H D M – 4

Highway Development & Management

volume SEVEN

MODELLING ROAD USER AND ENVIRONMENTAL EFFECTS IN HDM-4

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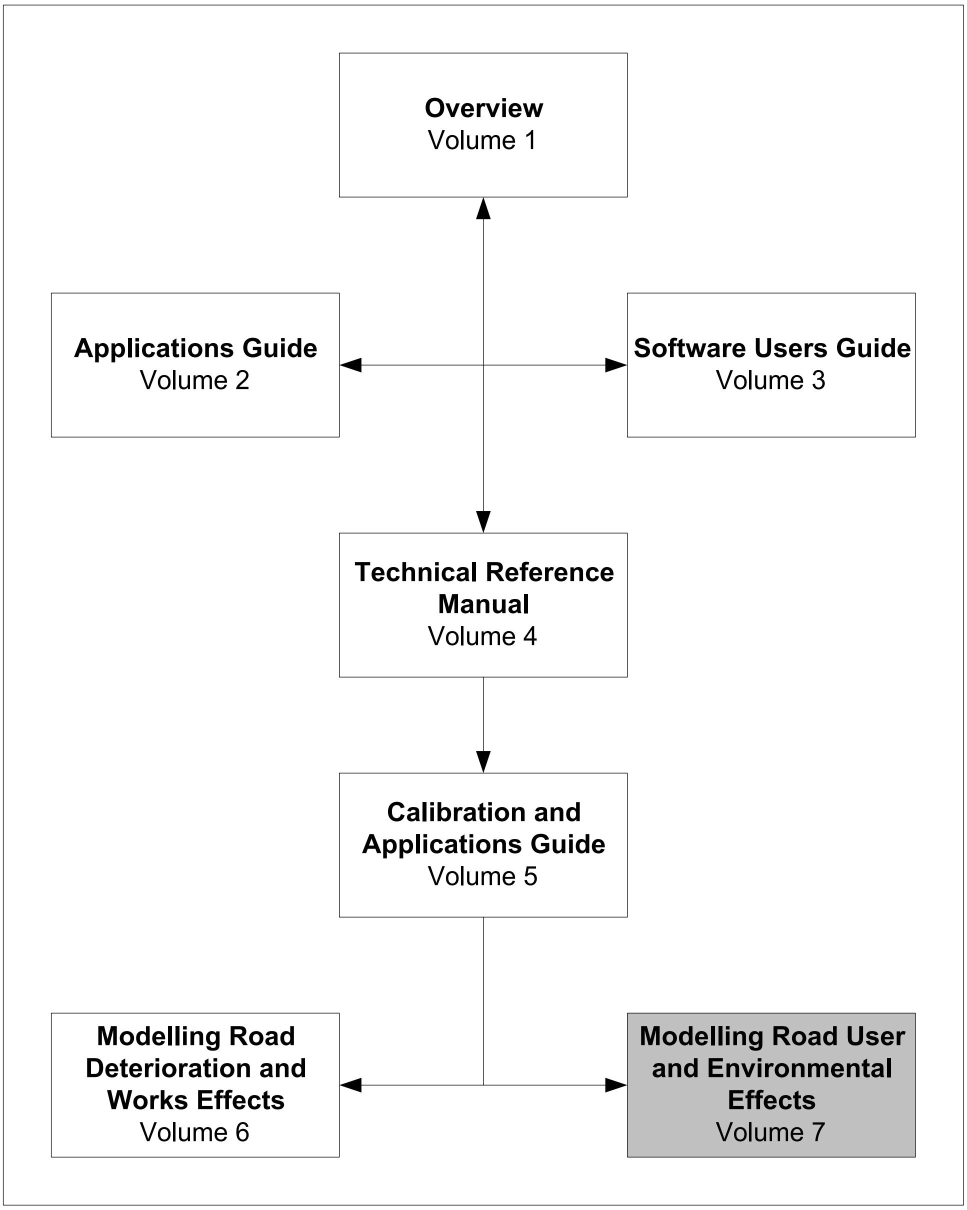
7

THE HIGHWAY DEVELOPMENT AND MANAGEMENT SERIES

About This Manual

This HDM-4 v1.0 edition of Modelling Road User and Environmental Effects provides details on the road user and environmental effects in the HDM model. It is one of seven volumes comprising the suite of HDM-4 documentation (see figure below). It is intended to be used by specialists interested in technical issues or responsible for setting up the HDM model. It provides the full background to the development and theoretical basis for the models in HDM-4 used for road user and environmental effects.



HDM-4 Documentation suite

The suite of documents comprise:

HDM-4 Overview (Volume 1)

A short executive summary describing the HDM-4 system. It is intended to be used by all readers new to HDM-4, particularly high level management within a road organisation

HDM-4 Applications Guide (Volume 2)

A task oriented guide describing typical examples of different types of analyses. It is to be used by the frequent user who wishes to know how to perform a task or create a study

HDM-4 Software User Guide (Volume 3)

Describes the HDM-4 software. It is a general purpose document which provides an understanding of the software user interface

HDM-4 Technical Reference Manual (Volume 4)

Describes the analytical framework and the technical relationships of objects within the HDM-4 model. It contains very comprehensive reference material describing, in detail, the characteristics of the modelling and strategy incorporated in HDM-4. It is to be used by specialists or experts whose task is to carry out a detailed study for a road management organisation

HDM-4 Calibration Reference Manual (Volume 5)

Suggests methods for adaptation and calibration of HDM-4 in different countries and allows for local conditions. It discusses how to calibrate HDM-4 through its various calibration factors. It is intended to be used by experienced practitioners who wish to understand the detailed framework and models built into the HDM-4 system

Modelling Road Deterioration and Works Effects in HDM-4 (Volume 6)

Describes the development and basis for the relationships in HDM-4 used for modelling road deterioration and works effects.

Modelling Road User and Environmental Effects in HDM-4 (Volume 7)

Describes the development and basis for the relationships in HDM-4 used for modelling road user and environmental effects.

Notes:

1 Volumes 1, 2 and 3 are designed for the general user

2 Volumes 4 to 7 are only to be used by experts who wish to obtain low level technical detail

Structure of the Manual

The information in this document is structured into three sections as follows:

- **Introduction**. This gives an overview of road user and environmental effects, how they are used in studies and representative vehicles.
- **Road User Effects**. This section presents details of the different components which comprise road user effects.

- **Environmental Effects**. This deals with vehicle emissions, noise emissions, and energy balance considerations.

ISOHDM products

The products of the International Study of Highway Development and Management Tools (ISOHDM) consist of the HDM-4 suite of software, associated example case study databases, and the Highway Development and Management Series collection of guides and reference manuals. This Volume is a member of that document collection.

Customer contact

Should you have any difficulties with the information provided in this suite of documentation please do not hesitate to report details of the problem you are experiencing. You may send an E-mail or an annotated copy of the manual page by fax to the number provided below.

The ISOHDM Technical Secretariat welcomes any comments or suggestions from users of HDM-4. Comments on Modelling Road User and Environmental Effects in HDM-4 should be sent to the following address:

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Change details

This is the first edition of the HDM-4 documentation.

Related documentation

HDM-4 documents

The Highway Development and Management Series Collection is ISBN: 2-84060-058-7, and comprises:

Volume 1 - HDM-4 Overview, ISBN: 2-84060-059-5

Volume 2 - HDM-4 Applications Guide, ISBN: 2-84060-060-9

Volume 3 - HDM-4 Software User Guide, ISBN: 2-84060-061-7

Volume 4 - HDM-4 Technical Reference Manual, ISBN: 2-84060-062-5

Volume 5 - HDM-4 Calibration Reference Manual, ISBN: 2-84060-063-3

Volume 6 - Modelling Road Deterioration and Works Effects in HDM-4,
ISBN: 2-84060-102-8

Volume 7 - Modelling Road User and Environmental Effects in HDM-4,
ISBN: 2-84060-103-6

Terminology handbooks

PIARC Lexicon of Road and Traffic Engineering - First edition. Permanent International Association of Road Congresses (PIARC), Paris 1991. ISBN: 2-84060-000-5

Technical Dictionary of Road Terms - Seventh edition, English - French. PIARC Commission on Terminology, Paris 1997. ISBN: 2-84060-053-6

General reference information

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Many of the tools, background reports and additional material described in this report is available from the HDM links available at the HTC Consultants Ltd. web site:

http://www.htc.co.nz

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Asian Development Bank (ADB)

Department for International Development (DFID) in the United Kingdom

Swedish National Road Administration (SNRA)

The World Bank

In addition to these, significant contributions were made by:

Finnish Road Administration (FinnRA)

Intra-American Federation of Cement Producers (FICEM)

Many other organisations and individuals in a number of countries have also contributed in terms of providing information, or undertaking technical review of products being produced.

The study has been co-ordinated by the ISOHDM Technical Secretariat at the University of Birmingham in the United Kingdom. A number of organisations participated in the research including:

FinnRA

Specification of the strategic and programme analysis applications.

FICEM

Development of deterioration and maintenance relationships for Portland cement concrete roads.

The Highway Research Group, School of Civil Engineering, The University of Birmingham

Responsible for system design and software development.

Road Research Institute (IKRAM) in Malaysia supported by N.D. Lea International (NDLI)

Responsible for providing updated relationships for road deterioration and road user costs.

Transport Research Laboratory (TRL) in the United Kingdom

Responsible for review and update of flexible pavement deterioration relationships.

SNRA

Responsible for developing deterioration relationships for cold climates, road safety, environmental effects, and supporting HRG with system design.

Transit New Zealand

Sponsored research to develop the HDM-4 tyre consumption model.

All research organisations received support from local and regional staff, visiting experts and external advisers, to ensure that a high standard of quality and international consensus was achieved. A number of other countries and individuals have supported this work through supplying expert advice and reviewing the products.

Copyright statement

These HDM-4 products have been produced by the International Study of Highway Development and Management Tools (ISOHDM), sponsored by The World Bank, the Asian Development Bank, the Department for International Development (UK), the Swedish National Road Administration, and other sponsors. The HDM-4 products are jointly published by The World Road Association (PIARC), Paris and The World Bank, Washington, DC.

Volume 7 Modelling Road User and Environmental Effects in HDM-4

Modelling Road User and Environmental Effects in HDM-4

A1 Introduction

This report is one of a two-volume set presenting the key technical results from the International Study of Highway Development and Management Tools (ISOHDM) project. It covers road user and environmental effects while the second volume addresses pavement deterioration and works effects.

Originally developed by the World Bank, the Highway Development and Management Model (HDM) has become widely used as a planning and programming tool for highway expenditures and maintenance standards. HDM is a computer model which simulates physical and economic conditions over the period of analysis, usually a life-cycle, for a series of alternative strategies and scenarios specified by the user.

HDM is designed to make comparative cost estimates and economic evaluations of different construction and maintenance options, including different time-staging strategies, either for a given road project on a specific alignment or for groups of links on an entire network. It estimates the total costs for a large number of alternative project designs and maintenance alternatives year by year, discounting the future costs if desired at different postulated discount rates so that the user can search for the strategy with the lowest discounted total cost.

Three interacting sets of costs (related to construction, maintenance and road use) are added together over time in discounted present values, where the costs are determined by first predicting physical quantities of resource consumption and then multiplying these by unit costs or prices. Road user costs are the largest component of the total transport costs and are thus, arguably, the most important to accurately predict.

The preparation of this report was funded by the Asian Development Bank under the Technical Assistance Project 5819-REG. It draws heavily on the ISOHDM Technical Relationships Study (HTRS) which was also sponsored by the Asian Development Bank under another Technical Assistance Project (5549-REG).

The HTRS was completed in October 1995 and is described in NDLI (1995). Since that time further work has been done which refined the HTRS work, and in some instances completely replaced the NDLI (1995) models. The HTRS study report also did not cover the full range of road user and environmental effects since some components were outside of the scope of their brief. This report brings together all the RUE work into a single volume.

The report is divided into three sections:

- **Overview:** This gives an overview of road user and environmental effects, how they are used in studies and representative vehicles.
- **Road User Effects:** This section presents details of the different components which comprise road user effects.
- **Environmental Effects:** This deals with vehicle emissions, noise emissions, and energy balance considerations.

The material presented in this report supersedes all previously published material. For the sake of brevity, not all the background material contained in NDLI (1995) is presented and users interested in these should refer to the original HTRS report or the various working papers released during the HTRS. Access to much of this material is available through the links on the HTC Consultants Ltd. web site at www.htc.co.nz.

A2 Road User and Environmental Effects in Economic Appraisals

A2.1 Road User Costs in Economic Appraisals

A minimum of expense is, of course, highly desirable; but the road which is truly the cheapest is not the one which has cost the least money, but the one which makes the most profitable returns in proportion to the amount spent on it.

W.M. Gillespie—1847

The preceding quotation shows that the importance of economic appraisals in highway engineering was recognised over 150 years ago. However, it is only relatively recently that they have become widespread. As late as 1962 almost 40 per cent of American state highway departments "never" used economic evaluations (Winfrey and Zellner, 1971). In developing countries, it is only in the last 25 years that economic evaluations have become common—and this is due in part to the role of the international agencies such as the World Bank who require appraisals as part of their loan criteria.

In order to conduct an economic appraisal it is necessary to calculate the streams of costs and benefits which arise from the project. Most of the benefits stem from a reduction in the cost of transport. In some instances an increase in demand for transport provides scope for additional benefits. It is therefore essential to have a mechanism for predicting the cost of transport under different investment strategies in order to undertake the economic appraisal.

Box A2.1 Terminology

The following terminology is used in this report:

- **VOC – Vehicle Operating Costs**: The total cost of road transport. Comprised primarily of fuel, tyre, parts, labour, oil and capital costs. It may also include travel time and crew costs.
- **RUE – Road User Effects:** Similar to VOC, but RUE also encompass other components traditionally missed—such as emissions and safety. In HDM-4 RUE has been adopted as the standard terminology.
- **RUC – Road User Costs:** The costs which arise when the components of the RUE are assigned monetary values.
- **RDWE – Road Deterioration and Works Effects:** Encompasses pavement deterioriationa nd the effects of works improvements. The latter include maintenance as well as improvements such as widening.

The cost of transport is often referred to as 'vehicle operating costs' (VOC) or 'road user effects' (RUE). Figure A2.1 shows the components of the RUE. VOC and RUE are both used in this report. VOC to reflect the components specifically associated with vehicle operation (the left column of boxes in Figure A2.1); RUE to reflect all components of Figure A2.1.

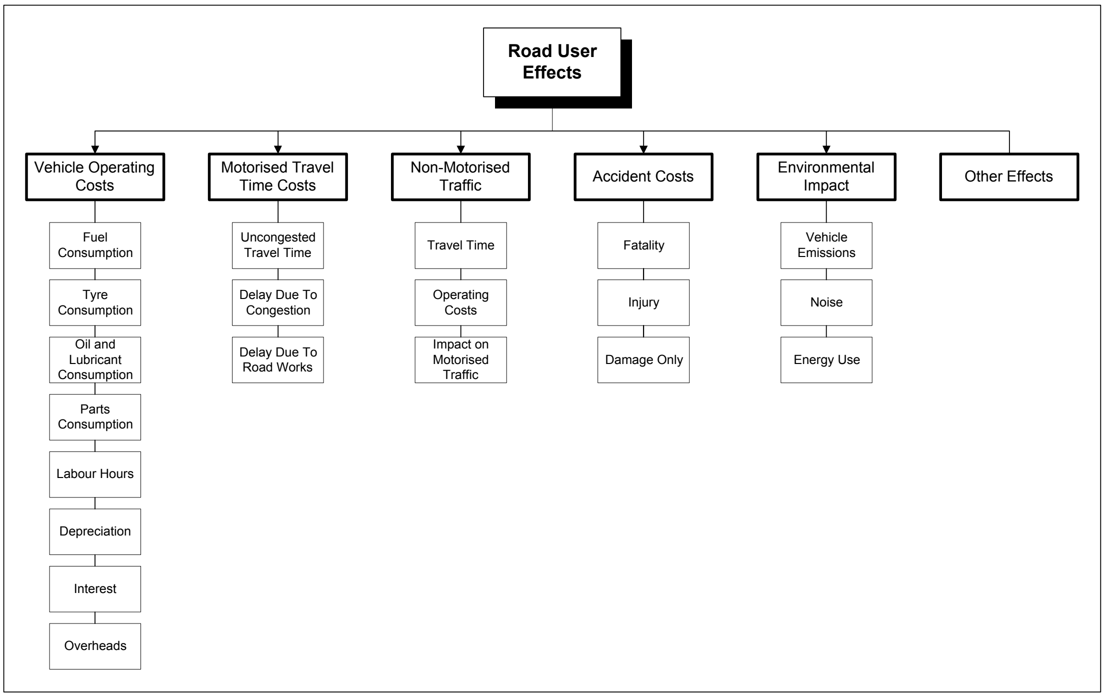


Figure A2.1: Components of Road User Effects

The relationship between the highway design standard, condition and vehicle operating costs has long been of interest to highway engineers. In 1881 Law and Kinnear-Clark published vehicle operating costs for horse-drawn vehicles. Research into fuel consumption began in the 1920s (Agg, 1923) and there were a number of studies which investigated the effects of geometry and road condition on vehicle operating costs, with particular emphasis on the advantages of gravel and paved surfaces over dirt. Moyer and Winfrey (1939) report one of the earliest surveys of vehicle operating costs.

The rapid expansion of the road networks in the 1950s led to an increase in research into vehicle operating costs. In the early 1950s the first appraisal manual incorporating vehicle operating costs was produced by the American Association of State Highway Officials (AASHO, 1952). During the 1950s and early 1960s a number of studies were conducted which tried to quantify the effects of highway and vehicle characteristics on vehicle operating costs. Winfrey (1963) produced a seminal work which synthesised the available experimental and survey data on vehicle operating costs. Winfrey (1969) was an update of this work which included accidents. This volume had a profound impact on economic appraisals for the next 10 years.

While most of the early research into vehicle operating costs was conducted in developed countries, particularly the USA, there was a growing need for economic appraisals in developing countries. This led the World Bank to sponsor a literature review which was published by De Weille (1966). The report had the objective of arriving "at a generally acceptable, more homogeneous approach to the problem of calculating vehicle operating cost savings". Drawing upon a variety of the existing sources—mainly Winfrey (1963)—a set of basic tables of vehicle operating costs was assembled for seven vehicle types.

Bonney and Stevens (1967), in the first major study of vehicle operating costs in developing countries, provided valuable insight into the costs of operating on bituminous, gravel and earth roads in Africa. This study noted that the previous work on the subject dated from the

1930s and "could not be used quantitatively with confidence in developing countries of the tropical regions, particularly because of the differences in climate and partly because of changes in vehicle design and operation over the last 25 years". As will be shown, this concern still exists today, although it has been ameliorated in some areas by adopting mechanistic models.

Table A2.1 summarises the major vehicle operating cost studies or publications since 1963.

Table A2.1: Major Syntheses of Vehicle Operating Costs

Reference	Cost Component	Source of Data or Relationships	Method	Date	Sample Size
Winfrey (1963) USA	Oil Fuel Tyres Maintenance Depreciation	Moyer (1939) Kent (1962) Claffey (1960) & Sawhill & Firey (1962) & Moyer (1939) Moyer & Tesdall (1945) & Kent (1960,1962) & Judgement Stevens (1961) & Moyer & Winfrey (1939) & Private sources, judgement Stevens (1961)	Experiment Experiment Experiment Experiment Experiment Experiment Experiment Survey	1938 1960 1959 1959 1938 1939-42 1960 1955-56	5 cars 45 line haul trucks 1 car, 2 trucks 9 trucks, 3 buses 5 cars 9 cars 45 line haul trucks, 40 city pickups 611 companies, 23,000 cars
De Weille (1966)	Oil Maintenance Tyres Depreciation Fuel	Moyer (1939) Stevens (1961) & Judgement Moyer & Tesdall (1945) & Stevens (1961) Judgement Winfrey (1963) & Kent (1960)	Experiment Survey Experiment Survey Model Experiment	1938 1955-56 1939-42 1955-56 1957-58	5 cars 611 companies, 23,000 cars 9 cars 611 companies, 23,000 cars 45 line haul trucks, 40 city pickups
Bonney & Stevens (1967) AFRICA	Fuel, oil, tyres, Maintenance, Depreciation, Crew wages	Own	Survey	1960-63	19 trucks, 46 buses
Claffey (1971) USA	Fuel Oil Tyres Maintenance	Stevens (1961)	Experiment Experiment Experiment Survey Own Survey	1963-69 1963-69 1963-69 1955-56 1963-69	5 cars, 1 truck, 1 bus 1 car, 1 truck 1 car, 1 truck 611 companies, 23,000 cars 1350 cars, 15 trucks
ARRB (1973) AUSTRALIA	Maintenance & Tyres Fuel All	Pelensky, *et al.* (1968) & Currey (1970) Pelensky (1970) Solomon (1970) & Solomon & Conroy (1974)	Survey Survey Experiment Survey		60 cars 299 cars 3 cars 950 trucks from 33 companies
Daniels (1974) AFRICA	Tyres Fuel Maintenance Depreciation	Bonney & Stevens (1967) EIU Surveys in Africa EIU Surveys in Africa	Survey Surveys Surveys Model	1960-63 1966-74 1966-74	19 trucks, 46 buses
Hide, *et al* (1975) -RTIM AFRICA	Fuel Tyres, Maintenance, Depreciation		Experiment Survey	1973-75 1973-75	1 car, 1 van, 3 trucks, 43 cars, 47 delivery vans, 78 trucks, 121 buses

AASHTO (1978) Red Book USA	Fuel Ttyres, Maintenance, Depreciation, Crew	Winfrey (1969) Claffey (1971) Curry & Anderson (1972) AASHO (1960)	As above As above	As above As above	As above As above
Zaniewski, *et al* (1982) TRDF USA	Fuel Truck VOC Car VOC Tyres Depreciation	 Ullmann (1980) Barriere, *et al. (*1974) Daniels (1974)	Experiment Survey Survey Model Survey	1979-82	8 rep. vehicles 12,489 trucks, 15 carriers
Watanatada, *et al* (1987a) - HDM III	Fuel Maintenance, Tyres Depreciation	Watanatada, *et al* (1987a) Chesher & Harrison (1987)	Experiment Survey Model	1975-79 1975-79	9 rep.vehicles 653 cars,442 trucks 231 cars
Bennett (1989a) - NZVOC NEW ZEALAND	All Fuel Fuel Depreciation	Watanatada, *et al* (1987) Biggs (1987) NITRR (South Africa) Bennett (1985)	As above Model Model Survey	As above 1984	As above
ISOHDM Study NDLI (1995)	All	Range of published sources	Literature review Limited field studies	1994-95	16 rep. vehicles

Source: Cox (1996a); Bennett (1985)

Initially, highway economic analyses were directed at minimising the construction costs. With the publishing of the vehicle operating cost information it was possible to determine the operating costs associated with a particular design, but there was no framework for considering the inter-relationship between the construction standard, maintenance standard, and the vehicle operating costs. As shown in Figure A2.2, this inter-relationship is complex, however, considering it allows for policies to be established which minimise the total transport costs to society.

In 1969 the World Bank initiated a programme to investigate this inter-relationship with respect to low volume roads. Phase I of the study was completed in 1971 which saw a research group at the Massachusetts Institute of Technology (MIT) develop a conceptual framework relating the construction and maintenance standards to vehicle operating costs. The objective was to determine the set of standards which minimised the total transport costs.

It was concluded from this initial study that sound empirical evidence was lacking for many of the cost relationships required for determining economic design and maintenance strategies. Accordingly, a major field study was undertaken from 1971-75 in Kenya which investigated paved and unpaved road deterioration as well as factors affecting vehicle operating costs (Hide, *et al*., 1975; Hodges, Rolt and Jones, 1975). The results of this study were used as a basis for developing th TRRL Road Transport Investment Model (RTIM) which evaluated the total transport costs for a single route (Robinson, *et al*., 1975). The Highway Design and Maintenance Standards Model (HDM) was developed in 1977 by incorporating features from the RTIM model with the MIT model. In 1981 the second version of the model (HDM-II) was released (Watanatada, 1981) and in 1987 HDM-III was released (Watanatada, *et al*., 1987a).

The development of HDM-III is shown in Figure A2.3. Since 1987 a number of refinements or new developments came from the model, ranging from a PC version to specialised models using components of HDM-III for predicting vehicle operating costs or pavement deterioration.

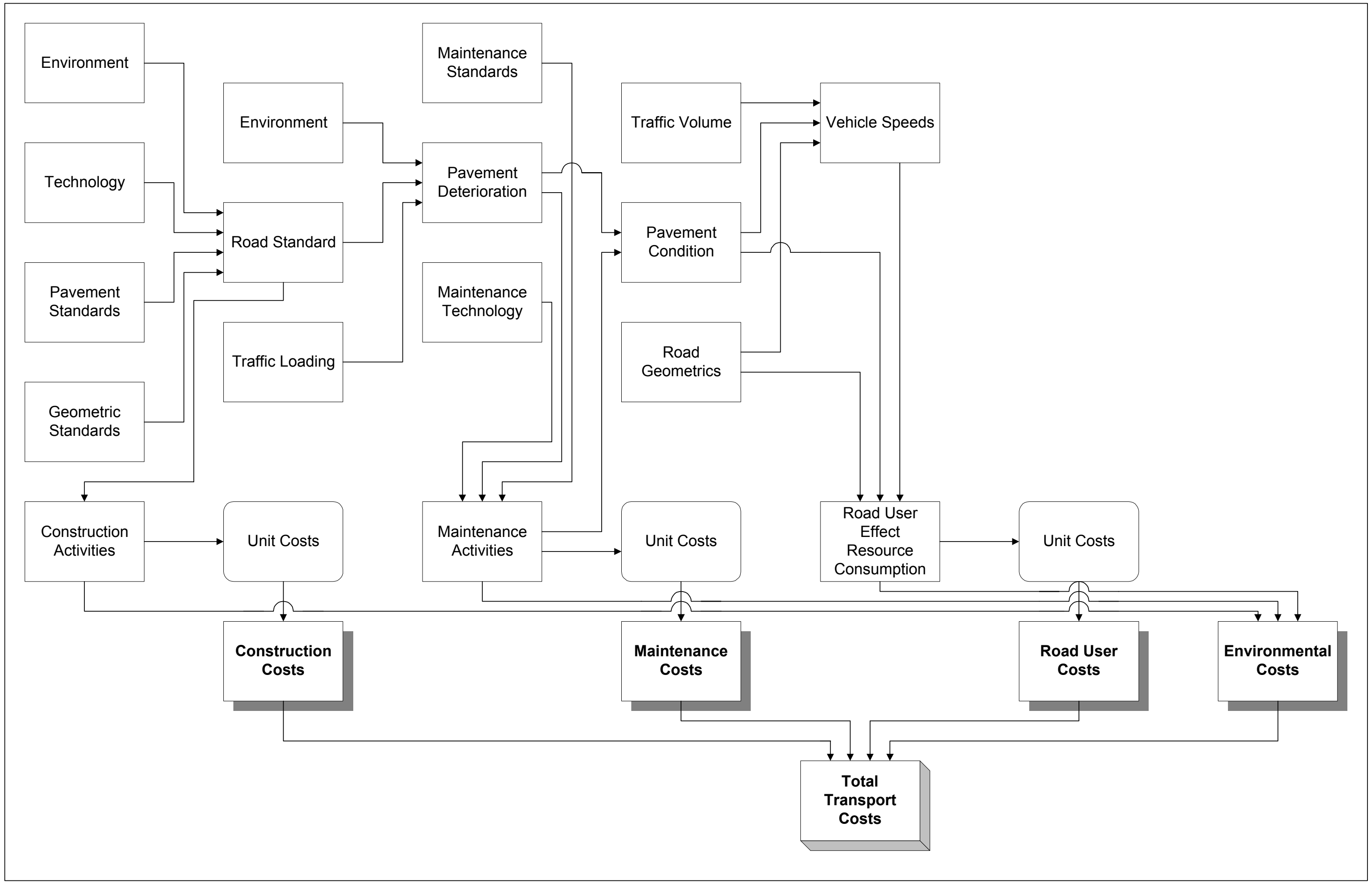


Figure A2.2: Inter-relationship Between Total Transport Cost Components

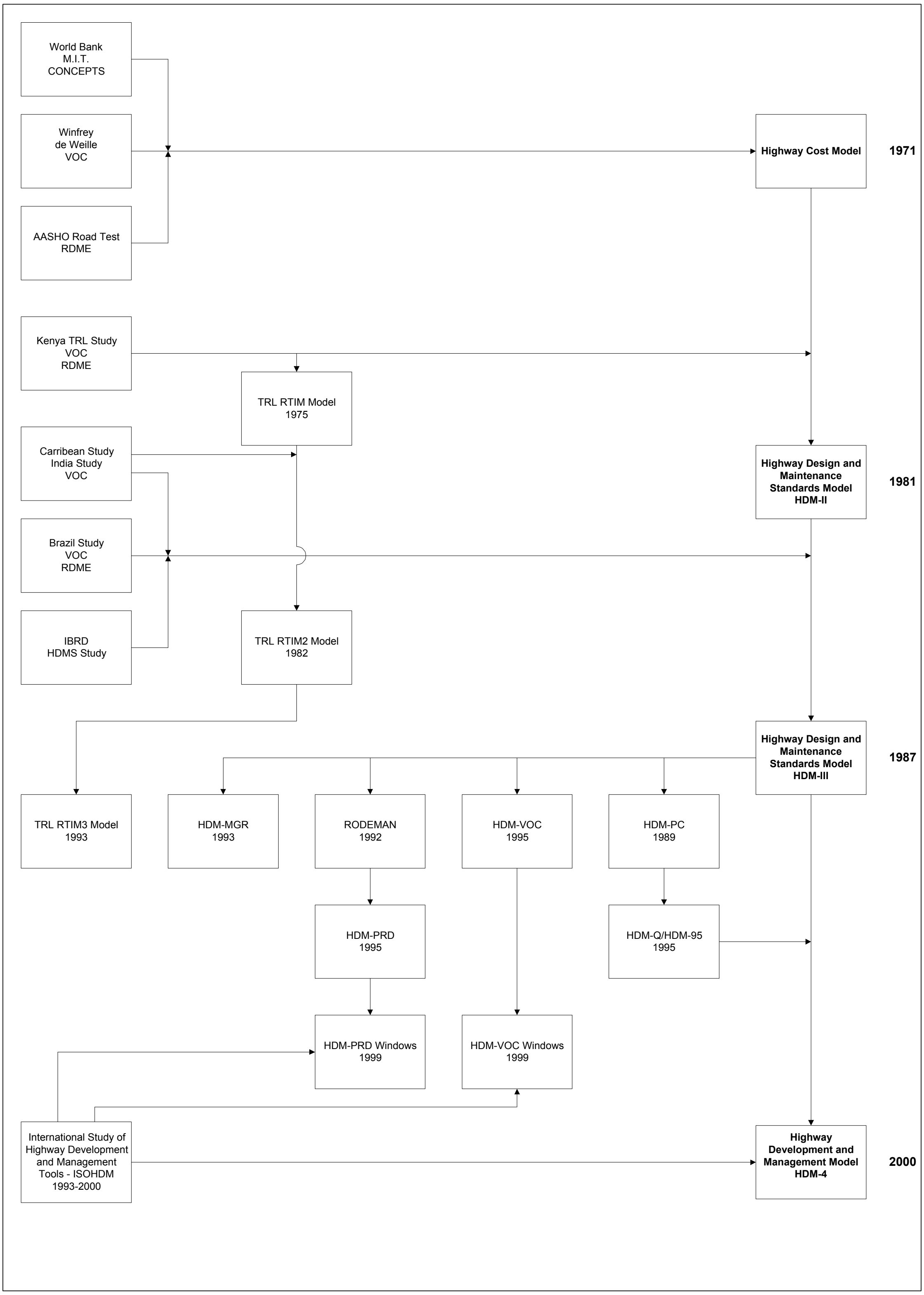


Figure A2.3: Development of HDM Model

HDM-III was applied in over 100 countries (NDLI, 1995) but it was recognised that there were limitations in the vehicle operating cost and pavement deterioration relationships. This led to the International Study of Highway Development and Management Tools (ISOHDM) which ran from 1993-2000 to update HDM-III and produce HDM-4

A2.2 Road User Effects in the HDM Model

Although HDM-III was widely used, it did not model the full range of road user effects. In its first release it was confined to free-flowing traffic. It was updated in 1995 to handle congestion effects, but this was only through the inclusion of a speed-flow model which reduced speeds with increasing volume-to-capacity ratio (Hoban, *et al*., 1994). This model did not consider the marginal effects of congestion on fuel or tyre consumption.

Chesher and Harrison (1987) describe the background to the HDM-III VOC relationships. In HDM-III there were four sets of VOC relationships available, developed from primary cost studies in Kenya, the Caribbean, India and Brazil. The user could select any one set of relationships and these were used to predict the amount of resources consumed for each of the above components. The exceptions to this were crew, overhead, depreciation and interest costs, where the same relationships are used for all applications.

As shown in Table A2.2, there were large differences in the scopes of the four primary cost studies. The Brazil study was the largest and the data were reanalysed by the World Bank to develop relationships for HDM-III. The Brazil relationships were therefore the most widely applied in HDM studies (Bennett, 1995a).

Table A2.2: Scope of Road User Cost Studies

	Kenya	Caribbean	India	Brazil
User Cost Survey				
Number of Vehicle Types	5	4	3	5
Total Number of Vehicles	289	68	939	1675
Largest truck GVW (t)	37	12	28	40
Number of Companies/Operators	-	-	121	147
Size of Network Monitored (km)	9300	-	40000	36000
Length of Observations (years)	2	2	3	4
Roughness (IRI)	3.3-9.0	3.5-11.4	5.4-12.9	1.8-14.9
Average Rise+Fall (m/km)	14.8-69.4	8.68	5.8-41.3	10.5
Horizontal Curvature (deg/km)	1.5-50	90-1040	26-675	6-294
Speed Studies				
Number of Sites	95	28	102	108
Number of Vehicle Types	5	4	6	6
Number of Observations	-	38,000	14,000	76,000
Roughness (IRI)	2.1-22.1	2.0-14.6	2.8-16.9	1.6-12.2
Vertical Gradient (per cent)	0.1-8.6	0-11.1	0.9	0-18.8
Horizontal Curvature (deg/km)	0-1980	0-1099	1-1243	0-2866
Road Width (m)	3.5-7.9	4.3-8.5	3.5-7.0	5.5-12.9
Fuel Experiments				
Number of Sections	95	82	-	51
Number of Vehicle Types	3	3	5	9
Number of Each Vehicle Type	1	1	1	1 or 2
Number of Observations	-	1161-2296	104-411	1192-5344
Roughness (IRI)	2.1-22.1	2.0-14.6	2.9-11.7	2.1-13.3
Vertical Gradient (per cent)	1.0-8.6	0-11.1	0-5	0.13
Horizontal Curvature (deg/km)	0-198	0-1099	-	0-340

NOTES: 1/ ‘ –’ data not available

As HDM-III has been applied to a wide range of conditions, modifications have often been made to the default VOC model. These may have seen calibration parameter values altered or the relationships replaced with alternatives. The various studies also used different representative vehicles to model the local fleet. A summary of the parameter values adopted in different studies as well as modifications made to the VOC model is given in Bennett (1995a).

The HDM-4 RUE work started with the HTRS project in Malaysia (NDLI, 1995). The starting point for the HDM-4 RUE model was the HDM-III model. The following specific areas were identified as priorities for research:

- Updating of fuel consumption model to better reflect modern vehicle technology;
- Consideration of the effects of traffic congestion on RUE;
- Updating of parts and labour model predictions to reflect modern vehicle technology;
- Revision and refinement of depreciation, interest and capital cost methodologies;
- Incorporating traffic safety impacts into HDM-4;
- Calculating the additional delay and VOC due to road works;
- Speed modelling; and,
- Modelling noise emissions.

As shown in Table A2.3, which is adapted from Anderson, *et al.* (1992), not all components have a major impact on the total costs, and not all have been well rearched. The focus of the research was on the more critical components.

Tyre consumption and vehicle emissions was investigated by the Swedish research team. Further work into tyres was done by the New Zealand research team and this formed the basis for the final tyre model incorporated into HDM-4. The Japanese research team had the responsibility of developing the non-motorised traffic model. Refinements were also made to the Malaysian RUE work to resolve issues arising from its application in the field.

It must be appreciated that the ISOHDM study did not undertake any major new field studies into RUE. Instead, it assembled the available information to establish the most appropriate set of models for HDM-4. Unfortunately, there have been relatively few studies conducted from which VOC relationships have been derived and this limited the scope for improvements in many areas. Bein (1993) reviewed the various VOC models and identified nine major models from different countries. However, as shown in Figure A2.4, most of them drew in at least in part from the HDM-III research. For some models, such as the Australian ARFCOM Model, the contribution of HDM was minor while in others, such as the New Zealand NZVOC Model, it was quite significant. As shown in Table A2.4 (Bein, 1993), not all are suitable to the full range of applications required of an economic appraisal model.

The area which has received the most attention is in modelling fuel consumption. Australia, South Africa and Sweden have all developed indigenous fuel consumption models. The area with the least research is in modelling maintenance parts and labour costs.

In spite of the relatively limited new research, it proved possible to expand the scope of RUE modelling in HDM-4 over HDM-III. All major components are now considered along with all the factors which influence the RUE. This increased scope is illustrated in Figure A2.5 and Figure A2.6 which contrast the RUE modelling in HDM-III and HDM-4.

Particularly noteworthy in HDM-4 are the modelling of the effects of congestion on RUE as well as non-motorised transport, safety, and vehicle emissions.

Table A2.3: Road User Cost Component Impact Matrix

		Road User Effect Component										
		Vehicle Operating Costs					Delay Costs			Accident Costs		
	Cost Impact	Fuel	Oil	Tyres	Maintenance and Repairs	Depreciation and Interest	Delay at Work Zone	Delay Due to Congestion	Delay at Intersections	Direct Costs	Indirect Costs	Cost Incurred Due to Deferral of Maintenance
Impact on User Costs	High	■						■				■
	Medium				■	■	■		■		■	
	Low		■	■						■		
Sensitivity of Pavement Type[1]	High			■			■					
	Medium	■	■									■
	Low	■			■	■			■	■	■	
Sensitivity to Pavement Condition[2]	High			■			■	■				■
	Medium	■	■		■			■		■	■	
	Low					■			■			
Extent of Research[3]	Major	■							■	■		
	Fair			■	■	■	■	■			■	
	Limited		■									■

Source: Adapted from Anderson, et al. (1992)

Notes: 1/ Sensitivity of the user cost to pavement type considers portland cement concrete and asphaltic concrete pavements.

2/ Pavement condition in terms of the Pavement Serviceability Index (PSI).

3/ Amount of research to date pertains to the user cost component of interest, not the subject as a whole.

4/ ■ Indicates that cost component has been assigned that cost impact.

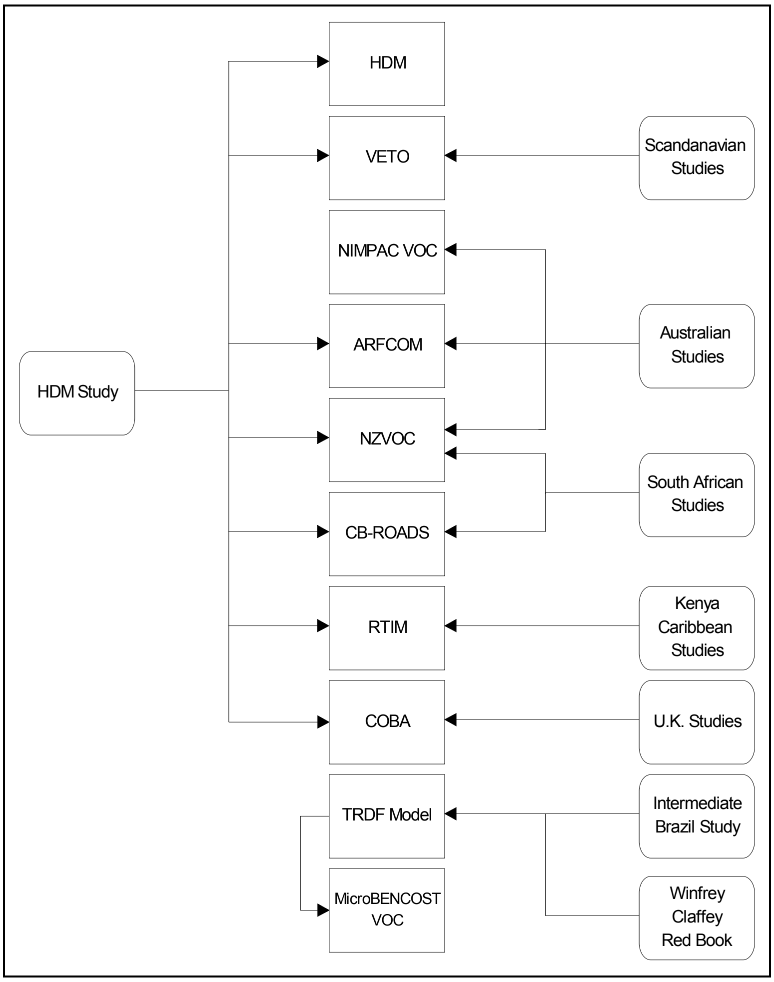


Figure A2.4: Basis for VOC Models

Source: Adapted from Bein (1993)

A2.3 Issues in Modelling Road User Effects

While there are many issues which arise in modelling road user effects, the two key ones are technology and transferability:

- **Technology:** What impact does changes in technology have on the costs?
- **Transferability:** How transferable are models from one country to another? How transferable from one region to another even within the same country?

Vehicle technology is constantly changing. This is not a recent development—Bonney and Stevens (1967) referred to technological changes as being a factor warranting their study—and they can be expected to continue. For example, Winfrey (1969) predicted truck costs from petrol powered vehicles. Diesel trucks are now almost exclusively used in developed countries, and in most developing countries. Electronic fuel injection and engine management systems coupled with aerodynamic improvements have led to quantum improvements in fuel consumption for almost all modern vehicles. In the near future electric or hybrid vehicles will begin service. Tyre technology has also changed with modern compounds offering significantly lower rolling resistance and longer tyre lives. While the mechanstic modelling of fuel and tyres should allow for these changes to be modelled, other components such as parts consumption do not lend themselves to such modelling and so are much more difficult to consider.

Table A2.4: Features of Major VOC Models

Feature	HDM	COBA9	VETO	NIMPAC	ARFCOM	TRDF, Micro-BENCOST	CB-ROADS
	Basis						
Statistical	•	•		•		•	•
Mechanistic	•		•		•		
	Level of Aggregation						
Simulation			•		•		
Project	•	•	•	•	•	•	•
Network	•	•		•	•		•
	Vehicle Operation						
Uniform Speed	•	•	•	•	•	•	•
Curves	•		•	•	•	•	•
Speed Change			•	•	•	•	
Idling			•	•	•	•	•
	Typical Vehicles						
Default	•	•	•	•	•	•	•
User Specified	•		•		•		
Modern Truck	•		•	•	•		•
	Road Variables						
Gradient	•	•	•	•	•	•	•
Curvature	•	•	•	•	•	•	•
Superelevation	•		•		•		
Roughness	•		•	•	•	•	•
Pavement Type	•		•	•	•		•
Texture			•		•		•
Snow, Water, Ice			•		•		
Wind, Temperature			•		•		
Absolute Elevation	•		•		•		
	VOC Components						
Fuel, Oil, Tyres, Maint. Deprec.	•	•	•	•	Fuel Only	•	•
Interest	•		•	•			•
Cargo Damage			•				
Overhead	•	•	•				
Fleet Stock		•	•				
Exhaust Emissions			•				

Source: Bein (1993)

How important can these changes be? Cox (1996a) indicates that a comparison of 1992 cost VOC to those from 1970 showed a 66 per cent reduction in costs over the 22 year period due to “substantial reductions in fuel, maintenance, tyres, oil and capital costs”. For trucks the reduction was 60 per cent. Cox (1994, 1996a) points out that the net effect of technology reducing VOC and the changing age and utilisation distributions means that there is a systematic error in the VOC predictions unless one takes these factors into account.

In HDM-4 it is possible to model technological change by providing different representative vehicles and having some of the fleet with negative growth rates so that over time they are retired from the fleet and replaced by different vehicles. The ability to model changing age and utilisation patterns is not available in HDM-4 as the data required to implement these features is seldom available.

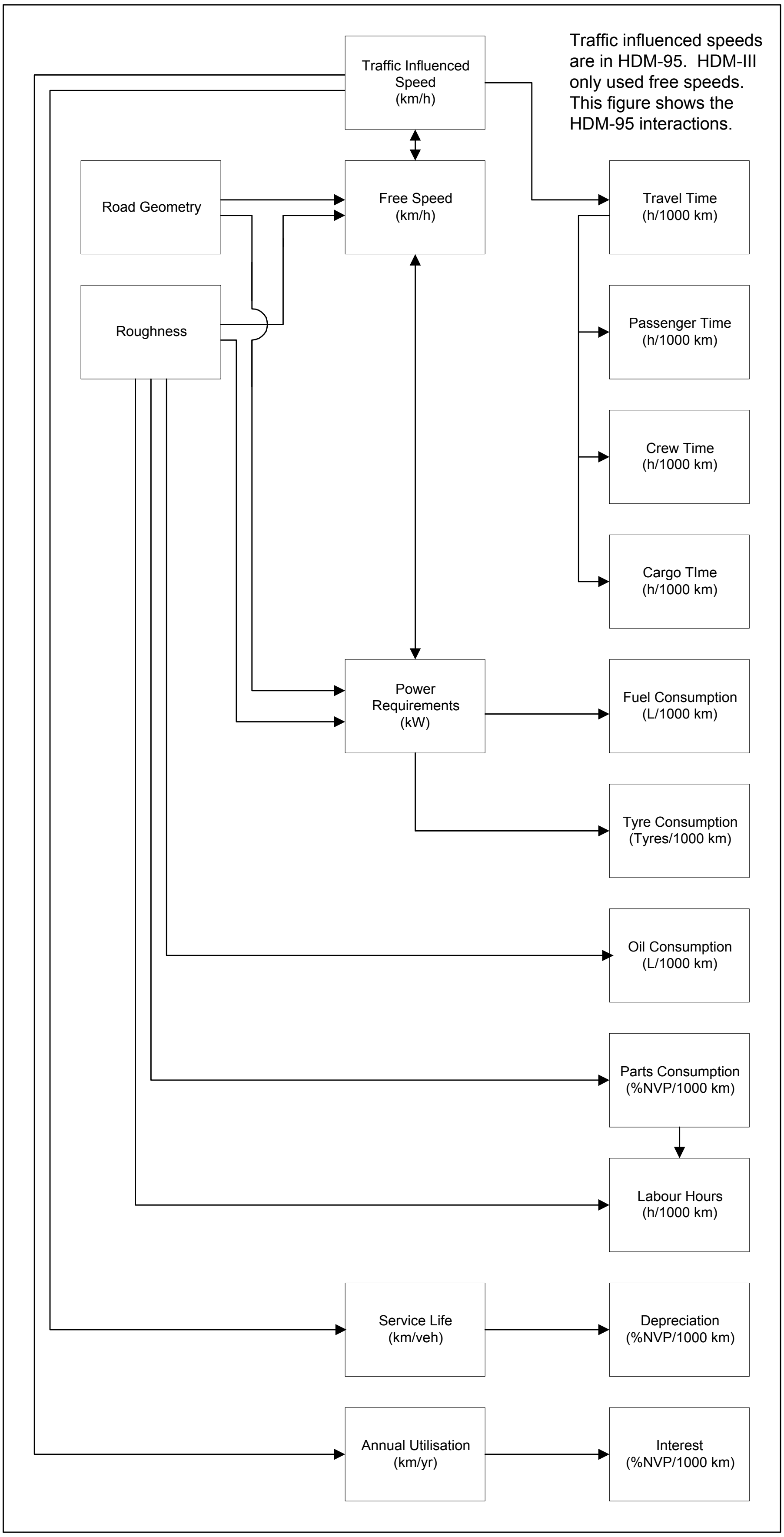


Figure A2.5: HDM-III/95 Road User Effects and Their Interactions

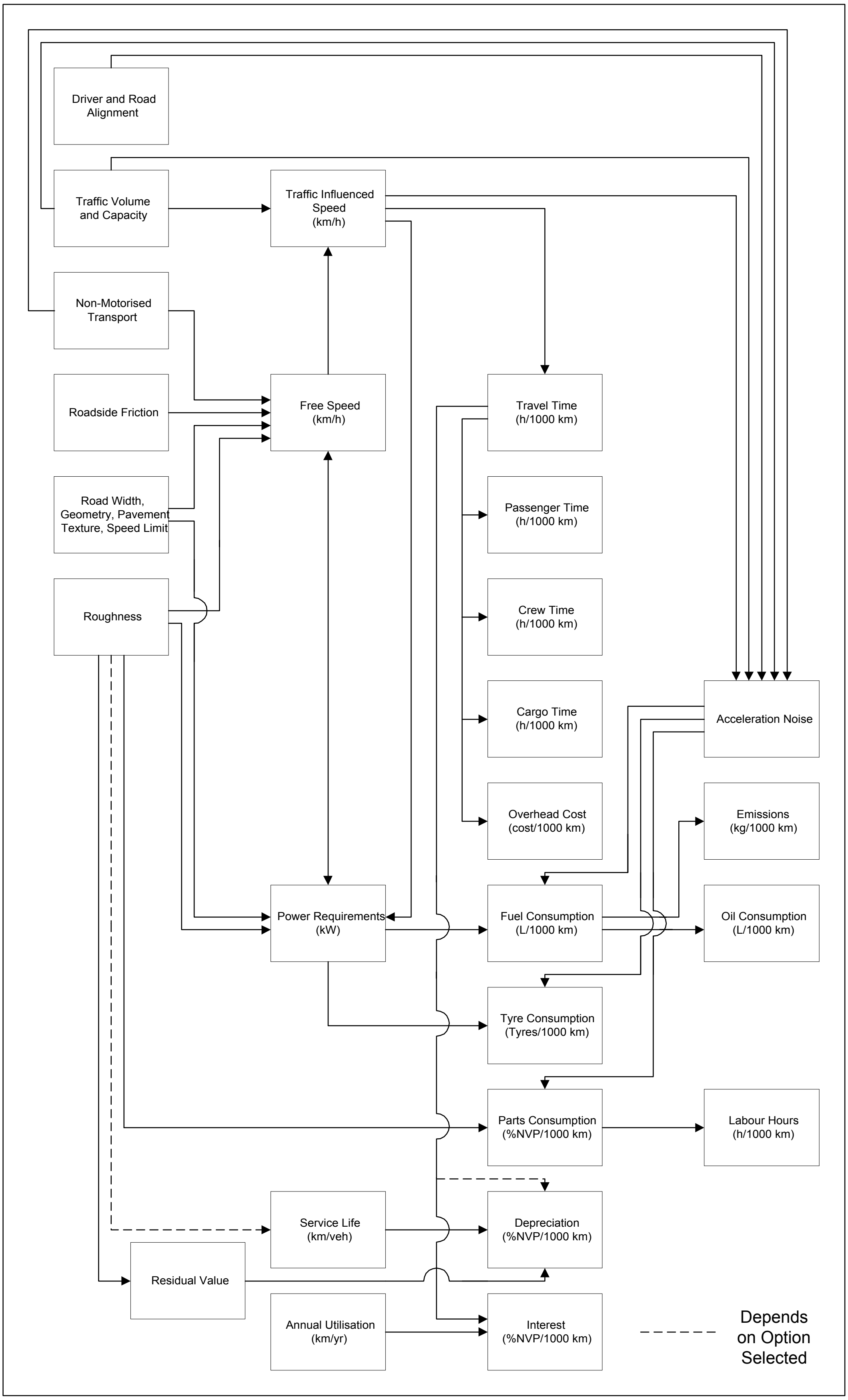


Figure A2.6: HDM-4 Road User Effects and Their Interactions

The transferability issue is the reason why it is **always** necessary to calibrate the HDM model. As an example of this, Figure A2.7 illustrates the differences arising between costs observed for trucks in Canada and those predicted by the HDM-III model with unadjusted truck default values (Lea, 1988). Not only were the total VOC predicted by HDM-III significantly different from those observed, but the relative contributions of the various components to the total operating costs were also very different.

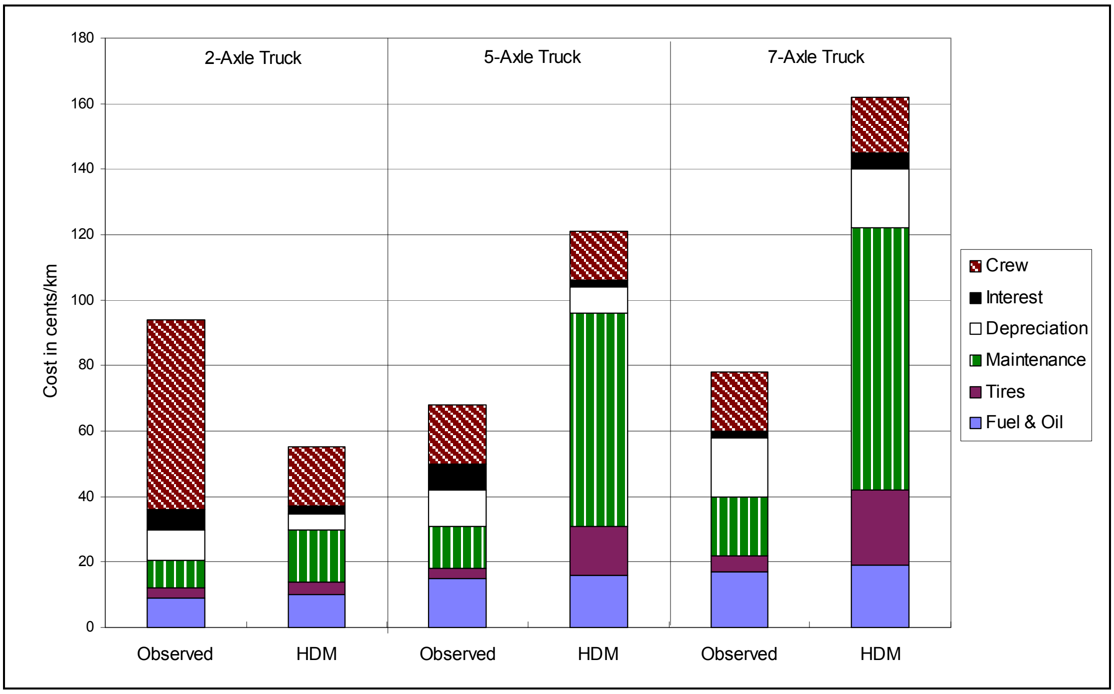


Figure A2.7: Comparison Between Observed and Uncalibrated HDM Predictions

These differences arise due in part to the different economic environments. For example, countries with high labour costs will tend to replace parts rather than repair them. They are also due to differences in driver behaviour.

In calculating the RUE one must therefore apply the models prudently and with suitable local calibration. This will ensure that the predictions are relevant and appropriate for the conditions under which they are being applied.

Box A2.2 Guidelines on Calibration the HDM Model

A separate report is available from the ISOHDM Study presenting guidelines on calibrating the HDM model. It covers the background to calibration as well as dealing with specific methods and the key parameters.

A3 Representative Vehicles

Since it is not possible to model each individual vehicle in the traffic stream, analysts resort to the use of *representative vehicles* for calculating vehicle operating costs. These are vehicles whose characteristics can be considered to be representative of all vehicles within a certain class.

There are two possible approaches used in describing representative vehicles. The first is to develop a “composite” vehicle. This is a hypothetical vehicle whose characteristics are representative of all vehicles in a given class. The second approach is to select an actual vehicle as the representative vehicle.

There are advantages and disadvantages to each approach. While the composite vehicle approach yields a vehicle whose characteristics are representative of the entire fleet, it relies upon having detailed data available for all vehicle characteristics which, in practice, are difficult to obtain. Even though the characteristics of an actual vehicle may not be completely representative for the entire class, it is generally much easier to obtain these data. In practice, most analysts adopt vehicle characteristics based on individual vehicles (Bennett, 1995a).

HDM-III allowed up to 10 representative vehicles in a single run. Table A3.1 lists the characteristics of these vehicles based on the original Brazil study data (Watanatada, *et al*., 1987a).

Not all applications have used the same representative vehicle classes. This is illustrated in Table A3.2 which summarises the representative vehicle classes adopted in different studies or applications of models similar to HDM. It is based on the data in Bennett (1995a).

As noted by TSPC (1992), motorcycles are a particularly significant component of the total vehicle fleet in many developing countries, especially in Asia. However, HDM-III did not have the facility to model motorcycles.

For HDM-4 the decision was taken to make the model flexible with respect to the number of representative vehicles which could be used in the analysis. The user is able to define any number of vehicles. This enables the user to have, for example, several heavy trucks but with different loading patterns. On the basis of the vehicles adopted in different studies, the default representative vehicles in Table A3.3 were adopted for HDM-4.

For modelling RUE in HDM-4, it is necessary to assign certain key characteristics to the representative vehicles. The basic data for each of the 16 representative vehicle types are also given in Table A3.3. These values were estimated by NDLI (1995) from a variety of sources. The additional data are presented in each chapter which addresses the RUE.

Table A3.1: HDM-III Brazil Representative Vehicle Characteristics

Characteristic	Representative Vehicle									
	Passenger Car			Utility	Heavy	Medium Truck		Heavy Truck		Artic.
	Small	Medium	Large		Bus	Petrol	Diesel	2 Axle	3 Axle	Truck
Aerodynamic Drag Coeff.	0.45	0.50	0.45	0.46	0.65	0.70	0.70	0.85	0.85	0.63
Projected Frontal Area (m^2)	1.80	2.08	2.20	2.72	6.30	3.25	3.25	5.20	5.20	5.75
Engine Power (kW)	37	110	150	46	111	128	77	111	111	216
Tare Weight (t)	0.96	1.20	1.65	1.32	8.10	3.12	3.27	5.40	6.57	14.73
Operating Weight (t)	0.96	1.20	1.20	1.62	10.40	5.12	5.27	10.00	12.57	27.73
Rated Gross Weight (t)	1.16	1.47	1.92	2.16	11.50	6.06	6.06	15.00	18.50	40.00
Number of Axles	2	2	2	2	2	2	2	2	3	5
Number of Tyres	4	4	4	4	6	6	6	10	10	18
Base Number of Retreads	-	-	-	-	3.39	1.93	1.93	3.39	3.39	4.57
Volume of Rubber (dm^3)	-	-	-	-	6.85	4.30	4.30	7.60	7.30	8.39
Elasticity of Utilisation	0.60	0.60	0.60	0.80	0.75	0.85	0.85	0.85	0.85	0.85

Source: Watanatada, et al. (1987a)
NOTES: "-" Not applicable

Table A3.2: Representative Vehicles Adopted in Different Studies

Reference	Country	MC	PC			LDV	LGV	LT	MT	HT	AT	LB	MB	HB
			S	M	L									
Chamala (1993)	Australia		•	•	•	•		•	•	•	•			•
Transroute (1992)	Bangladesh		•			•			•					•
NDLI (1994a)	Barbados		•	•		•	•		•	•				•
SWK (1993a)	Botswana		•			•	•		•	•	•	•		•
IBRD (1990)	Burundi		•			•	•		•	•	•	•		
NDLI (1988)	Canada		•	•	•	•		•	•	•	•			•
SWK (1993b)	Ethiopia		•			•	•		•	•	•	•		•
GITEC (1992)	Guatemala			•		•				•	•		•	•
TSA (1995)	Hungary		•					•	•	•	•			•
NDLI (1994b)	India		•	•					•	•				•
Hoff & Overgaard (1992)	Indonesia	•	•			•		•	•	•		•		•
Sammour (1992)	Jordan				•				•					•
SWK (1993c)	Lesotho		•			•			•	•		•	•	•
JKR (1991)	Malaysia		•					•	•	•	•			•
NDLI (1992)	Myanmar		•			•		•	•	•		•		•
NDLI (1993)	Nepal			•		•			•					•
Louis Berger (1990)	Nigeria		•				•		•	•				•
Bennett (1989a)	New Zealand		•	•		•	•	•	•	•	•			•
RUST PPK (1994)	P.R. China		•					•	•	•	•	•		•
Kampsax-Beca (1990)	PNG		•			•		•	•	•	•			
CESTRIN (1994)	Romania		•					•	•	•	•			•
IBRD (1995)	Russia		•					•	•	•	•			•
Arup (1992)	Tanzania		•			•			•	•				•
NDLI (1991)	Thailand		•					•	•	•		•		•
NDLI (1994c)	Trinidad		•			•		•	•	•		•	•	
SWK (1993d)	Uganda		•			•			•		•			•
du Plessis and Schutte (1991)	South Africa		•			•			•	•				•
TSPC (1992)	Sri Lania	•	•			•		•	•	•	•		•	•

Source: Adapted from Bennett (1995a)

Table A3.3: HDM-4 Default Representative Vehicle Classes and Characteristics

Vehicle Number	Type	Description	Abbrev-iation	Fuel Type	Number of Axles	Number of Wheels	Aero-dynamic Drag Coeff.	Projected Frontal Area (m^2)	Tare Weight (t)	Operating Weight (t)
1	Motorcycle	Motorcycle or scooter	MC	P	2	2	0.70	0.8	0.1	0.2
2	Small Car	Small passenger cars	PC-S	P	2	4	0.40	1.8	0.8	1.0
3	Medium Car	Medium passenger cars	PC-M	P	2	4	0.42	1.9	1.0	1.2
4	Large Car	Large passenger cars	PC-L	P	2	4	0.45	2.0	1.2	1.4
5	Light Delivery Vehicle	Panel van, utility or pickup truck	LDV	P	2	4	0.50	2.0	1.3	1.5
6	Light Goods Vehicle	Very light truck for carrying goods (4 tyres)	LGV	P	2	4	0.50	2.8	0.9	1.5
7	Four Wheel Drive	Landrover/Jeep type vehicle	4WD	P/D	2	4	0.50	2.8	1.5	1.8
8	Light Truck	Small two-axle rigid truck (approx. < 3.5 t)	LT	D	2	4	0.55	4.0	1.8	2.0
9	Medium Truck	Medium two-axle rigid truck (> 3.5 t)	MT	D	2	6	0.60	5.0	4.5	7.5
10	Heavy Truck	Multi-axle rigid truck	HT	D	3	10	0.70	8.5	9.0	13.0
11	Articulated Truck	Articulated truck or truck with drawbar trailer	AT	D	5	18	0.80	9.0	11.0	28.0
12	Mini-bus	Small bus based on panel van chassis (usually 4 tyres)	MNB	P	2	4	0.50	2.9	1.1	1.5
13	Light Bus	Light bus (approx. < 3.5 t)	LB	D	2	4	0.50	4.0	1.75	2.5
14	Medium Bus	Medium bus (3.5 - 8.0 t)	MB	D	2	6	0.55	5.0	4.5	6.0
15	Heavy Bus	Multi-axle or large two-axle bus	HB	D	3	10	0.65	6.5	8.0	10.0
16	Coach	Large bus designed for long distance travel	COACH	D	3	10	0.65	6.5	10.0	15.0

Source: NDLI (1995)

The application of representative vehicles in HDM-4 is done through a multi-level hierarchy as shown in Figure A3.1 and A3.2 There are three levels to this hierarchy:

- Categories - which differentiate between motorised and non-motorised traffic;
- Classes - groupings of similar vehicles, for example passenger cars;
- Types - the individual representative vehicle types, for example as in Table A3.3.

The motorised vehicle hierarchy in Figure A3.1 only shows the default vehicle types. As mentioned earlier, the user can define any number of additional types. The users can also define their own classes in HDM-4.

The HDM-4 framework will allow great latitude in selecting representative vehicles for an analysis. However, it will be up to the analyst to determine the appropriate number and mix of vehicles to use. One should avoid having more vehicles than the data warrant and their selection should be based primarily on factors such as VOC, loading, and occupancies. One common situation is where there are differential loading patterns in two directions — for example uphill and downhill — and these will have a major impact on the pavement loading and VOC. In such instances different vehicles should be used in each direction.

Figure A3.1: Motorised Vehicle Representation in HDM-4

Figure A3.2: Motorised Vehicle Representation in HDM-4

B1 Forces Opposing Motion

B1.1 Introduction

Mechanistic models predict that the VOC are proportional to the forces acting on the vehicle. Thus, by quantifying the magnitude of the forces opposing motion one can establish, for example, the fuel consumption. Mechanistic models are an improvement over empirical models since they can allow for changes in vehicle characteristics and are inherently more flexible when trying to apply the models to different conditions.

For HDM-4 mechanistic models were adopted for the modelling of fuel and tyre consumption only, as mechanistic theory has not been successfully applied to the other VOC components. Mechanistic/behavioural models were adopted for predicting speeds.

The total power required by a vehicle is given by:

$$Ptot = Ptr + Paccs + Peng \qquad \ldots(B1.1)$$

where Ptot is the total power required in kW
Ptr is the tractive power required in kW
Paccs is the power required to drive the accessories in kW
Peng is the power required to overcome engine drag in kW

Since the vehicle drivetrain is not perfectly efficient, it is necessary to supply more power than the tractive forces call for. This leads to the following equations for predicting the total power requirements:

$$Ptot = \frac{Ptr}{edt} + Paccs + Peng \qquad Ptr \geq 0 \qquad \ldots(B1.2)$$

$$Ptot = Ptr\ edt + Paccs + Peng \qquad Ptr < 0 \qquad \ldots(B1.3)$$

where edt is the drivetrain efficiency as a decimal

The accessory power is the power required to drive the vehicle accessories such as the cooling fan and the engine drag allows for internal friction.

The tractive power is the power applied at the wheels, and is equal to:

$$Ptr = Ftr\ \ v / 1000 \qquad \ldots(B1.4)$$

where Ftr is the tractive force applied to the wheels in N
v is the vehicle velocity in m/s

The tractive force is calculated as:

$$Ftr = Fa + Fr + Fg + Fc + Fi \qquad \ldots(B1.5)$$

where Fa Aerodynamic drag resistance in N
Fr Rolling resistance in N
Fg Gradient resistance in N
Fc Curvature resistance in N
Fi Inertial resistance in N

The forces acting on a vehicle as it ascends a grade of angle θ to the horizontal, are shown Figure B1.1. Each of these forces are described in the following sections, along with how they are quantified.

Figure B1.1: Forces Acting on a Vehicle on a Gradient

B1.2 Aerodynamic Resistance

B1.2.1 Introduction

The aerodynamic force represents the force required to push an object through the air. It is calculated as:

$$Fa = 0.5\ \rho\ CD\ AF\ vr^2 \qquad \ldots(B1.6)$$

where	Fa	is the aerodynamic force opposing motion in N
	ρ	is the mass density of air in kg/m^3
	CD	is the aerodynamic drag coefficient
	AF	is the projected frontal area of the vehicle in m^2
	vr	is the speed of the vehicle relative to the wind in m/s

The following sections describe quantifying the various components of this equation.

B1.2.2 Mass Density of Air

The density of moist air varies with pressure and temperature. Several researchers (St John and Kobett, 1978; Visser and Curtayne, 1985; and Biggs, 1990) have related the density of air to either temperature, altitude or a combination of the two. (Bennett, 1989a) established the following equation for calculating the mass density of air as a function of altitude and air temperature:

$$\rho = 0.0566 + 1.225\ (1 - 2.26\ \ 10^{-5}\ ALT)^{4.225} - 0.00377\ TAIR \qquad \ldots(B1.7)$$

where	ρ	is the mass density of air in kg/m^3
	ALT	is the altitude above sea level in m
	TAIR	is the temperature of the air in ^oC

For HDM-4 the user supplies the altitude or a default value which should be calculated using Equation B1.7. The default value recommended is 1.2 kg/m^3, which is based on a standard temperature of 15°C at an altitude of 200 m.

It should be recognised that the forces are not very sensitive to the values adopted for this parameter: an elevation change of 500 m or a temperature change of 15 °C results in a five per cent change in ρ. Given the number of other factors influencing RUE, the user should adopt a standard value for ρ and use it in all their analyses.

B1.2.3 Aerodynamic Drag Coefficient and Relative Velocity

The aerodynamic drag coefficient (CD) is a measure of three sources of air resistance:

- Form drag caused by turbulent air flow around the vehicle;
- Skin friction between the air and the vehicle; and,
- Interior friction caused by the flow of air through the vehicle.

Form drag and skin friction make up approximately 85 per cent and 10 per cent respectively of the total air resistance (Mannering and Kilareski, 1990). Accordingly, large square shaped vehicles have higher CD values than small rounded ones.

The CD is a function of vehicle direction relative to the wind. The apparent direction of the wind, which is the vector resultant of the vehicle direction and the wind direction, is termed the yaw angle (ψ). This is illustrated in Figure B1.2. The values of CD reported in the literature are usually from wind tunnel tests conducted with a 0 degree yaw (*ie* front on to the wind) and accordingly are the minimum for the vehicle.

To obtain typical values of CD which would be found on roads it is therefore necessary to adjust for the variation in wind direction. This can be done using a "typical" wind angle which increases the 0 degree yaw CD, leading to wind averaged CD, or CD(ψ).The ratio of CD(ψ)/CD is termed the CD multiplier (CDmult). Equation B1.6 is therefore rewritten as:

$$Fa = 0.5\ \rho\ CD(\psi)\ AF\ vr^2 \qquad \ldots(B1.8)$$

or,

$$Fa = 0.5\ \rho\ CDmult\ CD\ AF\ vr^2 \qquad \ldots(B1.9)$$

where CD(ψ) is the wind averaged CD
CDmult is the CD multiplier = CD(ψ)/CD

Biggs (1988) presented an approach for calculating CDmult, however Biggs (1995) recommended the use of the methodology presented by Sovran (1984). The Sovran (1984) approach gives a value of zero for CDmult when the yaw angle is 90°, whereas the method presented by Biggs (1988) gives a non-zero value. The Sovran (1984) approach for calculating CDmult is as follows:

Figure B1.2: Wind Forces Acting on Vehicle

<u>For $0 < \psi < \psi c$</u>

$$\text{CDmult} = 1 + h\left[\sin\left(\frac{90\ \psi}{\psi c}\right)\right]^2 \qquad \ldots(\text{B1.10})$$

<u>For $\psi c < \psi < 180 - \psi c$</u>

$$\text{CDmult} = (1 + h)\cos\left[\frac{90\ (\psi - \psi c)}{90 - \psi c}\right] \qquad \ldots(\text{B1.11})$$

<u>For $180\text{-}\psi c < \psi < 180$</u>

$$\text{CDmult} = -\left(1 + h\left[\sin\left(\frac{90\ (180 - \psi)}{\psi c}\right)\right]^2\right) \qquad \ldots(\text{B1.12})$$

where	ψc	is the yaw angle in degrees at which CDmult is a maximum
	ψ	is the yaw angle (*ie* the apparent direction of the wind)
	h	is the proportionate increase in CD at angle ψc

The yaw angle is given by:

$$\psi = \sin^{-1}((Vw\ \sin(\chi))/vr) \qquad \ldots(\text{B1.13})$$

where	Vw	is the wind velocity in m/s
	χ	is the direction of the wind relative to the direction of travel

The velocity of the vehicle relative to wind can be calculated from Figure B1.2 as:

$$vr^2 = v^2 + Vw^2 + 2\ v\ Vw\ \cos(\chi) \qquad \ldots(B1.14)$$

The role and quantification of the parameter h needs to be explained. With reference to Figure B1.3, depending on the yaw angle the value for CDmult (=CD(ψ)/CD) varies in a non-linear fashion. For a certain yaw angle CDmult is at a maximum at this point CDmult = 1 + h, or, CD(ψ) = (1 + h) CD. In Figure B1.3 ψc is 30^o and h is 0.4 which are the values recommended by Sovran (1984) for passenger cars.

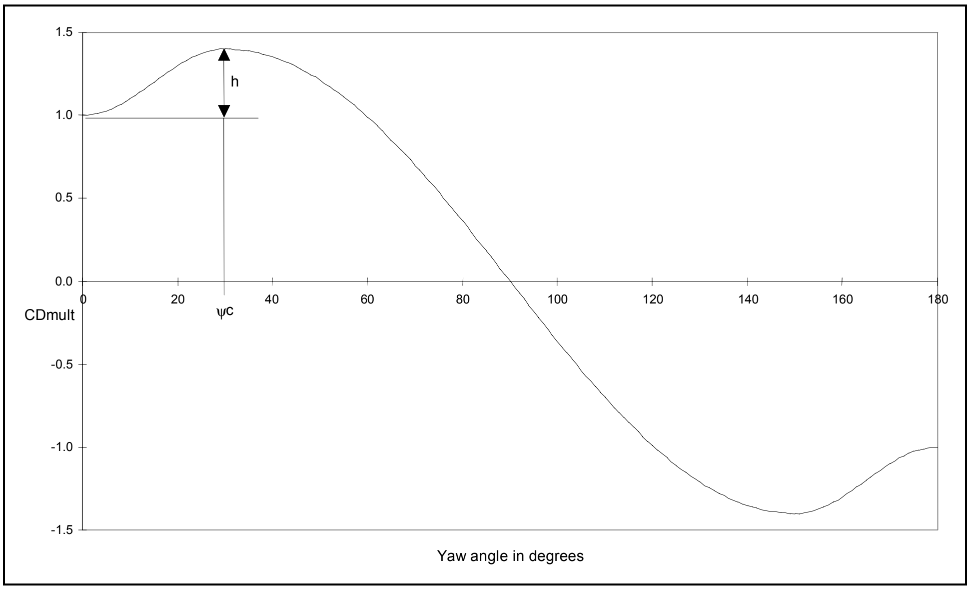


Figure B1.3: CDmult vs Yaw Angle

Wong (1993) indicates that ψc is also approximately 30^o for heavy vehicles also. Biggs (1995) gives the following values of h for trucks which compare well with the data in Wong (1993):

0.6	rigid trucks
0.8	single trailer trucks
1.2	double trailer trucks

In order to calculate an average value for CDmult it is necessary to allow for the distribution of wind directions acting on a vehicle.

Ingram (1978) calculated wind-averaged values of CD for heavy goods vehicles by establishing the distribution of wind speeds and directions imposed on a vehicle on various motorways around Britain. He compared his detailed values with those which would have arisen under the assumption that on a network there is an equal probability of wind arriving at all angles relative to the centre line of the roads. He concluded that:

> *[using] "the approximation reached by assuming that the national average wind speed is equally probable from all directions is quite adequate, giving an error of less than one per cent".*

This does not imply that the wind has an equal probability of arriving from all directions, as most localities have predominant wind patterns, but rather that the combined distribution of wind and road angles is equally distributed. Consequently, there is little use in undertaking a detailed analysis of wind direction, such as that done by Ingram (1978).

Using the approximation that wind has an equal probability of arriving at any angle relative to the centre line of the road within a network in conjunction with Equations B1.10 to B1.12, a wind-averaged CD multiplier can be established. By considering the relative velocity of the vehicle in these calculations it is possible to rewrite Equation B1.9 as:

$$Fa = 0.5\ \rho\ CDmult\ CD\ AF\ v^2 \qquad \ldots(B1.15)$$

Thus, the value of CDmult implicitly models the ratio of vr^2/v^2.

In order to calculate values for CDmult an application was written as part of the suite of 'HDM Tools' software[1]. The CD multiplier is calculated for angles of χ between 0 and 359 degrees, in intervals of 1 degree. From the 360 values of yaw angle, the value for CDmult and relative velocity are calculated for each angle. The average[2] of these values was then divided by the square of vehicle speed to obtain the value for CDmult. This value can then be used to update a database which in turn can be imported to HDM-4 to update its vehicle data base. Figure B1.4 shows the input data screen for this application.

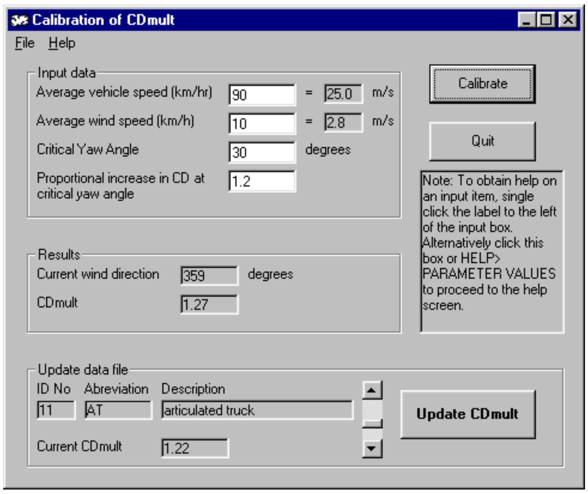


Figure B1.4: HDM Tools CDmult Calibration Program Screen

There are four data items required:

- Average vehicle speed;
- Average wind speed;
- Critical yaw angle (ψc); and,
- Proportional increase in CD at the critical yaw angle ('h').

As discussed earlier, it is recommended that ψc be assumed to be a constant value of 30^o for all vehicle classes. The value for h will vary between vehicle classes while the vehicle and wind speeds will vary depending on applications.

1 HDM Tools were written primarily by Ian Greenwood of Opus International Consultants Ltd. They consist of a suite of applications for assisting in calibrating the HDM models or defining key input data items. Details on obtaining HDM Tools are available from www.opus.co.nz/hdmtools or www.htc.co.nz.

2 The weight of the product at a wind angle of 0 and 180 degrees was halved to account for the full 360 degrees of possible wind angles.

To illustrate the sensitivity of CDmult to these three input data items, consider Figure B1.5. and Figure B1.6. Figure B1.5 shows CDmult as a function of vehicle speed and h, assuming a wind speed of 10 km/h. For a given value of h over a wide range of operating speeds the variation for most vehicles is less than 20 per cent. The average wind speed has a greater impact on the results, and so this should be estimated if possible based on local conditions.

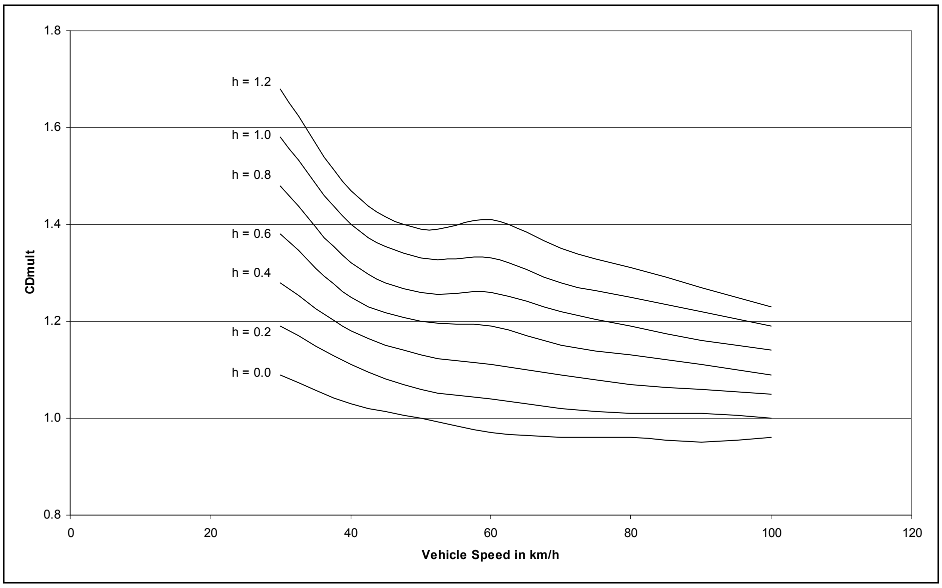


Figure B1.5: CDmult vs Vehicle Speed and h

It is therefore recommended that the analysis be conducted assuming an average vehicle speed of 75 km/h (21 m/s). This approximation results in a slight under prediction of the aerodynamic resistance at low vehicle speeds, and a slight over prediction at high vehicle speeds. At low vehicle speeds, the tractive force is dominated by rolling resistance, whereas at high vehicle speeds aerodynamic resistance dominates. Hence, it is more important to accurately model aerodynamic resistance at higher vehicle speeds.

For the wind speed a default value of 14 km/h (4 m/s) is recommended. This value is at the mid-range of the predominant wind speeds found by Ingram (1978).

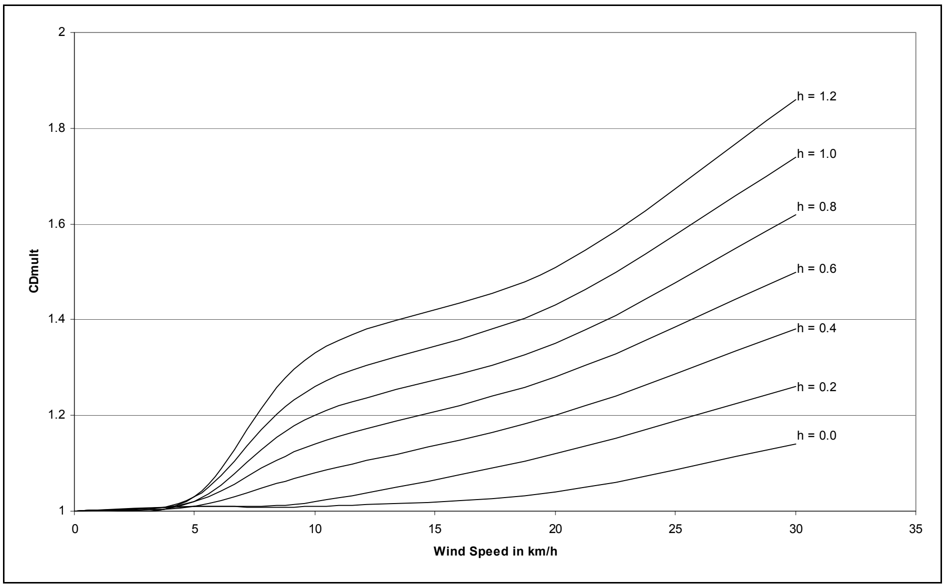


Figure B1.6: CDmult vs Wind Speed and h

Table B1.1 presents the recommended default values for use with Equation B1.15 based on the default wind speed of 14 km/h. This table also includes the values for h used in calculating CDmult.

Table B1.1: Aerodynamic Resistance Default Parameters

Vehicle Number	Type	h	CDmult[1/]	CD	AF (m^2)
1	Motorcycle	0.4	1.12	0.70	0.8
2	Small Car	0.4	1.12	0.40	1.8
3	Medium Car	0.4	1.12	0.42	1.9
4	Large Car	0.4	1.12	0.45	2.0
5	Light Delivery Vehicle	0.5	1.16	0.50	2.9
6	Light Goods Vehicle	0.5	1.16	0.50	2.8
7	Four Wheel Drive	0.5	1.16	0.50	2.8
8	Light Truck	0.6	1.19	0.55	4.0
9	Medium Truck	0.6	1.19	0.60	5.0
10	Heavy Truck	0.7	1.22	0.70	8.5
11	Articulated Truck	1.2	1.38	0.80	9.0
12	Mini-bus	0.5	1.16	0.50	2.9
13	Light Bus	0.6	1.19	0.50	4.0
14	Medium Bus	0.7	1.22	0.55	5.0
15	Heavy Bus	0.7	1.22	0.65	6.5
16	Coach	0.7	1.22	0.65	6.5

Source: *Author's estimates*
Notes: 1/ Calculated assuming wind speed of 14 km/h and vehicle speed of 75 km/h

B1.2.4 Projected Frontal Area

The projected frontal area, AF, is calculated as product of the maximum width of the vehicle by the maximum height less the area under the axles. Biggs (1988) indicates that the area under the axles is between five and 15 per cent for most vehicles. As with the aerodynamic drag coefficient, the projected frontal area is proportional to the yaw angle. However, these considerations have been included with the wind averaged aerodynamic drag coefficient so are not repeated here.

Table B1.1 presents the values for frontal area recommended for each of the representative vehicles. Updating of these values is straight forward as the data are readily available from manufacturer's specifications.

B1.2.5 Modelling Aerodynamic Resistance in HDM-4 - Summary

MODEL

The aerodynamic resistance model for HDM-4 is:

$$Fa = 0.5\ \rho\ CDmult\ CD\ AF\ v^2 \qquad \ldots(B1.16)$$

MASS DENSITY OF AIR

Given the range of climatic conditions over which HDM-4 is anticipated to be used, it is proposed that HDM-4 include a default value for the mass density of air which may be modified according to local conditions. The recommended default value is 1.2 kg/m^3 which is based on an air temperature of 15^{o}C and an altitude of 200 m. Equation B1.7 can be used to adjust this value for other temperatures or altitudes.

AERODYNAMIC DRAG COEFFICIENT AND VEHICLE VELOCITY

The aerodynamic drag coefficient at zero degrees yaw and the CDmult should be input by the user based on the values presented in Table B1.1.

PROJECTED FRONTAL AREA

The projected frontal area should be calculated from the actual vehicle characteristics. Values for the representative vehicles are given in Table B1.1.

B1.3 Rolling Resistance

The rolling resistance is described by Biggs (1988) as:

> *"... the total of all forces, apart from aerodynamic drag, acting on a free-wheeling vehicle (ie, with the clutch disengaged). Thus, it includes all frictional forces from the output of the gear box to the wheels and tyre resistance forces."*

In its simplest form the rolling resistance is calculated as:

$$Fr = M\ g\ CR \qquad \text{…(B1.17)}$$

where Fr is the rolling resistance in N
g is the acceleration due to gravity in m/s^2
CR is the coefficient of rolling resistance

There is an alternative formulation for this equation which takes into account vehicle lift. This aerodynamic effect serves to reduce the normal load on the pavement and therefore reduce the rolling resistance. However, for a model such as HDM-4 it was considered that the additional sophistication and data requirements of such a formulation were not warranted.

The rolling resistance is strongly influenced by tyre technology. Wong (1993) published data which indicates that for vehicles operating with speeds between five and 100 km/h, bias-ply tyres have an average of 28 per cent higher rolling resistances than radial tyres. This is similar to a value of 30 per cent reported by Biggs (1987). Francher and Winkler (1984) also reported much lower rolling resistances for radial compared to bias-ply tyres.

Cenek (1994) presents a detailed review of rolling resistance characteristics of roads. He shows that most studies (*eg* Bester, 1981; St. John and Kobett, 1978; CRRI, 1985; Brodin and Carlsson, 1986) have found that there is a speed dependence on CR. The common form of equation is:

$$CR = (Co + Cv\ v^{Cm}) \qquad \text{…(B1.18)}$$

where Co is the static coefficient of rolling resistance
Cv is the dynamic coefficient of rolling resistance in $1/(m/s)^{Cm}$
Cm is an exponent

The values reported for Cm are usually 2 for light vehicles and 1 for all other vehicles[1].

The static coefficient of rolling resistance was related to roughness by du Plessis, *et al.* (1990) who developed a model for trucks which used the roughness and tyre pressure as

[1] Cenek (1994) indicates that the value of 1 for Cm with heavy vehicles arises because with increasing speed there is an increase in the vertical stiffness of the tyre due to the centrifugal force increasing the vertical tension on the belt.

independent variables. Substituting in a standard value of 640 kPa results in the following equation:

$$Co = 0.00874 + 0.00043\ IRI \qquad \ldots(B1.19)$$

In HDM-III (Watanatada, *et al*., 1987a), only roughness was considered to influence CR and their formulation ignored the speed effects. This led to the following equations:

$$CR = 0.0218 + 0.00061\ IRI \quad \text{for cars and LCVs} \qquad \ldots(B1.20)$$

$$CR = 0.0139 + 0.00026\ IRI \quad \text{for buses and HCVs} \qquad \ldots(B1.21)$$

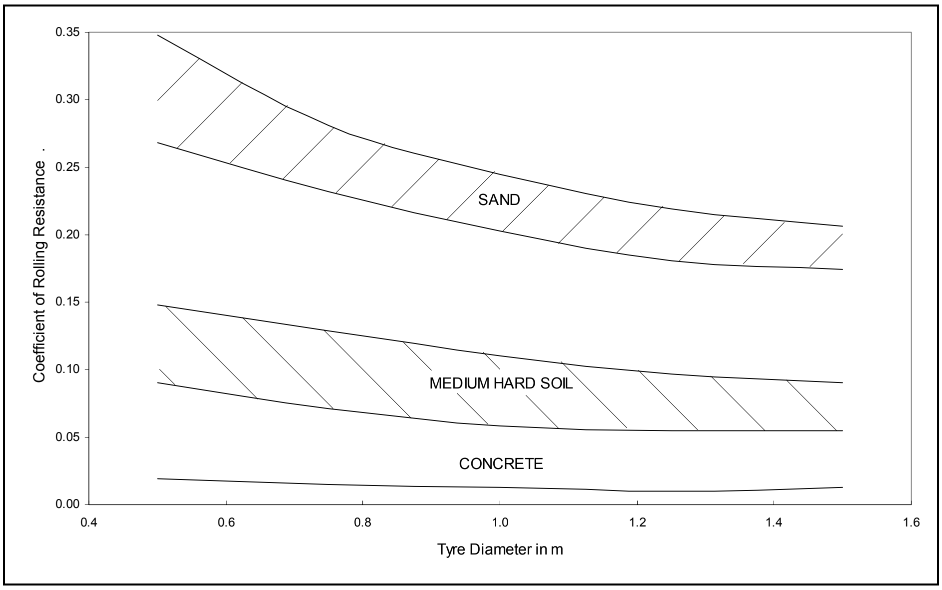
where IRI is the roughness in IRI m/km

The importance of texture in rolling resistance is highlighted by Cenek (1994) who states:

> *"[A 4 per cent reduction] in fuel consumption will be achieved by reducing surface texture depth from 2.2 mm to 1.4 mm as by reducing the roughness from 5.7 IRI to 2.7 IRI."*

Cenek (1994) concluded that truck tyres are not as sensitive to texture as passenger car tyres based on a study in the UK. This found that whereas texture would increase rolling resistance by 44 per cent for passenger cars, there would only be a six per cent increase in truck tyre resistance. This is probably due to the larger dimensions and higher inflation pressures of truck tyres.

Surface type has a major influence on the rolling resistance. Figure B1.7 illustrates the effect of tyre diameter on the rolling resistance of various surfaces (Wong, 1993). This figure also shows that tyre diameter has a greater effect on deformable or soft ground.



Source: Wong (1993)

Figure B1.7: Coefficient of Rolling Resistance for Various Surface Typres

Francher and Winkler (1984) multiplied the rolling resistance (CR) by the following factors as a function of pavement type:

Smooth concrete	1.0
Worn concrete, cold blacktop	1.2
Hot blacktop	1.5

Biggs (1988) presented the following formula for the calculation of rolling resistance which has was the basis for the HDM-4 equation:

$$Fr = CR2 \, (b11 \, Nw + CR1 \, (b12 \, M + b13 \, v^2)) \qquad \ldots(B1.22)$$

where	CR1	is a rolling resistance tyre factor
	CR2	is a rolling resistance surface factor
	b11, b12, b13	are rolling resistance parameters
	Nw	is the number of wheels

The coefficient CR1 is a function of tyre type and was given by Biggs (1988) as:

CR1	= 1.3	for cross-ply bias
	= 1.0	for radial
	= 0.9	for low profile tyres

Biggs (1988) indicates that coefficients b11 to b13 are a function of wheel diameter:

$$b11 = 37 \, Dw \qquad \ldots(B1.23)$$

$$b12 = 0.067/Dw \quad \text{for old model tyres} \qquad \ldots(B1.24)$$

$$= 0.064/Dw \quad \text{for latest model tyres} \qquad \ldots(B1.25)$$

$$b13 = 0.012 \, Nw/Dw^2 \qquad \ldots(B1.26)$$

where Dw is the diameter of the wheels in m

Bennett (1997) proposed the following model for the rolling resistance surface factor[1]:

$$CR2 = Kcr2 \, [a0 + a1 \, Tdsp + a2 \, IRI] \qquad \ldots(B1.27)$$

where	Kcr2	is a calibration factor
	Tdsp	is the texture depth in mm from the sand patch method[2]
	a0 to a2	are coefficients

Information from Cenek (1995) led to the values for CR2 in Table B1.2. These are based on the work of Walter and Conant (1974), Knoroz and Shelukhin (1964) which are similar to those given by Biggs (1988).

Table B1.2: Bennett (1997) Parameters for CR2 Model

Surface Class	Surface Type	Operating Weight <= 2500 kg				Operating Weight > 2500 kg			
		a0	a1	a2	Kcr2	a0	a1	a2	Kcr2
Bituminous	AM or ST	0.90	0.022	0.022	1	0.84	0.030	0.030	1
Concrete	JC or CR	0.90	0.022	0.022	1	0.64	0.030	0.030	1
Unsealed	GR	1.00	0.00	0.075	1	1.00	0.000	0.075	1
Unselaled	-	0.80	0.00	0.100	1	0.80	0.000	0.100	1
Block	CB, BR or SS	2.00	0.00	0.000	1	2.00	0.000	0.000	1
Unsealed	SA	7.50	0.00	0.000	1	7.50	0.000	0.000	1

1 This is a change from the model in NDLI (1995) model which included Tdsp and IRI in a single expression.

2 HTC (1999c) report the following equation to convert between the sand patch and mean profile texture depth: Tdsp = 1.02 MPD + 0.28.

Cenek (1999) analysed data from Cenek (1994) and Jamieson and Cenek (1999) to quantify CR2 as a function of texture depth, roughness and benkelman beam deflection. This resulted in the following equations for surface dressed pavements:

$$CR2 = 0.50 + 0.02\ Tdsp + 0.10\ IRI \qquad \text{Cars} \qquad \ldots(B1.28)$$

$$CR2 = 0.57 + 0.04\ Tdsp + 0.04\ IRI + 1.34\ DEF \qquad \text{Trucks} \qquad \ldots(B1.29)$$

where DEF is the Benkelman Beam rebound deflection in mm

Not only do these equations predict lower values for CR2 than the earlier work of Bennett (1997), but they show that for surface dressed pavements by far the greatest factor is the deflection. Cenek (1999) notes that the R^2 for the truck equation was low (0.33), but this reflected the fact that the data set had tyres of various sizes in it. Rather than including tyre size as an independent variable it was considered better to accept a lower level of predictive ability from the equation.

On the basis of this work, the following is the proposed CR2 model for HDM-4 with the coefficients given in Table B1.3.

$$CR2 = Kcr2\ [a0 + a1\ Tdsp + a2\ IRI + a3\ DEF] \qquad \ldots(B1.30)$$

Table B1.3: Final Parameters for CR2 Model

Surface Class	Surface Type	Operating Weight <= 2500 kg				Operating Weight > 2500 kg			
		a0	a1	a2	a3	a0	a1	a2	a3
Bituminous	AM or ST	0.50	0.02	0.10	0	0.57	0.04	0.04	1.34
Concrete	JC or CR	0.50	0.02	0.10	0	0.57	0.04	0.04	0
Unsealed	GR	1.00	0.00	0.075	0	1.00	0.000	0.075	0
Unselaled	-	0.80	0.00	0.100	0	0.80	0.000	0.100	0
Block	CB, BR or SS	2.00	0.00	0.000	0	2.00	0.000	0.000	0
Unsealed	SA	7.50	0.00	0.000	0	7.50	0.000	0.000	0

Kcr2 has a default value of 1.0.

Biggs (1995) indicates that the effect of snow is to increase CR2 by 0.2 to 0.4. Cenek (1995) suggests an increase in CR2 of 20 per cent for wet surfaces. The climatic effects can therefore be considered with the following equation:

$$FCLIM = 1 + 0.003\ PCTDS + 0.002\ PCTDW \qquad \ldots(B1.31)$$

where FCLIM is a climatic factor related to the percentage of driving done in snow and rain

PCTDS is the percentage of driving done on snow covered roads

PCTDW is the percentage of driving done on wet roads

The default values of PCTDS and PCTDW are both zero in HDM-4.

This yields the following formula for calculating the average rolling resistance of vehicles which has been adopted for HDM-4:

$$Fr = CR2\ FCLIM\ (b11\ Nw + CR1(b12\ M + b13\ v^2)) \qquad \ldots(B1.32)$$

Table B1.4 lists the recommended parameter values for the various rolling resistance model coefficients except CR2 whose values are established based on the information in Table B1.3. The values of CR1 are dependent on the tyre type and these are also given for each representative vehicle.

Table B1.4: HDM-4 Representative Vehicle Rolling Resistance Model Parameters

Vehicle Number	Type	Number of Wheels	Wheel Diameter (m^2)	Type of Tyre	CR1	b11	b12	b13
1	Motorcycle	2	0.55	Bias	1.3	20.35	0.1164	0.0793
2	Small Car	4	0.60	Radial	1.0	22.20	0.1067	0.1333
3	Medium Car	4	0.60	Radial	1.0	22.20	0.1067	0.1333
4	Large Car	4	0.66	Radial	1.0	24.42	0.0970	0.1102
5	Light Delivery Vehicle	4	0.70	Radial	1.0	25.90	0.0914	0.0980
6	Light Goods Vehicle	4	0.70	Bias	1.3	25.90	0.0914	0.0980
7	Four Wheel Drive	4	0.70	Bias	1.3	25.90	0.0914	0.0980
8	Light Truck	4	0.80	Bias	1.3	29.60	0.0800	0.0750
9	Medium Truck	6	1.05	Bias	1.3	38.85	0.0610	0.0653
10	Heavy Truck	10	1.05	Bias	1.3	38.85	0.0610	0.1088
11	Articulated Truck	18	1.05	Bias	1.3	38.85	0.0610	0.1959
12	Mini-bus	4	0.70	Radial	1.0	25.90	0.0914	0.0980
13	Light Bus	4	0.80	Bias	1.3	29.60	0.0800	0.0750
14	Medium Bus	6	1.05	Bias	1.3	38.85	0.0610	0.0653
15	Heavy Bus	10	1.05	Bias	1.3	38.85	0.0610	0.1088
16	Coach	10	1.05	Bias	1.3	38.85	0.0610	0.1088

Source: Author's estimates

B1.4 Gradient Resistance

The gradient resistance is the force component in the direction of travel necessary to propel a vehicle up a grade (positive resistance) or down a grade (negative resistance). Figure B1.8 is a force resolution diagram for a vehicle on a gradient. The weight of the vehicle can be resolved into both a parallel and perpendicular component with respect to the direction of travel. Therefore, the gradient resistance is given as:

$$Fg = M\ g \sin(\theta) \qquad \ldots(B1.33)$$

where Fg is the gradient force in N

θ is the angle of incline in radians

Since θ is small, $\theta \approx \sin(\theta) \approx \tan(\theta)$, and $\tan(\theta) = GR$, where GR is the gradient as a decimal. Using this approximation Fg can be written as:

$$Fg = M\ GR\ g \qquad \ldots(B1.34)$$

The approximation that $\theta \approx \sin(\theta) \approx \tan(\theta)$ introduces a 1.1 per cent error on a gradient of 15 per cent.

B1.5 Curvature Resistance

When a vehicle traverses a curve, the tyres deform by a finite amount resulting in a small angle (slip angle) occurring between the direction of travel and the direction of the wheels. Due to this slip angle, an additional force against motion (the curvature resistance) is placed on the vehicle. This curvature resistance has been found to be proportional to the side force applied to the wheels and the slip angle, *ie*:

$$Fcr = Ff \tan(\Phi) \qquad \ldots(B1.35)$$

where Fcr is the curvature force in N

Ff is the side friction force in N

Φ is the slip angle

Figure B1.8: Resolving Forces on a Gradient

The slip angle Φ is a measure of the amount of roll (or deformation) in the tyres. Moore (1975) indicates that the slip angle is dependent upon the magnitude of the side force applied to each wheel and inversely to the stiffness of the tyre. Since Φ is small, tan(Φ) ≈ Φ. Hence, the curvature resistance can be written as:

$$Fcr = Ff\,\Phi \qquad \text{…(B1.36)}$$

The basic equation for the balance of forces as a vehicle traverses a curve is:

$$M\,\frac{v^2}{R} = M\,g\,e + M\,g\,f \qquad \text{…(B1.37)}$$

where	R	is the radius of the curve in m
	e	is the superelevation in m/m
	f	is the side friction factor

The formula for calculating the side friction factor of a vehicle traversing a curve is:

$$f = \frac{\frac{M\,v^2}{R} - M\,g\,e}{M\,g} \qquad \text{…(B1.38)}$$

which can be simplified to:

$$f = \frac{v^2}{R\,g} - e \qquad \text{…(B1.39)}$$

By dividing the side force Ff by the number of wheels and the stiffness of the tyres, the slip angle Φ can be calculated as:

$$\Phi = \frac{\left(\frac{M\ v^2}{R} - M\ g\ e\right)}{Nw\ Cs}\ x\ 10^{-3} \qquad \ldots(B1.40)$$

Substituting in the equations for the side friction force and the slip angle gives:

$$Fcr = \max\left(0, \frac{\left(\frac{M\ v^2}{R} - M\ g\ e\right)^2}{Nw\ Cs} 10^{-3}\right) \qquad \ldots(B1.41)$$

Thus, the curve force, is proportional to the vehicle speed, curve geometry, number of wheels and the tyre stiffness. The tyre stiffness is given by:

$$CS = Kcs\ [a0 + a1\ \frac{M}{Nw} + a2\ \left(\frac{M}{Nw}\right)^2] \qquad \ldots(B1.42)$$

where Kcs is a calibration coefficient

Typical values for tyre stiffness coefficient Cs (in kN/rad) are given in Table B1.5[1].

Table B1.5: Tyre Stiffness Model Parameters

Coefficient	Mass < =2500 kg		Mass > 2500 kg	
	Bias	Radial	Bias	Radial
a0	30	43	8.8	0
a1	0	0	0.088	0.0913
a2	0	0	-0.0000225	-0.0000114
Kcs	1	1	1	1

Source: Biggs (1988)

The value adopted for the superelevation e has a significant impact on the magnitude of the curve forces. Watanatada, *et al.* (1987a) proposed the following relationship for predicting superelevation as a function of the degree of curvature:

$$e = 0.012\ CURVE \qquad \text{paved roads} \qquad \ldots(B1.43)$$

$$e = 0.017\ CURVE \qquad \text{unpaved roads} \qquad \ldots(B1.44)$$

While these relationships are based on a correlation between the curvature and the superelevation, for the purposes of predicting the curve forces it is important that the relationship reflect driver behaviour since that is where the input data for the relationship will come from.

Various researchers have found that there is a relationship between the side friction factor f and curve design standards (Bennett, 1994). Drivers operating on roads with frequent low radii curves will accept much higher side friction levels than on higher standard roads. This is embodied in many design standards. Annex B.1.1 describes how the relationship between superelevation and radius was established. This relationship is given below.

1 For passenger car high performance tyres the value for a0 in the Cs model would be 54 (NDLI, 1995).

$$e = \max(0, 0.45 - 0.68 \ln (R)) \qquad \ldots(B1.45)$$

B1.6 Inertial Resistance

INTRODUCTION

When a vehicle accelerates it needs to overcome inertial resistance. The inertial effects of the various rotating parts (engine, wheels and drivetrain) serves to increase the vehicles dynamic mass over its static mass. This increased mass is termed the *effective mass*. Thus, inertial resistance is given by:

$$Fi = M' a \qquad \ldots(B1.46)$$

where	Fi	is the inertial resistance in N
	M'	is the effective mass in kg
	a	is the acceleration in m/s^2

Watanatada, *et al*. (1987b) defined the effective mass as:

$$M' = M + Mw + Me \qquad \ldots(B1.47)$$

where	Mw	is the inertial mass of the wheels in kg
	Me	is the inertial mass of the engine and drivetrain in kg

The following sections describe each of these contributors to effective mass.

INERTIAL MASS OF THE WHEELS

Watanatada, *et al*. (1987b) gave the following equation for the calculation of the inertial mass of the wheels:

$$Mw = \frac{Iw}{r^2} \qquad \ldots(B1.48)$$

where	Iw	is the moment of inertia of the wheels in $kg\text{-}m^2$
	r	is the rolling radius of the tyres in m

The moment of inertia of a wheel is given by the product of its mass and its radius of gyration. Thus, the total inertial effects for all the wheels on the vehicle is:

$$Mw = TMW \frac{rg^2}{r^2} \qquad \ldots(B1.49)$$

where	TMW	is the total mass of the wheels in kg
	rg	is the radius of gyration of the tyre in m

INERTIAL MASS OF THE ENGINE AND DRIVETRAIN

Watanatada, *et al*. (1987b) gave the following equation for the calculation of inertial mass of the engine:

$$Me = \frac{Ie\ H^2}{r^2} \qquad \ldots(B1.50)$$

where Ie is the moment of inertia of the engine in kg-m^2
H is the total gear reduction

Bester (1981) gave the following formula for the moment of inertia of a truck engine:

$$Ie = 0.169 + 0.0188\ DENG^2 \quad \text{for a petrol truck engine} \quad \ldots(B1.51)$$

$$Ie = 0.169 + 0.0251\ DENG^2 \quad \text{for a diesel truck engine} \quad \ldots(B1.52)$$

where DENG is the engine capacity in litres

TOTAL EFFECTIVE MASS

From combining the above calculations for Mw and Me, M' can be written as:

$$M' = M + \frac{Iw}{r^2} + \frac{Ie\ H^2}{r^2} \quad \ldots(B1.53)$$

which equates to:

$$M' = M + TMW\ \frac{rg^2}{r^2} + \frac{Ie\ H^2}{r^2} \quad \ldots(B1.54)$$

The ranges of Iw and Ie for a passenger car as reported by Burke, *et al*. (1975) and Watanatada, *et al*. (1987b) are given in Table B1.6.

Table B1.6: Moment of Inertia Values for Passenger Cars

	Burke, *et al*. (1975)	Watanatada, *et al*. (1987b)
Iw (kg-m^2)	1.223 to 1.495	0.89 to 1.01
Ie (kg-m^2)	0.543 to 0.678	not supplied

THE EFFECTIVE MASS RATIO

A more convenient method of representing the effective mass is through the ratio of effective mass to mass (EMRAT = M'/M). Bester (1981) indicates that EMRAT would be expected to equal 1.082 for a car and between 5.0 (at 6 km/h) and 1.033 (at 100 km/h) for a truck. Trucks exhibit a lower value of EMRAT at higher speeds than do light vehicles because of their higher static masses. While the magnitude of the effective mass for the light vehicles is much lower than for trucks, relative to their static mass it is higher.

Since the effective mass is dependent on the gears in use, and gear selection is a function of road speed, it is necessary to take both of these considerations into account as well as other physical attributes of the vehicle to calculate the effective mass and EMRAT. The program 'GEARSIM' was written to perform these calculations. Part of the 'HDM Tools' software[1], GEARSIM uses a Monte-Carlo simulation to predict driver gear selection and, thus, the engine speed of the vehicle. This information is then used to establish the effective mass at different road speeds and from that, the effective mass ratio.

[1] Details on obtaining HDM Tools are available from www.opus.co.nz/hdmtools or www.htc.co.nz.

Figure B1.9 is an example of the input screen for running GEARSIM. Annex B1.2 contains a description of the simulation logic and the underlying assumptions. The user must define the essential vehicle attributes for performing the simulation. It then simulates the gear selection of a user-defined number of vehicles in 5 km/h increments for mean speeds from 5 to 100 km/h. The coefficient of variation is used to define the spread of speeds around this mean.

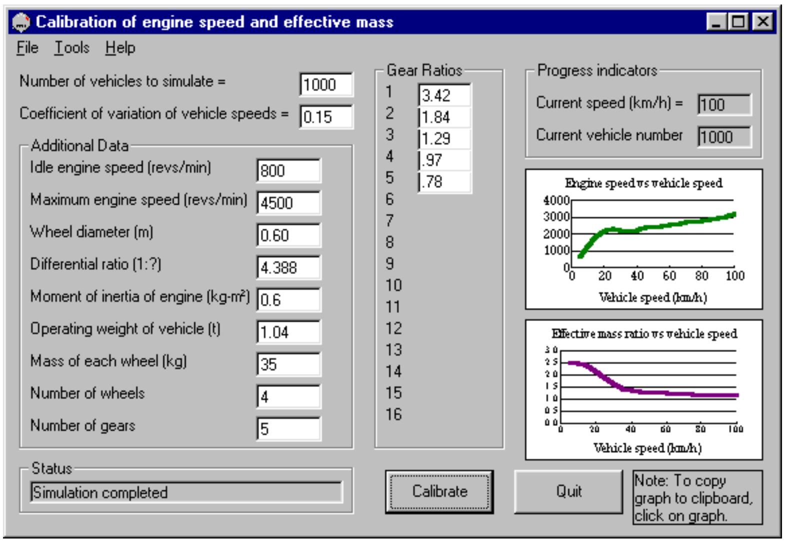


Figure B1.9: Example of GEARSIM Input Screen and Results

The program outputs two sets of data: the engine speed vs road speed and the EMRAT vs road speed. Figure B1.9 shows the graphs of these which are displayed when the simulation is completed. An example of the EMRAT profiles obtained from the simulation is shown in Figure B1.10 (NDLI, 1995). This clearly shows the decrease in EMRAT with increasing speed, and the differences between light and heavy vehicles.

From the data generated from the simulation the following general model which expresses the effective mass ratio as a function of velocity was adopted for HDM-4:

$$\text{EMRAT} = \text{a0} + \text{a1 atan}\left(\frac{\text{a2}}{\text{v}^3}\right) \qquad \ldots\text{(B1.55)}$$

where atan is the arc tan in radians

a0 to a2 are regression coefficients

Table B1.7 gives the values for the regression coefficients for the HDM-4 representative vehicle classes for use in the above model.

On the basis of the above discussion, the formula for the calculation of inertial resistance is:

$$\text{Fi} = \text{M EMRAT a} \qquad \ldots\text{(B1.56)}$$

It should be noted that HDM-4 itself does not consider inertial effects since it is predicated on constant speeds. However, the HDM Tools acceleration/fuel model uses these results to predict traffic interaction effects on fuel consumption.

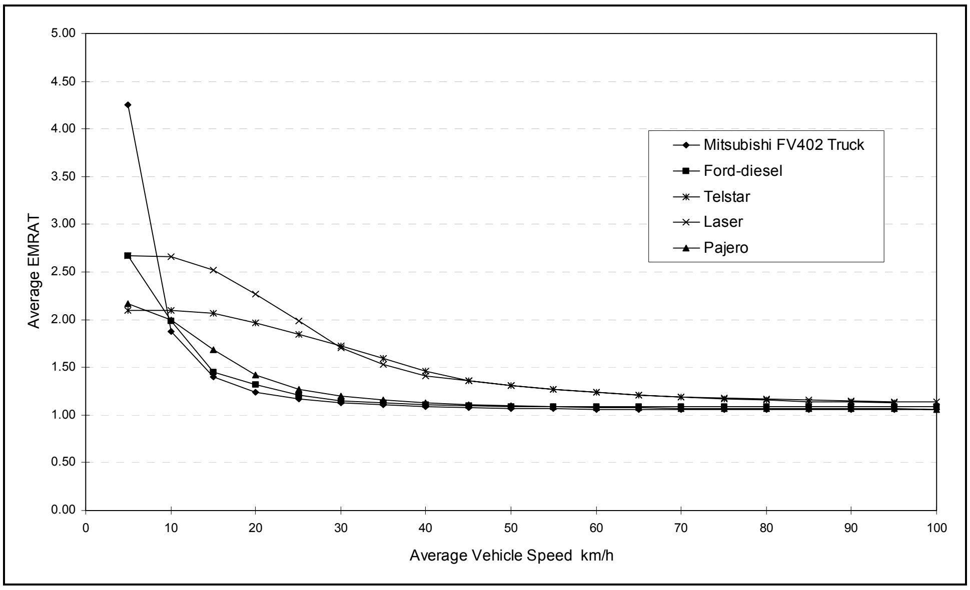


Figure B1.10: Average EMRAT versus Average Vehicle Speed for Selected Vehicles

Table B1.7: Coefficients for Predicting Effective Mass Ratio

Vehicle Number	Type	Effect Mass Ratio Model Coefficients			R^2	S.E.
		a0	a1	a2		
1	Motorcycle	1.10	0	0	-	-
2	Small Car	1.14	1.010	399.0	1.00	0.03
3	Medium Car	1.05	0.213	1260.7	0.88	0.04
4	Large Car	1.05	0.213	1260.7	0.88	0.04
5	Light Delivery Vehicle	1.10	0.891	244.2	0.95	0.11
6	Light Goods Vehicle	1.10	0.891	244.2	0.95	0.11
7	Four Wheel Drive	1.10	0.891	244.2	0.95	0.11
8	Light Truck	1.04	0.830	12.4	0.99	0.03
9	Medium Truck	1.04	0.830	12.4	0.99	0.03
10	Heavy Truck	1.07	1.910	10.1	0.96	0.13
11	Articulated Truck	1.07	1.910	10.1	0.96	0.13
12	Mini-bus	1.10	0.891	244.2	0.95	0.11
13	Light Bus	1.10	0.891	244.2	0.95	0.11
14	Medium Bus	1.04	0.830	12.4	0.99	0.03
15	Heavy Bus	1.04	0.830	12.4	0.99	0.03
16	Coach	1.04	0.830	12.4	0.99	0.03

Source: *Simulation of data as described in Greenwood and Bennett (1995)*

B1.7 Summary

The forces opposing motion can be quantified in terms of the vehicle characteristics. Wherever possible, VOC should be predicted using mechanistic principles since they allow for the extrapolation and application of the VOC equations to vehicles with different characteristics to those upon which the relationships were based.

On the basis of the discussion in this chapter the following are the equations recommended for calculating the forces opposing motion in HDM-4:

$$Fa = 0.5\ \rho\ CD\ CDmult\ AF\ v^2 \qquad \text{…(B1.57)}$$

$$Fr = CR2\ (1 + 0.003\ PCTDS + 0.002\ PCTW)\ (b11\ Nw + CR1(b12\ M + b13\ v^2)) \qquad \text{…(B1.58)}$$

$$Fg = M\ g\ GR \qquad \ldots(B1.59)$$

$$Fcr = \max \quad 0, \frac{\frac{M\ v^2}{R} \quad M\ g\ e^{\quad 2}}{Nw\ Cs}\ 10^{\ 3} \qquad \ldots(B1.60)$$

$$Fi = M \quad a0 + a1\ a \tan \ \frac{a2}{v^2} \quad a \qquad \ldots(B1.61)$$

The total tractive force can therefore be written as:

$$Ftr = Fa + Fr + Fg + Fcr + Fi \qquad \ldots(B1.62)$$

$$\begin{aligned} Ftr = {} & 0.5\ \rho\ CD\ CDmult\ AF\ v^2 \\ & + CR2\ (1 + 0.003\ PCTDS + 0.002\ PCTDW)\ (b11\ Nw + CR1\ (b12\ M + b13\ v^2)) \\ & + M\ g\ GR \ + \ \max \quad 0, \frac{\frac{M\ v^2}{R} \quad M\ g\ e^{\quad 2}}{Nw\ Cs}\ 10^{\ 3} \quad + M\left[a0 + a1\ a \tan\left(\frac{a2}{v^3}\right)\right] a \end{aligned}$$

$$\ldots(B1.63)$$

B2 Free speeds

B2.1 Introduction

In many projects the benefits from travel time savings—as represented by increased speeds—constitute a significant proportion of the overall benefits. Speeds also influence many of the road user effect (RUE) components, particularly fuel and tyre consumption. Accordingly, predicting speed is an important component of HDM-4.

This chapter presents the HDM-4 free speed prediction model. It commences with an overview of the influencing speeds. This is then followed by a description of the HDM free speed model and the application of the model in HDM-4. Speed-flow modelling is discussed in Chapter B3.

Box B2.1 Terminology

- **Spot speeds** are speeds measured at a point on the road. These are often called time speeds.
- **Journey speeds** are speeds over a section of road. These are often called space speeds.
- **Free speeds** are the speeds vehicles travel when unaffected by other traffic, but affected by road alignment.
- **Operating speeds** are speeds adopted when affected by other traffic and road alignment but at lower flow levels.
- **Congested speeds** are speeds adopted when affected by other traffic and road alignment at high flow levels (*ie* under congested conditions).
- **Desired speeds** are the 'ideal' speed that vehicles would travel at when unconstrained by other traffic or the geometry but affected by the overall alignment of the road and the road environment.

B2.2 Factors influencing speeds

There are a wide array of factors influencing the speed of a vehicle on a road. Figure B2.1 summarises the general groups of factors that influence speeds. Within each of these groups there is usually a number of individual factors. For example, Oppenlander (1966) in a review of 160 items in the literature listed over 50 specific factors that influence speeds. The following sections discuss each of these factors individually. Section B2.3 deals with the effect of road condition on speed in detail since this is what is most pertinant to HDM-4.

When considering the factors influencing speeds there are some which are invariably found to have an influence—such as grades and curves—and others which may or may not have impact depending upon local conditions. Table B2.1 illustrates this by comparing the results from four multivariate analyses of speed data. The individual studies considered more factors than listed here, this table only considers the common factors tested in all studies. It shows that very few factors were found to be significant in more than one study, with the results of the various studies often contradicting each other. It is for this reason that local calibration of speed models is always prudent.

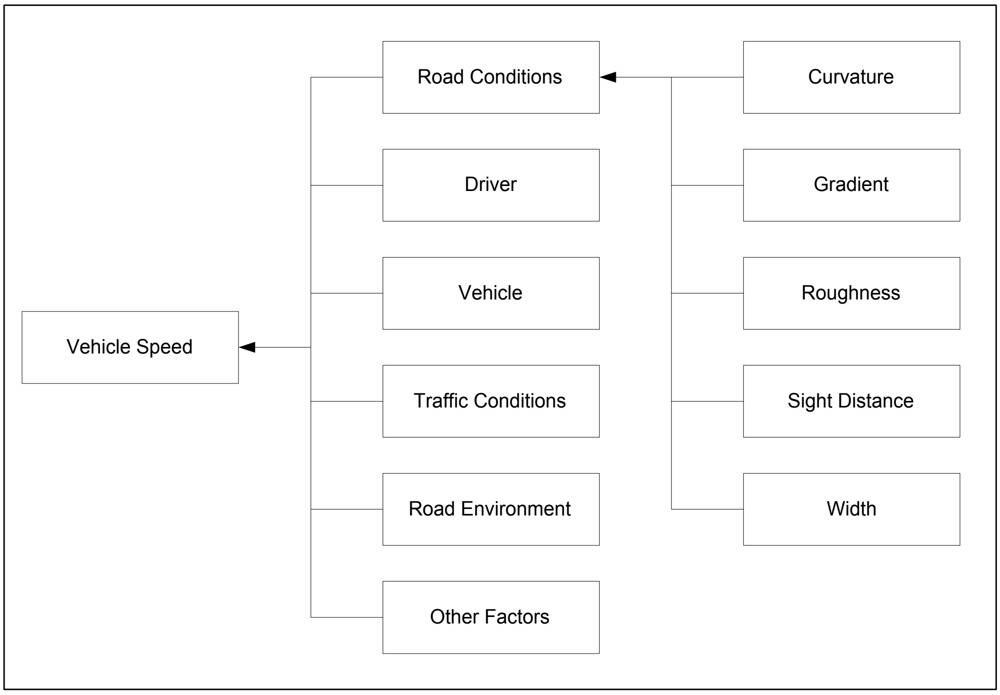


Figure B2.1: Factors Influencing Speeds

Table B2.1: Comparison of Multivariate Analysis Results

Factor	Wortman (1965)	O'Flaherty and Coombe (1971a,b,c)	Galin (1981)	Barnes (1988)
Country	USA	Britain	Australia	N.Z.
Type of Speed	Spot	Journey	Journey	Spot
Number of Factors Investigated	38	66	48	14
Minimum sight distance	✓	☒	-	-
Roadside establishments	✓	☒	✓	-
Combination trucks	✓	☒	-	-
Horizontal curvature	☒	✓	-	-
Two or more passengers	☒	✓	☒	☒
Bunching	-	✓	✓	-
Driver age	-	-	✓	✓
Driver sex	☒	☒	✓	☒
Vehicle age	-	-	✓	✓
Vehicle engine size	-	-	✓	✓
Weather (dry versus wet)	☒	☒	✓	-
Car ownership	-	-	☒	☒
Trip purpose	-	-	☒	☒
Trip distance	-	-	✓	✓

Notes: ✓ Significant effect on speed in multivariate analysis
☒ No significant effect on speed in multivariate analysis
\- Not tested in multivariate analysis

As an example of how the impact of road conditions can vary between locations, consider the results of the Kenya Road User Cost Study (Hide, *et al.*, 1975), the Caribbean Study (Morosiuk and Abaynayaka, 1982) and the India Study (CRRI, 1982). All developed multivariate models with the same fundamental structure. The models were of the form:

S = a1 + a2 RS + a3 F + a4 CURVE + a5 ALT + a6 BI + a7 PWR + a8 WIDTH

…(B2.1)

where RS is the rise in m/km
F is the fall in m/km
CURVE is the horizontal curvature in degrees/km
ALT is the altitude in metres above sea level
BI is the roughness in BI mm/km
PWR is the power to weight ratio in kW/t
WIDTH is the pavement width in m
a1 to a8 are regression coefficients

Each of the independent variables were not used in every study. Table B2.2 presents the values for the coefficients a1 to a8 by vehicle class. The Kenya and Caribbean models were based on journey speeds while the India model is based on journey speeds and spot speeds.

Table B2.2: Speed Coefficients for Kenya, Caribbean and India Models

Coeff.	Kenya Study[1/]				Caribbean Study[2/]			India Study[3/]		
	PC	LCV	CV	Bus	PC	LCV	CV	PC	CV	Bus
a1	102.60	86.90	68.10	72.50	67.60	62.60	51.90	55.7	46.5	51.2
a2	-0.372	-0.418	-0.519	-0.526	-0.078	-0.085	-0.222	-0.178	-0.175	-0.277
a3	-0.076	-0.050	0.030	0.067	-0.067	-0.067	-0.122	-0.155	-0.073	-0.159
a4	-0.111	-0.074	-0.058	-0.066	-0.024	-0.022	-0.017	-0.009	-0.014	-0.011
a5	-0.0049	-0.0028	-0.0040	-0.0042	-	-	-	-	-	-
a6	-	-	-	-	-0.00087	-0.00066	-0.00106	-0.0033	-0.0016	-0.0022
a7	-	-	-	-	-	-	0.559	-	-	-
a8	-	-	-	-	-	-	-	1.82	0.88	1.35

Source: *1/ Hide, et al. (1975)*
2/ Morosiuk and Abaynayaka (1982)
3/ CRRI (1982)

The coefficients in Table B2.2 illustrate the different effects of geometry on speeds measured in the three studies. The constant a1 represents the free speed and these differed by up to 51 km/h for the same vehicle class. Similarly, the effects of gradient and curvature on speed is also substantially different between the three studies. One again, this highlights the need to develop models based on local observations.

B2.2.1 Road Conditions

Traffic and highway engineers are primarily interested in the effects of road condition on speeds since these are the factors influenced by engineering decisions. The road condition factors consist of horizontal curvature, vertical gradient, pavement roughness, road width and sight distance. These are discussed in detail in Section B2.3.

B2.2.2 Driver

Since drivers seldom travel at the speed attainable by their vehicle, even in those countries without speed restrictions, the driver is usually the singularly most important factor governing speeds. A number of studies have investigated the full range of driver characteristics from age to familiarity with the road network to various socio-economic factors.

The findings of the studies have often been at odds with one another inasmuch as factors which were found to be significant in one study often proved to have no effect in other studies. This was illustrated in Table B2.1 which presented the results of some studies which used multivariate analyses to investigate a range of human factors and their influence on speeds.

B2.2.3 Vehicle

The vehicle characteristics dictate the performance limits of the vehicle, and thus its speed under certain conditions. The top speed of a vehicle is largely irrelevant in areas where there are speed limit restrictions as these are usually well below the attainable top speed. The ability to accelerate, which depends upon the used power-to-weight ratio, is important, especially where there are steep upgrades. The level of driving comfort provided by the vehicle can be a factor as noisy vehicles with high vibrations will often travel slower than quiet, smooth vehicles (Winfrey, 1969).

There has been a continued improvement in vehicle technology over time. For example, Mannering and Kilareski (1990) show that aerodynamic drag coefficients have decreased in the 20 years since 1968 from approximately 0.45 to 0.35, which is approaching the lower practical limit for this characteristic. There have also been major reductions in vehicle mass, particularly since the 1973 oil shock, increased engine efficiencies, as well as improvements to the suspension systems and tyre traction.

As a consequence of these changes, the performance of a modern passenger car is such that there is no physical reason why the speed of the vehicle should be influenced by moderate upgrades. Similarly, the improved traction of modern tyres makes it possible to traverse curves well in excess of the posted “safe speeds” without losing control. The ride offered by modern vehicles means that it is unlikely that drivers are even aware of roughness levels which in previous years would have caused them to reduce speeds. Because of these changes, it is imprudent to apply speed prediction models which were derived some years ago based on older fleets and it must be ensured that the current vehicle fleet is being modelled in any analyses.

B2.2.4 Traffic Conditions

On two-lane highways speeds are influenced through the mechanism of bunching. A stream of traffic is comprised of a population of vehicles each with their own desired speed. As the traffic volume increases faster vehicles catch up to slower ones. If there is an overtaking opportunity the faster vehicle will often pass and become free again, otherwise it will become a following vehicle until such a time as an overtaking opportunity presents itself. Thus, the speed of vehicles is dependant upon the volume and the available gaps in the opposing traffic stream.

Multi-lane highways have a similar effect, although the ability of users to change lanes greatly increases the ability to overtake. This is reflected in the capacities of multi-lane vs two-lane highways where muli-lane highways usually have on the order of a 50 per cent higher per lane capacity than two-lane highways.

Traffic effects on speed are discussed in detail in Chapter B3.

B2.2.5 Road Environment

The road environment has an influence on speeds through its impact on the desired speed of travel. High standard roads with good alignments will induce a higher desired speed than lower standard roads, as will flat versus rolling or mountainous terrain.

McLean (1981) gave values for the 85th percentile desired speed as a function of terrain and road type in Australia. These data were updated by AUSTROADS (1989) and are presented in Table B2.3. The data in this table indicate that wide variations, up to 50 km/h, will arise in desired speeds between diverse road environments.

Table B2.3: 85th Percentile Desired Speed by Terrain

Overall Design Speed (km/h)	Desired Speed (km/h)			
	Flat	Undulating	Hilly	Mountainous
40 - 50			75	70
50 - 70		90	85	
70 - 90		100	95	
90 - 120	115	110		
> 120	120			

Source: AUSTROADS (1989) and McLean (1981)

Not only does terrain influence speeds, but also the proximity to urban areas. For example, McLean (1981) found that speeds were lower leaving an urban area than travelling into it.

B2.2.6 Other Factors

WEATHER

In common with many other factors, the literature often presents contradicting results for the effects of rain and snow on speeds.

CRRI (1982) found in a study at a single site in India that the wetness of the road significantly affected the speeds. The same was found in Sweden (Kolsrud, 1985a) where mean speeds were reduced by one to two km/h. Olsen, *et al.* (1984) in the USA analysed 1175 hours of speed data for wet and dry pavements to investigate the effects of weather on speed. It was concluded that there were no weather effects since the majority of the sites showed no changes in speed and for those sites where there was a change, it was usually small enough to be ignored. McLean (1978c) also found no differences between speeds measured on wet versus dry days.

Winter conditions are reported to lead to speed reductions. Kolsrud (1985a) reports average speed reductions for cars of approximately eight km/h in Sweden, with up to 20 km/h on “fast” roads. In Canada, Yagar (1981a) found that winter speeds were five km/h below summer speeds. When the pavements became icy the speeds were 20 km/h below summer, and when snow covered and very slippery they were 30 km/h below summer speeds (Yagar, 1981a).

VISIBILITY

Light conditions affect the visibility on a road and can therefore be expected to influence speeds. CRRI (1982) in a study of day versus night speeds at a single site in India found that bus speeds were unaffected by light conditions while car speeds were reduced by approximately five per cent. Yagar (1981a) found in Canada that darkness reduced speeds by five km/h and that these effects were compounded by other factors such as weather and poor surface condition. However, Wahlgren (1967) did not find a statistically significant difference between speeds under good or poor visibility. Barnes and Edgar (1984) report a marginal reduction in night over day speeds in N.Z.

Bennett (1994) compared day and night spot speeds at 42 sites in N.Z. The speeds were measured at more than one location at each site so tests were made by speed measurement location as well. Speeds were therefore available for a total of 154 site-locations. It was found that 70 per cent of the site-locations did not have a significant difference in the speeds and that when there was a difference, some locations at the same site did not have a difference. It was concluded that the majority of vehicles were not affected by night conditions.

SPEED RESTRICTIONS

The international literature presents consistent observations on the influence of speed restrictions on vehicle speeds. When a speed limit is reduced, there is a corresponding decrease in speeds. These may be small, such as were observed in Finland when the move from an unposted to posted speed limit resulted in a two km/h decrease in speeds (Salusjärvi, 1981), or relatively large such as those in Canada where the 85th percentile speeds on motorways decreased by 16 km/h when the speed limit was reduced from 120 to 100 km/h (Gardner, 1978). In N.Z. MOT(1984) found there were major decreases in mean speeds at some sites, and minor increases at others. Yagar and van Aerde (1983) undertook a multivariate analysis of speed data from Canada and found that there was a significant effect of the posted speed limit on speeds. There is also a decrease in the standard deviations of speeds, for example in Canada it was reduced from 12.3 km/h to 8.8 km/h on motorways (Gardner, 1978).

Although reducing speed limits has an immediate effect, this is often not maintained unless there is adequate enforcement. Over time the speeds tend to increase, often surpassing their pre-change levels. MOT (1984) describes a fairly steady annual increase in speeds over time after N.Z. reduced their rural road speed limit in 1973 to 80 km/h from 100 km/h. By 1984 the posted speed limit bore little resemblance to the operating speeds since it was exceeded by 91 per cent of the cars, with 70 per cent exceeding 90 km/h. The resulting speed distribution was similar to those found in Australia for 100 or 110 km/h limit areas.

Increasing a speed limit may have an impact on speeds, but this depends upon whether the pre-change speed limit was rigorously enforced. For example, Kolsrud, *et al.* (1985b) concluded in Sweden after reviewing historical speed data that “a raised speed limit is accompanied by a certain speed increase”. However, Barnes and Edgar (1987) concluded “It would appear that the increased speed limit has had little impact on car speeds outside sections of open [rural] road having high design standards. This tends to support a hypothesis that drivers habitually select their speed on their perceptions of the capability of their vehicle and the road environment and hence are little influenced by speed limits until presented with high standard roads together with an enforcement presence.”

B2.3 The Effect of Road Conditions on Speed

B2.3.1 Upgrades

INTRODUCTION

The force balance equation presented in Chapter B1 indicates that in order for a vehicle to maintain its speed on an upgrade, the tractive force must exceed the gravitational, aerodynamic, curvature and rolling resistances. The tractive force depends upon the used power, *ie*:

$$Ftr = \frac{1000\ Pd}{v} \qquad \ldots(B2.2)$$

where Pd is the used driving power delivered to the wheels in kW
v is the vehicle velocity in m/s

In Equation B2.2 the tractive power is expressed in terms of the used driving power (Pd) as opposed to the rated engine power (Prat). The used power is always less than the rated power because of losses in the drivetrain . Also, since the maximum rated power arises at a set engine speed and it is impractical for the vehicle to always operate at this engine speed, driver behaviour further influences the power usage.

The tractive force is calculated as:

$$Ftr = Fa + Fr + Fg + Fc + Fi \quad \text{...(B2.3)}$$

where	Fa	Aerodynamic drag resistance in N
	Fr	Rolling resistance in N
	Fg	Gradient resistance in N
	Fc	Curvature resistance in N
	Fi	Inertial resistance in N

Figure B2.2 illustrates the forces acting on a heavy truck on an eight per cent gradient as a function of speed (Bennett, 1994). The figure presents the aerodynamic drag, rolling and gradient resistances along with the available engine force. The difference between the available engine power and the power used to overcome the forces opposing motion is termed the available driving power and this is also illustrated in the figure.

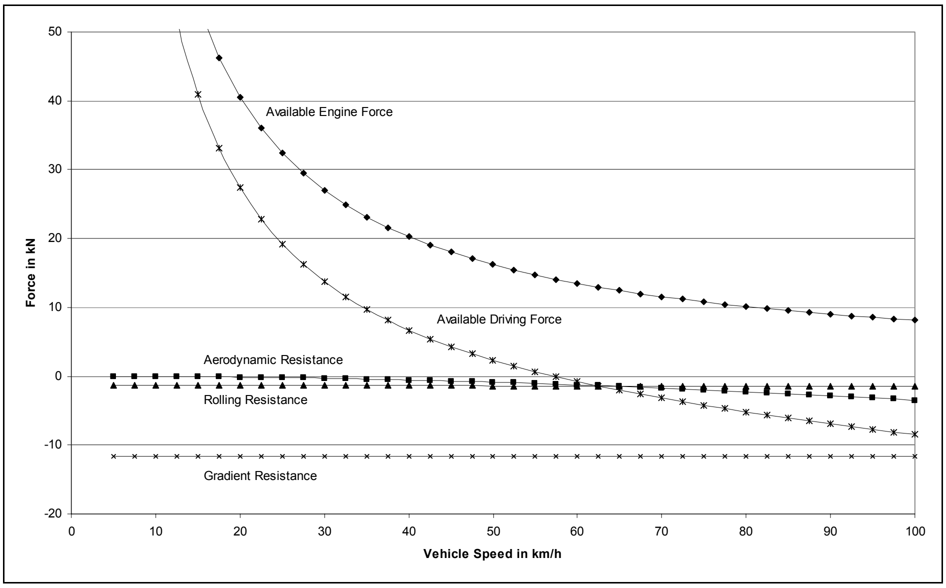


Figure B2.2: Forces Acting on Truck on 8 per cent Upgrade as a Function of Speed

In Figure B2.2 it can be observed that the available drive force is negative above 60 km/h. This is because the magnitude of the forces opposing motion are greater than the available engine force. The speed at which the available drive force is zero represents the terminal or ‘crawl’ speed of the vehicle. This is the steady state speed that the vehicle can maintain on the upgrade and it arises when the forces are in balance.

When the available drive force is positive, the vehicle has an ‘acceleration reserve’ (AR). This means that there is a net force available to accelerate the vehicle. If the driver uses this AR the used power may approach the available drive power. Alternatively, the driver may choose not to use this AR and so the used power will be equal to the forces opposing motion. A constant speed will thus be maintained on the upgrade.

The AR of a vehicle can be established from the force-balance equation. Substituting for the effective mass ratio and expressing the equation in terms of the acceleration, the following is the fundamental equation of motion for a vehicle:

$$a = \frac{1000\ Pd}{M\ EMRAT}\frac{1}{v} - \frac{1}{M\ EMRAT}\left[Fa + Fr + Fg + Fc\right] \qquad \ldots(B2.4)$$

where a is the acceleration in m/s^2
M is the vehicle mass
EMRAT is the effective mass ratio

Equation B2.4 indicates that the ability of a vehicle on an upgrade to accelerate is dependent on the used power-to-weight ratio, mass and the various forces opposing motion. For vehicles with low power-to-weight ratios, such as trucks, the vehicle will decelerate to a crawl or terminal speed where forces are in balance and the acceleration is equal to zero. This terminal speed can thus be obtained by solving Equation B2.4 for the velocity term. Thus, the limiting speed of a vehicle on a grade is a function of the power-to-weight ratio and, to a lesser degree, other vehicle attributes.

The gradient-speed research can be considered in two groups: those based on the power-to-weight ratio—which are termed **mechanistic models**—and those which used multivariate techniques. Each of these are addressed separately below.

MECHANISTIC UPGRADE SPEED MODELS

As shown above, the power-to-weight ratios plays a vital role in establish the speed of a vehicle on a gradient. If the power-to-weight ratio can be estimated, the speed of the vehicle can be calculated by solving the fundamental equation of motion for velocity. Because of their importance, power-to-weight ratios have received a fair deal of attention in the literature.

McLean (1989) indicates that two approaches have been used: determining the power-to-weight ratio for design trucks and determining it for typical vehicles. The design truck approach is used to derive design standards to ensure that traffic performance satisfies minimum criteria. Table B2.4 shows truck power-to-weight ratios from a number of different studies (McLean, 1989). By comparison, the values for passenger cars and range from 27.4 to 86.5 W/kg. Abaynayaka, *et al.* (1977) found in Ethiopia that once the power-to-weight ratio reached 18.6 W/kg, vehicle characteristics no longer were a factor governing truck speeds on gradients.

Table B2.4: Power-to-Weight Ratios Used in Design and Traffic Analyses

Source	Country	Basis of Value	Power-to-Weight Ratio (W/kg)	Crawl Speed on Six per cent Gradient (km/h)
1965 Highway Capacity Manual	U.S.A.	Typical Truck	5.1	23.2
Wright and Tignor (1965)	U.S.A.	Design Truck	4.1	18.8
AASHO (1965)	U.S.A.	Design Truck	4.1	18.8
Walton and Lee (1977)	U.S.A.	Design Truck	4.3	23.0
St. John and Kobett (1978)	U.S.A.	Design Truck	5.5	29.3
Ching and Rooney (1979b)	U.S.A.	Mean equilibrium gradient speed	10.0	51.6
Gynnerstedt, *et al.* (1977)	Sweden	Median Five+ axle truck-trailer	4.4	23.6
CRRI (1982)	India	Median	3.6	19.2

Source: McLean (1989)

There are two methods which can be used to calculate the power-to-weight ratio: interviewing vehicles as to their rated power or estimating it from observed speeds. Bennett (1988) calculated values this way having stopped a sample of vehicles, weighing them and recording their rated engine powers. However, it was noted that this did not accurately reflect the used engine power since it was necessary to multiply the rated engine power by an assumed drivetrain efficiency and an assumed power utilisation factor.

The speed approach uses recorded speeds on upgrades. Rearranging Equation B2.4 gives:

$$\frac{1000\ \text{Pd}}{\text{M EMRAT}} = \text{a v} + \frac{\text{v}}{\text{M EMRAT}}[\text{Fa} + \text{Fr} + \text{Fg} + \text{Fc}] \qquad \text{...(B2.5)}$$

On steep grades the acceleration tends to 0 as vehicles reach their crawl speed. Equation B2.5 can therefore be solved for Pd having observed the speed and used vehicle attributes to calculate the other forces opposing motion. McLean (1989) gives a variation of the above equation employing the speed measured at two points on a grade can be also used. This eliminates the need to have vehicles travelling at their crawl speed . The equation for applying this is:

$$\frac{1000\ \text{Pd}}{\text{M EMRAT}} = \frac{v_1^2 - v_0^2}{2\ \text{TSEC}} + \frac{v'}{\text{M EMRAT}}[\text{Fa} + \text{Fr} + \text{Fg} + \text{Fc}] \qquad \text{...(B2.6)}$$

where	v_0	is the velocity at the entrance to a section in m/s
	v_1	is the velocity at the end of the section in m/s
	TSEC	is the time taken to traverse the section in s
	v'	is the average speed in m/s calculated as the section length divided by the time[1]

As shown in Bennett (1994), who analysed the same data using both methods, similar results are obtained irrespective of which is used.

In order to apply either of the above equations it is necessary to have an estimate of the vehicle mass. While it may be convenient to assume a mean mass and use that to estimate the power, this will lead to a bias in the results since the crawl speed is non-linear as a function of mass. Bennett (1994) and HTC (1999a) used a Monte-Carlo simulation program to get around this problem. For each observed speed on a grade a large number of masses were randomly drawn from a distribution. The used power was estimated for each mass, and the mean used power for the vehicle was then calculated[2]. Table B2.5 shows the results from these two analyses for similar vehicle classes. The differences between the two countries are due to the use of quite different weight distributions as well as vehicle characteristics.

Table B2.5: Used Power Estimates from N.Z. and Thailand

Vehicle Description	Power-to-Weight Ratio (kW/t)		Used Power (kW)	
	N.Z.	Thailand	N.Z.	Thailand
Passenger Car	26.3	19.1	28.6	24.4
Light Truck	13.5	15.2	26.8	25.5
Medium Truck	11.1	8.9	82.2	41.6
Heavy Truck	11.3	6.9	132.9	89.0
Heavy Truck Towing	6.5	6.9	213.1	219.4

Source: Bennett (1994) and HTC (1999a)

1 v' should also be used in place of v in calculating the forces opposing motion. It should be noted that T is missing in the denominator of the speed term in McLean (1989).

2 The software used by HTC (1999a) is available at www.htc.co.nz.

Both of these analyses emphasise that the used power is in fact a distribution and that this varies with gradient: the steeper the gradient the higher the power usage.

It is important that the distributions be calibrated to local conditions. An example of this is shown in Figure B2.3 which gives the passenger car used power-to-weight ratio distributions from Indonesia (Sweroad and Bina Karya, 1994), Sweden (Brodin and Carlsson, 1986), N.Z. (Bennett, 1994) and Thailand (analysis of data from HTC, 1999a). Adopting an inappropriate distribution would result in an under or overestimate of the used power and thus have a major impact on the predicted speed.

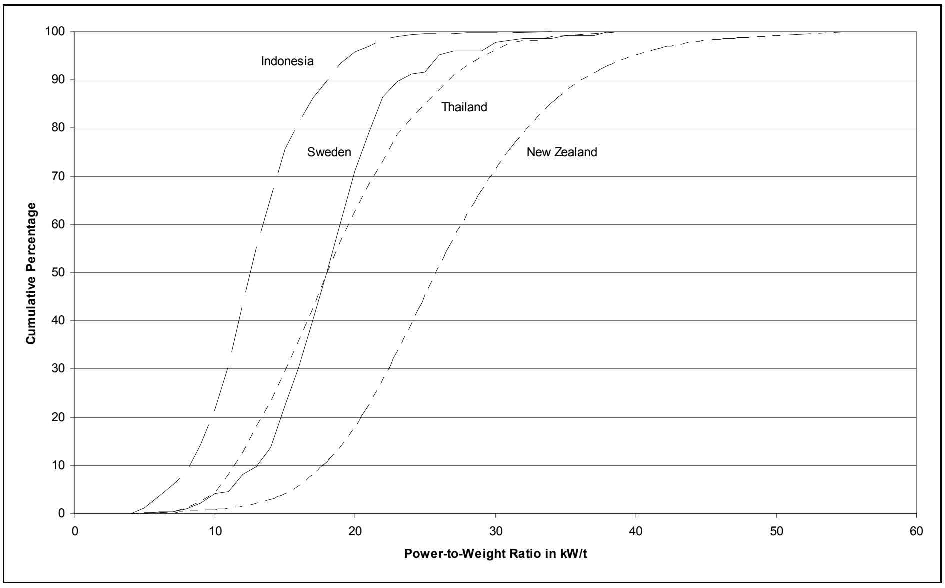


Figure B2.3: Car Power-to-Weight Ratio Distributions From Different Countries

An additional application of the mechanistic modelling is in generating speed-distance diagrams. By numerically integrating Equation B2.4, one can establish the acceleration rate on an instantaneous basis. Using this information in conjunction with an initial speed, one can establish the speed of a vehicle on any point of a grade. An example of this from Bennett (1994) is given in Figure B2.4.

MULTIVARIATE GRADIENT SPEED MODELS

Multivariate analyses consist of observing speeds on gradients and then fitting regression models with one or more independent variables to the model. These are simpler to apply than mechanistic models, but are not as flexible nor transferable. In the context of HDM studies, the most common examples are those from the Kenya, Caribbean and Indian road user cost studies which were included in HDM-III.

Table B2.6 compares the regression coefficients from these studies for different vehicles (Hide, *et al.*, 1975; Morosiuk and Abaynayaka, 1982; CRRI, 1982). These values give the change in speed in km/h as a function of the rise and fall.

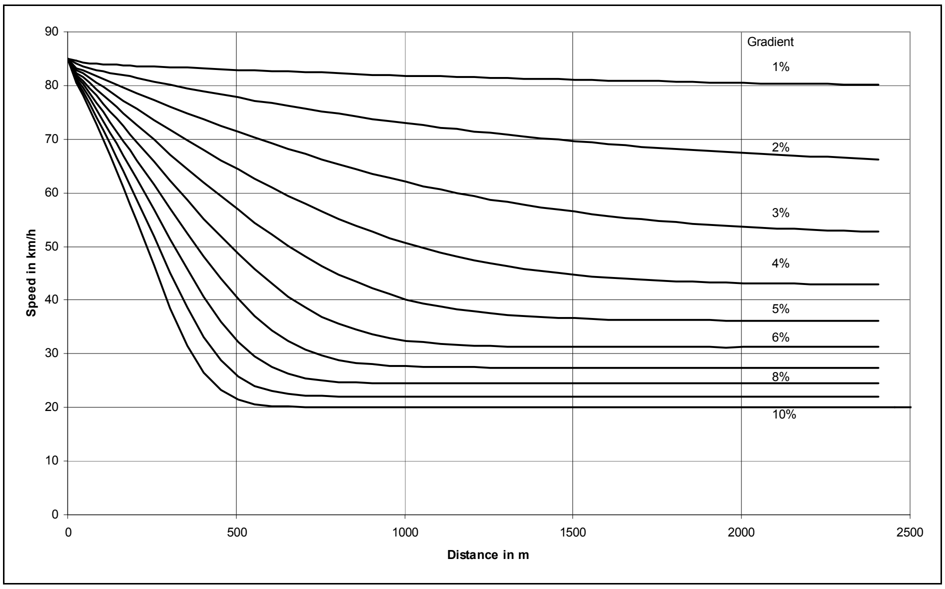


Figure B2.4: Speed-Distance Profile for Heavy Truck Towing

Table B2.6: User Cost Study Gradient Effect Coefficients[1/]

Gradient	Study	Gradient Coefficient by Vehicle Class[2/]			
		Passenger Car	Light Commercial Vehicle	Medium and Heavy Commercial Vehicle	Heavy Bus
Rise	Kenya	-0.372	-0.418	-0.519	-0.526
	Caribbean	-0.078	-0.085	-0.222	-
	India	-0.178	-	-0.175	-0.277
Fall	Kenya	-0.076	-0.050	0.030	0.067
	Caribbean	-0.067	-0.067	-0.122	-
	India	-0.155	-	-0.073	-0.159

Notes: 1/ Rise and fall expressed in m/km (10 per cent gradient = 100 m/km)
2/ km/h per rise and fall

With regard to upgrades (rise), the values in Table B2.6 show little consistency between countries. This suggests quite substantial differences in the responses to gradient between the three studies which most likely reflects different vehicle characteristics—specifically the power-to-weight ratios. The downgrade (fall) coefficients are more consistent, however, there are still wide variations in the magnitude for the same vehicle between the three studies. Since downgrades are more affected by driver behaviour than upgrades, where vehicle characteristics govern, this is indicative of variations in driver response to downgrades between countries.

The Brazil User Cost Study (GEIPOT, 1982) developed equations for predicting the effects of vertical gradient on speeds for six vehicle classes. These were not used in HDM-III, instead being replaced by a mechanistic model. The basic steady state speed equations were of the form:

$$S = a1 + a2\ GR + a3\ IRI \qquad \text{…(B2.7)}$$

where a1 to a3 are regression coefficients
IRI is the roughness in IRI m/km

B2.3.2 Downgrades

The speeds of vehicles on downgrades has received much less attention in the literature than on upgrades. While speeds on upgrades are governed by the power-to-weight ratio and the magnitude of the forces opposing motion, downgrade speeds are influenced by a number of factors such as the length and gradient of the slope, the following alignment, driver behaviour, as well as the vehicle characteristics.

The force balance equation presented earlier can also be used to estimate the braking requirements on downgrades. In order to stop a vehicle it is necessary to overcome the acceleration due to gravity. The parasitic drag (in the form of the rolling and aerodynamic resistances) contributes towards this which leads to the following equation for predicting the braking requirements:

$$Fbr = Fg - Fr - Fa \qquad \text{...(B2.8)}$$

where Fbr is the braking force required in N

Figure B2.5 illustrates the braking power required as a function of speed and gradient for a rigid heavy truck (Bennett, 1994). From this figure it can be observed that on level roads the braking force required is negative. This indicates that the rolling and aerodynamic resistances are (eventually) sufficient to eventually stop the vehicle. Similarly, when operating on a -2 per cent gradient above 65 km/h these resistances are sufficient to retard the speed of the vehicle. However, once the speed is below 65 km/h it is necessary to supply additional braking power to stop the vehicle.

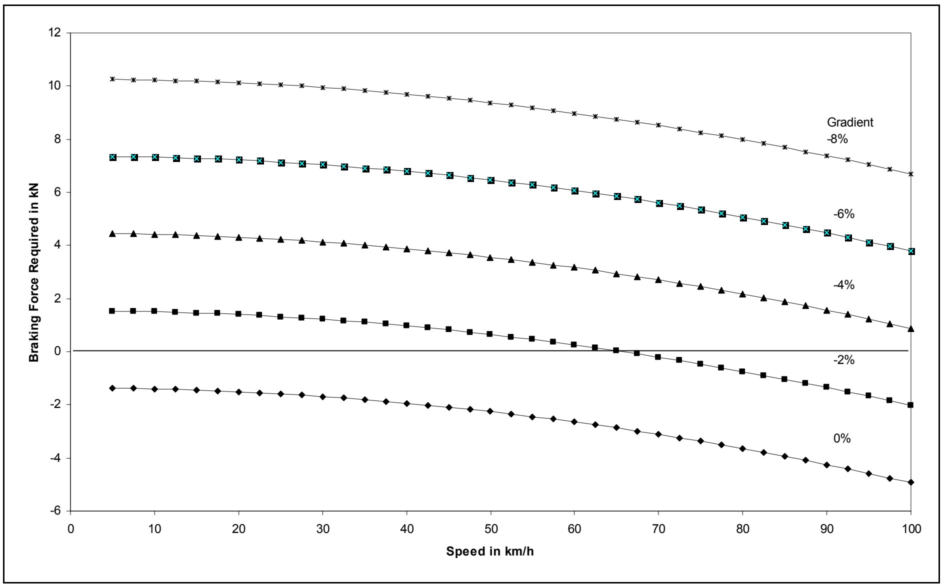


Figure B2.5: Forces Acting onTruck on Downgrade as a Function of Speed

Watanatada, *et al.* (1987a) used the force balance equation to quantify the maximum speed drivers will travel on downgrades. It was considered that braking would only be important on downgrades above five per cent. Ignoring the aerodynamic drag because it was "insignificant" they expressed the limiting speed as a function of the braking power.

The assumption of the limiting braking speed not applying until five per cent is probably based on the findings by several researchers that speeds are unaffected by gradients until they reach a certain level. For example, ITE (1976) indicate that average speeds on downgrades are increased over tangent speeds on gradients up to five per cent for trucks and up to three per cent for buses and passenger cars. When the gradients exceed these levels the speeds decrease over tangent speeds.

However, NAASRA (1980) considered that even on very steep downgrades, truck speeds were equal to their free speeds on level sections. On more winding sections, downgrade speeds were equal to the upgrade speeds. For cars "uphill and downhill speeds seem to differ by about 10 km/h to 15 km/h spaced about evenly on either side of the flat gradient value." The Australian computer simulation model TRARR (Hoban, *et al.*, 1985) used this approach in that vehicles are assumed to travel at their desired speed on downgrades.

Tom and Elcock (1988) studied truck speeds on eleven long downgrades in California ranging from 3.5 to 7.1 per cent gradients. The results showed that speeds were influenced by the length of gradient, rate of gradient, horizontal curvature and other factors. Although the authors did not use the results of this study to develop any definite relationships between truck speeds and downgrades, some conclusions can be drawn from this work. Truck speeds were equal to level section speeds when the gradients were below four per cent, reducing when the gradients were above this level. On long and steep downgrades the speeds were found to reduce to a constant speed and stay at this speed.

St. John and Kobett (1978) postulated that the downgrade speed was proportional to the truck brake capability (engine plus wheel) and the gross vehicle weight. It was noted that speeds were not usually influenced unless the gradient was over 1.6 km long and greater than four per cent. Crawl speeds began to rise when trucks were within 600 - 900 m of the bottom of the gradient (St. John and Kobett, 1978). The authors do not give details on the rate of increase but state that it was incorporated into the MRI simulation model.

Bennett (1994) undertook a mechanistic analysis of downgrade speeds. It was found that there were three situations with regard to power usage for vehicles on downgrades:

- No additional was provided and the only increase was due to gravity ($Pd = 0$);
- Additional power was provided so the vehicle accelerated faster than gravity ($Pd > 0$); or,
- The vehicle brakes were used to reduce the speed ($Pbr > 0$).

It was noted that the HDM-III model, as did most other researchers, only considered the third situation where there was positive braking power. The analysis indicated that the power usage was a function of gradient on downgrades and moderate upgrades and a model was developed which predicted the power as a function of gradient. However, the application of the model was complicated due to the analytical requirements to solve for speed. In the end it was concluded that it was appropriate to assume a mean limiting speed on downgrades which was based on the speed on flat sections. This was consistent with the earlier work in Australia (*eg* NAASRA, 1980 and Hoban, *et al.*, 1985) and represented the maximum possible speed of vehicles on downgrades.

B2.3.3 Horizontal Curvature

INTRODUCTION

Horizontal curves have long been recognised as having a significant effect on vehicle speeds. They have therefore been afforded a great deal of attention by researchers. The discussion of the forces acting on a vehicle traversing a curve in Chapter B1 resulted in the following force-balance equation:

$$M \frac{v^2}{R} = M\,g\,e + M\,g\,f \qquad \ldots(B2.9)$$

This can be rewritten as:

$$v = \sqrt{(e + f)\, R\, g} \qquad \ldots(B2.10)$$

where v is the vehicle velocity in m/s
R is the radius of curvature in m
e is the superelevation in m/m
f is the side friction factor

The above equation represents the maximum speed at which a vehicle can traverse a curve. Since this maximum speed is dependant upon the radius of curvature, the superelevation and the side friction factor, these have been the factors investigated in the literature.

RADIUS OF CURVATURE

McLean (1974b) and Good (1978) give an overview of the early research into the effects of the radius of curvature on speeds. McLean (1974b) reanalysed some of the early data to investigate other model formulations using multivariate techniques which were unavailable at the time of the original work.

The early research into curve effects by Taragin (1954), not surprisingly, found that the radius of curvature had a significant effect on speeds. This was also subsequently found by other researches, *eg* Emmerson (1970) and McLean (1974b). Different forms of mathematical expressions were used to predict curve effects. Figure B2.6 shows the predictions of these early studies along with more recent data from N.Z. (Bennett, 1994) and Thailand (HTC, 1999a).

The curves in this figure suggest that above a 200-300 m radius, curvature has little impact on speeds. This has been supported by a number of researchers in different countries. For example, Gambard and Louah (1986) found in France that it was only below 200 m radius that curvature had an effect on speeds.

Taragin (1954) developed different curves for various percentile speeds. These curves indicated that the higher percentile speeds were more affected by the curve radius than the slower drivers.

A comprehensive study of the effect of curvature on speeds was undertaken in Australia during the 1970s (McLean and Chin-Lenn, 1977; McLean 1978a, 1978b, 1978c, 1978d, 1978e). Data were collected on 72 curves constituting a total of 120 sites. A multivariate analysis was performed on the data. As with the other researchers, it was found that the radius of curvature was a significant factor influencing speeds. There were small, statistically significant effects due to sight distance, shoulder width and superelevation (McLean, 1978c). A preliminary model was developed which predicted the 85th percentile speed as a function of the sight distance and curvature. However, since the sight distance only contributed to a small portion of the total sum of squares, it was subsequently eliminated.

McLean (1979) subsequently grouped the data by desired speed and produced a series of regression curves for each data set. These led to the development of the design chart in Figure B2.7 which is the current Australian and N.Z. standard. Bennett (1994) found that these curves were appropriate for N.Z. with his more recent field data suggesting that some extrapolation was possible with lower curve radii acceptable in higher speed environments. However, it was noted that this would not be good design practice so they advised against any changes.

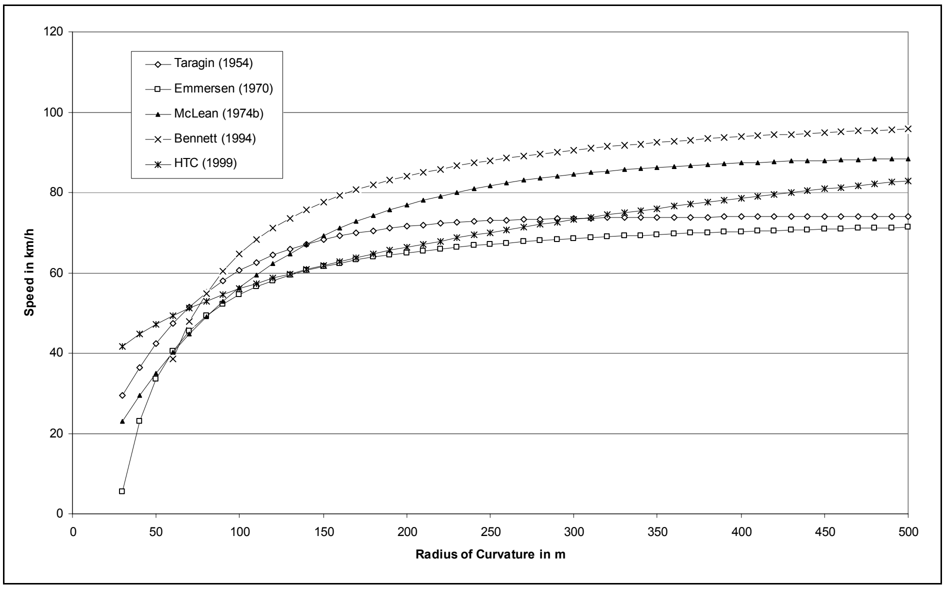


Figure B2.6: Research into Effects of Horizontal Curvature on Car Speeds

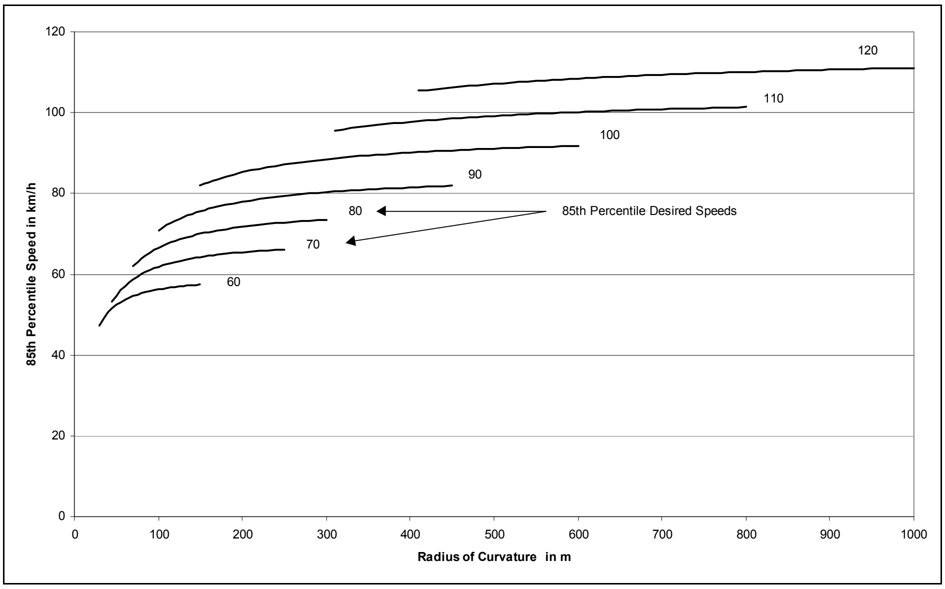


Figure B2.7: Family of Australian Horizontal Curve Speed Relationships

The findings that there are significant differences in curve speed as a function of the desired speed of travel has also been found by other researchers in the U.K. (Kerman, *et al.*, 1982), Sweden (Brodin and Carlsson, 1986), N.Z. (Bennett, 1994), and Thailand (HTC, 1999a).

The research described above illustrates a very important feature: namely, that the response of a driver to a curve is not only a function of the radius of curvature, but also of the approach speed to the curve, a measure of the desired speed of travel. Vehicles which are travelling faster will be more influenced by curvature than slower vehicles, with the magnitude of the influence increasing with increasing desired speed. Thus, equations which only consider the radius of curvature and not the desired speed of travel will not fully reflect the effects of curvature on speed.

SUPERELEVATION AND SIDE FRICTION

Because of the force balance equation for a vehicle operating on a curve it would be anticipated that superelevation would have an influence on the speed of a vehicle through the curve. However, this has not been the findings of most researchers.

Taragin (1954) found that superelevation had no effect on speeds. McLean (1976) found that it had no effect on high standard curves and this was confirmed in the subsequent regression analyses covering all standards of curves (McLean, 1978c). Bennett (1994) and HTC (1999a) in developing curve-speed functions found that in neither N.Z. nor Thailand did superelevation have a statistically significant impact on speed.

As described in McLean (1978d), there are two interpretations of the side friction factor: it is regarded as a design criterion or as an explanation of driver behaviour. The former ensures that drivers will not experience discomfort from high side friction values while the latter assumes that drivers adjust their speed according to the comfort criterion represented by a side friction-speed relationship.

The Australian curve speed research (McLean, 1978d) found that curves based on design speeds below about 90 km/h were consistently driven at speeds in excess of the design speed which indicated that drivers were willing to accept higher values of side friction on lower standard curves. Bennett (1994) found the same results in N.Z. Nicholsen and Wong (1992) concluded that it was “unlikely that drivers can accurately estimate side friction” based on their analysis of curve-speed data.

On the basis of these studies it can be concluded that under most conditions the side friction is an outcome of the speed selected as opposed to a factor influencing it.

B2.3.4 Pavement Roughness

Pavement roughness is the variation in the longitudinal profile of the surface. One of the earliest studies into roughness was by Karan, *et al.* (1976). Data were recorded on roughness, volume to capacity (v/c) ratios and spot speeds (using radar) at 72 locations. This paper is important in that it gives a good example of the problems associated with trying to quantify roughness effects through statistical modelling.

Firstly, the authors had very small sample sizes: 32 per cent of the sites had less than 60 vehicles and five per cent less than 18 vehicles. These small sample sizes will result in large standard errors. Unfortunately, the smallest samples corresponded to those pavements with the highest roughnesses thus limiting the accuracy of the resulting models. Secondly, and more importantly, no allowances were made for the effects of environment on speeds. Since speeds will naturally vary between locations, it is necessary to correct for this in the modelling.

Karan, *et al.* (1976) found a significant effect of roughness on speeds and presented four models for predicting these effects. However, the coefficients in their models are inconsistent and raise doubts as to the accuracy of the approach. In two models there is a 20 per cent difference in the roughness effects between 80 and 96 km/hr speed limit areas while in another model the roughness effects are the same.

du Plessis, *et al.* (1989) investigated roughness effects on speed with the objective of calibrating the HDM-III speed prediction model. Data were collected on 22 straight, level sections on two-lane and dual carriageway roads. The sites were strongly biased towards smooth pavements, with 64 per cent having a roughness below 2.3 IRI. Only five pavements had roughnesses above 4.6 IRI, four of which were unsealed roads. A series of regression models were developed from the data. However, when the models were tested for autocorrelation, it was found that for all vehicles except heavy trucks, road type had a greater impact on speed than roughness. Thus, all the regression models were rejected as invalid.

The various road user cost studies have developed equations which predict the effect of roughness on speeds. Table B2.7 shows the results from several of these studies for passenger cars on paved roads.

Table B2.7: Effect of Roughness on Passenger Car Speeds from User Cost Studies

Country	Reference	Decrease in Speed (km/h) per Increase in IRI m/km
Brazil	GEIPOT (1982)	2.00
Caribbean	Morosiuk and Abaynayaka (1982)	0.62
India	CRRI (1982)	2.57
Kenya	Watanatada (1981)	0.64

The Caribbean and India studies obtained roughness effects through a multivariate analysis where roughness was one term in the model. The Kenya study (Hide, *et al.*, 1975) did not quantify a statistically significant roughness effect but Watanatada (1981) extrapolated the Kenya unsealed road results to sealed roads.

The measurements of response type roughness meters constitute the vertical displacement of the chassis relative to the axle over a section of road. This is termed the average rectified slope (ARS) and is usually expressed in units of m/km or mm/m. Most studies, such as the user cost studies in Table B2.7, directly related speed effects to the ARS. However, Paterson and Watanatada (1985) found that the average rectified velocity was a better statistic to use since it "best represents the level of excitation in a moving vehicle".

For the Brazil Study data, the ARS was expressed as 1.15 IRI (Watanatada, *et al.*, 1987a). By establishing the maximum acceptable level of excitation of a vehicle, one can predict the speed as a function of this maximum level and the ARS.

Watanatada, *et al.* (1987b) developed the following equation for predicting the average rectified velocity at any speed:

$$\mathrm{ARV(v)=a2\ IRI\ v}\left[\frac{\mathrm{v}}{22.2}\right]^{(\mathrm{a0\ +\ a1\ ln(a2\ RI)})} \qquad \ldots(\mathrm{B2.11})$$

Table B2.8 gives the values of various coefficients as reported by Watanatada, *et al.* (1987b).

Table B2.8: Regression Coefficients for Predicting ARV

Surface Type	a0	a1	a2
Asphaltic concrete	0	0	1.15
Surface treated or gravel	1.31	-0.291	1.15
Earth or clay	2.27	-0.529	1.15

Source: Watanatada, et al. (1987b)

For the purposes predicting the speed, we are interested in the maximum average rectified velocity (ARVMAX) since this will give the maximum speed that a vehicle will travel at a given roughness level. This results in the following equation:

$$v = \exp \frac{\ln\left(\frac{ARVMAX}{a2\ IRI}\right) + \left(a0 + a1 \ln(a2\ IRI)\right)\ln(22.2)}{1 + a0 + a1 \ln(a2\ IRI)} \quad \text{…(B2.12)}$$

where ARVMAX is the maximum average rectified velocity in mm/s

For asphaltic concrete surfaces (where a0 and a1 =0) Equation B2.12 simplifies to:

$$v = \frac{ARVMAX}{a2\ IRI} \quad \text{…(B2.13)}$$

In HDM-III Equation B2.13 was used for all surface types. Using data from the Brazil study, Watanatada, *et al.*, (1987a) calculated the values for ARVMAX given in Table B2.9 for the HDM-III model. The values decrease as one moves from the softer suspensions of passenger cars to the firm suspensions of heavy commercial vehicles. This indicates that lighter vehicles will have higher speeds on rough roads than heavy vehicles.

Table B2.9: Maximum Average Rectified Velocity by Vehicle Class

Vehicle Class	Maximum Average Rectified Velocity (mm/s)	
	Brazil[1/]	Australia[2/]
Passenger Cars	259.7	203
Light Commercial Vehicles	239.7	200
Heavy Buses	212.8	-
Medium Commercial Vehicles	194.0	200
Heavy Commercial Vehicles	177.7	180
Articulated Trucks	130.9	160

Source: 1/ *Watanatada, et al. (1987a)*
2/ *McLean (1991)*

In Australia, McLean (1991) derived roughness effects values for Equation B2.13 indirectly from a user survey. On rougher pavement sections, users were asked to state the additional distance they would be prepared to travel to make the journey on a smooth road. For the roughest section in the survey (IRI = 6.7), the difference in journey distances were converted into different travel speeds for the same journey time. The parameter ARVMAX was adjusted to match the predicted steady state speed to the travel speed calculated for IRI = 6.7. The resulting value for ARVMAX was reasonably close to that in HDM-III. For trucks, the values adopted were based on those in HDM-III. Table B2.9 lists the Australian values for ARVMAX recommended by McLean (1991).

Elkins and Semrau (1988) presented equivalent models to Equation B2.13 for cars and trucks in the U.S.A. based on data from Brazil and the current U.S.A. analytical process. The predictions of these equations indicated a marked decrease in speeds on very smooth pavements. However, the authors note that their equations “are primarily based on engineering judgement”. In contrast, Cox (1996b) suggested that roughness did not impact on speeds for smooth roads, citing an Australian study which found it was only above 5 IRI m/km that speeds were affected by roughness.

While the multivariate analyses generally found statistically significant roughness effects, and the HDM-III model developed a limiting roughness-speed model, there are always problems ensuring that the effects are being correctly predicted. The approach used in developing all these models relied on statistical estimation to differentiate between the factors simultaneously influencing speeds. Given the multiplicity of factors influencing speeds it is not always certain that the roughness effects have been accurately isolated.

The work of Karan, *et al.* (1976) and du Plessis, *et al.* (1989) show there are major problems with attempting to investigate roughness effects by pooling data from different sites. The roughness effects can be expected to be small, particularly on roads with low levels of roughnesses such as are usually found in developed countries. When the other factors influencing speeds, particularly road environment, are taken into account, these effects are often lost.

NDLI (1997) circumvented this problem by conducting a before-and-after study of sites due for overlays in India, an approach first suggested by Cooper *et al.* (1980) in the U.K. The speeds were measured before the overlay and again after the overlay. Since the only feature altered was the roughness, any changes in speeds could be attributed to the roughness reduction. Figure B2.8 shows the before and after data from this study[1]. The speed changes were in the range of 0.4 km/h/IRI (buses) to 2.3 km/h/IRI (new technology passenger cars).

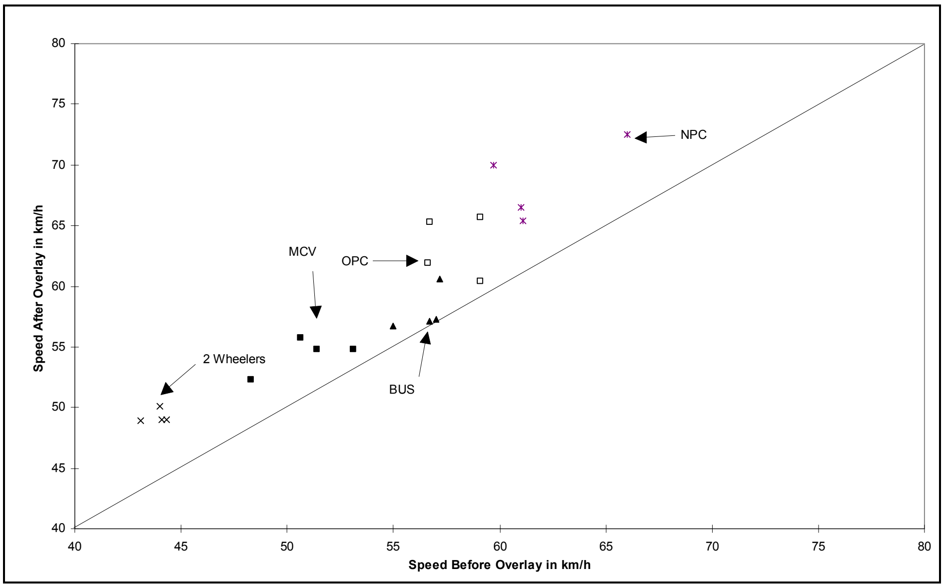


Figure B2.8: Effect of Roughness Reductions on Speed

B2.3.5 Sight Distance

In assessing the effects of sight distance on speeds there are two factors to consider: the available sight distance (SDA) and the safe stopping sight distance (SDS). As a vehicle travels along a road at a given speed, the SDA varies as a function of the terrain. The SDS is defined as:

$$SDS = PRD + BD \qquad \text{…(B2.14)}$$

where PRD is the perception/reaction distance in m
BD is the braking distance in m

The PRD is a function of the perception/reaction time (PRT) and the vehicle speed. Kosashi, *et al.* (1987) indicate that PRT values are in the range of 0.5 to 1.7 s. Noting that speeds exceeded the safe stopping distance on curves, McLean (1981) suggested “As fast driving on a constrained environment implies a high level of alertness, an alternative ‘absolute minimum’ design standard based on a 1.5 s reaction time for the 85th percentile speed has

1 The abbreviations in this figure are: 2 wheeler = Motorcycle; NPC = New technology passenger car; OPC = Old technology passenger car; HCV = Heavy truck; BUS = Heavy bus.

been suggested for constrained alignments”. The braking distance is a function of the vehicle speed and pavement friction.

It would be expected that SDA would have an effect on vehicle speeds inasmuch as the speed adopted by the driver would always be such that SDS < SDA. However, the research to date does not support this hypothesis. For example McLean (1976) stated that “for sites where available sight distance was below [the SDS], operating speeds did not drop accordingly”.

Los (1974) investigated the effects of sight distance on speeds in Australia and found that they did not have a statistically significant effect on speeds. This was also found by Yagar (1984) in Canada. Lefeve (1953) measured speeds on vertical curves with sight distance restrictions but found no consistent relationship between crest speeds and sight distance. McLean (1989) suggested that the small decrements in speed measured could have been due to drivers responding to the pneumatic tubes used for data collection. Wahlgren (1967) found a significant sight distance effect but commented “drivers do not react to local variations in sight distance very sharply but maintain a speed in keeping with the general standard of the road”. Bennett (1994) found that neither the approach nor curve site distances were correlated with the speed.

Some studies which have employed multivariate analyses of speeds have found significant effects. In India (CRRI, 1982) horizontal sight distance was found to affect mean speeds by 1.4 to 3.0 km/h per 100 m. In two studies, the mean speeds were not affected by sight distances but the higher percentile values were. McLean (1978c) found that the 85th percentile speeds increased by about 1.5 km/h per 100 m of sight distance. Taragin (1954) found that the 90th percentile speeds increased by about 8.4 km/h per 100 m of sight distance. Leong (1968) found an increase of 2.4 km/h per 100 m of sight distance for mean car speeds. In reanalysing the same data, Troutbeck (1976) found a smaller impact for sight distance and that it was particularly important with higher percentile speeds. Troutbeck and Crowley (1976) in analysing speed data from a number of states in Australia found a generally insignificant effect of sight distance on speeds. McLean (1976) commented that while sight distance had an effect on speeds it was “not to the extent that it could be regarded as a controlling parameter”.

Polus, *et al.* (1979) conducted a study in Israel designed specifically to address the issue of sight distance on speeds. They found an impact on speeds which increased with the higher percentile speeds. Their findings are partially at odds with McLean (1989) who concluded:

> *“It would appear that in motorised countries sight distance restrictions induce a small reduction in the speed adopted by the faster-travelling drivers, but that they have little, if any, effect on the speeds of other drivers.”*

McLean (1989) went on to indicate that there may be a second order effect on speeds through sight distance restrictions altering the drivers desired speeds. It was proposed that in terrain where there are numerous sight distance restrictions drivers have a lower desired speed than they would normally adopt. This likely explains why some multi-variate analyses found sight distance to have an impact: it was acting as a surrogate for another factor which was not considered in the analysis.

Boyce, *et al.* (1988) give a plausible explanation for the failure of many researchers to find a significant effect of sight distance on speeds:

> *“... the possibility of curtailed sight distances concealing a hazard is perceived as remote, so drivers do not generally adjust their speed to a level commensurate with sight distance restrictions.”*

B2.3.6 Road Width

As with sight distances, there is no consensus in the literature as to the effects of road width on vehicle speeds.

There have been a number of multivariate analyses which have included road width. Van Aerde and Yagar (1981) proposed that road width effects could best be modelled using an exponential function. At low road widths the speeds would be markedly reduced from the desired speed, with the effects decreasing with increasing road width until at a sufficiently large road width there was no influence. However, they point out a number of practical problems which would be encountered trying to calibrate such a model, the most important of which is the limited range of road widths observed in most studies. They subsequently adopted a simple linear model based on the assumption that an ideal lane width was 4.0 m and that the coefficient should reduce the speeds for widths below this ideal. For lane widths of 3.3 to 3.8 m it was found that the operating speed decreased by 5.7 km/h per m of width.

Table B2.10 summarises the effects of road width on speed reported in a number of different studies. Where results were available for more than one vehicle, only those for passenger cars are given. The results are only given for sealed pavements.

Table B2.10: Effect of Width on Speed from Various Studies

Country	Reference	Increase in Speed (km/h) per m road width	Type of Speed	Comments
Australia	Leong (1968)	0.98	Spot	5.5 to 7.6 m width
Britain	Ford (1977)	7.07	Journey	
Canada	van Aerde and Yagar (1981)	5.67	Spot	6.6 to 7.6 m width
Caribbean	Morosiuk and Abaynayaka (1982)	8.10	Journey	Below 5.0 m width
Germany	Lamm, *et al.* (1986)	5.00	Both	85th Percentile
Jamaica	Bunce and Tressidder (1967)	4.32	Journey	
Kenya	Abaynayaka, *et al.* (1974)	5.45	Journey	Below 5.0 m width
Kenya	Hide, *et al.* (1975)	7.10	Journey	Below 5.0 m width
South Africa	NITRR (1983)	2.20	Journey	
Sweden	VTI (1990)	0.70	Journey	

The values in Table B2.10 indicate that there is little agreement between the various studies. Whereas some researchers have found that it is only on narrow pavements that width effects become pronounced (*eg* Hide, *et al.*, 1975), others apply a width effect even to high standard roads such as are found in Germany (McLean, 1991) and Sweden (Brodin and Carlsson, 1986). It is also important to note that several researchers were not able to find a significant width effect (*eg* Troutbeck, 1976) and so not only is the magnitude of the effect of width on speed open to interpretation, but even its very existence.

In addressing the conflicting findings with regard to width effects, McLean (1989) states:

> *"The design speed concept, which has been adopted as a basis for road design in most countries since the 1940s, serves to balance and correlate the standards for all geometric features of a roadway. All else being equal, a road with generous cross section can also be expected to have a high standard of alignment."*

McLean (1989) goes on to suggest that width effects will be manifested on narrow pavements through a modification to the driver's desired speeds. As the width of the pavement decreases, the likelihood of being affected by a vehicle travelling in the opposite direction increases. Thus, the driver would reduce their desired speed. This thesis is supported by the findings of Abaynayaka, *et al.* (1974), Hide, *et al.* (1975), and Morosiuk and Abaynayaka (1982) who found that it was only when the pavement width was below 5.0 m that speeds were affected by width. The variations in speed by different cross-sections observed in India (CRRI, 1982) further support this thesis.

B2.4 The HDM Probabilistic Speed Prediction Model

One of the most comprehensive research efforts to date went into the development of the HDM-III speed prediction model (Watanatada, *et al.*, 1987a; Watanatada and Dhareshwar, 1986). This model was adopted, with some modifications, for HDM-4.

The model was developed using spot speed data collected in Brazil. As McLean (1989) notes "the theoretical development and estimation procedures for the model [are] difficult to follow, but the underlying principles appear sound". The basis for the model is that at any one time there are a series of constraints acting on the driver. These constraints consist of the driving power speed, the braking capacity speed, the curve speed, the surface condition speed, and the desired speed. At any time the speed of the vehicle is the minimum of these constraints, *ie*:

$$VSS = \min(VDRIVE, VBRAKE, VCURVE, VROUGH, VDESIR) \quad \ldots(B2.15)$$

where

where	VSS	is the steady state speed in m/s
	VDRIVE	is the limiting driving speed in m/s
	VBRAKE	is the limiting braking speed on negative gradients in m/s
	VCURVE	is the limiting curve speed in m/s
	VROUGH	is the limiting roughness speed in m/s
	VDESIR	is the limiting desired speed in m/s

The model can be interpreted that the driver attempts to maintain their desired speed (VDESIR) subject to the other four limiting (*ie* maximum) speeds. The various limiting speeds were quantified by Watanatada, *et al.* (1987a) using mechanistic and/or behavioural principles.

Figure B2.9 shows the predictions of the HDM-III speed prediction model for a truck operating on a straight gradient with low roughness. As a tangent section, the curvature constraining speed (VCURVE) is not important. Similarly, the roughness is low enough to effectively eliminate the roughness constraining speed (VROUGH). Thus, the only constraints affecting the speeds in this example are the desired speed (VDESIR), the driving constraint (VDRIVE) and the baking constraint (VBRAKE).

In Figure B2.9 there are three zones where different constraints affect the speeds:

- > 0.5 per cent gradient — VDRIVE dominates
- -4.5 to 0.5 per cent gradient — VDESIR dominates
- < -4.5 per cent gradient — VBRAKE dominates

Hoban (1987) gives an good review of the model, particularly the underlying assumptions. He notes that the model is: "conceptually more satisfactory than simple linear additive concepts, since it is more firmly based on actual vehicle and driver behaviour."

The limiting velocity model represented by Equation B2.15 is not unique. For example, the computer simulation model TRARR (Hoban, *et al.*, 1985) used a variation of it as have several other researchers (Watanatada and Dhareshwar, 1986). However, Watanatada, *et al.* (1987a) modified the model to treat each constraining speed as a random variable. In doing so, they developed a probabilistic form of the model which will hereinafter be referred to as the PLVM (probabilistic limiting velocity model). This will differentiate it from the standard model formulation which will be termed the MLVM (minimum limiting velocity model). The equations of these are as follows:

HDM-III PLVM

$$\mathrm{VSS} = \frac{\exp(\sigma^2/2)}{\left[\mathrm{VDRIVE}^{-1/\beta} + \mathrm{VBRAKE}^{-1/\beta} + \mathrm{VCURVE}^{-1/\beta} + \mathrm{VROUGH}^{-1/\beta} + \mathrm{VDESIR}^{-1/\beta}\right]^{\beta}}$$

…(B2.16)

MLVM

$$\mathrm{VSS} = \min(\mathrm{VDRIVE}, \mathrm{VBRAKE}, \mathrm{VCURVE}, \mathrm{VROUGH}, \mathrm{VDESIR})$$ …(B2.17)

where σ is a parameter derived from the Brazil data
β is a parameter derived from the Brazil data

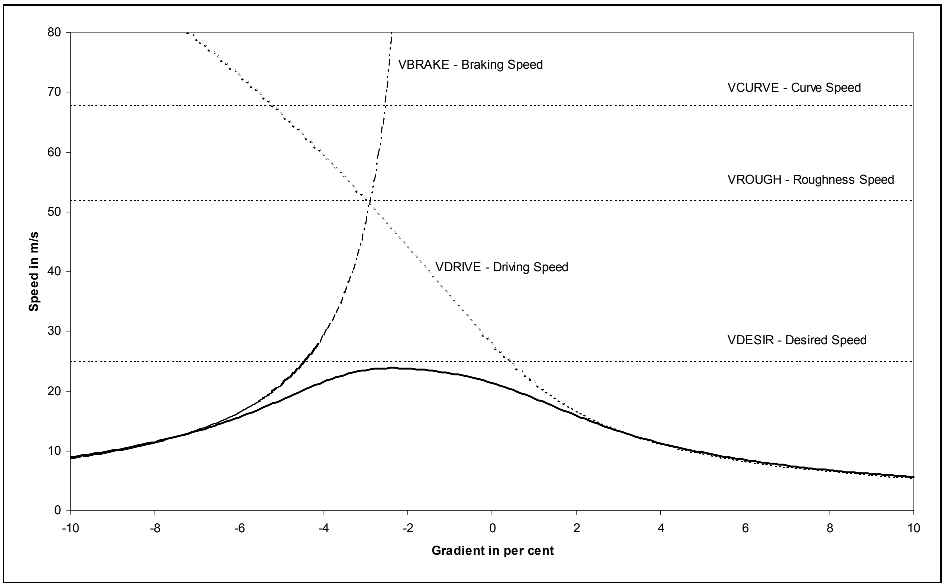


Figure B2.9: Example of HDM-III Speed Model Predictions

The PLVM gives a smooth transition between the constraining speeds whereas the MLVM results in discontinuities. This is illustrated in Figure B2.10 which shows the predictions of the two approaches for the same data in Figure B2.9.

With regard to the difference between the two models, Elkins and Semrau (1988) observed:

> *"When two or more speeds become equally dominant, the [PLVM] speed drops below the [MLVM] speed by a larger amount. Also, as more speeds begin to lower the [PLVM], they do so at a diminishing rate. Thus, the stochastic nature of driver perception is modelled such that as the driver reacts to a greater number of speed constraints, he or she will drive slower than the minimum of the constraining speeds."*

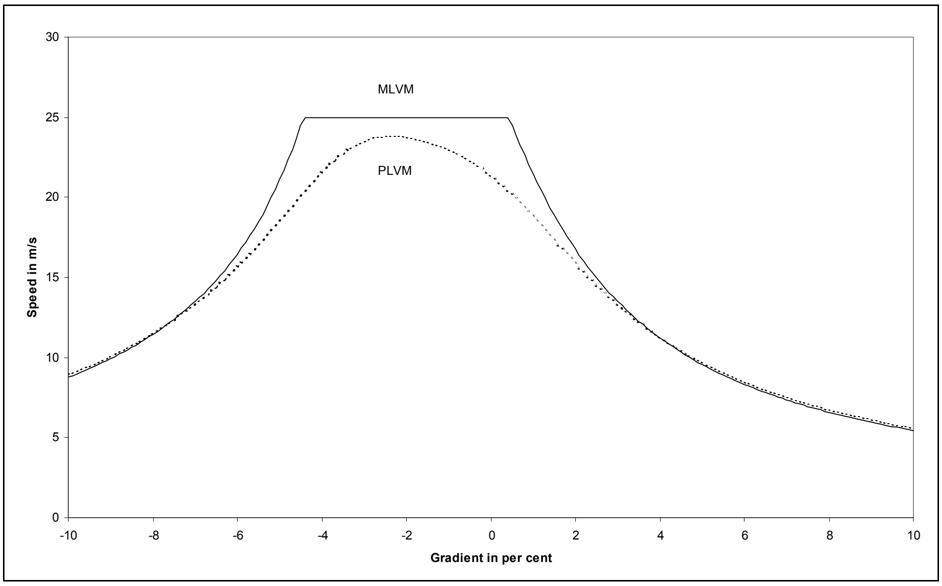


Figure B2.10: Comparison of PLVM and MLVM Predictions

The PLVM is a marked improvement over the other formulations used for predicting speeds. The model formulation not only reflects driver behaviour, but it avoids many of the deficiencies of other models such as those based on multiple linear regression analyses. The first derivatives of these latter models are constant which means that the effects of changing one variable does not change irrespective of the magnitude of the other variables. For example, the regression models predict that reducing the curvature on a mountainous road would have the same impact as reducing it on a flat section. By comparison, the PLVM model would predict that if the speed was being mainly constrained by gradient, reducing the curvature would have little, if any impact.

The model parameter β in the PLVM dictates the degree of interaction between the limiting speeds. McLean (1991) views the parameter β as a measure of behavioural response to combinations of speed affecting road variables while also being an overall measure of interaction. McLean (1991) suggests that there would be different levels of interactions under different operating conditions, *eg* combinations of roughness and curvature over curvature and gradients.

Hoban (1987) highlighted the importance of β and understanding its role and underlying assumptions when applying the model to new situations. Calibrations of the model had yielded values of 0.24 – 0.31 for Brazil vs 0.59 – 0.68 for India, “suggesting a wide variability in constraining speeds”. The consequence of these differences is that the predicted speed may be well below the minimum mean constraining speed. This is illustrated in Figure B2.11 which shows the effects of varying β from 0.01 to 0.6 for the same data presented earlier. At very low values of β the PLVM and MLVM predictions are the same which high values see the predicted speed much lower than the constraining speed.

In calibrating the HDM speed model to Thailand, HTC (1999a) found that there was a strong correlation between β and VDESIR. An attempt was made to estimate both simultaneously, such as was done by Viswanathan (1989), but it was found that the value of “β was ‘driving’ the analysis and the estimated values of desired speed were patently unreasonable.” This had also been noted by Hoban (1987) who considered that “the estimation procedure appears to produce unrealistically high values of desired speed”. HTC (1999a) circumvented this problem by assuming a desired speed and estimating the value of β for that speed.

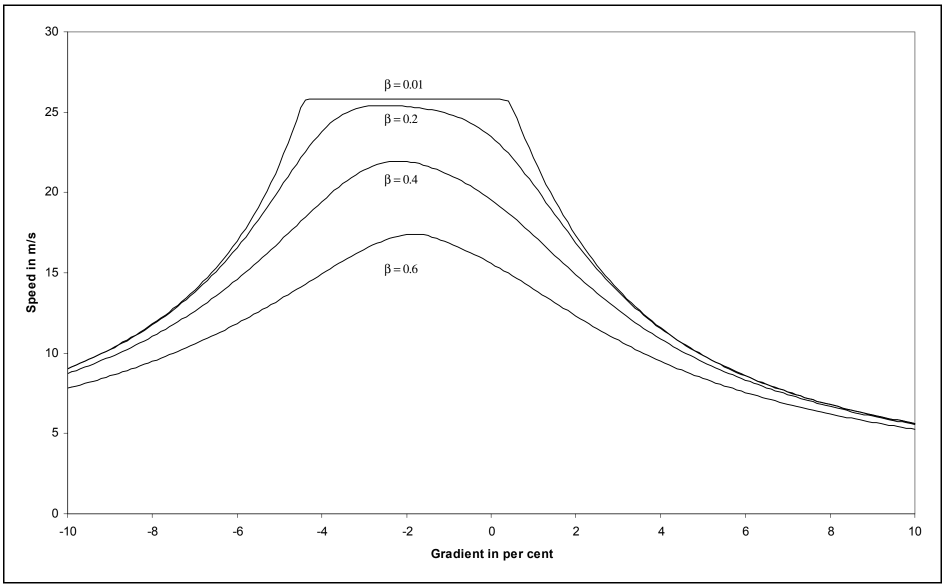


Figure B2.11: Effect of β on Predicted Speeds

HTC (1999a) compared the benefits of the PLVM approach over developing models from multiple-linear regression using the same data. As shown in Figure B2.12 (HTC, 1999a), the PLVM model showed less bias than the regression models, with the predictions falling evenly around the line of equality. The implications of using VDRIVE and VCURVE in a regression model was also investigated and this was found to offer slightly inferior predictions to the PLVM model. It was concluded that the PLVM "model therefore offers advantages over the alternatives, in spite of its complex formulation and the difficulties in estimating the model parameters."

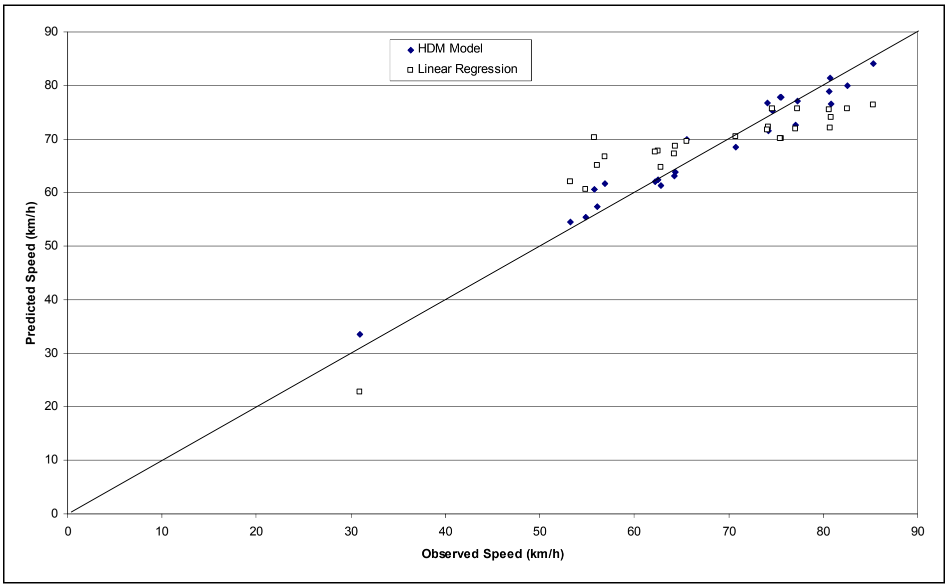


Figure B2.12: Comparison of PLVM and Linear Regression Models

B2.5 HDM-4 Free Speed Model

B2.5.1 Introduction

The HDM-4 free speed model built upon the framework of the HDM-III model. As such, it represented an incremental change instead of a major re-estimation. A preliminary model was proposed by NDLI (1995) but subsequent work found problems with what was proposed. Additional work was done to rectify these issues. The results presented here represent the final version of the HDM-4 free speed model.

B2.5.2 Constraining Speed due to Driving Power – VDRIVE

The constraining speed due to driving power, VDRIVE, is calculated from the mechanistic principles presented earlier. VDRIVE is influenced by the aerodynamic, rolling and gravitational resistances[1]. As shown in Chapter B1, these forces can be calculated as:

$$Fa = 0.5\ \rho\ CD\ CDmult\ AF\ v^2 \qquad \ldots(B2.18)$$

$$Fr = CR2\ FCLIM\ (b11\ Nw + CR1(b12\ M + b13\ v^2)) \qquad \ldots(B2.19)$$

$$Fg = M\ g\ GR \qquad \ldots(B2.20)$$

The maximum used driving power is calculated as:

$$Pd = \frac{v\ (Fa + Fr + Fg)}{1000} \qquad \ldots(B2.21)$$

Substituting for the individual forces results in the following equation:

$$1000\ Pd = z0\ v^3 + z1\ v \qquad \ldots(B2.22)$$

$$z0 = 0.5\ \rho\ CD\ CDmult\ AF + b13\ CR1\ CR2\ FCLIM \qquad \ldots(B2.23)$$

$$z1 = b11CR2\ FCLIM\ Nw + b12\ CR1\ CR2\ FCLIM\ M + M\ g\ GR \qquad \ldots(B2.24)$$

The speed is calculated by solving Equation B2.22 for velocity. This can be done numerically or deterministically using Descartes' rule of signs (see Annex B2.1).

Figure B2.13 shows the effect of gradient on VDRIVE for a passenger car and an articulated truck (NDLI, 1995). The driving power dominates only on positive gradients or on minor negative gradients where power is required to overcome rolling and aerodynamic resistance. The figure shows that the truck is much more influenced by the gradient than the car, due to its much lower power-to-weight ratio. The high speeds on downgrades are quite idealised since they do not take into account the mechanics of the vehicles. The structure of the HDM speed model is such that these effects only serve to eliminate driving power as a constraint on downgrades.

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1 The curve resistances are accounted for in the curve model. Since it is a steady state speed the acceleration is 0 so the inertial resistances are ignored.

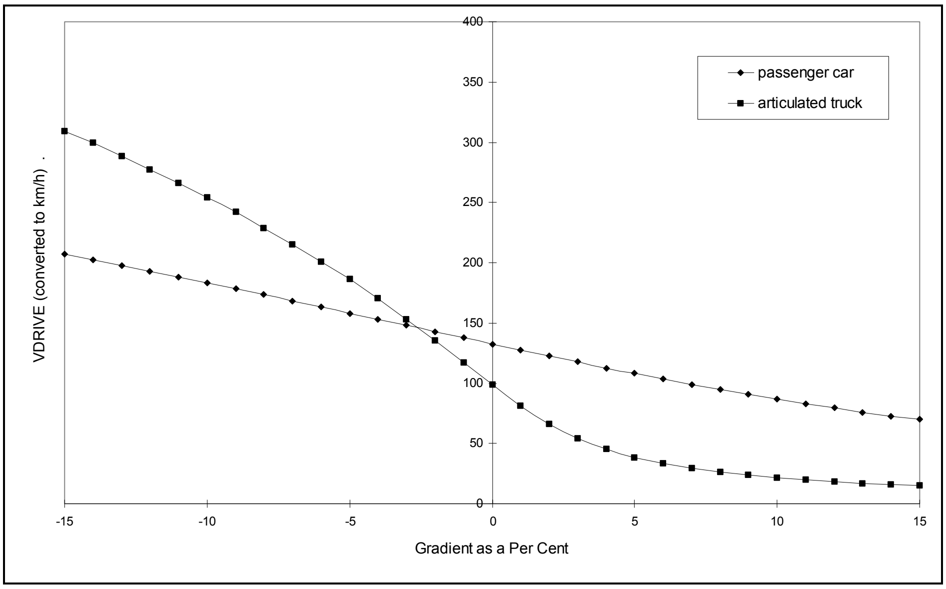

Figure B2.13: VDRIVE vs Gradient for Passenger Car and Heavy Truck

For each of the representative vehicles the driving power was calculated based on data from Bennett (1994) using the vehicle characteristics from Chapter A3. The results are given in Table B2.11.

B2.5.3 Constraining Speed due to Braking Power – VBRAKE

The constraining speed due to braking power—VBRAKE—applies on downgrade sections. It is therefore set to infinity on upgrades.

As described earlier, the modelling of downgrade speeds presents particular problems. In some studies (*eg* Bennett, 1994) light vehicles were found to travel at or near their desired speeds on downgrades. Other studies (*eg* Watanatada, *et al.*, 1987b) suggests that these vehicles can have a significant reduction in their speeds. Heavy vehicles have also been found to have minor speed reductions (Bennett, 1994) or major reductions (Archilla, 1992).

The speeds adopted on downgrades appear to be influenced by both by the magnitude and the length of the downgrade. Myers, *et al.* (1980) presented a method for calculating the maximum grade length for heavy trucks as a function of vehicle weight, velocity and grade magnitude. Myers, *et al.* (1980) indicate that a three km long 10 per cent downgrade has the same effect on truck speeds as a 10 km long five per cent downgrade. This would also explain the differences between the N.Z. speeds in Bennett (1994), which were measured on 1 to 1.5 km long sections, and those from Canada in Archilla (1992) which were measured on sections in excess of 5 km.

NDLI (1995) proposed that the for HDM-4 VBRAKE should be a function of vehicle mass, gradient length and magnitude. They combined the methodology presented by Myers, *et al.* (1980) and that used within HDM-III (Watanatada, *et al.*, 1987b) to develop a model which considered the critical length of a grade a vehicle could travel before brake failure.

Table B2.11: Default HDM-4 Free Speed Model Parameters

Vehicle Number	Vehicle Type	VDRIVE Pd kW	VBRAKE Pb kW	VCURVE a0 m/s	VCURVE a1	VROUGH ARVMAX mm/s	VROUGH a2	VDESIR a0 m/s	VDESIR a1	VDESIR WF	VLIMIT ENFAC	Model Parameters σ	Model Parameters β
1	Motorcycle	12	5	3.9	0.34	203	1.15	27.6	0.0020	1.00	1.10	0	0.151
2	Small Car	26	20	3.9	0.34	203	1.15	27.6	0.0020	0.75	1.10	0	0.151
3	Medium Car	33	20	3.9	0.34	203	1.15	27.6	0.0020	0.75	1.10	0	0.151
4	Large Car	36	20	3.9	0.34	203	1.15	27.6	0.0020	0.75	1.10	0	0.151
5	Light Delivery Vehicle	40	25	3.9	0.34	203	1.15	27.6	0.0020	0.75	1.10	0	0.151
6	Light Goods Vehicle	40	20	3.9	0.34	200	1.15	27.6	0.0020	0.75	1.10	0	0.151
7	Four Wheel Drive	45	25	3.9	0.34	200	1.15	27.6	0.0020	0.75	1.10	0	0.151
8	Light Truck	50	45	4.8	0.29	200	1.15	25.2	0.0028	0.73	1.10	0	0.191
9	Medium Truck	87	70	4.8	0.29	200	1.15	25.2	0.0028	0.73	1.10	0	0.164
10	Heavy Truck	227	255	4.6	0.28	180	1.15	24.1	0.0033	0.73	1.10	0	0.110
11	Articulated Truck	227	255	4.2	0.27	160	1.15	26.3	0.0039	0.73	1.10	0	0.110
12	Mini-Bus	40	26	3.9	0.34	203	1.15	27.6	0.0020	0.75	1.10	0	0.151
13	Light Bus	50	45	4.8	0.29	200	1.15	25.2	0.0028	0.73	1.10	0	0.191
14	Medium Bus	65	70	4.8	0.29	200	1.15	25.2	0.0028	0.78	1.10	0	0.191
15	Heavy Bus	120	120	4.6	0.28	180	1.15	24.1	0.0033	0.78	1.10	0	0.110
16	Coach	180	180	4.6	0.28	180	1.15	24.1	0.0033	0.78	1.10	0	0.110

The data from Archilla (1992) were used to calculate a critical gradient length (CGL) for heavy vehicles. For gradients whose length is less than CGL, VBRAKE has little effect on speeds. Once the gradient length exceeded CGL, VBRAKE was calculated based on mechanistic principles. The formula for calculating CGL was as follows (NDLI, 1995):

$$CGL = a0 \exp(a1\ GR) + a2 \qquad \ldots(B2.25)$$

where CGL is the critical gradient length in km
a0 to a2 are regression coefficients

Although the data within Archilla (1992) related to heavy vehicles, it was assumed that the same critical gradient length applied to all vehicle types, even though for light vehicles the CGL would be expected to be larger than that for heavy vehicles. The recommended default values for the regression coefficients in Equation B2.25 are given in Table B2.12.

Table B2.12: Default Critical Gradient Length Coefficients

Vehicle Type	a0 km	a1	a2 km	R^2adj	S.E
All vehicles	94.874 (15.8)	0.850 (25.9)	2.8 (16.2)	1.00	0.206

Source: NDLI (1995) as adopted from Archilla (1992)
NOTES 1/ 't' statistic given in parenthesis below each coefficient

The coefficient a2 indicates that irrespective of the gradient magnitude, if the slope is less than 2.8 km long, speeds will be unaffected (VBRAKE = □). This supports the results provided in Bennett (1994) which showed no significant reduction in speeds on straight grades up to 1.5 km long for all vehicle types.

When the gradient length exceeded the CGL, VBRAKE could be calculated as per VDRIVE, but with the substitution of -Pb for Pd. Where Pb is the braking power supplied by the engine. This results in the following formula to be solved for VBRAKE:

$$-1000\ Pb = z0\ VBRAKE^3 + z1\ VBRAKE \qquad \ldots(B2.26)$$

This equation could be solved in a similar manner to that given in Annex B2.1 for VDRIVE with the following amendments:

- When DT ≥ 0, there is no solution (VBRAKE = infinity); and,
- When DT < 0, there are two real positive solution, VBRAKE equals the minimum of these.

Although no precise data was available for calibrating the proposed model on downgrades, data presented in Watanatada, *et al*. (1987b), Archilla (1992) and Bennett (1994) were used to subjectively calibrate the model. The recommended default values for Pb were given in Table B2.11.

Bennett (1998a), in testing the HDM-4 beta model, showed that the critical gradient length concept embodied in the NDLI (1995) model gives rise to two quite different sets of downgrade speeds. These are shown in Figure B2.14 and Figure B2.15 for gradients with lengths less than and greater than the critical gradient length. In Figure B2.14 the speeds on downgrades for all vehicles except trucks are continuous. There are slight discontinuities around the -6 per cent gradient where the limiting constraining speeds shifted from the power to the desired speed. The data in Figure B2.15 show the decreases in speed on long gradients.

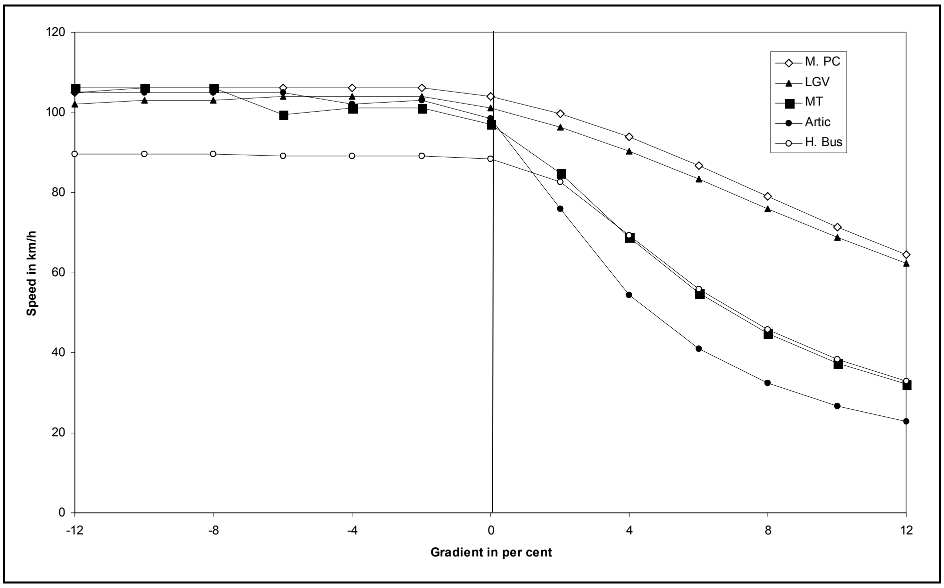

Figure B2.14: Effect of Gradient on Speed – Grades Shorter than Critical Length

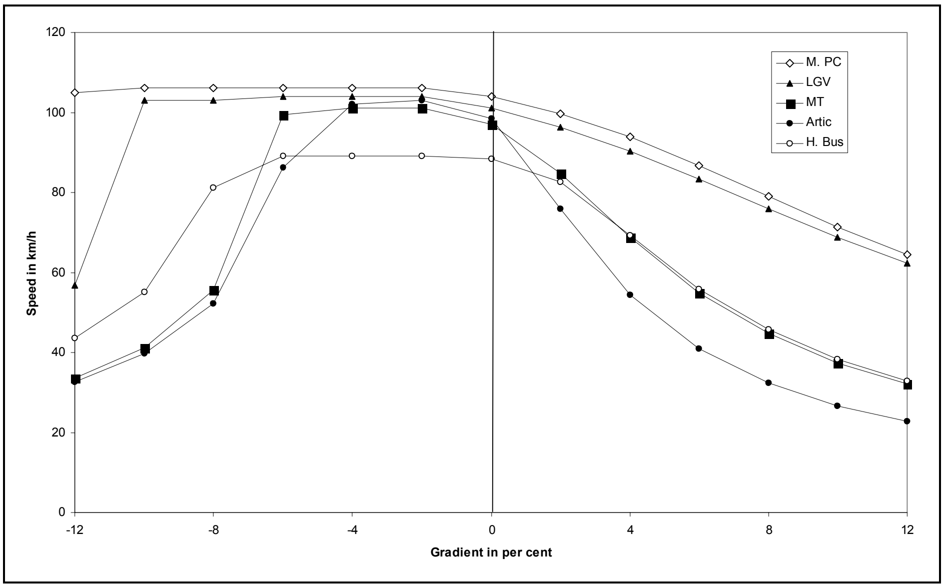

Figure B2.15: Effect of Gradient on Speed – Grades Longer than Critical Length

The sudden speed decreases in Figure B2.15 for the LGV and MT vehicles point to a problem in the model. This arises because of a sudden change in the constraining speeds. Another deficiency which the HDM-4 method recognises is that there are some combinations of vehicle attributes which will give rise to unsolvable solutions. Under these circumstances the vehicles are assumed to travel at their speed on 0 gradient. In the tests the medium and articulated trucks slowed down on steep grades whereas the heavy truck did not slow down. The latter was due to the vehicle attributes corresponding to an unsolvable solution for VBRAKE.

Bennett (1998a) proposed two solutions for the problem. One was to mirror the upgrade speeds with the downgrade speeds. Various studies had found this to be a reasonable approximation. The alternative was to revert to the HDM-III model, even though it was acknowledged that this was less than desirable. The HDM-III model had been rejected for HDM-4 since it was found that while having the appearance of being mechanistic it was, in fact, based on a mathematically convenient simplification of the mechanistic principles. A single parameter – termed the braking power – was used to dictate the shape of the function. This has the advantage of making the model easier to calibrate, although the parameter could not be readily quantified in the field.

At the time of writing HDM-4 v 1.0 has not been corrected and maintains the NDLI (1995) model.

B2.5.4 Constraining Speed due to Curvature – VCURVE

For HDM-4 a change was made to the curve speed model as used in HDM-III. Watanatada, *et al*. (1987b) took the force-balance equation for a vehicle operating on a curve and rewrote it to express the maximum velocity as a function of the radius, superelevation and side friction factor:

$$M \frac{v^2}{R} = M\,g\,e + M\,g\,f \qquad \text{…(B2.27)}$$

Which results in:

$$VCURVE = \sqrt{(e + f)\,R\,g} \qquad \text{…(B2.28)}$$

This approach assumes that the drivers select a level of side friction and then drive at a speed which utilises the desired friction. For this to be valid there would be a correlation between the superelevation and speed. However, as described in Section B2.3.3, most studies have failed to find this, except perhaps for the highest percentile drivers, which indicates that side friction is an outcome of the speed selected as opposed to governing it.

NDLI (1995) therefore proposed a simpler that curve speed model which was based mainly on the radius of curvature. They showed from the literature that the most robust speed models were those which used the approach speed as an independent variable (*eg* McLean, 1989; Bennett, 1994; HTC, 1999a). These have the form:

$$VCURVE = a0 + a1\ Vapp + \frac{a2}{R} \qquad \text{…(B2.29)}$$

where Vapp is the approach speed in m/s

For HDM-4 it was not considered appropriate to have the approach speed as an independent variable. McLean (1991) proposed the following simple model:

$$VCURVE = a0\ R^{a1} \qquad \text{…(B2.30)}$$

NDLI (1995) analysed data from Bennett (1994) with the model in Equation B2.30 and showed that it gave reasonable predictions for N.Z. It was recommended by NDLI (1995) that the results of McLean (1991) be used in HDM-4 instead of the N.Z. data since it was based on a better spread of data.

Table B2.13 gives the parameter values for this model from studies in Australia, N.Z. and Thailand. Figure B2.16 shows the predictions of this model using the coefficients from Australia, N.Z. and Thailand. Table B2.11 gives the default values for use with this model in HDM-4 for each of the default representative vehicles.

Table B2.13: Curve-Speed Model Parameters from Different Studies

Vehicle Class	Australia[1/]		N.Z.[2/]		Thailand[3/]	
	a0	a1	a0	a1	a0	a1
Motorcycle					6.1	0.17
Car + Taxi	3.9	0.34	5.9	0.24	5.3	0.24
Light Truck			6.6	0.21	5.1	0.24
Medium Truck	4.8	0.29	6.2	0.22	4.9	0.22
Heavy Truck	4.6	0.28	6.6	0.21	5.9	0.18
Heavy Truck Towing	4.2	0.27	7.3	0.20	5.4	0.19
Mini-Bus					5.2	0.24
Medium Bus + Coach					6.3	0.18

Source: *1/ McLean (1991)*
2/ NDLI (1995)
3/ HTC (1999a)

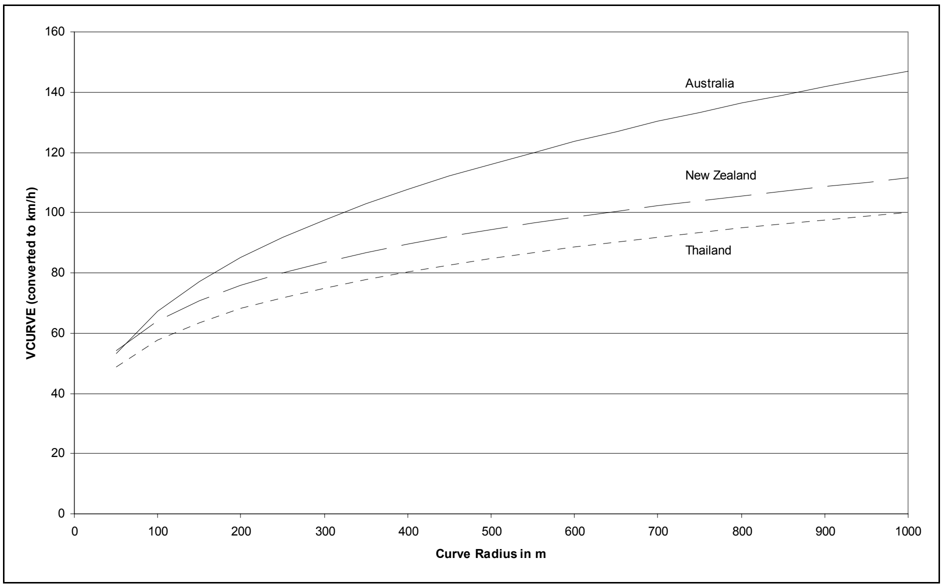

Figure B2.16: Predicted Limited Curve Speed vs Radius for a Passenger Car

B2.5.5 Constraining Speed Due to Roughness – VROUGH

Due to the absence of any improved formulation, for HDM-4 the same equation as in HDM-III was adopted for all pavement types:

$$\text{VROUGH} = \frac{\text{ARVMAX}}{\text{a0 IRI}} \qquad \text{…(B2.31)}$$

The development of this model was discussed in Section B2.3.3.

The default parameter values for a0 were recommended by NDLI (1995) to be those from McLean (1991) since it was considered that they better represented more modern vehicles than those in HDM-III. These parameters were presented earlier in Table B2.9 and are given in Table B2.11 for each of the HDM-4 representative vehicles. Figure B2.17 shows the predictions of the model for several classes of vehicle.

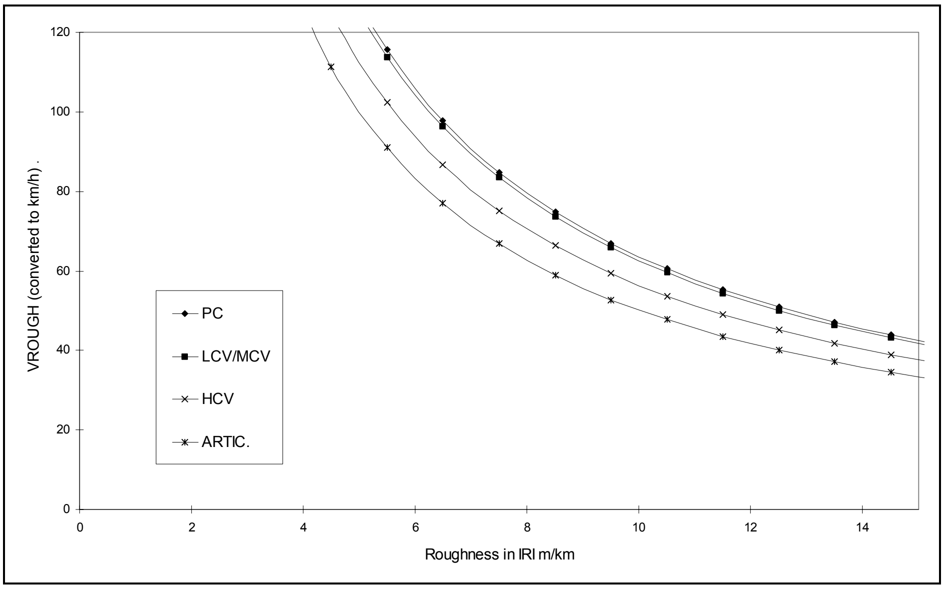

Figure B2.17: VROUGH vs IRI for Different Vehicle Types

B2.5.6 Constraining Desired Speed – VDESIR

INTRODUCTION

The desired speed of travel—VDESIR—accounts for a variety of factors which are not easily separated. Watanatada, *et al*. (1987b) suggest that these include fuel economy, vehicle wear, safety or blanket speed limits.

As discussed in Section B2.4, the HDM desired speed differs from the traditional desired speed due to its correlation with the model parameter β. The estimation procedure means that the value for VDESIR will be in excess of the desired speed measured in the field due to the draw-down effect of β. The interaction of VDESIR and β was found by HTC (1999a) in calibrating the HDM-4 speed model to Thailand:

> *"... it became apparent that β was 'driving' the analysis and the estimated values of desired speed were patently unreasonable. The analysis was therefore conducted by selecting a value for the desired speed and establishing the value of β. If the desired speed was too low, the resulting value for β was not statistically significant."*

The desired speed in HDM-4 is considered to be influenced by:

- A basic desired speed which is a function of road width and the number of lanes;
- The roadside friction;
- Non-motorised traffic; and,
- Speed limits.

NDLI (1995) included speed limits as a constraint in the HDM-4 model formulation. However, it was subsequently found that this resulted in unreasonable predictions from the model due to the interactions of the constraints. Speed limits are therefore considered to be an external constraint which reduce the desired speed in the absence of speed limits, *ie*:

VDESIR0 = VDES XFRI XNMT VDESMUL …(B2.32)

$$VDESIR = \min\left(VDESIR0, \frac{PLIMIT\ \ ENFAC}{3.6}\right) \qquad \ldots(B2.33)$$

where VDESIR0 is the desired speed in m/s in the absence of speed limits
VDES is the desired speed
XFRI is the roadside friction factor
XNMT is the NMT friction factor
VDESMUL is a desired speed multiplier for the road class
PLIMIT is the posted speed limit in km/h
ENFAC is an enforcement factor (default = 1.0)

Equation B2.33 indicates that vehicles will travel at a maximum of their desired speed or at the posted speed limit, whichever is smaller.

NDLI (1995) also included bendiness as a factor influencing desired speeds. This had been found by McLean (1989) and others to have a major influence on the speeds. However, when the model was applied it was found that under extreme conditions it gave unreasonable predictions so it was not included in HDM-4. Instead, it is recommended that users apply lower values of the desired speeds when analysing roads with high levels of bendiness.

PAVEMENT WIDTH

Pavement width is an important factor influencing speeds. The width model for HDM-4 was developed from Hoban, *et al.* (1994) and Yuli (1996) and is illustrated in Figure B2.18. It predicts that there is linear decrease in speed from the desired speed on two-lane highways (CW2) to narrow roads (CW1). At a certain point there is no further reduction in speed due to width. The model also predicts that there is an increase in speed beyond CW2. The slope between CW1 and CW2 is governed by the desired speeds of travel at these widths.

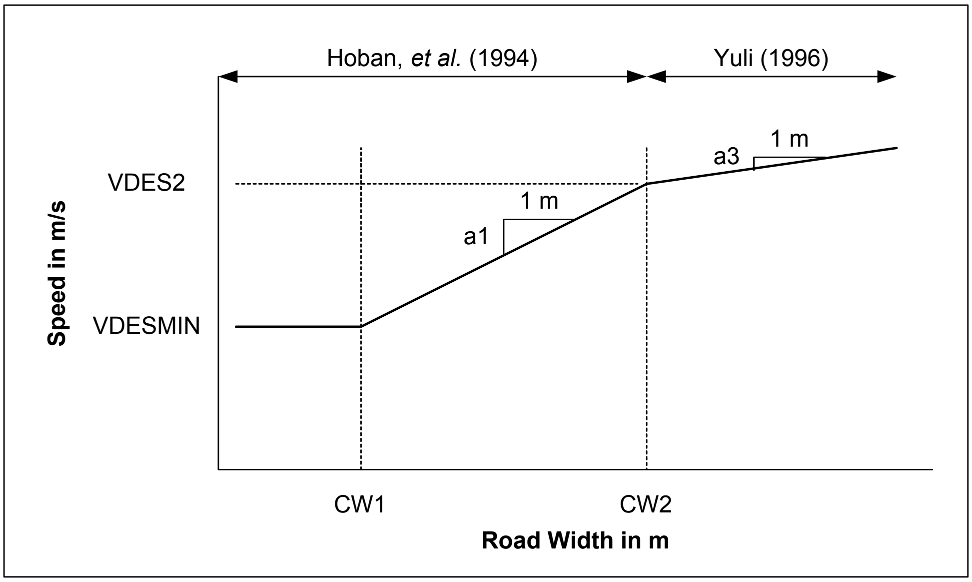

Figure B2.18: HDM-4 Width Model

With reference to Figure B2.18, there are three equations for calculating the width influenced desired speed:

$$VDES = VDESMIN \qquad \text{if } WIDTH \le CW1 \qquad \ldots(B2.34)$$

$$VDES = VDESMIN + a1\ (WIDTH - CW1) \qquad \text{if } CW1 < WIDTH \le CW2 \qquad \ldots(B2.35)$$

$$VDES = VDES2 + a3\ (WIDTH - CW2) \qquad \text{if } WIDTH > CW2 \qquad \ldots(B2.36)$$

where
VDES is the width influenced desired speed in m/s
VDESMIN is the minimum desired speed on single lane roads in m/s
VDES2 is the minimum desires speed on two lane roads in m/s
CW1 is the width below which VDESMIN applies
CW2 is the width where VDES2 applies
a1 is the rate of increase in desired speed for single to two lane road in m/s per m of road width
a3 is the rate of increase in desired speed for two lane roads wider than CW2 in m/s per m of road width

On narrow roads, as usually represented by single lane roads, there is a minimum speed of travel. This applies to all widths below CW1 m. The minimum speed on two-lane roads is assumed to apply at width CW2. Between CW1 and CW2 the speeds are assumed to linearly increase. Speeds will increase further if the road is wider than CW2, but usually at a reduced rate—which may be set to 0.

The slope from single to two lane roads is calculated from the other data as:

$$a1 = \frac{VDES2 - VDESMIN}{CW2 - CW1} \qquad \ldots(B2.37)$$

The minimum speed is expressed as a fraction of the desired speed on a two lane roads as:

$$VDESMIN = a2\ VDES2 \qquad \ldots(B2.38)$$

where a2 is the ratio of desired speeds on single lane to two lane roads

The default values for HDM-4 were established from the literature and are given in Table B2.14. These are applied to all vehicle classes, but the user my modify them if data are available.

Table B2.14: Default HDM-4 Speed-Width Parameters

Parameter	Definition	Value
CW1	Width below which minimum desired speed applies	4.0 m
CW2	Width where two lane desired speed applies	6.8 m
a2	Ratio of minimum to two lane desired speeds	0.75

The values for a3 by vehicle class are:

- Passenger cars and light commercial vehicles: 2.9 m/s/m;
- Trucks: 0.7 m/s/m; and,
- Buses: 0.6 m/s/m.

ROADSIDE FRICTION AND NON-MOTORISED TRANSPORT FACTORS

The factors XFRI and XNMT are multipliers which cater for the effect of roadside friction and non-motorised transport (NMT) on speed[1]. A factor of 0.9 implies that all speeds are reduced by 10 per cent from their undisturbed value. They each have values of 0.6 - 1.0 so if they are both used at the same time there is a potential reduction of 64 per cent in the desired speed.

It must be recognised that these parameters are simplifications of very complex processes. As noted by Hoban, *et al*. (1994) “The effects of pedestrians, bicyclists, animal-drawn vehicles, roadside stalls, bus stops, parking and access points vary considerably in time and space, have different effects on speed and overtaking, and have different effects at low and high traffic flows”.

The underlying assumption with these factors are that they have the same impact across all flow levels. In all likelihood these effects vary as a function of flow—particularly with regard to NMT—but this has not been adequately researched in the literature to reach any firm conclusions in this matter. It has also not been established whether the factors influence only speed, as modelled here, or speed and capacity.

There is very little quantitative information on friction effects. Bang (1995) used data from Indonesia for investigating side-friction factor effects. On the basis of regression analysis, weightings were applied to the relative impact different friction events. This led to the following equation for interurban roads:

$$FRIC = 0.6\ PED + 0.8\ PSV + 1.0\ EEV + 0.4\ SMV \qquad \ldots(B2.39)$$

where PED is the pedestrian flow in ped/h
PSV is the number of vehicle stops and parking manoeuvres in events/h
EEV is the number of vehicles entering and exiting roadside premises in veh/h
SMV is the number of slow-moving vehicles in veh/h

The roads were then grouped into five friction classes, as shown in Table B2.15.

Table B2.15: Indonesian Interurban Roadside Friction Classes

Frequency of Events (both sides) FRIC	Typical Conditions	Side Friction Class
< 50	Rural, agriculture or undeveloped; almost no activities	Very Low
50 – 149	Rural, some roadside buildings and activities	Low
150 – 249	Village, local transport and activities	Medium
250 – 350	Village, some market activities	High
> 350	Almost urban, Market/business activities	Very High

Source: Bang (1995)

Instead of using a single value of XFRI, Bang (1995) gave a table with the speed reduction factors varying as a function of road width and road class. For this study an analysis was made of these data and this resulted in the following models for XFRI. All coefficients were statistically significant at 95% confidence; the standard errors were approximately 0.02; the $R^2 > 0.82$.

---

1 It should be noted that XFRI in HDM-4 differs from XFRI in HDM-Q and HDM-95. The latter included both NMT and roadside effects while in HDM-4 NMT are catered for by XNMT.

$$XFRI = 0.9615 + 0.0270\ WIDTH - 0.0297\ SFCL \quad \text{Four-Lane Divided} \qquad \ldots(B2.40)$$

$$XFRI = 0.9598 + 0.0290\ WIDTH - 0.0317\ SFCL \quad \text{Four-Lane Undivided} \qquad \ldots(B2.41)$$

$$XFRI = 0.9555 + 0.0350\ WIDTH - 0.0403\ SFCL \quad \text{Two-Lane Undivided} \qquad \ldots(B2.42)$$

where WIDTH is the pavement width in m
SFCL is the side friction class where 0 = Very Low; 1 = Low, 2 = Medium, 3 = High, 4 = Very High. See Bang (1995) for photographs showing the types of roads falling into each class.

As would be expected, the coefficients in the above equations indicate that increasing road width has a greater impact on two-lane undivided roads than four lane roads and that two-lane roads are also more sensitive to roadside friction.

In Thailand HTC (1999a) ranked levels of side friction from 0 to 5, where 0 was no side friction and 5 was the worst possible side friction. As shown in Figure B2.19, there was a relationship between free speed and the friction factor. This led to the following relationship for predicting the side friction factor:

$$XFRI = 1 - 10.2\ \frac{SFF}{SFI} \qquad \ldots(B2.43)$$

where SFF was a roadside friction factor (0 – 5)
SFI was the free speed under ideal conditions

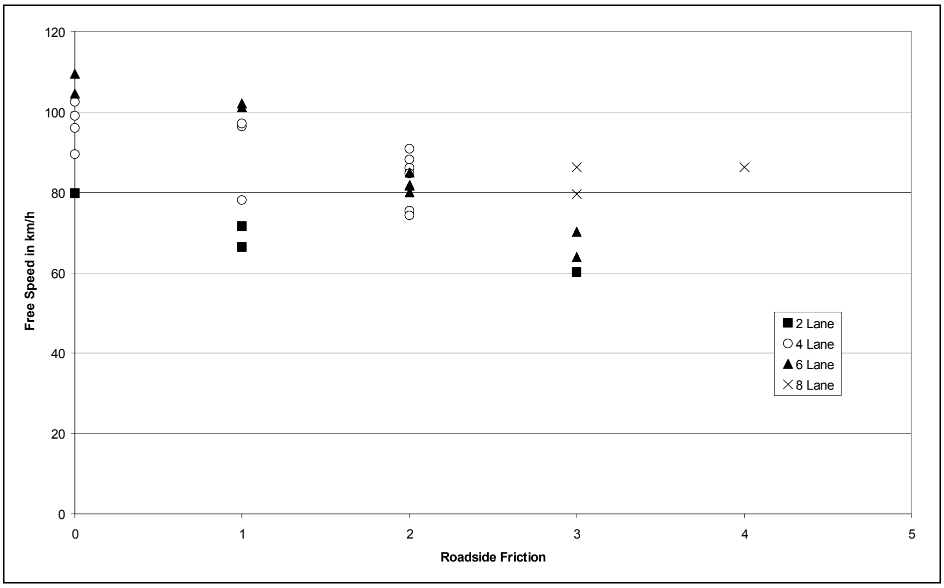

Figure B2.19: Thailand Free Speed vs Roadside Friction

Although the Indonesian and Thailand studies established quantitative means for estimating side friction, with the Indonesian study including NMT, these results are not likely transferable between countries. The only way of adequately estimating these values is by comparing the speeds on road sections and from these, establishing either values for XFRI and XNMT or else a quantitative relationship.

As an example of side friction factors, Annex B2.2 shows photos of road sections from Thailand and India with their different friction levels.

B2.5.7 Calibration of HDM-4 Speed Model

The calibration of the speed model was done using data from N.Z. and Australia (Bennett, 1994 and McLean, 1991). The individual speed constraints were calibrated separeately in a manner similar to that give by McLean (1991). Since mean speeds were used in the analysis instead of individual speeds, the model parameter σ was set to zero. The individual constraining speeds were then combined to calibrate the model parameter β.

Calibrating the individual speed constraints separately results in lower values of β than if all the constraining speeds had been calibrated together from individual vehicle data. This is due to the nature of the PLVM which artificially increases the magnitude of each constraining speed in order to give the required steady state speed when included in the model.

Both methods of calibrating the PLVM have their limitations. Calibration by the method used here may result in the under prediction of steady state speeds when two or more constraints are acting on the vehicle. However, calibration of the complete model can result in an over prediction of the steady state speed when one speed constraint operates in isolation.

Table B2.16 presents the results of the calibration, along with the associated statistical properties (NDLI, 1995). Table B2.11 gives the full set of calibrated parameters for each of the representative vehicles.

Table B2.16: Calibrated Coefficients for β

Vehicle Type	β	R^2-adj	S.E
Passenger Car	0.15 (9.4)	0.86	1.69
Light Truck	0.19 (13.4)	0.77	2.11
Medium Truck	0.16 (10.8)	0.77	1.99
Heavy Truck	0.09 (3.4)	0.87	1.65
Articulated Truck	0.11 (5.0)	0.81	2.26

B2.6 Correction for Time vs Space Speeds

The HDM-4 speed model predicts what is termed the **time mean** speed. This is defined as the (arithmetic) average speed of all vehicles passing a point on the road over a specified time period. An alternative measure, which is more relevant to HDM analyses is the **space mean speed** (also called the 'journey speed') which is the average of all vehicles occupying a given section of road over a specified time period.

The time and space speeds are calculated as:

$$\overline{St} = \frac{\sum \frac{d}{t_i}}{n} = \frac{\sum St}{n} \qquad \text{...(B2.44)}$$

$$\overline{Ss} = \frac{d}{\sum \frac{t_i}{n}} = \frac{n\,d}{\sum t_i} = \frac{n}{\sum \frac{1}{St}} \qquad \text{...(B2.45)}$$

where	$\overline{St}$	is the average time mean speed in km/h
	$\overline{Ss}$	is the average space mean speed in km/h
	St	is the spot speed of the vehicle
	d	is the section length in km
	t_i	is the travel time over section I in h
	n	is the number of vehicles

Wardrop (1952) proposed the following general formula for time and space mean speeds:

$$\overline{St} = \overline{Ss} + \frac{\sigma^2}{\overline{Ss}}$$

where σ is the standard deviation of the space mean speeds

To evaluate the implications of space mean and time mean speeds for HDM-4, Bennett (1996a) used a Monte Carlo simulation. The space and time mean speeds of 1500 vehicles were calculated for a one km section of road between 10 and 100 km/h in 5 km/h speed increments. A comparison was then made of the resulting mean values. The simulation was tested using Wardrop's formula and was found to give almost identical results.

It was found that there was a linear relationship between the time and space mean speeds and that the relationship was proportional to the coefficient of variation (COV) which is calculated as the standard deviation/mean. This is illustrated in Figure B2.20 which shows the relationship between the time and space mean speeds for the four different values of COV.

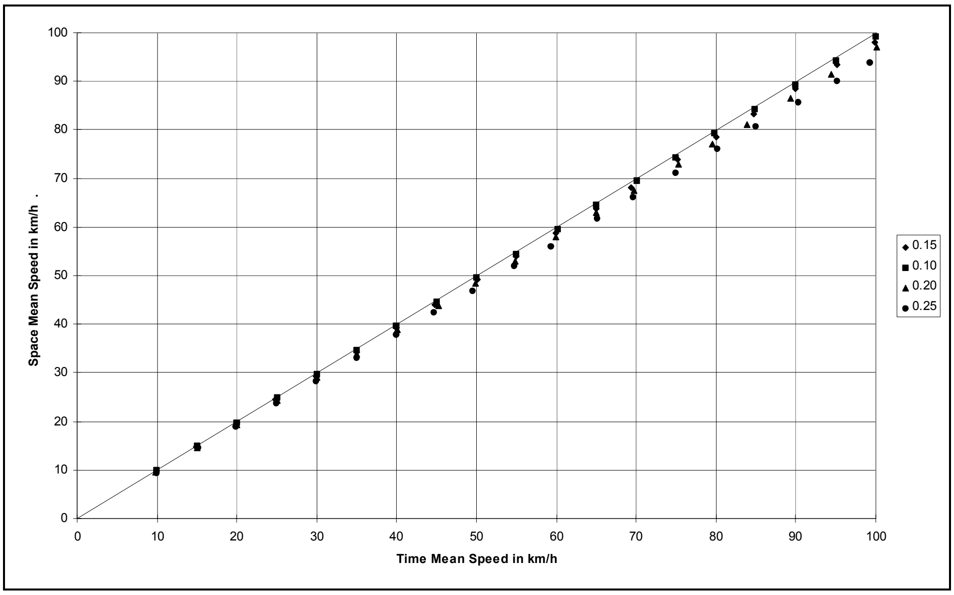

Figure B2.20: Comparison of Time and Space Speeds from Simulation

Values for the COV are in the range of 0.10 to 0.15 for both developed and developing countries (McLean, 1989; Bennett, 1994). Thus, the potential error from using time mean speed as opposed to space mean speed will generally be less than two per cent.

However, since this can be readily corrected the time mean speeds predicted by HDM-4 are converted to space mean speeds by multiplying them by a factor based on the COV. The correction factor a1 is calculated from the COV as (Bennett, 1996a):

$$\text{SPEEDBIAS} = 1.0000 + 0.0122\ \text{COV} - 0.8736\ \text{COV}^2 \qquad R^2 = 1.0 \qquad \ldots(\text{B2.46})$$

where SPEEDBIAS is the bias correction factor for speeds

Thus, the speeds to be used in the analyses can be expressed as:

$$\overline{Ss} = \text{SPEEDBIAS}\ \overline{St}$$

The value for $\overline{St}$ is that predicted by the HDM-4 speed prediction model, *ie* the free speed S.

B3 Effects of Volume on Traffic Flow

B3.1 Introduction

Traffic interactions result in speed fluctuations and a decrease from the free speed. These interactions lead to additional fuel and tyre costs while the decrease in speed leads to an increase in travel time. Consequently, it is important to consider traffic interactions in any analysis with moderate to high traffic levels.

Traffic interactions may arise in several different ways, depending on the type of facility:

- **Overtaking delays** occur on single or two-lane roads when a vehicle catches up with a slower moving vehicle and cannot overtake immediately. It is forced to travel at, or near to, the speed of the slower vehicle (often called a platoon or bunch leader) until an overtaking opportunity presents itself.
- **Demand delays** arise on multi-lane highways and are similar to overtaking delays. Multi-lane highways have greatly improved overtaking capabilities compared to two-lane highways, but as the demand increases the headways between vehicles decrease thereby decreasing the overtaking opportunities. Vehicles are forced to travel at more uniform speeds and, as the demand continues to increase the speeds decline and may eventually cause the facility to become jammed.
- **Crossing delays** are caused by interactions between vehicles travelling in opposite directions. While negligible on wide two-lane or multi-lane roads, they can be substantial on narrow roads where vehicles may be forced to move off the carriageway onto the shoulders. Crossing delays are addressed through the effects of width on vehicle speeds described earlier in Chapter B2.

Box B3.1 Terminology

- **HCM** is the US Highway Capacity Manual (TRB, 1992).
- **pcu** is a ‘passenger car unit’. This is a factor used to convert the traffic stream to a single, common baseline. For example, a heavy truck may affect the traffic flow in an equivalent manner to six single passenger cars so it is assigned a value of six for its pcu. The pcu considers both the space occupied by the vehicle and the impact of its performance on the traffic stream.
- **pcse** is the ‘passenger car space equivalency’. Adopted for HDM analyses, it is similar than the pcu except that it only considers the size of the vehicle and the space occupied, excluding any performance issues.

This chapter addresses the effects of volume on traffic flow. It considers the effects in two separate areas:

- **Speed effects** are the reduction in speeds from the free speed that arise with increasing traffic volumes; and,

- **Acceleration effects** are the vehicle accelerations and decelerations at different levels of traffic volumes. As traffic volumes increase there is an increase in the frequency and magnitude of accelerations and decelerations. This in turn has a significant impact on the road user costs.

Both of these are considered in HDM-4 to provide an integrated model for analysing the effects of traffic volumes on road user effects.

The chapter opens with an introduction to highway capacity speed-volume effects. This is then followed by discussions of passenger car equivalencies, the HDM speed-volume model, and the HDM-4 acceleration-volume model.

B3.2 Highway Capacity and Speed-Volume Effects

The concept of highway capacity is integral to the understanding and modelling of speed-volume effects. It is beyond the scope of this book to deal with the subject in detail, and readers are referred to standard reference documents such as the TRB (1992) Highway Capacity Manual (HCM).

Capacity is defined as:

> "The maximum rate of flow that can be accommodated by a given traffic facility under prevailing conditions" (TRB, 1992)

It is useful to conceive of the capacity by considering the **density** of the traffic. While capacity is a flow, expressed in veh/h, density is the number of vehicles occupying a given length or roadway, averaged over time and expressed in vehicles/km. It describes the proximity of vehicles to one another and reflects the freedom to manoeuvre. The direct measurement of density is difficult since it can only be done from above the road. It is therefore usually calculated from the observed flow as:

$$D = \frac{Q}{S} \qquad \ldots(B3.1)$$

where D is the density in veh/km
Q is the flow in veh/h
S is the speed in km/h

Flow itself is related to the headways adopted by the vehicles and can be calculated as:

$$Q = \frac{3600}{hwy} \qquad \ldots(B3.2)$$

where hwy is the average headway in s/veh

The relationship between speed, flow and density is illustrated in Figure B3.1 and this forms the basis for all uninterrupted flow capacity analyses. The data from which these figures were generated is from the Indonesian Highway Capacity Study for four-lane, divided roads (Sweroad, 1997).

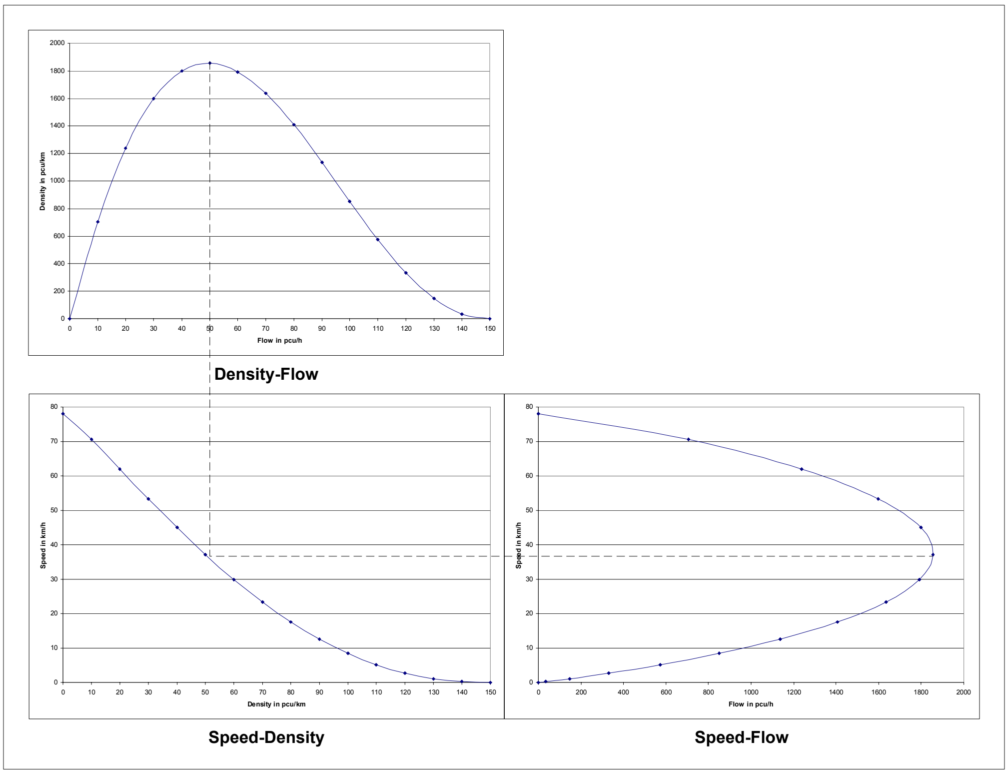

Figure B3.1: Characteristics of Traffic Flow

From the curves in Figure B3.1 it can be noted that:

- At the maximum (jam) density, the speed is zero—all vehicles are stopped on the road. Thus, the flow is also zero.
- When there is no flow, the density is zero. The speed is purely theoretical and it would be whatever the first driver would select.
- Between these two extremes the traffic results in a maximising effect. As the density increases, the flow also increases. This results in a relatively minor reduction in speeds. However, once the density reaches a certain point, the speeds begin to decrease.

The maximum flow—*ie* the capacity—arises at the point where the product of increasing density and decreasing speed results in a reduced flow. This is the critical density and the critical speed. If the capacity is exceeded the system breaks down and enters an unstable flow regime. This sees a loss of service in the facility and additional delays.

Speed-volume relationships have been established in a number of studies and are critical to the economic appraisal of many road improvement projects. They indicate the reduction in speed which accompanies increasing traffic demand, and thus are the mechanism for quantifying the benefits of capacity improvements or improved traffic management. Figure B3.2 shows some of the curves developed by the early researchers into speed-volume modelling. They proposed a range of models including the Underwood, Edie, Greenshield, and Greenburg models as well as two-segment, three-segment models. Each of these were found to fit different data sets with varying degrees of success and no ‘universal’ model was found to give the best overall results.

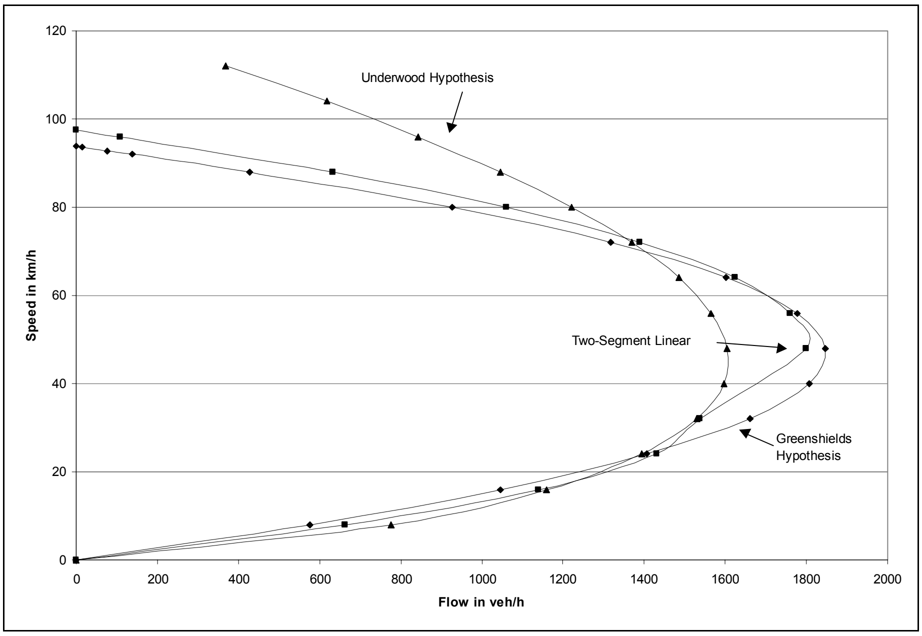

Figure B3.2: Examples of Speed-Volume Curves

The key attributes of these speed-volume relationships are:

- **Capacity:** This represents the maximum flow level on the speed-volume curve;
- **Slope:** The slope of the speed-volume curve varies with many factors but, as can be seen from Figure B3.2, a linear approximation has been found to be adequate in many studies;
- **Free-Speed:** This represents the maximum speed of traffic; and,
- **Free-Speed Plateau:** While Figure B3.1 and Figure B3.2 shows an immediate decrease in speeds with increasing flow, others have found evidence of a free-speed "plateau". This sees the free speed maintained by traffic at low to moderate flows. Speeds only decrease once a certain flow level is reached. Hall, *et al.* (1994) suggest that this plateau is a function of free speed, with higher free speed roads being affected by increasing flows sooner than lower free speed roads.

In developing the HDM speed-volume model Hoban (1987) considered all of these factors and developed a flexible model which could be applied in a range of conditions.

B3.3 HDM Speed-Volume Model

HDM-4 and HDM-95 uses the speed-volume model proposed by Hoban (1987) and shown in Figure B3.3[1]. It is built around the predicted free speed and five key parameters which vary by road class:

- **Qult:** The ultimate capacity of the road in pcse/h;
- **Qnom:** The nominal capacity in pcse/h where all vehicles are travelling at the same speed;
- **Qo:** The flow in pcse/h where interactions commence;

[1] This figure uses the term 'pcse' in place of 'pcu'. These represent 'passenger car space equivalencies' and their quantification is discussed in Section B3.4.

- **Snom:** The speed in km/h at nominal capacity; and,
- **Sult:** The speed in km/h at ultimate capacity.

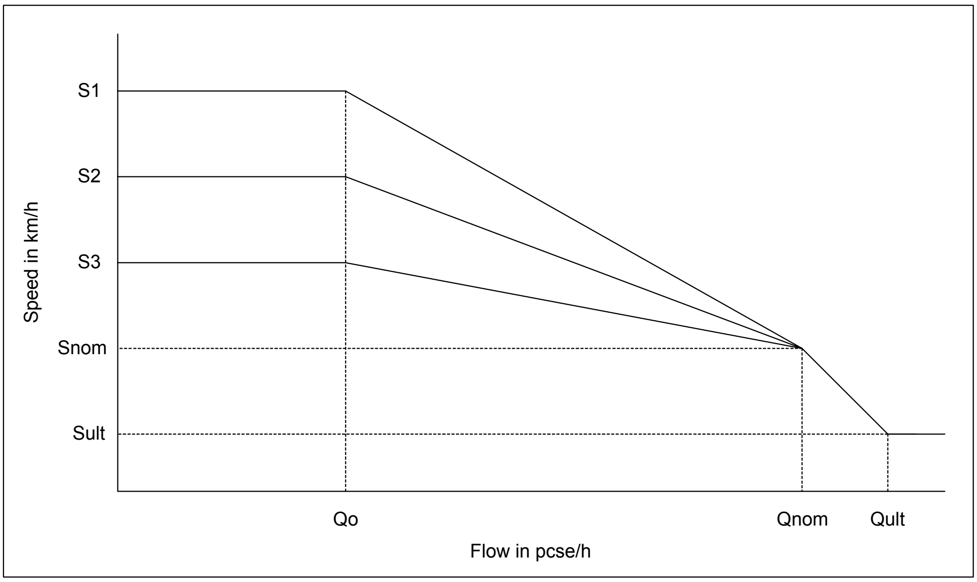

Figure B3.3: HDM Speed-Volume Model

The basic premise of the HDM speed-volume model is that below a certain flow level (Qo) vehicles travel at their free speed. When the flow exceeds Qo there are traffic interactions which cause the mean speed to decrease. On multi-lane highways this may be at a flow as high as 800-1000 pcse/h while on narrow single lane roads it may be a flow of 0. At the nominal capacity (Qnom) all vehicles are travelling at the same speed. If there are additional vehicles the speeds drop rapidly to the minimum speed (Sult) which is at the ultimate capacity (Qult). If the demand exceeds Qult the speed remains at Sult.

It is assumed in the model that all speeds between Qo and Qnom decrease linearly from their free speeds—in HDM estimated using the free speed model—to the speed at nominal capacity Snom. This speed is assumed to be the 15th percentile speed of the slowest vehicle type (usually trucks). It is approximated by the mean less one standard deviation, or 0.85 the free speed (Hoban, 1987), *i.e.*:

$$\text{Snom} = 0.85 \min(S_i) \qquad \text{…(B3.3)}$$

where S_i is the speed of vehcile class i

The model is applied using the following equations:

$$\text{SQ} = \text{S} \qquad \text{Q} < \text{Qo} \qquad \text{…(B3.4)}$$

$$\text{SQ} = \text{S} - \frac{(\text{S} - \text{Snom})(\text{Q} - \text{Qo})}{(\text{Qnom} - \text{Qo})} \qquad \text{Qo} \leq \text{Q} < \text{Qnom} \qquad \text{…(B3.5)}$$

$$\text{SQ} = \text{Snom} - \frac{(\text{Snom} - \text{Sult})(\text{Q} - \text{Qnom})}{(\text{Qult} - \text{Qnom})} \qquad \text{Qnom} \leq \text{Q} < \text{Qult} \qquad \text{…(B3.6)}$$

$$\text{SQ} = \text{Sult} \qquad \text{Q} \geq \text{Qult} \qquad \text{…(B3.7)}$$

where SQ is the traffic-influenced speed in km/h
S is the free speed

To adjust the speed for different road classes (*eg* multi-lane highways) it is modified using a clibration factor:

$$SQ = \max(SQ\ CALBFAC, Sult) \qquad \ldots(B3.8)$$

where CALBFAC is a road dependant calibration factor (range: 0.1 - 1.0)

Table B3.1 gives the default HDM parameter values for use with the model.

Table B3.1: Default HDM Speed-volume Model Parameters

Road Type	Width (m)	Qo/ Qult	Qnom/ Qult	Qult (PCSE/h)	Sult (km/h)
Single Lane Road	< 4	0.0	0.70	600	10
Intermediate Road	4 to 5.5	0.0	0.70	1800	20
Two Lane Road	5.5 to 9	0.1	0.90	2800	25
Wide Two Lane Road	9 to 12	0.2	0.90	3200	30
Four Lane Road	>12	0.4	0.95	8000	40

Source: Hoban, et al. (1994)

A final adjustment is made to the speed to account for the effects of time versus space mean speeds. This uses Equation B2.46 as follows:

$$SQ = SQ\ SPEEDBIAS \qquad \ldots(B3.9)$$

Roberts (1994) modified the model for Bangladesh by incorporating level of service concepts. The most significant departure was for the zone where Q > Qult. It was assumed that there was a linearly decreasing relationship with a speed of zero at 2 Qult. This contrasts with the assumption in HDM-4 that the speed Sult applies when Q > Qult.

As noted by HTC (1999a), the HDM speed-volume model is very flexible and, when calibrated, is able to accurately recreate observed speed-volume profiles. The speed at nominal capacity is critical as this governs the slope of the speed-volume curve. An example of this from Thailand is presented in Figure B3.4 (HTC, 1999a) where it will be observed that using calibrated parameters markedly improved the fit of the model over the predictions using the HDM defaults.

To apply the speed-volume model HDM breaks the AADT down into different flow bands. These consist of the number of hours per year that the traffic will be at certain volumes, with the hourly flow expressed as a percentage of the AADT. The total annual costs are established by calculating the costs based on the speeds in each flow level and multiplying them by the number of hours per year the flow levels arise. The total over the year is the total road user costs.

Figure B3.5 shows flow profiles from different countries and road classes. It will be noted that the flow profiles vary not only between countries, but also between different classes of roads. It is therefore essential that they be established based on local data. Bennett and Paterson (1999) give guidance on how this is done.

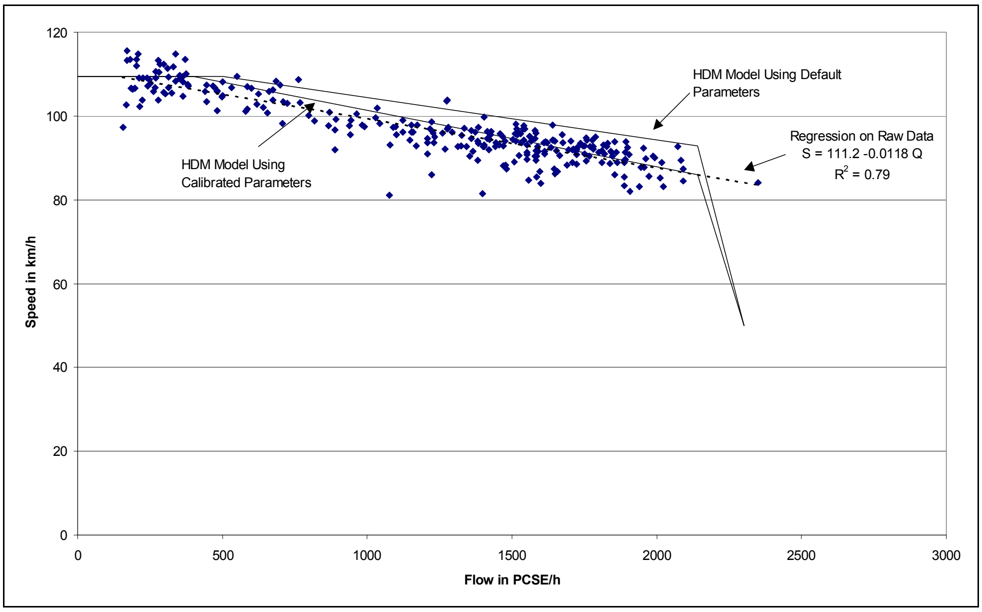

Figure B3.4: Example of Calibration of HDM Speed-Volume Model from Thailand

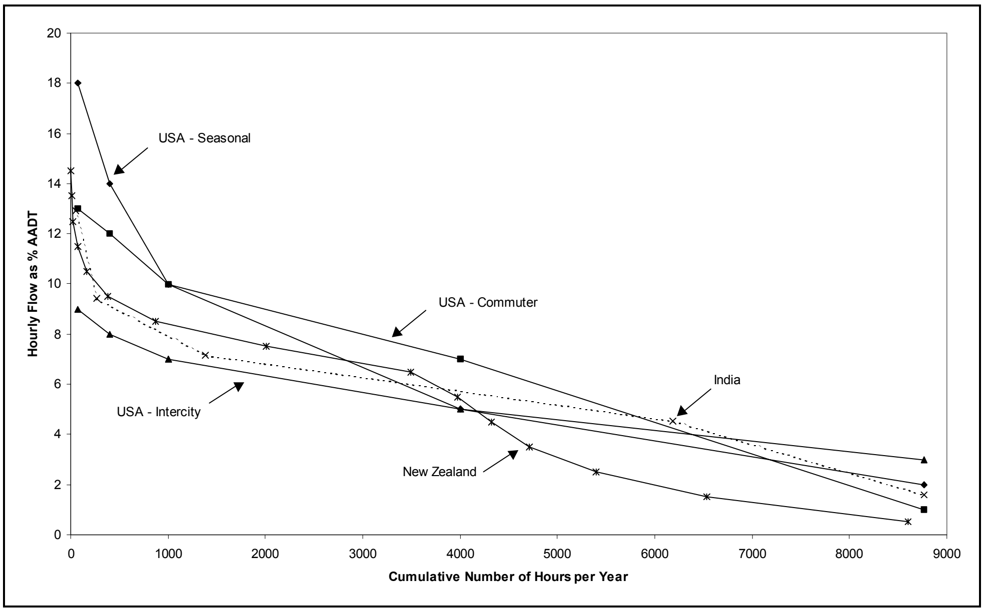

Figure B3.5: Examples of Traffic Flow Distributions

Table B3.2 is an example of the hourly flow distributions used in HDM studies in several countries. The table includes a distribution for non-motorised traffic from India since the same approach is applied in calculating their costs.

Table B3.2: Sample Hourly Flow Distributions

Thailand[1]		Australia[2]		India[3]		
Hours per year at that flow	Hourly Flow as % AADT	Hours per year at that flow	Hourly Flow as % AADT	Hours per year at that flow	Hourly Flow as % AADT	
					Motorised	Non-Motorised
2996	1.466	4760	1.5	2575	1.59	0.79
1921	4.072	2500	6.0	4805	4.53	4.20
3413	5.844	1000	8.0	1114	7.14	8.88
405	7.656	250	10.0	216	9.41	12.17
25	10.500	250	13.5	50	12.92	35.04

Sources: *1/ HTC (1999a)*
2/ Hoban (1987)
3/ NDLI (1997)

B3.4 Equivalency Factors

B3.4.1 Introduction

Traffic streams are comprised of a range of vehicles, from passenger cars to heavy trucks. For the purposes of defining the capacity it is necessary to convert these into a homogeneous traffic stream. There are two types of equivalency factors in use, and it is important to understand the difference between them in order to correctly apply the HDM speed-volume model.

- **Passenger Car Equivalencies (PCE):** These are also called passenger car units (PCU) have been commonly used in capacity studies, such as with the HCM. These represent the number of additional passenger cars which, if added to the traffic stream, would have the same impeding effect as the vehicle of interest—usually a truck. The PCE is based on two considerations: (1) the area occupied by the vehicle and (2) its impact on the traffic stream through its performance.

- **Passenger Car Space Equivalencies (PCSE):** These are used in HDM. These are **only** based on the area occupied by the vehicle and not on its performance impact on the traffic stream since, as described later, the performance is already explicitly modelled.

To illustrate the difference between these two measures, the following sections will describe how each are calculated.

B3.4.2 PCE – Basis and Estimation

As described in Chapter B2, the ability of a vehicle to accelerate is a function of the forces opposing motion and the power-to-weight ratio. On upgrades, if the vehicle has sufficient power to overcome the gravitational deceleration, is unaffected and maintains its speed. If there is insufficient power it decelerates to a speed where the forces are in balance and the acceleration is zero. This is termed the 'crawl speed' and other vehicles are impeded unless they are able to pass.

The heavy vehicle impedance effects vary with traffic volume, particularly for two-lane highways. At low flows and with ample overtaking opportunities truck impedance is relatively small. It is at a maximum for moderate flows with limited overtaking opportunities. For the same vehicle, the HCM PCEs are higher on two-lane highways than multi-lane highway because of the reduced overtaking opportunities (TRB, 1992).

The effects also vary with the percentage of heavy vehicles in the traffic stream with the values decreasing with an increasing percentage of heavy vehicles. This is because of two effects. Firstly, there are less passenger cars to be impeded. Secondly, trucks will impede other trucks as well as passenger cars so the aggregate effect is less.

The equivalency criterion used to define PCEs should be the same as that used to define flow performance (*eg* journey speeds). A variety of methods have been used to derive PCEs and McLean (1989) gives a discussion of them in detail. Among those used are:

- **Direct Method:** Based on observing flows of mixed traffic and comparing their performance to those of only passenger cars.
- **Equivalent Overtaking Method (Walker Method):** Equivalence is defined in terms of the overtaking rate about a single slow-moving truck necessary for cars to achieve a particular distribution of speeds, relative to the aggregate overtaking rate within a stream of cars with the same distribution of achieved speeds.
- **Equivalent Delay Method:** Similar to the Walker method but includes the delays associated with waiting for an overtaking opportunity.
- **Headways Method:** This is based on the ratio of the headway (including vehicle length) for trucks to the headway for cars. It is predicated on the fact that platoon leaders are usually heavy trucks.
- **Platoon Leaders:** This is based on the ratio of truck platoon leaders to car platoon leaders.

Some studies have tried to develop PCEs from empirical speed-low studies, but these have not been overly successful.

McLean (1989) indicates that for two-lane highways, the Walker method was used in the 1965 HCM. The headway method was used both in 1965 and 1985 to determine generalised road segment PCE values. Simulation has proved to be a useful tool and that was used in the 1985 HCM for the specific grade section PCEs. The equivalency criterion was the average speed of the up grade traffic.

B3.4.3 PCSE – Basis and Estimation

The discussion in Section B3.4.2 showed that PCE values can be established in several different ways. The common thread to all the estimation methods is that the PCE reflects two considerations:

- The effect of the vehicle’s performance on the traffic stream—*ie* the additional delay caused by their slower speed and greater difficulty in overtaking; and,
- The additional space occupied by the vehicle.

As described in Section B3.4.2, the HCM PCE values are a function of gradient. On level roads where the speeds are uniformly low and overtaking is generally not possible the space component represents a lower bound for the PCE. Since the HDM speed prediction model already accounts for the differences in performance between vehicles, this minimum PCE has been adopted for HDM capacity analyses. Called the ‘passenger car space equivalent’, or PCSE, it represents the additional space occupied by a vehicle relative to a passenger car.

Hoban, *et al.* (1994) describe the recommended PCSE values for HDM. This work builds upon the earlier work of Hoban (1987). The PCSE were established based on the assumption that each vehicle has a typical length as well as typical leading and following headways[1]. Using an assumed speed of 72 km/h, Hoban, *et al.* (1994) calculated the basic PCSE values shown in Table B3.3. These basic values only accounted for the longitudinal space occupied by vehicles. Additionally, larger vehicles tend to impact adjacent lanes, with this "adjacent lane" effect being greater for larger vehicles and for narrower roads (Hoban, *et al.*, 1994). This led to the final PCSE values varying by width as shown in Table B3.3.

Table B3.3: HDM Default PCSE Values by Vehicle Class

Vehicle Class	Avg. Length (m)	Time Headway (s)	Space Headway (m)	Total Space (m)	Basic PCSE	Recommended Values (Includes "Basic" plus adjacent lane effects)		
						Two-Lane Four-Lane	Narrow Two-Lane	One-Lane
Car	4.0	1.6	32	36.0	1.0	1.0	1.0	1.0
Pickup	4.5	1.8	36	40.5	1.1	1.0	1.0	1.0
Heavy Bus	14.0	2.2	44	58.0	1.6	1.8	2.0	2.2
Light Truck	5.0	2.0	40	45.0	1.3	1.3	1.4	1.5
Medium Truck	7.0	2.2	44	51.0	1.4	1.5	1.6	1.8
Heavy Truck	9.0	2.4	48	57.0	1.6	1.8	2.0	2.4
Truck and Trailer	15.0	2.5	50	65.0	1.8	2.2	2.6	3.0

Notes: 1/ Basic data from Hoban, *et al.* (1994)
2/ Time headway calculated from space headway using 72 km/h speed
3/ Truck and Trailer average length increased to 15 from 11 given in Hoban, *et al.* (1994) based on difference between total space and space headway

The PCSE values in Table B3.3 were calculated by firstly establishing the total space occupied by passenger cars (default = 36 m). The 'Basic PCSE' values for the other vehicle classes were then determined from the ratio of their total space to the passenger car space. This basic value was then subjectively adjusted for width effects to obtain the recommended values in the final three columns.

HTC (1999a) used data from high flow sites in Thailand to calculate PCSE. They broadly confirmed the default HDM values from Table B3.3, although it was found that heavy trucks had higher PCSE values. They also proposed a value of 0.33 for motorcycles, reflecting the fact that under extremely congested conditions motorcycles are able to travel between lanes and therefore often have a negligible impact on traffic.

B3.5 Estimating Capacity

To apply the HDM speed-volume model it is necessary to provide two values for capacity:

- **Nominal Capacity:** The capacity where the traffic stream is travelling at a constant speed. This is the capacity which is sustainable over long time periods; and,
- **Ultimate Capacity:** The maximum possible capacity before flow breaks down.

When considering capacity, it must be appreciated that capacity is not a single, fixed value. Instead, it varies over time, between countries, and even between similar roads in the same country. Examples of this are:

1 The PCSE is analogous to truck equivalencies calculated using the headway method. This defines the equivalency of a truck as the ratio of the average headway for trucks in the stream (in s) to the average headway for cars in the stream (in s).

- ❑ **Time Variations:** The HCM updates have shown that there is an increase in capacity over time. For example, the ideal capacity of two-lane highways was 2000 pcu/h in the 1964 HCM but 2800 pcu/h in the 1985 HCM.
- ❑ **Between Countries:** The following capacities in pcse/h have been reported for two-lane highways in different countries: Colombia 2950 (Radelat, 1994); Finland 2500 (Pursala, 1994); India 2500 (Kadiyali, 1991); Japan 2500 (Nakamura, 1994); Thailand 2800 (HTC, 1999a). As an example of these differences, Figure B3.6 shows the capacities as a function road width from several countries. This figure serves to highlight the need to establish locally calibrated values since the results are markedly different between the various countries.

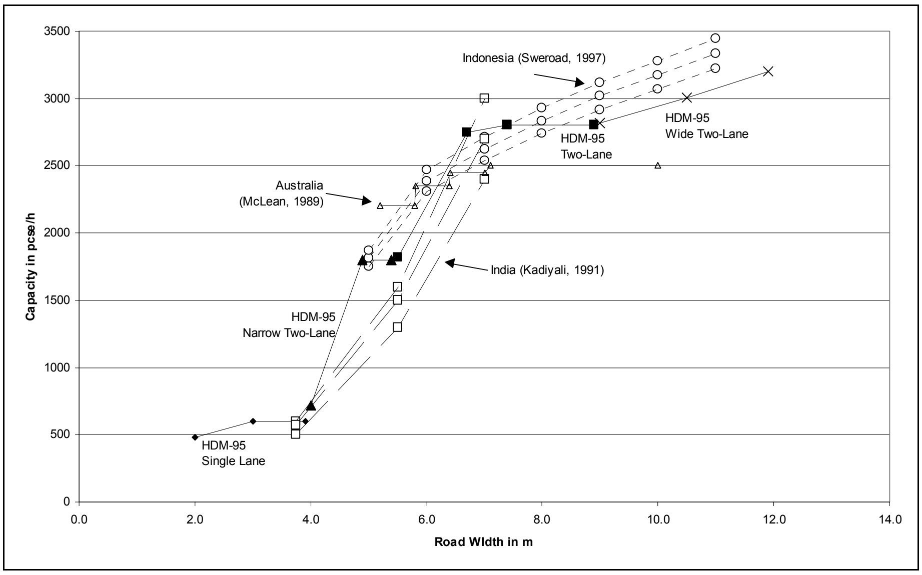

Figure B3.6: Effect of Width on Capacity

- ❑ **Between Roads:** Figure B3.7 is an example of the variations in capacity between different road classes from France (Cohen, *et al.*, 1994). The two-lane highway values are for both directions while the multi-lane values are per lane. This figure clearly shows the variations in capacity not only between road classes, but even between roads within the same class. For example, the two-lane highway capacities vary from 1500 – 2200 pcse/h. Thus, in estimating capacity one needs to measure it at a range of sites and establish an overall representative value instead of basing it on measurements at a single site.

As shown in Figure B3.8, there are several different methods available for estimating capacity (Minderhoud, *et al.,* 1997). These authors make an important observation:

> *"Attempts to determine the capacity of a road by existing methods will generally result in a capacity value estimate, but the validity of this value is hard to investigate because of the lack of a reference capacity value, which is supposed to be absolutely valid. A clear, reliable method does not appear to be available at this time."*

Irrespective of which method is used, it is necessary to sample over sufficient time intervals to obtain a reliable estimate. One normally measures the flow in five to 15 minute periods and converts this to an hourly flow. In most situations capacity is a difficult condition to reliably encounter so one must often extrapolate data collected at lower flows to estimate the capacity.

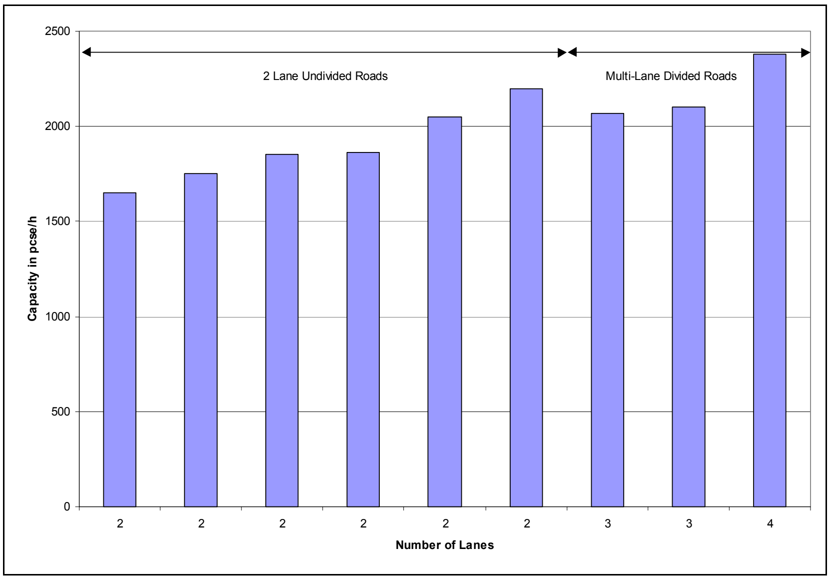

Figure B3.7: Example of Variation in Capacity Between Roads in France

Figure B3.8: Capacity Estimation Methods

The headway method presented earlier as Equation B3.2 can be used to estimate the ultimate capacity. This is based on the theory that at high flows there are still two populations of vehicles: constrained (followers) and unconstrained (leaders). The distribution of following headways is expected to be the same as for constrained drivers in any stationary traffic stream. Although relatively easy to collect, it is considered that this method tends to overestimate capacity due to the assumption that the distribution of constrained drivers can be compared at capacity with that below capacity.

Another approach is to collect data on roads which will reach capacity at some point during the study period. The road capacity is taken to be the maximum flow, or the mean of several very high flows, observed during the analysis period. This is illustrated in Figure B3.9.

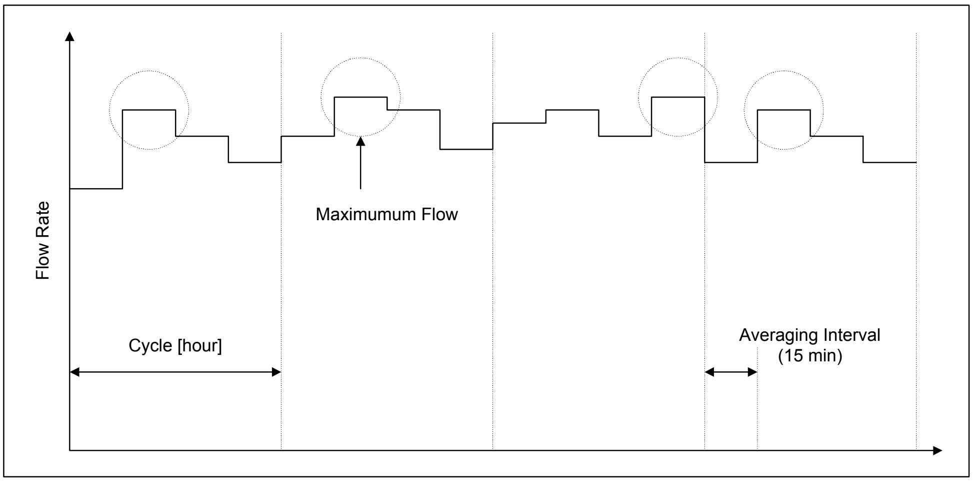

Figure B3.9: Estimating Capacity from Observed Flows

Irrespective of which method is applied it is necessary to estimate both the ultimate (Qult) and nominal (Qnom) capacities. The difference between them can be illustrated through Figure B3.10 which shows the flows from five minute intervals at a six-lane divided motorway in Thailand (HTC, 1999a).

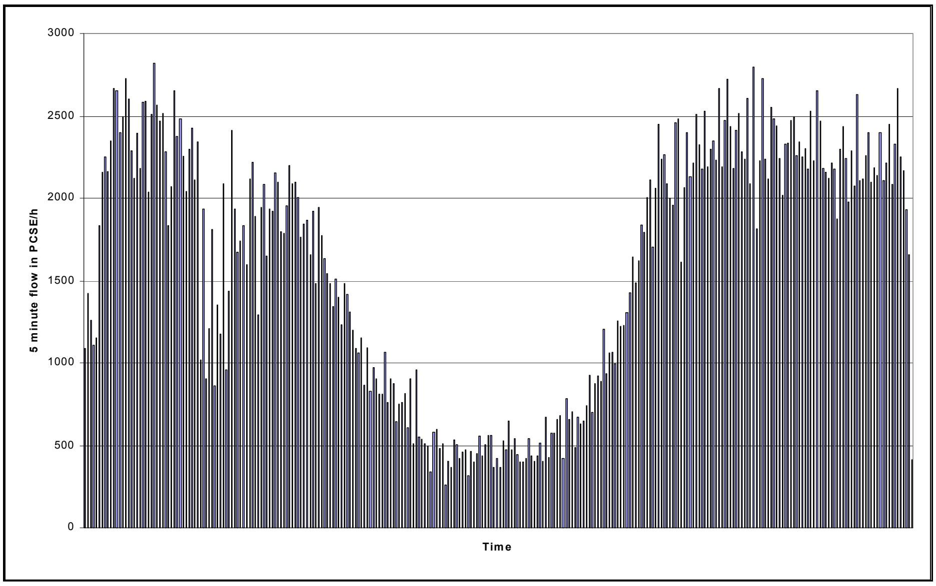

Figure B3.10: Five Minute Flow Levels for Six-Lane Divided Motorway in Thailand

The ultimate capacity prevails over short time periods so is best represented by the maximum flow observed on a road. Here, the peak flow observed based on five minute intervals was 2181 pcse/h. However, this level was reached only once in the study and so is indicative of the ultimate capacity of the road.

The nominal capacity is the maximum flow rate that can reasonably be expected to be maintained so it should be based on a longer period. HTC (1999a) used 10 and 15 minute averages for these values, adopting a nominal capacity of 2107 pcse/h for this site.

It should also be appreciated that the ultimate capacity is synonymous with the HCM capacity under 'ideal' conditions. The HCM recommends ideal capacities of 2000 PCSE/h/lane for multi-lane highways and 2800 PCSE/h for two-lane highways which are identical to those proposed for HDM given in Table B3.1. These ultimate capacities are reduced in the HCM to reflect factors such as lane and shoulder width and, for two-lane highways, directional split and percentage of no-passing zones. These factors need to be taken into account in any capacity study trying to quantify Qult and Qnom.

B3.6 Acceleration-Volume Effects

B3.6.1 Introduction

In Section B3.2 it was shown how as traffic flow and traffic density increase there is a decrease in speeds due to traffic interactions. Traffic interactions cause vehicles to accelerate and decelerate and, thus, have an impact on road user costs. An example of these differences is illustrated in Figure B3.11 which shows idealised acceleration distributions for traffic in congested and uncongested conditions. It will be noted that the congested distribution has a much higher magnitude of acceleration than the uncongested distribution, reflecting the ‘stop-start’ conditions.

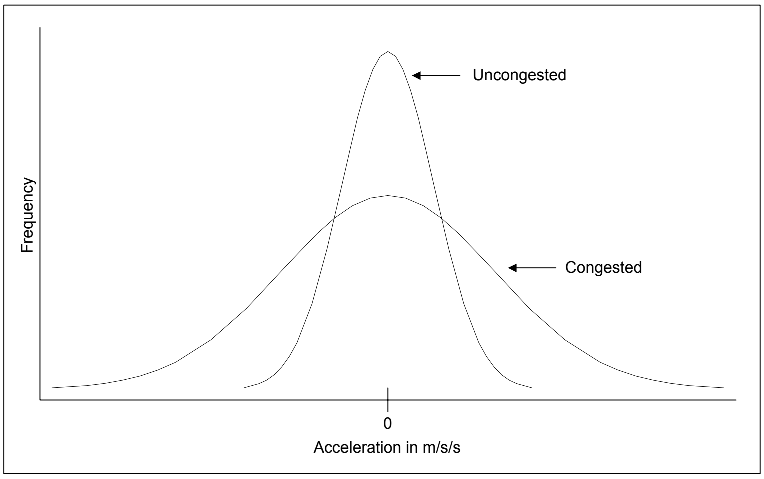

Figure B3.11: Congested vs Uncongested Acceleration Levels

Under ideal conditions drivers would maintain a steady-state speed without any decelerations or accelerations. However, in practice this is not possible since drivers are forced to modify their speeds in response to:

- Traffic interactions;
- The road alignment;
- Roadside friction; and,
- Pavement surface condition.

In addition, due to their reliance on sensory perception, drivers do not maintain their steady-state speed in the absence of any of the above factors so there are always some accelerations or decelerations as the vehicle travels along the road. These can be significant, such as when induced by severe alignment or congestion, or minor, such as those which naturally arise as the driver tries to maintain a speed.

Research into car-following theory found that the accelerations[1] of a driver approximately follow a Normal distribution (Bester, 1981). The standard deviation of the accelerations — called the **acceleration noise**—gives an indication of the severity of speed changes. Low values of acceleration noise indicate that there are minor speed changes; large values major speed changes.

The total acceleration noise for a vehicle can be considered to be comprised of two components; the natural acceleration noise and the traffic induced acceleration noise. These may be combined as follows to give the total acceleration noise:

$$\sigma a = \sqrt{\sigma at^2 + \sigma an^2} \qquad \text{…(B3.10)}$$

where σa is the total acceleration noise in m/s^2
σat is the noise due to traffic interactions in m/s^2
σan is the natural noise ascribed to the driver and road in m/s^2

The natural noise can be described as:

$$\sigma an = f(\sigma adr, \sigma aal, \sigma asf, \sigma anmt, \sigma airi) \qquad \text{…(B3.11)}$$

where σadr is the noise due to natural variations in the driver's speed in m/s^2
σaal is the noise due to the road alignment in m/s^2
σasf is the noise due to roadside friction in m/s^2
$\sigma anmt$ is the noise due to speed limits in m/s^2
$\sigma airi$ is the noise due to roughness in m/s^2

The following sections describe how these noises are calculated.

B3.6.2 Traffic Acceleration Noise

On the basis of car-following theory Bester (1981) reported that the relationship between traffic noise and traffic density was:

$$\sigma at = \sigma atm\,(1 - 6.75\ RELDEN + 13.5\ RELDEN^2 - 6.75\ RELDEN^3) \qquad \text{…(B3.12)}$$

where σatm is the maximum traffic noise in m/s^2
RELDEN is the relative density

The relative density was defined as:

$$RELDEN = \frac{k}{kjam} \qquad \text{…(B3.13)}$$

where k is the traffic density in veh/km
kjam is the jam density in veh/km

Equation B3.12 has a minimum value for a relative density of approximately 0.33. Below this density traffic interactions are considered negligible and only the natural noise is significant.

[1] In this book the generic term acceleration is often used to describe both accelerations (positive acceleration) and decelerations (negative acceleration).

The HDM-4 model uses flow as opposed to density, so an investigation was made of the predictions of Equation B3.12 in this context. Using the HDM-4 speed-flow model for a multi-lane road, Equation B3.12 predicts traffic interactions to begin at approximately 90 per cent of flow capacity. However, as described by Greenwood and Bennett (1995), experiments with an instrumented vehicle showed that traffic noise began to be manifested at much lower volumes than would be predicted from Equation B3.12.

Another consideration in modelling acceleration behaviour is the fact that the volume where speeds begin to decrease in the HDM speed-volume model (Qo) is also the volume where by definition traffic acceleration noise becomes noticeable. Thus, the intercept of the natural and traffic noises needs to be a variable as opposed to a constant as in Bester (1981).

A new model was therefore developed by which used the relative flow as an independent variable and which also could take into consideration Qo as a variable. The model developed was as follows (NDLI, 1995):

$$\sigma at = \sigma atm \; \frac{1.04}{1+e^{(a0+a1\;VCR)}} \qquad \ldots(B3.14)$$

$$VCR = \frac{Q}{Qult} \qquad \ldots(B3.15)$$

where VCR is the volume-to-capacity ratio
a0 and a1 are regression coefficients quantified as follows:

$$a0 = 4.2 + 23.5\left(\frac{Qo}{Qult}\right)^2 \qquad \ldots(B3.16)$$

$$a1 = -7.3 - 24.1\left(\frac{Qo}{Qult}\right)^2 \qquad \ldots(B3.17)$$

The basic form of the equation was confirmed through field experiments, although, as described in NDLI (1995), it could not be rigorously quantified because of the dynamic nature of the volume-to-capacity ratio at intermediate volumes. The acceleration noise is measured by a vehicle moving through a traffic stream and it is impossible to accurately estimate the volume-to-capacity ratio to correlate with the acceleration noise being experienced by the vehicle. However, at the extremes the volume-to-capacity ratio is readily quantified so these extremes were focused on by NDLI (1995) in developing the model.

In the discussion which follows the traffic noise ratio (σatrat) will often be used. This is defined as:

$$\sigma atrat = \frac{\sigma at}{\sigma atm} = \frac{1.04}{1+e^{(a0\,+\,a1\,VCR)}} \qquad \ldots(B3.18)$$

When recording the acceleration noise, only the total noise can be measured. At low flow levels the total noise is assumed to be equal the natural noise since the other traffic does not have an impact. Under maximum flow conditions—*ie* at capacity—the maximum total noise (σamax) is observed. It is therefore desirable to calculate the total acceleration noise at any flow level in terms of the two measurable variables, σamax and σan.

Rewriting Equation B3.18 gives the traffic noise at any flow as:

$$\sigma at = \sigma atm \; \sigma atrat \qquad \text{...(B3.19)}$$

The maximum traffic noise is expressed in terms of the maximum total noise and the natural acceleration noise as:

$$\sigma atm = \sqrt{\sigma a\max^2 - \sigma an^2} \qquad \text{...(B3.20)}$$

Substitution of the above into Equation B3.10 yields the following formula for calculating the total acceleration noise:

$$\sigma a = \sqrt{\sigma an^2 + \left(\sigma atrat \sqrt{\sigma a\max^2 - \sigma an^2}\right)^2} \qquad \text{...(B3.21)}$$

which may be rearranged to:

$$\sigma a = \sqrt{\sigma an^2 \left(1 - \sigma atrat^2\right) + \sigma a\max^2 \; \sigma atrat^2} \qquad \text{...(B3.22)}$$

Equation B3.22 predicts the total acceleration noise on the basis of the maximum total and the natural acceleration noises along with the traffic noise ratio which is a function of flow. Figure B3.12 illustrates the predictions of Equation B3.22 for various values of Qo/Qult. A maximum traffic noise of 0.6 m/s^2 were used in preparation of the figure. It also shows the natural noise of 0.10 m/s^2 which, when added to the traffic noise, gives the total acceleration noise.

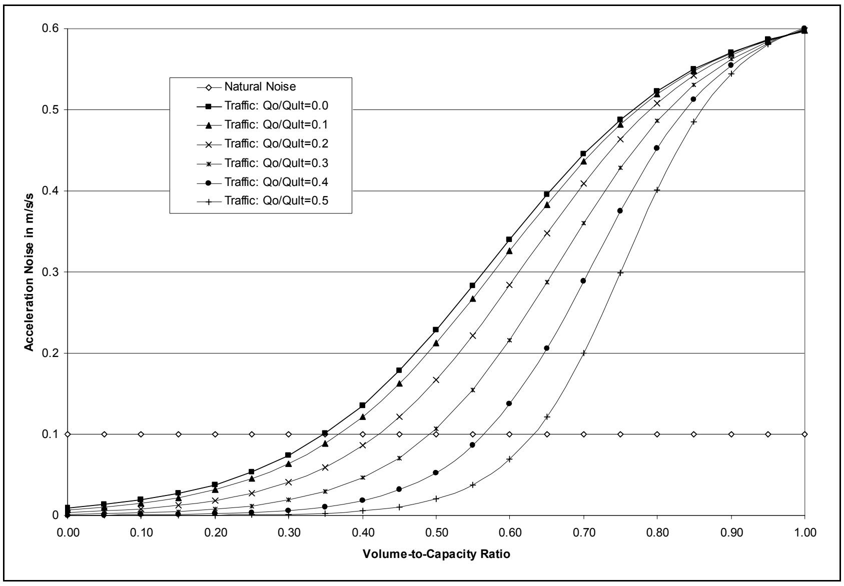

Figure B3.12: Effect of Volume-to-Capacity Ratio on Acceleration Noise

In HDM-4 the maximum total acceleration noise is limited to the range of 0.1 to 0.75, with further limits being able to be defined for travel on gradients where vehicle performance will further limit the acceleration noise.

B3.6.3 Natural Acceleration Noise

The total natural noise is given by:

$$\sigma an = \sqrt{\max((\sigma adr^2 + \sigma aal^2), \sigma asf^2, \sigma anmt^2, \sigma airi^2)} \quad \ldots(B3.23)$$

For HDM-4 the driver noise (σadr) and the alignment noise (σaal) were combined into a single value as it is difficult to differentiate between these two components (NDLI, 1995). The other three components of natural noise — side friction (σasf), non-motorised transport (σanmt) and roughness (σairi) — are modelled as linear functions as shown in Figure B3.13. The values are calculated as[1]:

$$\sigma asf = 2.5\ (1 - XFRI)\ \sigma xfrimax \quad \ldots(B3.24)$$

$$\sigma anmt = 2.5\ (1 - XNMT)\ \sigma xnmtmax \quad \ldots(B3.25)$$

$$\sigma airi = \min\left(\sigma iri\max, \sigma iri\max \frac{RI}{AMAXRI}\right) \quad \ldots(B3.26)$$

where

σxfrimax	is the maximum acceleration noise due to side friction (default = 0.20 m/s^2)
σxnmtmax	is the maximum acceleration noise due to non-motorised transport (default = 0.40 m/s^2)
σirimax	is the maximum acceleration noise due to roughness (default = 0.30 m/s^2)
AMAXRI	is the roughness at which the maximum acceleration noise arises (default = 20 IRI m/km)

These individual factors are discussed below.

NOISE DUE TO NATURAL SPEED VARIATIONS AND ROAD ALIGNMENT

The noise due to natural speed variations (σadr) is in many respects analogous to the basic desired speed of a vehicle. It is the noise which will arise on high standard roads in the absence of any other constraints such as alignment, roadside friction or roughness.

The noise due to road alignment (σaal) arises because of a road's gradients and curvature. This will be proportional to the severity of the route and the number of speed changes induced by the alignment.

Although these effects are separate, in practical terms it is virtually impossible to differentiate between the natural and alignment noises since both are being measured simultaneously. For this reason, both are combined in Equation B3.23.

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1 The constant 2.5 is used to reflect the limit of 0.6 – 1.0 for the factors XFRI and XNMT. It results in an effective range of 0 – 1 for the calculated values for σasf and σanmt.

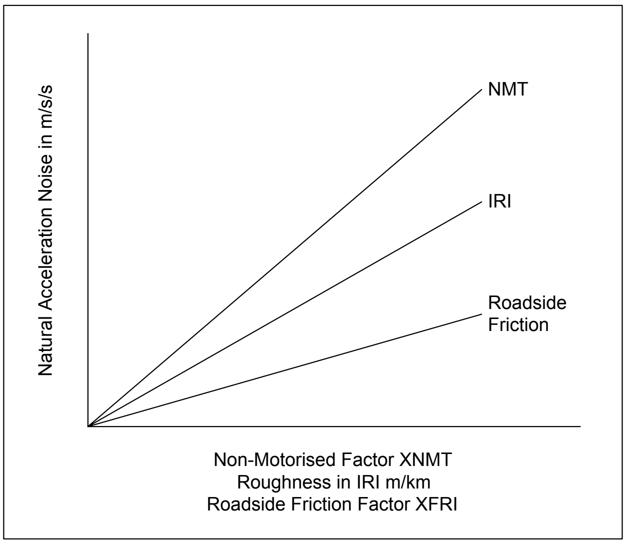


Figure B3.13: Natural Noise Modelling in HDM-4

NOISE DUE TO ROADSIDE FRICTION

Roadside friction can manifest itself as noise (σasf) in several ways:

- Through inadequate pavement width;
- Through roadside developments and driveways; or,
- Through accidents or incidents of interest in the opposing lane.

NOISE DUE TO NON-MOTORISED TRAFFIC

Pedestrians and non-motorised traffic cause traffic noise (σanmt) when they are either adjacent to the carriageway or encroach on the carriageway. The former results in a limited impact on traffic whereas encroachment can lead to significant speed fluctuations.

NOISE DUE TO ROUGHNESS

The noise due to roughness (σairi) arises as vehicles are forced to slow down due to sections with high roughness or to manoeuvre their vehicles around broken or badly patched sections of pavement.

B3.6.4 Quantifying the Acceleration Noise

INTRODUCTION

In order to quantify the acceleration noise model parameters for HDM-4, experiments were undertaken by NDLI (1995) under different operating conditions. By measuring traffic noise on a high standard motorway, at low traffic flows the natural noise due to drivers could be directly quantified. Similarly, by collecting data under stop-start conditions, information on the traffic-induced acceleration noise were obtained.

NDLI (1995) tested three cars; one 4WD, one mini-van and two medium trucks (one laden and one unladen). Multiple runs were made with the same vehicle using different drivers. A total of 10 different drivers were tested in the study. Figure B3.14 is an example of a speed profile from NDLI (1995).

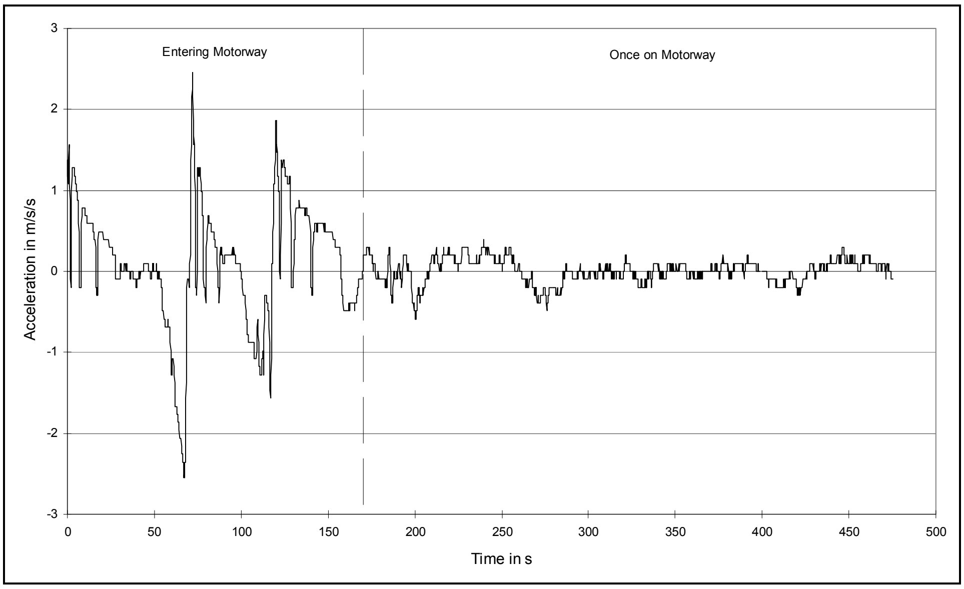


Figure B3.14: Example of Acceleration Data on Motorway - Pajero 4WD

The natural noise was deemed to be that observed along the motorway under free flow conditions. This was quantified for all the vehicles and drivers. Typically, the natural noise was around 0.1 m/s^2, with vehicles having higher power-to-weight ratios resulting in higher values than those with a lower power-to-weight ratios.

The natural traffic noise is defined as a function of the noise due to the driver, alignment, roadside friction, non-motorised traffic and roughness (see Section B3.6.3). The motorway standards were such that the only components present were the driver noise and the traffic noise.

As the traffic noise is a function of the volume-to-capacity ratio, and this is difficult to accurately quantify except at either extreme (*ie* either at free flow or capacity), the traffic noise could not be accurately related to different levels of congestion. The data collection therefore focused on collecting data at jam flow when the volume-to-capacity ratio equals 1.

ACCELERATION NOISE DISTRIBUTION

As described in Chapter B4, the acceleration noise model is applied to calculate the effects of speed cycles on fuel and tyre consumption using a Monte-Carlo simulation. This approach is predicated on the data being normally distributed so it is important to understand the underlying distribution of the acceleration noise data.

Bester (1981) found that although the acceleration noise distribution was reported in the literature to follow a normal distribution, his experimental data were not normally distributed. The data collected by NDLI (1995) also showed that the acceleration noise does not follow a normal distribution, with the distribution skewed to the left (*ie* the deceleration tail is longer than that of the acceleration). This finding is not completely unexpected, as the deceleration tail is limited by driver behaviour whereas the maximum acceleration rate is a function of the vehicle power-to-weight ratio. This was confirmed by Hine and Pangihutan (1998) who found in Indonesia that the maximum deceleration exceeded the maximum acceleration.

A typical plot of acceleration noise is shown in Figure B3.15, with the observed distribution overlain by the normal distribution with equal mean and standard deviation (NDLI, 1995). NDLI (1995) found that although the acceleration noise distributions failed tests for normality, the data were approximately normally distributed over much of the range, with most of the error occurring within 0.05 m/s^2 of the mean value or at the tails. Since the majority of the error was at such a low acceleration rate, and the error was within the accuracy of the measurement system, it was considered by NDLI (1995) that the assumption of the acceleration noise being normally distributed would not significantly influence the analysis results. As shown in Chapter B4, this assumption does not materially affect the final results since the simulated fuel consumption was found to be similar to the observed fuel consumption.

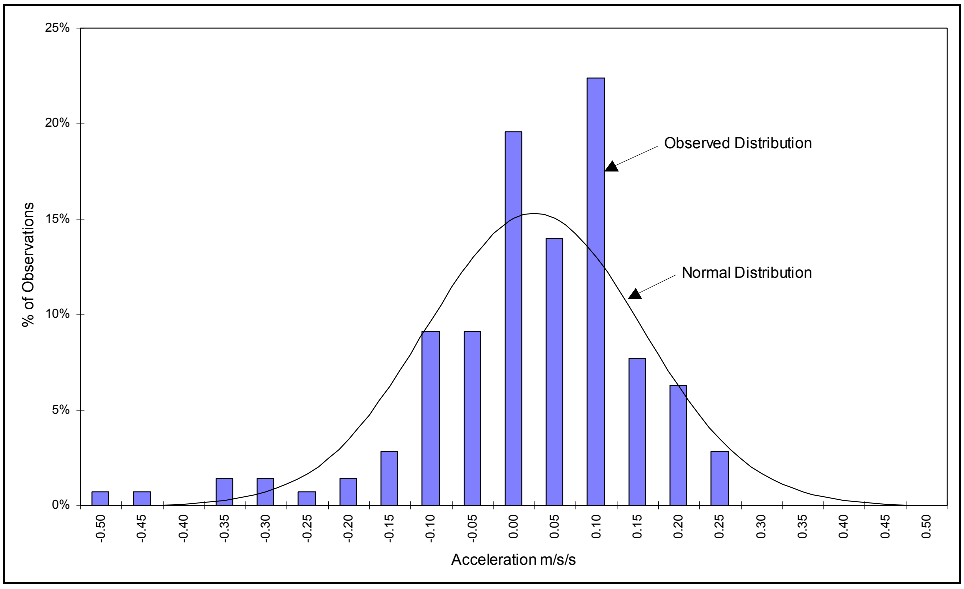


Figure B3.15: Distribution of Acceleration Noise Under Free Flow Conditions for Passenger Car

NATURAL AND TRAFFIC ACCELERATION NOISES

Table B3.4 gives the values for the natural, traffic and acceleration noise for each of the vehicles tested by NDLI (1995), averaged for the different drivers. From Table B3.4 it can be seen that the maximum acceleration noise is approximately 0.6 m/s^2 for all the tested vehicles, and this value has been adopted for HDM-4. Greenwood (1999) confirmed this value in his study of passenger cars in Thailand.

Table B3.4: Measured Natural and Traffic Acceleration Noises by Vehicle Type

Vehicle Type	Natural Noise (m/s^2)	Maximum Traffic Noise [1] (m/s^2)	Maximum Acceleration Noise (m/s^2)
Passenger Car	0.11	0.57	0.58
4WD Diesel	0.15	0.61	0.63
Van	0.11	0.59	0.60[2]
MCV - Unladen	0.21	0.48	0.52
MCV - Laden	0.02	0.57	0.57

Notes: 1/ The maximum traffic noise was not directly measured and was calculated using Equation B3.10.
2/ This value was not measured and was assumed to equal 0.60.
Source: *NDLI (1995)*

The natural noise appears to be related to the power-to-weight ratio of the vehicle, with a higher ratio leading to a higher natural noise level. Neither of these observations are surprising given that at jam density the entire traffic stream has similar start-stop patterns, while under free flow conditions those vehicles that are under powered (low power-to-weight ratio) have very little available power to accelerate. Hine and Pangihutan (1998) found a strong relationship between acceleration noise and truck loads in their study in Indonesia. However, since data are lacking for establishing this relationship, a constant value of 0.10 m/s^2 was adopted for the natural noise in HDM-4.

B3.6.5 Validation of Acceleration Noise Model

There have been two studies conducted to calibrate/validate the HDM-4 acceleration noise model proposed by NDLI (1995): Hine and Pangihutan (1998) in Indonesia and Greenwood (1999) in Thailand.

The Indonesian study collected data on speed profiles in 30 s intervals for a light passenger vehicle and a medium truck on a 17 km congested road in Java. The acceleration data for both vehicles was much more normally distributed than in NDLI (1995), but this was likely due to the coarser recording intervals (0.35 m/s^2 vs 0.05 m/s^2). The maximum levels observed were four times those observed by NDLI (1995) which contrasts with Greenwood (1999) who found similar levels to NDLI (1995).

It was found that the acceleration noise was a function of speed and, for trucks, gross vehicle weight: the heavier the weight, the lower the acceleration noise. A "relatively poor fit could be found between observed acceleration noise and traffic noise", but this was ascribed in a large extent to difficulties in predicting speeds from the volume-to-capacity ratio.

In Thailand the focus was on calibrating the HDM-4 model so measurements were made under low flow and extremely congested conditions. Additional measurements were made to confirm the basic HDM-4 relationship.

Figure B3.16 is an example of the acceleration noise vs the mean speed for a two-lane highway (Greenwood, 1999). It will be noted that as the speeds decrease due to traffic interactions there is an increase in the acceleration noise—confirming the approach in the HDM-4 model.

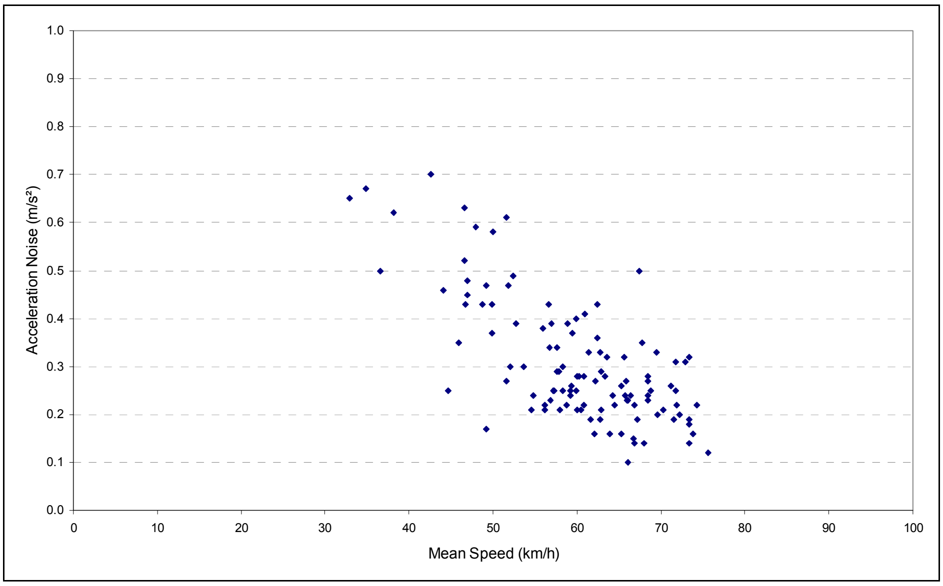


Figure B3.16: Acceleration Noise vs Speed From Thailand

Table B3.5 shows the recommended values for acceleration noise by road class from Thailand (Greenwood, 1999). These indicate a decrease in the natural noise with increasing road standard and that two-lane highways have higher total acceleration noises than multi-lane roads.

Table B3.5: Acceleration Noise Values by Road Class

Road Type	Natural Acceleration Noise (m/s²)	Maximum Traffic Acceleration Noise (m/s²)	Maximum Total Acceleration Noise (m/s²)
2 Lane Undivided	0.20	0.62	0.65
4 Lane Undivided	0.20	0.57	0.60
4 Lane Divided	0.20	0.57	0.60
6 Lane Divided	0.15	0.58	0.60
6 Lane Highway/Motorway	0.15	0.58	0.60

Source: Greenwood (1999)

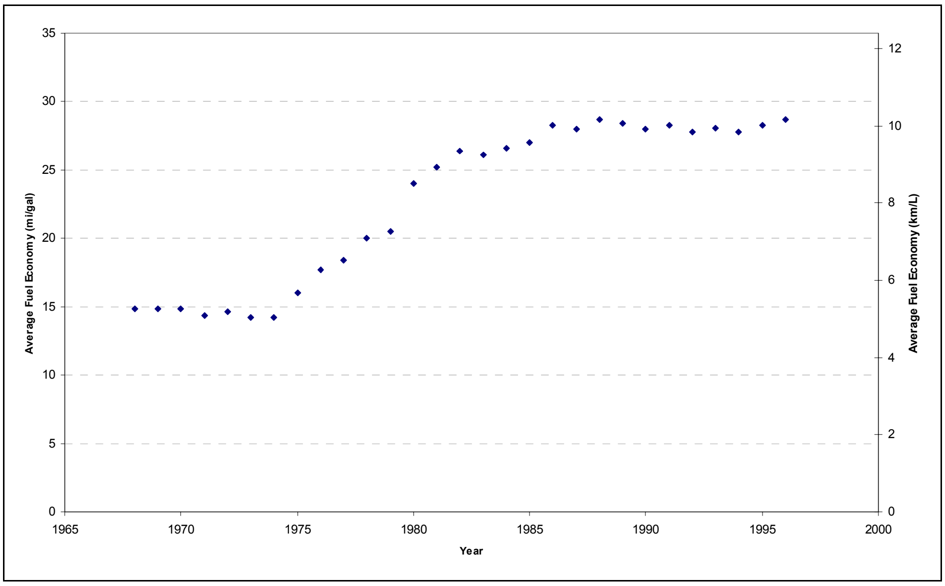
Greenwood (1999) also tested the effects of vehicle type and drivers on the acceleration noise. There were marked differences between drivers, but only limited differences between the 1.6 vs 2.0 litre passenger cars. The latter was ascribed to the relatively small differences in the power-to-weight ratio between the vehicles. This confirms the need to use several different drivers in any calibration exercises.

B4 Fuel Consumption

B4.1 Introduction

Fuel consumption is a significant component of VOC, typically accounting for between 20 and 40 per cent of the total VOC (HTC, 1999b). It is influenced by traffic congestion, road condition and alignment, vehicle characteristics and driving style, so it is sensitive to virtually any investment decisions on the road network.

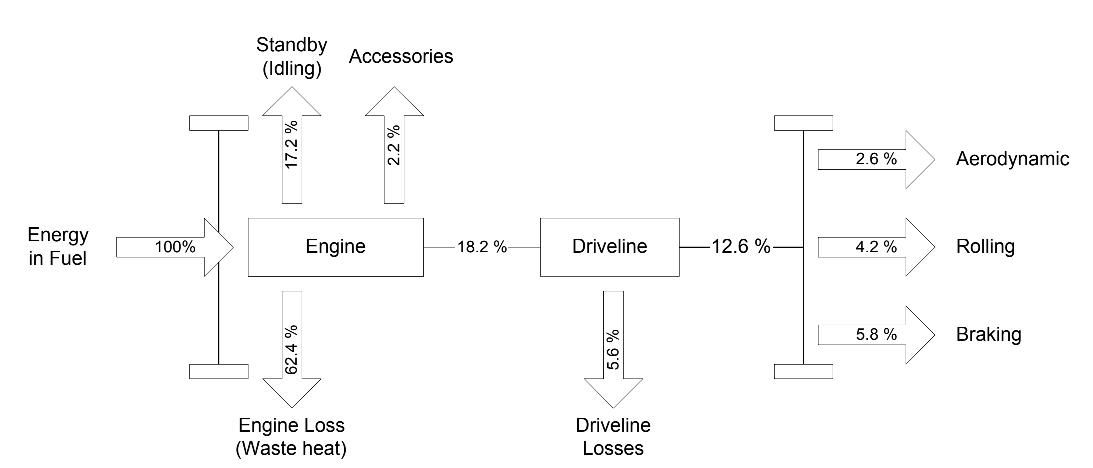
As is illustrated in Figure B4.1, there has been a significant change in the average fuel economy of passenger cars over the past 30 years, with an approximate improvement of nearly 200 per cent. It is of interest that the majority of change occurred over a relatively short 10 year period (during the oil crisis years of the mid 1970s to the mid 1980s). Heaventich, *et al*. (1991) indicate that although the average fuel economy has undergone dramatic improvement, the economy of the lowest five per cent of passenger cars has undergone a relatively minor improvement over this same period.



Source: Amann, 1997

Figure B4.1: Change in Fuel Economy Since 1968

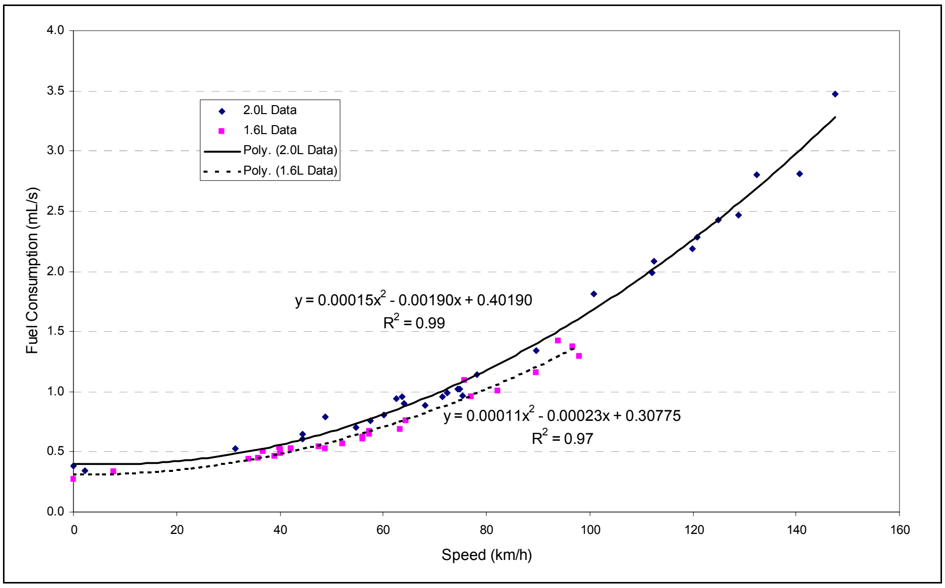
The energy in the fuel consumed by a vehicle is utilised to overcome a variety of demands is illustrated in Figure B4.2. Of the total fuel consumed, over 60 per cent of the total energy is expended as heat through the coolant system and the exhaust. Only 18 per cent of the total energy in the fuel is available to propel the vehicle along the road under typical urban driving conditions. For highway driving conditions, this percentage is increased to over 25 per cent, primarily by reducing the standby component.



Source: Adapted from Amann (1997)

Figure B4.2: Energy Train for a Typical Urban Car

Fuel consumption is often related to vehicle speed in literature. Figure B4.3 illustrates the typical relationship between mean vehicle speed and fuel consumed. These data were collected from steady speed runs in Thailand for 1.6 litre and 2.0 litre passenger cars (Greenwood, 1999). As can be seen from the graph, increasing vehicle speed from 70 to 140 km/h is a doubling in speed, but yields a three-fold increase in fuel consumed.



Source: Greenwood (1999)

Figure B4.3: Fuel Consumption versus Vehicle Speed

This chapter describes the fuel consumption model adopted for HDM-4. It commences with a summary of the various types of fuel models available from the literature and presents the modelling approach and fuel model adopted for HDM-4. The implementation of this fuel model for predicting steady state fuel consumption is given. The chapter closes with a discussion of the additional fuel consumption due to traffic congestion

In the discussion which follows the fuel consumption is expressed in two ways: mL/s or L/1000 km. Figure B4.4 shows the predictions for the default HDM-4 articulated truck with both sets of units. The **instantaneous** fuel consumption in mL/s is converted to the **specific** fuel consumption in L/1000 km by dividing with the vehicle velocity. The instantaneous fuel consumption is used internally in HDM-4 but is applied as the specific fuel consumption.

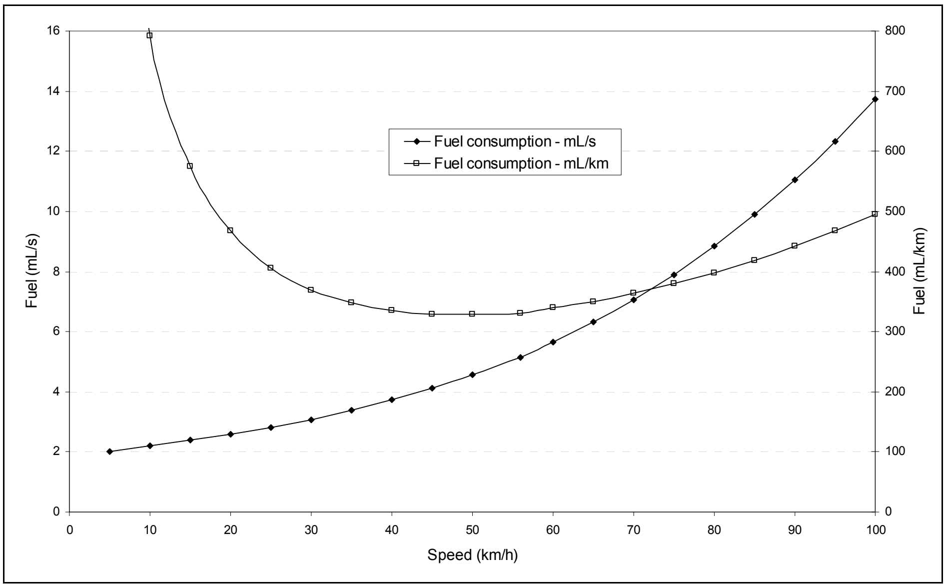


Figure B4.4: Comparision of Different Fuel Consumption Units

B4.2 Factors Influencing Fuel Consumption

Virtually any change to the road, the traffic conditions, or the vehicle, will yield a change in the fuel consumption rate of the vehicle. Additionally, parameters which are not easily modelled–such as driving style–also have a significant influence.

Chapter B1 dealt with the forces opposing the motion of vehicles. A change in any of the road attributes that effect the forces opposing motion, will in turn impact on the fuel consumption. The primary attributes relating to the road are:

- Gradient;
- Roughness;
- Texture; and,
- Curvature.

Controlled experiments with instrumented vehicles can be done to establish the magnitude of these effects. Sime, *et al.*, (2000) describes an unusually accurate study wherein four driverless trucks operating on a test track before and after a pavement rehabilitation had their fuel monitored every 0.5 s for an eight week period just before and seven weeks after the rehabilitation. Since the vehicle's gross weight, speed and aerodynamic profiles were fixes and the vehicles were very well maintained, any changes in fuel consumption were due to changes in roughness. An average reduction of 4.5 per cent was observed, and this was in spite of the average wind speed increasing from 5.6 to 7.9 km/h. Unfortunately, the roughness was not measured so the reduction cannot be related to the change in roughness.

Traffic conditions impact on the fuel consumption by both yielding different mean operating speeds and a greater level of fluctuation in speeds as discussed in Chapter B3. As the mean speed is related in part to the road parameters, changes in these can have the dual impact of changing both the forces opposing motion and the mean speed of the vehicle–both of which influence the fuel consumption.

Vehicle parameters that impact on the forces opposing motion, all impact on the fuel consumed. Factors such as vehicle mass, frontal area and design, engine design and tyres all play a significant role in the overall fuel consumed. Other factors, such as the style of driving also have a profound influence on the fuel consumption rate.

For example, Elder (1983) found that at 80 km/h an eight per cent decrease in tyre pressure led to a 2.1 per cent increase in fuel consumption. In India (Kadiyali, *et al*., 1981) it was found that at 80 km/h radial tyres used 1.5 per cent less fuel than bias ply tyres. The ambient conditions also play an important role, with the impact of wind strength and direction impacting on the forces against the vehicle. Armstrong (1983) determined that running engines below their operating temperatures consumed 40-60 per cent excess fuel.

The maintenance of the vehicle, consisting of the tuning of the engine, condition of aerodynamic aids and the shape of loads on the back of trucks, although mainly beyond the realms of current modelling, all have a significant impact on the fuel consumed. However, when comparing one road investment alternative to another, these factors tend to be constant so are of lesser importance from a modelling perspective.

Clearly from the above, there are numerous factors impacting on the fuel consumption of vehicles. The remained of this chapter discusses various models that have been developed to predict the fuel consumption.

B4.3 Review of Fuel Consumption Models

B4.3.1 Introduction

Research has been conducted into the fuel consumption of motor vehicles almost since they were first invented. In recent years a number of studies have been undertaken with the objective of quantifying the effects of speed, road geometry and surface condition on fuel consumption. Initially, researchers used coarse empirical data (*eg* de Weille, 1966), this was then superseded by experimental studies that related the fuel consumption to specific operating conditions and modelled it using an empirical approach. More recently, the fuel consumption has been modelled using **mechanistic** principles that relate the consumption to the forces opposing motion.

This section briefly reviews some of the major fuel consumption models developed. The models are grouped into empirical and mechanistic based models.

B4.3.2 Empirical Models

The early empirical models related fuel consumption principally to vehicle speed. A number of studies found that the relationship between the specific fuel consumption (in L/1000 km) and vehicle speed was U-shaped. There are relatively high fuel consumption rates at both low and high speeds with the minimum fuel consumption arising at an “optimum” speed, generally around 40 - 60 km/h.

The reason behind this U-shape can be simply explained by considering the two extremes. At high speeds, the aerodynamic forces, which are related to the square of velocity, become dominant requiring large quantities of fuel to be consumed. While at low speeds which equate to low tractive power requirements, the idle fuel consumption that powers the engine drag and accessories dominates.

The common empirical formulation for fuel consumption is given by Equation B4.1:

$$FC = a0 + \frac{a1}{S} + a2\ S^2 + a3\ RISE + a4\ FALL + a5\ IRI \qquad ...(B4.1)$$

where FC is the fuel consumption in L/1000 km
S is the vehicle speed in km/h
IRI is the roughness in IRI m/km
RISE is the rise of the road in m/km
FALL is the fall of the road in m/km
a0 to a5 are constants

The coefficients established for the above model from studies in the Caribbean, India and Kenya for different vehicles is given in Table B4.1.

Figure B4.5 is an example of the effect of speed on the predictions for passenger cars using these coefficients. It shows that there are marked differences in the speed effects for the different vehicle types, not only between countries but also for different vehicles in the same country. As will be shown in the section describing mechanistic modelling, this is a reflection of the physical properties of the different vehicles.

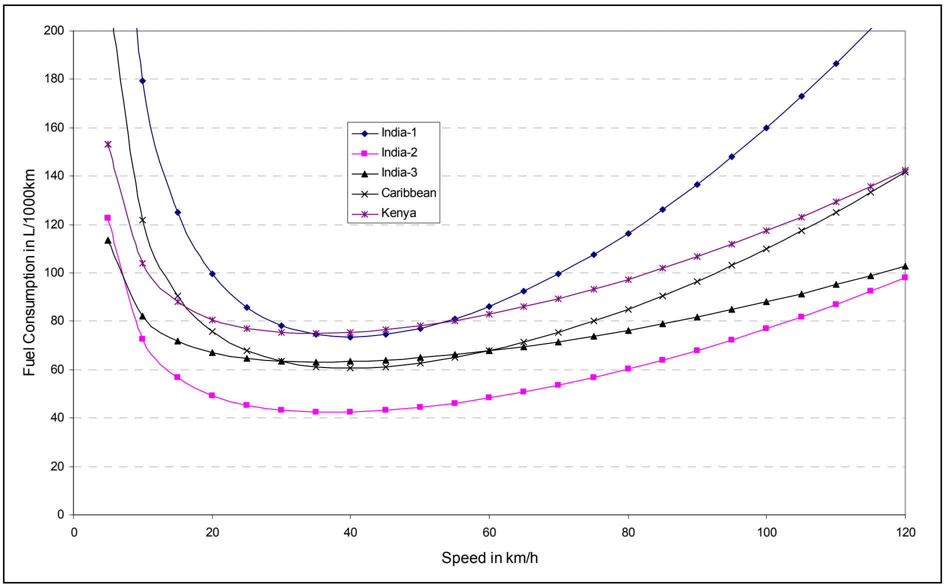


Figure B4.5: Effect of Speed on Passenger Car Fuel Consumption

B4.3.3 Mechanistic Models

Mechanistic models predict that the fuel consumption of a vehicle is proportional to the forces acting on the vehicle. Thus, by quantifying the magnitude of the forces opposing motion one can establish the fuel consumption. Mechanistic models are an improvement over empirical models since they can allow for changes in the vehicle characteristics and are inherently more flexible when trying to apply the models to different conditions. Because of their numerous advantages over empirical models, mechanistic models were adopted for use in HDM-4.

Table B4.1: Coefficients Estimated for Empirical Fuel Consumption Model

Vehicle	Country	Fuel Model Coefficients						Other Variables	Source
		a0	a1	a2	a3	a4	a5		
Passenger Cars	India	10.3	1676	0.0133	1.39	-1.03	0.43		Chesher & Harrison (1987)
	India	21.85	504	0.0050	1.07	-0.37	0.47		IRC (1993)
	India	49.8	319	0.0035	0.94	-0.68	1.39		Chesher & Harrison (1987)
	Caribbean	24.3	969	0.0076	1.33	-0.63		+ 0.00286 $FALL^2$	Chesher & Harrison (1987)
	Kenya	53.4	499	0.0059	1.59	-0.85			Chesher & Harrison (1987)
Light Commercials	India	30.8	2258	0.0242	1.28	-0.56	0.86		Chesher & Harrison (1987)
	India	21.3	1615	0.0245	5.38	-0.83	1.09		IRC (1993)
	Caribbean	72.2	949	0.0048	2.34	-1.18		+ 0.0057 $FALL^2$ + 1.12 (GVW - 2.11) RISE	Chesher & Harrison (1987)
	Kenya	74.7	1151	0.0131	2.91	-1.28			Chesher & Harrison (1987)
Heavy Bus	India	33.0	3905	0.0207	3.33	-1.78	0.86		IRC (1993)
	India	-12.4	3940	0.0581	0.79		2.00	+ 0.0061 CKM	Chesher & Harrison (1987)
Truck	India	44.1	3905	0.0207	3.33	-1.78	0.86		IRC (1993)
	India	141.0	2696	0.0517	17.75	-5.40	2.50		IRC (1993)
	India	85.1	3905	0.0207	3.33	-1.78	0.86	- 6.24 PW	Chesher & Harrison (1987)
	India	266.5	2517	0.0362	4.27	-2.74	4.72	-6.26 PW	Chesher & Harrison (1987)
	India	71.70	5670	0.0787	1.43			- 9.20 PW - 3.98 WIDTH	Chesher & Harrison (1987)
	Caribbean	29.2	2219	0.0203	5.93	-2.60		+ 0.85 (GVW - 7.0) RISE + 0.013 $FALL^2$	Chesher & Harrison (1987)
	Kenya	105.4	903	0.0143	4.36	-1.83		-3.22 PW	Chesher & Harrison (1987)

One of the more comprehensive mechanistic fuel consumption models available is the ARFCOM model (Biggs, 1988) and its approach is summarised in Figure B4.6. This shows how the total power requirements are based on the tractive forces, the power required to run accessories, and internal engine friction. The fuel consumption is then taken as proportional to the total power requirements.

There have been three major studies that have developed mechanistic models: the HDM-III study, research in South Africa, and research in Australia. As described by Greenwood and Bennett (1995), all have similarities in that they use the magnitude of the forces opposing motion as the basis for the predictions.

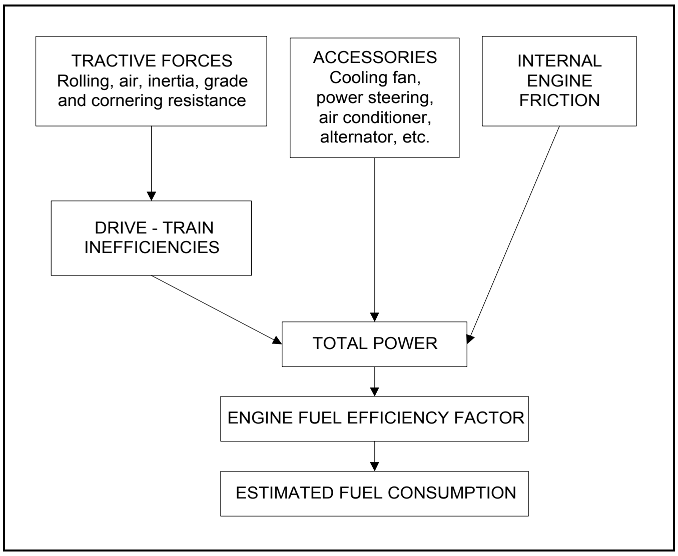


Figure B4.6: ARFCOM Approach to Modelling Fuel Consumption

With both the South African and Australian models fuel experiments or other data were used to calculate an efficiency factor ξ. The fuel consumption was the product of the total power requirements and this factor. AFRCOM offers improved predictions over the South African model through explicitly considering the fuel required to maintain engine operation.

The HDM-III model in many respects was a hybrid mechanistic model. Although it used the forces opposing motion as the basis for its calculations, it resorted to the use of a series of regression parameters to convert the forces to the fuel consumption. Unlike the South African model, it explicitly considered engine speed, although a constant value was adopted.

After reviewing the three groups of mechanistic models, their appropriateness for modelling various tasks were established. These tasks were:

- **Forces opposing motion:** Can the model predict the fuel consumption using different equations for predicting the magnitude of the forces opposing motion;
- **Internal vehicle forces:** Does the model explicitly consider the fuel required to overcome the internal vehicle forces such as engine drag;
- **Engine speed**: Does the model explicitly consider engine speed effects;
- **Appropriate for acceleration fuel consumption:** Can the model be used to predict the fuel consumption during acceleration and deceleration; and
- **Transferable to different vehicles**: How suitable is the model for being applied to different vehicle technology.

The results of this assessment is given in Table B4.2. It can be seen that the model which offered the greatest potential for meeting the varied requirements of HDM-4 was ARFCOM. Accordingly, ARFCOM was selected as the basis for the HDM-4 fuel consumption model[1]. The sections which follow discuss the implementation of ARFCOM for HDM-4.

Table B4.2: Assessment of Mechanistic Models

Fuel Model	Forces Opposing Motion	Internal Vehicle Forces	Engine Speed Effects	Appropriate for Acceleration Fuel	Transferable to Different Vehicles
HDM-III	•		•		
South African	•			•	•
ARFCOM	•	•	•	•	•

B4.4 Modelling Fuel Consumption in HDM-4

B4.4.1 Basic Model

As described above, it was decided to adopt the ARFCOM fuel model as the basis for the fuel model for HDM-4. This model uses mechanistic principles for calculating the fuel use and offered the greatest flexibility and scope for fuel modelling, without going into extremely detailed models such as those based on engine maps.

ARFCOM predicts that the fuel consumption is proportional to the vehicle power requirements (Biggs, 1988):

$$IFC = f\,(Ptr,\ Peng + Paccs) \qquad \text{...(B4.2)}$$

where
- IFC is the instantaneous fuel consumption in mL/s
- Ptr is the power required to overcome tractive forces in kW
- Paccs is the power required to power engine accessories in kW
- Peng is the power required to overcome internal engine drag in kW

All tractive forces are influenced by the drive-train efficiency factor (edt). When the tractive power is positive, this results in additional power needing to be supplied to overcome the forces. Under deceleration or on negative grades the tractive power can be negative in which case kinetic and potential energy is used to overcome the engine and accessory power requirements. In this case the power transfer is reduced by the drivetrain efficiency. This leads to the following relationships for predicting the total power (Biggs, 1988):

$$Ptot = \frac{Ptr}{edt} + Paccs + Peng \qquad Ptr \geq 0 \qquad \text{...(B4.3)}$$

$$Ptot = edt\ Ptr + Paccs + Peng \qquad Ptr < 0 \qquad \text{...(B4.4)}$$

where Ptot is the total power requirements in kW

Thus, substituting Ptot into Equation B4.2, the fuel consumption under all values of Ptr is given by:

$$IFC = \max(MinFuel,\ \xi\ Ptot) \qquad \text{...(B4.5)}$$

[1] The ISOHDM Study would like to acknowledge the contribution of ARRB Transport Research in making available the ARFCOM source code.

where MinFuel is the minimum fuel consumption in mL/s
ξ is the fuel-to-power efficiency factor in mL/kW/s

The minimum fuel consumption has traditionally been set equal to the idle fuel consumption level. Substituting the idle fuel consumption into Equation B4.5 yields Equation B4.6.

$$IFC = \max(\alpha, \xi \, Ptot) \qquad \text{...(B4.6)}$$

where α is the idle fuel consumption in mL/s

Modern engines have the ability to completely shut off the fuel when the tractive power of the vehicle is supplying sufficient power to overcome the accessories and engine drag components. The effect of utilising a zero or idle fuel rate for the minimum fuel consumption value, only has an impact when considering roads with high gradients, or high traffic congestion levels wherein the number of hard braking manoeuvres is high. Further discussions on this are made in the following section.

B4.4.2 Minimum Fuel Consumption

Under general driving conditions the fuel consumption continually varies owing to minor changes in vehicle speed or road condition. Under high rates of deceleration, the fuel consumption takes on an absolute minimum level dependent on the technology of the engine. Computer controlled fuel injected engines, and some of the more recent carburettored engines, have the ability to restrict the fuel consumption to a level below the idle fuel consumption rate (often to zero fuel) during periods when the tractive power is sufficient to power the accessories and the engine drag.

Figure B4.7 illustrates the use of the idle and minimum fuel consumption levels during a period of driving in congested conditions for a vehicle that can run at zero fuel consumption (Greenwood, 1999). Under periods of heavy deceleration, the fuel consumption drops to zero. When the vehicle is stationary and no power is generated by the forces opposing motion to overcome the accessories power and engine drag, the fuel consumption steadies at the idle fuel consumption rate. As can be seen from Figure B4.7, there are very few periods of zero fuel consumption, such that the assumption that the minimum fuel consumption equals the idle fuel consumption will not introduce large errors in the modelling process.

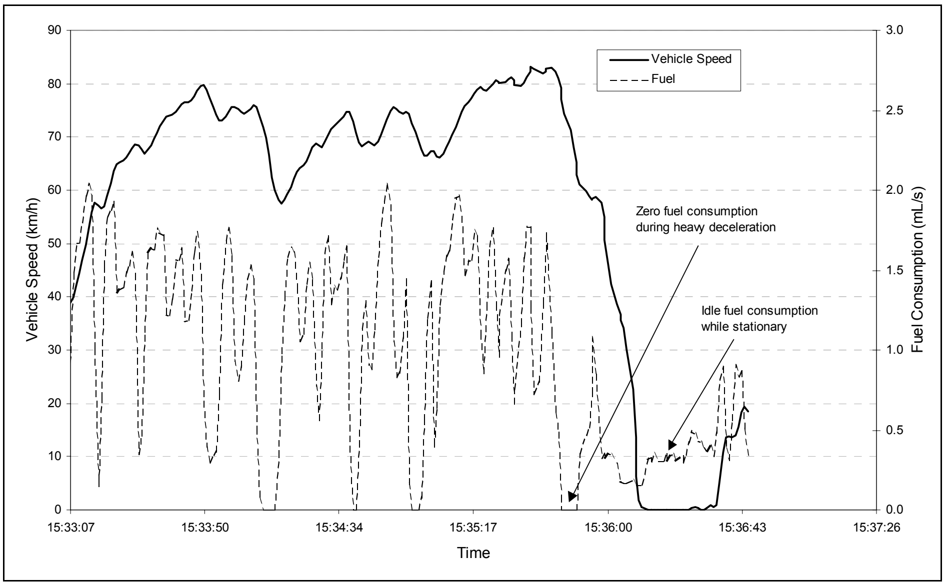


Figure B4.7: Fuel Consumption during Congested Drive Cycle

The minimum fuel consumption value takes account of this ability by enabling values to be entered that reflect the technology. For vehicles that do not have the ability to completely shut off the fuel, the minimum fuel consumption equals the idle fuel consumption. For those vehicles that can shut off the fuel, this value equals 0.

Box B4.1 Software Status

Within the HDM-4 software the minimum fuel consumption is hard coded as equalling the idle fuel consumption. Given that the HDM-4 software assumes steady speed conditions, this is not an issue except on steep downgrades. The current HDM Tools program that simulates traffic interactions (ACCFUEL.EXE) runs from a file exported from the HDM-4 database. As the minimum fuel consumption is not included in this file, the program also assumes that the minimum fuel equals the idle fuel consumption. A future update to the software will correct this situation.

B4.4.3 Idle Fuel Consumption

Bowyer, *et al.* (1986) performed a regression analysis to calculate the idle fuel rate as a function of engine capacity. Their resulting equation ($R^2 = 0.67$, SE = 0.086 mL/s) was:

$$\alpha = 0.220\ DENG - 0.0193\ DENG^2 \qquad ...(B4.7)$$

where DENG is the engine capacity in litres

The idle fuel consumption was tested by Greenwood (1999) for two cars in Thailand. The engine capacity, observed and predicted idle fuel consumption rates are given in Table B4.3. As can be seen in the table, the above predictive equation of Bowyer, *et al.* (1986) appears to yield values of the correct magnitude (within 10 per cent) given the fact that it does not explicitly account for accessory loading, tuning or other items.

Table B4.3: Observed and Predicted Idle Fuel Consumption Rates

Vehicle / Engine Capacity	Observed Idle Fuel Consumption	Predicted Idle Fuel Consumption	Observed / Predicted
1.6 L Toyota Corolla	0.27 mL/s	0.30 mL/s	0.90
2.0 L Toyota Corolla	0.38 mL/s	0.36 mL/s	1.06

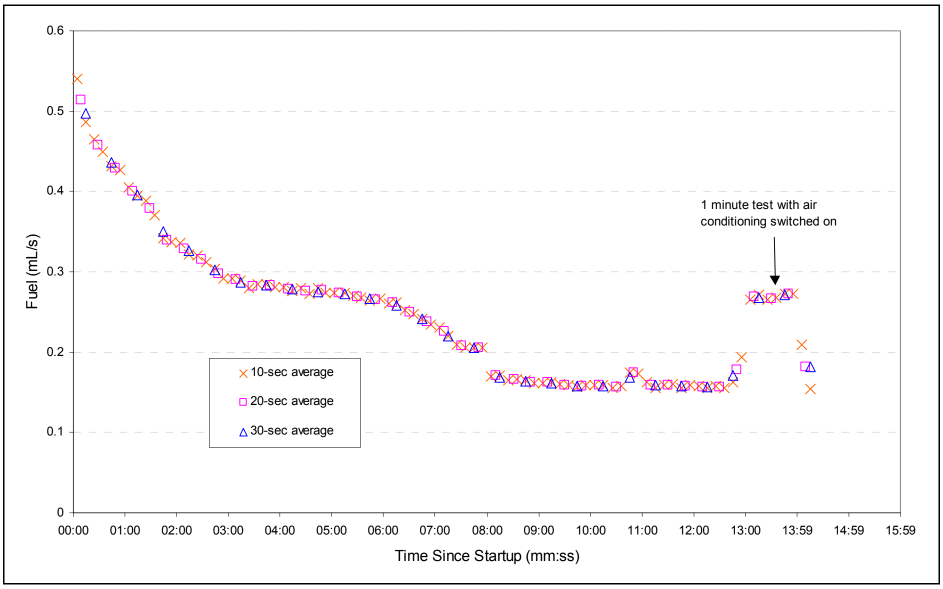
On the basis of the results in Table B4.3 it is concluded that the work of Bowyer, *et al.* (1986) is still applicable to the modern car fleet. Table B4.4 lists the recommended default values of α for inclusion in HDM-4.

When a vehicle is idling the tractive forces are equal to zero, thereby enabling Equations B4.3 and B4.5 to be simplified. Rearranging and solving for the idle fuel consumption yields:

$$\alpha = \xi\ (Peng + Paccs). \qquad ...(B4.8)$$

Although Equation B4.8 in theory enables the prediction of the idle fuel consumption, the reality is that of the parameters in the above equation, the idle fuel consumption is likely to be the only known. Therefore Equation B4.8 is more likely to be used to estimate the engine and accessory drag, or the fuel efficiency, than it is to be used to estimate the idle fuel consumption rate.

Greenwood (1999) ran several cars from cold start to observe the change in idling fuel consumption during the warm-up period of the engine. The result for a mid-size passenger car (1.6L Toyota Corollaa) is shown in Figure B4.8 with the effect of switching on the air conditioning unit also clearly evident from the figure.



Source: Greenwood (1999)

Figure B4.8: Change in Idling Fuel Consumption with Time from Start-Up for a 1.6L Toyota Corona

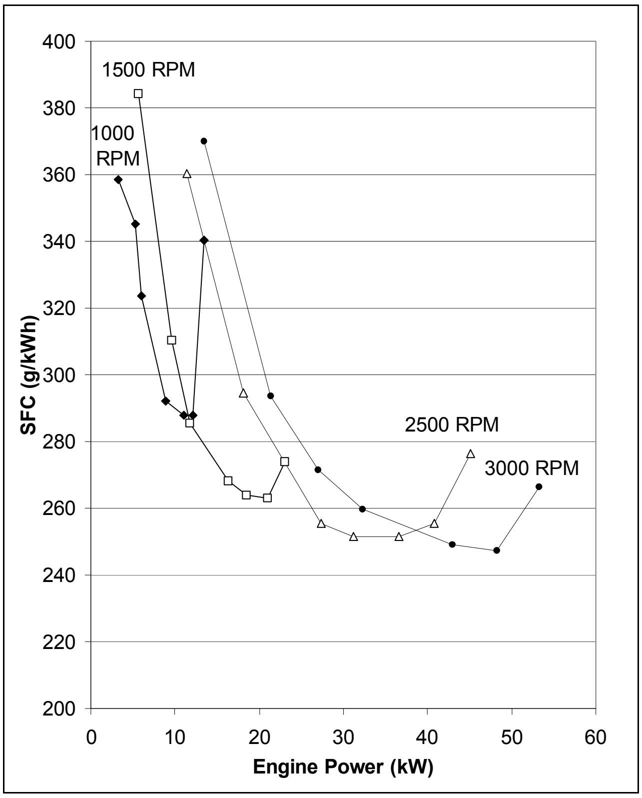
Given the wide range of the idle fuel consumption from a single vehicle, and the previously stated fact that 17.2 per cent of the total energy is utilised in idle mode (refer Figure B4.2), the decision on what value to adopt can have a significant impact on the estimated running costs of a vehicle for urban conditions. For rural conditions, where a much smaller proportion of time is spent idling, the impact of the idle fuel consumption is greatly reduced.

B4.4.4 Engine Efficiency

When dealing with engine efficiency (and other factors such as power output) it is necessary to distinguish between the two types of data presented. Broadly the two types are:

- **Nominal engine recordings:** These exclude the impact of accessories, engine drag and the likes; and,
- **Brake readings:** These include for the accessories, engine drag and often transmission losses.

With the second of these two methods it is not necessary to model as discrete components the losses for engine and accessories power as these are implicitly included in the fuel efficiency factors. When using the brake readings, engine efficiency rapidly rises at low power levels. This is because when idling, there is no output power to the wheels (*ie* no brake power) yet fuel is being consumed. An example of such an engine efficiency map from brake readings is illustrated in Figure B4.9.



Source: Autotech (1993)

Figure B4.9: Typical Engine Map Using Brake Measurements

For HDM-4 it was decided to base the fuel modelling around the ARFCOM model (Biggs, 1988) and therefore the former of these two methods are utilised, where the efficiency factor and power usage specifically excludes accessories and engine drag.

Biggs (1988) found that engine efficiency decreases at high levels of output power, resulting in an increase in the fuel efficiency factor ξ. Biggs (1988) used the following relationship to account for this decreased efficiency.

$$\xi = \xi b \ (1 + ehp \ (Ptot\text{-}Peng)/Prat) \qquad ...(B4.9)$$

where ξb is the base engine efficiency in mL/kW/s

ehp is the proportionate decrease in efficiency at high output power

Prat is the rated engine power in kW

Table B4.5 lists the engine efficiency factors given by Biggs (1988). Using the recommended HDM-4 default values for Prat in Table B4.4, the values for ξb and ehp in Table B4.4 were calculated for the 16 vehicle types.

Table B4.4: HDM-4 Default Fuel Model Parameters

Vehicle Number	Type	Engine Speed Model Parameters a0 RPM	a1 RPM/(km/h)	a2 RPM/(km/h)²	a3 RPM/(km/h)³	RPMIdle rev/min	α mL/s	ξb mL/kW/s	ehp	Prat kW	edt	Peng_a0	PctPeng %
1	Motorcycle	-162	298.86	-4.6723	-0.0026	800	0.12	0.067	0.25	15	0.95	0.20	80
2	Small Car	1910	-12.311	0.2228	-0.0003	800	0.25	0.067	0.25	60	0.90	0.20	80
3	Medium Car	1910	-12.311	0.2228	-0.0003	800	0.36	0.067	0.25	70	0.90	0.20	80
4	Large Car	1910	-12.311	0.2228	-0.0003	800	0.48	0.067	0.25	90	0.90	0.20	80
5	Light Delivery Vehicle	1910	-12.311	0.2228	-0.0003	800	0.48	0.067	0.25	60	0.90	0.20	80
6	Light Goods Vehicle	2035	-20.036	0.3560	-0.0009	800	0.37	0.067	0.25	55	0.90	0.20	80
7	Four Wheel Drive	2035	-20.036	0.3560	-0.0009	800	0.48	0.057	0.10	60	0.90	0.20	80
8	Light Truck	2035	-20.036	0.3560	-0.0009	500	0.37	0.057	0.10	75	0.86	0.20	80
9	Medium Truck	1926	-32.352	0.7403	-0.0027	500	0.50	0.057	0.10	100	0.86	0.20	80
10	Heavy Truck	1905	-12.988	0.2494	-0.0004	500	0.70	0.056	0.10	280	0.86	0.20	80
11	Articulated Truck	1900	-10.178	0.1521	0.00004	500	0.70	0.055	0.10	300	0.86	0.20	80
12	Mini Bus	1910	-12.311	0.2228	-0.0003	800	0.48	0.067	0.25	60	0.90	0.20	80
13	Light Bus	2035	-20.036	0.3560	-0.0009	500	0.37	0.057	0.10	75	0.86	0.20	80
14	Medium Bus	1926	-32.352	0.7403	-0.0027	500	0.50	0.057	0.10	100	0.86	0.20	80
15	Heavy Bus	1926	-32.352	0.7403	-0.0027	500	0.60	0.057	0.10	120	0.86	0.20	80
16	Coach	1926	-32.352	0.7403	-0.0027	500	0.70	0.057	0.10	150	0.86	0.20	80

Source: Author's estimates based on various studies and vehicle manufacturers specifications

Table B4.5: Engine Efficiency Parameters

Engine Type	ξb		ehp
Diesel	0.059	Old technology engines	0.10
	0.058 - [Pmax 10^{-5}]	New technology engines	
Petrol	0.067		0.25

Source: Biggs(1988)

To predict the fuel consumption one must calculate the tractive power, the accessories power and the engine drag. These are then used in conjunction with the efficiency factor to establish the fuel use. An example of the contributions of the various components to the total fuel consumption is illustrated in Figure B4.10 for the default HDM-4 medium passenger car.

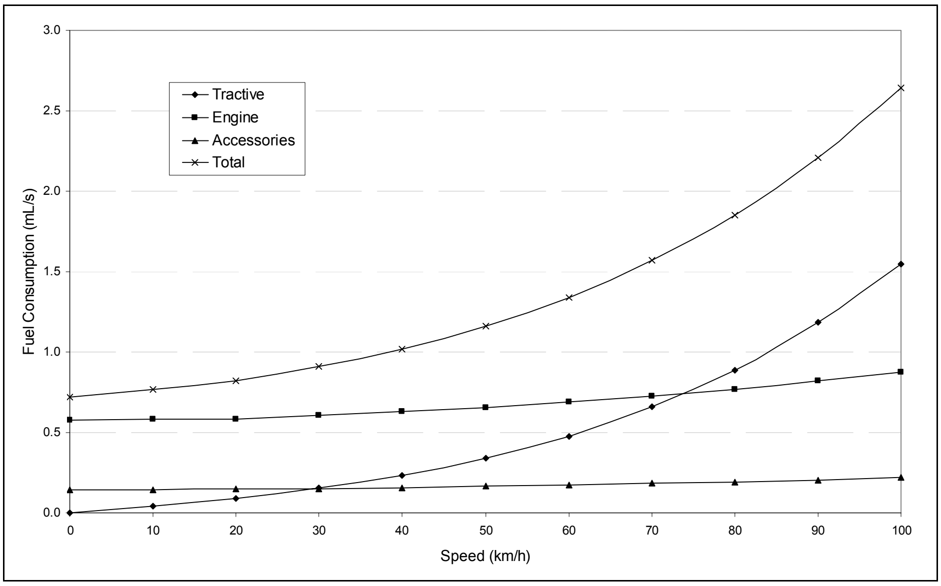


Figure B4.10: Components of Total Fuel Consumption

B4.5 Quantifying Power Requirements

The HDM-4 fuel model predicts that the fuel consumption is proportional to the power requirements. This section describes how these power requirements are quantified and builds on the introduction to this topic given in Chapter B1 and Section B4.4.

B4.5.1 Tractive power

The tractive power is the power required to overcome the forces opposing motion. The quantification of the tractive forces was discussed in Chapter B1 and so is not repeated here. In summary the tractive power is given by:

$$Ptr = Ftr\ v / 1000 \qquad ...(B4.10)$$

$$Ftr = Fa + Fr + Fg + Fc + Fi \qquad ...(B4.11)$$

Equations were presented in Chapter B1 for calculating the forces based on mechanistic principles. As noted in Chapter B1 and Section B4.4, as the vehicle drivetrain is not perfectly efficient a higher level of power is required to be supplied by the engine than that required to overcome just the tractive power.

B4.5.2 Accessory Power and Engine Drag

Accessories resistance is defined as the power to drive the vehicle accessories such as the cooling fan, power steering, air conditioner, alternator *etc.* These items, while often large consumers of power when in operation, tend to be utilised intermittently, thereby making a definitive statement on accessories power for any given vehicle difficult. Engine drag is the level of power consumed to overcome internal friction in the engine itself, and is related to the speed of the engine and various other parameters.

In preparing the ARFCOM model, Biggs (1988) presented a series of relationships for predicting both the accessories power and the engine drag of an engine. The basic relationships as reported by Biggs (1988) are given below as Equations B4.12 and B4.13. Of note is the comment by Biggs (1988) that the determination of the parameter values for the engine drag equation was quite problematic with low coefficients of determination and high standard errors.

$$\text{Pacs} = \text{EALC}\ \frac{\text{RPM}}{\text{TRPM}} + \text{ECFLCP}\ \max\left(\frac{\text{RPM}}{\text{TRPM}}\right)^{2.5} \qquad \text{...(B4.12)}$$

where EALC is the accessory load constant in kW
ECFLC is the cooling fan constant
TRPM is the load governed maximum engine speed in rev/min

$$\text{Peng} = \text{ceng} + \text{beng}\left(\frac{\text{RPM}}{1000}\right)^{2} \qquad \text{...(B4.13)}$$

where ceng is the speed independent engine drag parameter
beng is the speed dependent engine drag parameter

An alternative model to Equation B4.13 was developed by NDLI (1995) that attempted to address some of these problems, however, this was found to have its own problems. Greenwood (1996) revised the previous work and produced a single relationship to predict the combined accessories and engine power requirements as a function of engine speed. This revised relationship is presented as Equation B4.14 below.

$$\text{Pengaccs} = \text{kPea Prat}\left(\text{Paccs\_a1} + (\text{Paccs\_a0} - \text{Paccs\_a1})\frac{\text{RPM} - \text{RPMIdle}}{\text{RPM100} - \text{RPMIdle}}\right) \qquad \text{...(B4.14)}$$

where kPea is a calibration factor (default = 1)
Paccs_a0, Paccs_a1 are parameter values
RPMIdle is the idle engine speed in rpm
RPM100 is the engine speed when travelling at 100 km/h in rpm

The combined value of Pengaccs may be split into its two components as indicated below in Equations B4.15 and B4.16.

$$\text{Peng} = \text{Pengaccs PctPeng} / 100 \qquad \text{...(B4.15)}$$

$$\text{Paccs} = \text{Pengaccs}\ (100 - \text{PctPeng}) / 100 \qquad \text{...(B4.16)}$$

where PctPeng is the percentage of Pengaccs ascribed to engine drag (default = 80%)

Paccs_a1 represents the proportion of the rated engine power utilised to overcome engine drag and accessories loading at idle, while Paccs_a0 is the equivalent value when the vehicle is travelling at 100km/h. The parameter value Paccs_a1 is related to the idle fuel consumption and may be calculated as below.

$$\text{Paccs\_a1} = \frac{-b + \sqrt{b^2 - 4ac}}{2a} \qquad \text{...(B4.17)}$$

$$a = \xi b \text{ ehp kPea}^2 \text{ Prat } \frac{100 - \text{PctPeng}}{100} \qquad \text{...(B4.18)}$$

$$b = \xi b \text{ kPea Prat} \qquad \text{...(B4.19)}$$

$$c = -\alpha \qquad \text{...(B4.20)}$$

where a,b,c are model parameters

B4.6 Predicting Engine Speed

The ARFCOM model, like HDM-III, uses engine speed as an independent variable in its predictions. However, as described in Bennett and Dunn (1989), the AFRCOM engine speed equations lead to a discontinuous relationship between vehicle speed and engine speed when the vehicle shifts into top gear. Such discontinuities lead to inconsistent fuel consumption predictions and should therefore be avoided in HDM-4. While one solution would be to adopt a constant engine speed, such as was done for HDM-III, this creates a bias in the predictions and was also considered unsuitable.

Accordingly a Monte-Carlo simulation was undertaken to develop a continuous function between vehicle speed and engine speed. The description of the model is contained in Chapter B1, under the section on calculating the effective mass ratio (EMRAT) values, with further details given in Annex B1.2 so it is not repeated here.

The result of the simulation program and accompanying analysis is the following relationship for predicting engine speed as a function of vehicle speed. All parameter values for the equations are given in Annex B1.2 of Chapter B1.

$$RPM = a0 + a1\ SP + a2\ SP^2 + a3\ SP^3 \qquad \text{...(B4.21)}$$

$$SP = \max(20, S) \qquad \text{...(B4.22)}$$

where a0 to a3 are model parameters
S is the vehicle road speed in km/h

This formulation is different to that from NDLI (1995) which was incorporated into HDM-4 v 1.0. The NDLI (1995) model consisted of three sets of equations which applied at different speeds. However, the polynomial model above gives equivalent predictions and is simpler to apply so it has been recommended as a replacment for the NDLI (1995) model. Annex B1.2 includes the NDLI (1995) model and its parameters.

B4.7 Effects of Traffic Interactions on Fuel Consumption

B4.7.1 Introduction

The methodology presented in Chapter B3 was used in conjunction with the HDM-4 fuel consumption model to establish the additional fuel consumption due to traffic congestion. The calculations were performed using the program ACCFUEL[1]. As described in Chapter B3, the vehicles were simulated travelling along an idealised section of road at different levels of congestion. As the congestion increased, so did the acceleration noise and thus the fuel consumption. To evaluate the predictions of the model, sample calculations were made for three markedly different classes of road.

B4.7.2 Effects of Acceleration Noise on Fuel Consumption

The presence of acceleration noise would be expected to increase the fuel consumption. This was found to be the case as illustrated in Figure B4.11, which shows the effect of different levels of acceleration noise on passenger car[2] fuel consumption at different speeds.

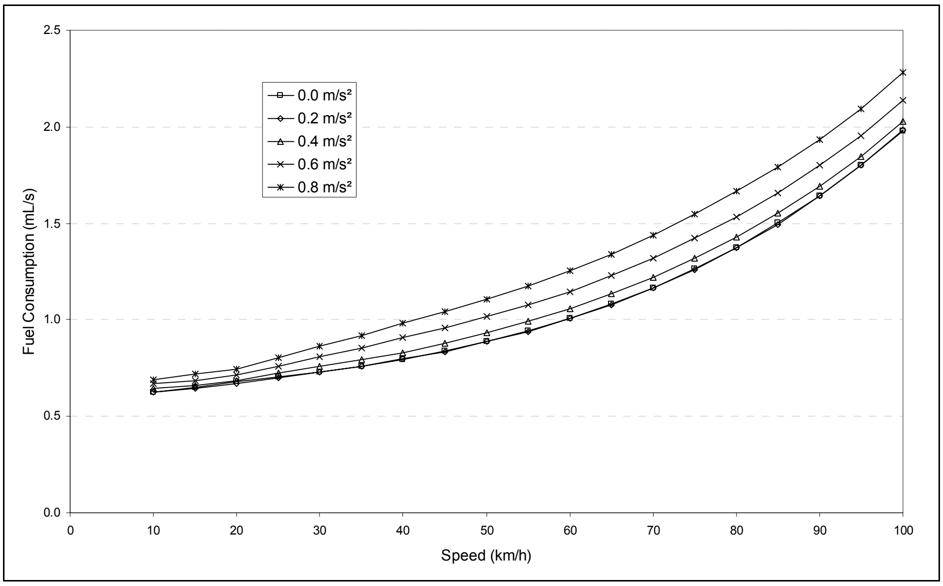


Figure B4.11: Effect of Acceleration Noise on Passenger Car Fuel Consumption (mL/s)

The results in Figure B4.11 show that it is only when the acceleration noise is above 0.2 m/s^2 that there is a significant increase in the fuel consumption. The magnitude of the difference is greatest at low speeds since at higher speeds there is a much higher rate of fuel consumption to overcome the forces opposing motion.

It will be noted in Figure B4.11 that below 20 km/h the fuel consumption curves converge. This is due to the restrictions placed on the speed range for vehicles travelling at low speeds (see Chapter B3).

Figure B4.12 is the same data from Figure B4.11 except converted from instantaneous (mL/s) to a distance base (mL/km). This figure indicates that the congestion effects are quite significant, particularly when the vehicle is travelling below its optimum speed.

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1 HDM Tools may be obtained from www.opus.co.nz/hdmtools or www.htc.co.nz.

2 The vehicle simulated is the default HDM-4 medium passenger car.

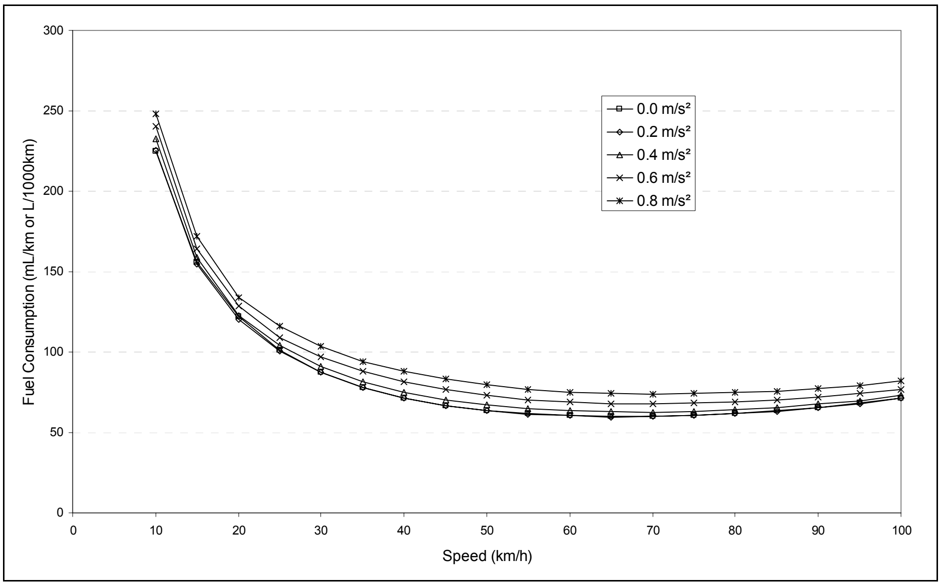

Figure B4.12: Effect of Acceleration Noise on Passenger Car Fuel Consumption (mL/km)

The results for articulated trucks are illustrated in Figure B4.13 and Figure B4.14, which are equivalent to those presented above for passenger cars. Here, because of the higher vehicle mass, the acceleration noise has a much larger effect on the fuel consumption.

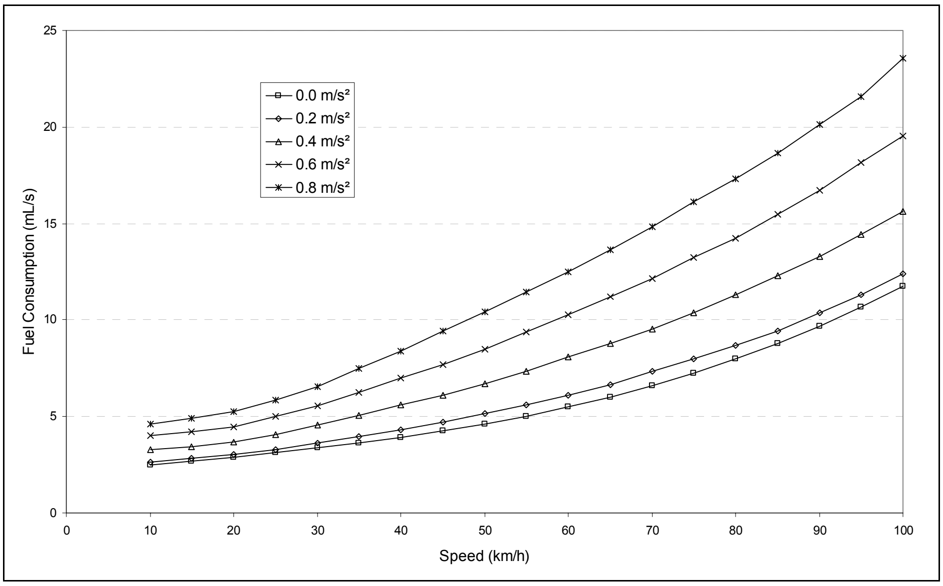

Figure B4.13: Effect of Acceleration Noise on Articulated Truck Fuel Consumption (mL/s)

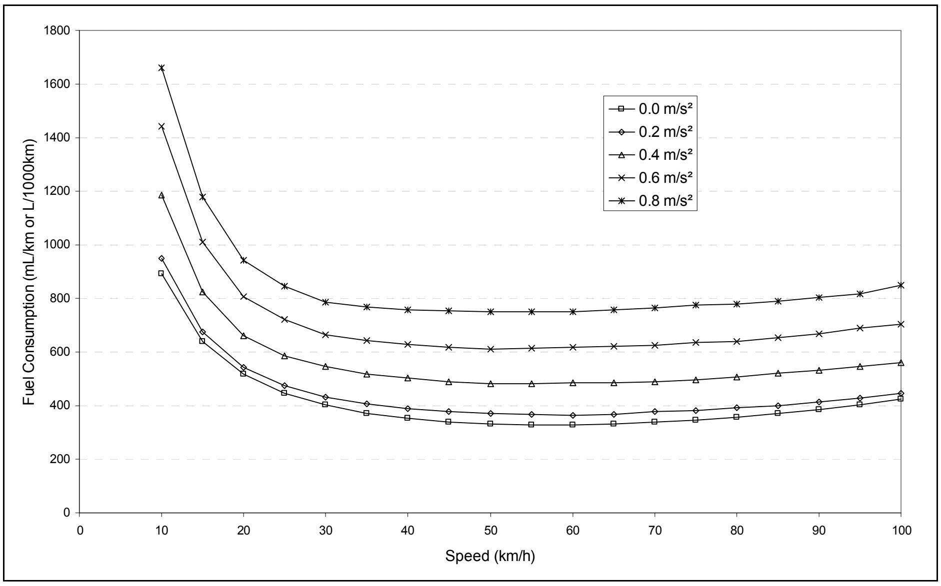

Figure B4.14: Effect of Acceleration Noise on Articulated Truck Fuel Consumption (mL/km)

Figure B4.15 illustrates the passenger car fuel consumption ratio and Figure B.4.16 the articulated truck ratio. This ratio is defined as follows:

$$\mathrm{dFUEL} = \frac{\mathrm{FCCONG}}{\mathrm{FCSTEADY}} - 1 \qquad \text{...(B4.23)}$$

where dFUEL is the incremental increase in fuel consumption owing to congestion

FCCONG is the congested fuel consumption in mL/km

FCSTEADY is the steady speed fuel consumption in mL/km

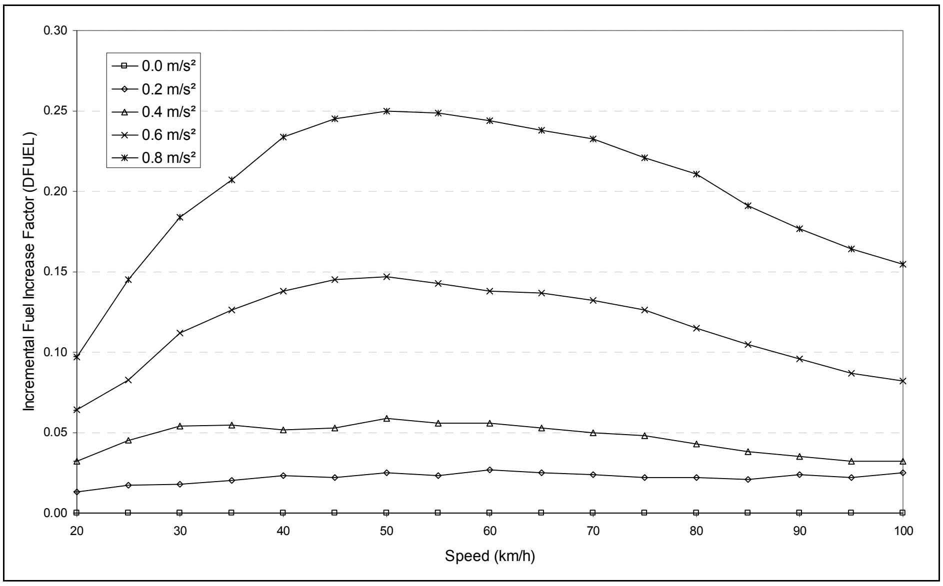

Figure B4.15: Passenger Car Fuel Consumption Ratio

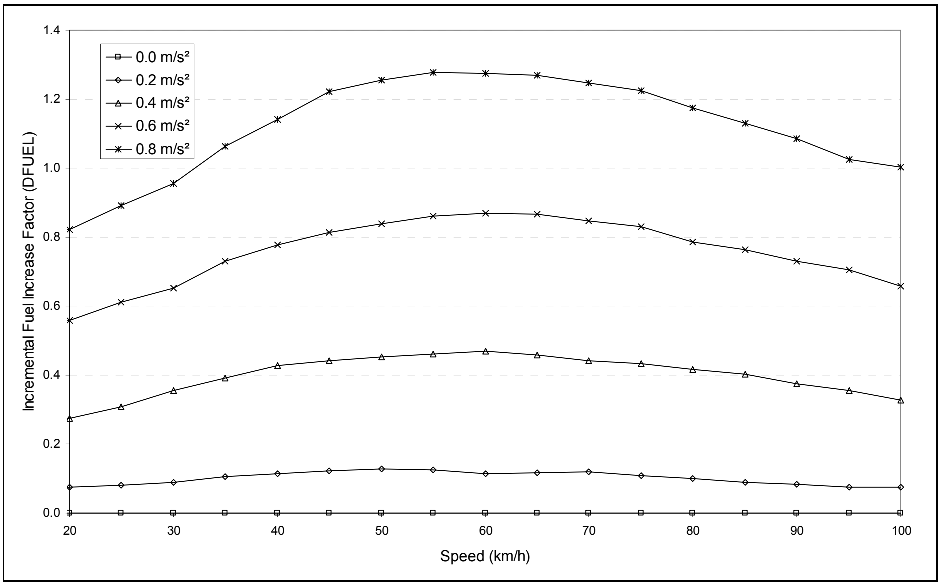

Figure B.4.16: Articulated Truck Fuel Consumption Ratio

The ratios are very sensitive to the vehicle characteristics, particularly the idle fuel rate. This is because the idle rate can be quite a significant component of the uncongested fuel rate, especially at low speeds, which is the denominator. However, the additional fuel due to congestion is unaffected by the changes in the idle fuel rate since the latter is a constant for all cases.

The fuel consumption is therefore calcualted as:

$$IFC = \max(\alpha, \xi \, Ptot \, (1 + dFUEL)) \quad \text{...(B4.24)}$$

B4.7.3 Fuel Bias

The traffic stream is comprised of a stream of vehicles travelling at different speeds and, thus, different fuel consumption rates. Since fuel consumption is non-linear with speed, the mean fuel consumption does not correspond to the fuel consumption at the mean speed.

Bennett (1996a) investigated the implications of this using a Monte Carlo simulation with the HDM-4 fuel model. The analysis was run for 1500 vehicles and all 16 of the default HDM-4 vehicles. The default characteristics for these vehicles were used in the analysis.

The speeds were assumed to be normally distributed about a mean speed with a coefficient of variation (COV = standard deviation/mean) from 0.05 to 0.25 in 0.05 increments. For mean speeds from 10 to 100 km/h, in 5 km/h increments, the individual speeds of the 1500 vehicles comprising this mean were established. The instantaneous fuel (in mL/s) and the specific fuel (in mL/m) were calculated. The ratio of these values to the fuel travelling at the mean speed was then determined.

The bias for the fuel consumption was significant and its magnitude varied between vehicle type. The bias generally followed the "U" shape which is associated with the effect of speed on specific fuel consumption. The bias was found to be consistent between 20 and 80 km/h for most vehicle types. The average was calculated by vehicle class and the differences between classes was a maximum of 0.3 per cent. It was therefore decided to adopt a single factor for all classes. A regression was made of the average correction as a function of COV which resulted in the following equation:

$$FUELBIAS = 1.0000 - 0.0182\, COV + 0.7319\, COV^2 \qquad R^2 = 1.0 \qquad \text{...(B4.25)}$$

A lower limit on the fual bias is given by the parameter dFUEL so the equation is applied as:

$$FUELBIAS = \max(dFUEL, 1.0000 - 0.0182\ COV + 0.7319\ COV^2) \qquad ...(B4.26)$$

B4.7.4 Congestion Effects for a Traffic Stream

The discussion in Section B4.7.2 showed the effects of congestion on the fuel consumption of a single vehicle. However, more relevant to HDM-4 analyses are the effects of congestion on the entire traffic stream. This is because the traffic stream is comprised of vehicles travelling at a range of speeds and these speeds are in turn influenced by the volume-to-capacity ratio. To illustrate the effects of congestion on fuel consumption analyses were conducted for three markedly different roads:

- A high standard two-lane highway from N.Z.;
- A low standard two-lane highway from India; and,
- A low standard one-lane highway from India.

The HDM-4 speed-flow model characteristics for these roads were given in Chapter B3.

The simulation program ACCFUEL was run for each of these roads and the fuel consumption was calculated both for steady state and congested conditions. The acceleration noise at different flow levels was predicted using the equations presented in Chapter B3. The predicted additional fuel consumption in mL/s and mL/km for passenger cars is illustrated in Figure B4.17 and Figure B4.18 respectively.

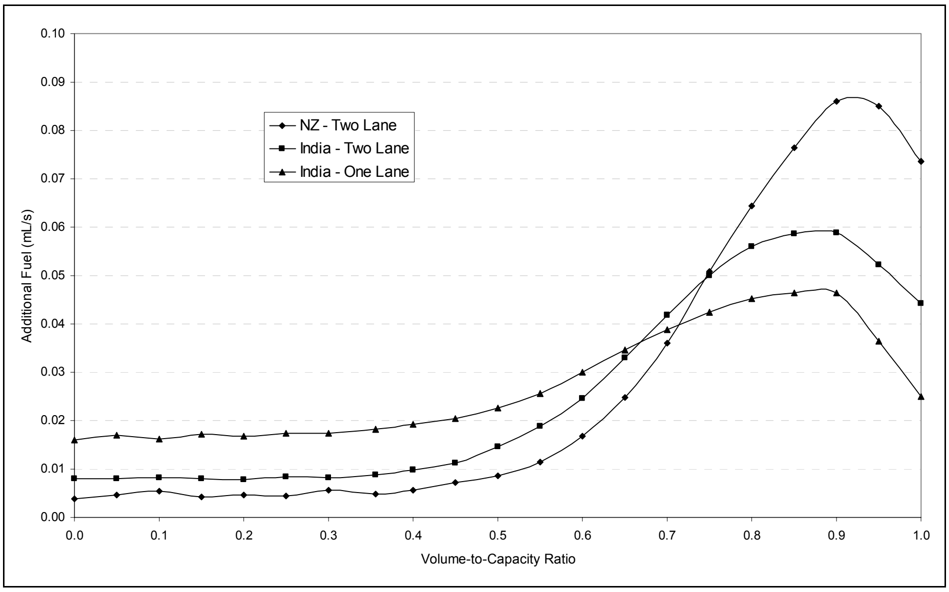

Figure B4.17: Effect of Flow on Additional Fuel Consumed (mL/s)

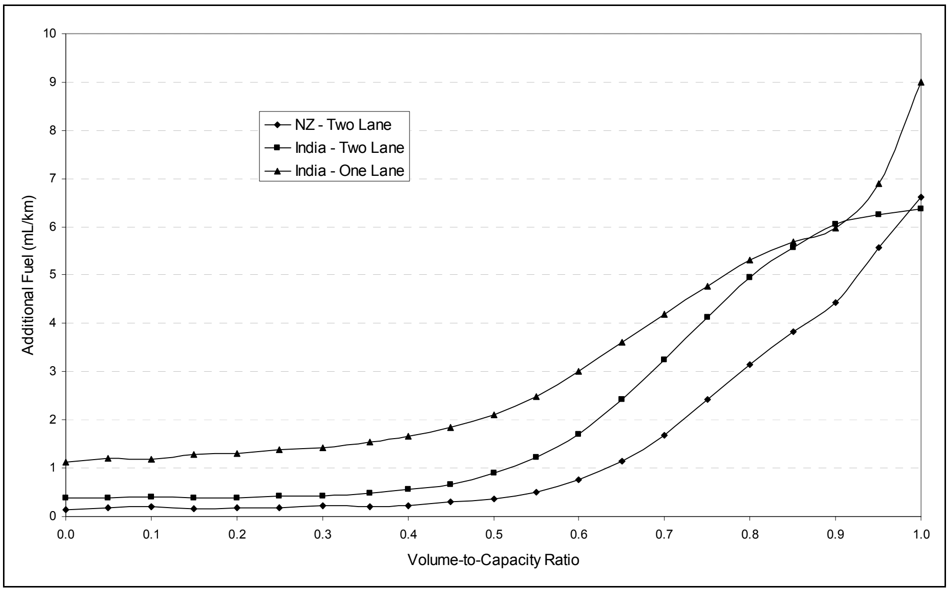

Figure B4.18: Effect of Flow on Additional Fuel Consumed (mL/km)

The predictions follow the expected pattern wherein for the same volume-to-capacity ratio there is an inverse relationship between the standard of road and the additional fuel consumed due to congestion. More fuel is consumed on the lower standard road than the higher standard road for the same volume-to-capacity ratio.

When expressed in mL/s (Figure B4.17) the additional fuel for the Indian one-lane road appears to be inconsistent in that there is a marked decrease in the additional fuel as one moves from volume-to-capacity ratios of 0.7 to 1.0. However, this is also due to the decrease in speeds and the corresponding relative increase in the steady state fuel consumption. When the data are converted to mL/km (Figure B4.18) this apparent inconsistency is eliminated.

Figure B4.19 illustrates the relative fuel consumption as a function of the volume-to-capacity ratio. The results are very similar to the absolute fuel consumption in mL/s which was illustrated in Figure B4.17, except that the N.Z. results are much lower than those for the Indian roads.

At each flow level there is a unique speed. Figure B4.20 and Figure B4.21 show the variation in the additional fuel consumed with speed.

B4.8 Summary

This chapter has presented the HDM-4 fuel consumption model. The model adopted is based on the ARRB ARFCOM mechanistic model, with alteration to the prediction of engine speed, accessories power and engine drag.

The fuel model predicts that fuel is proportional to the power requirements of the vehicle. it can be expressed as:

$$IFC = \max(FCmin, \xi\ Ptot\ (1 + dFUEL)) \qquad \text{...(B4.27)}$$

where

IFC	is the instantaneous fuel consumption in mL/s
FCmin	is the minimum fuel consumption in mL/s
ξ	is the fuel-to-power efficiency factor in mL/kW/s
Ptot	are the total vehicle power requirements
dFUEL	is the additional fuel due to accelerations

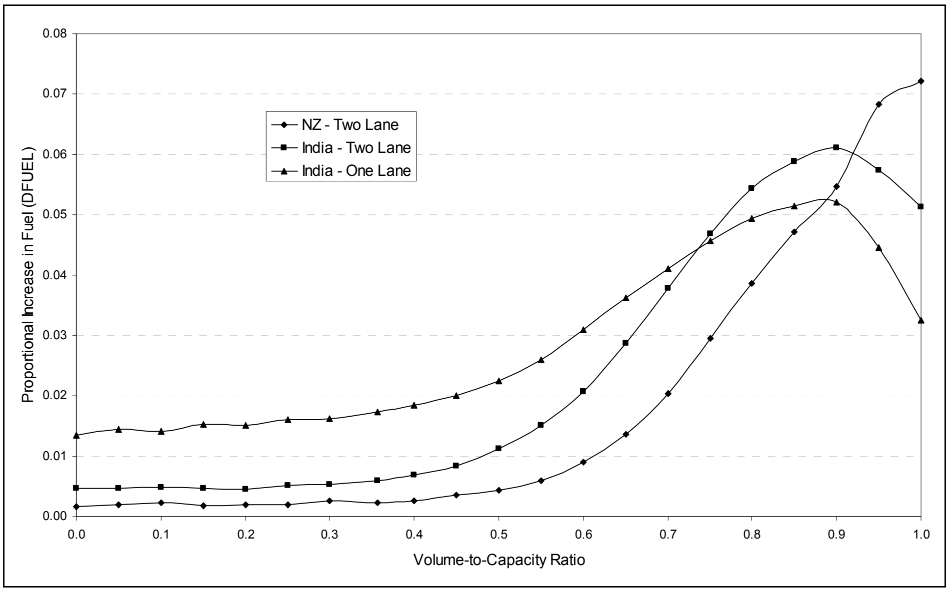

Figure B4.19: Effect of Flow on Passenger Car Relative Fuel Consumption

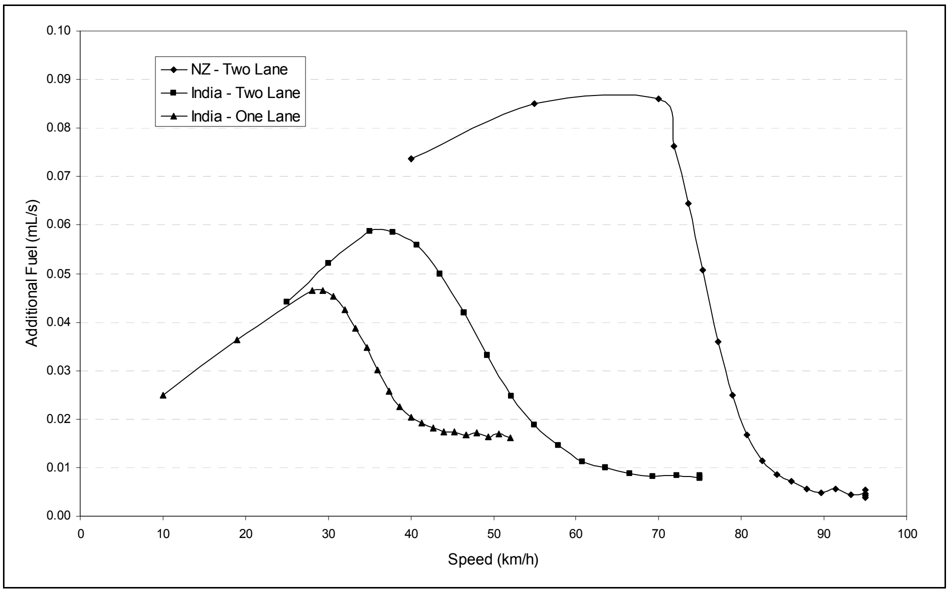

Figure B4.20: Effect of Speed on Additional Fuel Consumed (mL/s)

The total power requirements are comprised as the power to overcome the tractive forces (see Chapter B1), to overcome engine drag and to power vehicle accessories. Owing to the inefficiencies of the drivetrain, the total power requirement is calculated by two alternative methods depending on whether the tractive power is positive or negative. The two formulae are:

$$\text{Ptot} = \frac{\text{Ptr}}{\text{edt}} + \text{Paccs} + \text{Peng} \qquad \text{Ptr} \geq 0 \qquad \text{...(B4.28)}$$

$$\text{Ptot} = \text{edt Ptr} + \text{Paccs} + \text{Peng} \qquad \text{Ptr} < 0 \qquad \text{...(B4.29)}$$

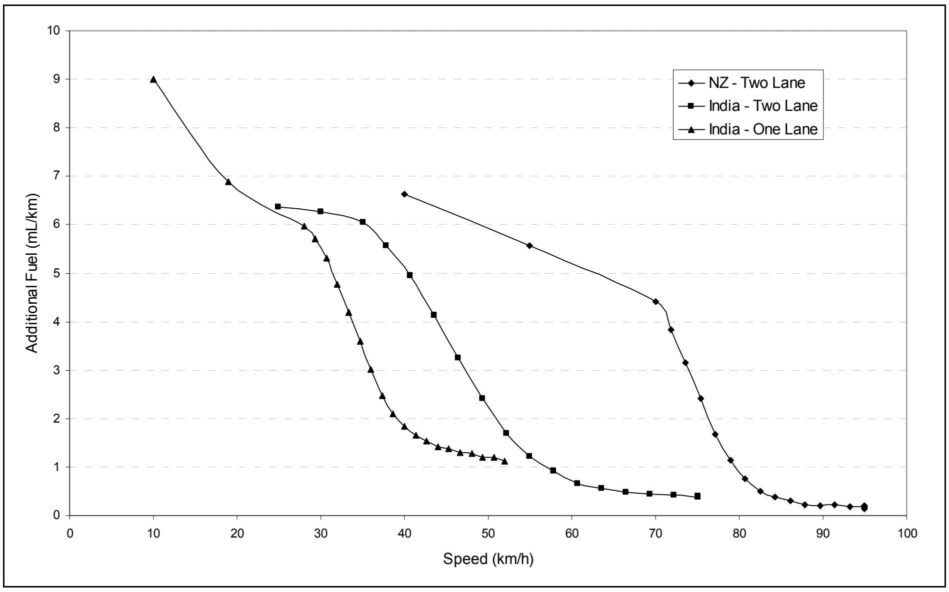

Figure B4.21: Effect of Speed on Additional Fuel Consumed (mL/km)

Engine efficiency decreases at high levels of output power, resulting in an increase in the fuel efficiency factor ξ. The following relationship to account for this decreased efficiency:

$$\xi = \xi b\ (1 + ehp\ (Ptot\text{-}Peng)/Prat) \qquad \text{...(B4.30)}$$

The model was used to predict the effects of traffic congestion on fuel consumption. While oriented towards fuel, the same approach can be applied to any other vehicle operating cost component that is predicted using mechanistic principles, for example tyre consumption.

The congestion modelling approach was described in Chapter B3 and is based on the thesis that there is an increase in accelerations with increasing congestion. This increase manifests itself through an increase in the acceleration noise, which is the standard deviation of acceleration.

Using mechanistic principles, the fuel consumption is proportional to the acceleration. A Monte-Carlo simulation technique was used which simulated a vehicle driving along a road at different levels of congestion. The output from this simulation was the additional fuel consumed at different levels of congestion, with a parameter dFUEL used to represent the incremental increase in fuel consumption owing to congestion. dFUEL is defined as:

$$dFUEL = \frac{FCCONG}{FCSTEADY} - 1 \qquad \text{...(B4.31)}$$

B5 Tyre Consumption

B5.1 Introduction

Tyre consumption can be a major component of the total road user costs, particularly for heavy trucks. For example, Opus-TRL (1999) report that in N.Z. tyre costs constitute 18 per cent of the total user costs for heavy trucks towing, compared to only five per cent for passenger cars. In the Brazil study the tyre costs for trucks were 23 per cent of the total in rolling terrain (Watanatada, *et al.,* 1987b).

Tyres are consumed continuously as vehicles travel. There are two principal modes of tyre consumption:

- **Tread wear**: This is the amount of the tread worn due the mechanism of the tyre coming into contact with the pavement surface.
- **Carcass wear**: This is a combination of fatigue and mechanical damage to the tyre carcass. It is defined as the number of retreads (recappings) to which a tyre carcass can have before the carcass is unsuitable for further retreads. In many countries, blowouts or ablative wear—*ie* the 'tearing' of the tread caused by surface material—are significant factors governing tyre life.

Each kilometre travelled results in a loss of tread. In addition, the forces acting on the vehicle result in stresses and strains to the tyre carcass. When the tyre tread reaches a certain depth, the tyre is either discarded or, if the economics warrant it and the carcass is in adequate condition, the tyre will be retreaded which sees new tread added to the existing carcass. Thus, in modelling tyres it is necessary to explicitly consider retreading, particularly for commercial vehicles where retreading is a common practice in many countries.

Figure B5.1, which is adapted from Nordström and Andersson (1995), shows the factors influencing tyre consumption. They can be broken down into two distinct groups: those which influence the rate of tyre wear per unit of energy and those which dictate the energy to the tyre. The principal factors include:

- **Pavement condition:** Tyre consumption will increase with increasing roughness since that leads to increased vertical loading on the tyres. The type of surface, its condition and its texture all play important roles in the tyre consumption.
- **Road alignment:** Tyre consumption increases significantly with the severity of road alignments, particularly with respect to horizontal curvature.
- **Traffic conditions:** Traffic interactions give rise to accelerations and decelerations which have a strong impact on tyre consumption.
- **Vehicle loading:** The tyre consumption is proportional to the vehicle loading and overloading, particularly when coupled with poor road condition, can lead to an increased frequency of carcass failures, especially on driven axles.
- **Climate:** The ambient temperature can have a significant impact on tyre consumption, particularly in tropical countries where it can lead to blowouts or stripping of tread.
- **Tyre properties:** The type of tyre (radial vs bias ply), if it is a new tyre or a retread, the properties of the rubber compounds, and the inflation pressure all have an impact on the tyre consumption rate.

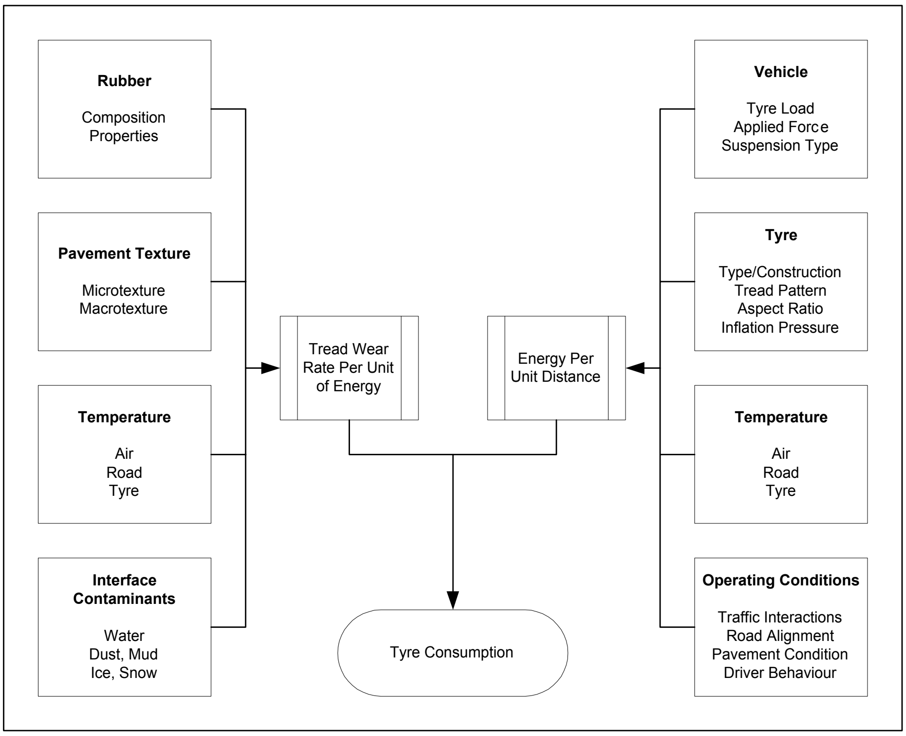

Figure B5.1: Factors Influencing Tyre Consumption

The mechanistic tyre consumption model adopted for HDM-4 has been designed to explicitly or implicitly consider the main factors influencing tyre consumption. It is flexible enough to cater for a range of tyre types and operating conditions. It also models carcass failure and the impact of retreads on tyre costs.

This chapter commences with a review of the research into predicting tyre consumption. This is followed by a description of the HDM-4 mechanistic tyre model. A set of calibrated parameters for this model is given along with examples of its predictions.

B5.2 Predicting Tyre Consumption

B5.2.1 Introduction

There are two approaches which have been used to develop tyre consumption models:

- **Controlled experiments:** These see the tyre consumption measured on a small number of tyres with a high degree of accuracy, or,
- **Fleet surveys:** The tyre consumption of vehicle fleets is monitored. Since the precise operating conditions are not known, it is only practical to relate this tyre consumption to aggregate descriptions of road condition.

There are advantages and disadvantages to each approach. While controlled experiments offer the potential for getting accurate measurements, they are difficult to conduct[1] and by necessity only cover a small range of tyres. They also do not generally give details on tyre

[1] Cenek, *et al*. (1996) describe the practical difficulties encountered in trying to measure texture effects on tyre consumption, particularly with regard to weighing tyres to establish the tread loss.

consumption under 'real life' conditions, although they are sometimes adjusted using additional data from fleet surveys.

Fleet surveys have their own set of difficulties which Chesher and Harrison (1987) summarise as:

> *"... tyre data are difficult to collect because, at least in large organisations, tyres are moved from vehicle to vehicle. Further, tyre life varies greatly from tyre to tyre even under identical operating conditions. Tyre life also varies considerably according to load carried, position on the vehicle, speed of operation, driver behaviour and depends on company policy regarding standards of maintenance of tyres and vehicles and regarding frequency and standard of recap. Even with the relatively large samples obtained in India and Brazil it is difficult to extract information on the relationship between tyre consumption and highway characteristics."*

Thus, although they have the advantage of encapsulating the actual tyre consumption under real life conditions, it is difficult to develop predictive models from the data insofar as the models may give unexpected results.

For example, Papagiannakis (1999) analysed tyre data for truck fleets operating on smooth roads in North America. Even though the average roughnesses were only in the range of 1.4 – 1.9 IRI m/km, it was suggested that there was a significant (> 50 per cent) increase in tyre wear over this range. Since the routes were comprised of interstate highways, these differences cannot be ascribed to speed differences and it is not clear why such changes would be observed at such low roughnesses when mechanistic theory suggests less of an impact.

Symonds (1996) investigated tyre consumption using fleet data in Australia. Among their findings was that the variation in tyre consumption for similar vehicles operating on the same route was greater than the tyre consumption differences between routes—something also commonly encountered in maintenance and repair cost modelling.

One particular complication with fleet data are the widely differing tyre lives found in service. For example, Daniels (1974) reports new passenger car lives in Africa ranging from 27,000 to 40,000 km. Large variations in life between vehicles was found in the India and Brazil user cost studies (Chesher and Harrison, 1987).

As described by Le Maître, *et al.* (1998), there are marked variations in tyre life between drivers and geographical areas. As shown in Figure B5.2, the distribution of tyre life follows a lognormal distribution, with some drivers—representing extreme use—experiencing very short tyre lives, while others with mild use having long lives. Le Maître, *et al.* (1998) show that severity can be "interpreted by the differences in acceleration levels which each driver imposes on the vehicle" and that this can lead to a several fold difference in tyre wear between different drivers.

There are two types of models which have been developed for predicting tyre consumption:

- **Mechanistic:** These relate the tyre consumption to the fundamental equations of motion and are generally developed from controlled experiments.
- **Empirical:** These are more aggregate models, usually developed from fleet survey data.

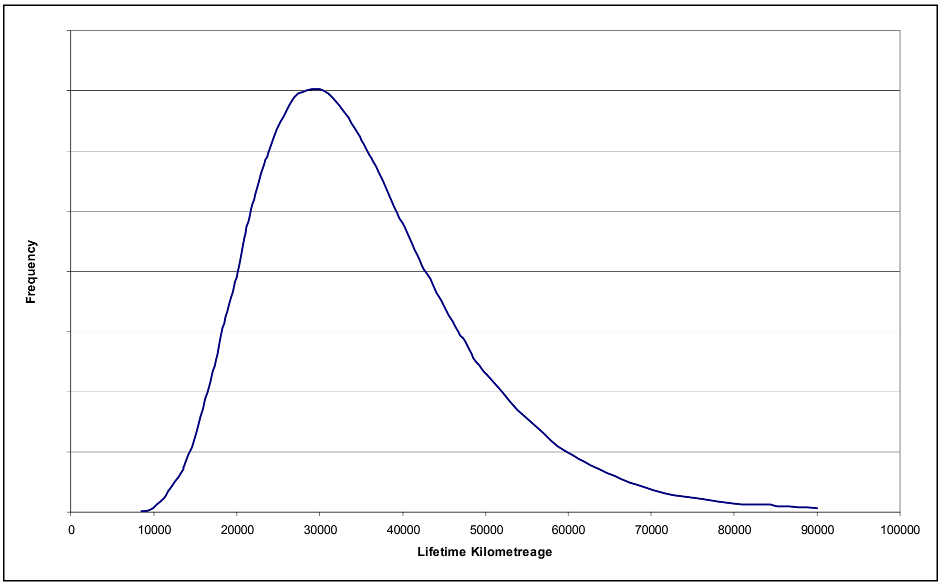

Figure B5.2: Typical Tyre Life Distribution

Du Plessis and Schutte (1991) suggest that when modelling tyre consumption "empirical models seem to perform best, since they capture the tyre policies implemented by operators which would be difficult, if not impossible, to model using purely theoretical purposes".

However, mechanistic models have the advantage of being able to cater for the full range of operating conditions encountered by vehicles in a consistent and theoretically robust manner. For this reason, HDM-4 uses mechanistic models but with coefficients which reflect the results of empirical studies.

Irrespective of the type of model, it is essential that retreaded tyres be explicitly considered. Retreaded tyres generally cost less and may have different performance properties to new tyres. The amount in use depends upon economic factors. In some countries they are common while in others rare. For example, Cenek, *et al.* (1993) report that 20 per cent of N.Z. passenger car tyre sales were retreads while a sample of buses had 57 new tyres which were retreaded a total of 176 times making retreads 75 per cent of the tyres.

B5.2.2 Mechanism of Tread Wear

TYPES OF TREAD WEAR

Most research into tyre consumption has focused on developing tread wear models. Tread wear is the "accumulated loss of tread rubber resulting from an interaction between the tyre and the road in different operating and environmental conditions" (Bergman and Crum, 1973). There are three basic types of tread wear:

- **Abrasive wear:** This arises from the contact of the tyre with the pavement surface and is primarily due to tensile failure of the tread rubber. It can be viewed as a 'tearing' of the tread caused by the contact with the pavement surface and is generally caused by microtexture. Texture effects are quite important, for example Cenek, *et al.* (1996) report a range of 300 - 600 per cent in tyre abrasion rates between different surface textures.

- **Fatigue wear:** This arises from the fatigue failure of the tread and can be described by the slow removal of a thin outer layer of tread by oxidation and repeated deformation, rather than by the more rapid abrasive wear.

- **Thermal decomposition wear:** This arises under extreme conditions, such as when wheels are locked during braking or spun during extreme acceleration. The temperatures generated exceed the decomposition temperature of rubber and rubber separates from the tread and is adhered to the pavement surface (*ie* a skid mark).

In the context of HDM the primarily focus is on abrasive wear, with some fatigue wear implicitly modelled through the tread wear parameters.

The primary cause of tread wear during straight ahead driving and braking is **tyre slip**. This arises because tyres deflect during the transmission of driving and braking forces from the wheel to the road which results in longitudinal or circumferential motion of the tyre relative to the wheel rim. This relative motion is known as tyre slip.

As described by Bergman and Crum (1973), tyre slip can be explained as follows:

> *"By entering into contact with the road surface, tyre elements establish grip with the road. As a result of this grip, tyre elements are instantaneously anchored to the road, therefore, their instantaneous motion relative to the ground is stopped for a finite distance within the footprint. [Since] the wheel rim continues to rotate uniformly about its axis … tyre elements in the front of the contact area decelerate to zero from their initial velocity upon entry into contact with the ground…* This results in a compressive deformation of the tyre carcass in front of the contact area and tension behind the contact area. When elastic forces reach the limit established by the grip between the tyre and road, tyre elements in the rear portion of the contact area start to slide relative to the road…The rubbing of tread rubber sliding on the road surface results in tread wear."

During cornering all four wheels operate at a slip angle which is a function of the cornering stiffness—itself a function of the elasticity of the carcass and tread—and the wheel load. The wear is proportional to the slip angle and the size of the tyre contact area.

SLIP-ENERGY MODEL

Moore (1975) indicates that abrasion losses are proportional to the frictional energy dissipation. This can be expressed as:

$$\Delta TWT = K0\ \mu\ NFT\ \lambda \qquad \ldots(B5.1)$$

where

ΔTWT	is the change in tread wear
K0	is a calibration factor reflecting pavement properties
μ	is the coefficient of friction
NFT	is the normal force on the tyre in N
λ	is the tyre slip

With reference to Figure B5.3, Watanatada, *et al.* (1987b) define the tyre slip as:

$$\lambda = \frac{Tsv}{|Vra|} \qquad \ldots(B5.2)$$

where

Tsv	is the tyre slip velocity defined as the vectoral difference between the travel and circumferential velocities of the wheel
Vra	Is the velocity of the road relative to the axle

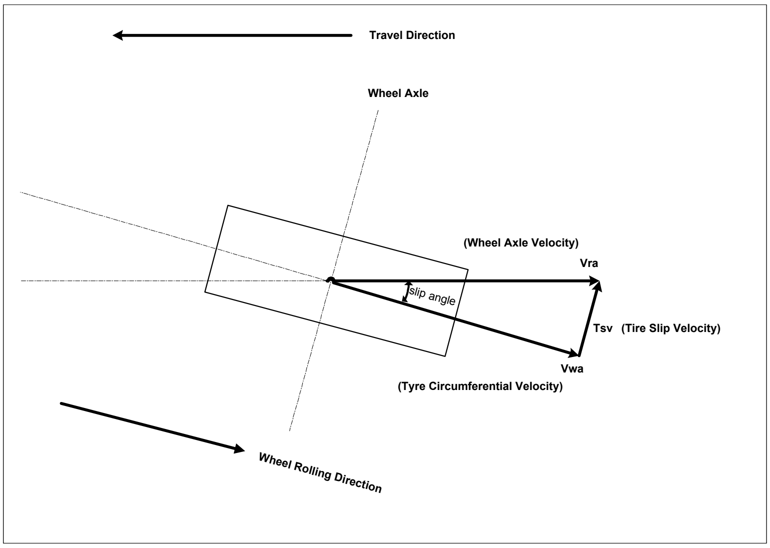

Figure B5.3: Illustration of Slip Velocity

When the wheel rolls freely in a direction that makes the slip angle η to the direction of travel, we have zero circumferential slip and the lateral slip is given by:

$$\lambda_l = \sin \eta \qquad \text{…(B5.3)}$$

where λ_l is the lateral slip
η is the slip angle

For small slips $\lambda_l = \eta$ so the slip angle is used in place of the lateral slip λ_l.

When the wheel rolls for straight ahead driving, the slip angle and the lateral slip are zero, and the circumferential slip is given by:

$$\lambda_c = 1 - \frac{|Vwa|}{|Vra|} \qquad \text{…(B5.4)}$$

where λ_c is the circumferential slip
Vra is the velocity of the wheel relative to the axle

λ_c represents the metres of slip per metre of wheel travel.

For small slips the slip angle can be written as (Watanatada, *et al*., 1987b):

$$\lambda_c = \frac{b}{k_c} \frac{CFT^n}{NFT^{(n-1)/2}} \qquad \text{…(B5.5)}$$

$$\lambda_l = \frac{b}{k_l} \frac{LFT^n}{NFT^{(n-1)/2}} \qquad \text{…(B5.6)}$$

where CFT is the circumferential force on the tyre in N
LFT is the lateral force on the tyre in N
k_c is the circumferential stiffness of the tyre
k_l is the lateral stiffness of the tyre
b is a function of the tyre's dimensions and mechanical properties

Substituting for the coefficient of friction, defined as the circumferential or lateral force divided by the normal force, results in the following equations[1]:

$$TWT_c = K0 \frac{b}{k_c} \frac{CFT^{n+1}}{NFT^{(n+1)/2}} \qquad \text{…(B5.7)}$$

$$TWT_l = K0 \frac{b}{k_l} \frac{LFT^{n+1}}{NFT^{(n+1)/2}} \qquad \text{…(B5.8)}$$

where TWT_c is the tread wear resulting from circumferential forces in dm^3/1000 km

TWT_l is the tread wear resulting from lateral forces in dm^3/1000 km

K0 is the slip energy coefficient in dm^3/MJ

The dimensionless term $\frac{b}{k}$ is called the 'slip coefficient' and the term K0 is the 'slip energy coefficient'. The latter represents the amount of tyre wear per unit slip energy. The product of these represents the **tread wear coefficient**.

As described in Watantatada, *et al*. (1987b), the magnitude of the exponent n has been subject to debate. Although some researchers have found values greater than 1, these have generally been for unusual conditions. Various road tests have shown that a value of 1 is appropriate so the above models simplify to:

$$TWT_c = K0 \frac{b}{k_c} \frac{CFT^2}{NFT} \qquad \text{…(B5.9)}$$

$$TWT_l = K0 \frac{b}{k_l} \frac{LFT^2}{NFT} \qquad \text{…(B5.10)}$$

For HDM-III Watanatada, *et al*. (1987b) assumed that the circumferential and lateral stiffnesses are equal and expressed the forces as the total tangential force. The above equations therefore reduced to:

$$TWT = C0tc + TWT_c + TWT_l \qquad \text{…(B5.11)}$$

or,

$$TWT = C0tc + K0 \frac{b}{k} \frac{TFT^2}{NFT} \qquad \text{…(B5.12)}$$

$$TFT = \sqrt{CFT^2 + LFT^2} \qquad \text{…(B5.13)}$$

where TWT is the total tread wear in dm^3/1000 km

---

1 There is an error in the equations presented in Watanatada, *et al*. (1987b). Equations 11A.6a and 11A.6b have the term: $NFT^{(n+1)/2}$. However, the work of Shallamach (1981)—referenced as the basis for these equations—and the theoretical derivation above indicate that this term should in fact be $NFT^{(n-1)/2}$. The final equations from their derivation in Watanatada, *et al*. (1987b)—11A.7a and 11A.7b—are correct which suggests that this was a typographical error.

TFT	is the total tangential force which is the vectorial sum of the mutually perpendicular lateral and circumferential components in N
C0tc	is the tread wear rate constant in dm^3/1000 km

B5.2.3 Mechanism of Retreaded Tyre Wear

Du Plessis and Schutte (1991) note that “It is normal practice for bus and goods vehicle tyres to be retreaded to prolong service life, provided the carcass is still serviceable”. The decision to retread obviously depends on the cost of recapping and the expected lifetime of the tyre thereafter, and these trade-offs need to be considered in the modelling of tyre costs.

There have been different findings with regard to the life of retreaded tyres compared to new tyres. Bonney and Stevens (1967) report “the mileages obtained from new and retreaded tyres were generally similar for any particular route”. However, as shown in Table B5.1, in India there was a marked decrease in tyre life with increasing number of retreads. Table B5.1 also shows the effect of retreads on tyre life from Brazil where after an initial decrease of 30 per cent the tyre life stabilised for subsequent retreads.

Table B5.1: Effect of Retreading on Tyre Life

Stage of Tyre Life	India[1]			Brazil[2]			N.Z.[3]
	Average km Travelled by Survivors	Travel as Percentage of New Tyre	Per Cent Surviving	Average km Travelled by Survivors	Travel as Percentage of New Tyre	Per Cent Surviving	Per Cent Surviving
New Tyres	32.,916	100	100	26,939	100	100	100
First Retread	21,515	65	67	18,628	69	54	100
Second Retread	20.853	63	27	18,707	69	27	84
Third Retread	18,191	55	7	18,495	69	13	72
Fourth Retread	10,364	31	1	18,581	69	5	40
Fifth Retread	-	-	-	19,165	71	2	11
Sixth Retread	-	-	-	-	-	0.4	2

Sources: 1/ *Chesher and Harrison (1987)*
2/ *Watanatada, et al. (1987b)*
3/ *Cenek, et al. (1993)*

There are *a priori* reasons to expect a decrease in tyre life for retreaded tyres over new tyres. Although retreading adds new material to the tyre, it is not possible to restore it to its original strength. In addition, the process by which the tyre is retreaded involves heating the tyre to a high temperature which further weakens its structure. On the basis of their data given in Table B5.1, Watanatada, *et al.* (1987b) assumed that “the wear rate of retreads is constant regardless of the number of retreadings to which the carcass has been subjected”. They suggested a value of 1.33 for the wear rate of retreads compared to new tyres. This compares with a value of 2.0 adopted by CRRI (1982).

The potentially lower performance of retreads has often been recognised by the trucking industry and they have modified their practices accordingly. For example, Tillman (1983) recommended that while new or first retreads are suitable for any operating conditions or position on the vehicle, by the time a tyre has been retreaded more than three times or repaired twice it should either be used on short-haul trailers or as a drive wheel only for city traffic to minimise “overstressing”.

Table B5.1 also shows that there are significant differences in carcass lives—that is, the suitability of a tyre for retreading—between countries. Whereas in India only one per cent of tyres survived to have a fourth retread, and five per cent in Brazil, 40 per cent of tyres in N.Z.

were suitable for their fourth retread. This difference may be explained in part by tyre type (cross-ply in Briazil vs steel belted radial in N.Z.), but is also likely due to pavement and route characteristics. Jackson and Maze (1982) indicate that for similar trucks operating under similar conditions, radial tyres have a markedly higher probability of being suitable for retreading than cross-ply tyres, with 30 per cent of radials suitable for a third retread compared to 10 per cent for cross-ply tyres.

The suitability for retreading can also be expected to vary by tyre type, for example Cenek, *et al.* (1993) indicate that on average car tyres can be retreaded 1.5 times vs four times for trucks.

The effect of operating conditions on tyre carcass life can be significant. If operating conditions are excessive, the carcass may suffer premature failure and not be suitable for retreading. In Africa Curtayne, *et al.* (1987) found that "45 per cent of tyre [scrappings] were caused by sharp, angular stones in the gravel wearing courses that penetrated the tyre casings." This ablative effect was also found by Findlayson and du Plessis (1991) who noted that in their study of buses that the depot with the lowest roughness had the highest average number of retreads per casing.

B5.2.4 Integrating Tread Wear and Carcass Failure Modelling

For HDM-III Watanatada, *et al*. (1987b) developed an integrated framework for considering tread wear and carcass failure. With minor modifications, this integrated model has been adopted for HDM-4.

The total distance travelled by a tyre carcass can be expressed as:

$$\text{DISTOT} = \text{DISNEW} + \text{NR DISRET} \qquad \ldots(\text{B5.14})$$

where	DISTOT	is the total distance in 1000 km travelled by a tyre carcass
	DISNEW	is the distance travelled in 1000 km when the tyre is new
	DISRET	is the distance travelled in 1000 km by a retreaded tyre
	NR	is the number of retreads for the tyre carcass

The tread wear rates of new and retreaded tyres can be expressed as:

$$\text{TWN} = \frac{1000}{\text{DISNEW}} \qquad \ldots(\text{B5.15})$$

$$\text{TWR} = \frac{1000}{\text{DISRET}} \qquad \ldots(\text{B5.16})$$

where	TWN	is the tread wear per 1000 km as a decimal for new tyres
	TWR	is the tread wear per 1000 km as a decimal for retreaded tyres

The total distance travelled can therefore be expressed as:

$$\text{DISTOT} = \frac{1}{\text{TWN}} + \frac{\text{NR}}{\text{TWR}} \qquad \ldots(\text{B5.17})$$

Expressing the tread wear rate of retreaded tyres as a fraction of the new tyre tread wear rate results in the following equation:

$$\text{DISTOT} = \frac{1 + \text{RTWR NR}}{\text{TWN}} \qquad \ldots(\text{B5.18})$$

where RTWR is the life of a retreaded tyre relative to a new tyre as a decimal ($\leq$ 1.0)

The tread wear can be replaced by the predicted volume of rubber loss from Equation B5.11 (in dm^3/1000 km) and the volume of wearable rubber on the tyre (in dm^3). So this equation can be reduced to[1]:

$$\text{DISTOT} = \frac{(1 + \text{RTWR NR})\,\text{VOL}}{\text{TWT}} \qquad \text{…(B5.19)}$$

The total tyre cost over the life of the tyre carcass is the sum of the new tyre cost and the cost of retreads, divided by the distance travelled:

$$\text{CTYRE} = \frac{\text{CTNEW} + \text{NR CTRET}}{\text{DISTOT}} \qquad \text{…(B5.20)}$$

where CTYRE is the total tyre cost per 1000 km
CTNEW is the cost of a new tyre
CTRET is the average cost of a retreaded tyre

Expressing the tyre cost as a fraction of the new tyre price this can be expressed as:

$$\text{EQNT} = \frac{1 + \text{RREC NR}}{\text{DISTOT}} \qquad \text{…(B5.21)}$$

where EQNT is the number of equivalent new tyres consumed per 1000 km
RREC is the ratio of the cost of retreads to new tyres (CTRET/CTNEW)

Substituting for DISTOT in Equation B5.19 gives the following equation:

$$\text{EQNT} = \frac{1 + \text{RREC NR}}{1 + \text{RTWR NR}}\,\frac{\text{TWT}}{\text{VOL}} \qquad \text{…(B5.22)}$$

The total tyre consumption for a vehicle is given by the product of EQNT and the number of tyres per vehicle, *ie*:

$$\text{EQNTV} = \text{NTV}\;\frac{1 + \text{RREC NR}}{1 + \text{RTWR NR}}\,\frac{\text{TWT}}{\text{VOL}} \qquad \text{…(B5.23)}$$

where NTV is the number of tyres per vehicle

This is the HDM-4 mechanistic tyre model[2].

---

1 The assumption here is that new and retreaded tyres have the same volume of rubber. There is no evidence to the contrary in the literature. See Bennett and Paterson (1999) for a discussion on how one calculates the volume of wearable rubber.

2 The same basic model was used for HDM-III but with two differences. Firstly, Watanatada, *et al.* (1987b) assumed that retreaded tyres had the same life as new tyres (RTWR=1.0), even though their data and text clearly showed that a value of 0.75 was more appropriate (see Table B5.1). It is postulated that this is because when they estimated the model parameters for TWT this included retreaded tyres. Secondly, the model had a bias correction error term of 0.0075 added to it to account for a non-linear transformation of linear prediction relationships (Watanatada, *et al*., 1987b). This term had a significant impact on the magnitude of the predictions and this may be one of the reasons for the relatively high predictions of the

B5.2.5 Tyre Consumption Research

TREAD WEAR

Although tyre consumption is a significant component of the total user costs, particularly for heavy vehicles, it has received much less attention in the literature than the other RUE components.

Claffey (1971) found that truck tyre consumption on a four-lane urban arterial with traffic signals was three times that on a controlled access road with free flowing traffic. However, curvature had an even greater effect with a 60° curve having five times the tyre consumption of a 30° curve. For passenger cars, curvature was also found to have a major impact on tyre consumption, with a 12° curve at 80 km/h having over 30 times the tyre wear over that on a straight road.

The effects of surface type was also tested by Claffey (1971). It was found that there was a 75 per cent increase in tyre wear on an asphalt pavement (wet coefficient of friction 0.48) over a concrete pavement (wet coefficient of coefficient 0.36). A hard gravel road covered with a think spreading of loose stones had a 460 per cent increase in the rate of tyre wear over the concrete pavement.

Chesher and Harrison (1987) describe the basis for the tyre consumption models developed as part of the Kenya, Caribbean, India and Brazil road user cost studies. Instead of conducting detailed experiments into tyre consumption, the data were collected through user surveys. The Brazil study monitored the history of individual tyres while the other studies focused on vehicles—essentially counting the number of tyres used by survey vehicles during the survey period and relating this to the kilometerage travelled by the vehicles. It is important to note that the studies had different types of tyres, for example in Brazil and India the passenger car results are for bias ply tyres whereas Kenya was mainly radial tyres. The Caribbean was a mixture of both.

Figure B5.4 illustrates the predicted tyre life as a function of roughness from these studies using the equations as given in Chesher and Harrison (1987). All models predict an increase in tyre consumption with increasing roughness, although there are significant differences in the slopes and magnitude of the predictions between studies. Hide (1982) suggests that the difference between the Caribbean and Kenya equations may be attributable to differences in tyre types or the presence of severely potholed pavements in the Caribbean.

The truck and bus models from Brazil and India included variables representing highway geometry. The predictions from both studies are similar insofar as increases in rise and fall or curvature serve to increase the tyre consumption. An increase in rise and fall from 20 to 50 m/km "increased tyre consumption by from 30 to 43 per cent for medium weight vehicles" (Chesher and Harrison, 1987). With regard to curvature, it is noted that in India vehicles did not experience large increases in tyre consumption with increasing horizontal curvature in part because of driver behaviour where "the curvature experienced by the vehicle [is] smaller than that measured in the survey as a result of drivers taking lines through corners which tend to straighten bends out. In addition, there are probably subtle speed effects" (Chesher and Harrison, 1987).

tyre consumption compared to the regression models developed from the same data (see Figure 11.3 and 11.4 in Watanatada, *et al.* (1987b)).

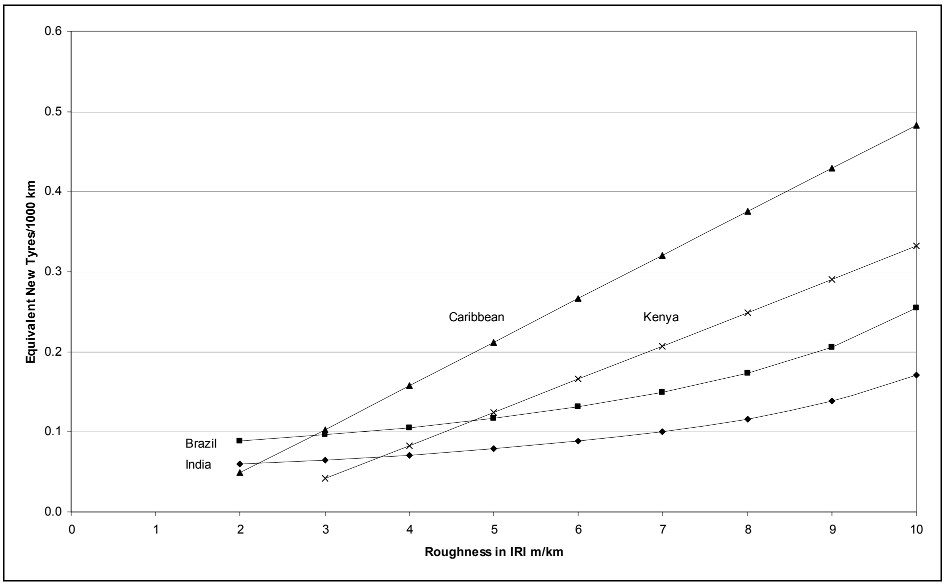

Figure B5.4: Effect of Roughness on Passenger Car Tyre Consumption

Du Plessis and Meadows (1990) operated three passenger cars on pre-selected routes on a daily basis for about three months. As shown in Figure B5.5, it was found that there was a significant difference between the paved and unpaved road tyre consumption. The results were compared to the tyre models from the different road user cost studies to confirm the most suitable model to adopt for South Africa.

Findlayson and du Plessis (1991) analysed costs for trucks from a forestry transport operation. The following equation was developed for predicting tyre life:

$$KMT = 166.47 - 31.83 \ln(13\ IRI) \qquad \ldots(B5.24)$$

where KMT is the tyre life in '000 km

This model shows very high tyre wear on rough roads which was "thought to be caused by the higher incidence of premature tyre failures and the increased ablative wear on unimproved logging roads" (Findlayson and du Plessis, 1991).

Traffic conditions can also have a significant impact on tyre consumption. For example, Thoreson (1993) reported a 23 per cent increase in tyre consumption for urban vs rural areas, even though the urban areas had lower speeds. This increase reflects the effects of stop-start conditions on tyre wear.

With regard to the slip-energy tyre consumption model, The amount of tread wear per distance travelled is determined by the pavement surface properties, tyre properties, vehicle load, and operating conditions. Zaniewski, *et al.* (1982) used the slip-energy model to update and extend the earlier empirical results into tyre consumption. This work as well as others was used as the foundation for the HDM-III slip-energy model. Watanatada, *et al*. (1987b) gave a review of the results of different researchers and present coefficients for the different model coefficients by vehicle and tyre type. Table B5.2 summarises these results for some different surface types.

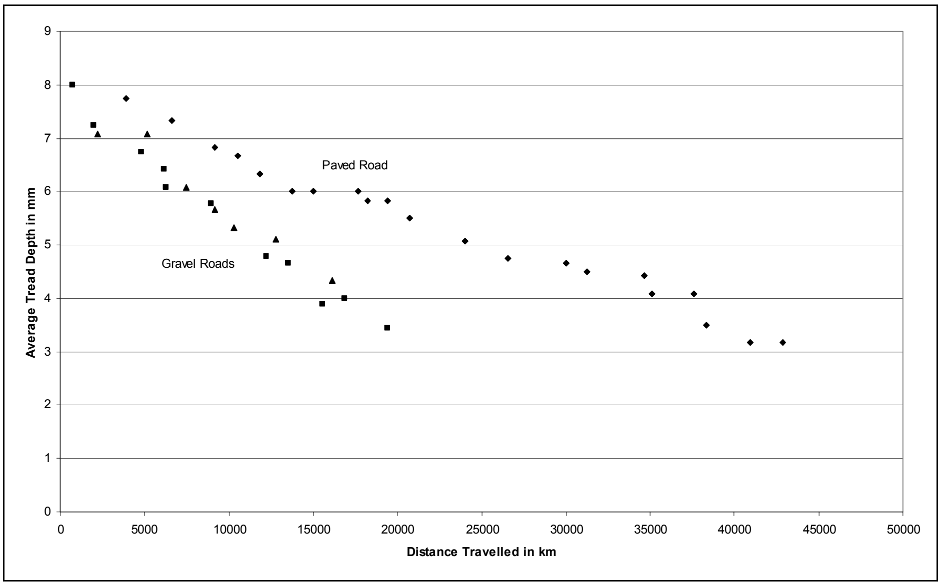

Figure B5.5: Effect of Distance Travelled on Tread Depth

Table B5.2: Summary of Representative Tyre Wear Coefficients

Surface Type	Tyre Type	Slip coefficient b/k	Slip Energy Coefficient K0	Tread Wear Coefficient K0 b/k
AC with crushed aggregate –	Car Radial 1	0.031	0.0275	0.00086
abrasion 20-25%	Car Radial 2	0.038	0.0219	0.00084
AC with crushed aggregate –	Car Radial 1	0.029	0.0131	0.00038
abrasion 26-29%	Car Radial 2	0.023	0.0174	0.00040
AC with uncrushed	Car Radial 1	0.024	0.0163	0.00028
aggregate – abrasion 35-37%	Car Radial 2	0.026	0.0105	0.00027
Open graded AC overlay	10.00-15 Truck Radial	0.077	0.0735	0.00566
Dense graded AC road mix	10.00-15 Truck Radial	0.091	0.0257	0.00252

Source: Watanatada, et al. (1987b)

For HDM-III, Watanatada, *et al.* (1987b) used Equation B5.12 as the basis for predicting tyre consumption. However, due to an inability to estimate the lateral forces with any confidence the model only contained the circumferential force. This model is given below:

$$TWT = C0tc + K0\frac{b}{k}\frac{CFT^2}{NFT} \qquad \text{...(B5.25)}$$

On the basis of a statistical analysis of the Brazil truck tyre data, the values for the coefficients were estimated which resulted in the following final model:

$$TWT = 0.164 + 0.01278\ \frac{CFT^2}{NFT} \qquad \text{...(B5.26)}$$

It was not possible to develop slip-energy model coefficients for passenger cars or light trucks using the Brazil data.

Zukang, *et al.* (1992) calibrated Equation B5.25 for China using data from 12 vehicle fleets in Shanghai and Beijing. As in Brazil, they found that the volume of rubber with retreads was 75 - 77 per cent of a new tyre. The coefficients were quantified by comparing the actual rubber volume with predicted from HDM-III. The resulting values are given in Table B5.3.

Table B5.3: Chinese HDM Truck Tyre Parameters

Tyre Size	Volume of Rubber (dm^3)	Constant C0tc	Tread Wear Coefficient K0 b/k	Base Number of Retreads
9.00/20	5.88	0.0684	0.0053	2.63
9.00R20	5.42	0.0684	0.0053	2.63
11.00/20	7.69	0.1335	0.0104	2.56

Source: Zukang, et al. (1992)

In comparing the HDM-III tread wear coefficient of 0.01278 against the previous values, Watanatada, *et al.* (1987b) noted that its magnitude was much higher than that reported in the previous studies (see Table B5.2). They postulated that this was due including non-survivors in the analysis and that tyres in Brazil were of lower quality than in the previous studies (*ie* cross-ply instead of radials). The Chinese data are lower than the Brazilian data and, for the 9.00 size tyres, agree well with the data in Table B5.2 for the 10.00 tyre size.

CARCASS WEAR

For HDM-III the following tyre carcass model was developed from the Brazil data (Watanatada, *et al.*, 1987b):

$$NR = NR0 \exp(-0.03224\ IRI - 0.00118 \min(300, CURVE)) - 1 \qquad \ldots(B5.27)$$

where NR0 is the base number of retreads for very smooth, tangent roads
CURVE is the horizontal curvature in degrees/km

The values for NR0 varied by vehicle type as follows:

1.93	9.00/20 tyres
3.39	10.00/20 tyres
4.57	11.00/22 tyres

Figure B5.6 shows the effect of roughness on the number of retreads for a horizontal curvature of 90 degrees/km. It will be noted that at high roughnesses the predictions go negative. Watanatada, *et al.* (1987b) indicate that this represents “carcass failure before the tyre is ready for the first retread”.

Cenek, *et al*. (1993) noted that that the predictions of this equation were consistent with N.Z. experience at lower roughnesses. They indicated that for typical driving conditions the average number of retreads suggest values for NR0 of four for cars and eight for trucks, although they noted a need for more detailed local calibration. These differences also likely reflect the higher quality tyres in use in N.Z. compared to Brazil (radial vs cross-ply tyres). Conversely, the data in Table B5.3 shows the base number of retreads from the Chinese study to be markedly lower than Brazil, suggesting poorer quality tyres, although the lower tread wear coefficients to not support this thesis.

An omission from this carcass model is in their failure to take into account consumer choice. Although it may be technically feasible to retread a tyre, there needs to be an economic incentive for users to purchase retreads. This decision is based on the relative cost of retreads to new tyres, their relative preformance, and other intangible factors such as consumer preference.

For example, although Cenek, *et al.* (1993) suggested a value of four for the base number of retreads of cars in N.Z., retreads only constitute approximately 20-25 per cent of new tyre sales. Thus, from a modelling perspective this lower value should be used in calculating the equivalent tyre consumption.

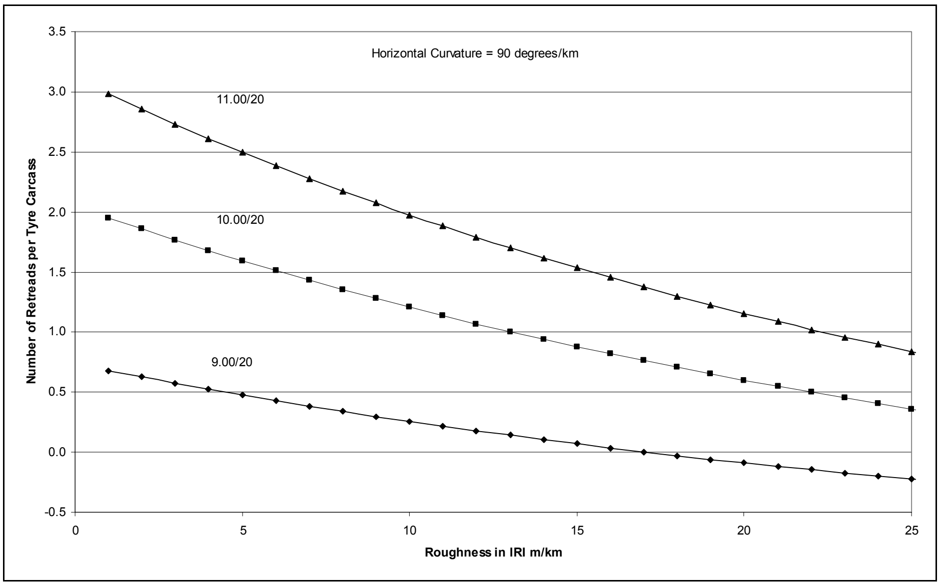

Figure B5.6: Effect of Roughness on Number of Retreads

B5.3 HDM-4 Tyre Consumption Model

B5.3.1 HDM-4 Mechanistic Tyre Model

MODEL FORMULATION

In common with HDM-III, the HDM tyre consumption model is based upon the principles of slip-energy. As described in Section B5.2.2, tyre slip is the circumferential motion of the tyre relative to the wheel rim. Slip-energy is the product of the total distance slipped by a tyre multiplied by the horizontal force on the tyre.

For HDM-4 the model has been extended to include horizontal curvature forces and also traffic interaction effects. This is done using the revised model formulation presented below Carpenter and Cenek (1999):

$$TWT = FLV\left(C0tc + \frac{Ctcte\ FNC\ CFT^2}{NFT} + \frac{Ctcte\ FNL\ LFT^2}{NFT} \right) \qquad \ldots(B5.28)$$

or,

$$TWT = FLV\left(C0tc + Ctcte\ TE \right) \qquad \ldots(B5.29)$$

where FLV is a factor for local effects, vehicle type, *etc.*
FNC is a factor representing the variation in circumferential forces
FNL is a factor representing the variation in lateral forces
TE is the tyre energy in MNm/1000 km

Since both acceleration and deceleration forces contribute to tyre wear, the terms CFT^2 and LFT^2 in Equation B5.28 represent the mean of the absolute values of CFT and LFT. The factors FNC and FNL were introduced by Carpenter and Cenek (1999) to account for the effects of random fluctuations, or noise, in the values of CFT and LFT. These changes are important since they allow the model to be used to predict the effects of traffic interactions on tyre consumption.

Cenek and Carpenter (2000) report that most tyre consumption is due to lateral as opposed to circumferential forces. In applying the model it was recommended that a limit be placed so that the lateral tread wear was a minimum of 2.5 the magnitude of the longitudinal tread wear.

The tyre consumption is calculated by substituting TWT into the integrated equation presented earlier:

$$EQNTV = NTV \frac{1 + RREC\ NR}{1 + RTWR\ NR} \frac{TWT}{VOL} \qquad \text{…(B5.30)}$$

The number of retreads is calculated using the following modified equation:

$$NR = \min(NRC, NR0 \exp(-0.03224\ IRI - 0.00118 \min(300, CURVE)) - 1) \qquad \text{…(B5.31)}$$

where NRC is the percentage of new tyres sold that are retreads as a decimal

Nordström and Andersson (1995) proposed modifying the HDM-4 TWT predictions by multiplying all terms by a temperature factor. This factor was to account for the “influence of temperature which is a significant factor that can differ considerably from country to country and also in different regions within one country”. It is calculated as[1]:

$$KTEMP = \frac{(1 + A_T\ (TAIR - T0))(1 - A_W\ WET)}{(1 - A_W\ W0)} \qquad \text{…(B5.32)}$$

where	KTEMP	is the temperature correction factor for tyre consumption
	A_T	is the representative air temperature coefficient
	A_W	is the representative wet road coefficient
	TAIR	is the air temperature in °C
	T0	is the reference air temperature in °C
	WET	is the representative percentage for wet roads
	W0	is the reference wet road percentage

Although Nordström and Andersson note that “the coefficients have to be determined experimentally”, they suggested a value of 0.005 for A_W and 0.025 for A_T, possibly based on Shallamach (1981). Using these values with assumed temperatures and wet road conditions for Sweden they suggest that there would be a 21.25 per cent increase in tyre consumption from a temperature increase between 20 and 25 °C.

Ignoring the wet road component, this equation supports the findings of Kadiyali and Ummat (1980) who found a five per cent increase in tyre consumption when there was an increase in air temperature from 28 to 30 °C. However, the data from Kadiyali and Ummat (1980) suggest that the relationship may in fact be non-linear. This is supported by the data from Gelling (1994), although the latter does not provide sufficient information to establish a transferable relationship between different tyre types.

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[1] Nordström and Andersson (1995) report an even more detailed formulation which includes speed and load factors, but these have been excluded for simplicity.

Temperature corrections using Equation B5.32 are not appropriate for typical applications of the HDM model since the model will cover extended periods of different environmental conditions. However, it is useful when calibrating the model since differences in the basic tyre consumption as predicted by HDM may be attributable to different ambient conditions when the model parameters were calibrated.

QUANTIFICATION OF TYRE MODEL PARAMETERS

Introduction

The HDM-4 tread wear model given in Equation B5.28 includes three additional parameters that were not in the HDM-III model: FLV, FNC and FNL. Otherwise, the model is identical to the basic HDM-III formulation.

Tyre Factor FLV

The factor FLV was introduced to reflect localised effects on tyre consumption which would not be embodied in the standard model parameters, such as local effects, vehicle type, road roughness, macrotexture, aggregate abrasion, tyre type, weather, regional effects, *etc.* It effectively serves as a 'rotation' calibration factor[1].

Carpenter and Cenek (1999) do not provide any details on values for FLV as a function of operating conditions so the value is set to 1.0 until further work is done in this area.

Speed Variation Factor FNC

The tyre factor FNC is a function of speed variations. If vehicles travel at a perfectly steady state speed the value is 1.0. As the frequency and magnitude of the speed variations increase, the value of FNC increases. This factor is calculated using the acceleration noise model presented in Chapters B3 and B4.

As described in Carpenter and Cenek (1999), the values for FNC were calculated using the same method as the additional fuel due to speed changes (see Chapter B4). For a range of acceleration noises the mean of CFT^2 was calculated and divided by CFT^2 for a constant speed. This gives the increase in CFT^2—and thus the tyre consumption—arising from speed changes.

As shown in Carpenter and Cenek (1999), speed changes have the greatest effects at low speeds due to the inertial effects and effective mass (see Chapter B1). The effects are also proportional to mass, with heavy vehicles having the greatest impacts. For motorcycles the value is approximately half that of a passenger car, which in turn is approximately half that of a heavy truck.

For use in HDM-4 the methodology was modified to calculate a parameter called 'dTYRE'. This is analogous to the dFUEL described in Chapter B4 for congestion fuel consumption. dTYRE represents the additional tyre consumption over steady state tyre consumption arising from speed changes.

dTYRE is calculated using the ACCFUEL program from the HDM Tools software. For each representative vehicle it calculates the tyre consumption using the HDM-4 model at a steady

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[1] As described by Bennett and Paterson (1999), calibration can be achieved by translation or rotation. A translation consists of adding a set amount to the predictions. This serves to shift the predictions while not influencing the impacts of the independent variables. Rotation factors are used to multiply the predictions and therefore alter the impact of the independent variable. For example, if a factor of 0.5 was used for FLV this would mean that the effects of roughness, speed *etc.* on tyre consumption would be halved.

state speed and then for a range of acceleration noises. This results in a matrix of values which is used to modify the tyre consumption at steady state speed as follows:

$$EQNTV = EQNTV\ (1 + dTYRE) \qquad \ldots(B5.33)$$

Figure B5.7 is an example of the dTYRE predictions for a medium truck as a function of the mean speed and the acceleration noise. The speed effects decrease with increasing speed due to the changes of the effective mass of the vehicle.

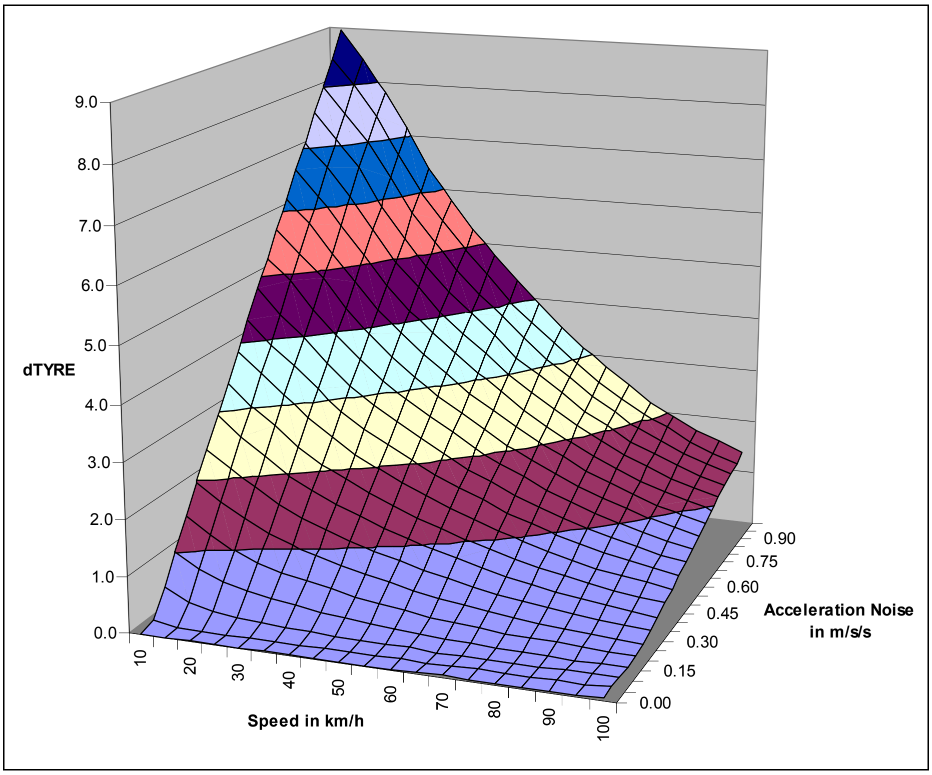

Figure B5.7: Effect of Speed and Accelerations on Tyre Consumption

Lateral Force Factor FNL

To estimate the value for FNL, Carpenter and Cenek (1999) used data from two routes with different severities in terms of the horizontal curvature (42 vs 425 degrees/km). Using appropriate values for FNC and TWT it was found that there was little variation in FNL between the routes, in spite of the large difference in rout severity. It was concluded that “FNL is likely to be relatively constant for many routes” and that “it is not likely to be significantly influenced by vehicle type”.

On the basis of their analysis it was proposed that a constant value of 2.5 be adopted for all routes and all vehicle types.

HDM-4 Model Parameters

On the basis of the previous work and their own experiments Carpenter and Cenek (1999) proposed the default tyre consumption model parameter values for the HDM-4 representative vehicles in Table B5.4. It will be noted that the values for Ctcte are markedly lower than those used in HDM-III. This reflects the different tyre technologies and the values are more consistent with other researchers than those in HDM-III.

Table B5.4: Carpenter and Cenek (1999) Tyre Consumption Model Parameters

Representative Vehicle	Description	C0tc (dm^3/1000 km)	Ctcte (dm^3/MNm)
1	Motorcycle	0.001	0.0009
2	Small Car	0.001	0.0005
3	Medium Car	0.001	0.0005
4	Large Car	0.001	0.0005
5	Light Delivery Vehicle	0.001	0.0005
6	Light Goods Vehicle	0.001	0.0005
7	Four Wheel Drive	0.001	0.0005
8	Light Truck	0.001	0.0003
9	Medium Truck	0.001	0.0003
10	Heavy Truck	0.001	0.0003
11	Articulated Truck	0.001	0.0003
12	Mini Bus	0.001	0.0003
13	Light Bus	0.001	0.0003
14	Medium Bus	0.001	0.0003
15	Heavy Bus	0.001	0.0003
16	Coach	0.001	0.0003

Source: Carpenter and Cenek (1999)

However, when testing the model, the values for C0tc were found to be too low and resulted in unreasonably high tyre lives. Alternative values were quantified using estimated tyre lives. The tread wears were calculated using typical road conditions and tyre lives proposed by Cenek and Carpenter (2000). The mechanistic forces were calculated using the default HDM-4 model parameters for each of the representative vehicles. While reasonable parameters were obtained for some vehicles, it was not possible to use them with absolute confidence.

Further work is therefore required to establish model parameters which will give sensible predictions with the mechanistic tyre model. Until this is done, the interim solution described in the following section was adopted for HDM-4.

B5.3.2 Interim HDM-4 Tyre Model

Due to the problems with the proposed new mechanistic tyre model, an interim model was adopted for HDM-4 v 1.0 based on the HDM-III model (Watanatada, *et al.*, 1987a). The mechanistic theory presented earlier resulted in Equation B5.21 which was presented earlier. In HDM-III a bias correction factor of 0.0027 was added to the equation. This resulted in the following equation:

$$EQNT = \frac{1 + RREC\ NR}{DISTOT} + 0.0027 \qquad \text{...(B5.34)}$$

The number of retreads is calculated using the equation (Odoki and Kerali, 1999b):

$$NR = \max\,[0, NR0\, \exp(-0.03224\ RImod) - 1] \qquad \text{...(B5.35)}$$

where RImod is the modified roughness (see below)

The rate of tread wear is calculated as:

$$TWT = C0tc + Ctcte\ TE \qquad \text{...(B5.36)}$$

where TE is the tangential energy in J-m

The tangential energy, circumferential, laterial and normal forces on the tyres are calculated as:

$$TE = \frac{CFT^2 + LFT^2}{NFT} \quad \text{...(B5.37)}$$

$$CFT = \frac{(1 + CTCON\ dFUEL)(Fa + Fr + Fg)}{NUM\_WHEELS} \quad \text{...(B5.38)}$$

$$LFT = \frac{Fc}{NUM\_WHEELS} \quad \text{...(B5.39)}$$

$$NFT = \frac{M\ g}{NUM\_WHEELS} \quad \text{...(B5.40)}$$

Table B5.5 gives the parameter values for use with this tyre model for each representative vehicle.

Table B5.5: Tyre Model Parameter Values

Vehicle Number	Vehicle Type	NR0	C0tc	Ctcte	VOL (dm^3)	VEHFAC	RImod
1	Motorcycle	1.30	0.00639	0.00050	0.35	2	RIav
2	Small Car	1.30	0.02616	0.00204	1.40	2	RIav
3	Medium Car	1.30	0.02616	0.00204	1.40	2	RIav
4	Large Car	1.30	0.02616	0.00204	1.40	2	RIav
5	Light Delivery Vehicle	1.30	0.02400	0.00187	1.60	2	RIav
6	Light Goods Vehicle	1.30	0.02400	0.00187	1.60	2	RIav
7	Four Wheel Drive	1.30	0.02400	0.00187	1.60	2	RIav
8	Light Truck	1.30	0.02400	0.00187	1.60	2	RIav
9	Medium Truck	1.30	0.02585	0.00201	6.00	1	min(7, RIav)
10	Heavy Truck	1.30	0.03529	0.00275	8.00	1	7
11	Articulated Truck	1.30	0.03988	0.00311	8.00	1	min(7, RIav)
12	Mini-Bus	1.30	0.02400	0.00187	1.60	2	Riav
13	Light Bus	1.30	0.02173	0.00169	1.60	2	Riav
14	Medium Bus	1.30	0.02663	0.00207	6.00	1	7
15	Heavy Bus	1.30	0.03088	0.00241	8.00	1	min(7, RIav)
16	Coach	1.30	0.03088	0.00241	8.00	1	min(7, RIav)

Source: Odoki and Kerali (1999b)

The distance travelled by the tyre carcass is given by the following equation:

$$DISTOT = \frac{(1 + NR)\ VOL}{TWT} \quad \text{...(B5.41)}$$

The total tyre consumption is given by the equation:

$$TC = \frac{NUM\_WHEELS\ EQNT}{MODFAC} \quad \text{...(B5.42)}$$

where MODFAC is a tyre life modification factor

The tyre life modification factors were proposed by Harrison and Aziz (1998) to depend upon the roughness, tyre type and congestion level. It is calculated as:

$$MODFAC = VEHFAC\ TYREFAC\ CONGFAC \quad \text{...(B5.43)}$$

where VEHFAC is a vehicle specific modification factor
TYREFAC is a tyre type modification factor

CONGFAC is a congestion modification factor

The values of congestion effects factor (CONGFAC) to be used in Equation B5.43 is a function of the volume to capacity ratio, as follows (Harrison and Aziz, 1998):

CONGFAC = 0.7 VCR < 0.85

CONGFAC = 1.0 VCR $\geq$ 0.85

The vehicle factor is given in Table B5.5; the tyre factor in Table B5.6.

Table B5.6: Tyre Type Correction Factors

Tyre Type	Paved roads	Unpaved Roads	
		IRI ≤ 6 m/km	IRI > 6 m/km
Bias	1.00	1.00	1.00
Radial	1.25	1.20	1.00

Source: Harrison and Aziz (1998)

Although this is an interim solution, there are a number of unsatisfactory issues with the above model:

- The parameter values given in Table B5.5 were not based on a rigorous re-estimation or theoretical quantification;
- The basic model form does not address the deficiencies identified and rectified in Section B5.3; and,
- The modification factors (MODFAC) are not based on a theoretical analysis and also result in discontinuities in the predictions.

Accordingly, the model's predictions should be viewed with caution until a more rigorous model is implemented.

B5.4 Summary

The tyre consumption modelling in HDM-4 still leaves much to be desired. Although the mechanistic theory of tyre consumption is well founded and has the flexibility required by a model such as HDM-4, the full implementation has not been done due to an inability to obtain satisfactory model parameters for use with the model. In its place, a model based on HDM-III but using modification factors has been implemented. This model is not entirely satisfactory as the factors adopted will lead to discontinuities in the predictions.

It is therefore imperative that further work be done to develop suitable parameters to apply the full theoretical model presented here. Until this is done, the predictions from the HDM-4 model for tyre consumption should be viewed with caution.

B6 Maintenance and Repair Costs

B6.1 Introduction

Vehicle maintenance and repair costs (hereafter called maintenance costs) are comprised of two components: parts consumption and labour hours. Maintenance costs are often a significant contributor to the benefits from road improvements, in HDM-III constituting up to 80 per cent of the total benefits for certain types of projects.

HDM-III contained separate relationships for parts and labour. The Brazil parts consumption relationship was a function of the roughness and the vehicle age, expressed as the cumulative kilometreage, while the labour hours was a function of the parts consumption and roughness.

This chapter discusses the maintenance model adopted for HDM-4 and presents coefficients for its application.

Box B6.3 Terminology

The following terminology is used in this chapter:

- **Maintenance cost**: The total maintenance and repair cost for a vehicle including parts consumption and labour costs.
- **Standardised Parts Costs**: The parts consumption expressed as a percentage of the replacement vehicle price.

B6.2 Modelling Maintenance Costs

Of all the VOC components, maintenance costs are both difficult to measure empirically and to predict. This is because:

- The costs usually arise infrequently over the life of the vehicle, particularly for larger components;
- The maintenance practices of the owners/operators have a major impact on the costs, for example Chesher and Harrison (1987) note that the variation in costs between operators was greater than that due to different road conditions;
- The maintenance costs for similar vehicles can vary significantly between manufacturers;
- Vehicles operating in harsh conditions may be of more robust construction and therefore have lower maintenance costs than standard vehicles and,
- Associating maintenance costs to operating conditions is difficult since vehicles tend to operate over a range of roads so the costs are averaged out.

There is a relationship between maintenance costs, capital costs, and vehicle service life. The 'Optimal Life' (OL) approach, presented in Chapter B7, shows that the optimal time for scrapping a vehicle is a function of the maintenance costs. This in turn dictates the total capital costs. However, the models developed for predicting maintenance costs have been developed in isolation from the capital cost models. Thus, they include operators who may have limited maintenance expenditure but a high rate of depreciation and those which practice a high level of preventative maintenance and therefore minimise vehicle downtime and reduce the rate of depreciation.

There are also areas where distortions can arise. For example warranty periods—which in developed countries may span three years or more—taxation and depreciation regulations may cause operators to minimise their financial costs. Another situation which arises is when organisations with in-house labour may not have their time or full costs assigned to the repair task.

As will be shown, there are also major regional differences in maintenance and repair costs. One reason for this is the cost of labour which dictates whether a part is replaced or repaired. In countries such as India where labour is relatively inexpensive relative to the cost of parts there is a tendency to repair rather than replace. The balance between labour and parts depends on many factors including, but not limited to, the relative cost of labour, rates of taxation, import duties, mark-ups, tax liabilities, *etc*.

Because of these issues, the practice has been to separate labour costs from parts consumption. The labour costs are predicted in terms of the number of labour hours required. This is then multiplied by the cost of labour in cost/h. Parts consumption is expressed in terms of the standardised parts costs. This is the parts consumption expressed as a percentage of the replacement vehicle price. This approach was adopted after the Kenya User Cost Study when it was found that parts costs were strongly correlated to replacement vehicle price. This has the advantage of making the models more transferable to different environments, although Chesher and Harrison (1987) note:

> *"…this is to a large extent a cosmetic operation since there is little reason to expect there to be a static relationship between vehicle prices and maintenance costs transferable across environments even under common highway conditions…*

Du Plessis and Schutte (1991) note the difficulties in establishing maintenance cost models:

> *"With the exception of fuel consumption, VOC models are generally developed from aggregate-empiric relationships, in which a large number of observations collected from transport operators are regressed against certain road condition parameters (such as road roughness). There is almost always a large scatter in data of this nature, since effects other than those modelled are inevitably present. For example maintenance parts data for a fleet of vehicles will exhibit not only vehicle age and road condition effects, but also influences such as maintenance policies, driver behaviour and the trade-offs between capital and labour exercised by the operator.*

For this reason the calibration of the maintenance cost model is an essential element of any HDM analysis.

B6.3 HDM-III Maintenance Model

HDM-III allowed users to predict VOC using relationships derived from road user cost studies in Brazil, India, Kenya and the Caribbean. The Brazil relationships were the ‘standard’ relationships in HDM-III and Bennett (1995a), in a review of experiences applying the HDM-III model, shows that they were the ones applied in most studies. The discussion here will focus on the Brazil and India models since they are most relevant to HDM-4.

BRAZIL MODEL

The Brazil relationships have the following form[1] (Watanatada, *et al.*, 1987a):

$$PARTS = C0SP\ CKM^{kp} \exp(CSPIRI\ IRI) \quad \text{for } IRI \leq IRI0SP \quad \ldots(B6.1)$$

$$PARTS = CKM^{kp} (a0 + a1\ IRI) \quad \text{for } IRI > IRI0SP \quad \ldots(B6.2)$$

$$a0 = C0SP \exp(CSPIRI\ IRI0SP)(1 - CSPIRI\ IRI0SP) \quad \ldots(B6.3)$$

$$a1 = C0SP\ CSPIRI \exp(CSPIRI\ IRI0SP) \quad \ldots(B6.4)$$

$$LH = C0LH\ PARTS^{CLHPC} \exp(CLHIRI\ IRI) \quad \ldots(B6.5)$$

where
- PARTS is the standardised parts consumption as a fraction of the replacement vehicle price per 1000 km
- CKM is the vehicle cumulative kilometreage in km
- IRI is the roughness in IRI m/km
- IRI0SP is the transitional roughness beyond which the relationship between parts consumption and roughness is linear
- C0SP is the parts model constant
- CSPIRI is the parts model roughness coefficient
- LH is the number of labour hours per 1000 km
- C0LH is the labour model constant
- CLHIRI is the labour model roughness coefficient

Table B6.1 lists the model parameters for use with the HDM-III parts and labour models (Watanatada, *et al.*, 1987a).

Table B6.1: HDM-III Maintenance Model Parameters

Vehicle Class	Parts Model Parameters				Labour Model Parameters		
	kp	C0SP ($x 10^{-6}$)	CSPIRI ($x 10^{-3}$)	IRI0SP	C0LH	CLHPC	CLHQI
Passenger Car	0.308	32.49	178.1	9.2	77.14	0.547	0
Utility	0.308	32.49	178.1	9.2	77.14	0.547	0
Large Bus	0.483	1.77	46.28	14.6	293.44	0.517	0.0715
Light and Medium Truck	0.371	1.49	3273.27	0	242.03	0.519	0
Heavy Truck	0.371	8.61	459.03	0	301.46	0.519	0
Articulated Truck	0.371	13.94	203.45	0	652.51	0.519	0

Source: Watanatada, et al., (1987a)

The structure of the parts model is quite complicated. This is because while trucks were found to have a linear response to roughness, passenger cars, utilities and buses had an exponential response. To prevent unreasonable predictions, above a certain roughness (IRI0SP) a linear relationship was adopted above that level.

The vehicle cumulative kilometreage (CKM) was calculated as half the lifetime kilometreage (0.5 LIFE0 AKM0). However, as noted by Cox (1994), this led to a bias in the predictions since utilisation is a function of age. For this reason the HDM-4 allows the user to define an age and utilisation distribution which is used to establish the cumulative kilometreage.

1 The relationships have been converted from using QI units to IRI using 1 IRI = 13 QI.

The review of experiences with HDM-III (Bennett, 1995a) found that few studies calibrated the maintenance model. This is undoubtedly due to the difficulties in gathering sufficient data for model calibration.

Figure B6.1 illustrates the predictions of the HDM-III parts model for vehicles of age 100,000 km. It can be observed that there is a significant difference in the predicted effects of roughness between the various vehicle classes. Passenger cars and utilities are very sensitive to roughness, particularly above 6 IRI m/km, while buses are relatively insensitive to roughness. The sensitivity of passenger cars and utilities to roughness has been of concern to a number of analysts (Bennett, 1995a). For example, an increase in roughness from two to 10 IRI m/km parts consumption increases by a factor of 4.1 for passenger cars as opposed to only 1.5 for buses.

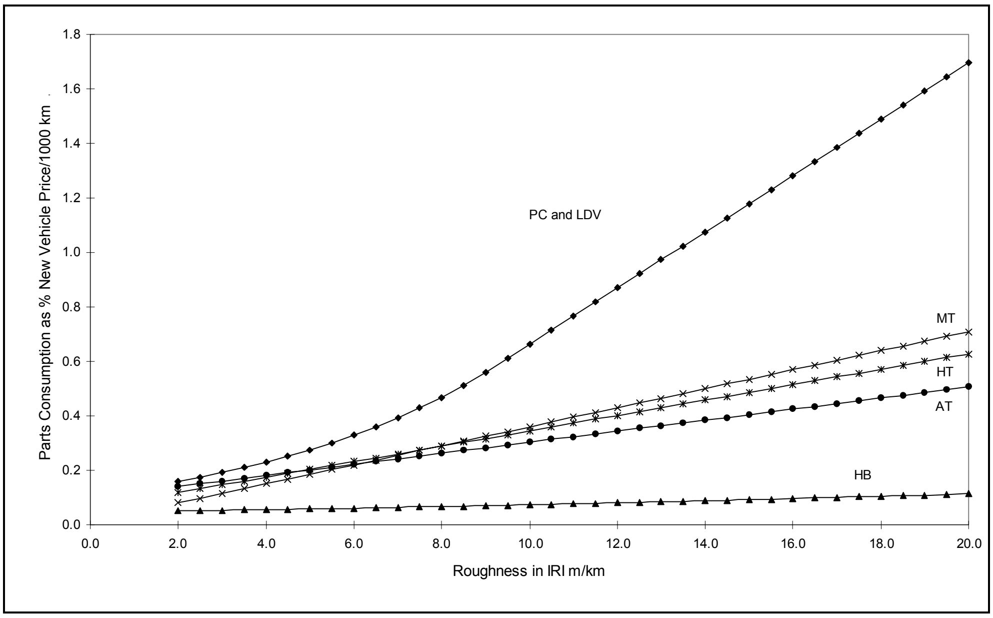

Figure B6.1: Effect of Roughness on HDM-III Parts Consumption

Figure B6.2 shows the effect of vehicle age on the HDM-III parts consumption at a roughness of 3 IRI m/km. As with roughness, passenger cars and utilities are the most sensitive to age; buses the least sensitive. For an increase in age from 50,000 to 200,000 km parts consumption rises by a factor of 1.5 for passenger cars and utilities, 1.7 for trucks and 2.0 for buses.

During the development of HDM-III an attempt was made to model parts consumption mechanistically (Watanatada, 1983; Dhareshwar, 1983). The analysis postulated a simple model of the form:

$$\text{PARTS} = \text{a0} + \text{a1 SPE} + \text{a2 PPE} \qquad \ldots(\text{B6.6})$$

where	a0 to a2	are constants
	SPE	is the suspension energy
	PPE	is the propulsive energy

The model was modified to take into account age as:

$$\text{PARTS} = \text{a0} + (\text{a1 SPE} + \text{a2 PPE})\ \text{CKM}^{\text{a3}} \qquad \ldots(\text{B6.7})$$

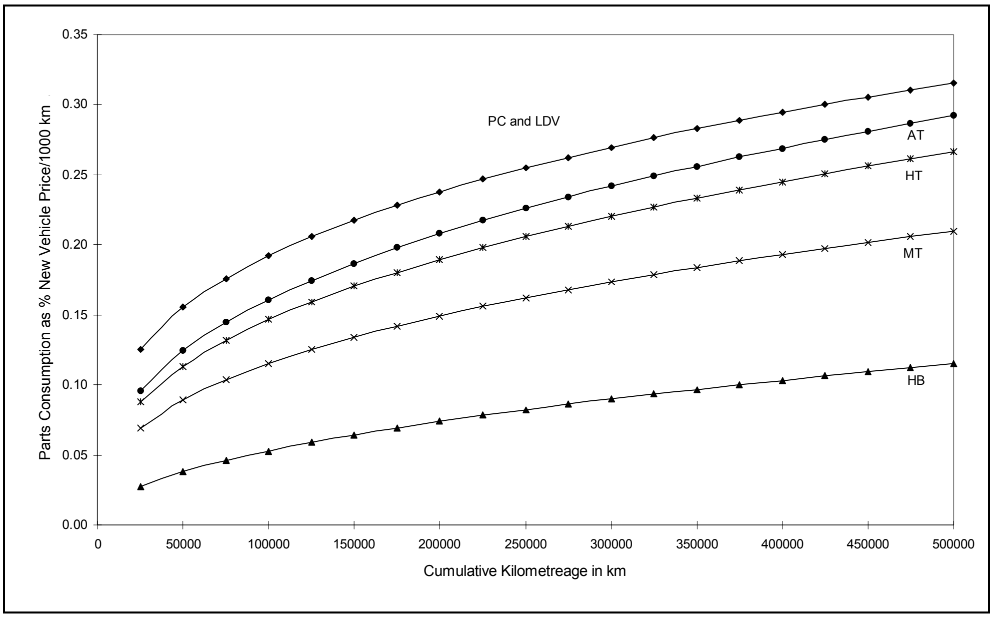

Figure B6.2: Effect of Vehicle Age on HDM-III Parts Consumption

Dhareshwar (1983) fitted the model using bus and truck data from Brazil and obtained similar, or better, statistics than were obtained using the simple Brazil model shown earlier. However, it was noted that the predictions were very sensitive to the propulsive energy term - particularly when there were gradients. Since this work was not continued, it can only be assumed that the problems with this formulation outweighed its benefits.

It should be noted that the same data used to develop the HDM-III maintenance models were used by GEIPOT (1981) to develop regression models. These models differ slightly from HDM-III in that the predictions extend to lower roughnesses (below IRI = 3) and the linear extension at high roughnesses.

INDIA MODEL

Watanatada, *et al.* (1987a) provided a set of maintenance relationships developed from data collected in the India User Cost Study. These models are different to those developed by the India RUC Study (CRRI, 1982) and include less variables.

Spare parts

There were three different model forms in the Indian relationships for standardised parts costs, one for cars and utilities, one for buses and one for trucks. There are similarities between these relationships and those for Brazil but there are also important differences.

For cars and utilities, for example, there was no dependence on vehicle age and consumption was simply an exponential function of roughness. For buses, parts consumption was the product of two functions, total travel raised to the power of 0.359 (broadly similar to Brazil) and an exponential function of roughness, curvature, gradient and road width. The relationship for trucks was similar to that for buses except that the exponential term also includes mass.

Maintenance labour

There are four different models for maintenance labour, one for cars, one for utilities, one for buses and one for trucks. The models have a similar form to Brazil. Labour hours are proportional to standardised parts cost raised to a power of around 0.5. In the case of buses and trucks, the labour hours are also multiplied by an exponential function of road roughness.

B6.4 Other Maintenance and Repair Models

B6.4.1 Introduction

In spite of the importance of maintenance costs in economic appraisals, there have been few studies conducted into this cost component. This section summarises the major studies.

B6.4.2 USA Research

Winfrey (1969) presented maintenance costs based on the results of surveys. These were updated by Claffey (1971) using an approach which has been termed the **constituent component** approach.

A list of parts requiring non-routine maintenance was assembled along with the distance travelled before the maintenance was required. Weighting these by the cost of repairs one can establish an average cost. Table B6.2 shows the results of this method for a "standard size" passenger car (Claffey, 1971). This suggests a cost of $0.0072/km, or $7.2 per 1000 km.

Table B6.2: Average Maintenance Costs Using Constituent Component Approach

Vehicle Part	Distance Travelled Before Repairs ('000 km)		Cost of Repairs ($)		Average Cost (cents/km)
	Range	Mean	Range	Mean	
Automatic transmission	86-165	106	100-255	178	0.17
Engine block	80-144	112	65-130	93	0.08
Shock absorbers	48-96	70	28-51	37	0.05
Brake system	64-123	86	40-58	41	0.05
Distributor	16-34	22	5-20	12	0.05
Exhaust	48-90	62	18-33	26	0.04
Carburettor	51-96	72	21-40	29	0.04
Universal	48-86	70	20-31	28	0.04
Rear axle	160-181	170	54-75	66	0.04
Generator	67-96	83	18-40	32	0.04
Water pump	54-88	69	18-30	24	0.03
Springs	64-160	109	28-46	40	0.04
Fuel pump	70-102	83	12-20	15	0.02
Oil Pump	147-221	174	16-28	21	0.01
Radiator	106-154	122	10-25	16	0.01
Fan belt	64-109	82	3-6	4	0.00
Total					0.72

Source: Claffey (1971). Converted to km from miles.

Papagiannakis (1999) gives the results of a study into heavy truck parts consumption. Figure B6.3 shows the effects of age on maintenance cost as well as the percentage of the cost due to labour. It will be noted that there is a significant increase in the maintenance costs with age and at the same time the percentage of the costs due to labour decreases. This indicates, not unexpectedly, an increase in the number of parts replaced as the vehicle ages.

One finding of this study which is at variance with other studies is that roughness effects manifest themselves on parts consumption even at low roughnesses (< 2 IRI m/km). This is possibly due to inaccuracies in estimating the roughness of the roads which the vehicles were operated on.

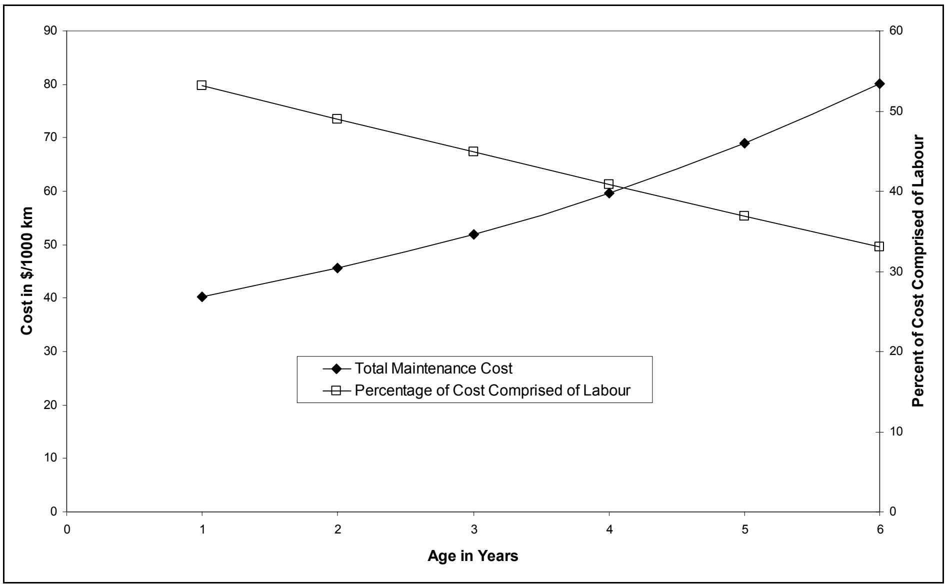

Figure B6.3: Effects of Age on Cost and Labour Component

Sime, *et al.*, (2000) presents the results from analysing truck data from a test track. Four heavy trucks were operated automatically and their costs monitored over a period of eight weeks before and seven weeks after a rehabilitation. The roughnesses before and after the rehabilitation were not reported.

It was anticipated that the increased roughness would be reflected through an increase in the frequency of failure of the truck and trailer components, for example through fractures of frames. As shown in Figure B6.4, there was a significant reduction in the failure rate of trailer springs after the pavement rehabilitation, reducing to almost the level of when the vehicles and pavement were new.

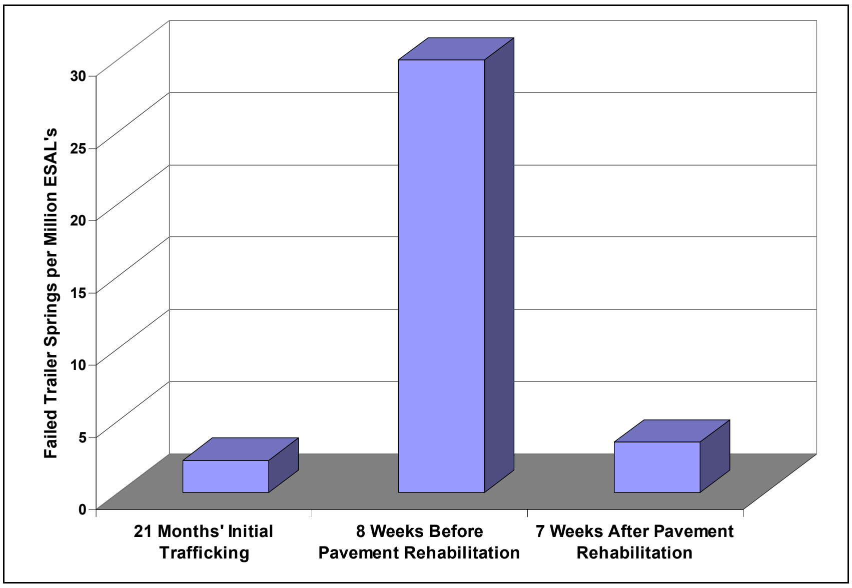

Figure B6.4: Effect of Rehabilitation on Trailer Spring Failure

B6.4.3 South African Research

As part of their considerable studies into VOC, the CSIR in South Africa developed models for parts consumption. The research can be grouped into two areas: that orientated at speed effects and that at roughness effects.

SPEED EFFECTS ON PARTS CONSUMPTION

As described by du Plessis editor (1989), the speed effects cost research saw maintenance costs divided into routine and non-routine maintenance components. The sum of these two was the total maintenance costs. This total cost was then modified to account for operating conditions, principally speed.

For cars, the routine costs were generally assumed to be those corresponding to the manufacturers recommended maintenance policy. The non-routine costs were then either assumed to be a fixed percentage of the routine costs or calculated based on estimates of the distance life of various components on the vehicle.

Initially, the speed effects were estimated based on a third order polynomial regression of the Winfrey (1969) costs. However, this was later superseded by an approach which divided the costs into the following categories:

- Those distance based and thus independent of speed;
- Those mainly influenced by time;
- Those mainly influenced by road speed;
- Those mainly influenced by engine speed; and,
- Those mainly influenced by stop/go and speed change cycles.

Routine maintenance costs were distance based. The time based costs were converted to their present worth and divided by the utilisation to obtain a per km cost. Since the utilisation can be a function of speed, these were also to some degree influenced by speed. The costs directly influenced by road speed were assumed to be linearly proportional to speed. The engine speed costs were those due to the power requirements of the engine and were assumed to be proportional to fuel consumption[1]. All brake and clutch repairs were assumed to be due to speed changes.

The total costs were then predicted using a polynomial of the following form[2]:

$$PCST = a1 + a2\ S + \frac{a3}{S} + a4\ S^2 \qquad \ldots(B6.8)$$

where PCST is the maintenance cost in cents/km
a1 to a4 are constants

The key problem with applying this method is the sensitivity of the results to the assumed average speed of the vehicle. du Plessis editor (1989) shows that there are significantly

1 It was noted by du Plessis editor (1989) that making these costs proportional to fuel consumption resulted in their overestimation at low speeds. It was recommended that they instead by made proportional to the power requirements as dictated by the magnitude of the forces opposing motion.

2 As shown in Chapter B4, this is the same formulation as many empirical fuel models.

different results if one assumes that the costs apply at a low urban speed instead of a higher rural speed.

ROUGHNESS EFFECTS ON PARTS CONSUMPTION

du Plessis and Meadows (1990) operated three identical rental cars over preselected routes, two with unsealed and one with a sealed surface, for 40,000 km. Marked differences were observed in the maintenance costs on the unsealed versus sealed roads. It was found that there was damage both due to roughness and also the unsealed surface. The small sample size did not permit for a relationship between parts consumption and roughness to be developed. However, the authors noted that the costs were similar to what would arise with the Brazil model at that roughness.

du Plessis, *et al.* (1989) used 12 months of cost records from a single bus operator running 740 vehicles on wide ranging conditions to develop relationships. This led to the development of the following model for predicting parts consumption:

$$PARTS = \exp(-0.5254 + 0.6779\ ln\ CKM + 0.3338\ ln\ (13\ IRI)) \times 10^3 \qquad \text{...(B6.9)}$$

They were not successful at establishing robust labour hours models.

An interesting aspect to this study was the effect of rebuilding of vehicles which resulted in a discontinuity in the parts-age relationship at the time of rebuilding.

Findlayson and du Plessis (1991) studied five separate forestry operations which had cost data. This led to the development of the following models:

$$PARTS = \exp(-3.0951 + 0.4514\ ln\ CKM + 1.2935\ ln\ (13\ IRI)) \times 10^3 \qquad \text{...(B6.10)}$$

The results of these studies along with the original Brazil research were used by du Plessis and Schutte (1991) to recommend the above models for buses and trucks in South Africa. The other vehicle classes had their costs predicted with the original Brazil models.

LABOUR MODELS

The South African research led to the development of the following labour hours relationships (du Plessis and Schutte, 1991):

Buses

$$LH = 0.763 \exp (0.0715\ IRI) \left(\frac{PARTS}{NVPLT}\right)^{0.517} \qquad \text{...(B6.11)}$$

Trucks

$$LH = \max(3, -0.375 + 0.017 \left(\frac{PARTS}{NVPLT}\right) + 0.182\ IRI) \qquad \text{...(B6.12)}$$

where NVPLT is the replacement vehicle price less tyres

B6.4.4 New Zealand Research

The New Zealand vehicle operating cost model (Bennett, 1989a) used the HDM Brazil relationships for spare parts calibrated for local use by the addition of a constant term—*ie* a translation calibration. The cost of maintenance labour was then calculated as the product of parts cost and the factor 55/45.

Opus-Beca (1998) describe the results of a study into maintenance costs using various sources of data, mainly commercial fleet databases. It was found that:

- The mean parts costs for light vehicles were within 20 per cent of those predicted by the HDM-III model at low roughnesses; medium trucks were 40 per cent higher and heavy trucks 100 per cent higher;
- The maintenance costs were constant over the first 3-4 years of vehicle life before increasing;
- The proportion of maintenance costs due to routine servicing as opposed to specific maintenance varied over the life of the vehicle. They decreased from 75 per cent for new cars to less than 20 per cent for cars seven years or older;
- Parts cost constituted 44 - 61 per cent of the total maintenance costs, with this percentage not varying greatly with vehicle age;
- There was evidence of body type influencing maintenance costs: for example, tanker trucks were found to have 75 per cent higher costs than bulk material costs;
- It was not possible to differentiate costs between urban and rural roads or by road condition. This was due to limitations in the data instead of indicating that these effects do not exist.

Cenek and Jamieson (1999) investigated roughness effects using an medium truck equipped with accelerometers on the body and suspension. The vehicles were driven at different speeds over sites with different roughnesses and the accelerations were recorded. Figure B6.6.5 is an example of the body accelerations versus roughness for medium trucks[1]. It can be seen that the body accelerations increase with increasing roughness and this can be used to calibrate the roughness effects in the HDM parts consumption model. It will also be noted that at lower roughnesses the accelerations do not have much of an impact. This can be used to confirm the roughness level below which there is no impact on parts consumption.

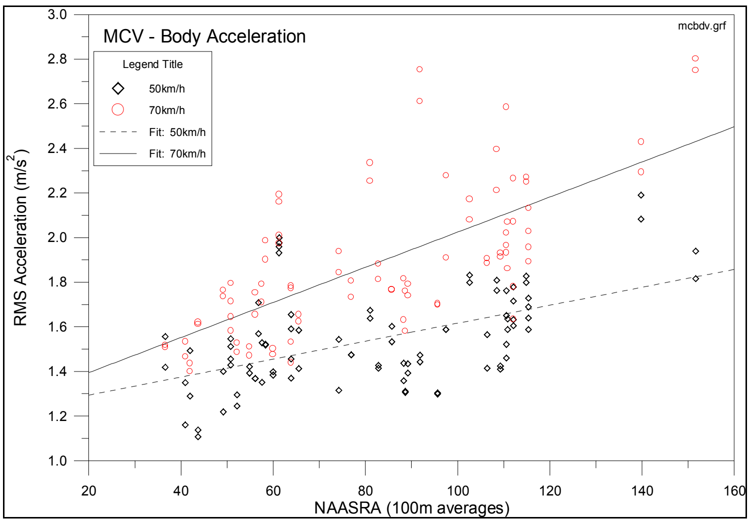

Figure B6.6.5 Effect of Roughness on Body Acceleration

[1] The roughness units in this figure are NAASRA counts/km. A conversion to IRI is 1 IRI = 26.4 NAASRA counts/km.

It was found that body accelerations were closely aligned to the roughness effects from the HDM-III parts consumption model and the axle accelerations with the HDM-4 parts consumption model. The results suggested that the low roughness cut off for parts consumption used with HDM-III and in HDM-4 was not appropriate, at least for trucks, since accelerations were being transmitted to the body even at roughnesses of 2.3 IRI m/km. It was also found that there were speed effects and it was proposed that a combination of speed and roughness—*ie* the roughness expressed as IRI m/h instead of IRI m/km[1]—would be a better predictor of parts consumption. This led Cenek and Jamieson (1999) to suggest that the:

> "*...benefits of roughness reduction in high speed environments may be considerably underestimated*."

One finding of this research which is not considered in HDM is the speed-roughness dependence. The results clearly show that the higher the speed, the greater the impact of roughness on body accelerations. This was something also suggested by Paterson and Watanatada (1985) who recommended that the average rectified velocity is a better determinant of speed than average rectified distance.

B6.4.5 Swedish Research

The VETO model was developed by the Swedish Road and Traffic Research Institute (VTI) (Hammarström and Karlsson, 1987). It contains two approaches to the calculation of spare parts and maintenance labour, one empirical and one mechanistic. The former relies on the HDM Brazil relationships while the latter employs a "wear index" for vehicle components.

The mechanistic model is a detailed simulation of an idealised two-dimensional vehicle travelling over a surface with a specified profile. The model works on the basis that the wear and tear of components depends upon the product of the number of stress cycles they have been subjected to and the stress amplitude raised to the sixth power. The number of cycles is assumed to be constant per unit length of road (independent of roughness) while the stress amplitude for each component is proportional to the RMS value of the dynamic component of the wheel load. The model does not take into account the static load.

The model has been calibrated by looking at the life expectancy of different components. Only four components have been studied and so the model does not yet provide a total cost calculation. Nevertheless, it is interesting to note that the change in vehicle wear with increasing roughness that it calculates is far higher than the change in parts cost predicted by the empirical model.

In spite of developing a complex model, Hammarström and Karlsson (1991) concluded that:

> *"... it would probably be virtually impossible to develop a model which could be used for calculating the relationship between total repair costs and road unevenness, component by component."*

Hammarström and Henriksson (1994) produced coefficients for calibrating HDM-III to Swedish conditions. The study produced scaling constants (C0SP) and was also able to examine how parts consumption changes with vehicle age (kp). However, it did not provide information on the effect of roughness on parts consumption.

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1 When roughness is expressed as a distance, *eg* in m/km, this is termed the **average rectified distance**. When expressed as a time basis, *eg* m/h, this is termed the **average rectified velocity**.

B6.4.6 RTIM3

The RTIM3 relationships are based on the results of field studies in Kenya and the Caribbean (Cundill, 1993).

SPARE PARTS

There are three models for standardised parts costs, one for cars and light goods, one for medium and heavy goods vehicles, and one for buses. For cars and light goods vehicles, parts consumption is related to roughness and the logarithm of total travel by rather involved sigmoidal functions. For medium and heavy goods vehicles, parts consumption is also related to gradient and curvature. For buses, the relationship is simpler, parts consumption being given by the product of the square root of total travel (as in Brazil) and a sigmoidal function of road roughness.

MAINTENANCE LABOUR

In all cases, the model form is linearly dependent on the standardised parts consumption (as opposed to a square root relationship for India and Brazil) multiplied by a sigmoidal function which decreases with increasing roughness. The function falls only slightly for buses and trucks but it falls by a factor of 1.8 for cars and light goods.

B6.5 Perceived Weaknesses With Existing Models

The users of current road investment models—not just HDM—have a serious problem in deciding how to do their calculations. Conventional wisdom is that maintenance costs are key elements of VOC and yet acceptable alternative approaches can give very different estimates.

The user of HDM-III, for example, could elect to select VOC equations from Brazil, India, Kenya or the Caribbean, and obtain quite different answers depending upon which one was chosen. The user could also elect to suppress the sensitivity of maintenance costs to roughness changes below an IRI of 3. On the other hand, it was not possible to mix VOC elements and combine, say, parts from Brazil with fuel consumption from India. If using the Brazil equations, one was allowed to change the coefficients to permit local calibration, but the India, Kenya and Caribbean equations did not permit local calibration.

The review of experience with HDM-III (Bennett, 1995a) showed clearly the wide variety of approaches that were adopted. In addition to the recommended procedures for changing coefficients to give local calibration, many users have made their own modifications so that model predictions agree with specific information or judgements that they have to hand.

Recurrent observations were:

- There were inconsistencies in the Brazil predictions with passenger cars often having significantly higher parts consumption rates to other vehicles and that this was probably due to the types of vehicles in the study;
- The almost complete insensitivity of buses roughness was hard to understand given that these often have similar chassis to trucks which exhibit a much greater sensitivity, even allowing for different loads;
- Users believed that HDM-Brazil often overestimates parts consumption;
- Users found the model difficult to calibrate; and,
- Because of the non-linearity of the parts consumption relationships, assuming that all vehicles are midway through their life gave a distorted estimate.

Since the field studies in Brazil, India, Kenya and the Caribbean, there have been many smaller studies looking at parts consumption in different situations. Studies have been carried out by TRL on cars, utilities and trucks in Botswana and St Helena and trucks in Pakistan (Hine, 1994); The CSIR have carried out studies in South Africa (du Plessis and Schutte, 1991); VTI in Sweden (Hammarström and Henrikson, 1994); New Zealand (Opus-Beca, 1998; Opus-TRL,1999); and USA (Papagiannakis, 1999).

However, there have been no major studies which would allow for the re-estimation of the parts consumption model parameters for modern vehicles in the same manner as the Brazil study. The improvements in tyre and suspension technology of modern vehicles over the 1970s technology that prevailed at the time of the Brazil study may be one of the reasons behind the view that the HDM relationships are overpredicting roughness effects. For example, Poelman and Weir (1992) found that roughnesses below 3-4 IRI m/km did not result in any strains being transmitted to the chassis. This is in contrast to the predictions of the Brazil parts consumption model, although it supports the convention adopted in HDM-III of eliminating parts benefits below 3 IRI m/km. However, it should be noted that Opus-TRL (1999) and Papagiannakis (1999) suggest that there are roughness effects at low roughnesses.

There may also have been a shift in the allocation of maintenance costs between parts and labour.

Other issues which have been raised are:

- Some analysts prefer the use of a sigmoidal model over a continually increasing roughness model under the thesis that on particularly rough roads there is a change in the type of vehicle used or *some* drivers alter their behaviour to minimise roughness effects; and,
- The use of standardised parts results in a distortion of the costs since vehicles which are particularly robust, and thus cost more, are predicted to have higher parts costs due to their higher capital value, even though their robustness should reduce the parts costs.

The need to address the perceived weaknesses in HDM-III was complicated by the lack of data upon which to base any modifications. The most major studies available are those from South Africa (du Plessis and Schutte, 1991), but these were oriented mainly at trucks and buses, not passenger cars.

On the basis of the available data, a revised approach to modelling parts consumption for HDM-4 was developed. This approach is outlined in the following sections.

B6.6 HDM-4 Parts Consumption Model

B6.6.1 Introduction

Owing to the absence of significant new data, the HDM-4 parts consumption model by necessity is largely based on the HDM-III model. It has been developed to address the following perceived shortcomings in the HDM-III model:

- The need to limit the model predictions at low roughness levels;
- The desire for a simpler and more easily followed model structure;
- The refinement of the model parameters with regard to the effects of roughness on parts consumption;
- The ability to model different trade-offs between parts and labour.

Cundill (1995a and 1995b) proposed a parts consumption model for HDM-4 which addressed these issues and this model is given in NDLI (1995). However, subsequent work resulted in changes to the model.

This section commences with a summary of the NDLI (1995) model and the changes that were made to it. This is followed by the parts consumption model adopted for HDM-4 and its basis.

B6.6.2 Preliminary HDM-4 Parts Consumption Model

MODEL FORM

The HDM-III parts model is linear for trucks, exponential for passenger cars and buses, although the exponential effects for buses are such that the function is essentially linear over the full range of roughnesses[1]. From a user's perspective, a linear function is much easier to interpret and calibrate so the following model was proposed for the HDM-4 parts model by NDLI (1995):

$$PARTS = C0 \left(\frac{CKM}{100000}\right)^{kp} (1 + CIRI\,(RI - 3)) \qquad \ldots(B6.13)$$

where C0 is a calibration parameter for the absolute magnitude of the predictions
CIRI is a calibration parameters for the effect of roughness on the predictions
RI is the adjusted roughness in IRI m/km (see below)

The above relationship was standardised to a distance of 100,000 km on a smooth pavement (IRI < 3). At this roughness, the calibration parameter C0 represents the parts consumption. It was considered that this parameter could be readily estimated by analysts when applying the model in the field.

The calibration parameter CIRI represents the increase in parts consumption as a function of roughness. There is a 100 CIRI per cent increase in the parts consumption for each increase in IRI. For local calibration, it was proposed that a vehicle operator make an estimate of the ratio of parts costs for different surfaces and so be able to estimate CIRI directly.

The constant kp can be estimated by examining the rate at which parts consumption increases with vehicle age. The term 2^{kp} gives the increase in parts consumption as cumulative travel is doubled.

In an economic appraisal the key factor will usually be the product C0 CIRI. This is indicative of the rate of increase in parts consumption as a function of roughness. The higher this product, the greater the effects of roughness.

ADJUSTED ROUGHNESS

The adjusted roughness, RI, has been adopted in place of the IRI for modelling of pavements with low roughnesses. In HDM-III the user could elect to ignore benefits from parts consumption for roughnesses below 3 IRI. This was based on the common sense assumption that roughness changes in this region would have little, if any effect. Subsequent research

[1] Replacing the exponential model used with passenger cars and heavy buses results in a maximum error of less than seven per cent for passenger cars and two per cent for heavy buses. Given the other uncertainties in the parts modelling this was considered to be acceptable.

(Poelman and Weir, 1992) showed that, at least for passenger cars, this was a valid assumption and that below 3-4 IRI most roughness effects are absorbed by the tyres and suspension. However, Opus-TRL (1999) and Papagiannakis (1999) suggest that that there are in fact roughness effects at these levels.

It is an issue requiring further research.

A disadvantage to the HDM-III approach was that it resulted in a sudden discontinuity at 3 IRI where the slope of the relationship would suddenly change. It was therefore decided that it was better to have a smoother transition using the following equations:

$$RI = IRI \qquad IRI \geq IRI0 \qquad \ldots(B6.14)$$

$$RI = a0 + a1\ IRI^{a2} \qquad IRI < IRI0 \qquad \ldots(B6.15)$$

where IRI0 is the limiting roughness for parts consumption in IRI m/km

This can also be expressed as:

$$RI = \max(IRI, \min(IRI0, a0 + a1\ IRI^{a2})) \qquad \ldots(B6.16)$$

Although Equation B6.16 has three independent variables, Cundill (1995b) shows that they are related to the limiting roughness and a single model parameter a3:

$$a0 = IRI0 - a3 \qquad \ldots(B6.17)$$

$$a1 = \frac{a3}{IRI0^{\frac{IRI0}{a3}}} \qquad \ldots(B6.18)$$

$$a2 = \frac{IRI0}{a3} \qquad \ldots(B6.19)$$

The parameter a0 represents the minimum roughness to use in the calculations. Based on HDM-III this should have a value of 3.0. Thus, it is possible to rewrite Equation B6.17 as:

$$a3 = IRI0 - 3 \qquad \ldots(B6.20)$$

Figure B6.6 shows the predictions of the smoothing function using values of 4, 3.5, 3.25 and 3.10 for IRI0. It can be seen that depending upon the values selected, there are significant differences in the slope as one approaches the limiting value of 3. For HDM-4 it was recommended that a default value of 3.25 be adopted for IRI0 since this limits the overestimation near the transition point. However, users are able to define alternative values for both IRI0 and a3 in HDM-4 to reflect local vehicles and conditions.

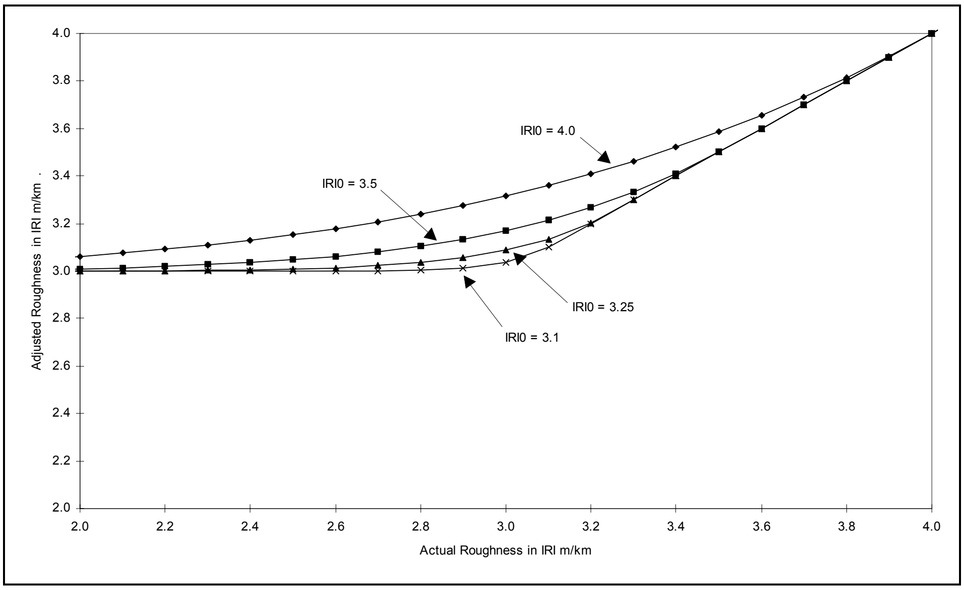

Figure B6.6: Low Roughness Smoothing Curve

TRAFFIC INTERACTION EFFECTS

It was noted by NDLI (1995) that traffic interactions (*ie* congestion) would result in additional parts consumption. It was proposed that this would be considered by multiplying the parts consumption from Equation B6.13 by the factor (1 + CPCON dFUEL). CPCON was an elasticity which reflected the relationship between the additional fuel due to interactions (dFUEL) and the additional parts consumption. It was recommended that CPCON have a default value of 0.10.

CALIBRATION OF PROPOSED HDM-4 PARTS CONSUMPTION MODEL TO AVAILABLE DATA

The parts consumption model presented above was calibrated from the various studies. The results are shown in Table B6.3. Cundill (1995b) and NDLI (1995) describe the basis for these values and the implications of their predictions.

The values in Table B6.3 were based on an analysis of the results from the various road user cost studies and other available data. However, Cox (1996a) shows that there have been significant reductions in the maintenance costs over time in Australia. For passenger cars the costs have decreased from 15.2 c/km in 1970 to 6.0 c/km in 1992 (1992 prices) while for articulated trucks the costs are reported to have decreased even more significantly. Cox (1996a) also shows evidence of a similar trend from Sweden where costs decreased by up to 50 per cent between 1976 and 1987.

The same was found in N.Z. Bennett (1985) reported that the 1983 passenger car costs were approximately 8 - 9 c/km whereas Opus-Beca (1998) suggest that the 1995 values were less than half this level. When the decrease in considered in terms of 1983 dollars there has been an even more significant decrease in maintenance costs. Truck costs also showed a marked decline, although not of the same magnitude as passenger cars.

Hammerström and Henriksson (1994) in calibrating the HDM-III model to Sweden found that the calibration coefficients for passenger cars were significantly lower than the default Brazil value, with a kp = 0.0741 as opposed to the default value of 0.308. However, the results for trucks indicated a higher value for kp than the default HDM-III value.

Table B6.3: Results of Parts Model Calibration

Vehicle Class	Study	Parts Model Parameters			C0 CIRI	2^{kp}
		C0 ($x 10^{-3}$)	CIRI	kp	($x 10^{-3}$)	
Passenger Cars and Utilities	Brazil	1.818	0.306	0.308	0.556	1.238
Light and Medium Trucks	Brazil	1.154	0.303	0.371	0.350	1.293
Heavy Trucks	Brazil	1.466	0.193	0.371	0.283	1.293
Articulated Trucks	Brazil	1.608	0.126	0.371	0.203	1.293
Heavy Buses	Brazil	0.519	0.061	0.483	0.032	1.398
Passenger Cars	India	0.710	0.357	0.000	0.253	1.000
Truck (7t)	India[1]	0.154	0.600	0.340	0.092	1.266
Truck (14t)	India[1]	0.223	0.600	0.340	0.134	1.266
Truck (24t)	India[1]	0.470	0.600	0.340	0.282	1.266
Heavy Buses	India	0.327	0.071	0.358	0.023	1.282
Passenger Cars and Utilities	Kenya	1.939	1.225	1.000	2.375	2.000
Trucks	Kenya	1.463	0.297	1.000	0.435	2.000
Heavy Buses	Kenya	0.188	1.211	0.500	0.228	1.414
Passenger Cars and Utilities[2]	Caribbean	-	-	1.000	2.696	2.000
Trucks	Caribbean	0.628	1.604	1.000	1.007	2.000
Trucks	South Africa	0.369	0.603	0.451	0.223	1.367
Heavy Bus	South Africa	0.473	0.067	0.678	0.032	1.600
Utility	Botswana	-	-	0.499	-	1.413
Passenger Car	St. Helena[3]	1.608	0.306	-	0.492	-
Utility	St. Helena[3]	0.658	0.306	0.492	0.201	1.406
Utility	St. Helena[3]	0.676	0.306	0.509	0.207	1.423
Medium Truck	St. Helena[3]	0.870	0.303	0.313	0.264	1.242

Source: Cundill (1995b)

Notes: 1/ Using truck relationships from Chesher and Harrison (1987).
2/ Relationship gives unrealistic negative intercept for C0 and this affects CIRI. However, the term C0 IRI is still valid.
3/ Inferred using Brazil values for CIRI.

The available evidence therefore suggests that technological improvements have resulted in a decrease in the magnitude of maintenance costs over time. This made it questionable to continue with the application of the Brazil model values in HDM-4.

After considering the above issues, NDLI (1995) proposed the parts consumption model parameters for each representative vehicle class in Table B4.4. The “new technology” values are subjective estimates considered to apply to modern vehicle technology. The “old technology” values are largely those from the Brazil study presented earlier in Table B6.3, although the parameters for the articulated truck have been altered to make the roughness effects similar to other trucks.

For all vehicles except passenger cars and buses the “new technology” values for C0 and CIRI are approximately 75 per cent of the “old technology” values. This indicated a 25 per cent improvement in technology since the 1970s when the Brazil study was conducted. Passenger cars were considered to have had a more significant improvement whereas buses, which already had relatively low costs compared to trucks, are considered to not have had any improvement.

The net impact of these changes is shown in the last two columns of Table B4.4. For light vehicles there is a 59 per cent reduction in the roughness effects; for trucks 42 to 43 percent. The changes to the age term have a limited impact on the age effects at less than seven per cent for all vehicle classes.

B6.6.3 Refinements to Preliminary Model

The 1995 TRL Workshop on Road User Effects (Bennett, 1996a) discussed the HDM-4 parts consumption model proposed by NDLI (1995).

The linear parts consumption model (Equation B6.13) was considered to be an improvement over the HDM-III formulation.

The discussion centred around the actual parameter values. It was agreed that the passenger car and utility parameters in HDM-III led to a significant overprediction of roughness effects whereas the bus parameters underpredicted roughness. Trucks were noted to have had the most reliable data and thus the most reliable relationships.

On the basis of the discussion it was proposed for HDM-4:

- The default roughness effects for all vehicles be the same;
- These will be based on the Brazil truck equations;
- A 25 per cent decrease in consumption due to technological improvements; and,
- A 25 per cent increase in the age term to reflect the bias introduced by sampling only survivors in the original survey.

Bennett (1996a) re-quantified the model parameters with these assumptions. The differentiation between 'Old' and 'New' technology from NDLI (1995) was not maintained as it was considered more appropriate to have a single set of parameters for all vehicles.

When the Optimal Life depreciation method was in the process of being implemented for HDM-4 it was found that Equation B6.13 had a major deficiency through the standardisation to 100,000 km (*ie*, due to the term CKM/100000). As described in Bennett (1996b), this meant that irrespective of what value was adopted for kp the parts consumption was always the same at 100000 km (C0 when RI < 3 IRI). Thus, any changes to kp served to rotate the predictions around 100000 km. This served to fundamentally alter the meaning of kp as well as impacting on the roughness effects. Consequently, the model reverted back to using the coefficient CKM alone. The ISOHDM Steering Committee also recommended that the model be continuous over all roughnesses. This led to the following parts consumption model (Bennett, 1996d):

$$PARTS = \{K0pc\ [CKM^{kp}\ (a0 + a1\ RI)] + K1pc\}\ (1 + CPCON\ dFUEL) \qquad \ldots(B6.21)$$

where

CPCON	is the congestion elasticity factor (default = 0.1)
dFUEL	is the additional fuel consumption due to congestion as a decimal
K0pc	is a rotational calibration factor (default = 1.0)
K1pc	is a translational calibration factor (default = 0.0)
a0 to a3	are model parameters

Bennett (1996d) estimated parameter values for this model from the HDM-III model predictions. The exponential models were replaced by linear models which gave similar predictions in the range of 3-10 IRI m/km. Using a cumulative kilometreage of 100,000 km for the distance travelled, the parameters a0 and a1 were established

Table B6.4: Preliminary HDM-4 Parts Consumption Model Parameters

Vehicle Class	Description	Parts Consumption Model Parameters										Difference Between Old and New Tech. in per cent	
		Old Technology					New Technology						
		C0	CIRI	kp	CO CIRI	2^kp	C0	CIRI	kp	CO CIRI	2^kp	CO CIRI	2^kp
1	Motorcycle	0.364	0.306	0.308	0.111	1.238	0.200	0.230	0.230	0.046	1.173	59	5
2	Small Car	1.818	0.306	0.308	0.556	1.238	1.000	0.230	0.230	0.230	1.173	59	5
3	Medium Car	1.818	0.306	0.308	0.556	1.238	1.000	0.230	0.230	0.230	1.173	59	5
4	Large Car	1.818	0.306	0.308	0.556	1.238	1.000	0.230	0.230	0.230	1.173	59	5
5	Light Delivery Vehicle	1.818	0.306	0.308	0.556	1.238	1.000	0.230	0.230	0.230	1.173	59	5
6	Light Goods Vehicle	1.818	0.306	0.308	0.556	1.238	1.000	0.230	0.230	0.230	1.173	59	5
7	Four Wheel Drive	0.667	0.250	0.250	0.167	1.189	0.650	0.200	0.200	0.130	1.149	22	3
8	Light Truck	1.154	0.303	0.371	0.350	1.293	0.870	0.230	0.280	0.200	1.214	43	6
9	Medium Truck	1.466	0.193	0.371	0.283	1.293	1.100	0.150	0.280	0.165	1.214	42	6
10	Heavy Truck	1.466	0.193	0.371	0.283	1.293	1.100	0.150	0.280	0.165	1.214	42	6
11	Articulated Truck	1.466	0.193	0.371	0.283	1.293	1.100	0.150	0.280	0.165	1.214	42	6
12	Mini-bus	1.818	0.306	0.308	0.556	1.238	1.000	0.230	0.230	0.230	1.173	59	5
13	Light Bus	1.154	0.303	0.371	0.350	1.293	0.870	0.230	0.280	0.200	1.214	43	6
14	Medium Bus	0.519	0.061	0.483	0.032	1.398	0.519	0.061	0.483	0.032	1.398	0	0
15	Heavy Bus	0.519	0.061	0.483	0.032	1.398	0.519	0.061	0.483	0.032	1.398	0	0
16	Coach	0.519	0.061	0.483	0.032	1.398	0.519	0.061	0.483	0.032	1.398	0	0

Source: *NDLI (1995). Based on work in Cundill (1995b) and judgement.*

B6.6.4 Final HDM-4 Parts Consumption Model

Equation B6.21 was adopted for the final HDM-4 parts consumption model. As noted above, the parameter values calculated by Bennett (1996d) were based on a cumulative kilometreage of 100,000 km. Since the default HDM-4 vehicle attributes resulted in values greatly in excess of this figure for many vehicle classes, Bennett (1998c) recalculated the values for a0 and a1 based on the CKM that will arise with the default HDM-4 values.

The 1995 HDM Workshop on Road User effects recommended that the roughness effects be based on those for heavy trucks since they had the most reliable data in the original Brazil study. To reflect improvements in technology these roughness effects were reduced by 25 per cent. The coefficients a0 and a1 in the parts model were estimated based on the following constraints:

- The predicted parts consumption should be the same as the original parts consumption at a roughness of 3 IRI m/km; and,
- The change in roughness in the range 3-10 IRI m/km corresponds to 75 per cent of the roughness effect for heavy trucks at that cumulative kilometreage. For buses, a value of 50 per cent of the roughness effect was used.

The values of a0 and a1 were established for each of the representative vehicle classes subject to these constraints. Table B6.5 shows the final parameters. These values are plotted in Figure B6.7 and Figure B6.8 along with the original HDM-III predictions. It will be noted that the greatest changes are with passenger cars where the roughness effects are about 1/3 what they were in HDM-III.

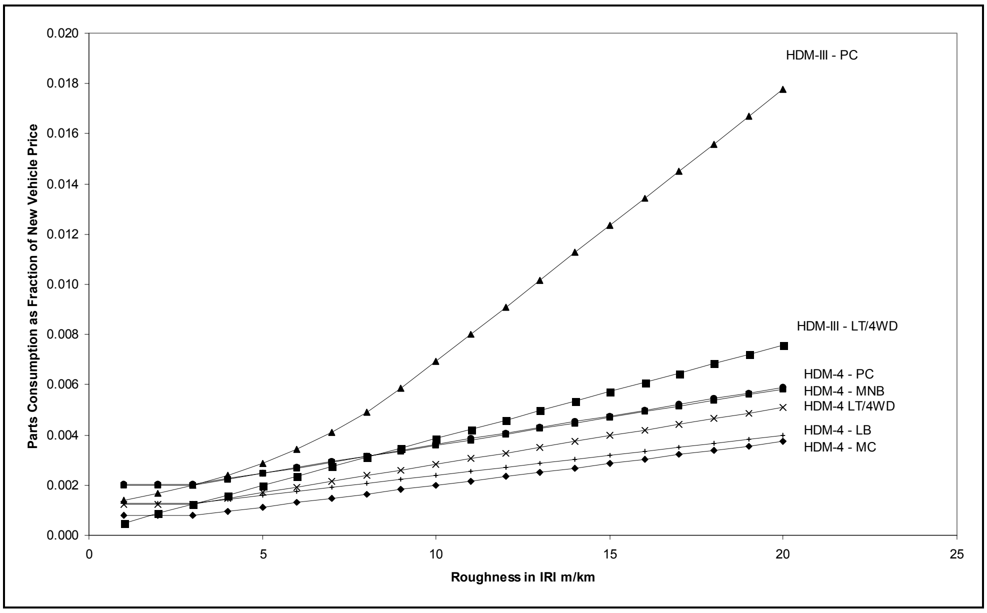

Figure B6.7: Comparison of HDM-4 and HDM-III Parts Consumption: Light Vehicles

Table B6.6 compares the roughness effects from HDM-4 and HDM-III for roughness changes between three and 10 IRI m/km. Values are given for the default CKM as well as 100,000, 200,000 and 300,000 km. It will be noted that for most vehicles there has been a decrease in the effects of roughness on parts consumption, particularly for passenger cars.

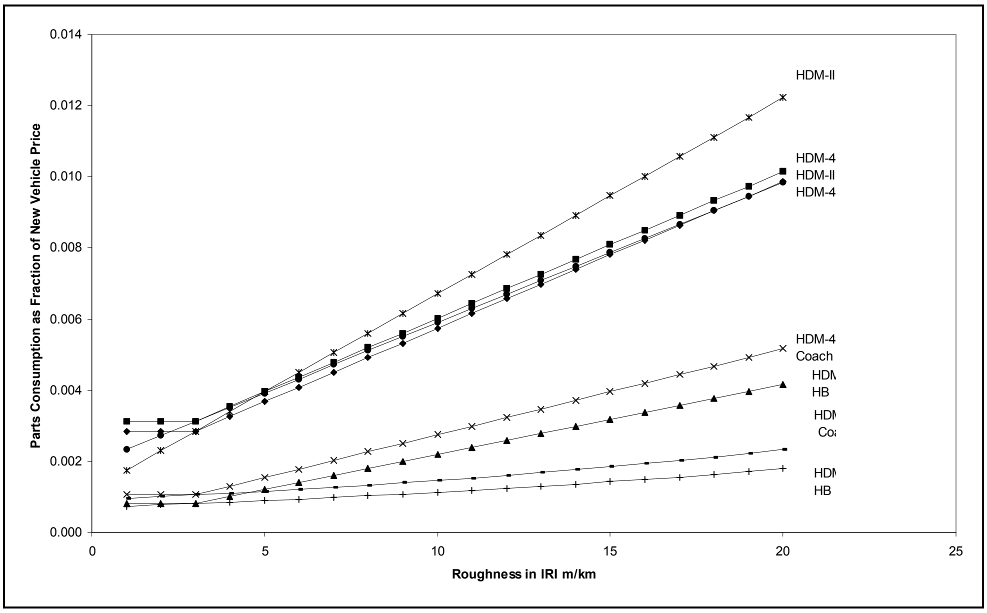

Figure B6.8: Comparison of HDM-4 and HDM-III Parts Consumption: Heavy Vehicles

Table B6.5: Recommended Parts and Labour Model Parameters

Vehicle	Description	Parts Consumption Model				Labour Model	
		CKM	kp	a0 x 10^{-6}	a1 x 10^{-6}	a0	a1
1	MC	50,000	0.308	9.23	6.20	77.14	0.547
2	PC	115,000	0.308	36.94	6.20	77.14	0.547
3	PC	115,000	0.308	36.94	6.20	77.14	0.547
4	PC	115,000	0.308	36.94	6.20	77.14	0.547
5	LDV	120,000	0.308	36.94	6.20	77.14	0.547
6	LGV	120,000	0.308	36.94	6.20	77.14	0.547
7	4WD	120,000	0.371	7.29	2.96	77.14	0.547
8	LT	120,000	0.371	7.29	2.96	242.03	0.519
9	MT	240,000	0.371	11.58	2.96	242.03	0.519
10	HT	602,000	0.371	11.58	2.96	301.46	0.519
11	AT	602,000	0.371	13.58	2.96	301.46	0.519
12	MNB	120,000	0.308	36.76	6.20	77.14	0.547
13	LB	136,000	0.371	10.14	1.97	242.03	0.519
14	MB	245,000	0.483	0.57	0.49	293.44	0.517
15	HB	420,000	0.483	0.65	0.46	293.44	0.517
16	COACH	420,000	0.483	0.64	0.46	293.44	0.517

Source: Bennett (1998c)

Notes: 1/ The AT labour model coefficient a0 from Brazil was 652.51. As described in the text, it has been replaced with the heavy truck value of 301.46 to obtain more reasonable predictions.

Table B6.6: Comparison of Roughness Effects – HDM-4 and HDM-III

Vehicle	Increase in Parts Consumption as Fraction of New Vehicle Price (x 10^{-3}) From 3-10 IRI m/km by CKM							
	Default CKM		100,000		200,000		300,000	
	HDM-III	HDM-4	HDM-III	HDM-4	HDM-III	HDM-4	HDM-III	HDM-4
MC	-	1.21	-	1.50	-	1.86	-	2.11
PC	4.91	1.57	4.71	1.50	5.83	1.86	6.60	2.11
LDV/LGV	5.04	1.59	4.76	1.50	5.90	1.86	6.68	2.11
4WD/LT	2.62	1.59	2.44	1.49	3.16	1.92	3.68	2.23
MT	2.74	2.06	1.98	1.49	2.56	1.93	2.98	2.24
HT	3.86	2.89	1.98	1.48	2.56	1.92	2.98	2.23
AT	2.77	2.89	1.42	1.48	1.84	1.92	2.14	2.23
MNB	5.04	1.59	4.76	1.50	5.90	1.86	6.68	2.11
LB	2.74	1.11	2.44	0.99	3.16	1.28	3.68	1.49
MB	0.31	1.38	0.20	0.90	0.28	1.25	0.34	1.52
HB	0.40	1.69	0.20	0.84	0.28	1.18	0.34	1.44
COACH	0.40	1.69	0.20	0.84	0.28	1.18	0.34	1.44

B6.7 HDM-4 Labour Hours Model

The HDM-III labour model was presented earlier as Equation B6.5. This related the number of labour hours to the parts consumption and, for buses, road roughness.

As would be expected, there were large regional variations in the labour hours predictions which reflected the different maintenance practices and other factors such as the relative costs of parts versus labour. Table B6.7 shows labour hours according to the Brazil, India and Kenya relationships from HDM-III. The table also shows standardised parts cost and the ratio of labour hours to parts, the last being important if you are thinking of labour costs as a “mark-up” on parts costs.

Table B6.7: Maintenance Labour from the Different Models

Study	Vehicle Class	PARTS (x 10^{-3})	LH	Ratio
	Car and Utility	1.82	2.45	1.4
Brazil	Light and Medium Truck	1.15	7.23	6.3
	Heavy Truck	1.47	10.20	7.0
	Bus	0.52	7.29	14.1
	Car	0.71	16.83	23.7
	Utility	0.71	24.29	34.2
India	Truck (7t)	0.15	8.34	54.2
	Truck (14 t)	0.22	10.63	47.7
	Truck (24t)	0.22	20.31	33.8
	Bus	0.33	15.68	48.0
	Car and Utility	1.94	1.36	0.7
Kenya	Truck	1.46	4.14	2.8
	Bus	0.19	0.47	2.5

Notes: 1/ LH = labour hours per 1,000 km
2/ PARTS = standardised parts cost per 1,000 km
3/ Ratio = LH/(PARTS x 10^{-3})
4/ Results are for cumulative kilometreage = 100,000 km and IRI = 3

The fact that LH/PARTS increases with vehicle size reflects that standardised parts cost is not a good explanatory variable. A better choice would probably be something between parts cost and standardised parts costs.

The India values for LH are higher than Brazil and, since the standardised parts costs are lower, the values of LH/PARTS are higher still. The numbers vary with vehicle age and road roughness but the difference is usually large—LH/PARTS for India can be as much as 20 times higher than the equivalent Brazil value.

The high values of LH in India must reflect the much lower wage rates and hence the pressure to substitute labour for parts where possible, so parts will be lower, labour hours higher and the ratio of labour to parts, higher still. Another factor at play is the use of multiple labourers for a single task.[1]

By contrast, the Kenya values for LH and LH/PARTS are lower than Brazil. At higher values of roughness and vehicle age, the Kenya equations can give values of LH which are higher than Brazil but the ratio of LH/PARTS remains typically between 20 and 85 percent of the Brazil value.

On the basis of the above, NDLI (1995) considered that it was most appropriate to continue with the use of either the Brazil or the India labour hour equations. The India equations should be used only in countries with very low unit labour costs. As a useful guide, it would be expected that if using the India equations, the total repair costs (parts plus labour) should be lower than would have been obtained with the Brazil equations.

It should be noted that when the Indian labour model is used it is advisable to also use coefficients for the parts model from India. This is because of the close relationship between parts replacement policy and labour costs.

In light of the refinements made to the parts consumption model, and the desire for simplification, Bennett (1996d) proposed the following simple linear model for labour hours:

$$LH = K0lh\ [a0\ PC^{a1}] + K1lh \qquad \text{...(B6.22)}$$

where

LH	is the number of labour hours per 1000 km
K0lh	is the rotation calibration factor (default = 1)
K1lh	is the translation calibration factor (default = 0)
a0 and a1	are constants

Since the roughness effects only affected buses, and they were minor, it was considered that their exclusion would not materially reduce the validity of the model's predictions.

Bennett (1996d) proposed the labour model parameters based on HDM-III Brazil which are given in Table B6.8.

Figure B6.9 shows, except for articulated trucks, the predicted labour hours using the new HDM-4 parts consumption model parameters with the default HDM-4 values of CKM. It will be noted in Figure B6.9 that the articulated vehicle predictions are markedly higher than the other vehicle classes—about three times the magnitude of those for heavy trucks. This was also in HDM-III (see Figure 5.10(c) on page 184 of Watanatada, *et al.*, 1987). This is the likely cause behind the observation that for some vehicle classes the BETA HDM-4 software was significantly over-predicting the maintenance and repair costs (Thoresen and Roper, 1998).

For this reason, the default HDM-4 articulated truck labour hours parameters were based on heavy trucks instead of the original HDM-III articulated truck. This will serve to make the predictions of a similar magnitude to heavy trucks but still higher to reflect trailer costs.

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[1] One had to be careful in applying the HDM-III India model since at an average speed of 60 km/h it would take 15 h to travel 1000 km. Thus, for buses the model predicts 1 hour of labour for every hour of travel, and even more for heavy trucks..

These labour hours are probably still too high, at least for modern vehicles, so local calibration is essential.

Table B6.8: Parameter Values for HDM-4 Labour Hours Model

Vehicle	Description	HDM-4 Labour Model Parameters			
		Brazil		India	
		a0	a1	a0	a1
1	Motorcycle	77.14	0.547	1161.42	0.584
2	Small Car	77.14	0.547	1161.42	0.584
3	Medium Car	77.14	0.547	1161.42	0.584
4	Large Car	77.14	0.547	1161.42	0.584
5	Light Delivery Vehicle	77.14	0.547	611.75	0.445
6	Light Goods Vehicle	77.14	0.547	611.75	0.445
7	Four Wheel Drive	77.14	0.547	611.75	0.445
8	Light Truck	242.03	0.519	2462.22	0.654
9	Medium Truck	242.03	0.519	2462.22	0.654
10	Heavy Truck	301.46	0.519	2462.22	0.654
11	Articulated Truck	652.51	0.519	-	-
12	Mini-bus	77.14	0.547	611.75	0.445
13	Light Bus	242.03	0.519	637.12	0.473
14	Medium Bus	293.44	0.517	637.12	0.473
15	Heavy Bus	293.44	0.517	637.12	0.473
16	Coach	293.44	0.517	637.12	0.473

Source: ***Bennett (1998c) from an a****nalysis of data in Watanatada, et al. (1987a)*

Notes: 1/ The articulated truck Brazil parameter will lead to a significant over-estimation of the labour hours (see Figure B6.9). It is therefore recommended that the heavy truck value of 301.46 be used instead of the articulated truck 652.51 (see Table B6.5).

2/ One would only apply the Indian labour hours model with lower coefficients for parts consumption from the default HDM-4 values.

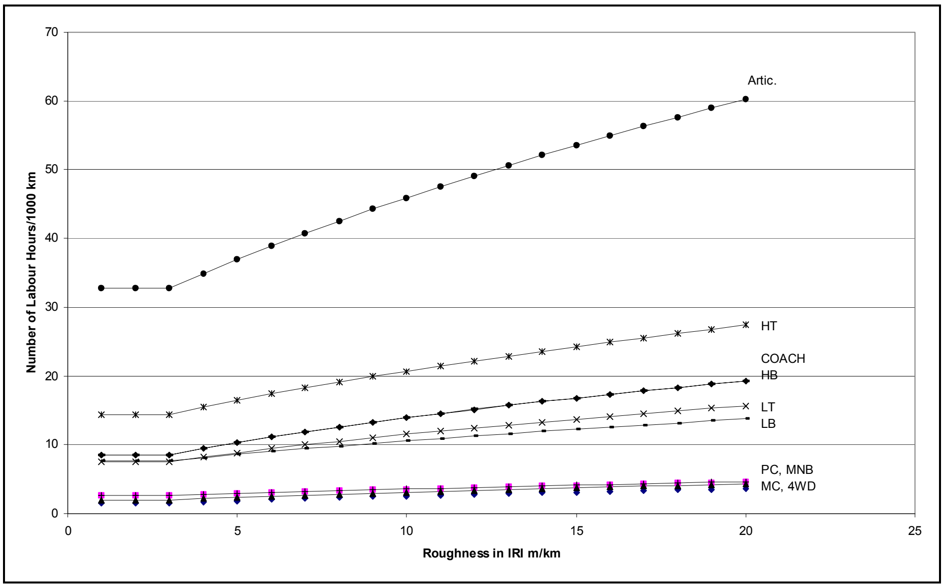

Figure B6.9: HDM-4 Labour Hours Predictions

B6.8 Modelling Age Effects

The parts model is non-linear with respect to age (CKM). In HDM-III the value for CKM was taken at the mid-point in the vehicle's life (Watanatada, *et al.*, 1987a):

$$CKM = 0.5\ LIFE\ AKM \quad \ldots(B6.23)$$

As described by Cox (1994), the use of the mid-point kilometreage results in a systematic bias in the results. This can lead to a mis-estimation of the parts costs, although given all the uncertainties it is likely not be that significant.

However, the HDM-4 software has been designed so that the user be allowed to enter both the utilisation over the life of the vehicle and an age spectrum and that these data be used to calculate the parts consumption at different vehicle kilometreages.

B6.9 Modelling Traffic Interaction Effects

The additional vehicle accelerations and decelerations associated with congestion can be expected to increase parts consumption. Although the South African work related maintenance costs to speed (see Equation B6.8), this was considered to be an unreliable way of modelling congestion effects since it implies that higher speeds in the absence of congestion would result in increased maintenance costs.

The congestion model introduced in Chapter B3 and applied in Chapter B4 predicts that the acceleration noise is a function of the volume-to-capacity ratio. At higher flows there are more accelerations and decelerations so the acceleration noise is higher. The fuel consumption was shown to be a function of the acceleration noise and expressed using the factor dFUEL (see Section B4.7). If it is assumed that the fractional change in parts consumption on a given road section due to traffic congestion is related to the fractional change in fuel consumption due to congestion, the parts consumption can be expressed as:

$$\text{PARTS} = \text{PARTS}\ (1 + \text{CPCON dFUEL}) \qquad \text{...(B6.24)}$$

where	CPCON	is the incremental change of parts consumption related to congestion fuel (default = 0.10)
	dFUEL	is the increase in fuel consumption due to congestion as a fraction

The above suggests that for every 100 per cent increase in the fuel consumption there will be a 10 per cent increase in the parts consumption.

B6.10 Summary

For HDM-4 it is proposed that the parts model be simplified over that used in HDM-III.

$$\text{PARTS} = \{\text{K0pc}\ [\text{CKM}^{kp}\ (a0 + a1\ \text{RI})] + \text{K1pc}\}\ (1 + \text{CPCON dFUEL}) \qquad \text{...(B6.25)}$$

where	CPCON	is the congestion elasticity factor (default = 0.1)
	dFUEL	is the additional fuel consumption due to congestion as a decimal
	K0pc	is a rotational calibration factor (default = 1.0)
	K1pc	is a translational calibration factor (default = 0.0)
	a0 to a3	are model parameters

The labour hours are predicted as:

$$\text{LH} = \text{K0lh}\ [a0\ \text{PC}^{a1}] + \text{K1lh} \qquad \text{...(B6.26)}$$

where	LH	is the number of labour hours per 1000 km
	K0lh	is the rotation calibration factor (default = 1)
	K1lh	is the translation calibration factor (default = 0)
	a0 and a1	are constants

It is possible to eliminate the effects of roughness on parts consumption at low roughnesses. This is done using the following smoothing relationship has been adopted:

$$RI = \max(IRI, \min(IRI0, a0 + a1\ IRI^{a2}) \qquad \ldots(B6.27)$$

where IRI0 is the limiting roughness for parts consumption in IRI m/km
a0 to a3 are model parameters

The quantification of parameters for the parts model was particularly problematic. There is a consensus of opinion that improvements in vehicle technology and other factors give modern technology vehicles lower maintenance costs, and that they are also less sensitive to roughness. However, there have been relatively few studies done to verify this opinion, particularly with passenger cars. Values were estimated based on the Brazil study results.

A generalised labour model was presented. It was recommended that for predicting labour hours the best approach was to adopt the Brazil labour model for countries with a low labour to parts ratio and the India relationships for countries with a high labour to parts ratio. If the India labour relationships are being used it would be judicious to also adopt parts consumption parameter values based on the India study.

Congestion will increase parts consumption due to the additional forces acting on the vehicle. Instead of using mechanistic theory the marginal maintenance costs are assumed to be a function of the marginal fuel consumption due to congestion. An increase of 10 per cent in the maintenance costs for each 100 per cent increase in fuel consumption is proposed as the initial estimate.

B7 Utilisation and Service Life

B7.1 Introduction

A vehicle, or any physical property, has three measures of its life, namely the:

- **Service life:** the period over which the vehicle is operated;
- **Physical life:** the period which the vehicle exists (even if it is not being used); and,
- **Economic life:** the period which the vehicle is economically profitable to operate

The life can be expressed in either of two ways: **time** (in years) or **distance** (in km). Both are interchangeable and the distance measure is referred to as the **lifetime kilometreage**. It is expressed either directly or as the product of the service life and the average annual kilometreage.

There are three definitions for the hourly utilisation:

- **HAV** - the number of hours the vehicle is **available** per year

  This is the number of hours per year (8760), less the time allowed for crew rest, time lost loading, unloading, refuelling, finding cargo, repairs, *etc.*

- **HRD** - the numbers of hours **driven**

  This is the hours that the vehicle is operated. It can be calculated from the average annual kilometreage divided by the average annual speed.

- **HWK** - the number of hours **worked**

  This is similar to the hours driven (HRD), except it includes the time spent loading, unloading and refuelling.

Using a standard working week, a vehicle is typically available for approximately 1800 hours per year. However, since there are substantial periods of time when the vehicle is not in use, for example due to loading/unloading, the driving time would often be less than 50 per cent of this value. In the Brazil study, for example, the vehicles were available 839 - 2414 h, but only driven 652 - 1863 h (Watanatada, *et al*., 1987b). Trucks and buses had the highest utilisations; utilities the lowest.

Vehicle utilisation and service life are important characteristics in modelling VOC. They are used to predict the parts consumption, through the use of kilometreage as an independent variable, and also influence depreciation, crew, overhead and vehicle time costs.

B7.2 Utilisation and Service Life Modelling in HDM-III

B7.2.1 Utilisation Modelling

HDM-III contained three methods for predicting utilisation (Watanatada, *et al*., 1987a):

- Constant Kilometreage;
- Constant Hours; and,
- Adjusted Utilisation.

The constant kilometreage method assumed that the vehicle travelled a constant distance each year, irrespective of operating conditions. The constant hours method assumed that the number of hours driven each year was constant. The adjusted utilisation method was predicated on there being a relationship between speed and utilisation. As will be shown, the adjusted utilisation method was a generic approach which, with selection of the appropriate model parameters, reduces to the constant kilometreage and constant hour methods.

As described by Watanatada, *et al*. (1987b), the adjusted utilisation method is based on total time spent making a trip on a fixed route throughout the year. This is expressed as:

$$TT = TN + TD \qquad \ldots(B7.1)$$

where TT is the total travel time for the trip in h/trip
TN is the amount of time spent on non-driving activities as part of the round trip tour, such as loading, unloading, refueling, layovers, *etc*., in h/trip
TD is the amount of time spent driving on the trip in h/trip

The total driving time is a function of the route length:

$$TD = \frac{RL}{S} \qquad \ldots(B7.2)$$

where RL is the route length in km
S is the speed in km/h

The number of hours available is assumed to be constant and independent of vehicle speed. Vehicle operators are assumed to maximise their productivity by making as many trips as possible within the constraint of vehicle availability. This is predicted as:

$$NTRIPS = \frac{HAV}{TT} \qquad \ldots(B7.3)$$

where NTRIPS is the number of trips per year
HAV is the number of hours vehicle available for use in h/yr

The annual utilisation is then:

$$AKM = NTRIPS\ RL \qquad \ldots(B7.4)$$

Substituting in Equations B7.1 and B7.2 into Equations B7.3 and B7.4 gives:

$$AKM = \frac{HAV}{\frac{TN}{RL} + \frac{1}{S}} \qquad \ldots(B7.5)$$

The above is based on vehicles always being used and no excess capacity or periods of underutilisation in the fleet. In reality this does not happen, which gives rise to the following:

$$NTRIPS \leq \frac{HAV}{TT} \qquad \ldots(B7.6)$$

As suggested by Equation B7.6, there is an inelasticity associated with vehicle use. We define the term “Elasticity of Utilisation” which is effect of speed on utilisation. As described by Watanatada, *et al*. (1987b) this is calculated as:

$$EVU = \frac{TD}{TT} \qquad \text{…(B7.7)}$$

$$EVU = \frac{HRD}{HAV} \qquad \text{…(B7.8)}$$

$$EVU = \frac{1}{\frac{S\ TN}{RL} + 1} \qquad \text{…(B7.9)}$$

where EVU is the elasticity of utilisation

Combining the EVU with the previous equations leads to the following function for predicting the annual utilisation (Watanatada, *et al*., 1987a):

$$AKM = \frac{AKM0\ HRD0}{HRD0\ (1 - EVU) + \frac{AKM0\ EVU}{S}} \qquad \text{…(B7.10)}$$

where AKM0 is the baseline annual utilisation in km/yr
HRD0 is the baseline number of hours driven in h/yr

Equation B7.10 is the HDM-III adjusted utilisation equation. It predicts that the annual kilometreage will be proportional to the base annual kilometreage, the base number of hours, the elasticity of utilisation and the speed.

When the elasticity is 1, Equation B7.10 predicts that the annual kilometreage is:

$$AKM = HRD0\ S \qquad \text{…(B7.11)}$$

When the elasticity is 0, Equation B7.10 predicts that the annual kilometreage is:

$$AKM = AKM0 \qquad \text{…(B7.12)}$$

These are the respectively "Constant Hourly" and "Constant Kilometreage" methods from HDM-III.

Watanatada, *et al*. (1987b) analysed utilisation data from Brazil and established the elasticity values in Table B7.1 This table also includes the data upon which these values are based.

The values for EVU in Table B7.1 constituted the default HDM-III values (Watanatada, *et al*. 1987a). They are high, particularly for passenger cars. This was also noted by Watanatada, *et al*. (1987b) who stated:

> "[the passenger car utilisation] *is significantly on the higher side, even for commercial usage. It is probably due to the special nature of business in which the cars in the sample were employed, namely, courier work in and around Brasilia. While the Brazil values are treated as defaults, it is recommended that users determine representative values for the region of application."*

Table B7.1: HDM-III Utilisation Values

Vehicle Type	Annual Util. ('000 km)	Mean Trip Length (km)	Average Round Trip Speed (km/h)	Hours Driven (h/yr)	Hours Available (h/yr)	Non-Driving Time (h/trip)	Elasticity of Util. (EVU)
Passenger Cars	95	327	79.8	1162	1972	2.3	0.60
Utilities	39	46	60.2	652	839	0.2	0.80
Heavy Buses	102	328	64.0	1601	2302	1.6	0.75
Trucks	96- 121	188 - 1230	54.6 - 65.6	1400 - 1863	2200 - 2414	0.7 - 4.9	0.85

Source: Watanatada, et al. (1987b)

The EVU represents the additional utilisation that arises from travel time savings[1]. The value of the EVU will vary significantly between individual vehicle classes, and also between urban and interurban operations.

Vehicles with the highest values of EVU have little time spent on non-driving activities. For example, urban taxis are usually available for another trip immediately upon completing a trip. Urban buses may also have high values for EVU; however, this depends upon whether or not the time savings are sufficient to permit additional trips. A bus operating on a 60 min round-trip route may be able to make eight trips in a day. A savings of 5 minutes per trip would therefore have little impact on the utilisation, whereas it would for a bus with a 15-minute round-trip. Private vehicles are usually considered to travel for a fixed number of trips and thus, would have little additional utilisation.

The value adopted for the EVU will have a significant impact on the predicted annual utilisation. Figure B7.1 shows the effect of speed and EVU on the annual utilisation using the Heavy Bus data from Brazil presented in Table B7.1 with an assumed baseline service life of 10 years. Even with the low value of 0.1 for the EVU there is a significant decrease in the annual utilisation with decreasing speeds, although the effects of higher speeds do not lead to a major increase in the utilisation. Conversely, with EVU values above 0.8 there is almost a linear relationship between speed and utilisation.

Halcrow Fox (1982) suggested that the EVU for trucks will be high under the following conditions:

- They operate between seaports, industrial plants and warehouses which are on 24 hour shifts;
- A large proportion of trucks are operated by commercial trucking companies and/or by own-account operators with multi-vehicle fleets;
- The information system for matching the supply of, and demand for, transport is well developed;
- Modal interfaces are good and delays are minimal; and,
- There are few or no time restrictions on the operation of trucks in urban areas.

[1] This is also termed the "fleet substitutability factor" since it represents the reduction in the total fleet requirements to accommodate a given level of transport demand.

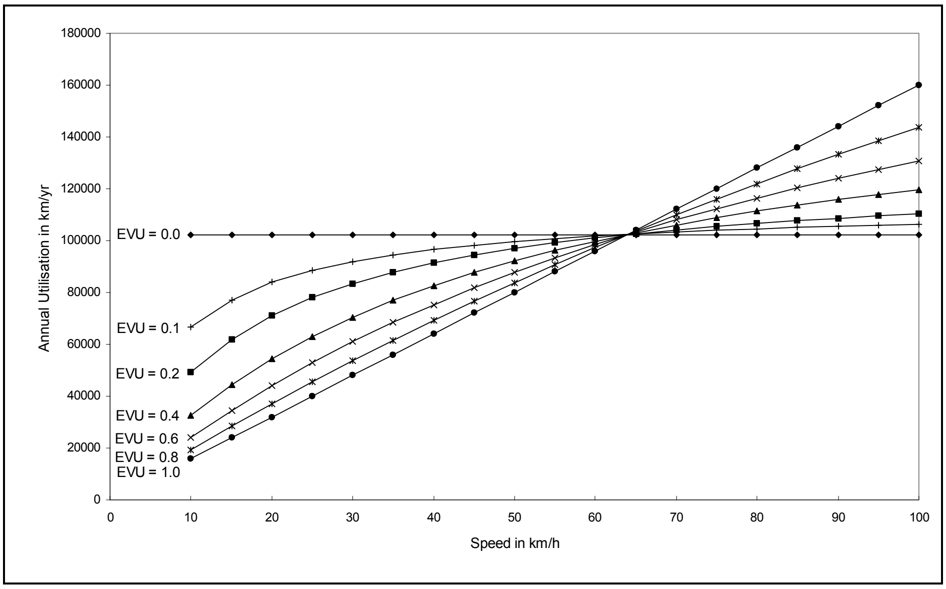

Figure B7.1: Effect of Speed and EVU on Annual Utilisation

As noted earlier, due to the unusual utilisation characteristics of Brazilian vehicles, the default values in HDM-III for the EVU (see Table B7.1) are high for most applications. However, on the basis of data in the review of HDM-III studies (Bennett, 1995a), approximately one-third of those who reported values for EVU adopted the default value. This is illustrated in Table B7.2 which is based on the data from a number of HDM-III analyses in different countries. For most vehicle classes the mean EVU adopted was on the order of 0.5. This suggests that the studies may have over-estimated the utilisation benefits from road improvements.

Table B7.2: Elasticity of Utilisation Values Adopted in HDM-III Studies

Vehicle Class[1]	Elasticity of Vehicle Utilisation					Per cent Using Default
	Mean	Std. Dev.	Minimum	Maximum	Number	
PC	0.40	0.24	0.10	0.80	13	31
LDV-LGV	0.54	0.31	0.15	0.90	15	30
LT	0.54	0.29	0.25	0.85	5	40
MT	0.52	0.28	0.20	0.90	11	27
HT	0.58	0.25	0.30	0.90	12	33
AT	0.64	0.22	0.40	0.90	7	29
LB	0.43	0.28	0.15	0.80	5	20
MB	0.62	0.39	0.17	0.85	3	66
HB	0.55	0.25	0.23	0.90	10	20

Source: Analysis of data in Bennett (1995a)

Notes: 1/ See List of Terms for abbreviations

The data suggests that instead of specifying the number of hours **driven**, some analysts appear to have specified the number of hours a vehicle is **available.** The hours available is the number of hours per year (8760 h/yr) less the time allowed for crew rest, time lost loading, unloading, refueling, finding cargo, repairs, *etc.* Due to these time losses the hours driven are always much lower than the hours available.

Such a mis-specification would lead to anomalous results due to the number of hours available, the number of hours driven, the average speed, and the elasticity of utilisation. This can be reflected as:

$$HRD = EVU\ HAV \qquad \ldots(B7.13)$$

$$S0 = \frac{AKM0}{HRD} \qquad \ldots(B7.14)$$

In HDM-III applications to ensure consistent results it was therefore important that the values for the elasticity of utilisation be selected by also considering the baseline annual speed and the annual utilisation.

B7.2.2 Service Life

HDM-III contained two methods for calculating service life: the Constant Life method and De Weille's Varying Service Life method.

The Constant Life method assumed that the service life is independent of speed. The Varying Life method assumed that there is an inverse relationship between speed and service life which is as follows (Watanatada, *et al*., 1987a):

$$LIFEFAC = \min\left[1.5;\ \frac{1}{3}\left(\frac{S0}{S} + 2\right)\right] \qquad \ldots(B7.15)$$

where LIFEFAC is the vehicle service life factor
S0 is the baseline average speed in km/h

The service life is given by:

$$LIFE = LIFE0\ LIFEFAC \qquad \ldots(B7.16)$$

where LIFE is the service life in years
LIFE0 is the baseline (*ie* user specified) average vehicle life in years

The limit of 1.5 LIFE0 was included by Watanatada, *et al*. (1987a) to eliminate unreasonably high service lives.

The baseline average speed is calculated as:

$$S0 = \frac{AKM0}{HRD0} \qquad \ldots(B7.17)$$

Figure B7.2 shows the effect of the predicted speed on the service life factor for four different base speeds: 25 , 50, 75 and 100 km/h. When the predicted speed is equal to this baseline speed the predicted service life is equal to the baseline service life (*ie* LIFEFAC = 1.0). However, if the speeds are reduced below the baseline speed there is a significant increase in the service life while increasing the speeds above the baseline speed results in a decrease in the service life.

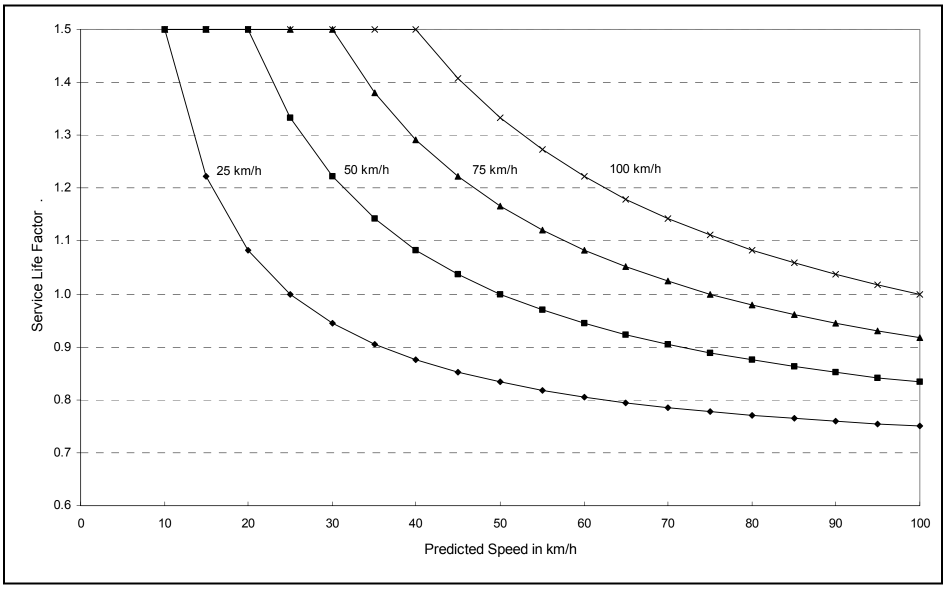

Figure B7.2: Effect of Predicted Speed on Service Life Factor

Bennett (1995a) found that a number of analysts had applied the Varying Life method with HDM-III. Using data given in Bennett (1995a) from studies in 18 countries, the baseline speeds in Table B7.3 were calculated.

The studies from which the values in Table B7.3 were derived covered a wide range of countries and operating conditions. However, the mean baseline speeds for almost all vehicles were less than 50 km/h, with some studies having values as low as 7 km/h. The free speed model would generally be predicting much higher speeds than these, particularly on good standard roads where they can approach 80 to 100 km/h. These higher predicted speeds coupled with low baseline speeds would lead to a decrease in the service life factor and thus a decrease in the service life (see Figure B7.2). This in turn would increase the depreciation and interest costs.

Although some would argue that there is an inverse relationship between vehicle speed and service life, it needs to be recognised that this has never been verified. Furthermore, the reliability of modern vehicles is such that the service life would not be as sensitive to speed as the 30 year old relationship postulated by de Weille. It should be emphasised that the relationship presented in Equation B7.15 has no theoretical or experimental foundation and, on the basis of the data presented in Table B7.3, has been applied in many studies with parameter values which would lead to unreasonably high sensitivities of service life to speed.

B7.2.3 Implications of HDM-III Utilisation and Service Life Method

With HDM-III analysts had their choice of three methods for predicting utilisation and two for predicting service life. As shown in Table B7.4, there has been a great deal of variation in the methods adopted in different studies (Bennett, 1995a).

Table B7.3: Service Life and Utilisation Data from HDM-III Studies

Vehicle Class[1]	Service Life (yr)				Hours Driven (h/yr)				Annual Utilisation ('000 km/yr)				Baseline Speed (km/h)				Number of Studies[2]
	Mean	Std. Dev.	Min.	Max.	Mean	Std. Dev.	Min.	Max.	Mean	Std. Dev.	Min.	Max.	Mean	Std. Dev.	Min.	Max.	
PC-S	9.7	3.2	6	15	505	461	188	2000	17.0	4.3	7	24	46.1	23.3	7.0	95.7	13
PC-M	9.4	4.4	4	15	537	135	417	730	25.8	16.3	16	50	49.9	33.7	27.4	100.0	4
PC-L	11.0	4.2	5	15	2375	1493	500	4000	39.3	19.4	17	60	20.4	9.6	12.5	34.0	4
LDV	7.8	2.8	4	13	946	485	279	2000	31.2	14.6	15	70	40.1	23.9	10.0	86.0	13
LGV	8.2	1.8	5	10	968	511	390	1500	34.8	12.4	20	54	54.7	26.4	28.0	77.8	4
LT	9.4	4.3	4	15	1454	796	600	3000	44.7	21.4	21	80	33.6	11.7	18.2	56.8	9
MT	10.7	3.6	4	16	1393	953	450	4000	44.6	17.0	23	80	40.0	18.2	12.5	71.9	17
HT	10.2	3.1	4	15	1279	479	650	2400	55.1	20.4	30	85	44.8	14.0	25.0	72.0	14
AT	8.9	3.1	4	15	1842	618	850	3000	71.1	21.8	44	116	43.3	19.9	22.0	77.3	10
LB	6.7	3.2	3	12	1309	1033	450	3150	64.8	39.1	30	125	57.0	18.1	33.3	82.0	6
MB	11.0	1.4	10	12	1980	1449	615	3500	110.0	14.1	100	120	47.2	26.3	28.6	65.8	2
HB	9.9	3.0	6	15	1977	985	650	4000	77.4	31.2	38	150	47.2	21.9	12.5	82.0	15

Source: Analysis of data in Bennett (1995a)

Notes: 1/ See List of Terms for abbreviations.
2/ Not all studies used all vehicle classes.

Table B7.4: Methods Adopted for Annual Utilisation and Service Life[1]

Country	Annual Utilisation			Service Life	
	Constant		Adjusted	Constant	Varying
	km	hours			
Australia	•		•	•	
Bangladesh			•		•
Barbados			•		•
Botswana			•		
Burundi	•		•	•	•
Ethiopia			•		
Guatemala			•	•	•
India	•				•
Jordan	•				•
Indonesia			•		•
Myanmar	•			•	
Nepal	•		•	•	
Nigeria	•			•	•
Romania			•		•
Russia	•		•	•	
Spain	•			•	
South Africa			•		•
Thailand	•			•	
Trinidad			•	•	•

Source: *Bennett (1995a)*

NOTES: 1/ In some studies different methods were used for different vehicles so more than one entry is made in the table.

In 11 of the 19 countries listed in Table B7.4 the varying life method was applied. No study was identified which applied the constant hourly utilisation method and many studies applied both the constant kilometreage and adjusted utilisation methods. The constant kilometreage was generally applied to light vehicles (*eg* passenger cars).

The service life and utilisation options in HDM-III allow the user to have four possible speed effects modelled:

- Constant Life - Constant Utilisation
- Constant Life - Varying Utilisation
- Varying Life - Constant Utilisation
- Varying Life - Varying Utilisation

Depending upon which combination of methods was used, one would get significantly different impacts of speed on depreciation and interest costs.

As will be discussed in Chapter 9, in HDM-III depreciation and interest costs were modelled using a straight-line depreciation technique:

$$\text{DEPFAC} = \frac{1}{\text{LIFE AKM}} \qquad \text{...(B7.18)}$$

$$\text{INTFAC} = 0.5\ \frac{\text{AINV}}{100}\ \frac{1}{\text{AKM}} \qquad \text{...(B7.19)}$$

where DEPFAC is the vehicle depreciation factor as a fraction of the new vehicle price per km

INTFAC is the interest factor as a fraction of the depreciable vehicle price per km

AINV is the annual interest rate on investments in per cent

By making utilisation or service life sensitive to speed one influences the depreciation and interest costs.

Figure B7.3 illustrates the effect of speed on the HDM-III depreciation factor (Equation B7.18) using these four combinations. This figure was established using the Heavy Bus data from Brazil presented in Table B7.1 with an assumed baseline service life of 10 years.

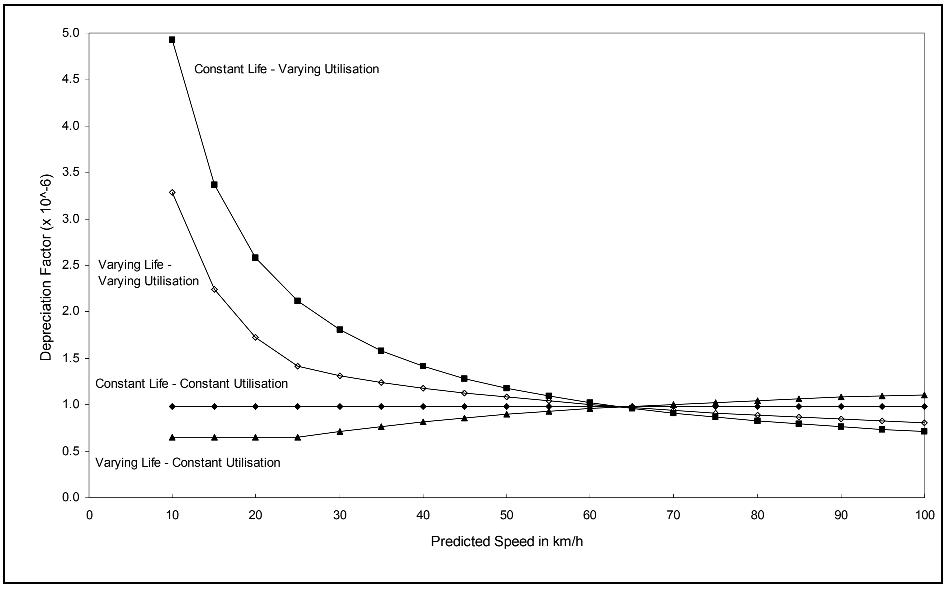

Figure B7.3: Effect of Speed and Utilisation Option on HDM-III Depreciation Factor

It can be seen from Figure B7.3 that depending upon which options were selected there would be markedly different depreciation costs as a function of speed. The differences can be summarised as:

Case	Effect of Predicted Speeds Being Lower Than Base Speeds on Costs
Constant Life - Constant Utilisation	None
Constant Life - Varying Utilisation	Increases Costs
Varying Life - Constant Utilisation	Decreases Costs
Varying Life - Varying Utilisation	Varies Depending Upon EVU

The combination of varying life and varying utilisation can lead to some unusual predictions. This is illustrated in Figure B7.4 which shows the same data as Figure B7.3, but with EVU values of 0.25, 0.50, 0.75 and 1.00. When the EVU become small enough the predictions initially decrease with decreasing speed before increasing again.

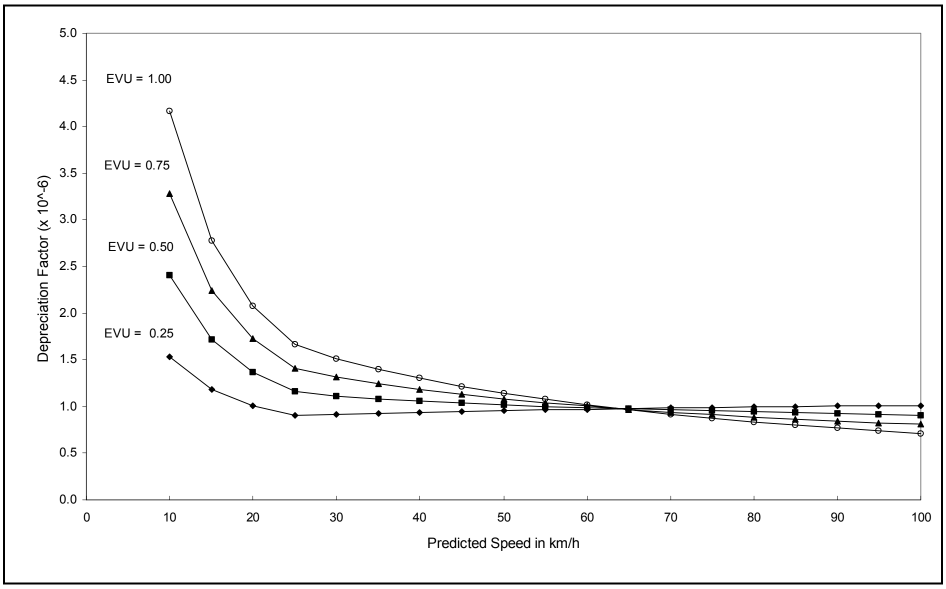

Figure B7.4: Effect of EVU on Depreciation Factor With Varying Life

B7.2.4 Conclusions on HDM-III Methods

On the basis of the above review, it was concluded that for HDM-4 several changes should be made to the modelling of utilisation and service life:

- Given the lack of theoretical or empirical evidence supporting de Weille's varying life method, and the sensitivity of the predicted depreciation costs to speed when it is used, it should not be employed in HDM-4;
- The service life should, if possible, be a function of the operating conditions; and,
- Only the adjusted utilisation method is attractive for its flexibility and ability to cater for constant kilometreage or constant hourly utilisation scenarios, however, the parameters such as the EVU needs to be carefully selected to ensure reasonable results.

B7.3 Other Techniques for Predicting Utilisation and Service Life

B7.3.1 Introduction

Several alternative methods to that in HDM-III have been used to predict utilisation and service life. They can broadly be grouped into regression models and optimal life models. Each of these are described separately in the sections that follow.

B7.3.2 Regression Models

Thoreson and Roper (1994) indicate that in an ARRB modified version of HDM-III the annual utilisation has been modified to make it a function of roughness. This serves to decrease the utilisation with increasing roughness, thereby increasing the depreciation costs. The following is the equation applied:

$$AKM = AKM0\ (1 - 0.026\ (\max(RI, 3) - 3)) \qquad \ldots(B7.20)$$

The basis for this equation is not known and Thoreson and Roper (1994) note "a need for empirical justification of the roughness effect demonstrated". This is particularly important since the same equation is applied to all vehicle classes, whereas one would expect roughness to influence the utilisation of different vehicles differently, such as was found in Brazil (Chesher, *et al*., 1980).

In South Africa, Schutte (1994) noted that the approach of calculating the per kilometre cost by dividing DEPCRT by utilisation is arguably incorrect since it implies a "static" model. Instead, the depreciation costs were defined to be a function of the vehicle life which was defined as (Schutte, 1994):

$$ELIFE = \frac{LIFEKM}{AKM} \qquad \text{…(B7.21)}$$

where LIFEKM is the lifetime utilisation in km
ELIFE is the effective vehicle life in years

Schutte (1994) proposed the following equation, for predicting the life kilometreage, the basis of which is not known:

$$LIFEKM = 200{,}000 - \max(0, 0.8\ (13\ RI - 40)^{2.266}) \qquad \text{…(B7.22)}$$

AASHTO (1977) proposed the following equations for predicting the utilisation and service life as a function of speed:

$$AKM = AKM0 \left(\frac{S}{S0}\right)^{EVU} \qquad \text{…(B7.23)}$$

Passenger Cars:

$$LIFE = \begin{cases} a1/AKM & \text{if } AKM \leq 16000 \\ a2 + a3/AKM + a4/AKM^2 & \text{if } AKM > 16000 \end{cases} \qquad \text{…(B7.24)}$$

Trucks and Buses:

$$LIFE = a5/AKM^{a6} \qquad \text{…(B7.25)}$$

where a0 to a6 are coefficients

Pienaar (1984) used a variation Equation B7.23 for predicting the effect of speed on service life:

$$LIFE = LIFE0 \left(1/\left(\frac{S}{S0}\right)^{EVU}\right)^{EVU} \qquad \text{…(B7.26)}$$

B7.3.3 Optimal Life Models

INTRODUCTION

The optimal life (OL) approach is used to predict the service life. It is predicated on there being an optimal point in the life of the vehicle for it to be scrapped. This is based on the cost of operating the vehicle. Originally proposed by Nash (1976), there have been two applications of it to RUE: that by Chesher and Harrison with another by Hine. Each of these are discusssed below.

CHESHER AND HARRISON

The method proposed by Nash (1976) and was refined by Chesher, *et al*. (1980); Chesher and Harrison (1987); and Chesher (1990). The following description is based on these later three references.

The costs of operating a vehicle at any given time is the sum of its running costs, defined as the fuel, lubricant, tyre, crew and maintenance costs—but **excluding** depreciation and interest costs. The total running costs from new until the optimal life can be calculated as:

$$\int_0^{OL} RUN(t)dt \qquad \ldots(B7.27)$$

where RUN — are the running costs at time t
OL — is the optimal year for scrapping

Running costs increase as a vehicle ages, otherwise there would never be a need to scrap the vehicle. The running costs are always non-zero at any point in the vehicle's life since there will always be costs for fuel, tyres, *etc.* However, it should be noted that only the maintenance costs change over time[1].

The present value of the stream of running costs is:

$$\frac{1}{1-e^{-ir\,OL}}\int_0^{OL} RUN(t)e^{-ir\,t}\,dt \qquad \ldots(B7.28)$$

where ir — is the interest rate as a decimal

Including the capital cost of the vehicle (less tyres since these are included with the running costs) the total present value is:

$$PVC = \frac{1}{1-e^{-ir\,OL}}\left(NVPLT + \int_0^{OL} RUN(t)e^{-ir\,t}\,dt\right) \qquad \ldots(B7.29)$$

where PVC — is the present value of the total cost
NVPLT — is the new vehicle price less tyres

By differentiating Equation B7.29 it can be shown that the optimum vehicle life arises when:

$$RUN(OL) = ir\ PVC \qquad \ldots(B7.30)$$

Equation B7.30 is shown by Chesher and Harrison (1987) to be equivalent to:

$$NVPLT = \int_0^{OL} [RUN(OL)\ RUN(t)]\, e^{ir\,t}\ dt \qquad \ldots(B7.31)$$

1 This is an underlying assumption of the OL model as presented by Chesher and Harrison (1987). In reality, the other VOC components such as tyres and fuel consumption can change over time but these changes are minor.

The value of the vehicle at time t is equal to the present value (at time t) of the remaining stream of net revenues to be provided by the vehicle:

$$\mathrm{VEHVAL}(t) = e^{ir\,t} \int_{t}^{OL} [\mathrm{RUN}(OL)\ \mathrm{RUN}(x)]\, e^{ir\,x}\ dx \qquad \ldots(B7.32)$$

where VEHVAL is the value of the vehicle at time t

The depreciation is by definition the change in the vehicle value over time. Differentiating Equation B7.32 gives:

$$\mathrm{DEP}(t) = \frac{d\mathrm{VEHVAL}(t)}{dt} = \ ir\mathrm{VEHVAL}(t)\ \ \mathrm{RUN}(t)+\mathrm{RUN}(OL) \qquad \ldots(B7.33)$$

where DEP(t) is the depreciation cost at time t

The interest costs at time t are the product of the interest rate and the vehicle value. Equation B7.33 can therefore be rewritten as:

$$\mathrm{RUN}(OL) = \mathrm{DEP}(t) + \mathrm{INT}(t) + \mathrm{RUN}(t) \qquad \ldots(B7.34)$$

Figure B7.5 illustrates the optimal life. By virtue of Equation B7.31, the optimal life arises when the shaded area above the running costs is equal to the discounted vehicle price.

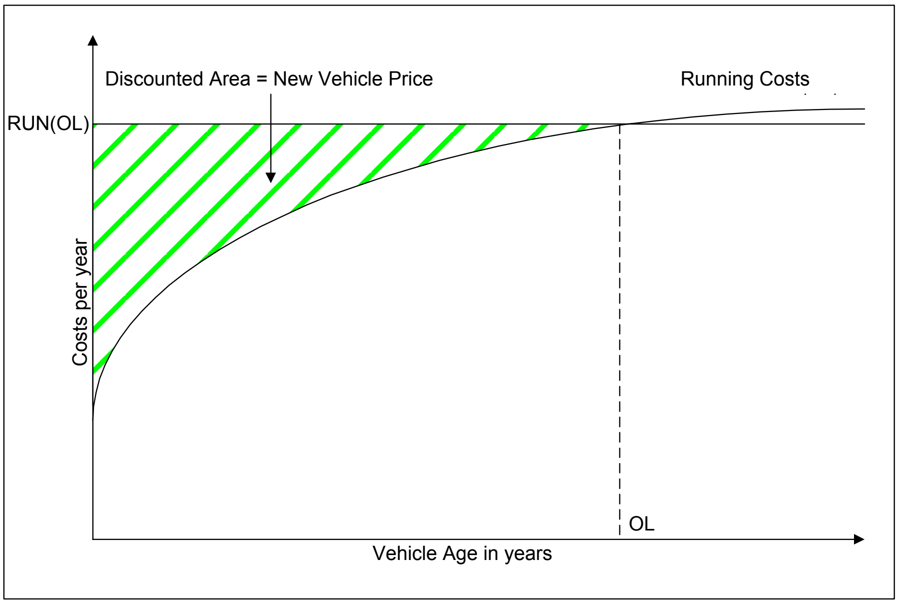

Figure B7.5: Costs, Vehicle Age and Optimal Scrapping

Equation B7.34 implies the following:

- At the date of scrapping, the depreciation and interest costs are zero;
- The total vehicle operating costs are invariant with age;
- The portion of the total vehicle operating costs due to depreciation and interest decrease with increasing age;

- If we know the optimal life, OL, then by calculating the terminal running costs we have an age invariant prediction of the total vehicle operating costs.

The running costs are comprised of both costs that are static and others that vary with age. Since essentially the only VOC component which varied significantly with age was the maintenance and repair costs, it is therefore possible to rewrite the above expressions as:

$$NVPLT = \int_{0}^{OL} [MAINT(OL)\ MAINT(t)]\, e^{ir\,t}\ dt \qquad \ldots(B7.35)$$

$$MAINT(OL) = DEP(t) + INT(t) + MAINT(t) \qquad \ldots(B7.36)$$

where MAINT are the maintenance and repair costs

Chesher and Harrison (1987) give an example of applying the OL method to the data collected in the Road User Cost studies and compare the results to those from the value-age (VA) method. It is shown that the OL results are more consistent and reasonable than those from the VA method.

However, it was found by Chesher and Harrison (1987) that the lives predicted by the OL method were longer than the vehicle lives observed in the studies. This was ascribed to several factors:

- The maintenance equations likely underpredict the true maintenance cost since omissions of costs are more likely to arise than spurious additional costs;
- Maintenance expenditures arise at discrete points in time and for older vehicles can be quite significant. This would lead owners to scrap the vehicles early;
- It is likely that the older vehicles observed in the surveys were those with lower costs as higher cost vehicles would have been retired early;
- It is likely that the fuel, tyre and oil costs also likely increase with age, although these effects were not quantified in the studies; and,
- In practice, vehicles are scrapped not just because of their costs but due to technological obsolescence.

The Chesher and Harrison OL method was applied in South Africa by du Plessis and Schutte (1991). Their results indicate that there is a decrease in the optimal life with increasing roughness. However, the predictions of the depreciation and interest costs for all vehicles except passenger cars give a decrease in costs with increasing roughness which is contrary to what is expected. du Plessis and Schutte (1991) were not able to offer an explanation for this decrease with increasing roughness.

While the Chesher and Harrison OL method has many attractions, there are potential problems with it that need to be recognised:

- Rational behaviour on the part of operators in assessing “optimum scrapping date” is assumed;
- In the analytical solution proposed by Chesher and Harrison (1987) it is assumed that utilisation is independent of vehicle age, which is often not the case (Cox, 1995);
- It does not allow for a residual value of the vehicle; and,
- It assumes that the total operating costs are independent of vehicle age.

To test the transferability of the Chesher and Harrison OL method for HDM-4, it was tested using values from the Brazil study as well as values adopted by analysts in three different studies from different countries. Two of these studies calibrated the Brazil parts model and all three had their own unique cost data.

The analysis was undertaken using a discount rate of 10 per cent. For each vehicle class, the OL was calculated—in years and kilometres—at a range of roughnesses from one to 20 IRI. The total running cost at the optimal life (maintenance and capital costs) were also established. Table B7.5 shows the results of this analysis and it shows that the optimal lives for the Chesher and Harrison (1987) approach are extremely high for most vehicles. In some instances, the values are patently unreasonable.

As described later in Section B7.5, it was considered that thee overpredictions arose due to a deficiency in the Chesher and Harrison (1987) OL method. This was addressed for HDM-4 by a modification to the method.

HINE (1982)

Hine (1982) proposed a similar approach to that of Chesher and Harrison (1987) for calculating the optimal life. The optimal life was defined as "the point of kilometreage which minimises the costs per kilometre". The total maintenance costs at any kilometreage can be defined as the integral of the parts and labour costs. As shown in Annex 9.1, this can be expressed as[1]:

$$\text{MCSTCKM} = \frac{\text{m0 CKM}^{\text{kp+1}}}{\text{kp}+1} + \frac{\text{m1 CKM}^{\text{m2kp+1}}}{\text{m2 kp} + 1} \qquad \text{…(B7.37)}$$

where MCSTCKM is the maintenance and repair costs at cumulative kilometreage CKM

CKM is the cumulative kilometreage in km

kp is the age exponent in the HDM-III parts consumption model

m0 to m2 are simplifications of the HDM-III parts and labour model terms

Including the capital costs of the vehicles, the total maintenance and capital costs on a per kilometre basis is:

$$\text{KMCST} = \frac{\text{m0 CKM}^{\text{kp}}}{\text{kp}+1} + \frac{\text{m1 CKM}^{\text{m2 kp}}}{\text{m2 kp} + 1} + \frac{\text{NVPLT}}{\text{CKM}} \qquad \text{…(B7.38)}$$

where KMCST is the total maintenance and capital costs on a per kilometre basis

The total maintenance and capital costs are at a minimum when the derivative of Equation B7.38 is zero.

1 Hine (1982) only considered parts consumption explicitly. Labour costs were assumed to be 45 per cent of the parts costs. Since HDM-III has different expressions for these two components, in describing the Hine (1982) approach they are considered separately here.

Table B7.5: Results of Evaluation of Chesher and Harrison (1987) OL Method

Vehicle Class	Country[1]	Annual Util. ('000 km/yr)	Replace-ment Vehicle Price (cost/ veh)	Labour Cost (cost/h)	Roughness of 3 IRI m/km			Roughness of 6 IRI m/km			Roughness of 9 IRI m/km		
					Optimal Life		Run. Cost at OL	Optimal Life		Run. Cost at OL	Optimal Life		Run. Cost at OL
					(km) (000s)	(yr)	(cost/ 1000 km)	(km) (000s)	(yr)	(cost/ 1000 km)	(km) (000s)	(yr)	(cost/ 1000 km)
PC (Private)	Brazil	20	43000	13	4655	232.75	332.75	1722	86.08	409.72	783	39.17	536.77
PC (Commercial)	Brazil	93	43000	13	1194	12.83	227.15	744	8.00	323.22	481	5.17	466.81
HB	Brazil	94	316970	13	2945	31.33	715.79	2554	27.17	765.68	2233	23.75	822.04
MT	Brazil	101	138621	13	1380	13.67	579.66	808	8.00	859.33	598	5.92	1101.52
AT	Brazil	101	437704	13	1069	10.58	2169.70	816	8.08	2647.52	673	6.67	3092.29
PC	Bangladesh	20	270000	38		>400			>400		7085	354.25	2103.37
HB	Bangladesh	65	1330000	42	3602	55.42	3410.58	2275	35.00	4034.43	1701	26.17	4625.81
MT	Bangladesh	50	670000	38	4308	86.17	2337.89	2354	47.08	2813.92	1642	32.83	3270.39
AT	Bangladesh	60	4220000	42	5650	94.17	11438.30	3285	54.75	13355.20	2325	38.75	15238.96
PC	Hungary	11.4	11200	5		>400		3871	339.58	144.53	1344	117.92	174.56
HB	Hungary	70	195200	5	3885	55.5	460.66	3278	46.83	487.21	2794	39.92	517.63
MT	Hungary	35	97600	5	3060	87.42	470.77	1266	36.17	633.64	820	23.42	785.09
AT	Hungary	50	305000	5	1475	29.50	1525.14	1050	21.00	1832.85	825	16 .50	2118.80
PC	Sri Lanka	15	237000	40		>400			>400			>400	
HB	Sri Lanka	65	833000	40	3602	55.42	2190.49	2513	38.67	2489.29	1955	30.08	2772.82
MT	Sri Lanka	40	385000	40	3973	99.33	1601.60	2113	52.83	1916.06	1450	36.25	2219.67
AT	Sri Lanka	30	1200000	40	4653	155.08	5972.42	2300	76.67	6991.38	1503	50.08	7983.15

NOTES: 1/ Brazil: Data from Chesher and Harrison (1987). Costs in 1976 Cruzeiros.
Bangladesh: Data from Transroute (1992). The calibration coefficients provided for the individual vehicles in Bangladesh were also used in the calculations. Costs in 1992 Tika.
Sri Lanka: Data from Transport and Studies Centre (1992). The calibration coefficients provided for the individual vehicles in Sri Lanka were also used in the calculations. Costs in 1992 Rp.
Hungary: Data from Transport Science Association (1995). This application used the default HDM-III calibration coefficients. Costs in 1994 C.

This leads to the following equation[1]:

$$\text{NVPLT} = \frac{\text{m0 kp CKM}^{\text{kp}+1}}{\text{kp}+1} + \frac{\text{m1 m2 kp CKM}^{\text{m2 kp}+1}}{\text{m2 kp}+1} \qquad \ldots(\text{B7.39})$$

The optimum vehicle life is the cumulative kilometreage (CKM) which solves Equation B7.39.

This approach of using calculus has a major difference from that of Chesher and Harrison (1987) in that it ignores discounting[2]. In common with the Chesher and Harrison (1987) approach it also ignores technical progress. Although NDLI (1995) indicated that it gave different results to that from Chesher and Harrision, this was incorrect. Hine (1995) indicates that it in fact yields equivalent results.

B7.4 Development of HDM-4 Optimal Life Method

B7.4.1 Introduction

NDLI (1995) proposed a model for predicting utilisation and service life in HDM-4, however, a number of deficiencies with this model were identified at the 1995 TRL Workshop on Road User Effects. Bennett (1996a) describes how the workshop participants considered that it was imperative to have some form of OL method in HDM-4 in order for the depreciation and interest costs to be consistent and rational. This section describes how the method was developed.

B7.4.2 Preliminary Developments

Bennett (1996a) describes how solutions to two of the problems encountered by NDLI (1995) in testing the OL method were suggested at the workshop:

UNDERPREDICTING AGE EFFECTS

The age effects in the parts model are biased since only survivors were sampled, not the high cost vehicles which had been removed from service. Chesher (1990) showed that this could lead to approximately a 25 per cent underprediction of the age effects for a vehicle. Accordingly, it was proposed that the parts consumption age term be increased 25 per cent for all vehicles in HDM-4.

EFFECT OF VARYING UTILISATION

Chesher (1995) describes an adaptation of the OL method to consider varying utilisation with age. This shows that varying utilisation leads to shorter lives and higher costs.On the basis of these observations, further tests of the OL method were conducted with the same approach as in NDLI (1995). It was found that the 25 per cent increase in the age term served to significantly shorten the predicted OL and that by adjusting this term by different percentages, less than 25 per cent for some vehicles, one could obtain "reasonable" predicted values for the OL.

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[1] There is an error in the equation as used in the draft working paper by Bennett (1995b). This has been corrected in the material presented here.

[2] It should be noted that the application of this technique by Hine (1982) considered discounting and other effects.

The HDM maintenance model can be re-written as:

$$PC = K0pc\ [CKM^{kpfac\ kp}\ (a0 + a1\ RI) + K1pc\ ]\ (1 + CPCON\ dFUEL) \quad \ldots(B7.40)$$

$$LH = K0lh\ [a2\ PC^{a3}] + K1lh \quad \ldots(B7.41)$$

where PC is the parts consumption
LH is the number of labour hours
CKM is the vehicle cumulative kilometreage in km
kp is the parts model age exponent
kpfac is the age exponent calibration factor (default=1.0)
CPCON is the elasticity of congestion fuel on parts (default=0.1)
dFUEL is the additional fuel due to congestion as a fraction
K0pc, K0lh are calibration factors (default=1.0)
K1pc, K1lh are calibration factors (default=0.0)

The additional coefficient kpfac in Equation B7.40 adjusts the age exponent which serves to alter the magnitude of the time stream of running costs. This is illustrated in Figure B7.6.

Figure B7.6: Effect of Varying kpfac on Costs

By virtue of the underlying propertie of the OL method, when the kpfac value is increased there is an increase in the running costs and a shortening of the time to the OL. Conversely, decreasing kpfac will see the running costs decreased and the time to the OL extended. Thus, by modifying kpfac one can calibration the OL prediction. This is illustrated in Figure B7.7 which shows the impact of varying kpfac on the predicted OL. The ‘modified’ curve represents that which would arise with an increased kpfac; the ‘default’ curve with kpfac = 1. The OL is arises when the discounted area is equal to the replacement vehicle price. Thus, the discounted areas A B C and A D E are equal.

Bennett (1996a) investigated the implications of the method suggested in Figure B7.7. It was suggested that the user define a ‘target’ OL at a roughness of 3 IRI m/km: 300,000 km for cars and utilities; 500,000 km for light and medium trucks/buses; 750,000 for other vehicles. By defining the replacement cost of the vehicle and the annual utilisation the age exponent could be varied so that the predicted OL was equal to the target OL.

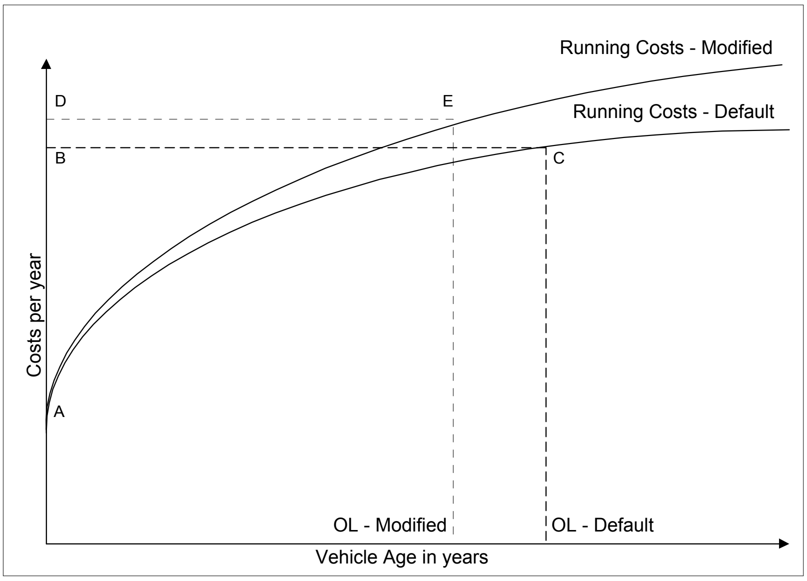

Figure B7.7: Effect of Varying Age Exponent on OL

A series of analyses were conducted using the original Brazil data as well as data from other studies. It was found that the age factor adjustments varied significantly between applications but that in almost all cases it was possible to calibrate the OL method using this approach. It was therefore recommended for inclusion in HDM-4.

B7.4.3 Refinements to Model from Thailand

The modified OL method proposed by Bennett (1996a) was tested using data from Thailand. It was found that there were two shortcomings with the approach.

PARTS MODEL

The HDM-4 parts consumption model proposed by NDLI (1995) was normalised to 100,000 km. This served to 'rotate' the predictions around this value instead of the origin as suggested in Figure B7.7. It was therefore recommended that the model form revert back to a linear form rotating about the origin.

CALIBRATION TO KNOWN RATES

A more fundamental problem was found when calibrating both the parts model and the OL model. In Thailand data were available on the maintenance costs at a certain age. This gave two points for calibrating the function: the target OL (*eg* 300,000 km) and the known parts consumption at half the OL. It did not prove possible to obtain a calibration with both of these values in spite of a number of different approaches being tried. If one calibrated the parts function so that the predicted rates corresponded at half the OL, the predicted OL was significantly higher than the target OL. Conversely, if one calibrated to the target OL the predicted parts consumption at half the OL was significantly higher than the known value.

This problem is illustrated in Figure B7.8. The known parts consumption is "A". The default OL for the parts model ("D") is greater than the known life "C". However, if the value for kpfac it modified so that the predicted OL is equal to C, the parts consumption A is increased to "B".

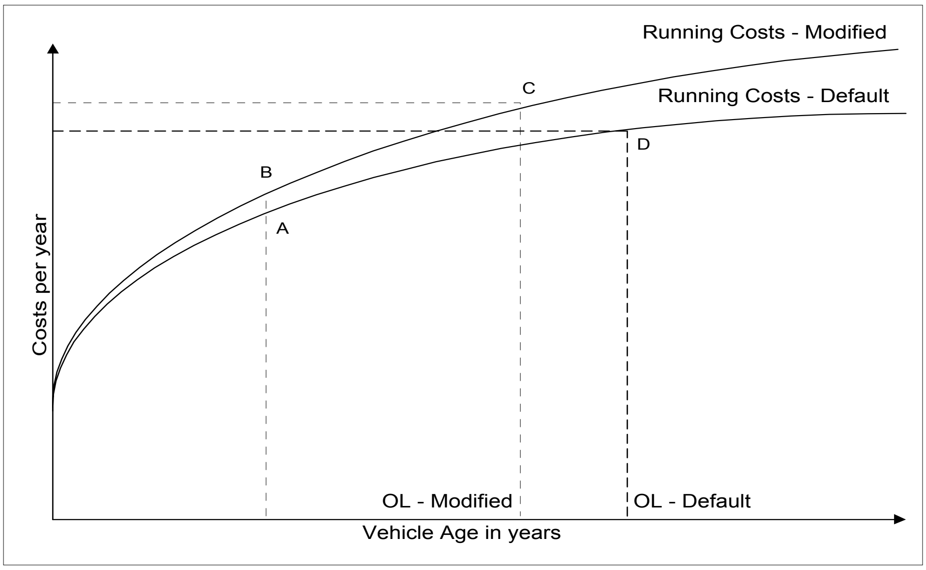

Figure B7.8: Parts Calibration Problem

B7.4.4 Final Optimal Life Implementation for HDM-4

At first glance the Thailand results shown in Figure B7.8 presents a conundrum since it is important to calibrate all models in HDM. However, it needs to be appreciated that the Chesher and Harrison (1987) OL method does not consider all factors which influence scrapping. In additional to the time stream of maintenance costs, the following factors also influence the decision of an operator to scrap a vehicle:

- **The probability of a breakdown:** The vehicle may be functional but as it ages there is an increased probability of a major breakdown. If the financial costs of such a breakdown are significant, the vehicle will be scrapped before its OL.
- **The sampling of survivors:** In all the road user cost studies the vehicles which were sampled were those that were still in operation. Vehicles which had significantly higher operating costs had been removed from the fleet. Chesher (1990) shows that this sampling bias led to an underestimation of the age exponent in the Brazil study for buses of about 25 per cent.
- **Technical progress:** Vehicles are scrapped because of technical improvements made in newer vehicles, for example significant reductions in running costs.
- **Maintenance costs at end of life:** As the vehicle approaches its OL the operator postpones forthcoming costs so the actual cost is artificially low.
- **Decreased utilisation:** If the vehicle has significant periods of inactivity there is an increase in the fixed costs due to the reduction in utilisation.

It can therefore be argued that instead of using the actual running costs it is necessary to use a 'shadow' cost for calculating the OL. This would be higher than the actual costs and reflect the intangibles above that cannot be modelled. This approach was intimated by Bennett (1996a), although not in the same manner.

The implication of adopting a shadow cost approach is that one can calibrate the OL method solely through varying the age exponent and not worry that the predicted maintenance costs upon which the OL are based are higher than the actual maintenance costs. The difference in values would be attributable to the intangibles above.

The shadow cost approach presents a different problem in that the Chesher and Harrison (1987) OL method defines the capital costs at an time as:

$$\text{DEP(t)} + \text{INT(t)} = \text{RUN(OL)} - \text{RUN(t)} \qquad \ldots\text{(B7.42)}$$

Since the increased age factor will distort the values for the running costs, it is no longer correct to define the capital costs in this manner. To overcome this problem the following approach was adopted for HDM-4:

- The user defines the target OL. This is the expected service life of the vehicle in km. In HDM-III the user already supplies this value by defining the service life and average annual utilisation.
- The user also defines the replacement vehicle price and utilisation behaviour.
- The age exponent calibration factor (kpfac) is calibrated using these values.
- The effect of roughness on the OL is calculated using these values and the calibrated age exponent. This gives the lifetime kilometreages for the vehicle at a range of roughnesses.

It should be noted that in this approach financial instead of economic costs should be used to establish the OL. This is because operators make their decisions based on financial costs.

The approach also does not call for any additional data over that which was used in HDM-III.

Figure B7.9 shows the predicted effects of roughness on the OL with this approach using data from Thailand, where the life is expressed as a percentage of the target OL. Although there were nine vehicle types used in the analysis – ranging from motorcycles through cars to trucks and buses – it can be observed that there is a great deal of similarity in the predictions. This was unexpected given the different parts consumption equations and replacement values of the vehicles. The lower curve on the figure is for the heavy bus while the upper figure is for the motorcycle.

Although the predictions in Figure B7.9 go as high as 20 IRI, the original Brazil data were only based on roughnesses up to 11.5 IRI. Also, in Thailand there are few roads with roughnesses above 10 IRI. It was therefore decided to fit a regression to the data points up to 10 IRI.

Because of the similarities between the predicted OL values they were averaged. The following equation was fitted to the data (R^2 = 0.99; S.E. = 1.31):

$$\text{LIFEKMPCT} = \min\left(100, \frac{100}{1 + \exp(-65.8553\ \text{RI}^{-1.9194})}\right) \qquad \ldots\text{(B7.43)}$$

$$\text{LIFEKM} = \frac{\text{LIFEKM0 LIFEKMPCT}}{100} \qquad \ldots\text{(B7.44)}$$

where LIFEKMPCT is the lifetime kilometreage as percentage of baseline life
LIFEKM0 is the baseline (*ie* user specified) service life in km
LIFEKM is the predicted service life in km
RI is the roughness in IRI m/km

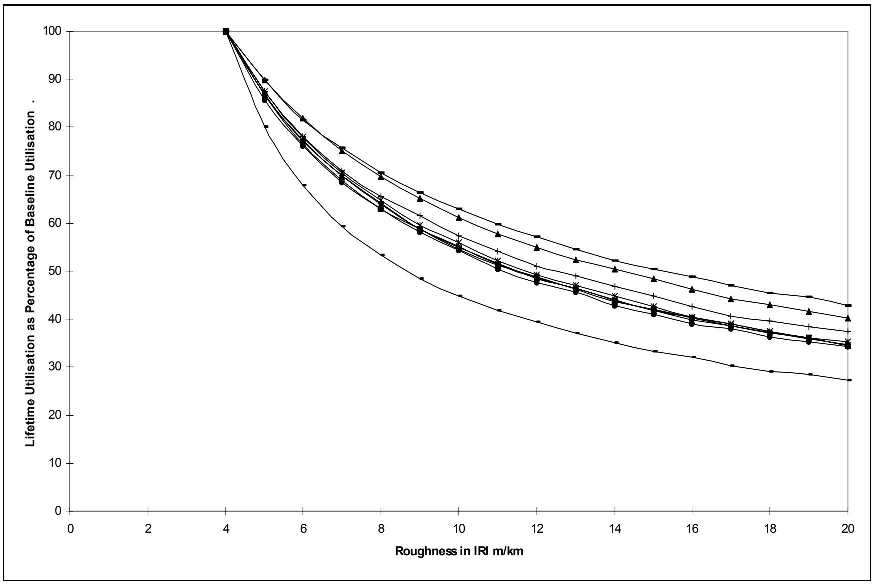

Figure B7.9: Effect of Roughness on Optimal Life - Thailand Data

Figure B7.10 shows the predicted effect of roughness on the lifetime utilisation using the above equation. It can be observed that the predictions are much less sensitive to roughness above 10 IRI than with the original data as shown in Figure B7.9.

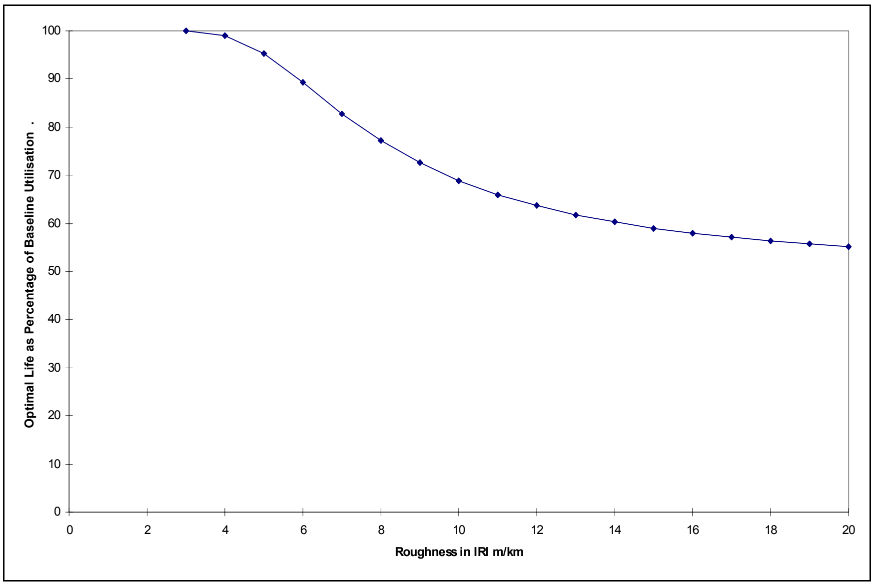

Figure B7.10: Effect of Roughness on Lifetime Utilisation

To undertake the analyses necessary to use this implementation with HDM-4 a program OLCALIB was written as part of the HDM Tools suite (Bennett and Paterson, 1999). Figure B7.11 shows the input screen for the program. The program uses the following data:

- Replacement value of the vehicle;
- Cost of maintenance labour, and
- Lifetime utilisation (km).

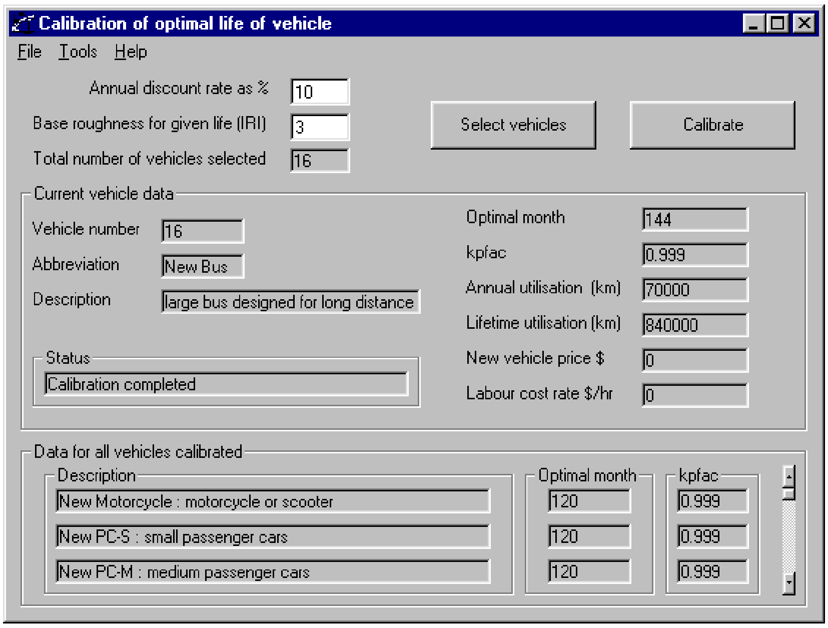

Figure B7.11: Optimal Life Simulation Input Screen

To run the program it is necessary to supply a base roughness level. This is the roughness at which the lifetime utilisation applies. For example, if the average roughness on the network is 6 IRI m/km and the average lifetime utilisation is 100,000 km you would supply the value of 6. However, if you estimated that on smooth roads the lifetime utilisation would be 200,000 km and this value is entered in the data file, you would enter 3 IRI m/km for the roughness.

The user selects the vehicles to analyse and the optimal life at different roughnesses is generated. This is done by varying the parts consumption coefficient kp based on the user-supplied values. Once it has the value of kp that matches the optimal life information, it then calculates the life at a roughness of the 3 IRI m/km (which if the base roughness is 3 equals the lifetime utilisation). It then goes on to calculate the percentage of life at various roughness levels relative to the life at a roughness of 3 IRI m/km (that is, not the supplied lifetime utilisation). Table B7.6 is an example of the program output. This would then be analysed to develop the regression model parameters.

Table B7.6: Example of Optimal Life Analysis Output

Representative vehicle	Roughness (IRI m/km)	Optimal life as a percentage of baseline life
1	3.0	97.1
1	3.5	89.6
1	4.0	83.3
1	4.5	77.9
1	5.0	73.3
1	5.5	69.6
1	6.0	65.8
1	6.5	62.9
1	7.0	60.0
1	7.5	57.5
1	8.0	55.4
1	8.5	53.3
1	9.0	51.3
1	9.5	49.6
1	10.0	47.9

B7.5 Utilisation and Service Life Modelling in HDM-4

B7.5.1 Utilisation

In HDM-4 vehicle utilisation is expressed in terms of the annual kilometreage driven during the annual working time. Working time (HRWK0) as defined by Hine (1996) is the time spent undertaking the essential tasks of a return trip. This excludes time spent idle, when the driver is eating, sleeping or otherwise resting, but includes time spent driving, loading, unloading and refuelling.

The average annual utilisation (AKM0) is either entered as a constant value by the user or if the user specifies an age distribution along with the percentage of vehicles at each age:

$$AKM0 = \sum_{i=1}^{n} \frac{AKMVi\ PCVTi}{100} \qquad \text{…(B7.45)}$$

where AKM0 is the average annual utilisation in km/yr
AKMVi is the annual kilometreage of vehicles with age i
PCTVi is the percentage of vehicles of age i in the fleet

The average annual working time (HRWK0) is either entered as a constant value by the user or if the user specifies an age distribution along with the percentage of vehicles at each age:

$$HRWK0 = \sum_{i=1}^{n} \frac{HRWKi\ PCTVi}{100} \qquad \text{…(B7.46)}$$

where HRWK0 is the average annual utilisation in hr/yr
HRWKVi is the annual working hours of vehicles with age i
PCTVi is the percentage of vehicles of age i in the fleet

B7.5.2 Service Life

The following two methods of calculating vehicle service life are provided in HDM-4:

- Constant Life method; and,
- Optimal Life method.

The user chooses which of the two methods should be used for calculating vehicle parts consumption and for modelling capital costs.

CONSTANT VEHICLE LIFE METHOD

The vehicle life, LIFE, is assumed to be constant irrespective of vehicle speed or operating conditions and is equal to the user-specified value.

OPTIMAL VEHICLE LIFE METHOD

The optimal life of the vehicle under the condition of different road roughness values is determined as follows:

$$LIFEKM = \frac{LIFEKM0\ LIFEKMPCT}{100} \qquad \text{…(B7.47)}$$

where LIFEKM is the optimal lifetime utilisation in km
LIFEKM0 is the average service life in km
LIFEKMPCT is the optimal lifetime kilometreage as a percentage of baseline life.

The average service life (or baseline life) is calculated from the expression

$$\text{LIFEKM0} = \text{AKM0 LIFE0} \quad \text{…(B7.48)}$$

where AKM0 is the user defined average annual utilisation in km
LIFE0 is the user defined average service life in years

The optimal life as a percentage of the user defined baseline vehicle life.

$$\text{LIFEKMPCT} = \min\left(100, \frac{100}{1 + \exp(\text{a0 } \text{RI}_{\text{adj}}^{\text{a1}})}\right) \quad \text{…(B7.49)}$$

where RI_{adj} is the adjusted road roughness in IRI m/km
a0, a1 are regression coefficients (Default values for all vehicle types are: a0 = -65.8553, a1 = -1.9194)

The default values for vehicle utilisation parameters in HDM-4 are shown in Table B7.7.

Table B7.7: Proposed Default Vehicle Utilisation Model Values

Vehicle Number	Vehicle Type	AKM0 (km/yr)	LIFE0 (years)	HRWK0 (hrs/yr)
1	Motorcycle	10000	10	400
2	Small Car	23000	10	550
3	Medium Car	23000	10	550
4	Large Car	23000	10	550
5	Light Delivery Vehicle	30000	8	1300
6	Light Goods Vehicle	30000	8	1300
7	Four Wheel Drive	30000	8	1300
8	Light Truck	30000	8	1300
9	Medium Truck	40000	12	1200
10	Heavy Truck	86000	14	2050
11	Articulated Truck	86000	14	2050
12	Mini-Bus	30000	8	750
13	Light Bus	34000	8	850
14	Medium Bus	70000	7	1750
15	Heavy Bus	70000	12	1750
16	Coach	70000	12	1750

B8 Capital Costs

B8.1 Introduction

Capital costs are comprised of depreciation and interest costs. Depreciation is the loss in value of a vehicle which is not restored by repairs or maintenance. It arises principally because of three factors:

- Use;
- Time; and,
- Technical obsolescence.

Interest costs are the opportunity cost of vehicle ownership. These consist of the income that would have been received had the capital invested in the vehicle been invested elsewhere.

Together, these two costs—called the capital costs—can constitute a significant component of the total VOC, sometimes exceeding the fuel and tyre consumption costs. Capital costs, and their allocation, are sensitive to both the utilisation of a vehicle and its service life.

This chapter discusses the proposed modelling of capital costs in HDM-4. It commences with a review of the techniques applied in HDM-III. This is followed by a summary of other studies and techniques. It closes with the methodology adopted for modelling these costs in HDM-4.

B8.2 Capital Cost Modelling in HDM-III

B8.2.1 Depreciation Modelling

In HDM-III the average annual depreciation was determined based on the vehicle service life and utilisation, both of which may be either variable or constant. It was calculated as (Watanatada, *et al*., 1987b):

$$DEPFAC = \frac{1}{LIFE\,AKM} \qquad \ldots(B8.1)$$

Or,

$$DEPFAC = \frac{1}{LIFEKM} \qquad \ldots(B8.2)$$

where DEPFAC is the vehicle depreciation factor as a fraction of the new vehicle price per km
LIFE is the vehicle service life in years
AKM is the average annual utilisation in km/yr
LIFEKM is the lifetime utilisation in km

This approach is referred to as "straight-line" depreciation. The product of the vehicle depreciable value and the depreciation factor gives the depreciation cost per km.

The depreciation cost is given by the product of the replacement vehicle price, less tyres to avoid double counting, and the depreciation factor:

$$DEPCST = DEPFAC \; NVPLT \qquad \text{…(B8.3)}$$

where NVPLT is the replacement vehicle price less tyres
DEPCST is the depreciation cost per km

B8.2.2 Interest Modelling

Watanatada, *et al*. (1987b) calculated the interest charge "as the average of the residual vehicle value, decreasing in a linear fashion from full purchase price at the end of year 0 to zero at the end of year LIFE" (pg. 297). Mathematically, this is expressed as:

$$INTFAC = 0.5 \; \frac{AINV}{100} \; \frac{1}{AKM} \qquad \text{…(B8.4)}$$

where INTFAC is the interest factor as a fraction of the depreciable vehicle price per km
AINV is the annual interest rate on investments in per cent

The interest cost is the product of the interest factor and the depreciable vehicle price.

Thoreson and Roper (1994) note that Equation B8.4 "is derived from the conventional average return on capital calculation method" which uses the following equation:

Interest on Capital Invested = 0.5 (New Price + Scrap Price) * Interest Rate

B8.2.3 Effects of Operating Conditions on Capital Costs

As described in Chapter B7, the effect of operating conditions as capital costs were taken in account through their impact on the annual utilisation and service life. In HDM-III both of these were, optionally, sensitive to speed changes. A detailed review of the methods used in HDM-III was given in Chapter B7 and will not be repeated here.

B8.3 Predicting Capital Costs

B8.3.1 Introduction

There are a number of other techniques which have been used to predict the depreciation costs. They can be broadly grouped into four classes:

- Value-age;
- Capital recovery;
- Optimal life; and,
- Other techniques.

The following sections summarise these various techniques. This is followed by a discussion on how the effects of operating conditions on depreciation have been modelled as well as other application techniques.

B8.3.2 Value Age

The "value-age" (VA) method has been applied in a number of different studies. As described by Chesher and Harrison (1987), it consists of a relationship between the capital value of the vehicle and its age. Interest costs are calculated as a function of the predicted vehicle value and depreciation costs are then obtained from the change in vehicle value over time.

Chesher and Harrison (1987) point out three deficiencies with this method:

- The predicted values are not related to the quality of the routes which the vehicles have travelled over;
- The values are usually not related to the distances travelled, only the vehicle age; and,
- It fails to account for the relationship between vehicle maintenance and depreciation.

The road user cost studies in Kenya, India, the Caribbean and Brazil all developed VA relationships. Table B8.1 lists these various relationships. The passenger car predictions have been plotted in Figure B8.1. These have been converted to the depreciation (in per cent) using the following relationship:

$$DEPPCT = 100\left(1 - \frac{VEHVAL}{NVPLT}\right) \qquad \ldots(B8.5)$$

where DEPPCT is the vehicle depreciation in per cent in year t
VEHVAL is the vehicle value in year t

Table B8.1: Value-Age Relationships from Road User Cost Studies

Country	Vehicle Class[1]	Relationship	Restrictions
Kenya	PC and LGV	VEHVAL(t) = 0.78 NVPLT	t = 1
		VEHVAL(t) = (0.793 - 0.077 t) NVPLT	$2 \leq t \leq 9$
	HB, MT, HT	VEHVAL(t) = (1.317 - 0.625 $t^{1/3}$) NVPLT	$1 \leq t \leq 8$
India	PC	VEHVAL(t) = exp (-0.081 t) NVPLT	
	HB, MT, HT	VEHVAL(t) = exp (-0.147 t) NVPLT	
Caribbean	PC and LGV	VEHVAL(t) = 0.78 NVPLT	t = 1
		VEHVAL(t) = (0.795 - 0.078 t) NVPLT	$2 \leq t \leq 9$
	MT, HT	VEHVAL(t) = (1.553 - 0.662 $t^{1/3}$) NVPLT	$1 \leq t \leq 12$
Brazil	PC - Commercial	VEHVAL(t) = (0.859 - 0.143 t) NVPLT	$1 \leq t \leq 5$
		VEHVAL(t) = 0.14 NVPLT	t > 5
	PC - Private	VEHVAL(t) = 1.07 exp (-0.173 t) NVPLT	$1 \leq t \leq 12$
		VEHVAL(t) = 0.13 NVPLT	t > 12
	LGV	VEHVAL(t) = 0.75 exp (-0.124 t) NVPLT	$1 \leq t \leq 15$
		VEHVAL(t) = 0.11 NVPLT	t > 15
	HB	VEHVAL(t) = 0.95 exp (-0.169 t) NVPLT	$1 \leq t \leq 12$
		VEHVAL(t) = 0.12 NVPLT	t > 12
	MT	VEHVAL(t) = 0.83 exp (-0.175 t) NVPLT	$1 \leq t \leq 12$
		VEHVAL(t) = 0.10 NVPLT	t > 12
	HT	VEHVAL(t) = 0.84 exp (-0.160 t) NVPLT	$1 \leq t \leq 12$
		VEHVAL(t) = 0.12 NVPLT	t > 12

Source: *Chesher and Harrison (1987)*
NOTES 1/ See List of Terms for Abbreviations

The results in Figure B8.1 show that there are significant differences in the depreciation rates between countries, and even for similar vehicles in the same country but with different utilisation characteristics. The similarities between the Kenya and Caribbean equations are unusual, particularly since the equations for trucks in those countries are quite different. They should therefore be viewed as an exceptional situation.

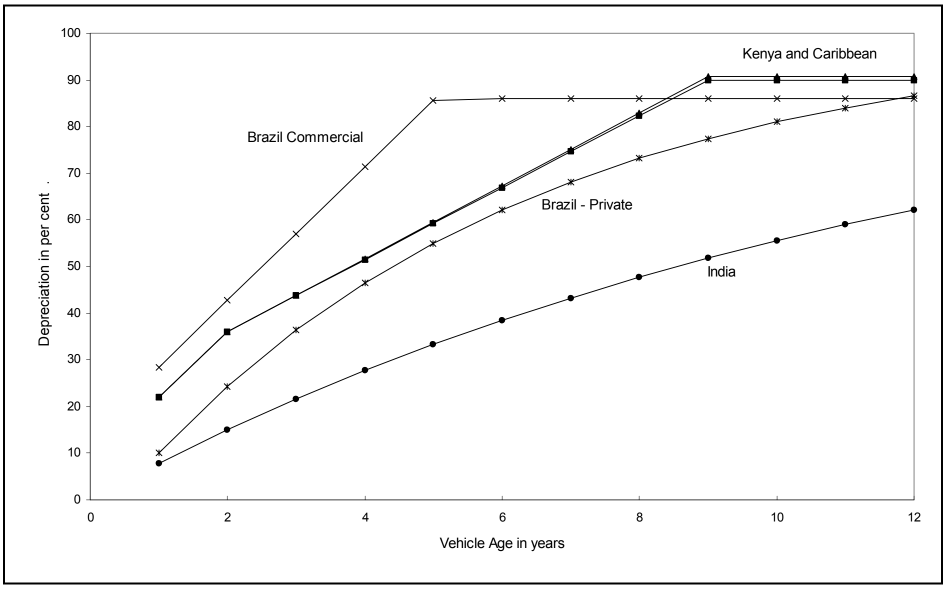

Figure B8.1: Road User Cost Studies' Passenger Car Depreciation

Chesher and Harrison (1987) show the predicted depreciation, maintenance and interest costs for vehicles using the Brazil, India and Caribbean studies. They note that while the predictions generally increase with increasing age, for Indian vehicles they decrease with age which "suggests some mis-matching of the studies' maintenance cost equations and their vehicle value-calendar age equations. The effect may be due to the exclusion from the latter of kilometre age and to highway condition effects". This led Chesher and Harrison (1987) to state:

> *"We suggest that the VA method should be used with caution and [that the] vehicle value relationships are consistent with the running costs equations that they use."*

B8.3.3 New Zealand Studies

There have been three studies conducted in N.Z. which resulted in VA equations: Ministry of Transport (1980), Bennett (1985) and Bennett (1989b). These studies developed VA equations of the following general form:

$$\text{DEPPCT} = \text{a0 AGE}^{a1}\ \text{CKM}^{a2} \qquad \ldots(\text{B8.6})$$

where	AGE	is the vehicle age in years
	CKM	is the vehicle cumulative kilometreage in km
	a0 to a2	are regression coefficients

The interest costs were calculated based on the average value of all vehicles in the fleet.

Two different techniques were used for calculating the depreciation: the "Replacement Value" and "Economic" techniques.

The replacement value technique was used in the Kenya, Caribbean and India Road User Cost studies. It defined the depreciation as the difference between the current market value of a similar vehicle and the resale value.

The economic technique converted the original sales price of the vehicle to an economic cost by deducting the sales tax applying in the year of manufacture. This was updated to the current year using the CPI. The current year's sales tax was then added to the updated price to get the current equivalent market value. The depreciation was defined as the difference between the equivalent market value and the resale price.

Table B8.2 presents the values for the model coefficients from the various studies. Figure B8.2 is a plot of the effect of age on the 1988 economic passenger car depreciation. It can be observed that the depreciation is very sensitive to age but relatively insensitive to utilisation.

Table B8.2: New Zealand Depreciation Model Values

Vehicle Class	Method	Year	Depreciation Model Coefficient a0	a1	a2
PC	Economic	1979	0.7173	0.5018	0.3108
	Economic	1987	4.1179	0.4270	0.1650
	Economic	1988	12.4586	0.2637	0.1079
	Replacement	1988	12.0000	0.2751	0.1108
LGV	Economic	1988	39.4946	0.2843	0.0000
	Replacement	1988	31.0542	0.4105	0.0000
MT, HT	Economic	1985	0.0973	0.4137	0.0955
	Economic	1988	11.1900	0.4625	0.0709

Source: *Bennett (1989b)*

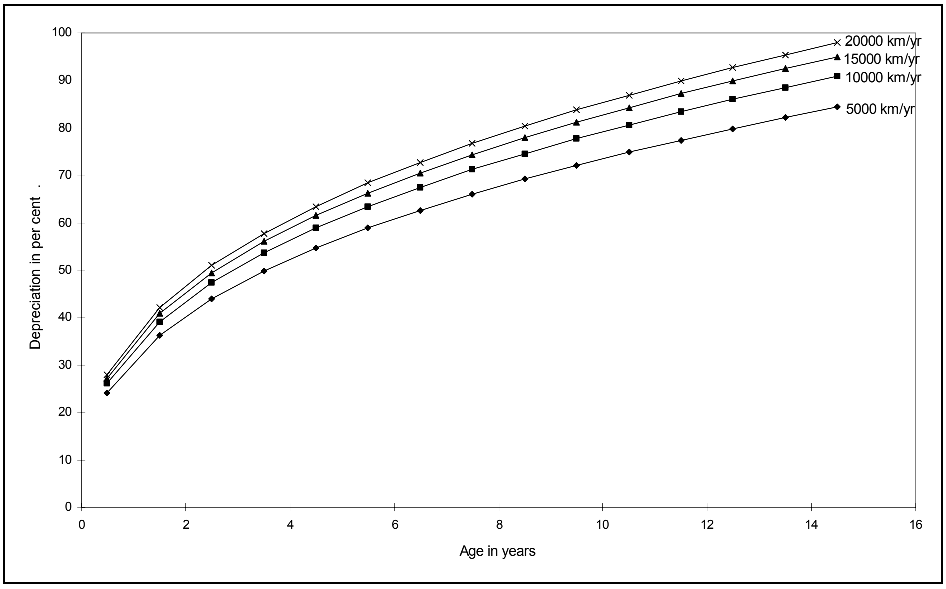

Figure B8.2: Effect of Age on 1988 Economic New Zealand Passenger Car Depreciation

Bennett (1989b) compared the predictions from the two depreciation techniques. It was concluded that the two techniques give very similar results so either could be used for analysing resale data. However, it was recommended that the economic technique be used because:

- It is based on the changing value of a commodity over time which is important in highway evaluations;
- It uses the original sales prices of vehicles in the calculations which is both easier to obtain, and arguably more meaningful, than the estimates of similar current vehicles that are used with the replacement value technique.

Since the depreciation relationships contain a time and use component it is possible to investigate the relative contribution of time and use to depreciation. Bennett (1989b) calculated the depreciation for two scenarios: assuming that there was no use and assuming that all the use happened instantaneously. The difference between these two scenarios gave an indication of the depreciation between time and distance. The results of this analysis are given in Table B8.3.

Table B8.3: Allocation of New Zealand Depreciation Between Time and Use

Vehicle Class[1]	Year	Technique	Percentage of Depreciation Due to	
			Time	Use
PC	1979	Economic	60	40
PC	1987	Economic	73	27
PC	1988	Economic	72	28
PC	1988	Replacement	72	28
HT	1984	Economic	80	20
HT	1988	Economic	87	13

Source: *Bennett (1989b)*
NOTES: 1/ See List of Terms for abbreviations

In spite of the differences in the predicted depreciation rates between 1987 and 1988 for passenger cars, the allocation of depreciation was consistent. However, it was significantly different to the passenger car allocation from the study nine years earlier. The significant differences in the coefficients and predictions of the depreciation models for different years in Table B8.3 indicated that the depreciation rate was not stable over time. It was concluded by Bennett (1989b) that there had been a decreasing sensitivity of the depreciation rate to use between the various studies, *ie* the depreciation was becoming more influenced by time. The depreciation rates were markedly different for data collected only 17 months apart due to a liberalisation of N.Z. import restrictions on used vehicles. This led to the conclusion that (Bennett, 1989b):

> *"the usefulness of developing detailed relationships for predicting depreciation is questionable if they are only pertinent for a short period of time".*

B8.3.4 RTIM2

The RTIM2 model (Parsley and Robinson, 1982) used a VA method; however, it also included the vehicle fleet age spectrum in the calculations. This age spectrum could remain constant throughout the analysis period or be modified by wastage and ageing mechanisms to reflect the natural scrapping due to accidents, technical obsolescence and the introduction of new vehicles. The program worked by establishing the number of surviving vehicles at the end of each year and then using this to determine the depreciation costs. The depreciation was calculated for passenger cars as:

$$\text{DEPPCT} = d0v_1^y + d1v_2^y + \sum_{i=3}^{8} d2v_i^y + d3v_9^y + d4\Delta v_1^y + \sum_{i=2}^{8}(d5 - d6(i-2))\Delta v_i^y \qquad \ldots(\text{B8.7})$$

where v_i^y is the number of vehicles of age i in analysis year y
d0 to d6 are the depreciation rates for year vehicles of age i

B8.3.5 RTIM3

The program RTIM3 (Cundill, 1993) used the following equations for calculating depreciation and interest costs:

$$\text{DEPCST} = \frac{\text{AD PN NVPLT}\,(100 - \text{PCTPRV})}{10^9}\,\frac{\text{TTKKM}}{\text{HRD}} \qquad \ldots(\text{B8.8})$$

$$\text{INTCST} = \frac{100 \text{ ir PN NVPLT } (100 - \text{PCTPRV})}{10^9} \frac{\text{TTKKM}}{\text{HRD}} \qquad \ldots(\text{B8.9})$$

where	INTCST	is the interest cost per km
	AD	is the annual depreciation as a percentage of the current vehicle price
	PN	is the current vehicle value as a percentage of the price of a new vehicle
	PCTPRV	is the percentage of private vehicles
	TTKKM	is the travel time in h/1000 km
	HRD	is the number of hours the vehicle is driven per year
	ir	is the interest rate as a decimal

There are a number of points to be noted with the above approach:

- The user defines the average depreciation (AD) as an input value. It is the average decrease in the **financial** cost of the representative vehicle, "typically in the region of 10 per cent";
- The costs are not calculated for vehicles engaged in private use;
- The costs are based on the average value of the vehicle fleet, expressed as the product of PN NVPLT; and,
- The costs are proportional to speed, but the effects are a function of the number of hours per year the vehicle is utilised.

Overhead costs were calculated in RTIM3 by dividing the total costs by the number of hours per year the vehicle is used. The predicted speed was then used to incorporate the effects of operating conditions on these costs, *ie*:

$$\text{OCKM} = \frac{\text{OA } (100 - \text{PCTPRV})}{100 \text{ HRD S}} \qquad \ldots(\text{B8.10})$$

where	OCKM	is the overhead cost per km
	OA	is the total annual overhead cost

It will be noted that in the above equation "private use" costs are excluded.

B8.3.6 Capital Recovery

The capital recovery technique (CRT) was proposed by Schutte (1979 and 1981). It was later modified by Pienaar (1984) to allow for a direct calculation of the per kilometre depreciation costs. It was applied in Indonesia (Halcrow Fox, 1982) and by Bennett (1989b) in N.Z. after finding that the VA equations were not stable over time.

As with the straight-line depreciation technique, the total depreciation costs are defined as the difference between the cost of the new vehicle, less tyres, and the residual value of the vehicle. However, the CRT adjusts both the new and residual values for the effects of time on capital by converting them to an annual cost. The CRT is illustrated by the cash-flow diagram in Figure B8.3.

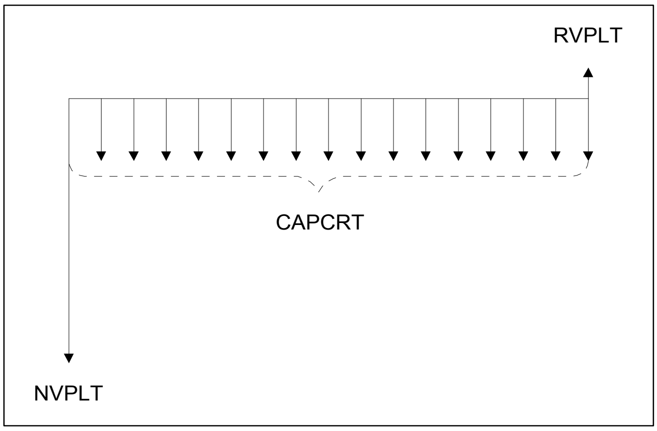

Figure B8.3: Capital Recovery Technique Cash Flow Diagram

The capital costs (*ie* depreciation plus interest) are calculated as:

$$\text{CAPCRT} = \text{NVPLT}\frac{\text{ir}\,(1+\text{ir})^{\text{LIFE}}}{(1+\text{ir})^{\text{LIFE}}-1} - \text{RVPLT}\frac{\text{ir}}{(1+\text{ir})^{\text{LIFE}}-1} \qquad \ldots\text{(B8.11)}$$

where CAPCRT is the annual depreciation and interest costs in cost/veh/yr
RVPLT is the residual vehicle price less tyres

The depreciation may be expressed as either an annual or a monthly cost. To convert the annual interest rate to a monthly rate the following equation is used:

$$\text{irm} = \sqrt[12]{(1+\text{ir})} - 1 \qquad \ldots\text{(B8.12)}$$

where irm is the monthly interest rate as a decimal

Schutte (1994) notes that:

> *"[since] a capital item, by definition, has a long economic life during which it is capable of producing a specified output, the time value of money should be considered in calculating capital cost. The price of the new vehicle (net of indirect taxes and the price of tyres) could be considered as the (discounted) present worth of the resource cost incidental to each year of its economic life (or each output unit produced during its economic life). When the capital item is abandoned at the end of its economic life, it will however have an opportunity value; this value will result from the fact that it could then be used for other purposes, for example, it could be sold for scrap metal. Therefore, the objective is to allocate the initial price of the vehicle (minus the present worth of its residual value) to each year of its economic life".*

B8.3.7 Optimal Life

As described in Chapter 7, the "optimal life" (OL) method was proposed by Nash (1976) and was refined by Chesher, *et al*. (1980); Chesher and Harrison (1987); and Chesher (1990). Hine (1982) used a variation of the OL method.

In all applications the theory is that cost minimisation arises when a vehicle is scrapped at its optimal life. Using this assumption in conjunction with the VOC allows one to calculate this optimal life under different conditions. This value can then be used in conjunction with a straight-line depreciation technique (*eg* Equation B8.2) to estimate the costs.

B8.3.8 Australian Studies

Thoreson and Roper (1994) indicate that the Australian NIMIPAC model calculates depreciation and interest costs using the following equation:

$$CAPKM = DISTDEP + \frac{TDPINT}{S} \qquad \ldots(B8.13)$$

where DISTDEP is the distance based depreciation adjusted for surface type in cost/km

TDPINT is the marginal time depreciation and interest cost in cost/h

The distance based depreciation is calculated from the replacement vehicle price, less tyres, and a depreciation rate in per cent/1000 km. The time based cost is calculated as:

$$TDPINT = \frac{ir NVPLT FLEET}{HAV} \qquad \ldots(B8.14)$$

where FLEET is the proportion of vehicles susceptible to fleet reduction effects[1]

Cox (1995) did further work on depreciation. He suggested that the average depreciation for the fleet should be based on the point in time when the travel of all younger vehicles is equal to that of all older vehicles. This was termed the “on the road” vehicle age and was found to range from six years for articulated trucks to 8.5 years for passenger cars. It was noted that the HDM-III approach gave minor cost differences when compared with values calculated using the on the road vehicle age.

For calculating capital costs it was assumed that vehicles were scrapped when they reached 90 per cent of their total cumulative travel and that the scrap value was five per cent of the purchase price (Cox, 1995). It was noted that the service life for passenger cars has risen from 200,000 km in 1971 to approximately 300,000 km in 1991 due to improvements in vehicle technology.

B8.3.9 Indonesia

Halcrow Fox (1982) used the capital recovery technique for calculating the total depreciation costs. The time based component of depreciation was calculated from an assumed maximum life with “minimal or no use”. For passenger cars, this was taken to be 33 years. The time based depreciation was corrected for the residual value, which resulted in the following equation:

$$TIMEDEP = \frac{1}{MAXVLIFE} \; \frac{RVPLT}{NVPLT\ LIFE} \qquad \ldots(B8.15)$$

where MAXVLIFE is the estimated maximum vehicle life with minimal or no use

TIMEDEP is the time based depreciation cost as fraction of the replacement vehicle price per year

1 Based on the description of Halcrow Fox (1982), the fleet susceptibility factor is analogous to the EVU.

Using the capital recovery technique, the total annual depreciation cost was calculated and expressed as a fraction of the replacement vehicle price, *ie*:

$$\text{DEPANFAC} = \frac{\text{DEPCRT}}{\text{NVPLT}} \qquad \text{…(B8.16)}$$

where DEPANFAC is the annual depreciation factor as a fraction of the new vehicle price per year

The annual interest factor was taken as the difference between the annual depreciation factor and the depreciation:

$$\text{INTANFAC} = \frac{\text{DEPCRT}}{\text{NVPLT}} - \frac{\text{NVPLT} - \text{RVPLT}}{\text{NVPLT LIFE}} \qquad \text{…(B8.17)}$$

$$\text{INTANFAC} = \frac{\text{DEPCRT LIFE} - \text{NVPLT} + \text{RVPLT}}{\text{NVPLT LIFE}} \qquad \text{…(B8.18)}$$

where INTANFAC is the annual interest factor as a fraction of the new vehicle price per year

The annual factors were converted to hourly factors by dividing them by the number of hours the vehicle was “operated”. This was considered to include “time spent in motion on the road, cargo handling time and time spent waiting for cargo or for passengers to board” (Halcrow Fox, 1982). This is therefore analogous to the time the vehicle is available for use (HAV).

B8.3.10 Sweden

Hammarström and Karlsson (1987) divided the depreciation costs into time and use components. The actual costs were estimated using a sophisticated method partly based on the probability of vehicle survival. The depreciation costs were adjusted for operating conditions via speeds.

B8.3.11 U.S.A.

Winfrey (1969) presented a number of different approaches for calculating depreciation. Including the “Sinking Fund”, “Present-Worth”, “Declining-Balance” and “Sum-Of-The-Years-Digits” methods.

Depreciation costs calculated by Winfrey (1969) continued to be used in the U.S.A. for some time. AASHTO (1977) based its depreciation costs on a method proposed by Winfrey (1969), where the costs were modified to take into account vehicle speed. Zaniewski, *et al*. (1982) used a survival curve method proposed by Daniels (1974) to modify Winfrey’s costs as a function of service life.

Claffey (1971) investigated various factors influencing the depreciation rate of vehicles. He studied 64 cars recently removed from the road because of progressive deterioration and obtained their lifetime kilometreage. A relationship between speed and utilisation was postulated along with one between annual utilisation and service life. It was concluded that highway improvements resulted in additional utilisation with little reduction in service life, thus decreasing the per kilometre depreciation cost.

B8.4 Factors Influencing Capital Costs

B8.4.1 Introduction

It has long been recognised that operating conditions have an influence on capital costs. While the OL technique can account for these through the variation of running costs under different conditions, others used different approaches for accounting for such effects, principally through the effect of operating conditions on utilisation and service life.

Another factor besides operating conditions influencing the costs is their allocation. Since studies have found that there are time and distance based components of depreciation.

B8.4.2 Utilisation and Service Life

The effects of utilisation and service life on costs is normally considered by adjusting the utilisation and life characteristics as a function of the operating conditions.

In Chapter 7 the HDM-4 service life and utilisation models were described. For HDM-4 service life is modelled as a function of roughness based on a variation of the optimal life method proposed by Chesher and Harrison (1987). Lifetime utilisation is modelled as a function of the roughness based on the outcomes from applying the optimal life (OL) method. Alternatively one can assume a constant utilisation.

B8.4.3 Allocation of Depreciation Components

Depreciation has both time and use related components. Ideally, the time related component should be treated as a standing cost while the distance related component is a running cost. Only the distance related component will be influenced by road investments, although if there are speed increases and additional utilisation the allocation of the time related component will be affected.

Halcrow Fox (1982) note that there is a relationship between the allocation of depreciation and utilisation. Highly utilised vehicles will have short service lives and thus, higher distance based depreciation than vehicles with low utilisation levels.

In most instances the depreciation has been allocated between time and use by assuming a percentage split. Some values reported in the literature are given in Table B8.4.

Daniels (1974) presented a technique for determining the time and use related components of depreciation. This technique was later applied by Butler (1984) to the U.S.A. It was based on the assumption that depreciation can be expressed as a function of a constant time component and a distance component which is linearly dependent on utilisation.

Table B8.4: Reported Allocations of Time and Distance Depreciation Components

Country	Source	Percentage Allocation of Depreciation	
		Time	Distance
Australia	Abelson (1986)	60	40
Denmark	Macdonald (1987)	0	100
Germany	Macdonald (1987)	50	50
N.Z.	Bennett (1989b)[1]	70	30
N.Z.	Bennett (1989b)[2]	85	15
U.K.	Macdonald (1987)	60	40
U.S.A.	AASHTO (1977)[3]	Varies	Varies

NOTES: 1/ Passenger cars
2/ Commercial vehicles and buses
3/ Allocation is a function of utilisation and speed

The N.Z. values for depreciation (Bennett, 1989b) are based on the predictions of a model that gave depreciation as a function of time and distance travelled. It was noted that there had been a change in the allocation over time with depreciation in 1988 being due more to time than it was in 1980.

B8.4.4 Effect of Trip Length

Schutte (1994) argues that in dealing with changed route lengths it is necessary to account for the fact that the annual utilisation will only be partially influenced by the route length. For example, if the lifetime utilisation was 200,000 km and the annual utilisation 20,000 km/yr, this implies an effective life of 10 years. If the entire network length was reduced by 10 per cent, there would be a 10 per cent decrease in AKM and the effective life would increase to 11.1 years. However, instead of the entire network being shortened road building projects typically only shorten a single link. Regarding vehicles using this link there are two cases:

Exclusive Use

The vehicles using this link (*ie* its ADT) exclusively operate on the link, and nowhere else in the network.

Partial Use

Some vehicles using this link also use other links on the network so its ADT consists of:

- The portion of ADT making exclusive use of the link; and,
- The portion of ADT that uses other link in the network.

Schutte (1994) therefore used various values for AKM to predict the effective life and thus the depreciation costs. As would be expected, the predicted costs were very sensitive to the split between exclusive and partial use. It was concluded that “savings from distance reductions will be substantially lower” than traditionally thought since little traffic makes exclusive use of a link.

The same argument was also applied to roughness improvements since the improvement of road surface condition will only affect one link of the network and not the entire network. As with distance reductions, roughness reductions were found to give significantly lower benefits than with traditional analyses, although not to the same degree as with distance reductions.

B8.5 Modelling Capital Costs in HDM-4

Bennett (1996c) describes the development of the final models for capital cost modelling in HDM-4 using data from Thailand.

B8.5.1 Residual Value

It can be assumed that the residual value of the vehicle (RVPLT) is proportional to roughness: vehicles operated on rougher roads will have a lower residual value since they will have suffered more wear and tear. Bennett (1996c) assumed that the residual values for all vehicles were 15 per cent at a roughness of 5 IRI m/km; five per cent at a roughness of 15 IRI m/km, and a minimum of two per cent. Using a linear relationship was expressed as:

$$RVPLTPCT = \max[2, 15 - \max(0, (RI - 5))] \quad \ldots(B8.19)$$

where RVPLTPCT is the residual vehicle price in per cent.

For implementation into HDM-4 the following generic form of Equation B8.19 was used:

$$RVPLTPCT = \max[a0, a1 - \max(0, (RI - a2))] \quad \ldots(B8.20)$$

where a0 is the minimum residual value of the vehicle in per cent (default = 2)

a1 is the maximum residual value of the vehicle in per cent (default value = 15)

a2 is the average roughness IRI below which the maximum value arises (default value = 5)

B8.5.2 Depreciation

The depreciation is calculated using the equation:

$$DEP = 1000 \, \frac{(1 - 0.01\, RVPLTPCT)}{LIFEKM} \quad \ldots(B8.21)$$

where DEP is the depreciation cost as a fraction of the replacement vehicle price, less tyres

The 0.01 in Equation B8.21 converts the residual value from a percentage to a fraction. The denominator of Equation B8.21 represents the lifetime utilisation. This is calculated in one of the following ways, depending upon the service life method.

OPTIMAL LIFE SERVICE LIFE METHOD

The lifetime utilisation is established directly from the optimal life method and used in the calculations.

CONSTANT SERVICE LIFE METHOD

As described in the HDM-4 Technical Reference Manual, a somewhat arbitrary approach was adopted, depending upon the percentage of private use of the vehicle. The following are the equations adopted:

$$LIFEKM = AKM0\ LIFE0 \qquad \text{Private Use} > 50\% \quad \ldots(B8.22)$$

$$LIFEKM = S\ HRWK0\ LIFE0 \qquad \text{Private Use} \leq 50\% \quad \ldots(B8.23)$$

The rationale behind this delineation is not clear since it will likely lead to a discontinuity in the area where private use approaches 50 per cent. It may also lead to excessive depreciation costs when S is much lower than the average speed used for establishing AKM0.

The calculation of the depreciation cost is done using the equation:

$$DEPCST = DEP\ NVPLT \quad \ldots(B8.24)$$

$$NVPLT = NVP - NUM\_WHEELS\ NTP \quad \ldots(B8.25)$$

where DEPCST is the depreciation cost in cost/1000 km

NVPLT is the replacement vehicle price, less tyres

NVP is the replacement vehicle price

NUM_WHEELS is the number of wheels on the vehicle

NTP is the new tyre price

An example of the predictions using the optimal life method is given in Figure B8.4. This was calculated using data for nine vehicle classes with unit cost data from Thailand and the function of lifetime utilisation versus roughness given earlier in Figure B7.10.

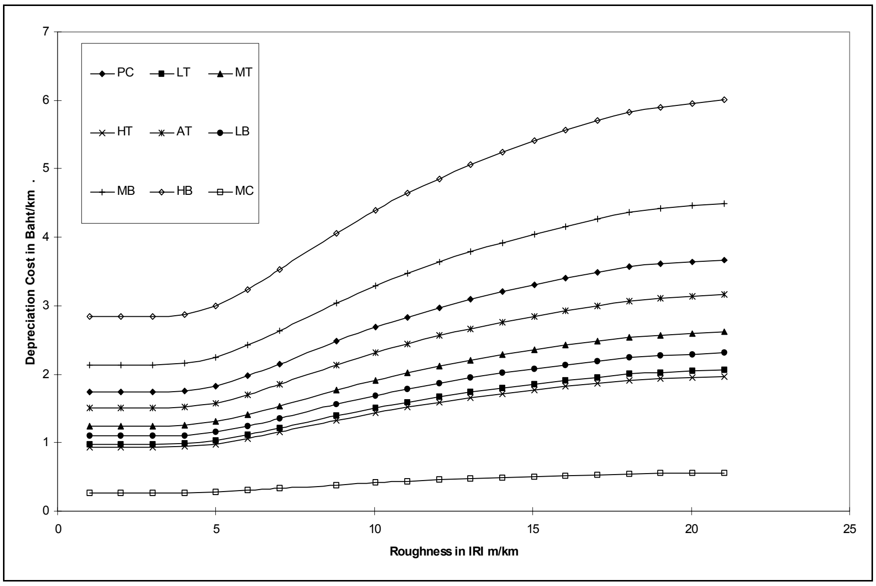

Figure B8.4: Example of the Effect of Roughness on Depreciation Costs

B8.5.3 Interest

The interest cost factor is calculated in a similar manner to HDM-III, however, instead of the denominator being the annual utilisation AKM, it is now the product of the speed and hours worked:

$$INT = 0.5\ \frac{AINV}{100}\ \frac{1000}{S\ HRWK0} \qquad \ldots(B8.26)$$

where INT is the interest cost as a fraction of the replacement vehicle price

Given the approach adopted for depreciation (Equations B8.22 and B8.23), it would have been more consistent to have the denominator AKM0 when the percentage of private use was more than 50 per cent.

The interest costs are calculated as:

$$INTCST = INT\ NVP \qquad \ldots(B8.27)$$

B8.5.4 Total Capital Costs

The total capital costs are given by the sum of the depreciation and interest costs, *ie*:

$$CAPCST = DEPCST + INTCST \qquad \ldots(B8.28)$$

where CAPCST is the total capital cost in cost/1000 km

B9 Engine Oil Consumption

B9.1 Introduction

Engine oil consumption constitutes a very small component of the total running costs and has therefore not been researched to the same detail as other RUE components. Because of this, NDLI (1995) proposed to exclude oil consumption from HDM-4 but feedback from reviewers indicated that for completeness it should be included.

This chapter presents the oil consumption model adopted for HDM-4. It commences with a brief summary of the research into oil consumption. This is then followed by a description of the HDM-4 model and examples of its predictions.

B9.2 Predicting Oil Consumption

Oil consumption arises through two factors:

- Oil contamination; and,
- Oil loss.

Contamination may occur because of impurities resulting from the combustion process. It may also be due to external sources, for example dust on unsealed roads. It is because of contamination that vehicles change their oil, usually after travelling a set distance as recommended by the manufacturer.

Oil losses arise through faulty seals or by the combustion process wherein oil may leak past the pistons and be burned with the fuel.

Claffey (1971) investigated engine oil consumption and noted:

> *"The magnitude of the oil consumed to replace contaminated oil depends in part on whether operation is at high sustained speed with engines properly warmed up (long trips), so that sludge is burned away, or at low speed on short trips with frequent stops when unburned combustion remnants are able to accumulate. Oil replacement because of contamination is also accelerated by operation on dusty roads when some dust particles get by filters to form an abrasive oil contaminant."*

It was found by Claffey (1971) that speed had an impact on oil consumption as did the frequency of stop-go cycles. Oil consumption rates were proposed for urban and free-flow traffic, as well as the marginal consumption due to stop-go cycles.

For HDM-III Watanatada, *et al.* (1987a) proposed the following model for predicting oil consumption:

$$OC = a0 + a1\ RI \qquad \text{…(B9.1)}$$

where OC is the oil consumption in L/1000 km
a0 and a1 are coefficients

The default coefficients for this model are given in Table B9.1. Since these could not be changed in HDM-III, they were applied in all analyses.

Table B9.1: HDM-III Oil Model Coefficients

Vehicle Class	a0	a1
Passenger Cars and Utilities	1.55	0.15
Light Trucks	2.20	0.15
Medium and Heavy Trucks	3.07	0.15
Articulated Trucks	5.15	0.15
Heavy Buses	3.07	0.15

Source: Watantada, et al. (1987a)

The India study (CRRI, 1982) is notable since it endeavoured to relate oil consumption to operating conditions. A series of equations were developed which related oil consumption to various road geometry parameters such as gradient, width and roughness. The study found that there were significant operator effects—*ie* the rate of oil consumption was highly dependent on the policies of the companies participating in the study. For example, when the individual operators were included through the use of dummy variables in the equation the R^2 was 0.75 compared to R^2 below 0.2 with pooled data. Not surprisingly, the vehicle operator effects mainly influenced the intercept which represents the oil lost due to contamination.

Kadiyali (1991) in updating the CRRI (1982) costs adopted Equation B9.2 which was from the original CRRI (1982) work. Table B9.2 gives the parameters for use with this model. This equation was adopted by IRC (1993) for economic appraisals in India.

$$OC = a0 + a1\ RF + a2\ \frac{RI}{W} \qquad \ldots(B9.2)$$

where RF is the rise and fall in m/km
W is the pavement width in m

Table B9.2: Indian Oil Consumption Model Parameters

Vehicle Class	a0	a1	a2
Motorcycles	0.39	0.00750	0.08
Passenger Cars	1.94	0.03769	0.43
Light Truck	1.00	0.02400	0.11
Medium and Heavy Trucks	2.48	0.06010	0.29
Heavy Buses	3.66	0.01271	0.48

Source: Kadiyali, et al. (1991)

CRRI (1982) also collected data on other oils needed for lubrication including gear, differential, brake and steering oils and used these to develop models. Data were also collected on the consumption of grease. Models were developed from the data and are in IRC (1993), but the quantity of these lubricants is so small—indeed, they were measured in terms of consumption per 10,000 km instead of the 1,000 km for engine oil—that they were not considered for HDM-4.

Schutte (1981) proposed a speed based oil model for South Africa based on the work of Claffey (1971). This model was:

$$OC = a0 + a1\ S \qquad \ldots(B9.3)$$

where S is the vehicle speed in km/h

Pienaar (1984) investigated oil costs by disaggregating the total costs into those due to contamination and those due to losses. The contamination costs were calculated by establishing the mean distance between oil changes recommended by various manufacturers and weighting these figures by sales data. This was used in conjunction with the average engine oil capacity to calculate the rate of oil contamination.

To investigate the effects of operating conditions on oil consumption experiments were conducted with a fleet of ten vehicles (Pienaar, 1984). It was found that oil loss was a function of engine speed and that it was proportional to the fuel consumption. Table B9.3 gives the values of the oil loss as a fraction of the fuel consumption for the different vehicle classes. These indicate that diesel vehicles have lower losses than petrol vehicles and Pienaar (1984) suggested that this is due to two factors, namely:

- At similar speeds a diesel engine operates at a lower engine speed than a petrol engine; and,
- In comparable situations a petrol engine heats up more than a diesel engine.

Table B9.3: Pienaar (1984) Oil Model Coefficients

Vehicle Class	Distance Until Change (km)	Engine Oil Capacity (L)	Rate of Oil Consumption (L/1000 km)	Rate of Oil Loss (L/L)
Passenger Cars	9290	4.1	0.44	0.0028
Light Delivery Vehicles	7300	4.9	0.67	0.0028
Medium Trucks	9000	13.6	1.73	0.0021
Heavy Trucks	10000	30.6	3.06	0.0021
Heavy Buses	8000	19.6	2.43	0.0021

Source: Pienaar (1984)

B9.3 HDM-4 Oil Consumption Model

The HDM-4 oil consumption model was based on the work of Pienaar (1984) which related oil consumption to operation and contamination. The operational losses are a function of fuel consumption and the contamination are a function of the distance between oil changes and the oil capacity.

The following is the model adopted for HDM-4:

$$OIL = OILCONT + OILOPER\ SFC \qquad \ldots(B9.4)$$

where OIL is the oil consumption in L/1000 km
OILCONT is the oil loss due to contamination in L/1000 km
OILOPER is the oil loss due to operation in L/1000 km
SFC is the fuel consumption in L/1000 km

The loss due to contamination is determined as follows:

$$OILCONT = \frac{OILCAP}{DISTCHNG} \qquad \ldots(B9.5)$$

where OILCAP is the engine oil capacity in L
DISTCHNG is the distance between oil changes in 1000 km

Based on data from Pienaar (1984), the values in Table B9.4 were adopted as the defaults for HDM-4.

Table B9.4: Default HDM-4 Oil Consumption Model Parameters

Representative Vehicle	Description	Distance Between Oil Changes (km)	Engine Oil Capacity (L)	Rate of Oil Contamination OILCONT (L/1000 km)	Oil Loss Due to Operation OILPER (L/L)
1	Motorcycle	5000	2.0	0.40	0.0014
2	Small Car	10000	4.0	0.40	0.0028
3	Medium Car	10000	4.0	0.40	0.0028
4	Large Car	10000	4.0	0.40	0.0028
5	Light Delivery Vehicle	7500	5.0	0.67	0.0028
6	Light Goods Vehicle	7500	5.0	0.67	0.0028
7	Four Wheel Drive	7500	5.0	0.67	0.0028
8	Light Truck	9000	14.0	1.56	0.0021
9	Medium Truck	9000	14.0	1.56	0.0021
10	Heavy Truck	10000	31.0	3.10	0.0021
11	Articulated Truck	10000	31.0	3.10	0.0021
12	Mini Bus	7500	5.0	0.67	0.0021
13	Light Bus	8000	14.0	1.75	0.0028
14	Medium Bus	8000	14.0	1.75	0.0021
15	Heavy Bus	8000	20.0	2.50	0.0021
16	Coach	8000	20.0	2.50	0.0021

Source: Bennett (1996a)

The HDM-4 oil model is an improvement over HDM-III in that it can be readily altered to reflect local maintenance practices. By establishing the typical distances between oil changes as well as the average oil capacity the parameter OILCONT can be readily quantified. The oil loss due to operation component also allows for the impact of factors such as roughness and speed changes on oil consumption.

B10 Travel Time, Crew and Overhead Costs

B10.1 Introduction

Travel time and crew costs are important in many economic appraisals of road projects. The amount of time used, usually in terms of h/1000 km, is calculated from the predicted speed in km/h (see Chapters B2 and B3 for a discussion of how speeds are predicted). This is multiplied by the unit cost of time to establish the cost.

Establishing a value of time is probably the singularly most difficult cost to quantify in an economic appraisal. There have been a number of reports dedicated to this issue. Symonds (1997) gave a review of the more recent work with Cox (1983) giving a good historical view. Symonds (1997) also gave a good discussion of the issue of non-linearity of travel time savings[1].

While some studies have simply ignored passenger time, this is not wise as it can bias the results in favour of projects with high transport costs. Conversely, an unrealistically high value for passenger time can bias projects in favour of those with large speed increases.

This chapter opens with a discussion of how one quantifies the value of travel time. It then presents the HDM-4 travel time and crew cost model. This is followed by the overhead cost model.

B10.2 Quantifying a Value for Passenger and Crew Time

In quantifying a value for passenger and crew time it should be recognised that there are two sectors of the economy: the **formal** and the **informal** sectors. The formal sector is comprised of wage earners. The informal sector is those without salaries - such as many of the rural population in developing countries.

For simplicity, travel is usually separated into **work** and **leisure** or **non-work** travel. Work travel is comprised of those on business activities. Leisure, comprised of other activities (for example, social, visiting family, personal business, going to school, *etc.*) is often a significant component of travel in developing countries.

Savings in time when journeys are related to work clearly have a value; if less time is spent travelling more time in the working day can be used for economically productive purposes. Another way of looking at this is the employer pays the employee an hour’s wages for no return. The employer would be willing to pay equal to an hour’s wages to reduce travel time by one hour. It can be argued that due to overheads and social charges the employer would be prepared to pay even more, but the common practice in developing countries is to equate the value of work time to the earnings rate of the traveller. In developed countries, where there are often large social costs, this gross wage is increased by the employer’s on-costs.

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1 This relates to the question of to the appropriateness of treating a large number of small time savings as equal to a single large time saving of the same magnitude. There are several arguments in favour of this, but the most common is that since small time savings cannot be perceived, they should not be valued lower than large time savings - hence the non-linearity of the value of time savings. Symonds (1997) reviews the issue and concludes *“... there are no compelling theoretical or empirical reasons to adopt non-linear assumptions on values of time savings, and there are strong practical reasons in favour of a constant [value]”*. They note that the exception to this is leisure time where there are different levels of disutility but that for practical reasons linear values should be used.

The use of wage rates is complicated by the fact that official statistics on wages will probably underestimate the earnings of travellers. Wage statistics do not usually cover the earnings of the highest paid workers, and the wages of those travelling during working time may be higher than the average. Often, there are also regional variations in wages that make it impractical to adopt a national average. For example, in Thailand the 1995 average wage was Baht 9900/month in Bangkok versus 4500 in the north-eastern region of the country.

Those in the informal sector or travelling in leisure time are not considered to be productive in the same way as those travelling in work time. Ultimately, the value of non-working time should reflect Government policy. If the policy is to maximise GDP, ignoring leisure time preferences and increasing the welfare of passengers, then a zero value should be placed on non-work time. It must be recognised that assigning a zero value to the time for those in the informal sector will serve to bias the results in favour of those who contribute to the modern, cash economy.

There is evidence that the leisure time savings are valued, particularly since these travellers still prefer their trips to be faster than slower and are often willing to pay more for this to happen. How much a person is prepared to pay for a quicker trip is based upon their income and wealth. It is therefore common practice to assume a value of personal time related to the individual's income. Various percentages have been assumed in different studies, usually in the range 20-50 per cent, but 20-25 per cent seems to be the most common.

Many who travel in personal time do not earn any income and so using this approach would have no value for their time. In affluent societies this would not be true, but it is argued by some that in some countries a zero value of time is appropriate.

In these instances the mean income is commonly used to calculate the value of time. The alternative approach is to calculate the value of time based only on those working and then to apply the value to all travellers. This will yield a higher value of time than using the mean income of travellers.

To summarise, there are three sets of passenger time values to be considered:

- Employed, travelling in work time;
- Employed, not travelling in work time; and,
- Unemployed or in non-paid activities.

There is evidence that travel time values are higher for traffic travelling under congested as opposed to free-flow conditions. MVA, *et al.* (1987) suggested a factor of 1.4 for the situations they studied, and an even higher factor under more congested conditions.

When establishing the value of crew time, particularly for truck and bus operators, it is important to include extra income that may be obtained above the base salary, for example:

- Daily allowances to cover food and rest;
- Carrying passengers (trucks) or extra, non-reported passengers (buses); and,
- Backhaul of goods by trucks where the operator instead of the owner keeps the income.

These can be significant, for example in India NDLI (1997) estimated that the total salaries for the truck driver and helper were Rs 4500 but that there was a further monthly income of Rs 3000 from collecting passengers and Rs 3000 from daily allowances. About 20-25 per cent of drivers managed to backhaul goods, earning a further Rs 7000 per month.

It is also common to differentiate between modes of travel. This, for example, sees different passenger time costs for passengers in private transport and those in buses or other public transport.

Table B10.1 gives an example of calculating passenger time and crew costs, using basic data from India (NDLI, 1997) along with some assumed additional values. The notes at the bottom of the table detail how the individual values were calculated. This method gives an average value for all vehicles which is how HDM-III operated; in HDM-4 it is possible to differentiate between working and non-working time explicitly.

Table B10.1: Example of Calculating Passenger Time and Crew Costs

		Two-wheeler	**Car**	**Medium truck**	**Heavy truck**	**Heavy bus**
(1)	Income - Work Trips (Rs/h)	15	32	-	-	43
(2)	Income - Non-Work Trips (Rs/h)	16	47	-	-	31
(3)	Time Value - Non-Work Trips (Rs/h)	3.2	9.4	-	-	6.2
(4)	Time Value - Not Employed (Rs/h)	0.6	1.8	-	-	1.2
(5)	Persons per vehicle	1.5	3.5	2.0	2.0	32.0
(6)	Passengers per vehicle	1.5	3.3	-	-	30.0
(7)	Percentage in employment	75	75	-	-	50
(8)	Number employed	1.1	2.5	-	-	15.0
(9)	Number not employed	0.4	0.8	-	-	15.0
(10)	Percentage travelling in work time	36	19	-	-	34
(11)	Number travelling in work time	0.5	0.6	-	-	10.2
(12)	Number employed but not on work trip	0.6	1.9	-	-	4.8
(13)	**Passenger Time cost Rs/h**	9.7	38.5	0.0	0.0	486.4
(14)	Crew per vehicle	-	0.2	2.0	2.0	2.0
(15)	Cost of Driver (Rs/month)	-	600	3,000	3,000	3,000
(16)	Cost of Helper(Rs/month)	-	0	1,500	1,500	1,500
(17)	Additional Income—Allowances (Rs/month)	-	0	3,000	3,000	3,000
(18)	Additional Income—Phantom Passengers (Rs/month)	-	0	3,000	3,000	3,000
(19)	Additional Income—Phantom Backhauls (Rs/month)	-	0	1,500	1,500	0
(20)	Monthly Crew Time (Rs/month)	-	600	13,000	13,000	10,500
(21)	**Crew Time Cost Rs/h (200 h/month)**	0.0	3.0	65.0	65.0	52.5
(22)	**Cargo Cost (Rs/h)**	0.0	0.0	2.5	5.0	0.0
(23)	**Total Cost (Rs/h)**	9.7	41.5	67.5	70.0	538.9

Source: *NDLI (1997)*

Notes:
(3) = (2) x 0.20. Assumed 20% for value of work time.
(4) = (3) x 0.20. Assumed 20% of value of non-work time.
(8) = (6) x (7)
(9) = (6) - (8)
(11) = (8) x (10)/100
(12) = (8) - (11)
(13) = (1) x (11) + (3) x (12) + (4) x (9)
For cars which had less than one crew, (15) = (14) x Rs 3,000
(19) is calculated from the extra revenue (here Rs 7,000/month) times the percentage of backhauls (here, approximately 20 per cent).
(20) = (15) + (16) + (17) + (18) + (19)
(21) = (20)/200. Assumed 200 working h/month
(23) = (13) + (21) + (22)

If goods spend less time in transit the amount of goods held in inventory may be able to be reduced. Some goods will not benefit from the time savings, for example, if they arrive before the business opens and thus cannot be unloaded. Thus, only a portion of the goods in transit should be included in the calculations. The value will depend upon the nature of operations in the country, or even in the area of the country, but values of 50-75 per cent are common.

The value of cargo time is calculated using the opportunity cost and the following equation:

$$\text{CARGO} = \frac{\text{PCTCGT OPC VACAR}}{365 \text{ x } 24} \qquad \ldots\text{(B10.1)}$$

where CARGO is the cargo cost in cost/h
PCTCGT is the fraction of vehicles whose cargo will benefit from time savings
OPC is the opportunity cost of the cargo as a decimal
VALCAR is the value of the cargo

B10.3 Calculating Travel Time in HDM-4

B10.3.1 Working Passenger-Hours

The annual number of working passenger-hours per 1000 vehicle-km is given by the following expression:

$$PWH = \frac{1000\ PAX\ PCTWK}{100\ S} \qquad \ldots(B10.2)$$

where PWH is the annual number of working passenger-hours per 1000 veh-km
PAX is the number of passengers (non-crew occupants) in the vehicle
PCTWK is the percentage of passengers on work-purpose journey

B10.3.2 Non-working Passenger-Hours

The annual number of non-working passenger-hours per 1000 vehicle-kilometres is given by the following expression:

$$PNH = \frac{1000\ PAX\ (100 - PCTWK)}{100\ S} \qquad \ldots(B10.3)$$

where PNH is the annual number of non-working passenger-hours per 1000 veh-km

B10.3.3 Crew Hours

The number of hours per crew member per 1000 vehicle-km is calculated using the formula:

$$CH = \frac{1000\ (100 - PP)}{100\ S} \qquad \ldots(B10.4)$$

where CH is the number of hours per crew member per 1000 veh-km
PP is the percentage of vehicle use on private trips

Thus, if the vehicle is being used for private trips the crew time is not included in the analysis.

B10.3.4 Cargo Holding Hours

The annual number of cargo holding hours per 1000 vehicle-km is calculated using the formula:

$$CARGOH = \frac{1000}{S} \qquad \ldots(B10.5)$$

where CARGOH is the annual number of cargo holding hours per 1000 veh-km

B10.3.5 Calculating Time and Crew Costs

The equations above establish the number of hours per 1000 veh-km for passenger working and non-working time; crew time, and cargo time. These values are multiplied by the appropriate unit cost for time to establish the time cost. To account for the speed-flow effects, the speed used in the calculations is the traffic-influenced speed. The individual values are multiplied by the portion of the year that the flow is at each of those speeds and summed to get the total annual cost.

B10.4 Overhead Costs

Overhead costs are calculated using the following equation.

$$OC = \frac{1000\, OA\, (100 - PP)}{100\, S\, HRWK} \qquad \ldots(B10.6)$$

where OC is the overhead cost per 1000 veh-km
OA is the annual overhead cost in cost/year

B11 Safety

B11.1 Introduction

Due to its complexity and uncertainty, traffic safety has traditionally been excluded from many highway investment analyses. For example, in HDM-III there was no mechanism for modelling traffic safety beyond treating it as an exogenous cost.

The growing realization of the substantial costs of road accidents (often over one per cent of GNP) has renewed the desire to include possible accident savings along with reductions in vehicle operating costs and time savings in economic appraisals. At the onset of the ISOHDM project traffic safety was identified as a key factor to be included in HDM-4.

The Transport Research Laboratory prepared a position paper recommending how traffic safety should be modelled in HDM-4 (TRL, 1995). The paper, entitled “Predicting Changes in Accident Rates in Developing Countries Following Modifications in Road Design” was used by ISOHDM (1995a) as the basis for a proposed method of including traffic safety in HDM-4. An international study was then conducted to try and gather the necessary data for use with HDM-4.

In the discussion which follows the term “accident rate” will be used. This is defined as:

$$ACCRATE = \frac{ACCYR}{EXPOSURE} \qquad \text{…(B11.1)}$$

where	ACCRATE	is the accident rate in accidents per 100 million veh-km
	ACCYR	is the number of accidents per year
	EXPOSURE	is the annual exposure to accidents

The exposure is usually expressed in terms of 100 million veh-km and is calculated as follows:

$$EXPOSINT = \frac{AADT\ 365}{10^8} \qquad \text{…(B11.2)}$$

$$EXPOSSEC = \frac{AADT\ 365\ SECTLEN}{10^8} \qquad \text{…(B11.3)}$$

where	EXPOSINT	is the accident exposure at intersections in million veh
	EXPSSEC	is the accident exposure between intersections in million veh-km
	AADT	is the total traffic entering the intersection or using the section in veh/day
	SECTLEN	is the length of the section in km

This chapter presents the recommended safety impact model for HDM-4. It commences with excerpts from the TRL position paper. This is followed by the recommended HDM-4 traffic safety model. The chapter closes with the results of the international study into accident rates.

B11.2 Modelling Traffic Safety

B11.2.1 Introduction

As described in TRL (1995), an evaluation was undertaken of the current ability to predict changes in accident rates in developing countries following modifications in road design. A review of work conducted in the developing world together with the lessons learned from developed countries' experiences follows below. It should be noted that the very different features of road and traffic conditions in developing countries (*eg* high pedestrian and non-motorised vehicle usage, conflicting land uses, and road user behaviour), substantially limits the potential for transferring information from developed countries to the developing world.

B11.2.2 Developing Countries Research

TRL (1995) identified five in-depth case studies conducted on the relationship between road geometry and accidents.

TRL studied the Nairobi-Mombassa road in Kenya and 29 routes scattered throughout Jamaica (Jacobs, 1976). The Road User Cost Study in India (CRRI, 1982) adopted the same approach and chose the Bombay-Pune road (114 non-urban km) and a second study of 34 routes with varying roadway characteristics (Hills and Jones-Lee, 1981). In Chile (Diaz, *et al*., 1983), 42 observations made on 15 two-lane paved road sections comprised the last case study. The most recent of these was presented in 1984.

Junction frequency, horizontal curvature and vertical curvature were common variables while the TRL studies also analysed surface irregularity. Road width was considered in the Indian study as well as in Kenya and Jamaica. Section lengths varied from uniform kilometre long sections to roads of 509 kilometres long. No study used geometric homogeneity to describe section length despite the assumption that the mean values of geometric design parameters used were representative.

The models developed differed in the choice of dependent variable, the variables found significant, and the analysis method. The TRL studies used personal injury accidents per million vehicle kilometre whilst the Chilean study chose total accidents per kilometre. The Indian studies developed equations for both indicators but preferred the latter as it was both simpler and had a higher R^2 value. While the models developed all had high R^2 values and significance levels, only junction frequency was found significant in all the equations.

The Chilean study argued that a flexible quadratic equation which considered second order effects was better at predicting accidents than the linear regression models used by the previous studies. The regression coefficient for flow increased by 40 per cent under the flexible form compared to the linear equation and the flexible form had a higher R^2 value.

None of these studies compared surfaced roads with earth roads. Nor did surface irregularity (often seen as a proxy for surface type) prove significant in either of the TRL studies, with it being highly correlated to the more dominant variable road width in Jamaica. A separate study on the Kenyan data found gravel roads to have a higher accident rate with 18 per cent of total accidents but only five per cent of the traffic .

A South African study (Bitumen and Tar Association, 1989) found total accidents to decrease when gravel roads were paved and while the proportion of fatalities and property damage-only accidents decreased, the share of injury accidents (both serious and slight) increased.

Unfortunately (particularly in light of TRL's activities in the early 1980s) little work has occurred in the past decade although accident and road inventory data has been collected in

Papua New Guinea but has yet to be fully analyzed. Preliminary analysis (simple regression only) found grade to have a strong correlation with accidents.

The South African formula for predicting accidents on rural highways was applied on the main highway connecting South Africa with Swaziland (Jordaan, 1995). The model is simple and easy to use but requires assumptions that may only apply to the more advanced developing countries. It uses base rates that are calculated for road type before adjusting these figures to specific road conditions. After determining total personal injury collisions (per 100 million veh-km), national fixed ratios are applied to determine accident severity rather than using local figures.

B11.2.3 Developed Countries Research

Despite the extensive number of studies, developed countries (perhaps surprisingly) have not been able to model accidents adequately on the basis of geometric parameters. Nor have they even been able conclusively to determine a preferred analysis method or the role of traffic flows.

Much relevant research has been recently sponsored in the US due to the Surface Transportation Act of 1982 which mandated a study on the cost-effectiveness of geometric design standards. The subsequent Transportation Research Board publication (TRB, 1986) developed a model for evaluating safety benefits but it was not proposed for national use and various state models still exist.

A review of European countries cost benefit methods for new roads (ECU, 1994) found most to assume different accident rates by road type or area location rather than geometric design as in Belgium and Spain. Due to the “severe constraints on the robustness of impact assessments”, it was concluded that assessment methods should be limited to criteria whose impacts are easily identifiable and significant.

Due to the acknowledged inability to predict accidents based on design characteristics, the UK road investment model, COBA, uses look-up tables for accident predictions (DOT, 1993). Roads are divided only according to classification and urban/rural settings with default personal injury accident rates (per million veh-km) used when local information is unavailable. The estimated number of accidents is thereby determined by road type, length, and traffic flow (urban roads with few junctions may use the rural road rate). COBA assumes that accident numbers are independent of flow although there is evidence to the contrary, with accident rates decreasing as traffic flow increases (McGuigan, 1987).

A recently proposed theoretical model from Sweden (Johansson, 1994) calculated risk separately for protected (motorised vehicle occupants) and unprotected road users. It also adopted a speed/flow model which increases the accident risk proportionally to increases in speed. This is the first model that reflects the potential of road improvements to increase accident risks. However, data requirements vary between being extremely specific with accident types and side ditch design details needed while exposure risks used can be based on national averages.

McLean (2000a and 2000b) investigated predicting accidents for Australia. For rural roads, the attributes affecting the accident rate in Table B11.1 were found (McLean, 2000a). For urban areas McLean 2000b noted that accident rates “are significantly affected by a number of attributes not necessarily captured within the broad [road] stereotype”, including:

- median treatment
- frontage access type and frequency
- abutting land-use and development

It was also noted that the above attributes are subject to a number of influences which are likely to differ between metropolitan regions. These include:

- historical road reserve widths
- planning policies
- development practices

"Hence, there can be marked variation in the characteristics of a road of given broad stereotype between metropolitan regions, or between sub-regions within a single metropolis" (McLean, 2000b).

Table B11.1: Factors Influencing Rural Accident Rates

Attribute	Comment
Traffic Volume	Conflicting results in the literature
Cross-Section Standard	For single carriageway roads, considers sealed/paved width only
Horizontal Curves	
Gradient	Literature implies an interaction with horizontal curves
Traffic Mix	
Roadside Hazards	Not included in inventories and difficult to quantify
Access Conditions	Not included in inventories
Overtaking Lanes	International literature indicates a safety benefit.

Source: McLean (2000a)

McLean (2000a and 2000b) analysed available data and established a base accident rate and adjustment factors for operating factors, applied through a multivariate model which was multiplicative.

B11.2.4 Conclusions on Modelling Traffic Safety

Whilst the developed countries have had limited success modelling accidents with respect to geometric design parameters, the complexity of developing countries road and traffic characteristics can only imply greater difficulties. Traffic flows are by no means homogeneous with significant numbers of heavy commercial vehicles, pedestrians and non-motorised road users. Wide variations in speed merit consideration as speed variations have a higher correlation to accidents than does mean speed alone.

As no consensus has been achieved between models, nor even between dependent variables or analysis methods, it is not possible to advocate the use of any one equation as a guideline for use elsewhere. In addition to the statistical discrepancies, the use of geometric means to represent road sections that were determined by geographical features or uniform lengths rather than homogeneous geometric design increases skepticism in the model. In Jamaica, a review of the data found wide ranges of values being obscured by the use of means.

The prospect of producing a model applicable to developing countries in general needs to be reconsidered. Unlike developed countries where fatality rates (deaths per 10,000 vehicles) vary little, the road safety situation varies tremendously in developing countries. For example, is it possible to develop one model for the developing world where fatality rates vary by at least twenty-fold between countries?

The complexity and uncertainty of accident predictions does not imply less need for their consideration. Accident costs are known to be substantial and need to be included in the design stage. A compromise adopted in the developed world, as seen by the example of COBA, in the UK is to develop a simple look-up table. Instead of pursuing equations based on geometric features, developing countries could focus on collecting before and after data on roads that have been widened or surfaced. Look up tables giving indication of changes in accident rate as a result of improving from one category to another (*eg* earth road to bituminous surface) could then be developed, for example as was done by McLean (2000a and 2000b) in Australia. Their use should be restricted to the host country as rates would be expected to vary widely between countries.

B11.3 The HDM-4 Traffic Safety Impact Model

B11.3.1 Introduction

The review of traffic accident predictive models by TRL (1995), summarised above, concluded that while developed countries have had limited success in modelling accidents, the same cannot be said of developing countries. The various studies showed little consensus between models, dependent variables, and even analysis methods.

The paper recommended that HDM-4 contain a series of “look-up tables” such as have been adopted for the COBA model in the U.K. These would give the change in accident rates as a result of improving from one road category to another. This method was also recommended at the International Workshop on HDM-4 which was held in Malaysia in November 1994 (ISOHDM, 1995b).

It needs to be appreciated that it is not always necessary to include accidents in an analysis. For example, the Tranfund New Zealand Project Evaluation Manual (Transfund, 1994) only includes accidents if the site has had two or more injury accidents recently or there are other specific causes to include traffic accident benefits.

B11.3.2 Modelling Traffic Safety Impacts in HDM-4

The basic equations for modelling traffic safety in HDM-4 are:

$$ACCYR = EXPOSSEC\ ACCRATE \qquad \ldots(B11.4)$$

$$ACCOST = ACCYR\ UNITCOST \qquad \ldots(B11.5)$$

where ACCCOST is the accident cost for the type of accident
UNITCOST is the unit cost for the type of accident

The accident rates are defined in HDM-4 using a tabular approach which is simple and easy to conceptualise. It is built around a broad, macro description of the expected accident rates which can be defined in several different ways.

ISOHDM (1995a) suggested Table B11.2. In this table there are “accident groups” and “accident types”. An accident “group” is a set of conditions which the accident rates apply to. The groups may be, for example, by road type, by geometry, by traffic, or any combination of factors. For each accident type, there is an accident rate (x1, x2, *etc.*) which is in terms of accidents per 100 million veh/km.

DOT (1994) used the values given in

Table B11.3 for the number of personal injury collision (PIC) rates per 100 million veh-km. Note that in this table the Accident Groups are in terms of road type. On the basis of the base number of PIC, the length of road and the AADT the number of personal injury collisions is obtained. A distribution of personal injury collisions is then used to divide the rate into the number of fatal, serious and slight collisions as well as damage only.

Table B11.2: ISOHDM (1995a) Recommended Accident Table

Accident Group	Accident Rate by Type of Accident			
	A	B	C	D
1	x1	x2	x3	x4
2	x5	x6	x7	x8
3	x9	x10	x11	x12
N				

Table B11.3: DOT (1994) Recommended Accident Table

Road Type	Surfaced 3 m Shoulders	Unsurfaced 3 m Shoulders	
	Long Radii	Long Radii	Short Radii
Freeway 18 m median	24	-	-
Freeway 6 m median	29	-	-
4 lane-interchanges	39	47	-
4 lane-at grade	69	83	-
2 lane-interchanges	34	41	-
2 lane-at grade	64	77	88
2 lane gravel	-	90	104
2 lane dirt	-	96	110

For HDM-4 NDLI (1995) recommended that accidents be included through a table such as that in Table B11.2. The number of "Accident Groups" would be defined within the model in a flexible manner. The groups would be defined as a function of any combination of independent variables using logical statements. For example, the group would be defined as:

- Road class = 1 .and. AADT > 5000 .and. AADT < 10000
- Pavement type = AC .and. skid resistance < 5
- Median = .T. .and. width < 3.5 .and. AADT > 1000

The most common applications would be to simply define the accident rate as a function of road type and perhaps volume, although in some situations where more detailed data or analytical techniques are available a more sophisticated set of groups would be established.

When there are improvements which will influence the accident rate, the user will need to define a group for the "after" rate. The analysis would then use the "before" rate for the do-nothing and the "after" rate for the improvement. It may be necessary to define several different "after" rates depending upon the treatments being investigated.

The following are some examples of applying the tabular data. It will be observed that the tabular approach allows for a great deal of flexibility in recognising data availability and the various improvement options.

Very Coarse Macro Data

If the country only has very coarse macro data, for example only knowing the total number of accidents and the total number of veh-km, they can only calculate a single value which is the gross accident rate. The table would therefore only have one entry:

Accident Group	Accident Rate by Type of Accident				
	Fatality	Injury	Damage Only	Pedestrian	All
All	-	-	-	-	x1

Macro Data by Accident Type

This is similar to the above, but the macro data have been broken down by individual accident types.

Accident Group	Accident Rate by Type of Accident				
	Fatality	Injury	Damage Only	Pedestrian	All
All	x1	x2	X3	x4	-

Macro Data by Road Type

Macro data are not available by individual accidents but by a road type.

Accident Group	Accident Rate by Type of Accident				
	Fatality	Injury	Damage Only	Pedestrian	All
2 lane	-	-	-	-	x1
4 lane	-	-	-	-	x2
Express-way	-	-	-	-	x3

Data by Road and Accident Type

Data are available by individual accidents and by road type.

Accident Group	Accident Rate by Type of Accident				
	Fatality	Injury	Damage Only	Pedestrian	All
2 lane	x1	x2	x3	x4	-
4 lane	x5	x6	x7	x8	-
Express-way	x9	x10	x11	x12	-

Data By Road and Accident Type - and Treatments

In this example, the application of a treatment is expected to modify the accident rate (may be increase or decrease depending on treatment).

Accident Group	Accident Rate by Type of Accident				
	Fatality	Injury	Damage Only	Pedestrian	All
2 lane	x1	x2	x3	x4	-
4 lane	x5	x6	x7	x8	-
Express-way	x9	x10	x11	x12	-
2 lane & Treat. 1	x13	x14	x15	x16	-

B11.4 Default Values for the HDM-4 Traffic Safety Impact Model

The model presented in Section B11.3 is conceptually very flexible and simple to apply. However, it was considered desirable that a default form of the model be adopted for HDM-4 and, if possible, default accident rate parameters recommended.

After reviewing the techniques adopted by various publications which have adopted similar models, it was decided to adopte three types of accidents:

- Fatality;
- Injury; and,
- Damage only.

Costs may be supplied for each of these accidents.

There are three classes of accidents:

- Section;
- Intersection; and,
- Total.

The section accidents apply to accidents on a section of road, or between intersections sometimes termed as mid-block section. The intersection accidents apply to accidents at an intersection. The total accidents are the aggregate number of accidents for all sections and intersections.

An international survey was undertaken to obtain default values for use with the HDM-4 traffic safety model. Letters were sent to over 38 organisations involved with traffic safety in different countries. They were requested to provide typical data to use in the HDM-4 model for their country. Only 15 organisations responded to the survey and out of that 10 countries managed to supply the information requested. The results are presented in Table B11.4. The following are values obtained for African countries for 1993.

Country	Personal Accident Injuries per 10,000 vehicles
Botswana	546
Lesotho	714
Namibia	7
Malawi	948
Mozambique	1821
South Africa	157
Swaziland	323
Tanzania	490
Zambia	555
Zimbabwe	353

B11.5 Summary

At the start of the ISOHDM study it was envisaged that HDM-4 would contain a model for predicting the effects of geometric changes on traffic safety. However, it was concluded that there was insufficient information available to develop or implement such a model.

Instead, a simpler approach based on tables of accident rates against road groups was adopted. A study was undertaken to obtain initial parameter values for use with the model. These were obtained for a number of different countries and the results are presented in Table B11.4.

Table B11.4: Results of International Study of Accident Rates

Road Class	Personal Injury Accidents per million veh-km								
Mid-block Section	**Norway**	**Ireland**	**Germany**	**Belgium**	**U.S.A.**	**Indonesia**	**Malaysia**	**Australia**	**Israel**
Urban Roads	0.32	-	-	-	0.61	0.554	-	1.8	1.00
Sub-Urban Roads	0.20	-	-	-	1.40	-	-	1.8	-
Rural Single-Lane Highways	0.17	0.13	-	-	1.02	-	-	-	-
Rural Two-Lane Highways	0.14	0.13	-	-	2.33	0.083	-	0.5	-
Rural Dual Carriageway Divided Highways	0.07	-	-	-	1.25	-	-	0.7	-
Expressways	0.10	-	0.147	0.17	1.11	0.420	-	0.8	0.38
Urban Intersection	**Personal Injury Accidents per million Vehicles Entering Intersection**								
Priority - Cross	0.18					0.08	-	-	
Priority - T	0.10					0.156	-	-	
Roundabout - 4 Leg	0.03					-	-	-	
Roundabout - 5 Leg	0.05					-	-	-	
Traffic Signals - Cross	0.10					0.266	-	-	
Traffic Signals - T	0.05					0.110	-	-	
General Accident Rates	**Rates per million veh-km**								
Fatality Rates							0.141		
Casualty Rates							0.897		

Source: *Survey conducted by NDLI (1995)*

B12 Non-Motorised Transport

B12.1 Introduction

In many countries non-motorised transport (NMT) constitute a significant component of the traffic stream. While many analyses have tended to view them simply as side-friction or a nuisance to motorised transport (MT) vehicles, they are in fact an essential means of moving passengers and freight in their own right. Hence, NMT impacts need to be considered in the analysis of many road projects and strategies.

Padeco (1996; 1997) present a comprehensive discussion and analysis of NMT modelling issues. Readers interested in further information should refer to these soruces.

B12.2 NMT Modelling in Economic Appraisals

In spite of the high numbers of NMT in some countries, NMT modelling has not received a great deal of attention in the literature.

For HDM-95 Hoban, *et al.* (1994) combined NMT and side friction into a single "roadside friction" factor (XFRI) which had a value between 0 and 1. This factor was used to reduce the free speeds of vehicles. Even though NMT would be expected to influence the capacity, this was not considered and thecapacity was expressed in terms of road width only. The effect of NMT were also not explicitly considered in the speed-flow modelling.

Roberts (1994) assigned a PCU value to NMT and the total NMT and motorised transport (MT) flow were used in the speed-flow modelling. As with Hoban, *et al.* (1994), NMT were not considered to influence capacity.

For Indonesia Hoff & Overgaard (1992; 1994) considered that NMT and MT flows were two separate streams and they will not mix when adequate provisions are made for NMT flow. The important factor is not the number of NMT, but whether or not they interfere with MT.

Shoulder usage is an important consideration on whether or not the streams mix since this increases the effective width of the pavement and reduces the interactions. The decision table presented in Table B12.1 was used to establish whether or not shoulders could be used by NMT. Shoulders were only used when they were firm, with smooth/minor rutted surfaces, and when they were not more than 10 cm lower than the road surface.

Table B12.1: IRMS Shoulder Usability Decision Table

Shoulder Width (m)	Shoulder Type	Shoulder Condition	Average Elevation	Usable ?
-	-	-	No Shoulder	No
-	-	No Shoulder	-	No
< 1	-	-	-	No
-	Soft	-	-	No
≥ 1	Firm	Smooth/Minor Rutting	Higher/Level/ Below Road Surface	Yes
≥ 1	Firm	Smooth/Minor Rutting	> 10 cm Below Road Surface	No
≥ 1	Firm	Severe Rutting	-	No

Source: Hoff & Overgaard (1992)

It was assumed that each NMT occupied an effective length of road, defined as that required for an MT to overtake an NMT. The ratio of this effective length to the average NMT spacing gave a factor which yielded an effective road width for the NMT. If the total effective width of NMT in both directions (assuming unbalanced flows) was greater than the total usable shoulder width (on both sides) then the residual width was deducted from the carriageway width to give the effective pavement width for MT. This effective width was then used in the capacity calculations.

The NMT were assumed to reduce the speeds of the MT through a reduction in carriageway width. The desired speed was then predicted using the following model:

$$VDESIR = VDESIR0 - VDIFF \;\; WRFACT \qquad \ldots(B12.1)$$

where	VDESR	is the desired speed on the road under examination
	VDESIR0	is the nominal desired speed associated with a two lane road
	VDIFF	is the maximum difference in vehicle speeds due to width reductions
	WRFACT	is a width reduction factor which is a function of the effective width of the carriageway

Further reductions in speed were then calculated for roughness and curvature. An additional delay was then added to account for the situations whereby the effective width of the pavement was of insufficient width for the free passage of MT.

Bennett and Greenwood (1996a) proposed a variation of the Hoff & Overgaard (1992) model for HDM-4. The basic concept was that the NMT reduce the capacity of the road available to the MT through a reduction in available road width. The amount of reduction is calculated from the effective occupancy of the road by the NMT, whereby an occupancy of one would result in zero capacity available to MT.

A key assumption of the model was that if the shoulder is usable the NMT will opt to travel on the shoulder instead of the carriageway. Once the shoulder is occupied by NMT there will be encroachment into the MT stream by NMT. The NMT stream is assumed to be in a single line until they are at their minimum spacings at which time they form a second stream. Figure B12.1 is an example of the effect of NMT on capacity using the proposed approach.

There were a several limitations with the proposed model:

- It gave priority to the NMT in that it was assumed that all the NMT flow would occur and that the MT would make room. No consideration was given to the case where the NMT capacity was reduced due to a large MT flow. The result of this assumption was that the capacity of the carriageway dropped to zero as the NMT flow increased. However, it would be anticipated that some small finite MT capacity would exist even under extremely high levels of NMT traffic. This was considered to be a major shortcoming of the model which ideally needed to be addressed in some form of equilibrium demand situation.

- It was assumed that there was no desire amongst the NMT to pass one another. In addition, the NMT were assumed to be travelling in one direction only on each side of the road.

- It did not consider unbalanced flows, although these could be modelled.

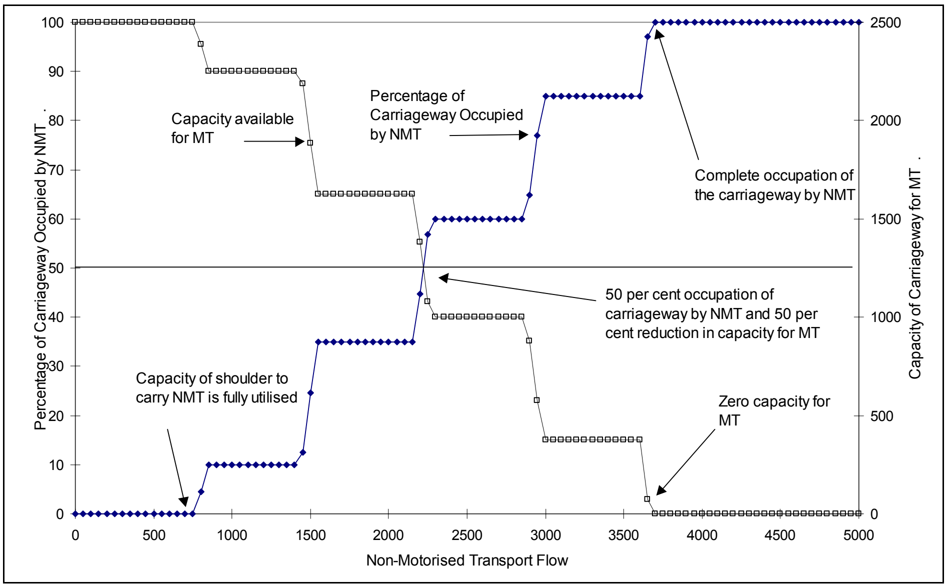


Figure B12.1: Effect of NMT on Capacity

Some of these deficiences were addressed by Bennett and Greenwood (1996b) but since further refinements were required the model was not developed any further.

Padeco (1996) considered the issue of modelling NMT in HDM-4 in detail. They proposed that NMT be modelled using five different representative vehicles:

- Pedestrian;
- Bicycle;
- Cycle-rickshaw;
- Animal cart; and,
- Farm tractor.

Although farm tractors are not technically NMT, in terms of operating performance resemble non-motorised vehicles more closely than motorised vehicles so they were included with NMT.

It was noted that road improvements catering for NMT would have wide ranging impacts on both NMT and MT. These included (Padeco, 1996):

- **Impact on MT Flow and Speed:** This would be represented by a side friction approach that takes the form of a speed reduction factor.
- **Impact on MT Operating Costs:** This impact would be incorporated by using the MT speeds calculated by the MT speed reduction factor.
- **Impact on Safety-Related Costs:** The magnitude of accident cost savings is potentially large, but statistically significant information is hard to obtain, particularly in developing countries. However, it was recommended that an NMT perceived level-of-safety submodel be included in HDM-4 to (as a minimum) identify and prioritise road sections that need improved NMT facilities.
- **Impact on NMT Flow and Speed:** Change of NMT speeds due to road improvements is essential to reflect time cost savings of NMT users.

- **Impact on NMT Operating Costs:** Reductions in NMT operating costs due to road improvements are an important element in the analysis.
- **Impact on Road Maintenance Costs:** It was considered that the collective influence of NMT traffic on pavement deterioration is minimal compared to impacts from MT traffic. In the case of unsealed roads, however, NMT may significantly impact deterioration, passability during wet seasons and maintenance costs. These impacts, which may increase road user costs considerably, have yet to be studied and therefore require investigation.
- **Impact on NMT Demand:** A wide variety of factors may be identified as having an effect on NMT demand. However, many of these factors (*eg* government policies, social attitudes toward NMT, environmental concerns, tourism and recreation) are difficult to quantify, and most factors are interrelated to some extent with no one factor controlling. Therefore, data collection and analyses required to analytically describe the dynamics of NMT demand were considered beyond the scope of the ISOHDM study.

Padeco (1996) went on to propose relationships for modelling the impact of NMT on MT flow and speed; perceived NMT level of safety; NMT speed; and NMT operating costs. The proposed models were somewhat complicated which led to refinements by Padeco (1997).

The work of Padeco (1996 and 1997) were developed by ISOHDM (1997) to establish the preliminary NMT modelling specifications for HDM-4. The approach adopted was to use mechanistic principles similar to those for MT which provided a more integrated framework, although at the expense of simplicity. Odoki and Kerali (1999a) revised this work to establish the final specifications. The sections which follow describe the modelling approach adopted.

B12.3 NMT Representation

The vehicle classification system for HDM-4 proposed by Kerali, *et al*. (1994) adopts a flexible approach in which a representative vehicle can be user specified based on one of several HDM-4 standard vehicles. Padeco (1996) recommend that the NMT category be added, with the following default representative modes/classes: pedestrian, bicycle, cycle-rickshaw, animal cart, and farm tractor. The user are able to specify their own set of NMT types within each class, as illustrated in Table B12.2.

B12.4 NMT Modelling in HDM-4

B12.4.1 Introduction

As described above, Padeco (1996) identified the following NMT impacts that need to be considered in economic analysis of road investment projects:

(i) Impact on MT Flow and Speed

(ii) Impact on MT Operating Costs

(iii) Impact on NMT Flow and Speed

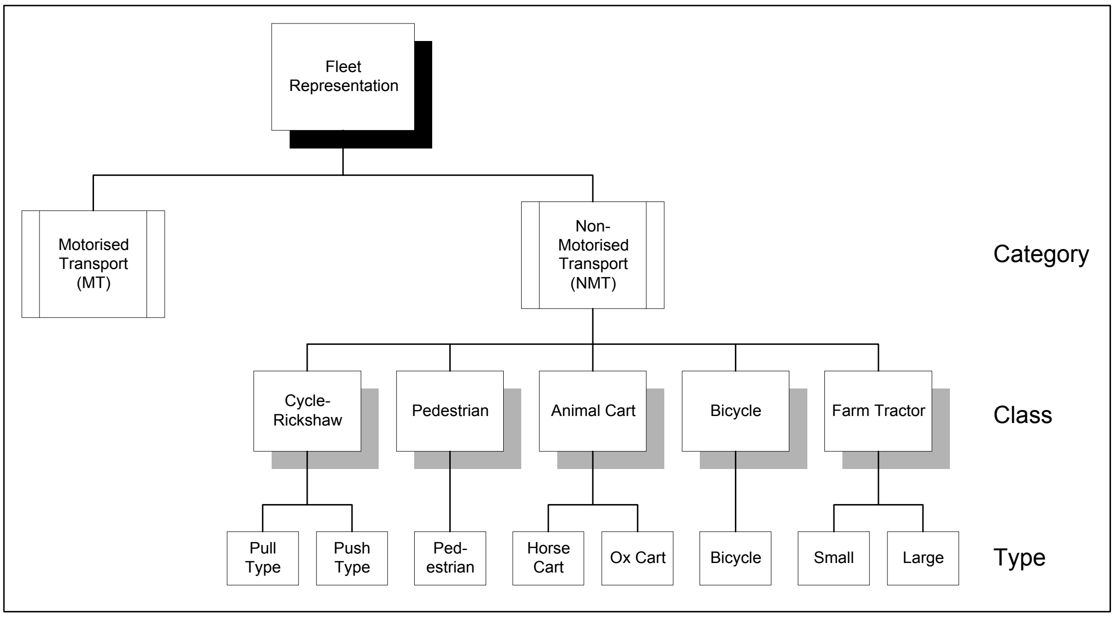
(iv) Impact on NMT Operating Costs

(v) Impact on Road Deterioration and Maintenance

(vi) Impact on MT and NMT Safety-Related Costs

(vii) Impact on NMT Demand.

The first four NMT impacts given in (i) to (iv) are modelled in HDM-4, and are discussed further in the sections that follow. Impacts on road deterioration and maintenance, road safety, and NMT demand are not currently considered in HDM-4.



Source: Padeco (1996)

Figure B12.2: NMT Representation in HDM-4

B12.4.2 Impact of NMT on MT Speed and Operating Costs

The impact of NMT on MT speed and operating costs is modelled in HDM-4 through the “side-friction” or speed reduction approach. For each road section, an MT speed reduction factor (XNMT) is user defined, and this is incorporated in the desired speed (VDES) relationship for MT vehicles (see Section B2.5.6). The XNMT value is maintained over the analysis years until a road improvement alters the characteristics or degree of conflict between NMT and MT.

Thus, XNMT factors account for the effects of NMT on motorised traffic speeds and operating costs, and hence the benefits to motorised traffic of improving shoulders and NMT facilities.

As described in Chapter B3.6.3, the NMT has an impact on the MT through the additional accelerations they induce. This is represented through the ‘acceleration noise’. The higher the NMT flow, the greater the level of acceleration and thus the MT RUE.

B12.4.3 NMT Speed Model

The estimation of NMT speeds will be used to measure NMT user benefits in terms of time savings. There are several possible factors influencing NMT speeds. These include the following, Padeco (1997):

- MT traffic volume and speed;
- NMT traffic volume;
- Roadside activities;
- Roadway grade;
- Rolling resistance;
- Road width (where NMT can safely travel) and/or number of lanes;
- Method of separating NMT/MT traffic (*eg*, markings, physical separation, NMT-only street);
- Roughness of road surface (particularly shoulder roughness); and,
- Inclement weather.

However, to capture the effects of all these factors would necessitate the formulation of a complex NMT speed model and calibration procedure.

A simplified speed model based on the concept of minimum limiting velocity approach for MT presented in Section B2.4 is proposed for HDM-4 as described below. NMT speed analysis over a road section will first be carried out separately for each of the possible two traffic flow directions, known as the uphill segment and the downhill segment, and the results averaged for a round trip.

The speed is calculated as follows:

$$v = XMT \max[0.14, \min(VDESIR, VROUGH, VGRAD)] \qquad \ldots(B12.2)$$

where VGRAD is the speed limited by gradient in m/s

XMT speed reduction factor due to motorised traffic and roadside activities, allowable range: min 0.4 to max 1.0 (default = 1.0)

The limit of 0.14 m/s corresponds to a speed of 0.5 km/h. The following describes how the speeds are calculated.

DESIRED SPEED (VDESIR)

Typical values of VDES for NMT (in km/h) are given in Table B12.2.

ROUGHNESS SPEED (VROUGH)

The effect of roughness on NMT speed is calculated as follows:

$$VROUGH = VDES + a0\ RI \qquad \ldots(B12.3)$$

where a0 is the roughness dependent model coefficient

NMT use the carriageway and also the road shoulders (if provided). The proportion of NMT traffic that use the carriageway and that which use the shoulders will need to be specified, so that carriageway roughness is applied to the former traffic, and the shoulder roughness applied to the latter traffic proportion. However, the current HDM-4 version does not model separately the deterioration of shoulders over time. The carriageway roughness will be used in the above model for the time being. Future enhancements may therefore include identification of separate NMT performance predictions for travel on shoulders, with user-specified percentage of NMT flows on the shoulders.

Table B12.2 shows the default parameter values for the NMT speed-roughness model.

Table B12.2: Default NMT Speed Model Parameters

NMT Type	NMT Speed Model Parameter			
	VDES - Paved (km/h)	VDES - Unpaved (km/h)	Roughness a0	Gradient a1
Pedestrian	5.1	4.6	-0.048	-9.2
Bicycle	21.3	18.0	-0.225	-49.0
Cycle-Rickshaw	18.6	15.4	-0.197	-47.0
Bullock Cart	3.8	3.2	-0.036	-6.0
Farm Tractor	30.0	24.0	-0.250	-63.0

Source: Odoki and Kerali (1999a)

VGRAD

The effect of gradient on NMT speed is calculated as follows:

$$VGRAD = VDES + a1\ GR \qquad \ldots(B12.4)$$

Table B12.2 contains values for the coefficient a1.

As described in Odoki and Kerali (1999a), the gradient speed model implementation includes the critical gradient length approach as was used with MT (see Chapter B2.5.3).

ROLLING RESISTANCE (VROLL)

This was included in the preliminary specifications for HDM-4 (ISOHDM, 1997), but were removed by Odoki and Kerali (1999a). The effect of rolling resistance on NMT speed was calculated as follows:

$$VROLL = VDES + a1\ FR \qquad \ldots(B12.5)$$

where FR is the rolling resistance in N
a1 is the rolling resistance coefficient

The rolling resistance FR was calculated using the same model as with MT vehicles presented in Chapter B1. The wheel factor CR1 depends upon the wheel type. For steel wheels it has a value of 0.9; for pneumatic tyres 1.0.

B12.4.4 NMT Time and Operating Costs

NMT time and operating cost are calculated as follows:

$$TOC = TMC + VOC \qquad \ldots(B12.6)$$

where TOC is the time and operating cost of NMT per vehicle-km
TMC is the time cost of NMT in cost/km
VOC is the operating cost of the NMT in cost/km

TIME-RELATED COST

The time-related cost, TMC, comprises passenger time cost and the cargo holding cost:

$$TMC = PAXC = CARGC \qquad \ldots(B12.7)$$

where PAXC is the passenger time cost for NMT in cost/veh-km

	CARGC	is the cargo holding cost for NMT in cost/km

To calculate the passenger time the following equation is used:

$$PAXC = \frac{PAX \; CTIME}{S} \qquad \ldots(B12.8)$$

where	PAX	is the number of passengers per NMT vehicle
	CTIME	is the time cost per passenger in cost/h

Similarly, the cargo holding cost is calculated as:

$$CARGC = \frac{CAGV}{S} \qquad \ldots(B12.9)$$

where	CAGV	is the average hourly value of cargo holding time in cost/h

VEHICLE OPERATING COST

The vehicle operating cost is calculated as:

$$VOC = CAPCST + RMC + CRWC + ENC + OVHD \qquad \ldots(B12.10)$$

where	CAPCST	is the capital cost in cost/km
	RMC	is the repair/maintenance cost in cost/km
	CRWC	is the crew cost in cost/km
	ENC	is the energy cost in cost/km
	OVHD	is the overhead cost (excluding pedestrians) in cost/km

CAPITAL COST

The capital cost is derived from the purchase cost, depreciated over the average life of each NMT, and the interest cost. It is calculated as:

$$CAPCST = DEPCST + INTCST \qquad \ldots(B12.11)$$

$$CAPCST = \frac{PCHC}{LIFE0 \; AKM} + \frac{PCHC \; AINV}{200 \; AKM} \qquad \ldots(B12.12)$$

where	DEPCST	is the depreciation cost in cost/km
	INTCST	is the interest cost in cost/km
	PCHC	is the purchase cost of the NMT
	LIFE0	is the average service life of NMT in years
	AKM	is the average annual kilometreage of NMT in years
	AINV	is the average interest charge in per cent

The annual utilisation AKM is calculated from the hours worked as:

$$AKM = S \; HRWK0 \qquad \ldots(B12.13)$$

where	HRWK0	is the average number of NMT working hours per year
	S	is the annual average speed of NMT in km/h.

REPAIR AND MAINTENANCE

The repair and maintenance cost, RMS, consists of the costs of maintaining the NMT vehicle. This includes factors such as tyres, lubricants, brakes, *etc.* It is predicted as a function of roughness and NMT age using the following equation:

$$RMC = (a0 + a1\ RI)\ CKM\ PCHC\ 10^{-3} \qquad \text{…(B12.14)}$$

where CKM is the cumulative kilometreage in km calculated as 0.5 AKM LIFE
a0 and a1 are coefficients as given in Table B12.3

Table B12.3: Default NMT Maintenance and Repair Coefficients

NMT Type	Maintenance Model Coefficients	
	a0 x 10^{-6}	a1 x 10^{-6}
Pedestrian	N/A	N/A
Bicycle	1.600	0.267
Cycle-Rickshaw	0.712	0.178
Bullock Cart	2.780	0.617
Farm Tractor	-	-

Source: Odoki and Kerali (1999a)
Notes: '-' Values for farm tractors were not given

CREW COST

The crew cost is calculated as:

$$CRWC = \frac{CRWV}{S} \qquad \text{…(B12.15)}$$

where CRWV is the average crew wages in cost/h

ENERGY COST

As described by Kerali, *et al.*(1997), energy cost per km, ENC, is calculated as follows:

$$ENC = ENUSD\ UCEN \qquad \text{…(B12.16)}$$

where ENUSD is the average energy consumption of NMT in Joules/km
UCEN is a user-defined unit cost of energy used by NMT in cost/Joule.

Energy consumption per km, ENUSD is calculated for uphill and downhill sections separately as:

$$ENUSD = (Fr + Fg)\ DIST \qquad \text{…(B12.17)}$$

where Fr is the rolling resistance in N
Fg is the gradient resistance in N
DIST is the distance travelled in m (= 1000)

The average energy consumption for a round trip is calculated by the expression:

$$ENUSD = \frac{2\ Kef}{\left[\left(\frac{1}{ENUSD_u}\right)+\left(\frac{1}{ENUSD_d}\right)\right]} \qquad \text{…(B12.18)}$$

where $ENUSD_u$ is the uphill energy consumption in Joules/km
$ENUSD_d$ is the downhill energy consumption in Joules/km
Kef is an energy efficiency factor to account for energy used to overcome other forces opposing motion (default=1.1).

The gradient resistance is calculated separately for the uphill and downhill segments using the following expression:

$$Fg = WGT\_OPER \; g \; GR \qquad \ldots(B12.19)$$

OVERHEAD COST

The overhead cost covers all other cost components including taxes, administration, *etc.* The average cost is calculated as:

$$OVHD = \frac{OHC}{S \; HRWK0}$$

where OHC is the overhead cost in cost/yr

B12.5 Summary

The NMT model presented here represents a forward step towards considering NMT in economic appraisals. However, as described by Padeco (1996) and Padeco (1997), there are many areas where fundamental research into NMT issues is completely lacking. For example, the effects of NMT on capacity is not considered in HDM-4.

The NMT cost model covers all elements of NMT costs, however, this is at the expense of simplicity and it is quite data intensive, and relativley unvalidated. Odoki and Kerali (1999a) present the default parameters for NMT modelling in Table B12.4 these will require calibration before they can be applied with confidence. It is hoped that the inclusion of NMT in HDM-4 will lead to additional research in this area with the intent of developing a model which fully reflects the benefits from road projects which consider NMT.

Table B12.4: Default HDM-4 NMT Parameters

Key Parameters	Units	NMT Type			
		Bicycle	Rickshaw	Bullock Cart	Pedestrian
Wheel type		Pneumatic	Pneumatic	Wood	N/A
Number of wheels		2	3	2	N/A
Wheel diameter	M	0.7	0.7	1.0	N/A
Operating weight	kg	100	300	1200	80
Payload	kg	35	235	900	15
Average life	year	10	6	3	N/A
Baseline annual kilometres	km/year	2500	7200	4000	N/A
Number of working hours	hours/year	150	500	1300	N/A
Number of passengers		1	3	0	1
Purchase cost	US Dollars ($)	75	130	180	N/A
Annual interest rate	per cent	12	12	12	N/A
Crew wages	$/hour	-	0.22	0.22	N/A
Energy cost	$/MJoules	0.10	0.10	0.07	0.10
Overheads	Per year	0	0	0	N/A
Passenger time value	$/hour	0.36	0.36	N/A	0.36
Cargo holding time value	$/hour	N/A	N/A	0.49	N/A

Source: Odoki and Kerali (1999a)
Notes: N/A - Not appropriate

B13 The Effects of Road Works on Traffic and User Costs

B13.1 Introduction

Road construction or maintenance works result in additional delays, vehicle operating costs and emissions to road users. When the traffic volume becomes large, these road user effects can be significant and may have an impact on the type of treatment selected: treatments which minimise the disruptions to road users may be preferable even though they have higher capital costs.

HDM-III did not consider the impact of work zones on users—hereinafter called work zone effects—although it was possible to consider them as an exogenous cost in the analysis. For HDM-4 these costs were considered important so a simple procedure was developed for calculating the RUE due to work zones.

This chapter presents the model for predicting the effects of road works on road users. In the discussion which follows the term **work zone** is used. This is the portion of road where pavement construction or maintenance arises. As will be shown, the work zone effects encompass not only this portion of road, but also in advance and beyond it.

This chapter also presents models for estimating the associated vehicle operating costs in terms of travel time and fuel associated with speed cycle changes. The functions provided are equally suitable to any deliberate change in speed, such as those occurring at traffic signals and the like.

B13.2 Work Zone Effects

Work zones lead to two principle effects on road users, both of which are a direct result of capacity reductions. These effects are:

- Reductions in operating speed and associated speed changes; and
- Increase in travel times and associated queue formation.

The situation in urban areas is distinctly different to that in rural areas. Urban areas generally have several alternative routes available that allow commuters already familiar with the extent of queuing at the site to divert to other routes—assuming these have adequate capacity available. Under such circumstances, the only way of adequately modelling the effects is through some form of network assignment model.

By comparison, in rural areas there are often few, if any, alternative routes. Consequently, the users often queue at the road site. Taking into account diversion is therefore much easier since it can usually be based on a single alternative link assignment model.

The methodology provided here does not allow for the diversion situations described above, but rather assumes that drivers have no option other than to queue to use the existing road, which is operating at a lower capacity.

In assessing the impact of work zones there are a number of factors that need to be considered:

- Time of day and duration of activity;
- Traffic volume (veh/day or veh/h);
- Speed-volume characteristics for the road; and,
- Road capacity both before and through the work zone.

A work zone results in a number of different effects on the traffic stream. An example of this is illustrated in the Figure B13.1. Vehicles are travelling at an approach speed. In advance of the work zone they are forced to decelerate. If there is a queue, they will be stationary for a time as well as moving up through the queue. Once they reach the front of the queue they will accelerate up to the speed they will travel at through the work zone. Upon reaching the end of the work zone they will then accelerate back to their initial speed.

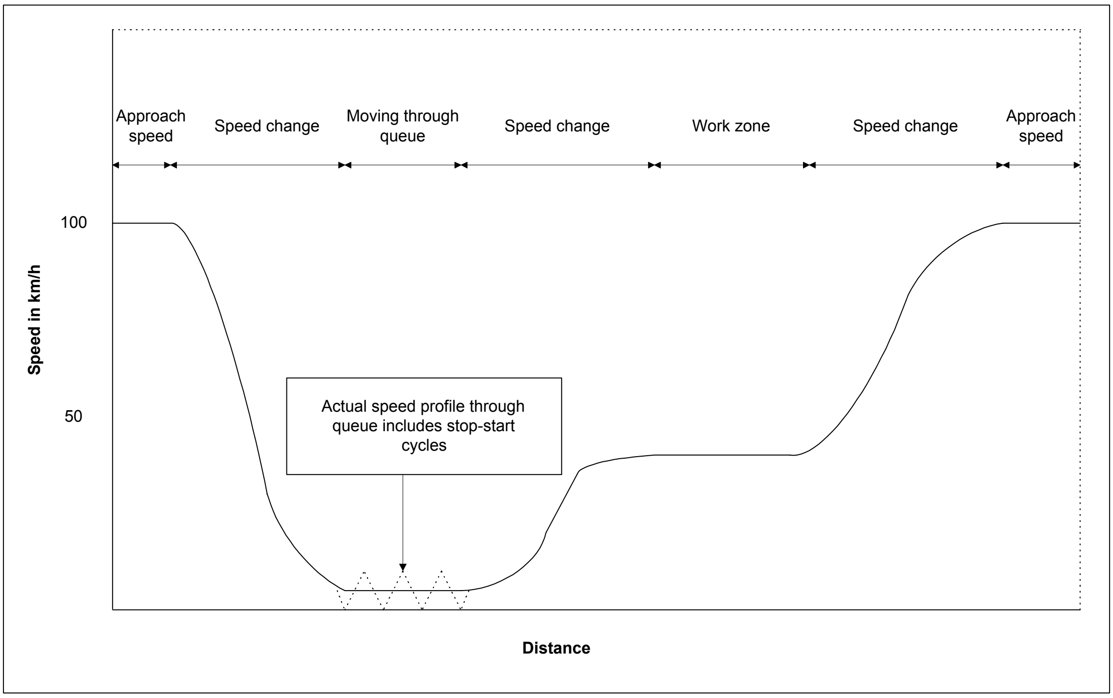


Figure B13.1: Example of Effects of Work Zones on Road Users

For the purposes of modelling the effects of work zones it is therefore necessary to consider:

- Speed change cycles;
- The effects of being queued; and,
- The effects of travelling through the work zone at a reduced speed.

The rate of queue build up and dissipation is a particularly important consideration in modelling the road user effects. As illustrated in Figure B13.2 (Memmott and Dudek, 1982), the size of the queue at any time is a function of the arrival and departure rate. Depending upon the situation, these may be governed by the capacity of the road (as in Figure B13.2), by traffic control devices, or both.

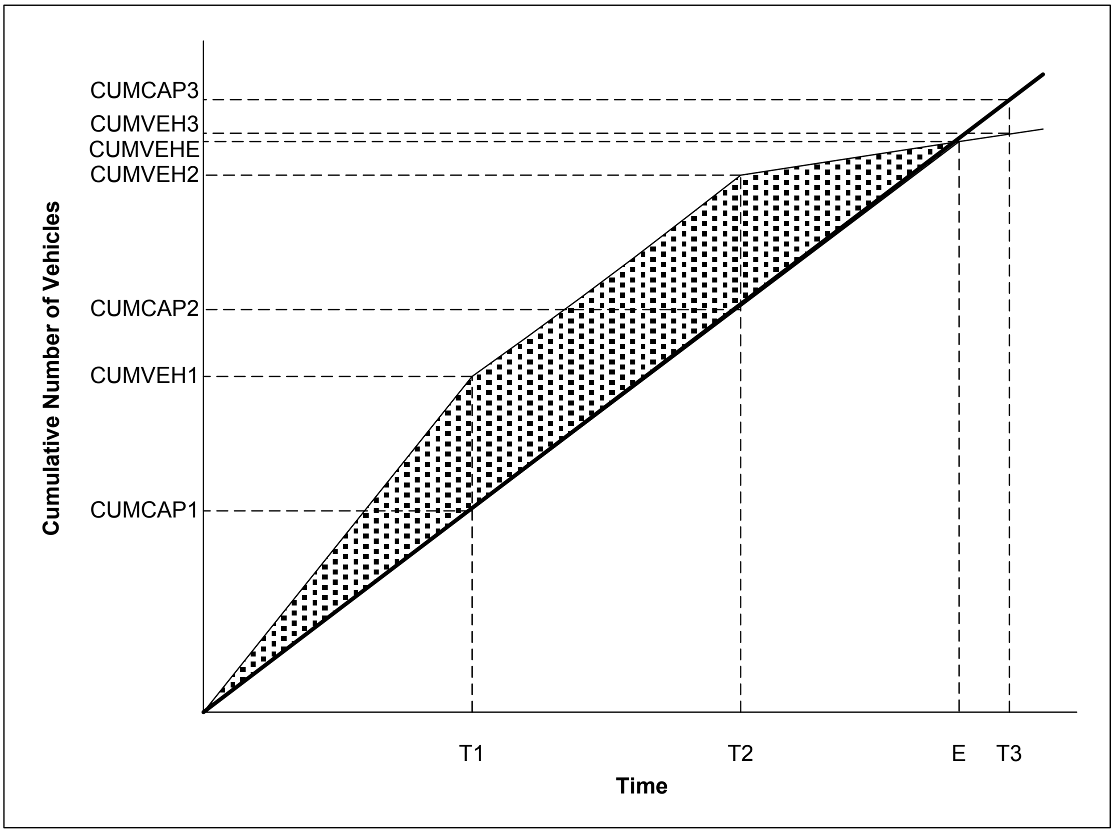


Figure B13.2: Build Up and Dissipation of Queues

In Figure B13.2 the size of the queue at any time is given by the difference between the top line (cumulative arrivals) and the diagonal line (cumulative departures). For example, at T1 the queue size would be CUMVEH1 - CUMCAP1. The area between the two lines is the total delay to those queued, in veh-h.

A key issue is that the modelling framework must include all the different types of work zone that are encountered (see Figure B13.3). These work zone types can be modelled as three basic cases with a fourth as a special combination case:

- **Case 1: Multi-lane Highway With One or Both Directions Affected.** This arises with a multi-lane facility where one or more of the lanes are closed.
- **Case 2: Two-lane Highway With Both Directions Affected.** This arises when a single lane of a two-lane highway is closed. The traffic from both directions then travel alternating in the single open lane.
- **Case 3: Multi-lane Highway, Two-Lane Highway or One-Lane Highway Temporarily Closed.** In this case, the vehicle flow is completely stopped over a period of time.
- **Case 4:** A special combination of Case 3 plus either Case 1 or 2.

Figure B13.3: Types of Lane Closures

The modelling for Cases 1 and 2 also needs to include the situation when the arrival of vehicles (temporarily) exceeds the capacity of the work zone.

Three distinct components need to be considered when modelling the effects of road works:

TRAVEL THROUGH THE WORK ZONE

Due to the reduced capacity of the work zone the vehicles will travel through at a reduced speed and under congested conditions. This generally leads to higher RUE.

QUEUING EFFECTS

Queuing at a work zone arises when:

- The vehicle arrivals approach or exceed the restricted capacity of the work zone[1] ;
- The work zone is temporarily closed (*ie* work zone capacity is zero).

For vehicles delayed there are two states:

- **A moving queue** where the vehicles move forward at a slow speed (based on the restricted capacity of the work zone);
- **A stationary queue** where all the vehicles are stopped (temporarily for a period of time).

In HDM-4, the RUE for these situations are calculated as follows:

- Moving queued vehicles - fuel and travel time costs are based on the calculated speed through the queue; and,
- Stationary queued vehicles - fuel consumption is related to the idle rate and the time of stationary, the travel time costs are also proportioned to the time stationary.

SPEED CHANGE CYCLE EFFECTS

As vehicles approach the work zone they must decelerate from their approach speed. They may then join queues or else travel directly through the work zone at a reduced speed. After the work zone they will accelerate back to their original speeds (see Figure B13.1). These speed changes incur additional travel time and vehicle operating costs and equations are given later in this chapter for calculating these values.

A flow chart which shows how work zone effects are modelled is given in Figure B13.4 . It should be noted that although diversions are included in the flow chart, due to their complexity they are not addressed in the methodology.

The following sections describe the HDM-4 work zone model as currently implemented in the HDM Tools software program ROADWORK[2].

---

1 Due to the randomness or fluctuations in the arrival rate, queuing will occur prior to the average arrival rate exceeding the capacity of the work zone.

2 As described in Bennett and Paterson (1999), HDM Tools is a suite of software programs designed to aid the use of the primary HDM-4 software. The program ROADWORK applies the logic of this chapter to cost the impacts of road works on users.

Figure B13.4: Flowchart for Establishing Work Zone Effects

B13.3 The HDM-4 Work Zone Effects Model

B13.3.1 Introduction

Greenwood, *et al.* (1995) reviewed the various available techniques and evaluated ones from the U.K., U.S.A. and N.Z. in detail. On the basis of their review a composite model was developed which combined the work from the U.S.A. on motorways with the N.Z. two-lane highway model. This approach, although providing simple solutions, had limitations with respect to random arrivals for Case 2 lane closures and queue build-up and dissipation under extended periods of complete closure.

NDLI (1995) presented a simulation model developed from the work of Greenwood, *et al.* (1995) which overcame the deficiencies for Case 2 lane closures and presented output from this model. The use of the simulation output in HDM-4 analyses was complicated so Bennett (1998b) proposed an analytical solution that was based on regression equations developed from the simulation output. However, as noted by Bennett (1998b), this approach had deficiencies, particularly at the boundaries where the regression equations were not good predictors. It was therefore decided to incorporate the output from the simulation program directly to HDM-4 and to use these values for calculating work zone effects. The simulation model was rewritten for Windows to reflect these changes and is described in Bennett and Paterson (1999).

It should be appreciated that under certain configurations of road works, deterministic solutions are available. However, the simulation approach adopted here will work for all situations. The deterministic models as proposed by Greenwood, *et al.* (1995) are given in Annex B13.1. While many of these deterministic models are not used in the current modelling logic, they are provided to aid the reader in understanding the theory of queuing delays.

B13.3.2 Simulation Model Description

The simulation model generates the arrival time of each vehicle and, from the capacity of the work zone, calculates the departure time. Depending on the traffic volumes and capacities, the model determines whether vehicles are required to stop, or just undergo a reduction in speeds. The simulation logic is outlined in Figure B13.5. The following are key elements to the simulation model.

CAPACITY OF WORK ZONE

Many factors affect the capacity of the work zone including lane width, lateral clearances, grade, vehicle composition, road alignment and pavement conditions. The estimated capacity of the work zone should be based on the worst conditions at a point along the length, as this will be the critical value. This is a required input for all cases.

Unfortunately, there is very little published empirical data available to estimate the capacity of the work zone. Table B13.1 gives the capacity of the work zone for multi-lane highways as reported by Dudek and Richards (1981) and Memmott and Dudek (1982), based on field studies.

For high standard two-lane highways, Tate (1993) found the capacity of the work zone lanes ranged from 900 to 1200 veh/h/lane. Like the data summarised in Table B13.1 for multi-lane highways, considerable capacity variations were recorded from site to site. It is recommended that in the absence of local data a value of 1000 veh/h/lane be used for the capacity of work zones on two-lane highways.

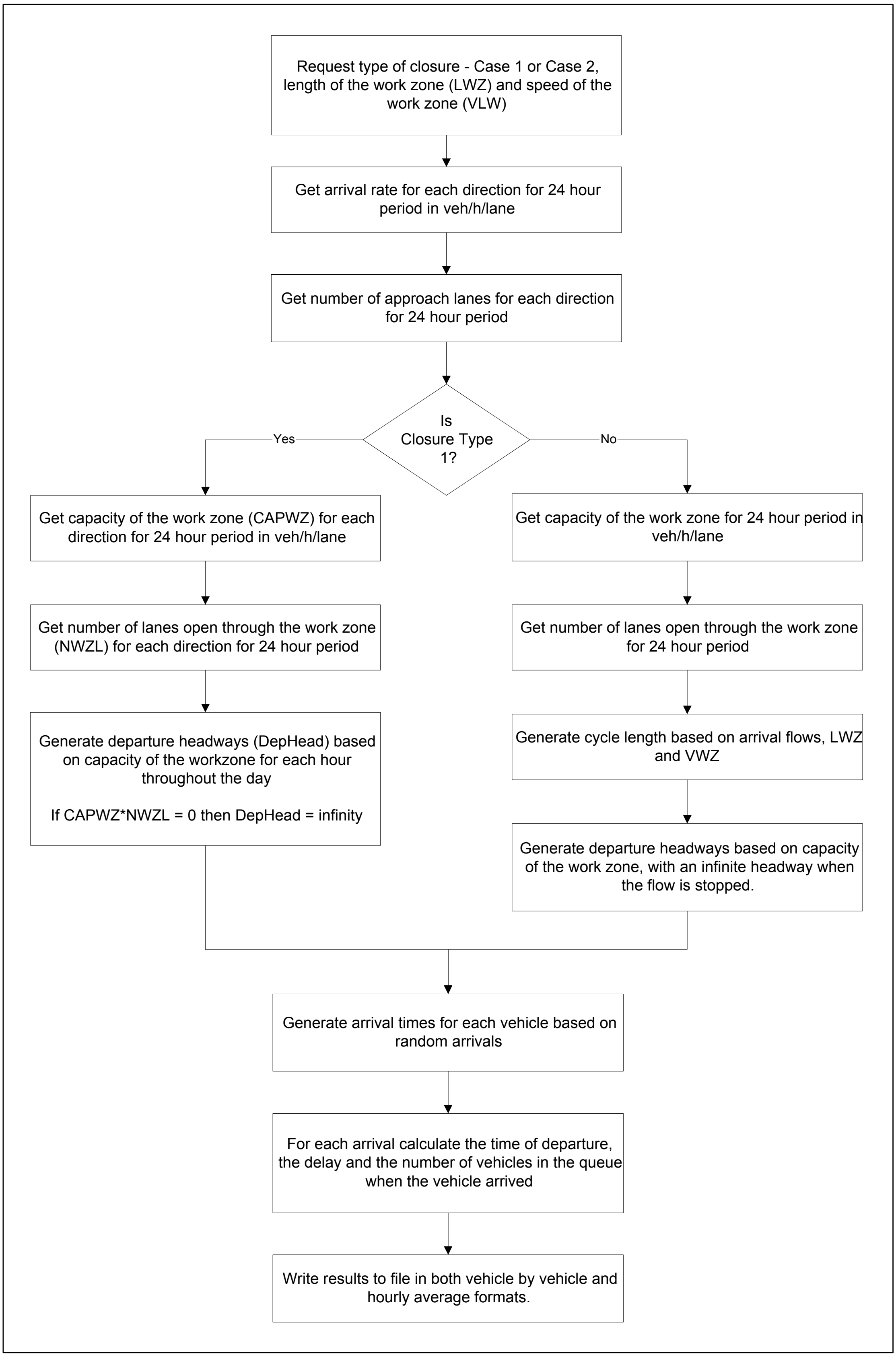


Figure B13.5: Flowchart of Simulation Program

Table B13.1: Multi-lane Highways - Estimated Restricted Capacity Through Work Zone

Normal Number of OpenLanes in One Direction	Number of Open Lanes Through Work Zone in One Direction in veh/h/lane				
	1	2	3	4	5
2	1300				
3	1200	1500			
4	1200	1500	1530		
5	1200	1400	1500	1550	
6	1200	1300	1400	1500	1600

Source: NDLI (1995) based on Dudek and Richards (1981) and Memmott and Dudek (1982)

For inclusion into HDM-4, the above work zone capacities have been converted into Passenger Car Space Equivalent (PCSE) volumes. The PCSE values for a variety of road types are given in Table B13.2 and were discussed in Chapter B3.

Table B13.2: Passenger Car Space Equivalents (PCSEs) by Road Type

Road Type	Car	LDV	Heavy Bus	Light Truck	Medium Truck	Heavy Truck	Articulated Truck
Single Lane	1.0	1.0	2.2	1.5	1.8	2.4	3.0
Intermediate	1.0	1.0	2.0	1.4	1.6	2.0	2.6
Two Lane	1.0	1.0	1.8	1.3	1.5	1.8	2.2
Wide Two Lane	1.0	1.0	1.8	1.3	1.5	1.8	2.2
Four Lane	1.0	1.0	1.8	1.3	1.5	1.8	2.2

Source: Hoban, et al., (1994)

To convert the work zone capacities as given by Dudek and Richards (1981), Memmott and Dudek (1982) and Tate (1993) into PCSE values it is necessary to know the composition of the traffic stream. In the absence of such data, it has been assumed that 85 per cent of the traffic stream is made up of light vehicles (PCSE = 1.0) and the remaining 15 per cent is evenly distributed among the heavy vehicles. The resulting work zone capacities recommended as default values for HDM-4 are given in Table B13.3.

Table B13.3: Default HDM-4 Work Zone Capacities

Case 1 Lane Closure - No Lane Sharing					
Normal Number of Open Lanes in One Direction	Number of Open Lanes Through Work Zone in One Direction PCSE/h/lane				
	1	2	3	4	5
2	1430				
3	1320	1650			
4	1320	1650	1680		
5	1320	1540	1650	1705	
6	1320	1430	1540	1650	1760
Case 2 Lane Closure - Lane Sharing					
Capacity of Single Open Lane for Both Directions of Traffic				1100	

Source: NDLI (1995) adapted from Dudek and Richards (1981), Memmott and Dudek (1982) and Tate (1993)

HOURLY DISTRIBUTION OF TRAFFIC

Since the simulation model works on an individual vehicle basis, the hourly distribution of traffic volume is a critical input parameter. The simulation program was written to accommodate input of hourly traffic volumes or to utilise a standard flow profile, which gives the percentage of the AADT per hour. The standard profile was developed from traffic data on a two-lane highway in N.Z., which had an AADT of approximately 6500 veh/day. Figure

B13.6 contains both the standard profile in use, plus profiles from a N.Z. motorway site and roads in Thailand.

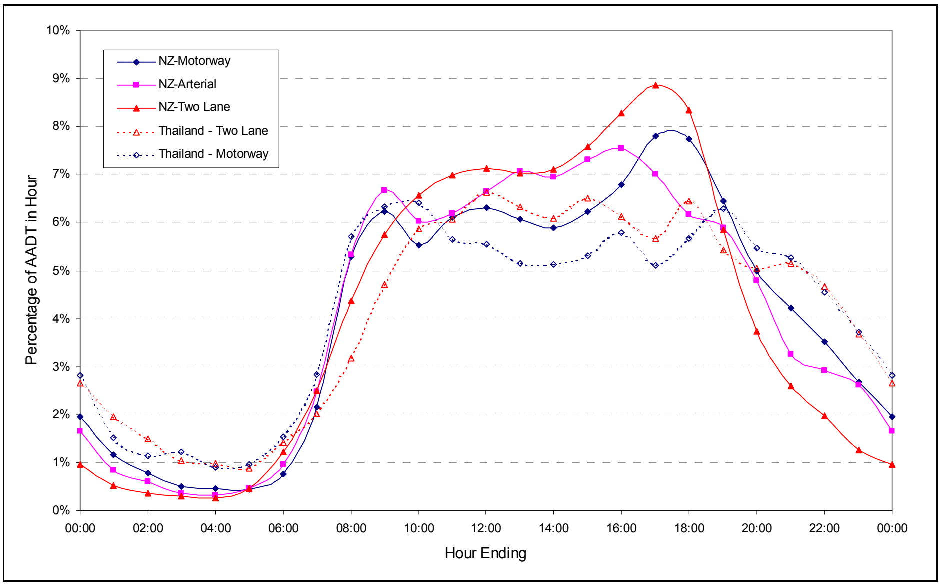


Figure B13.6: Examples of 24 Hour Flow Profiles

As the flow profile of a road is very dependent upon local conditions, the results presented in this report can only be used to give an indication of the form of the output from the simulation. For HDM-4 it is anticipated that users will input their own flow profiles and/or hourly traffic volumes to improve the accuracy of the predictions.

HEADWAY DISTRIBUTION

In addition to the use of hourly traffic volumes, headway data are required. Headways within each hour were assumed to follow a shifted negative exponential distribution (Bennett, 1994). For each vehicle within an hour period, the headway was calculated on a random basis. These headways were then adjusted to sum 3600 seconds, to ensure that the correct number of vehicles arrived at the work zone within the hour. The formula utilised for calculating the headway of a vehicle based on the shifted negative exponential distribution is:

$$hw_i = [hm - (ha - hm)\ln(1 - RAND)] \frac{3600}{\sum_{i=1}^{ARRIVALS} hw_i} \qquad \ldots(B13.1)$$

where hw_i is the randomly generated headway in s
hm is the minimum headway for a following vehicle in s
ha is the average headway for the hour in s
RAND is a random number between 0 and 1
ARRIVALS is the number of arriving vehicles in the hour for the direction

The minimum headway (hm) was based on a single direction of traffic, and from data in Bennett (1994) was set equal to one s. For use in the simulation program it was necessary to divide hm by the number of approach lanes. The average headway for each hour was calculated from the number of arrivals within the hour as:

$$ha = 3600/ARRIVALS \qquad \ldots(B13.2)$$

The resulting distribution of headways for an AADT of 10,000 vehicles per day is given as Figure B13.7. Headways greater than 30 s which made up approximately 14 per cent of the total headways are not shown in Figure B13.7.

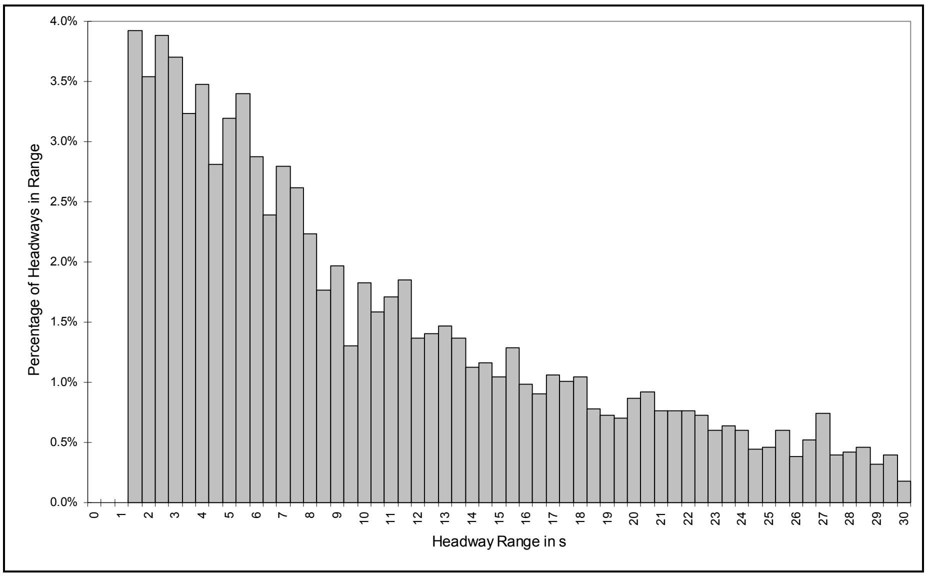


Figure B13.7: Distribution of Simulated Headways Over 24 Hour Period

B13.3.3 Examples of Simulation Model Output

To illustrate the output from the simulation model, a number of simulations were conducted. The average delay per vehicle and the average queue length as observed by the arriving vehicle, both on a 24 hour basis, were generated. As noted earlier, the results of the simulation program are highly dependent upon the 24 hour flow profile used, and hence the following figures are only intended to illustrate the output and not to provide definitive solutions for all cases.

Figure B13.8 and Figure B13.9 illustrate the average queue length and delay, respectively, for a Case 1 lane closure on a multi-lane highway. In Figure B13.9 it is interesting to note that doubling both the capacity and 24 hour demand does not result in a constant value for the average delay to vehicles *ie* a two-lane work zone can carry twice the traffic of a one lane work zone with a lower average delay per vehicle[1].

Figure B13.10 illustrates the average queue build up and dissipation over a 24 hour period for a work zone involving lane sharing (Case 2). As expected there is a large increase in the average queue length when the arrival rates approach capacity of the work zone during the peak periods.

Figure B13.11 and Figure B13.12 are for a work zone involving lane sharing (Case 2). As with the multi-lane highway (Case 1) the average queue length drops to zero as the AADT falls. However, unlike the Case 1 where no lane sharing is required, Case 2 has a minimum average delay associated with the minimum cycle length (60 s) and the lost time (2 s).

[1] This anomaly is explained analytically for constant flow conditions in the equation in Annex B13.1, which indicates that the average delay per vehicle is not linearly proportional to the ratio of arrivals to capacity.

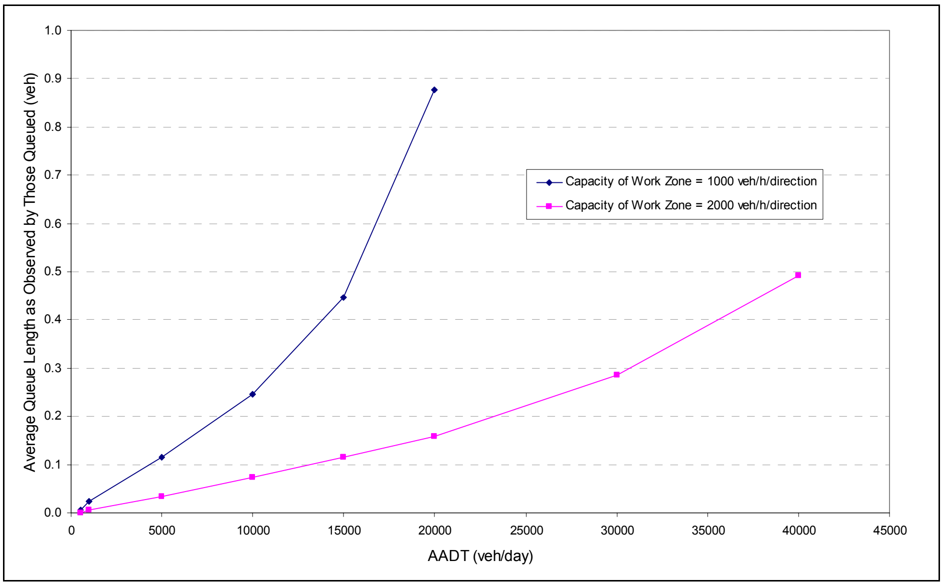


Figure B13.8: Average Queue Length versus AADT for a Work Zone with No Lane Sharing (Case 1)

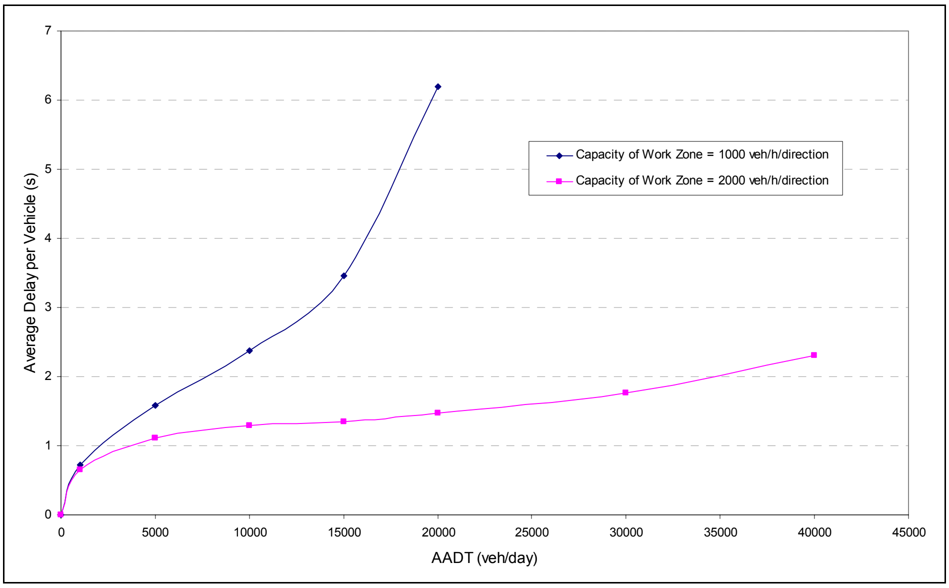


Figure B13.9: Average Delay per Vehicle versus AADT for a Work Zone with No Lane Sharing (Case 1)

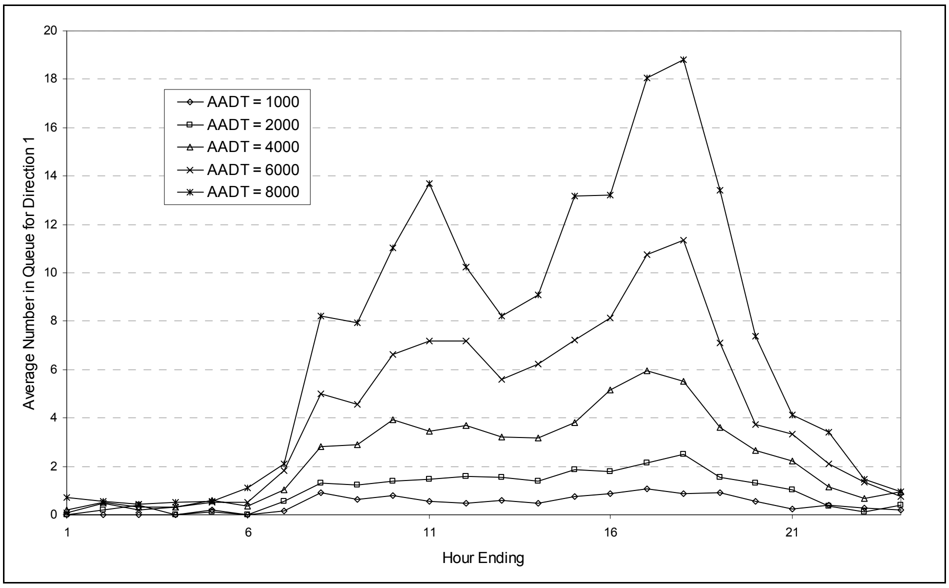


Figure B13.10: Example of Average Number of Vehicles Queued Throughout the 24 Hour Period for a Work Zone with Lane Sharing (Case 2)

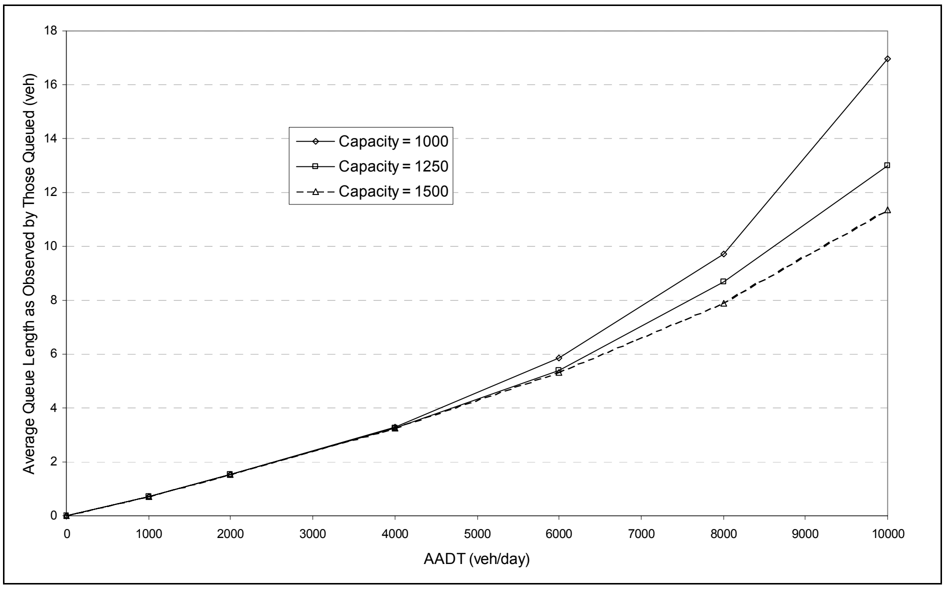


Figure B13.11: Average Queue Length versus AADT for a Work Zone with Lane Sharing (Case 2)

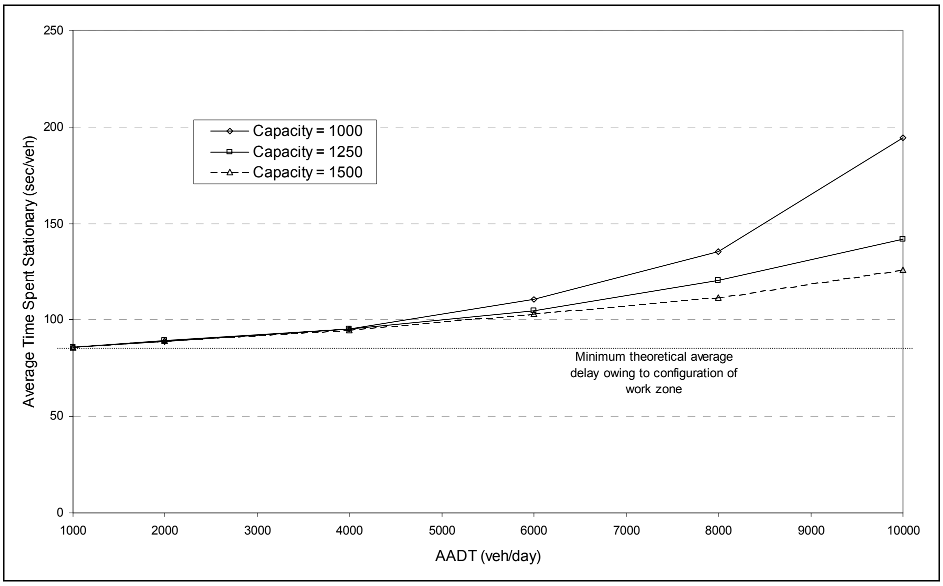


Figure B13.12: Average Delay per Vehicle versus AADT for a Work Zone with Lane Sharing (Case 2)

Figure B.13.13 though to Figure B.13.15 illustrate the impact of traffic volumes (the horizontal axis) and work zone length (the vertical axis) on the time spent stationary, time spent moving through the queue and the time spent travelling through the work zone. These values pertain only to the given set of input parameters on traffic distribution, lane capacities and work zone length used in generating the figures and should not be used as default values for the analysis of work zones.

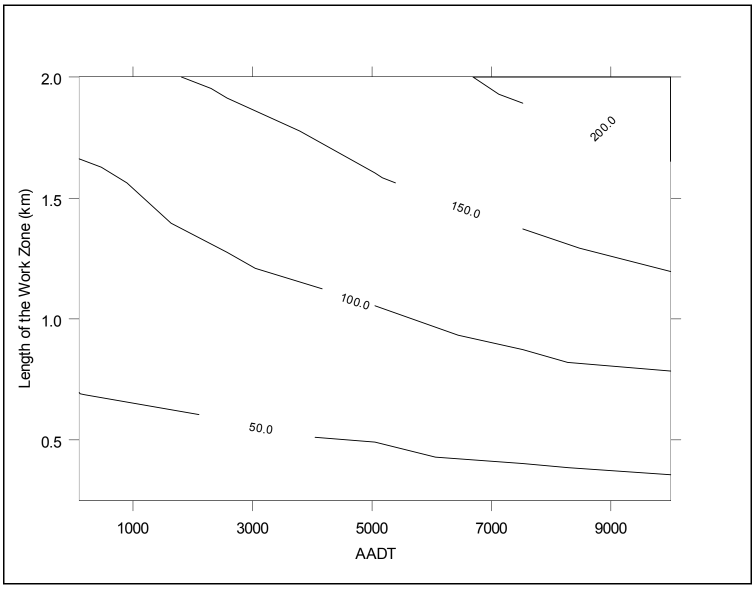


Figure B.13.13: Effect of AADT and Length of the Work Zone on the Average Time Spent Queued (Stopped)

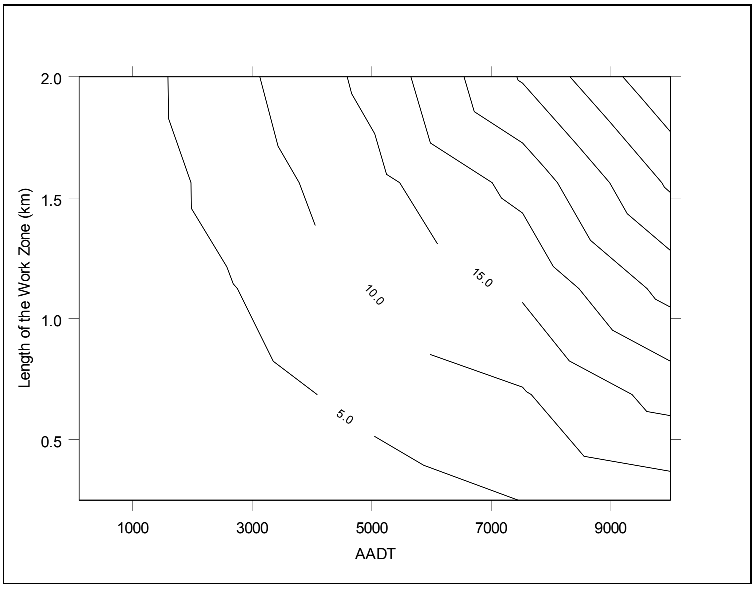


Figure B.13.14: Effect of AADT and Length of the Work Zone on the Average Time Spent Moving Through the Queue

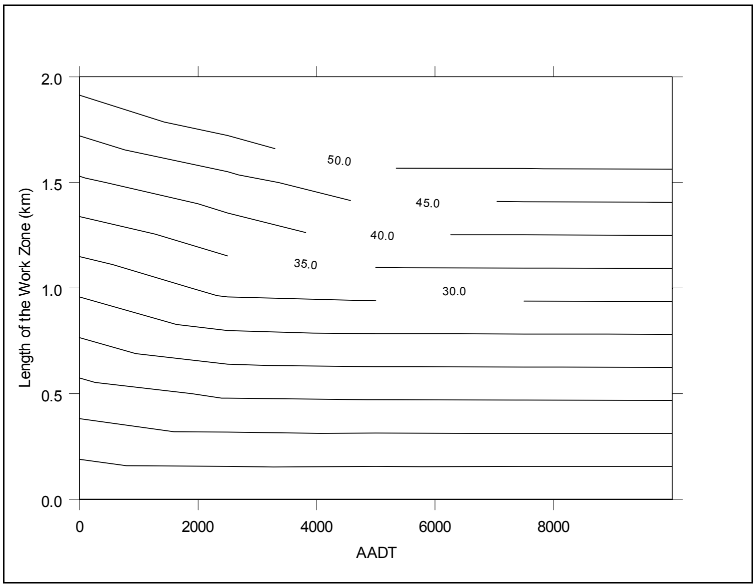


Figure B.13.15: Effect of AADT and Length of the Work Zone on the Average Time Spent Moving Through the Work Zone

B13.4 Additional Fuel and Travel Time Due To Work Zones

B13.4.1 Introduction

For the purposed of modelling vehicle operating costs in HDM-4, there are two types of traffic interactions to be considered:

- **Congestion:** The effect of increasing traffic volumes on speeds and RUE; and,
- **Work zone effects**: Specific cycles wherein a vehicle decelerates from an initial speed to a final speed or accelerates from a final speed to an initial speed.

Work zone effects differ from the congestion effects presented in Chapters B3 and B4 in that there are definitive initial and final speeds. Whereas the congestion model will be applied with most typical HDM-4 applications, the work zone model is specifically designed to work with HDM-4 for modelling the effects of work zones on speeds, although it could be used to evaluate other situations such as traffic signals or one-lane bridges.

The discussion that follows presents models for predicting the additional fuel consumption and travel time due to work zones. The approach can be applied to any other mechanistically modelled components such as tyre consumption and vehicle emissions, although at the time of writing it has not been extended or validated to these areas.

B13.4.2 Modelling Speed Work Zone Speed Change Cycles

Work zone speed change cycles arise when traffic is interrupted and forced to decelerate. An example of the type of cycle was illustrated in Figure B13.1. In Figure B13.1 vehicles are travelling at an approach speed. In advance of the work zone they are forced to decelerate. If there is a queue, they will be stationary for a time as well as moving up through the queue. In theory, as long as the capacity is finite the queue does not stop. However, since drivers do not respond in an ideal manner queuing inevitably results in stop/start conditions. Once a vehicle reaches the front of the queue it will accelerate up to the speed they will travel at through the work zone. Upon reaching the end of the work zone they will then accelerate back to their initial speed.

There are three speed change cycles in this example:

- Approach speed to stop;
- Stop to work zone speed; and,
- Work zone speed to approach speed.

There are two distinct components to be modelled in a speed change cycle: the deceleration and the acceleration phases. For HDM-4 the ARRB Polynomial Acceleration Model (Akcelik and Biggs, 1987) has been adopted. As described by Bennett (1989a), this model has a number of advantages over the others that are available.

The ARRB Polynomial Model is as follows:

$$a(t) = ar\ am\ \theta a\ (1 - \theta a^{m})^{2} \quad \ldots(B13.3)$$

where
- $a(t)$ is the acceleration at time t in m/s^2
- ar is a polynomial acceleration model parameter
- am is the maximum acceleration rate in m/s^2
- θa is the acceleration model time ratio
- m is a polynomial acceleration model parameter

The acceleration model time ratio θa is defined as:

$$\theta a = \frac{t}{ta} \quad \ldots(B13.4)$$

where t is the time in s
ta is the time to accelerate in s

Equation B13.3 satisfies two important conditions:

- Zero acceleration at the start and end of the acceleration; and,
- Zero jerk (da/dt) at the start and end of the acceleration.

Akcelik and Biggs (1987) give the following equations for applying the model:

$$A0 = 27\ \delta - 19 \qquad \ldots(B13.5)$$

$$A1 = A0 + 4 \qquad \ldots(B13.6)$$

$$A2 = 6\ \delta - 2 \qquad \ldots(B13.7)$$

$$m = \frac{-A1 + \sqrt{A1^2 - 4\ A0\ A2}}{2\ A2} \qquad \ldots(B13.8)$$

$$aav = \frac{velfin - velinit}{ta} \qquad \ldots(B13.9)$$

$$q = \frac{m^2}{(2\ m + 2)\ (m + 2)} \qquad \ldots(B13.10)$$

$$ar = \frac{(1 + 2m)^{2+1/m}}{4\ m^2} \qquad \ldots(B13.11)$$

$$am = \frac{aav}{arq} \qquad \ldots(B13.12)$$

where aav is the average acceleration in m/s^2
q is a polynomial acceleration model parameter
velfin is the final velocity in m/s
velinit is the initial velocity in m/s

The development of these equations and the model are given in the various background reports (Akcelik, *et al.*, 1983; Biggs and Akcelik, 1985).

The parameter δ is a 'shape' parameter that indicates whether the maximum acceleration arises early or late during the speed change. It can be predicted using the equation from Akcelik and Biggs (1987), or else assumed to be constant as was done by Bennett (1989a) and Bennett (1993).

Figure B13.16 is an example of the speed profile that one obtains from the Polynomial Model. The associated acceleration profile is given in Figure B13.17. These are for speed changes from 100-0-100 km/h with a value of δ = 0.5.

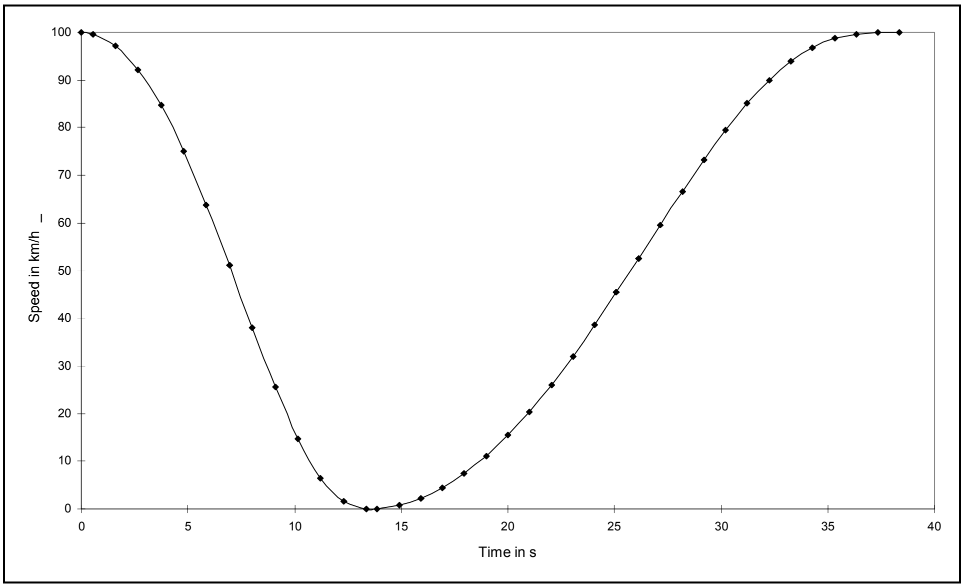


Figure B13.16: Speed-Time Profile for 100-0-100 km/h Speed Change Cycle

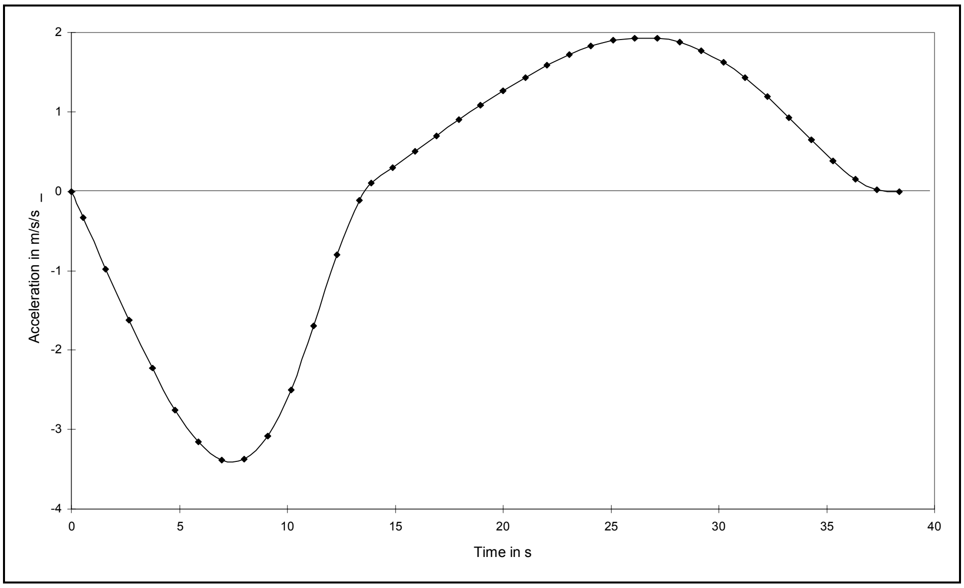


Figure B13.17: Acceleration-Time Profile for 100-0-100 km/h Speed Change Cycle

In order to apply the ARRB Polynomial Model it is necessary to have the times to accelerate and decelerate. For passenger cars the following equations from Bennett (1993) were adopted:

$$ta = \frac{\text{velfin} - \text{velinit}}{0.486 - 0.006\ \text{velinit} + 0.123\sqrt{\text{velfin} - \text{velinit}}} \qquad \ldots\text{(B13.13)}$$

$$td = \min\left(\frac{3.6\ (\text{velinit} - \text{velfin})}{1.7 + 0.238\sqrt{3.6\ (\text{velinit} - \text{velfin})} - 0.0324\ \text{velfin}}, \sqrt{193.20\left(1 - \frac{\text{velfin}}{\text{velinit}}\right)}\right)$$

$$\ldots\text{(B13.14)}$$

where ta is the time to accelerate in s
td is the time to decelerate in s
velinit is the initial velocity in m/s
velfin is the final velocity in m/s

The times to accelerate and decelerate for other vehicle classes relative to passenger cars are given in Table B13.4 (Bennett, 1993).

Table B13.4: Relative Times to Accelerate and Decelerate by Vehicle Class

Vehicle Class	Time to Accelerate/Decelerate Relative to Passenger cars (per cent)	
	Accelerate	Decelerate
Passenger Cars	100	100
Light Delivery Vehicles	115	100
Medium Trucks	120	105
Heavy Trucks	140	120
Heavy Trucks Towing	160	130
Heavy Buses	120	105

Source: Bennett (1993)

B13.4.3 Calculating Time and Vehicle Operating Costs

In order to calculate the travel time and VOC associated with speed change cycles, it is necessary to undertake an instantaneous analysis throughout each stage of the speed cycle. It was considered that this level of detail was not warranted in HDM-4. Instead, a model that was specifically developed to perform these calculations was used to generate data, which could in turn be incorporated into HDM-4.

The model used was the N.Z. Vehicle Operating Costs Model (NZVOC) and its development is described in Bennett (1989a). The version used for the analysis had some changes made and is described by Bennett (1993).

The NZVOC Model approach considers a vehicle to be travelling at a steady state speed when the travel is interrupted. This interruption consists of decelerating to a final speed and accelerating back to its original speed. If the vehicle stops, additional costs while idling are included exogenously. The additional cost due to the speed change cycle is the difference between the costs that would have arisen had the vehicle travelled over the same distance without interruption, *ie*:

$$ADDCST = (DECCST + ACCCST) - UNICST \quad \ldots(B13.15)$$

where ADDCST is the additional cost due to the speed change cycle
DECCST is the cost of decelerating from an initial to final speed
ACCCST is the cost of accelerating from the final speed back to the initial speed
UNICST is the cost of travelling the same distance at the original speed

Thus, the NZVOC model costs are related back to the distance travelled during the speed change cycle.

The analysis in Bennett (1993) resulted in a series of tables of the additional travel time (in s) and fuel consumption (in mL) for a matrix of initial and final speeds. The results are for a complete speed change cycle — deceleration and acceleration phases. However, for the purposes of HDM-4 it was considered necessary to separate out the deceleration and acceleration costs, since the acceleration phase would not always be a single continuous acceleration back to the original speed. Instead, it could consist of two accelerations: one from the queue to the work zone speed and the second from the work zone speed to the final speed (see Figure B13.1).

B13.4.4 Speed Change Cycle Costs

To establish the additional costs due to speed change cycles, the NZVOC Model was run using a special option which output the intermediate results. Tables were generated of the time and fuel used to decelerate, accelerate and to travel at the original speeds over the speed change cycle distance. The additional fuel and time for decelerations and accelerations was defined as the difference between the speed change values and the values to travel the distance at the original speed. Data were obtained for all combinations of initial and final speeds in the range of 0 to 100 km/h in 5 km/h increments.

The analysis was conducted for six vehicle classes:

- Passenger cars;
- Light commercial vehicles;
- Medium trucks;
- Heavy trucks;
- Heavy trucks towing; and,
- Heavy buses.

Subsequent to the Bennett (1993) project, the NZVOC Model was modified to include the ARRB ARFCOM fuel model which has been adapted for HDM-4. Accordingly, this analysis was done using ARFCOM, which has been adapted for HDM-4. The analysis was done for a range of subtypes within each class, operating at a range of loads, and so therefore is representative of the general traffic stream.

The data were analysed using the SPSS statistical analysis package to develop models. The models developed predict the additional time and fuel to travel the deceleration and acceleration distances over what would have arisen had those same distances been travelled at the original speed. Thus, they can be used directly to calculate the additional costs.

B13.4.5 Additional Time to Accelerate and Decelerate

The following model was developed for predicting the additional time to decelerate and accelerate over the time to travel the same distances at the original speed:

$$\text{adtimd} = \frac{3.6\ (\text{velinit} - \text{velfin})}{\text{a0} + \text{a1}\ (3.6\ \text{velinit}) + \text{a2}\ \sqrt{3.6\ (\text{velinit} - \text{velfin})}} \qquad \text{…(B13.16)}$$

$$\text{adtima} = \frac{3.6\ (\text{velfin} - \text{velinit})}{\text{a0} + \text{a1}\ (3.6\ \text{velfin}) + \text{a2}\ \sqrt{3.6\ (\text{velfin} - \text{velinit})}} \qquad \text{…(B13.17)}$$

where adtima is the additional travel time due to acceleration in s
adtimd is the additional travel time due to deceleration in s

Table B13.5 contains the values for the travel time coefficients. All coefficients were significant at 95 per cent confidence with the R^2 values all being in excess of 0.95. Figure

B13.18 is a comparison of the predictions of the deceleration equation for passenger cars compared to the original values calculated from the NZVOC Model.

Table B13.5: Parameter Values for Predicting Additional Time Due To Speed Change Cycles

Vehicle Class	Equation	a0	a1	a2	Standard Error
Passenger Cars	adtima	12.46	0.1059	-1.5102	0.29
	adtimd	9.25	0.2548	-2.0707	0.16
Light Delivery Vehicles	adtima	10.87	0.0924	-1.3175	0.33
	adtimd	9.25	0.2548	-2.0707	0.16
Medium Trucks and Medium Buses	adtima	10.42	0.0886	-1.2628	0.35
	adtimd	8.79	0.2421	-1.9672	0.17
Heavy Trucks	adtima	8.91	0.0758	-1.0806	0.40
	adtimd	7.68	0.2115	-1.7187	0.19
Articulated Trucks	adtima	7.84	0.0667	-0.9504	0.46
	adtimd	7.12	0.1962	-1.5944	0.21
Heavy Buses	adtima	10.31	0.0877	-1.2498	0.35
	adtimd	8.79	0.2421	-1.9672	0.17

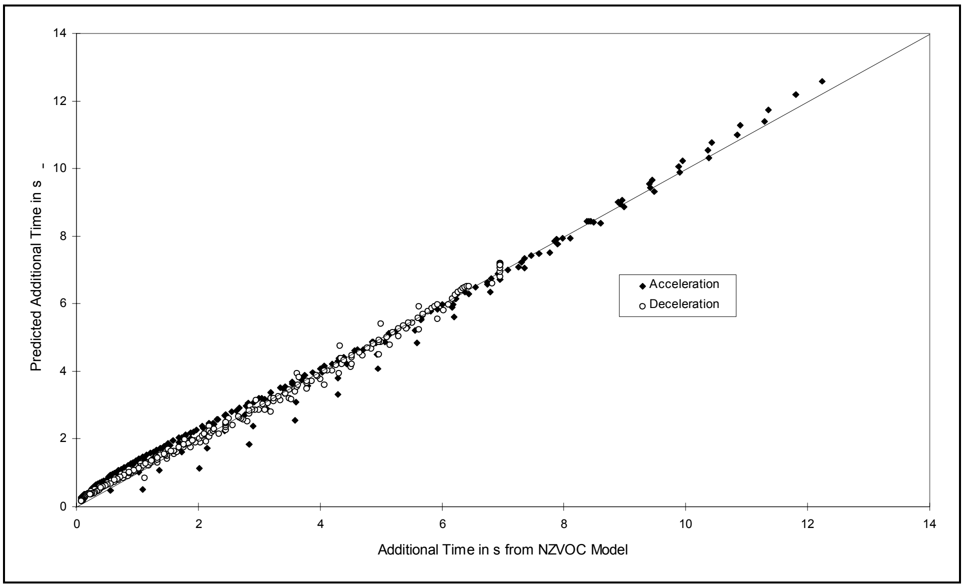


Figure B13.18: Fit of Additional Times Model for Passenger Cars

There is an inconsistency with the equations for speed changes with a final speed of 0 and initial speeds above 80 km/h. In this situation the times decrease with increasing speed. As the differences are small, it is recommended that the 80 km/h values be applied for all speeds above 80 km/h.

B13.4.6 Additional Fuel to Accelerate and Decelerate

In considering the additional fuel to accelerate and decelerate it needs to be appreciated that during the deceleration phase the vehicle is using less fuel to travel the original distance than it would have if there had been no interruption. However, when it needs to accelerate back to its original speed there is an excess of fuel used which offsets the savings during deceleration. This can be conceptualised from Figure B13.19 which shows the NZVOC fuel consumption profile for the 100-0-100 speed change cycle illustrated earlier in Figure B13.16 and Figure B13.17.

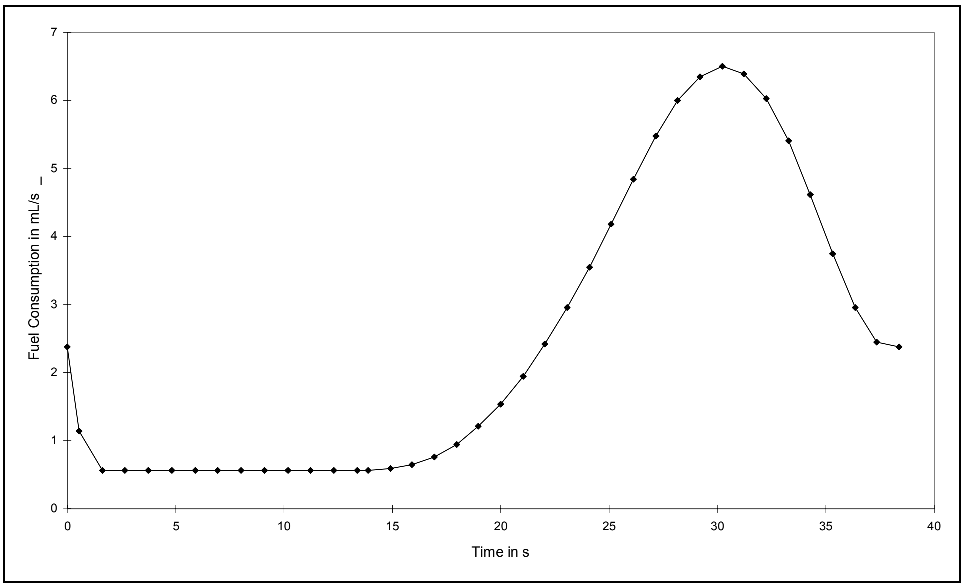


Figure B13.19: Fuel-Time Profile for 100-0-100 km/h Speed Change Cycle

A number of models were tested with the fuel consumption data. The following gave the best fit:

$$\text{adfueld} = \text{a0} + 3.6\ \text{a1}\ (\text{velinit} - \text{velfin}) + 3.6\ \text{a2}\ \text{velinit}\ \sqrt{3.6\ (\text{velinit} - \text{velfin})} \qquad \dots(\text{B13.18})$$

$$\text{adfuela} = \text{a0} + 3.6\ \text{a1}\ (\text{velfin} - \text{velinit}) + 3.6\ \text{a2}\ \text{velfin}\ \sqrt{3.6\ (\text{velfin} - \text{velinit})} \qquad \dots(\text{B13.19})$$

where adfueld is the additional fuel to decelerate in mL
adfuela is the additional fuel to accelerate in mL

Table B13.6 presents the parameter values for use with the above equations. All equations had statistically significant coefficients at 95 per cent confidence and coefficients of determination of 0.98 or better. NDLI (1995) show that the equations give a good fit of the NZVOC data.

B13.5 Predicting Work Zone Effects in HDM-4

On the basis of the methodology presented above, to calculate work zone costs HDM-4 requires three data items:

- The number of vehicles queued;
- The time stopped; and,
- The additional time moving through the queue.

Table B13.6: Parameter Values for Predicting Additional Fuel Due To Speed Change Cycles

Vehicle Class	Equation	a0	a1	a2	Standard Error
Passenger Cars	adfuela	-0.1470	-0.1524	0.0854	0.83
	adfueld	0.0202	0.2178	-0.0306	0.28
Light Delivery Vehicles	adfuela	-0.1727	-0.2210	0.1012	1.32
	adfueld	0.0233	0.2516	-0.0399	0.48
Medium Trucks and Medium	adfuela	-0.4314	-0.3485	0.2733	2.19
Buses	adfueld	0.0934	0.6478	-0.1021	1.44
Heavy Trucks	adfuela	-0.9874	-0.8960	0.6117	5.02
	adfueld	0.1628	1.1913	-0.1877	2.51
Articulated Trucks	adfuela	-2.2440	-2.3980	1.3578	12.20
	adfueld	0.1648	1.5614	-0.2455	2.97
Heavy Buses	adfuela	-0.6067	-0.4904	0.3837	3.03
	adfueld	0.1349	0.9314	-0.1463	2.04

These are a function of the traffic volume (AADT), the work zone length, work zone capacity, work zone speed, and the arrival pattern.

For HDM-4, the ROADWORK simulation model is used to generate these values for a range of traffic and work zone conditions. The model output is then read by HDM-4 and used to estimate the additional time and fuel consumption due to the work zone.

In the HDM-4 analysis, two sets of vehicles are considered: those who are forced to stop and queue and those which travel through the work zone without stopping. Each of these are treated separately. In both cases, the additional travel time and fuel are calculated.

Figure B13.20 shows the analysis procedure for vehicles which are forced to stop and queue. Annex B.13.2 gives the equations used in HDM-4 to calculate the various elements of this figure. It will be noted that there are four components to the analysis:

- Decelerating from approach speed to stop;
- While queued or stopped;
- Travel from stop through work zone; and,
- Accelerating from work zone to approach speed.

For vehicles which do not queue, the same components are analysed, except for the queued/stopped component which does not apply.

The total work zone effects are calculated as the sum of costs for the queued and not queued vehicles, weighted by their relative frequency in the traffic stream.

B13.6 Summary

Work zones result in a disruption to traffic. Vehicles are forced to decelerate and, if there is inadequate capacity through the work zone, stop and queue. They then will travel through the work zone at a lower speed than their approach speed before returning to their original speed once past the work zone.

This chapter has presented the logic for modelling the effects of road works on road users for HDM-4. A simulation program called ROADWORK has been developed which outputs the speed and queuing characteristics for a number of combinations of lane configurations and traffic volumes. The model predicts that capacity has a significant impact on the queue length and delay. Examples of the simulation output were given for two different types of work zones.

A methodology has also been presented to cost the additional time and fuel consumption associated with the speed change cycles encountered at road works. This methodology is equally applicable to costing the speed change cycles associated with traffic signals or other such locations where drivers make deliberate speed changes. For the random fluctuations in speed caused by general traffic congestion, please refer to Chapters B3 and B4.

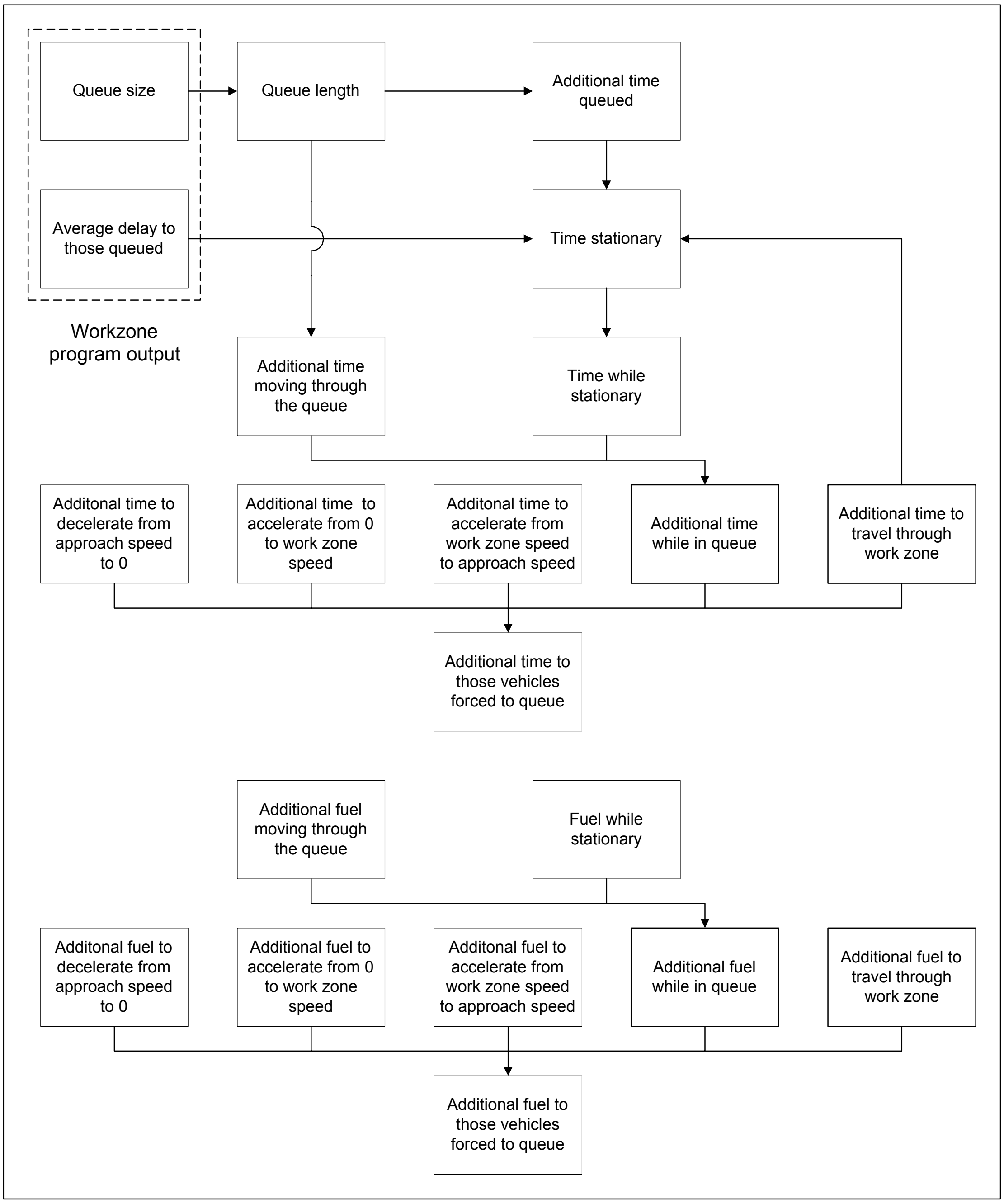


Figure B13.20: Work Zone Analysis Procedure: Queued Vehicles

B14 Heavy Vehicle Trailers

B14.1 Introduction

In many countries heavy vehicles tow trailers so the RUE need to be calculated for both the towing vehicle and the trailer. For HDM-III trailer costs were only considered with the articulated truck representative vehicle. It was not, for example, possible to associate trailers with rigid trucks, nor to have multiple trailers with a single towing vehicle. To circumvent this problem, for HDM-4 trailers are considered as separate entities from the towing vehicle. It is therefore possible to associate any trailer with a towing vehicle, although only with heavy vehicles since light vehicle trailers are not considered to have a significant enough level of RUE to warrant considering.

As shown in Table B14.1, the RUE components which are predicted specifically for trailers are parts, tyres, capital costs and utilisation. The presence of trailers also influences the speed, tyre, fuel and oil consumptions of the towing vehicle. Of these, the trailer maintenance and repair costs are by far the greatest component, followed by tyres and fuel.

Table B14.1: RUE Components for Trailers and Vehicles

Component	Trailer	Vehicle
Maintenance and Repairs	•	
Tyre Consumption	•	•
Capital Costs and Utilisation	•	
Speed		•
Fuel Consumption		•
Oil Consumption		•

The discussion which follows opens with trailer maintenance and repair costs, and how this is predicted in HDM-4. This is then followed by the influence of trailers on mechanistic forces since this impacts on the speed, tyre, fuel and oil consumption.

B14.2 Trailer Maintenance and Repairs

In most studies the trailer maintenance and repair costs were grouped with the towing vehicle to obtain the total costs. The only major exception to this was a recent N.Z. study (Opus-Beca, 1998) and the results of this study have been used to develop the HDM-4 trailer model.

Opus-Beca (1998) found that the parts consumption could be expressed as a constant percentage of the new vehicle price, irrespective of the trailer age. The resulting model was:

$$PC = 0.22 \qquad \ldots(B14.1)$$

where PC is the parts consumption as a percentage of the new vehicle price per 1000 km

The parts cost is therefore calculated as:

$$PCOST = 0.0022\ NVPLT \qquad \ldots(B14.2)$$

where PCOST is the parts cost per 1000 km
NVPLT is the trailer replacement value, less tyres

The total maintenance and repair costs were comprised of 44 per cent parts; 56 per cent labour. This corresponds to:

$$LCOST = 1.27\ PCOST \qquad \text{…(B14.3)}$$

where LCOST is the labour cost per 1000 km

It can be anticipated that there will be an impact of roughness on the maintenance costs, although the low roughnesses in N.Z. did not allow for this to be established. Assuming that there is a linear increase in the parts costs above a roughness of 3 IRI leads to the following general equation:

$$PC = \left(0.22 - RImin\ \frac{dPC}{dRI}\right) + \left(\frac{dPC}{dRI}\ RI\right) \qquad \text{…(B14.4)}$$

where dPC is the change in parts consumption as a decimal
dRI is the range of roughness over which the change in parts arises
RI is the adjusted roughness
RImin is the minimum adjusted roughness (default = 3 IRI m/km)

On the basis of truck parts consumption, it was assumed that there would be a 20 per cent increase between the roughnesses of 3 and 10. This leads to the following equation:

$$PC = 0.2011 + 0.0063\ RI \qquad \text{…(B14.5)}$$

The total maintenance and repair costs are therefore:

$$PCOST = (0.2011 + 0.0063\ RI)\ NVPLT \qquad \text{…(B14.6)}$$

$$LCOST = 1.27\ PCOST$$

or,

$$LCOST = (0.2554 + 0.0080\ RI)\ NVPLT \qquad \text{…(B14.7)}$$

HDM-4 expresses the labour cost as the product of the number of hours and the hourly cost of labour. Since we are calculating the total labour cost, we back-calculate the labour hours as:

$$LH = \frac{LCOST}{CST\_LABOUR} \qquad \text{…(B14.8)}$$

or,

$$LH = \frac{1.27\ PCOST}{CST\_LABOUR} \qquad \text{…(B14.9)}$$

For incorporating into HDM-4 the following general model form was adopted:

$$PC = K0pc\ [a0 + a1\ RI] + K1pc \qquad \text{…(B14.10)}$$

$$LH = K0lh\ [a2\ \frac{PC\ NVPLT}{CST\_LABOUR}] + K1lh \qquad \text{…(B14.11)}$$

$$PCOST = PC\ NVPLT \qquad \text{…(B14.12)}$$

$$LCOST = LH\ CST\_LABOUR \qquad \ldots(B14.13)$$

where	a0	is the constant in the parts consumption model (default = 0.2011)
	a1	is the roughness slope in the parts consumption model (default= 0.0063)
	a2	is the labour cost factor (default = 1.27)
	LH	is the number of labour hours
	CST_LABOUR	is the hourly cost of maintenance labour
	K0pc, K0lh	is the rotation calibration factor (default = 1)
	K1pc, K1lh	is the translation calibration factor (default = 0)

Since the above model is linear, it is strictly not necessary to have both translation and rotation calibration factors. However, these have been included in the event that a more sophisticated model is one day developed.

In applying the above equations they are only used for a vehicle which is towing a trailer. The trailer costs are already considered with the articulated truck.

B14.3 Effects of Trailers on Mechanistic Forces

Trailers will lead to additional forces opposing motion. With reference to Chapter B1, trailers influence the aerodynamic, rolling, gravitational, curve and inertial resistances. Each of these effects are addressed below.

B14.3.1 Aerodynamic Resistance

Using the methodology discussed in Chapter B1, it was found that the CD multiplier (CDmult) should be set to 1.22 for all vehicles.

The projected frontal area will be the maximum of that for the vehicle and trailer, *ie*:

$$AF = \max(AFveh, AFtrl) \qquad \ldots(B14.14)$$

where	AF	is the projected frontal area in m^2
	AFveh	is the projected frontal area of the towing vehicle in m^2
	AFtrl	is the projected frontal area of the trailer in m^2

No data were found for the drag coefficients for vehicles with trailers versus without. In the absence of any other data it was assumed that the presence of a trailer leads to a 10 per cent increase in the aerodynamic drag coefficient, *ie*:

$$CDvtrl = 1.1\ Cdveh \qquad \ldots(B14.15)$$

where	CDvtrl	is the truck and trailer aerodynamic drag coefficient
	CDveh	is the truck aerodynamic drag coefficient without the trailer

B14.3.2 Rolling Resistance

For the purposes of modelling rolling resistance, the vehicle is considered separately to the trailer. The total rolling resistance is therefore:

$$Fr = Frveh + Frtrl \qquad \ldots(B14.16)$$

where	Frveh	is the rolling resistance of the towing vehicle
	Frtrl	is the rolling resistance of the trailer

This approach has the advantage in that it allows the properties of the trailer tyres to differ from those of the towing vehicle, thereby ensuring improved predictions.

As discussed in Chapter B1, the HDM-4 rolling resistance equation is:

$$Fr = CR2\ (1 + 0.003\ PCTDS + 0.002\ PCTW)\ (b11\ Nw + CR1(b12\ M + b13\ v^2))$$

The same equation is used substituting in the number of wheels, wheel diameter and type of tyre for the trailers. From this, the trailer rolling resistance (Frtrl) is established. As described above, this is then added to the vehicle rolling resistance to get the total rolling resistance.

B14.3.3 Gradient and Curve Resisances

The gradient and curve resistances are a function of the total mass of the vehicle and trailer, *ie*:

$$M = Mveh + Mtrl \qquad \text{...(B14.17)}$$

where Mveh is the vehicle operating mass in kg
Mtrl is the trailer operating mass in kg

The curvature resistance also depends upon the tyre stiffness coefficient which in turn is a function of the mass and number of wheels. For simplicity of modelling it is assumed that the trailer tyre stiffness is the same as that for the towing vehicle. The same equations as with vehicles are used, but the input data consist of the total vehicle mass and the total number of wheels on the vehicle and trailer:

$$Nw = Nwveh + Nwtrl \qquad \text{...(B14.18)}$$

where Nwveh is the number of wheels on the vehicle
Nwtrl is the number of wheels on the trailer

B14.3.4 Inertial Effects

The inertial effects are calculated just as with the vehicle, but total mass of the vehicle and trailer are used instead of the vehicle mass alone.

B14.4 Speed

The same speed prediction model parameters used for the vehicle are used when it also has a trailer. The predicted speed will be lower since the mass will be that of the vehicle and trailer.

B14.5 Fuel Consumption

As described in Chapter B4, the fuel consumption is predicted in HDM-4 as:

$$IFC = \xi\ Ptot$$

where IFC is the fuel consumption in mL/s
ξ is the fuel conversion factor in mL kW/s
Ptot is the total power requirements in kW

The total power is calculated using the total forces for the vehicle and trailer combination. From this, the fuel consumption is established.

B14.6 Tyre Consumption

Since the vehicle and trailer may have different tyre types, the tyre consumption is calculated separately for the vehicle and trailer and then combined to give the total tyre consumption. The calculations are done using the standard HDM-4 methodology, with the tyre attributes for the trailer potentially different to those for the vehicle.

B14.7 Capital Costs and Utilisation

The annual utilisation will be assumed to be the same as that for the vehicle. Similarly, the reduction in service life due to roughness will be assumed to be the same as that for the vehicle.

The capital costs for the are calculated using the same technique as with the vehicle. The total capital costs are the sum of the two.

C1 Vehicle Emissions

C1.1 Introduction

Vehicles emit various chemical compounds as a direct result of the combustion process. The type and quantity of these emissions depends on a variety of factors including the tuning of the engine, fuel type and driving conditions. When dealing with vehicle emissions, researchers focus primarily on the following substances: Hydrocarbons (HC), Carbon Monoxide (CO), Carbon Dioxide (CO_2), Nitric Oxides (NOx), Sulphur Dioxide (SO_2), Lead (Pb) and particulate matter (PM). These compounds are considered to not only form the majority of the emissions, but also form the most damaging to the natural environment and human health.

Heywood (1997) states that in the United States, vehicles are estimated to produce about 40-50 per cent of the HC, 50 per cent of the NOx and 80-90 per cent of the CO emissions in urban areas. The effect of these emissions both locally and globally is of growing concern, and thus it is imperative that in the evaluation of roading options, the effect on vehicle emissions can be accurately modelled.

Although not a primary concern of this research, the above emissions are reported to create or add to the environment problems in Table C1.1 (EPA, 1997). Patel (1996) estimates that around 40,000 deaths per year in India’s major cities are caused by air pollution.

Table C1.1: Emissions and their associated Problems

Emission	Problems Caused by Pollutant
Hydrocarbons	Urban ozone (smog) and air toxics
Carbon Monoxide	Poisonous gas
Nitrogen Oxides	Urban ozone (smog) and acid rain
Carbon Dioxide	Global warming

Source: EPA (1997)

Paterson and Henein (1972) quoted Dr John Middleton as stating in 1968:

> *"The day may soon come – if it's not already here – in which the individual automobile can no longer be tolerated as a convenient form of transportation, simply because of its adverse effects on the health of people, not just the aesthetic of the atmosphere."*

In response to this issue, which is just as strong today, many countries have signed agreements to reduce their total emission levels, with particular focus on those emissions coming from motor vehicles. Since the 1960s when the above quote was taken, the desire to improve the emission output of vehicles has resulted in substantial reductions. Heywood (1997) shows that the average production of vehicle emissions in modern vehicles is around 2 - 10 per cent of the levels recorded in the 1960s prior to emission controls. This reduction is in response to ever-stricter standards placed on vehicle manufactures, which have responded with technology such as catalytic converters and cleaner burning fuels.

Heywood (1997) states that *"it sounds too simple to say half the total fleets' CO and HC emissions come form the worst 10 percent of the vehicles on the road, but it is indeed true!"* For this to be the case, there is clearly a significant variation in the quantity of emissions coming out of seemingly similar vehicles. Any predictive models will therefore need to either recognise this distribution, or clearly be used to predict average conditions only.

For estimating regional or national emissions the use of average emission rates is quite acceptable, and, owing to their generally simplified mathematical form, even preferable for such tasks. The USA Environmental Protection Agency (EPA) gives the following average emission values for a passenger car and a light truck (EPA, 1997). The data in Table C1.2 is based on an annual usage of 17,700 km (11,000 miles) and 22,500 km (14,000 miles) for the passenger car and light truck respectively.

Table C1.2: Annual Emissions and Fuel Consumption for Passenger Car and Light Truck

Vehicle	Pollutant	Average Amount[1] g/km	Average/year[1] kg/year
Passenger Car	Hydrocarbons	1.9	34.0
	Carbon Monoxide	14.3	262.7
	Nitrogen Oxides	1.0	17.7
	Carbon Dioxide	225.5	3992.0
	Gasoline	88	1561.3
Light Truck	Hydrocarbons	2.5	57.2
	Carbon Monoxide	19.9	447.7
	Nitrogen Oxides	1.2	28.1
	Carbon Dioxide	338.3	7620.5
	Gasoline	123.5	2782.0

Source: EPA (1997)

Notes: 1. Values converted from g/mile and lbs/year.
2. Conversion rates of 1 gallon = 3.785 litres and a specific gravity of 0.75 were used

Unlike many commodities, such as computers, where life is measured in months, motor vehicles are durable with lives often greater than 20 years. This long life has a marked impact on the overall effectiveness of new technologies. Typically less than 10 per cent (LTSA, 2000 and BTCE, 1996) of all cars are newly registered each year. As noted in BTCE (1996), any innovation in vehicle technology which improves the fuel or emission performance of new vehicles will therefore take several years to affect a sizeable proportion of the total fleet.

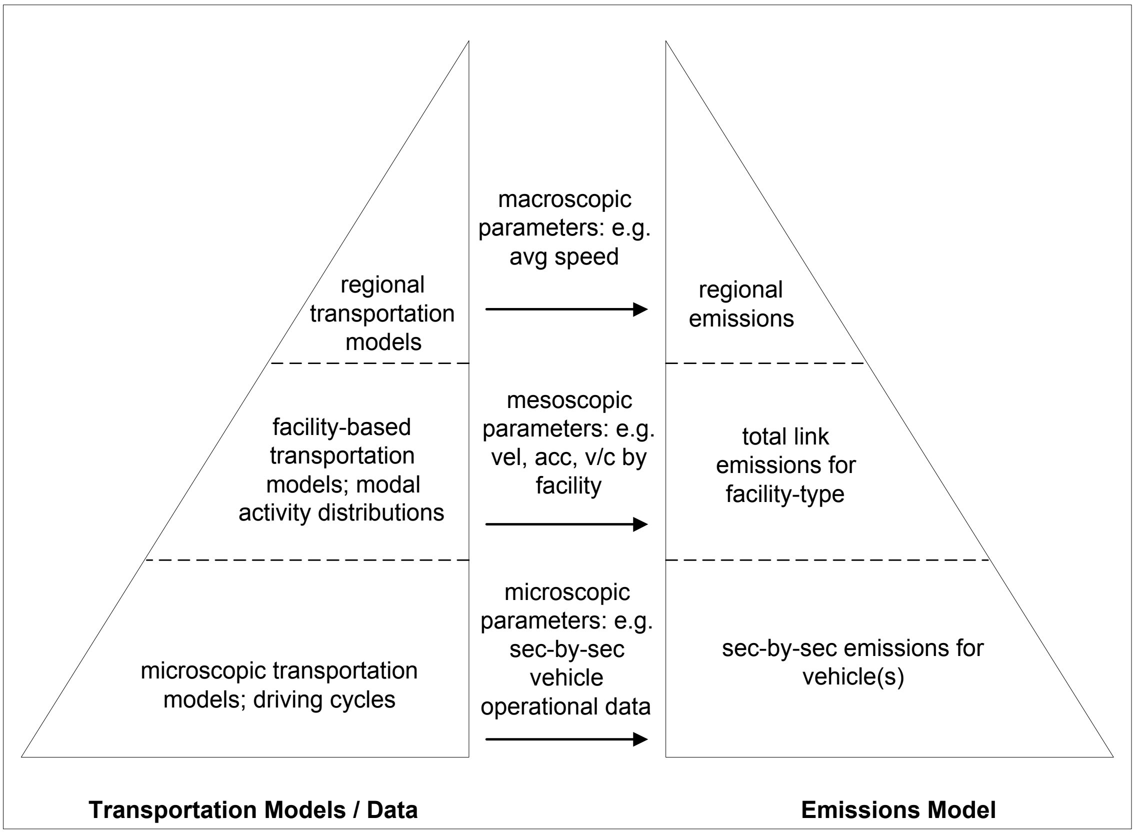
It is noted that some cahracteristics that differentiate between old and new technology, such as fuel efficiency and tyre rolling resistance, can already be addressed within the current modelling framework through the appropriate selection of parameter values.

Assuming that a sudden improvement in efficiency of 50 per cent was achieved for new vehicles, then after one year the overall impact on the total fleet would be less 5 per cent. The up side of this long gestation period, is that a greater level of stability in fleet characteristics can be assumed by the modeller.

C1.2 Modelling Emissions in HDM-4

As with most items that are modelled, a range of levels of sophistication is available for the modelling of vehicle emissions. An, *et al. (*1997) present the following diagram that relates the type of emission model to various levels of transportation models. HDM-4 can be best equated to the mesoscopic/macroscopic level, compared to the microscopic simulation modelling within the HDM Tools ACCFUEL[1] program. The latter sees actual drive cycles simulated compared to aggregate predictions of driver behaviour.

[1] ACCFUEL is one of the HDM Tools suite of software programs. It is used to simulate vehicle accelerations for estimating the effects of speed cycles on fuel and tyre consumption.



Source: An, et al. (1997)

Figure C1.1: Transportation/Emission Model Interface

By considering the required level of sophistication required in the output, one can set about collecting the appropriate input parameters. For instance, when evaluating two broad investment regimes, the collection and calibration of second by second data will generally be unwarranted, in which case the use of emission rates based on road link parameters will be sufficient. However, when evaluating various widening scenarios for capacity improvement projects, the use of second by second data should be considered.

C1.3 Factors Influencing Emissions

C1.3.1 Petrol versus Diesel Engines

There is a significant variation in the quantity of the various emissions produced between petrol and diesel engines. These variations arise as a consequence of the differences in both the raw fuel and the means by which it is combusted.

Petrol engines operate at significantly different temperatures and pressures to diesel engines, which in turn influences the production of a number of vehicle emissions. In addition to this variation because of the combustion methods, the chemical makeup of petrol and diesel are also different.

The models presented in the remainder of this chapter have been generated to enable the same basic model from to represent both petrol and diesel emissions. The consequence of this is that in some locations values of zero (or one) are used to effectively turn models 'off' in situaitons where they would be inappropriate.

C1.3.2 Hot versus Cold Emissions

One of the major influences on the level of emissions produced from vehicles is that of engine temperature. Once an engine is warm, its efficiency improves resulting in a cleaner burn of the fuel. Catalytic converters also require a high temperature (300 – 500^{o}C) to operate efficiently, although current research is aimed at reducing the lower end of operating conditions. The initial start-up phase of an engine therefore produces the most emissions on a unit basis and has the lowest ability to treat these in the catalytic converter. The SNRA (1995) estimate the proportion of emissions from cold start in Swedish cars as:

- 18 per cent of HC;
- 37 per cent of CO; and,
- 3 per cent of NOx.

In testing vehicles for the effect of cold starts the EFRU (1997) established the ratio of emissions during the first three minutes to those once the engine is warm (after six minutes of operation). The ratio is equivalent to the ratio of cold start to hot engine emissions and are given below for petrol fuel injected engines with catalytic converters:

- CO = 6.2;
- HC = 8.0; and,
- NOx = 1.7.

It was noted that for non-catalyst equipped vehicles, emissions of NOx increase as the engine warms owing to the enrichment of the mixture during startup suppresses the formation of NOx (EFRU, 1997). For vehicles with catalysts, the efficiency of the catalyst improves at a greater rate than the increase in engine out emissions[1], thus yielding the above results.

For HDM-4, the length of the journey time is not known and hence the composition of engines running hot and cold is also unknown. Given that this composition would be largely unaffected by most road investment projects—except where there is likely to be significant route shortening — the HDM-4 emissions models are based on hot engine emissions. The peculiarity of cold engine emissions and evaporation (refer next section), although often significant quantities in themselves, are specifically excluded from HDM-4. If cold engine emissions are considered important in a project then the project should be analysed using a dedicated emissions model, such as Mobile 5 produced by the USA EPA.

C1.3.3 Evaporation

When considering vehicle emissions, not all of the pollutants are emitted from the vehicles exhaust. As is readily observed when petrol is spilt on the ground, it quickly evaporates into the atmosphere. A similar process occurs within vehicles, whereby the production of HC from evaporation is often significant.

The rate of evaporation of fuel is directly related to two items:

- The air temperature; and,
- The level of vapour pressure in the fuel.

[1] Engine out emissions equate to the quantity of the various compounds being emitted from the engine, prior to treatment by the catalytic converter if present.

To minimise evaporation, fuel companies have the ability to alter the vapour pressure level as the ambient temperature alters during the year. SNRA (1995) indicate that this is standard practice in Sweden, where a lower vapour pressure than in winter is utilised during the warm summer period.

Evaporation is primarily a concern for the production of HC, with the SNRA (1995) estimating that evaporation accounts for approximately 43 per cent of the total production. The proportion of evaporation from the various modes is estimated as follows (SNRA, 1995):

- Diurnal (losses during the day) 45 per cent;
- Running losses (losses while vehicle is in operation) 33 per cent; and,
- Hot soak (during cool down period after use) 22 per cent.

It is interesting to note, that a significant proportion of HC production takes place in parked vehicles. As with cold start conditions, the impact of road investment alternatives on evaporation is minimal, and hence it is considered appropriate to exclude the modelling of this component from a model such as HDM-4.

C1.3.4 Effect of Legislation

Unlike fuel consumption, the level of emissions is closely regulated in many countries of the world. Where emissions legislation is enforced, in spite of differences in the physical characteristics of the vehicles, the level of emissions exiting the tailpipe tend to be very uniform—dictated by the legislation. Consequently, vehicles that may have differing fuel consumption levels for a given power demand, will often have similar emission rates with the use of appropriate catalytic converters.

C1.3.5 Two Stroke Engines

Two stroke engines are considered to be cheap to run – as is evident by their high representation in the vehicle mix in most developing nations of the world. However, despite being using relatively little fuel, what fuel they do consume is not utilised efficiently. In particular, there is a high percentage of the fuel that bypasses the combustion process completely and simply exits the exhaust pipe as a vapour. Sheaffer, *et al. (*1998) state that as much as 30 to 40 per cent of the air-fuel mixture at wide open throttle is short circuited unburned to the atmosphere.

Catalysts for two stroke engines are not common, but can achieve reductions in CO of 94 per cent and HC of 90 per cent. Such potential reductions, in combination with legislation may well see dramatic changes in the emissions from two stroke engines in the near future.

NOx production from two stroke engines is in general very low owing to the lack of either high temperatures or high engine pressures.

C1.4 Predictive Models

Predicting vehicle emissions has proved to be more difficult than that of fuel consumption owing to the greater variability in results. Factors such as ambient air temperature and journey length (*ie* proportion of trip with hot engines) play pivotal roles in determining emissions, yet are largely beyond the control of vehicle manufacturers and drivers. Vehicles which appear to be the same can also have vastly different emission output levels.

For HDM-4, it was necessary to provide models that could be used to evaluate one road project against another, as opposed to models that can evaluate one vehicle relative to another. Thus, models that predict average emission levels for each vehicle type were adopted.

To enable comparisons of emissions from one vehicle to another, manufactures and researchers have resorted to the use of typical drive cycles. These cycles are deemed to represent the various types of driving undertaken either in urban or rural conditions. Although these results are of use in comparing one vehicle with another under 'average' conditions, they do not enable one roading option to be compared with another, as only those options resulting in a shortened route would show any benefits.

The EPA (1997) state that 18 per cent of on-road driving conditions are not modelled within the standard FTP (Federal Test Procedure) drive cycles, therefore even questioning the reliability of using such results in the comparison of one vehicle to another. In particular the FTP drive cycle is not considered to account for the following driving conditions (EPA, 1997):

- High engine loads;
- Hard accelerations;
- Climbing grades;
- Warm starts; and,
- Accessory use.

As emissions typically increase under high loading conditions, the results produced from the test cycles could be considered as lower bounds of the on-road results.

For use in HDM analyses, it is necessary to be able to compare the average level of vehicle emissions under the different operating conditions. To do this, a relationship is required to predict emissions either as a function of operating conditions (road geometry, roughness, vehicle speed and total power), or as a function of another variable (such as fuel consumption).

Many vehicle emissions can be expected to have a high degree of correlation with fuel consumption owing to fuel being one of the primary agents in the combustion process. In support of this approach, An, *et al*. (1997) states *"analysis indicates a strong correlation between fuel use and engine-out emissions under specific conditions"*.

The SNRA produced a range of parameters and statistics for a simple linear model between fuel consumption and the various emissions. The SNRA (1995) state that the R-squared values "*could be considered high enough to accept a simple model describing exhaust emissions as linear functions of fuel consumption.*" The SNRA (1995) form of model was coded in Release 1 of HDM-4.

ETSU (1997) state that *"The emissions of nitrogen oxides do not depend on fuel consumption, however, and are more directly linked to engine speed and temperature of combustion."* They then go on to state *"The disadvantage of the fuel consumption model* [to model emissions] *is that in several cases the emission of a specific pollutant is not physically dependent on the level of fuel consumed."* ETSU (1997) present the following equation form for predicting the vehicle speed dependent emissions:

$$EOE_i = a1 + a2\,S + a3\,S^{a4} + a5\,e^{a6\,S} \qquad \text{...(C1.1)}$$

where	EOE_i	is the engine out emission in g/km for emission i
	i	is the emission type (*ie* HC, CO, NOx, SO_2, Pb, PM or CO_2)
	a1 to a6	are model parameters varying by emission type and vehicle
	S	is the vehicle speed in km/h

ETSU (1997) defined 64 sets of parameters for Equation C1.1 , however, only one utilised the exponential term. Therefore, the model could well be simplified to exclude the final component. This simplified model form is similar to the early emperical fuel consumption models discussed in Section B4. Therefore, although the ETSU (1997) state that the emission of some pollutants are not related to fuel, the form of the predictive models tends to suggest that a moderate level of correlation would in fact exist—supporting the findings of other researchers.

In using any model it is necessary to consider the transportability of the results to differing vehicles and/or countries. To this end ETSU (1997) state *"the data for deriving the empirical relationships between fuel consumption and pollutant emissions have been measured by laboratories in the developed world. It is not clear how suitable these relationships would be under different climatic conditions, with different vehicle types and levels of maintenance."*

ETSU (1997) recommend the use of what they term the "Detailed Traffic Flow/Engine Power Calculations" method. This method uses detailed information from a range of driving conditions, including parking manoeuvres and the like. However, owing to the data requirements being beyond those appropriate for HDM-4, ETSU (1997) state that this would *"undermine the accuracy of the method".* Therefore it is considered that the use of such models are inappropriate for use within HDM-4 and where such information is available then consideration should be given to a dedicated emission model such as Mobile5.

Based on the above work, it was concluded that the most appropriate emissions model to adopt for HDM-4 was one which predicted most emissions as a function of fuel. The exception was Carbon Dioxide which is modelled as a Carbon balance equation, wherein any Carbon not consumed by the other emissions, by default is emitted as CO_2. The method of modelling each of the emissions is shown in Table C1.3.

The remainder of this chapter presents the various model forms and parameter values required to estimate the above emissions.

Table C1.3: Selection of Estimation Method by Emission

Emission	Fuel Consumption	Carbon Balance
Hydrocarbons (HC)	✔	
Carbon Monoxide (CO)	✔	
Nitrogen Oxides (NOx)	✔	
Particulates (PM)	✔	
Carbon Dioxide (CO_2)		✔
Sulphur Dioxide (SO_2)	✔	
Lead (Pd)	✔	

C1.5 HDM-4 Emissions Model Form

The HDM-4 modelling approach has been adapted from that proposed by An, *et al. (*1997) to include the two model forms presented above. For this report, all models have been presented such as they predict emissions in g/km, where this is not the format of the models in their original formulation, appropriate conversions have been made. The basic model form is:

$$TPE_i = EOE_i\ CPF_I \qquad \text{...(C1.2)}$$

where TPE_i are the Tailpipe Emissions in g/km for emission i

CPF$_i$ is the Catalyst Pass Fraction for emission i

Diagrammatically this can be represented as shown in Figure C1.2, which is a modification of a similar plot presented by An, *et al.* (1997). In essence, the engine out emissions are estimated based on the fuel consumption rates, with CO_2 being calculated from Carbon balance assumptions. These engine out emissions are then treated by the catalytic converter, if present, to yield the tailpipe emissions observed by the environment.

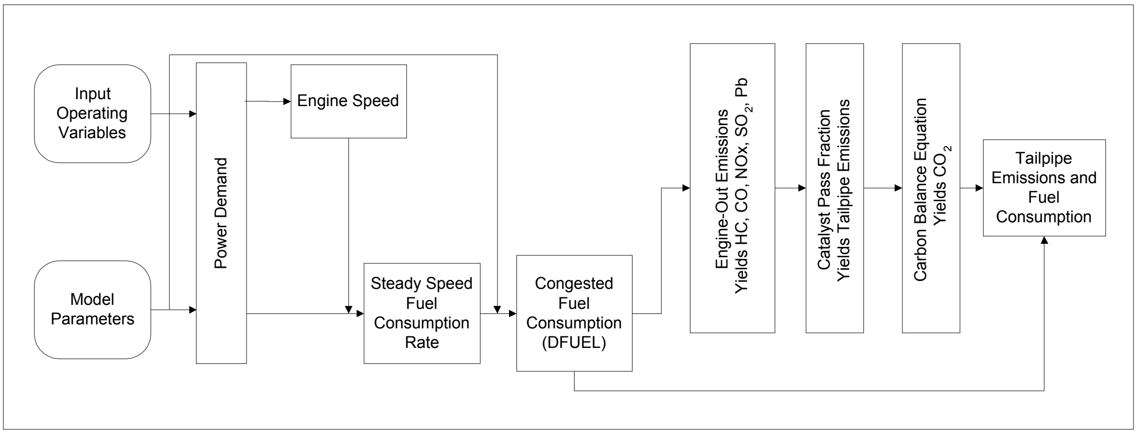


Figure C1.2: Calculation of Vehicle Emissions

The next section presents models for predicting the various engine out emissions that are related to fuel consumption. It then provides functions for the catalytic converter performance and finishes with the carbon balance equation for predicting CO_2.

C1.6 Fuel Dependent Emissions

C1.6.1 Introduction

Before relating emissions to fuel consumption, it is necessary to review the HDM-4 fuel consumption model presented in Section B.4. In essence, the model firstly predicts the steady speed fuel consumption, which is then adjusted to account for acceleration cycles through the use of the term dFUEL. dFUEL takes into account the proportional increase in fuel consumption caused by non-steady speed driving.

Two alternatives are available for predicting vehicle emissions as a function of fuel consumption; the first is to predict emissions as a function of the congested fuel consumption (*ie* the steady state plus dFUEL). Alternatively, we could predict them as a function of the steady state fuel consumption, then use a factor to increase for congestion.

The former of these two approaches results in a much tidier model, and on the basis of the work in An, *et al*. (1997), it is this technique that has been adopted for developing the HDM-4 emissions model. In addition, the prediction of certain emissions requires the actual fuel consumption to be utilised, which can only be through the use of the congested fuel consumption.

In relating emissions to fuel consumption, the assumption is primarily one of chemical balance. Therefore, the production of emissions is considered to be directly proportional to the fuel consumed, resulting in a nominal model form as given below. As is illustrated in the remainder of this chapter, for the prediction of some emissions a slightly modified form of this model is utilised, although the base component is still evident.

$$EOE_i = FC\left(\frac{g_{emissions}}{g_{fuel}}\right)_i \qquad \text{...(C1.3)}$$

$$FC = \frac{IFC \quad MassFuel \quad 1000}{v} \qquad \text{...(C1.4)}$$

where	FC	is the fuel consumption (including congestion effects) in g/km
	$g_{emissions}/g_{fuel}$	is the ratio of engine-out emissions per gram of fuel consumed for emission i
	IFC	is the instantaneous fuel consumption (including congestion effects) in mL/s
	MassFuel	is the mass of fuel in g/mL
	v	is the vehicle speed in m/s

As part of the ISOHDM study, the SNRA (1995) presented a vehicle emissions model similar in form to that shown in Equation C1.3, with the exception that a constant component was also included and no explicit allowance for catalytic converters was made.

The work of An, *et al. (*1997) included an allowance for the air/fuel equivalence ratio (ϕ) in emission production. The form of the fuel consumption model adopted for HDM-4 (refer Section B.4) does not incorporate the modelling of this item and hence an adjustment was necessary. An, *et al*. (1997) state that the air/fuel equivalence ratio varies between 0.98 and 1.02 during stoichiometric operating conditions (normal conditions) and increases up to a value of 1.3 during "rich" operating conditions, when power demand exceeds the maximum engine power.

For the range of conditions that the HDM-4 emissions model is expected to cover, rich operating conditions will be minimal and thus it has been assumed that the air/fuel ratio can be assumed as a constant. During the remainder of this work on emissions modelling, the air/fuel equivalence ratio has therefore been set equal to 1.0

C1.6.2 Swedish Emission Models

In the SNRA (1995) work mentioned earlier, relationships were presented for urban and rural driving, and new and old vehicle technologies, based on a linear regression analysis. This work was based on output generated from the VETO software and then appropriate conversions were made to enable emissions to be predicted as a function of the fuel consumption. Hammarström (2000) noted that the models generated are valid for a a link, but owing to the various adjustments made in fitting the models to a pre-determined form are not suitable for a second-by-second basis.

The form of the model used in the regression analysis is (SNRA, 1995) given below (note that the prediction is in terms of g/s and not g/km with the fuel consumption the instantaneous fuel consumption in mL/s):

$$\text{Emissions (g/s)} = a0 + a1\ IFC \qquad \text{...(C1.5)}$$

The analysis based on the combined rural and urban data yielded unexpected results in that some of the emissions were shown to decrease with increasing fuel consumption, despite both the individual urban and rural relationships showing that the emissions increased. It is postulated that this anomaly is due to the fact that the range of fuel consumption flows over which the measurements were taken in the urban and rural studies were not the same, and thus the average results are distorted and thus considered invalid. The SNRA (1995) note that for general driving conditions reasonable values would be obtained, however in some situations such as steep downgrades, negative emission levels would be predicted unless appropriate modifications were made to the models.

Table C1.4 contains the various coefficients from SNRA (1995) including R^2 values for the new technology vehicles. From a pure theoretical view point, the value of the constant for the various emissions, should always be zero, as when no fuel is being consumed, no emissions can be produced. From an examination of the R-squared values, clearly some of the relationships are tenuous, with the simple linear equation accounting for less than half of the variation in over 30 per cent of the equations.

As the work of SNRA (1995) does not explicitly consider the impact of catalytic converters, the results of the SNRA model are best considered as equivalent to the tailpipe emissions rather than engine out emissions. To use these model coefficients within HDM-4, the appropriate adjustment to account for the impact of catalytic converters and a change in units to g/km would be required.

However, given the anomalies in the predictions and the potential problems in applying the models in HDM-4, they were replaced with the work presented here.

C1.6.3 An, *et al. (*1997)

An, *et al*. (1997) present the following models for predicting vehicle emissions. As noted earlier, the air/fuel equivalence ratio (ϕ) is not modelled within the fuel consumption model used within HDM-4, but has been assumed to be equal to unity (1.0). For those predictive models within An, *et al*. (1997) that do include this variable, a value of 1.0 has been substituted and the resulting equation simplified.

CARBON MONOXIDE (CO)

For the prediction of CO, it is assumed that a direct relationship exists between fuel consumption and the production of CO in the engine.

$$EOE_{CO} = a_{CO}\ FC \qquad \text{...(C1.6)}$$

where EOE_{CO} is the engine-out CO emissions in g/km

a_{CO} is a constant = g_{CO}/g_{fuel}

HYDROCARBONS (HC)

Hydrocarbons are believed to be generated from two sources within a combustion engine. The first is from the burning of the fuel, while the second is from incomplete combustion. Therefore, the model for the prediction of HC being emitted from the engine is as follows:

$$EOE_{HC} = a_{HC}\ FC + \frac{r_{HC}}{v} 1000 \qquad \text{...(C1.7)}$$

where EOE_{HC} is the engine-out HC emissions in g/km

a_{HC} is a constant = g_{HC}/g_{fuel}

r_{HC} is a constant to account for incomplete combustion in g/s

Table C1.4: Coefficients from Swedish Study for New Technology Vehicles

Vehicle Type	Area	HC a0	HC a1	HC R^2	CO a0	CO a1	CO R^2	NOx a0	NOx a1	NOx R^2	Particulates a0	Particulates a1	Particulates R^2
Car	Rural	-0.000113	0.000436	0.877	-0.00200	0.00512	0.655	-0.00182	0.00340	0.992			
	Urban	0.0000349	0.000357	0.622	-0.00206	0.00467	0.200	-0.00150	0.0024	0.580			
	Total	-0.000272	0.000394	0.847	-0.00291	0.00554	0.658	-0.00314	0.00400	0.895	-1.48×10^{-5}	3.92×10^{-5}	0.758
Light Duty	Rural	-0.00691	0.000866	0.949	-0.00886	0.0145	0.620	-0.000839	0.00251	0.835			
	Urban	0.000150	0.000235	0.470	-0.00728	0.0126	0.687	-0.000269	0.00152	0.417			
	Total	-0.00533	0.000780	0.913	-0.00968	0.0148	0.713	-0.00166	0.00283	0.838	-2.16×10^{-5}	6.85×10^{-5}	0.560
Truck	Rural	0.0156	0.000808	0.140	0.0215	0.00623	0.830	0.0185	0.0185	0.886			
	Urban	0.0261	-0.000977	0.063	0.0454	0.00186	0.051	-0.0126	0.0332	0.961			
	Total	0.0233	-0.000785	0.038	0.0383	0.00275	0.106	0.0205	0.0187	0.842	-5.25×10^{-4}	1.40×10^{-3}	0.862
Artic-Truck	Rural	0.0106	0.00129	0.484	-0.00165	0.00921	0.982	0.0860	0.0178	0.974			
	Urban	0.0156	0.00208	0.592	0.0225	0.00573	0.365	0.0340	0.0306	0.971			
	Total	0.0244	0.000345	0.189	0.00776	0.00814	0.838	0.100	0.0175	0.783	2.81×10^{-3}	2.10×10^{-4}	0.164

Emissions (g/sec) = a0 + a1 IFC (mL/s)

Source: SNRA (1995)

NITRIC OXIDES (NOX)

Nitric Oxides are possibly the emission that this least related directly to fuel consumption. As a result of this, the model used to relate NOx production as a function of fuel consumption is more complex that the others presented in this chapter. The model form proposed by An, *et al. (*1997) is:

$$EOE_{NOx} = \max\left[a_{NOx}\left(FC - \frac{FR_{NOx}}{V} 1000 \right), 0 \right] \quad \text{...(C1.8)}$$

where EOE_{NOx} is the engine-out NOx emissions in g/km
a_{NOx} is a constant = g_{NOx}/g_{fuel}
FR_{NOx} is a the fuel threshold below which NOx emissions are very low in g/s

Values for the constants contained in Equations C1.6 to C1.8 are given in Table C1.5. Table C1.5 also contains some basic attributes of the different vehicles for which the parameter values were produced, such as mass and engine size.

Table C1.5: Constants for Prediction of Vehicle Emissions

Variable	1981 Toyota Celica	1986 Buick Century	1995 Honda Civic
Engine Capacity (litres)	2.4	2.8	1.6
Mass (kg)	1361	1531	1247
Pmax (kW)	78	84	76
a_{CO}	0.16	0.11	0.10
a_{HC}	0.013	0.013	0.012
r_{HC}	0.006	0.008	0.000
a_{Nox}	0.03	0.016	0.055
FR_{Nox}	0.39	0.49	0.17

Source: An, et al. (1997)

C1.6.4 ETSU (1997)

The ETSU (1997) present fuel dependent emission models for SO_2 and Pb. The models presented are of the same form as Equation C1.3.

SULPUR DIOXIDE (SO_2)

The quantity of SO_2 produced is related directly to the quantity of sulphur present in the fuel. Estimation of the model coefficient is made by assuming that all the sulphur in the fuel is converted to SO_2 (ETSU, 1997). Based on this assumption, the following relationship is derived for predicting SO_2 engine out emissions.

$$EOE_{SO_2} = 2\, a_{SO_2}\, FC \quad \text{...(C1.9)}$$

where EOE_{SO2} is the engine-out SO_2 emissions in g/km
a_{SO2} is a constant = g_{SO2}/g_{fuel}

A default value of a_{SO2} is estimated by ETSU (1997) from a variety of countries fuel supplies as:

- 0.0005 for petrol vehicles; and,
- 0.005 for diesel vehicles.

LEAD (Pb)

The quantity of Pb produced is related directly to the quantity of lead present in the fuel, which in recent years has been dramatically decreased (or eliminated) in many countries over health concerns. Estimation of the model coefficient is made by assuming that a proportion (default = 75 %) of the lead in the fuel is converted to lead emissions (ETSU, 1997). Based on this assumption, the following relationship is derived for predicting Pb engine out emissions.

$$EOE_{Pb} = Prop\_Pb\ a_{Pb}\ FC \qquad ...(C1.10)$$

where EOE_{Pb} is the engine-out Pb emissions in g/km
$Prop\_Pb$ is the proportion of lead emitted (default = 0.75)
a_{Pb} is a constant = g_{Pb}/g_{fuel}

A default value of a_{Pb} is estimated by ETSU (1997) from a variety of countries fuel supplies as:

- 0.000537 for petrol vehicles; and,
- 0 for diesel vehicles (*ie* diesel fuel should contain no lead).

C1.6.5 Catalytic Converters

Catalytic converters aim to reduce certain harmful emissions into chemical compounds that are less harmful to both human life and the environment. Primarily catalytic converters aim to convert any carbon (found in CO, HC and particulate matter) into CO_2, and NOx into amonia, nitrogen, CO_2 and N_2O (laughing gas) depending on operating conditions prevailing at the time. Significant changes have been made to the efficiency of the catalyst technology since the early oxidising catalysts. Current catalysts are made up of a range of materials in order to address a wider range of emissions, with current research focusing on the improved efficiency during cold start conditions.

The effectiveness of catalytic converters in reducing emissions is modelled through the term Catalyst Pass Fraction (CPF). The effectiveness of catalysts is dependent on the temperature of the catalytic converter, with significant variances often observed between hot-stabilised and cold-start conditions. For the model being developed for HDM-4, only the hot-stabilised portion is required.

An, *et al*. (1997) present the following equation for the prediction of CPF, with the relevant constants contained in Table C1.6. As with the emission prediction models described above, the air/fuel equivalence ratio has been set equal to 1.0 and the appropriate simplifications made. An, *et al*. (1997) note that the form of Equation C1.11 represents a decreasing probability for oxidation with increasing fuel consumption. An, *et al*. (1997) state that the form of the CPF equation for the prediction of NOx is not good, and that only an average CPF for these emissions is used[1].

$$CPF_i = 1 - \varepsilon_i \exp[-b_i\ IFC\ MassFuel] \qquad ...(C1.11)$$

where CPF_i is the catalyst pass fraction for emission I
ε_i is the maximum catalyst efficiency for emissions ei
b_i is the stoichiometric CPF coefficient

[1] An *et al*. (1997) do not include the prediction of NOx into an equation, but rather state that it is a constant value. However, for consistency across emissions, the appropriate constants in Equation 1.11 have been set to zero to enable one form of model for all emissions.

Table C1.6: Catalyst Pass Fraction Constants

Vehicle	CO		HC		NOx	
	ε_{CO}	b_{CO}	ε_{HC}	b_{HC}	ε_{NOx}	b_{NOx}
1981 Toyota Celica	.279	1.60	.474	1.30	0.000	0.0
1986 Buick Centrury	.999	0.46	.999	0.11	0.753	0.0
1995 Honda Civic	.999	0.05	.999	0.03	0.812	0.0

Source: An, et al. (1997)

Lead poisons catalytic converts, such that they are only suitable for use with unleaded fuels. There is therefore no reduction in lead emissions ($\varepsilon_{Pb} = 0$) from the installation of a catalytic converter as these are mutually exclusive occurrences.

In a similar manner to lead, SO_2 also poisons the catalyst. SO_2 is also not converted or absorbed by catalytic converters, such that the engine out emission of SO_2 equals the tailpipe emissions.

Particulate matter is unaffected by catalysts and is simply emitted from the tailpipe in the same quantity as it is emitted from the engine. Therefore CPF factors of $\varepsilon_{PM} = 0$ and $b_{PM} = 0$ are utilised.

Catalysts serve to convert CO and HC into CO_2, and do not treat CO_2 in itself. As is indicated in Equation C1.13 within the next section, the prediction of CO_2 already includes this effect by utilising tailpipe emissions in the carbon balance equation. Therefore, as there is a direct prediction of the tailpipe emissions for CO_2 there is no CPF.

It is evident from the constants in Table C1.6 that significant variation occurs between vehicles, with the newer model cars performing significantly better than the older models. To assist in understanding the values within Table C1.6, the CPF ratios have been plotted over a range of typical fuel consumption rates.

Figure C1.3 illustrates the variation of CPF with fuel consumption for the 1995 Honda Civic. The form of the equation for predicting CPF within An, *et al*. (1997) contains an additional variable in the exponential portion of Equation C1.11. This additional component dealt with periods of rich operation. As noted elsewhere, only the stoichiometric conditions are being modelled within this research. Subsequent to this, Figure C1.3 shows that the variation in CPF is essentially linear over the typical range of fuel consumption observed.

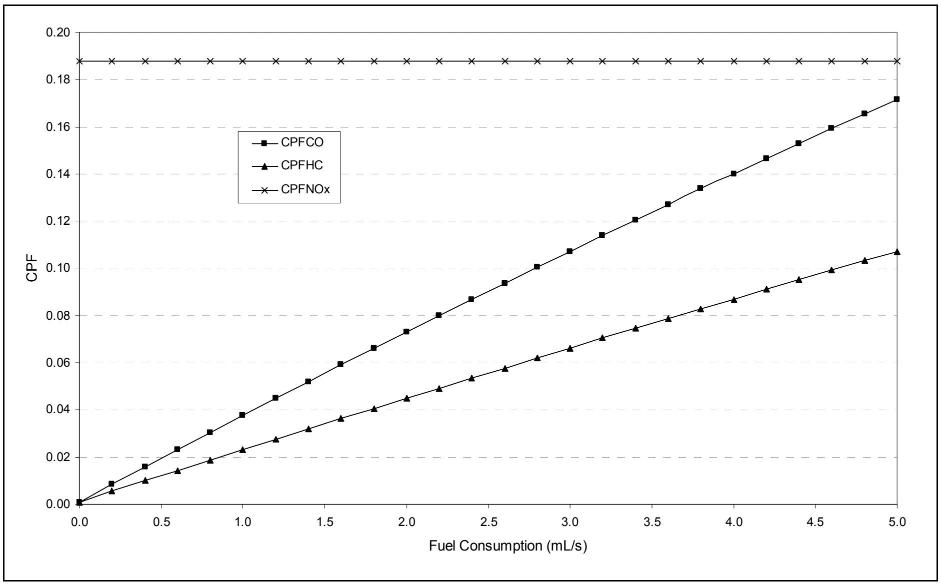


Figure C1.3: Variation of CPF with Fuel Consumption

Deterioration of the exhaust system is required to be modelled. The SNRA (1995) state that this is particularly important for new technology vehicles, where catalytic converters are present. The SNRA (1995) implemented a simple linear deterioration function with time to account for this deterioration. Although not included in either the model form in SNRA (1995) or the HDM-4 Technical Reference Manual (Odoki and Kerali, 1999b), SNRA (1995) state that the normal approach is to have an upper limit on the accumulated deterioration. A value of 10 for this upper limit is stated as normal practice for Sweden by SNRA (1995).

The addition of a deterioration component to the modelling of the CPF yields the following equation:

$$CPF_i = \left[1 - \varepsilon_i \exp\left(-b_i \ \text{IFC MassFuel}\right)\right] \min\left[(1 + \frac{r_i}{100} AGE) \, , \, MDF_i\right] \qquad \text{...(C1.12)}$$

where r_i is the deterioration factor for emission i in %/year
AGE is the vehicle age in years
MDF_i is the maximum deterioration factor for emission i (default = 10)

Values for the deterioration factor, r_i, are given in Table C1.7. These values are extracted from SNRA (1995) and Hammarström (1999), and indicate a significant increase in the production of HC and CO for new technology vehicles. In particular, the emission of CO is predicted to double every four years and HC every 5 years. Deterioration is caused by a combination of poisoning (by lead or sulphur) or via thermal degradation, wherein the noble metals disintegrate under extreme heat, thereby reducing the effective surface area of the catalyst.

Table C1.7: Deterioration Factors r_i

Vehicle Technology	Substance			
	HC	CO	NOx	Particulates
Old[1]	7	6	2	
New[1]	20	25	5	
New[2]	20	4.8	11	4.8

Source: 1/ SNRA (1995)
2/ Hammarstrom (1999)

Hammarström (1999) indicates that his values are based around a total travelled distance of 200,000 km over a period of 13 years. Hammarström (1999) states that the difference between the values he reports and those from the SNRA (1995) are that the newer values are from statistical data, while the older values are from a literature review at that time. The values of Hammarström (1999) could therefore be considered a more reliable source for a modern vehicle fleet.

When using different deterioration factors it is important to ensure that the deterioration factors are calibrated in conjunction with the base rates for a new vehicle (Hammarström, 2000). The application of the above deterioration rates, in conjunction with base emission rates generated from older vehicles, will combine to overpredict the quantity of emissions being produced.

C1.7 Carbon Balance

The estimation of CO_2 production is undertaken by solving the carbon balance equation given below as Equation C1.13. This equation, extracted from ETSU (1997) yields the quantity of CO_2 based on the overall Carbon consumed, less that extracted by other forms. It is noted that this equation utilises the tailpipe emissions as the catalytic converter (refer previous section) increases the output of CO_2 by converting CO and HC and particulate matter into CO_2.

$$TPE_{CO_2} = 44.011\left(\frac{FC}{12.011+1.008\,a_{CO2}} - \frac{TPE_{CO}}{28.011} - \frac{TPE_{HC}}{0.01385} - \frac{TPE_{PM}}{12.011}\right) \quad ...(C1.13)$$

where a_{CO2} is a fuel dependent model parameter representing the ratio of hydrogen to carbon atoms in the fuel

In the medium to long term, the concentration of HC and CO will tend to form into CO_2 (Hammarström, 2000). Therefore, the fianl concentration of CO_2 in the atmostphere owing to combustion will be higher than that calculated here. However, at the end of the tailpipe, when such breakdown has not occurred, the above will give reliable estimates of CO_2 output. By removing the HC and CO terms from Equation C1.13 and estimate of the long term CO_2 production can be made.

C1.8 Examples of Predictions

Based on the recommended predictive models and the parameter values given in Table C1.8 and C1.9, an estimation of vehicle emission of vehicle emissions for a passenger car and an articulated truck have been made. These results are based on steady speed conditions and are therefore likely to be the lower limit of the various predictions. Both vehicles were assumed to have a catalytic converter in place.

The results of the estimates are shown in Figure C1.4 and C1.5 for the tail pipe emissions. As would be expected with the relationship of the emissions to the fuel consumption, the emission rates also contain a distinct u-shape with a minimum around the 50 km/h range.

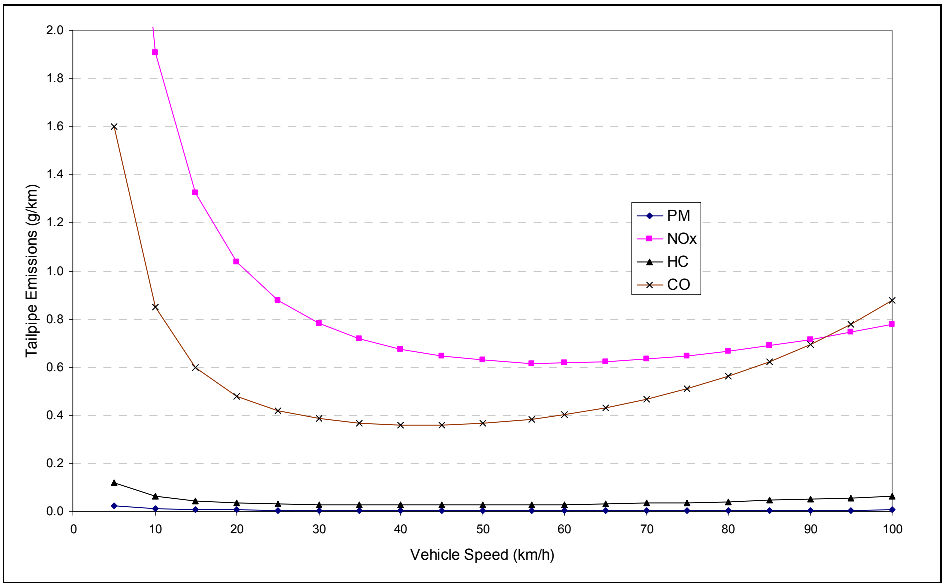


Figure C1.4: Emission Predictions for Default HDM-4 Large Passenger Car

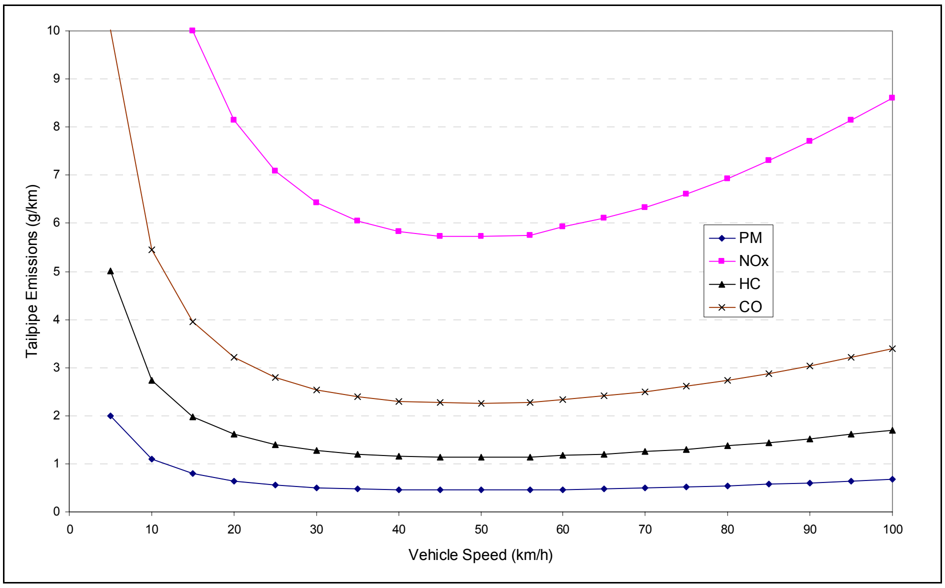


Figure C1.5: Emission Predictions for Default HDM-4 Articulated Truck

C1.9 Summary

For the prediction of vehicle emissions within this report the following relationships have been adopted

$$TPE_i = EOE_i \, CPF_i \qquad ...(C1.14)$$

EOE_i is based on the emission under consideration as given below:

HC $$EOE_{HC} = a_{HC}\, FC + \frac{r_{HC}}{v} 1000 \qquad ...(C1.15)$$

CO $$EOE_{CO} = a_{CO}\, FC \qquad ...(C1.16)$$

NOx $$EOE_{NOx} = \max\left[a_{NOx} \left(FC - \frac{FR_{NOx}}{v} 1000 \right), 0 \right] \qquad ...(C1.17)$$

SO_2 $$EOE_{SO_2} = 2\, a_{SO_2}\, FC \qquad ...(C1.18)$$

Pb $$EOE_{Pb} = Prop\_Pb \; a_{Pb}\, FC \qquad ...(C1.19)$$

PM $$EOE_{PM} = a_{PM}\, FC + \frac{r_{PM}}{v} 1000 \qquad ...(C1.20)$$

The catalytic pass fraction is given by:

$$CPF_i = [1 - \varepsilon_i \exp(-b_i \; IFC \; MassFuel)] \min\left[(1 + \frac{r_i}{100} LIFE), MDF_i \right] \qquad ...(C1.21)$$

CO_2 is predicted based on the assumption of carbon balance utilising the following relationship:

$$TPE_{CO_2} = 44.011\left(\frac{FC}{12.011+1.008\,a_{CO2}} - \frac{TPE_{CO}}{28.011} - \frac{TPE_{HC}}{0.01385} - \frac{TPE_{PM}}{12.011}\right) \quad ...(C1.22)$$

Parameter values for the 16 default HDM-4 vehicles for the above equations are given in Table C1.8 and C1.9. The mass of fuel (MassFuel) may be taken as (Heywood, 1988):

- Petrol: MassFuel = 0.75 g/mL; and,
- Diesel: MassFuel = 0.86 g/mL.

Table C1.8: Default Emission Model Parameter Values for HDM-4 Representative Vehicle Classes

HDM-4 Vehicle	Fuel Type	HC		NOx		CO	SO_2	Pb		PM		CO_2
		a_{HC}	r_{HC}	a_{NOx}	FR_{NOx}	a_{CO}	a_{SO2}	Prop_Pb	a_{Pb}	a_{PM}	b_{PM}	a_{CO2}
MC	P	0.060	0	0.020	0.00	0.20	0.0005	0.75	0	0.0001	0.0	1.8
PC-S	P	0.012	0	0.055	0.17	0.10	0.0005	0.75	0	0.0001	0.0	1.8
PC-M	P	0.012	0	0.055	0.17	0.10	0.0005	0.75	0	0.0001	0.0	1.8
PC-L	P	0.012	0	0.055	0.17	0.10	0.0005	0.75	0	0.0001	0.0	1.8
LDV	P	0.012	0	0.055	0.17	0.10	0.0005	0.75	0	0.0001	0.0	1.8
LGV	P	0.012	0	0.055	0.17	0.10	0.0005	0.75	0	0.0001	0.0	1.8
4WD	D	0.040	0	0.027	0.00	0.08	0.005	0.75	0	0.0032	0.0	2.0
LT	D	0.040	0	0.027	0.00	0.08	0.005	0.75	0	0.0032	0.0	2.0
MT	D	0.040	0	0.027	0.00	0.08	0.005	0.75	0	0.0032	0.0	2.0
HT	D	0.040	0	0.027	0.00	0.08	0.005	0.75	0	0.0032	0.0	2.0
AT	D	0.040	0	0.027	0.00	0.08	0.005	0.75	0	0.0032	0.0	2.0
MNB	P	0.012	0	0.055	0.17	0.10	0.0005	0.75	0	0.0001	0.0	1.8
LB	D	0.040	0	0.027	0.00	0.08	0.005	0.75	0	0.0032	0.0	2.0
MB	D	0.040	0	0.027	0.00	0.08	0.005	0.75	0	0.0032	0.0	2.0
HB	D	0.040	0	0.027	0.00	0.08	0.005	0.75	0	0.0032	0.0	2.0
COACH	D	0.040	0	0.027	0.00	0.08	0.005	0.75	0	0.0032	0.0	2.0

Source: *An, et al. (1997)*
ETSU (1997)
SNRA (1995)

Table C1.9: Default Emission Model Parameter Values for HDM-4 Representative Vehicle Classes

HDM-4 Vehicle	Fuel Type	HC			NOx			CO			SO_2			Pb			Particulates		
		ε_i	b_i	r_i	ε_i	b_i	r_i	ε_i	b_i	r_i	ε_i	b_i	r_i	ε_i	b_i	r_i	ε_i	b_i	r_i
MC	P	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
PC-S	P	0.999	0.03	20	0.812	0	11	0.999	0.05	4.8	0	0	0	0	0	0	0	0	4.8
PC-M	P	0.999	0.03	20	0.812	0	11	0.999	0.05	4.8	0	0	0	0	0	0	0	0	4.8
PC-L	P	0.999	0.03	20	0.812	0	11	0.999	0.05	4.8	0	0	0	0	0	0	0	0	4.8
LDV	P	0.999	0.03	20	0.812	0	11	0.999	0.05	4.8	0	0	0	0	0	0	0	0	4.8
LGV	P	0.999	0.03	20	0.812	0	11	0.999	0.05	4.8	0	0	0	0	0	0	0	0	4.8
4WD	D	0.900	0.00	20	0.250	0	11	0.900	0.00	4.8	0	0	0	0	0	0	0.5	0	4.8
LT	D	0.900	0.00	20	0.250	0	11	0.900	0.00	4.8	0	0	0	0	0	0	0.5	0	4.8
MT	D	0.900	0.00	20	0.250	0	11	0.900	0.00	4.8	0	0	0	0	0	0	0.5	0	4.8
HT	D	0.900	0.00	20	0.250	0	11	0.900	0.00	4.8	0	0	0	0	0	0	0.5	0	4.8
AT	D	0.900	0.00	20	0.250	0	11	0.900	0.00	4.8	0	0	0	0	0	0	0.5	0	4.8
MiniBus	P	0.999	0.03	20	0.812	0	11	0.999	0.05	4.8	0	0	0	0	0	0	0	0	4.8
LB	D	0.900	0.00	20	0.250	0	11	0.900	0.00	4.8	0	0	0	0	0	0	0.5	0	4.8
MB	D	0.900	0.00	20	0.250	0	11	0.900	0.00	4.8	0	0	0	0	0	0	0.5	0	4.8
HB	D	0.900	0.00	20	0.250	0	11	0.900	0.00	4.8	0	0	0	0	0	0	0.5	0	4.8
COACH	D	0.900	0.00	20	0.250	0	11	0.900	0.00	4.8	0	0	0	0	0	0	0.5	0	4.8

Source: *An, et al (1997)*
Clean Cat (2000)
Discount Converters Ltd (2000)
Greenwood (2000)
Hammarstrom (1999)
SNRA (1995)

C2 Vehicle Noise Impact

C2.1 Introduction

It was originally envisaged that the environmental impact of vehicle noise would be considered in HDM-4. However, it was subsequently decided that this was best left as an analysis that would be conducted external to HDM-4. Since NDLI (1995) presented models for predicting noise they are repeated here for completeness.

There are two sources of noise:

- The exterior noise - *ie* the rolling noise from the contact between tyres and pavement; and,
- The interior noise - *ie* the vehicle noise from engine, transmission, exhaust and suspension which is highest during acceleration, on upgrades, during engine braking, on rough roads and in interrupted traffic condition.

Since road investment projects deal with exterior noise effects, these are the focust of this chapter.

NDLI (1995) proposed adopting a simple procedure for evaluating noise emissions caused by the traffic due to the exterior noise emission. The chapter opens with definitions of noise and other key considerations. This is followed by a summary of the noise modelling techniques used in the UK and USA. On the basis of these, the methodology proposed for HDM-4 is presented.

C2.2 Definitions

Noise is measured in terms of sound level expressed by the logarithmic function of acoustic pressure having the unit of decibels (dB). Noise measurement specifications require the definition of the period of measurement, the noise parameter to be recorded and the position of the recording instrument. Some definitions used in this chapter are:

Leq The equivalent acoustic level. This is the sound level of a stable noise which contains the same energy as the variable noise over the same period. It represents the mean of the energy perceived during the period of observation, usually over 12 hours, and designated as Leq (12hr).

L10 The noise level exceeded 10 per cent of the time over a measured time period. For example, L_{10} hourly dB(A) is the noise level exceeded for just 10 per cent of the time over a period of one hour. The L_{10}(18 hour) dB(A) is the arithmetic average of the values of L_{10} hourly dB(A) for each of the eighteen one-hour periods between 0600-2400 hours.

Source line this is a line to the noise source which is taken to be 0.5 m above the carriageway level and 3.5 m in from the nearside carriageway edge.

Reception point The point at which the measuring equipment is located.

Road segment The section of road under analysis.

C2.3 Noise Modelling in the United Kingdom

C2.3.1 Introduction

The UK Department of Transport noise model is described in the report "Calculation of Road Traffic Noise CRTN" published by the Department of Transport (HMSO, 1988). It will be referred to as the CRTN technique hereafter.

CRTN give predictions of noise levels for free flow and congested conditions in selected standard units of L10 hourly or L10 (18 hour) dB(A). In this report only the L10 hourly dB(A) results will be discussed. The CRTN procedure for noise prediction from road scheme is given as in Figure C2.1.

C2.3.2 The Prediction Model

There are various stages in the procedure to predict noise levels and for each stage the relative corrections are made to the basic noise level. The concept of 'Basic Noise Level' is taken at the individual road segment 10m away from the edge of the roadway. The level is obtained based on the traffic flow, speed and composition of traffic, the gradient of the roadway and the road surface type.

The relationship used for model prediction for Basic Noise Level at speed 75 km/h, flat level road and no heavy vehicle is as follows:

$$L_{10}=42.2+10\text{Log}_{10}Q \qquad \ldots\text{(C2.1)}$$

where Q is the total hourly flow in veh/hr

The relationship is shown Figure C2.2.

C2.3.3 Correction Factors

Corrections are made to the predicted value of the Basic Noise Level from Equation C2.1 for the road segment to account for the following factors:

- Low volumes of traffic (*ie* flows below 200 veh/h);
- Average traffic speed and percentage of heavy vehicles;
- Road gradient;
- Road surface;
- Propagation; and,
- Site layout.

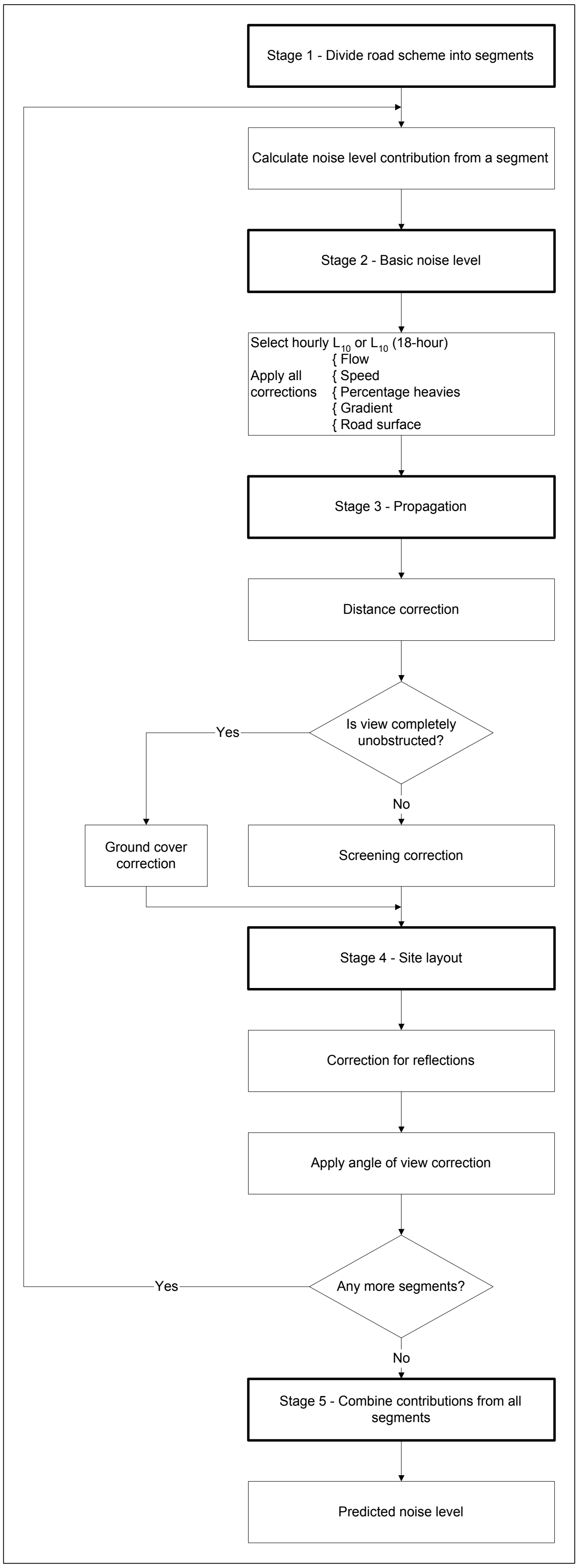


Figure C2.1: Flow Chart for the Stages in CRTN Noise Predicting Model

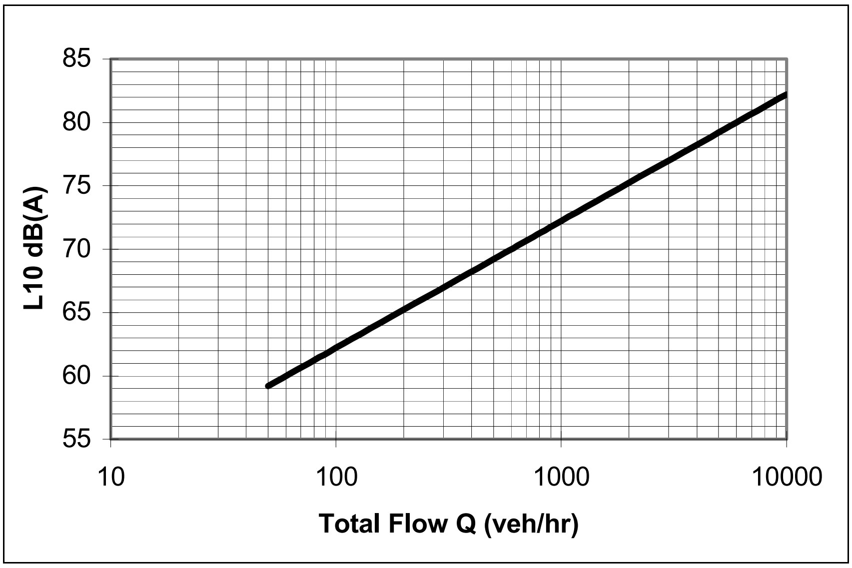


Figure C2.2: Prediction of Basic Noise Level L10

The predicted noise level requires the combination of noise level contributors from all the source segments which comprise the total road scheme. Since the correction factors are in a logarithmic form, the combination is done logarithmically when calculating the overall noise level (L), *ie*:

$$L = 10\ \mathrm{Log}_{10}\ [\Sigma\ \mathrm{Antilog}_{10}\ (Ln / 10)]$$

$$NL = 10\ \log 10 \sum_{i=1}^{nseg} \mathrm{anti}\log_{10}\left(\frac{NLi}{10}\right) \qquad \ldots(C2.2)$$

where NL is the noise level for all sections in dB(A)
NLi is the noise level for section i in dB(A)
nseg is the number of sections

C2.4 FHWA Noise Prediction Model

C2.4.1 Introduction

In the FHWA method (Barry and Regan , 1978), a similar concept of correction factors to that of the CRTN method are applied to the Basic Noise Level predicted at the reception point. This is referred to as the reference sound level. The reception point is taken as 15 m away from the centerline of the carriageway lane.

The reference sound level is the mean energy emission level, while adjustments are made according to the influences of traffic flow, varying distances from the roadway, finite length roadways and also shielding effects.

The basis of the model is the equivalent sound level, L_{eq}, although an adjustment factor for conversion to L10 is provided.

C2.4.2 The Prediction Model

The relationship for the model is given as the expression below for all the variables concerned:

$L_{eq}(h)_i = (L_o)_{Ei}$	Reference energy mean emission level
$+10\log\left(\frac{N_i \pi D_o}{S_i T}\right)$	Traffic flow adjustment
$+10\log\left(\frac{D_o}{DIS}\right)^{1+\alpha s}$	Distance Adjustment
$+10\log\left(\frac{\psi n_{\alpha s}(\phi 1, \phi 2)}{\pi}\right)$	Finite roadway adjustment
$+\Delta_S$	Shielding adjustment

…(C2.3)

where

L_{eq} (hi)	is the hourly equivalent sound level of the ith class of vehicles
$(L_o)_{Ei}$	is the reference energy mean emission level of the ith class of vehicles
N_i	is the number of vehicles in the ith class passing a specified point in a one-hour period
DIS	is the perpendicular distance, in m from the centerline of the traffic lane to the observer
Do	is the reference distance at which the emission levels are measured. In the FHWA model Do is 15 m
Si	is the average speed of the ith class of vehicles in km/h
T	is the time period over which the Leq is computed (usually 1 hour)
αs	is the site parameter depending on site conditions
ψn	is the function of segment adjustment *ie* adjustment for finite length roadways
Δs	is the attenuation in dB provided by shielding
ϕ	is the angle subtended by the finite road segment to the observer
π	is pi constant (3.1415)

The first two terms of the equation predict the hourly equivalent sound level (L_{eq}) generated by a flow of a single class of vehicles travelling at a constant speed on an effective infinite, flat road at a reference point of 15 m away from the center of the carriageway lane. The last three terms represent the adjustments (corrections) dealing with the site conditions.

The total L_{eq} is then the sum of the contributions of the various classes of vehicles on the roadway. There are three distinct classes used mainly passenger cars (PC), medium trucks (MT) and heavy trucks (HT). The total hourly equivalent sound level is computed as:

$$L_{eq}(h) = 10\log\left(10^{\frac{Leq(h)_{PC}}{10}} + 10^{\frac{Leq(h)_{MT}}{10}} + 10^{\frac{Leq(h)_{HT}}{10}}\right) \quad \text{…(C2.4)}$$

Figure C2.3 shows the computational sequence followed in the FHWA method for arriving at a predicted sound level.

Step 1 Compute the energy level for all vehicle class

Step 2 Adjustment for vehicle volume of class I Ni and the speed Si at distance Do i.e. NiDo/Si

Step 3 Adjustment for distance greater than 15 meters and ground condition

Step 4 Adjustment for finite length of road sections from orientation and ground effect

Step 5 Obstructions and barriers effect in attenuation of noise level

Step 6 Conversion to $L_{10}(h)i$

Step 7 Summation of all $L_{10}(h)i$ from prediction of vehicle groups

Figure C2.3: The Flow Chart Process in the FHWA Noise Prediction Model

C2.4.3 The Reference Energy Mean Emission Level

The reference energy mean emission level represents the emission levels of a large number of different types of vehicles at various speeds and determined through statistical techniques. Based on this analysis, the FHWA Noise prediction model relates the vehicles into three distinct acoustic source groups. The model uses the following relationship in speed applicable for only flat terrain at cruising speed of between 50-100 km/h:

$$(Lo)_{E_A} = 38.1 \log(S) - 2.4 \quad \ldots(C2.5)$$

$$(Lo)_{E_{MT}} = 33.9 \log(S) + 16.4 \quad \ldots(C2.6)$$

$$(Lo)_{E_{HT}} = 24.6 \log(S) + 38.5 \quad \ldots(C2.7)$$

where S is the average vehicle speed of the vehicle class in km/h.

Figure C2.4 shows the relationship between the Reference Energy Mean Emission Levels as a function of speed.

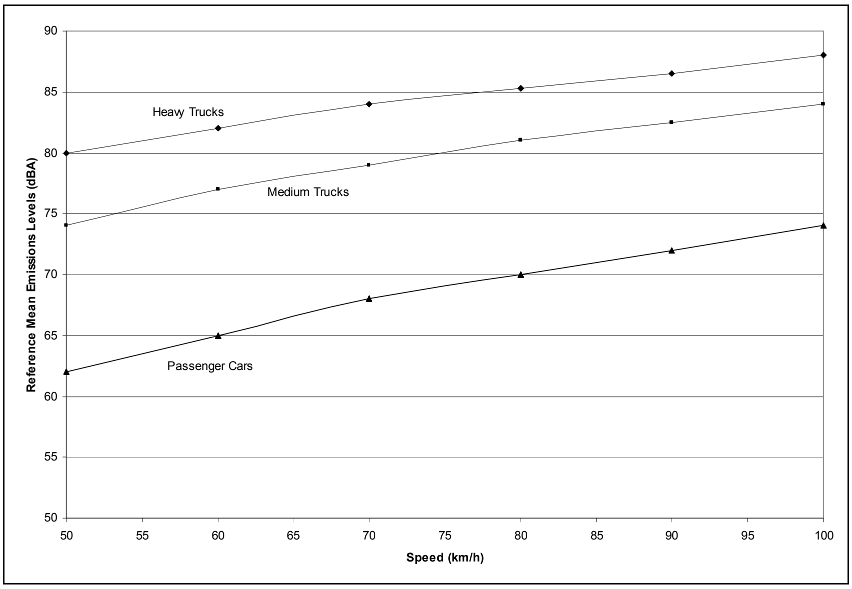


Figure C2.4: Reference Energy Mean Emission Level Versus Speed

C2.4.4 Conversion L_{eq} to L_{10} Value

The $L_{eq}(h)_i$ - $L_{10}(h)_i$ conversion is based on the N DIS/S ratio *ie* ratio of the volume of vehicle time distance/speed measured in m/km for a particular group of vehicle types. The assumption in the conversion factor was that the sources, *ie* the particular vehicle group, have equal power and are equally spaced. This assumption lead to conservative values of L_{10} $(h)_i$

Figure C2.5 shows the adjustment factor for converting L_{eq} (h)i to L_{10}(h)i relationship to the N DIS/S ratio. Care should be taken when considering two aspects of the traffic flow due to the two assumptions mentioned earlier.

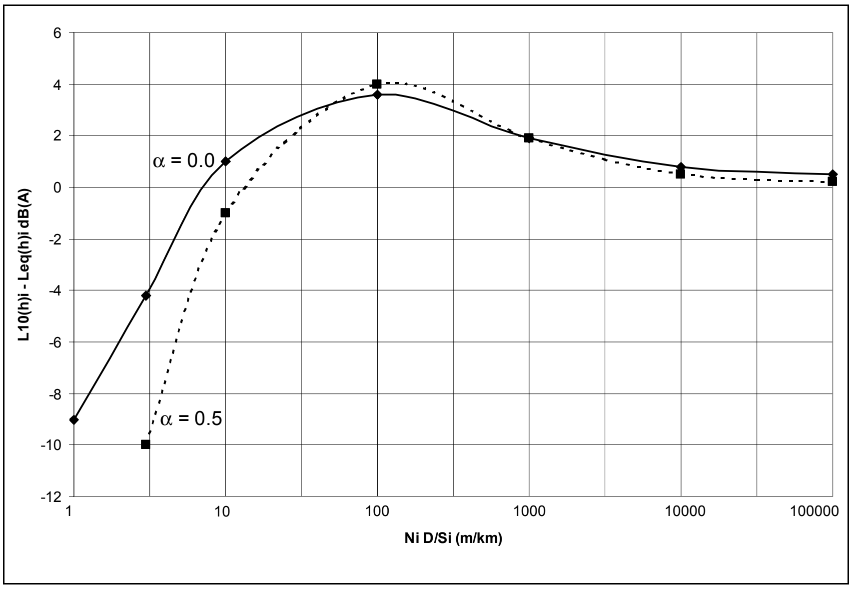


Figure C2.5: Adjustment Factor for Converting Leq(h)i to L10(h)I

Low Volume Traffic

The predictive models are based on large sample populations, *ie* average values. There is no assurance that on low volume roadways the measured sound levels are representative of the average conditions on which the predictions are made.

It is advisable that when N DIS/S ratio is less than 40 m/km, noise predictions be made in term of L_{eq} (h).

High Volume Traffic

Once the N DIS/S ratio is more than 40m/km, there is a larger volume of vehicles in the sampling measures, the individual noise becomes less critical and the overall effect is that the average values are approximated. The spacing of the vehicle tend to become more even throughout the measurement times thus become representative of the hourly volumes N_i.

C2.5 Proposed Model for HDM-4

C2.5.1 Introduction

The model proposed for HDM-4 is based on the CRTN model.

The prediction technique in HDM-4 will cover the prediction of the ‘basic noise emission level’ and the correction factors associated with:

- Vehicle classification and composition;
- Speed for free and non free flow (congestion) conditions; and,
- Road characteristics (*ie* gradient and road segments).

C2.5.2 Road Segments

In HDM-4 projects are divided into homogenous segments so that each segment will be treated as a separate road source and the noise contribution evaluated accordingly.

C2.5.3 Basic Noise Emission Level

As suggested in the CRTN model, HDM-4 will develop the reception point at 10 m away from the nearside edge of the carriageway for the calculation of the basic noise level.

The same equation is utilised for the prediction assuming a flat terrain, no heavy trucks and zero gradient and travelling at free speed of 75 km/h. The factor that influences the noise level is the total hourly flow in veh/hour (Q):

$$L_{10}=42.2+10\text{Log}_{10}Q \qquad \ldots(C2.8)$$

Additional corrections are to the basic noise level for:

- Congestion speed;
- Gradients;
- Pavement surface conditions; and,
- Distance correction.

C2.5.4 Correction Factors

SPEED CORRECTION

There are two factors that influence the speed of a vehicle:

- Interactions with other vehicles, particularly heavy vehicles; and,
- The road alignment.

The HDM-4 speed-flow model was presented in Chapter B3 and it predicts the free and interrupted flow speeds of a vehicle. As interactions increase, overall free speed of the traffic stream converge to the speed of the slowest vehicle class.

The HDM-4 free speed model predicts the free flow speed of the vehicle class based on the road gradient and curvatures. Since the noise production is a function of speed, the change in speed due to road alignment (gradient and curvatures), will influence the noise level emission.

The average speed of the traffic stream will be calculated using the HDM-4 speed model. This speed will then be substituted into the following equation to obtain the noise correction for speed:

$$\text{dBSPEED} = 33\ \text{Log}_{10}\left(\text{S}+40+\frac{500}{\text{S}}\right)+10\ \text{Log}_{10}\left(1+\frac{5\ \text{PCTHCV}}{\text{S}}\right)-68.8 \qquad \ldots(C2.9)$$

where dBSPEED is the correction due to speed in dB(A)
PCTHCV is the percentage of heavy vehicles in the traffic stream

It can be seen from the expression above that a value of 68.8 dB(A) (the constant in the expression) is the maximum combined noise level expected from the new or altered highway.

GRADIENT

A correction for the additional noise due to climbing a positive gradient. It should be noted that corrections for traffic speed on a gradient have already been taken into account in speed correction. The correction value is taken from the CRTN model:

$$dBGR = 0.3\ GR \qquad \ldots(C2.10)$$

where dBGR is the correction due to gradients due to gradients in dB(A)
GR is the gradient in per cent

ROAD SURFACE

Two types of road surface need to be considered; cement concrete surfaces and bituminous surfaces.

Table C2.1 shows the relationships used and the values in the correction factors according to speed and surface type. Typical values for texture depth are 0.5 - 0.9 mm with bituminous surfaces; 0.2 - 0.4 mm with concrete surfaces.

Table C2.1: Correction Factors for Road Surfaces

Acoustic Properties	Pervious Surfaces (Macadams)	Impervious Surfaces	
		Concrete	Bituminous
≥ 75 km/h	-3.5 dB(A)	10 log10 (90Tdsp+30) - 20 dB(A)	10 log10 (20Tdsp+60) - 20 dB(A)
< 75 km/h	-3.5 dB(A)	- 1.0 dB(A)	- 1.0 dB(A)

Notes: Tdsp is the texture depth in mm

DISTANCE CORRECTION

The distance correction is based on the "shortest slant distance" which is illustrated in Figure C2.6.

The distance correction is calculated along the shortest slant distance (d') from the source line to the reception point as :

$$dBDIST = -10 Log_{10}\left(\frac{d'}{13.5}\right) \qquad \ldots(C2.11)$$

where dBDIST is the correction for distance
d' shortest distance from effective source point, defined as:

$$d' = \left[(d+3.5)^2 + HR^2\right]^{1/2} \qquad d \geq 4.0\ m \qquad \ldots(C2.12)$$

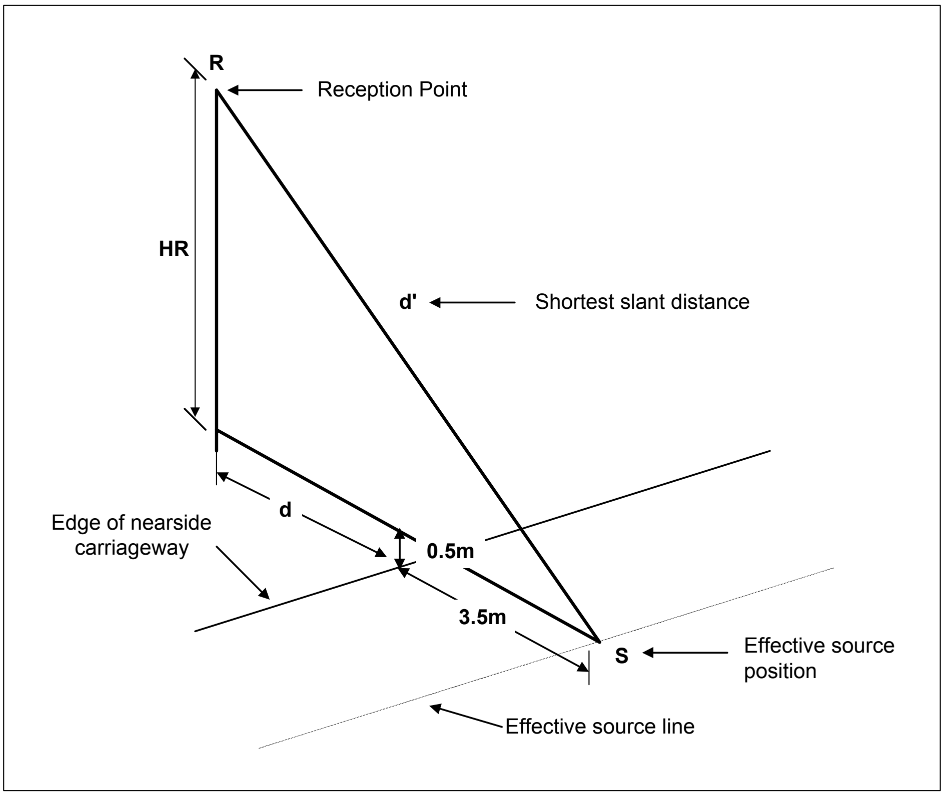


Figure C2.6: Calculation of Shortest Slant Distance

SECTION SIZE

The correction for size of section depends upon the angle ϕ (angle of view) subtended by the section boundary and the reception point as shown in Figure C2.7. The angle is obtained from the dimension L which is the length of the section and the distance d, *ie* distance R from the edge of the carriageway.

$$\phi = 2\arctan\left(\frac{\text{SECLEN}/2}{\text{d}}\right) \qquad \ldots\text{(C2.13)}$$

The correction relationship is as follows:

$$\text{dBSECT} = 10\text{Log}_{10}\left(\frac{\phi}{180}\right) \qquad \ldots\text{(C2.14)}$$

where dBSECT is the correction for section length

ϕ is the angle of view in radians

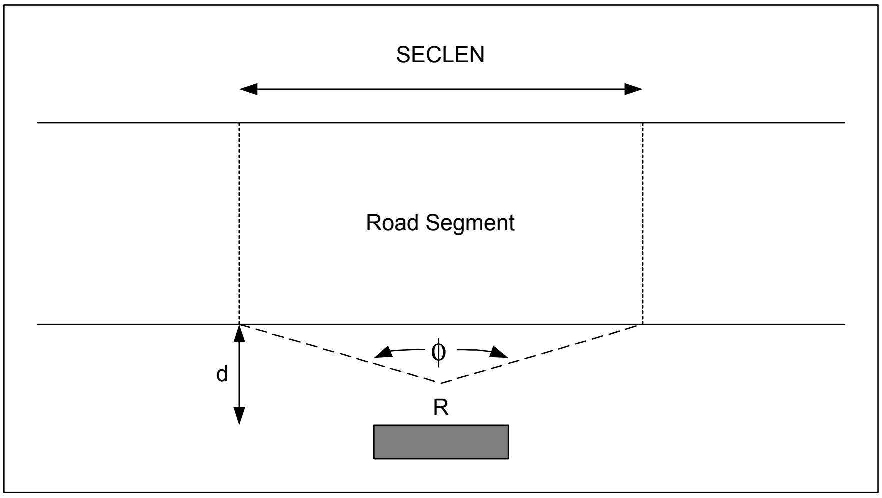


Figure C2.7: The Angle of View ϕ

CONTRIBUTION OF NOISE LEVEL FROM EACH ROAD SEGMENT

For projects which consist of more than one section, the predicted noise level at the reception point shall be calculated by combining the contributions from all the segments to give the overall noise level (L) as discussed earlier:

$$NL = 10 \log_{10} \sum_{i=1}^{nseg} \text{anti}\log_{10}\left(\frac{NLi}{10}\right) \qquad \ldots(C2.15)$$

where NL is the noise level for all sections in dB(A)
NLi is the noise level for section I in dB(A)
nseg is the number of sections

C2.6 Evaluation of Noise Level Effect in HDM-4

The evaluation of noise effect in HDM-4 should be to report the extent of noise impact to the areas in the vicinity of the road construction or improvement schemes. As suggested in the N.Z. Project Evaluation Manual (Transfund, 1994) the change in noise exposure should be identified in bands of 3 dB (A) *ie* :

- Area affected by 0 - 3 dB (A);
- Area affected by > 0 - 3 dB (A); and,
- Area affected by > 6 - 9 dB (A).

This approach is based on the distance effect incorporated within the calculation of the CRTN model for distance correction to the basic noise level, *ie* doubling the distance reduces the noise level by 3 dB (A).

Having predicted the noise level from the prediction model, the economic effect of the noise levels must be incorporated into HDM-4. The approach recommended is to select limiting noise levels that are acceptable to the bordering population. If the road surface generates exterior noise levels which are greater than the limiting values, noise attenuation activities are warranted.

For exterior noise, noise attenuation is usually achieved through building of physical and sometimes natural noise barriers.

The costing of noise should be done using the CRTN approach which considers its effects in property values.

The economic cost of the effect will be derived from the reduction of property values which is expressed as :

$$NSCST = \frac{[L_{eq}(24h) - 50]\ PV\ \dfrac{DR}{FR}}{\dfrac{NUM\_LANES}{2}} \qquad \text{…(C2.16)}$$

where PV is property value in cost/property
DR is depreciation rate per excess dB (A)
FR is property frontage width in meters
NSCST is the noise cost in cost/lane-m.

C2.7 Summary

Impact of noise traffic will be confined to situations where there is a significant population bordering a heavily trafficked road. This will usually arise in urban areas. A model was presented here which would be suitable for use in a model such as HDM-4. However, noise has not been included for HDM-4 v 1.0.

C3 Energy Balance Framework

C3.1 Introduction

Traditionally, the analyses conducted with HDM consisted of assessing the life-cycle economic implications of different investment strategies. The costs were comprised of construction, maintenance and vehicle operating costs. For HDM-4 energy balance analyses have been included which extends the analysis to include the life-cycle energy consumption. This allows for the relative efficiency of different modes of transport, both motorised and non-motorised, to be established.

A key component of the energy balance analysis is the ability to distinguish between renewable and non-renewable sources of energy. It is also important to consider energy usage from external factors—such as energy used to manufacture vehicles outside of the country—to be established. While this may not be an issue for the local policy makers, it has a bearing on global sustainability.

In keeping with the analytical framework of HDM-4, there are two levels of analysis:

- **Project Level:** This assesses the energy implications of the project alternatives against one another; and,
- **Strategic Level:** This is designed to consider the implications of different policy objectives, for example the energy implications of alternative fuels.

Collings (1999) presented a proposal for an energy balance framework to HDM-4. Odoki and Kerali (1999c) describe how this was implemented. The material presented here is based on these two sources.

C3.2 Energy Use Categories

There are three categories where the energy use can be calculated:

- Energy used by motorised vehicles;
- Energy used by non-motorised vehicles; and,
- Energy used during the construction, maintenance and rehabilitation of roads.

C3.2.1 Energy Used by Motorised Vehicles

Table C3.1 lists the energy use associated with vehicle production and use.

As described in Chapter B1, the forces opposing motion govern the power requirements, and thus the energy usage, of the vehicle. Thus, the energy consumed by motorised vehicles is influenced by vehicle size, design and age, road and traffic conditions. The energy balance analysis here considers fuel consumption, lubricating oil consumption, tyre wear, and vehicle parts consumption.

Table C3.1: Energy Use Associated With Vehicle Production and Use

Major Sections	Sub-sections
Fuel Production	Raw material extraction
	Feed stock transportation
	Processing
	Fuel distribution
Vehicle Manufacture	Raw material extraction
	Processing
	Component manufacture
	Component transportation
	Assembly
	Vehicle distribution
Vehicle Use	Fuel consumption
	Oil consumption
	Tyre wearing
Vehicle Maintenance and Support	Component manufacture
	Distribution

Source: ETSU (1997)

FUEL CONSUMPTION

The HDM-4 fuel consumption model presented in Chapter B4 predicts the fuel use in L/1000 km for petrol and diesel vehicles using mechanistic principles. For analyses involving comparison with these vehicles, the energy consumption of alternative fuelled vehicles can be estimated by applying scale factors which relate the calculated energy consumption of petrol or diesel vehicles to the energy consumption of various alternative fuelled vehicles.

Different transport fuels have different calorific values. Thus in order to compare like with like the average fuel use should be converted using the energy content values given in Table C3.2.

Table C3.2: Energy Content of Transport Fuels

Fuel	Energy Content (MJ/litre)
Petrol	34.7
Diesel	38.7
LPG[1]	25.5
CNG[2]	40
Ethanol	23.9
Methanol	18.1
Biodiesel	32.8

Source: ETSU (1996)
Notes: 1/ Assumes 90% Propane/10% Butane
2/ Units are MJ/m^3

ETSU (1997) note that an omission of the HDM-4 fuel model is its failure to consider the additional fuel—and thus energy—associated with cold starts. This is a particular problem for petrol vehicles on short trips where a significant portion of the journey is made under cold start conditions. While for rural projects this is seldom an issue, it may be of particular significance in urban areas where there are many shorter trips. ETSU (1997) proposed that factors be established based on the proportion of the journey run under hot vs cold conditions and the relative fuel consumption under those two conditions. Factors to account for this are presented later in this chapter.

LUBRICATING OIL CONSUMPTION

As described in Chapter B13, the HDM-4 oil consumption model predicts the oil consumption in L/1000 km. This is converted to an energy value using a conversion factor of 47.7 MJ per litre.

TYRE WEAR

As described in Chapter B5, the tyre consumption in HDM-4 is predicted as the number of tyres per 1000 km. This is converted to an energy value using a conversion factor of 32 GJ/tonne of tyres (Department of Trade and Industry, 1996). The weight of tyres is estimated using the values from Table C3.3.

Table C3.3: Tyre Weights by Vehicle Type

Vehicle Type	Tyre Weights (kg)
Motorcycle	2.0
Small Car	3.0
Medium Car	3.5
Large Car	4.0
Light Delivery Vehicle	4.0
Light Goods Vehicle	4.0
Four Wheel Drive	5.0
Light Truck	7.0
Medium Truck	12.4
Heavy Truck	12.4
Articulated Truck	13.7
Mini-bus	4.0
Light Bus	7.0
Medium Bus	9.8
Heavy Bus	11.2
Coach	11.2

Source: ETSU (1996)

MAINTENANCE AND REPAIRS

The contribution of maintenance and repairs to the overall energy use is very small compared to the other components. For example, Malibach, *et al.* (1995) based on transport in Switzerland estimated maintenance and support service energy use to be approximately four per cent of the total life-cycle energy use. In developing countries it would be even less. Thus, the energy use associated with maintenance and support services would not considered to be a significant aspect of the overall energy balance.

The HDM-4 parts consumption model predicts the parts consumption in terms of the fraction of the new vehicle price per 1000 km. The energy use associated with this vehicle parts consumption is estimated by multiplying the energy used to produce the vehicle (see Table C3.4) by the fraction of the parts price to the new vehicle price. To calculate vehicle parts energy consumption per vehicle kilometre, the energy used to produce the vehicle parts should be divided by the vehicle cumulative kilometerage, at the time the parts are replaced. The typical energy used during a year can be divided by annual average vehicle kilometres to give vehicle maintenance energy use per vehicle kilometre.

Table C3.4: Energy Use for Vehicle Production

Vehicle Type	Vehicle Mass (kg)	Vehicle Production Energy (GJ)
Motorcycle	200	20
Small Car	800	80
Medium Car	1000	100
Large Car	1200	120
Light Delivery Vehicle	1400	140
Light Goods Vehicle	1600	160
Four Wheel Drive	1800	180
Light Truck	4000	400
Medium Truck	6000	600
Heavy Truck	10000	1000
Articulated Truck	15000	1500
Mini-bus	3000	300
Light Bus	5000	500
Medium Bus	7000	700
Heavy Bus	10000	1000
Coach	7000	700

Source: ETSU (1997)

C3.2.2 Fuel Production and Vehicle Manufacture Energy Use

FUEL PRODUCTION

Figure C3.1 (ETSU, 1997) illustrates the stages that must be considered for petrol and diesel fuel production. The energy use values from each stage of the fuel cycle are calculated on an energy delivered basis, and then aggregated to give the total energy use per unit of energy delivered for the full fuel-cycle. In this way energy use for fuel production can be calculated per vehicle kilometre.

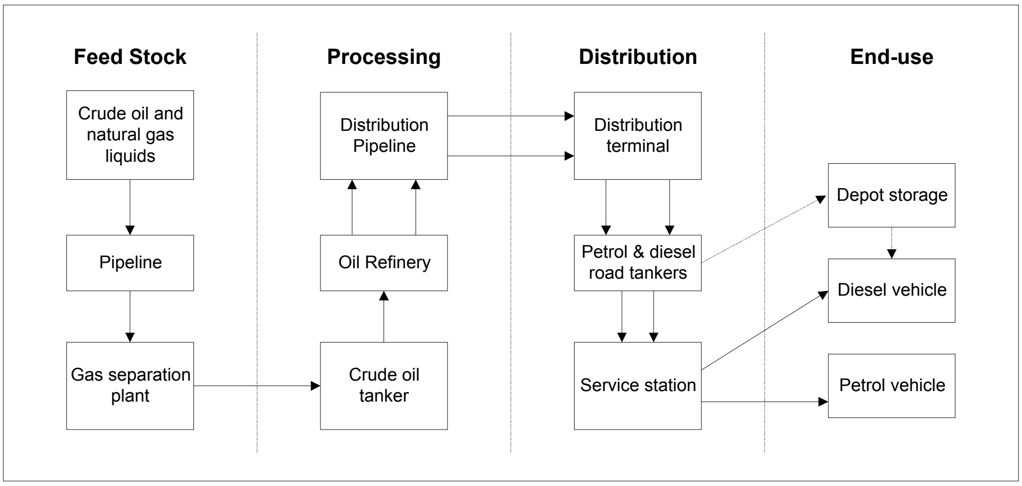


Figure C3.1: Petrol and Diesel Fuel Production Cycle

The energy use associated with each of the fuel production stages (shown in Figure C3.1 for petrol and diesel) will vary considerably from country to country. However, if no local data exist and neither do the resources to carry out a fuel-cycle analysis, then default data such as that shown in Table C3.5 could be used to estimate the energy use associated with fuel production. Fuel production energy use on a vehicle kilometre basis is derived from the energy consumption during vehicle use.

Table C3.5: Fuel-Cycle Energy Use Factors

Fuel	Energy Use MJ/MJ of fuel delivered
Petrol	0.169
Diesel	0.122
LPG[1]	0.122
CNG	0.061
Electric[2]	2.857
Biomethanol	0.514
Bioethanol	0.510
Biodiesel	0.655

Source: ETSU (1996)
Notes: 1/ Assumes 40% of LPG from refineries; 60% directly extracted
2/ Assumes a higher calorific value average generating efficiency of 35%

VEHICLE MANUFACTURE

As described in ETSU (1995), data concerning the energy used to manufacture and deliver a motor vehicle are very difficult to obtain due to the commercial sensitivities of the motor manufacturing industry. A recent survey of what data are available revealed a typical value of 100 GJ per medium-sized car with a weight of one tonne (ETSU, 1995). This represents approximately 15 per cent of total life-cycle energy use and is therefore a significant part of the overall energy balance.

To estimate the energy used to manufacture other vehicle types, a first order approximation can be obtained by scaling 100 GJ by the ratio between the weight of a medium sized car and the weight of the other vehicle. Based on this approach Table C3.4 shows some estimates of energy use for vehicle production for each of the default vehicle types in HDM-4. The energy used in the production process of each vehicle can simply be divided by the average total life-time vehicle kilometres to give vehicle production energy use per vehicle kilometre.

C3.2.3 Energy Used by Non-Motorised Vehicles

INTRODUCTION

Non-motorised traffic (NMT) account for the vast majority of the movement of people and goods in developing countries. For this reason the inclusion of NMT in the appraisal of developing country transport projects and policies is essential.

ETSU (1997) noted that with animal-drawn transport, the energy used can be justifiably included on the grounds that possibly the only reason that the animals are kept, and fed, is so that they can provide transport services to their owners. Thus the feed used can be looked upon as an energy loss equal to the opportunity cost of forfeiting its energy input for another purpose. However, with human powered transport such as cycling and walking, it is very difficult to say that the energy consumed would not have been consumed if the trip had not taken place. It is unlikely that people consciously ‘fill-up’ by consuming say 10 per cent more calories when they know they are going to be making a trip.

Nevertheless, energy is expended by walking and cycling, and it could be argued where food is a particularly scarce resource that this energy should be included as a cost to society. In addition, where individuals are using their energy for powering their business (*eg* cycle-rickshaws) and are therefore very active for most of the day, they probably need to consume several times the calories of a sedentary person.

The sections below describe the principles behind calculating energy use for various types of NMT.

ANIMAL-DRAWN VEHICLES

The energy required for animal-drawn carts can be split into two parts:- that required for the animal to walk the required distance, and, if a load is being pulled, the energy required to move that load. Data on energy use by animals are sparse. As a broad assumption, a relationship derived for human walking activity may be used: 1.8 kJ/km/kg (Replogle, 1992).

The two main loads that have to be overcome in pulling a laden cart are rolling resistance and hill climbing. Thus the energy required to pull a laden cart can be estimated from the following equation:

$$P = (FR + FG)\ v/1000 \qquad \ldots(C3.1)$$

where FR is the rolling resistance in N
FG is the gradient resistance in N
P is the power in kJ/km

Table C3.6 shows a range of energy use factors derived using this formula. The animal and load weights have been chosen to reflect those typically seen in certain parts of the world. For example the weight of a laden cart pulled by two oxen (common in more humid zones of west Africa (Starkey, 1993) is estimated to be 1000 kg (Dennis, 1995). An ox could be assumed to weigh 400 kg. For comparison a donkey pulling a 100 kg laden cart, common in the drier zones of West Africa, weighs around 150 kg (Starkey, 1993). A pack animal, commonly used in hilly areas, would tend to carry loads in the region of 30-70 kg. The values in Table C3.6 have assumed a rolling resistance for carts of 0.04 (Dennis, 1995), an average speed of 6.4 km/h and a zero gradient. The effects of gradient could be considered by using the equation from Chapter B1 for gradient resistance.

Table C3.6: Typical Energy Use Factors for Animal Transport

Animal Weight Including Load (kg)	Energy Factor in kJ/km by Cart Weight Including Load (kg)			
	No Cart	100	500	1,000
200	360	399	556	752
400	720	759	916	1,110
600	1,080	1,119	1,276	1,470
800	1,440	1,479	1,636	1,830

Source: ETSU (1997)

CYCLING

Cycling is a common mode of transport with over 800 million bicycles world-wide (United Nations, 1993). Of these, approximately 400 million are in Asia, with 300 million in China alone (Replogle, 1992).

Published figures regarding the energy used in cycling vary widely. Naturally, the specific energy use will depend upon the weight of the rider and bicycle, the friction due to the cycle and the speed of motion. Table C3.7 shows a range of cycling energy use factors quoted by Lewis (1995).

Table C3.7: Cycling Energy Use Factors

Weight Including Load (kg)	Energy Factor in kJ/km By Speed (km/h)			
	19.2	24	27.2	30.4
50	64	72	79	92
59	74	82	92	107
68	84	94	104	122
77	94	105	117	137
86	103	116	130	153
91	108	122	136	160

Source: ETSU (1997)

WALKING

Replogle (1992) estimates the energy used in walking by the formula: 1.8 kJ/km/kg. This is equivalent to 144 kJ/km for an 80 kg person.

C3.2.4 Energy Used During Road Construction and Maintenance

Energy use during the construction and maintenance of road networks is a significant aspect of the complete energy balance picture for road transport investments. Thus when comparing the energy implications of alternative policy or project options it is important that this type of energy use is considered.

ETSU (1997) were unable to obtain data on energy use during road construction and maintenance. The energy used in performing the various road works activities can be broadly considered under the following: production of materials (*eg* bitumen, cement, lime, stone-aggregates, *etc.*), delivery of materials to work sites, and the operation of equipment. However, for the purpose of energy balance analysis in HDM-4 such a detailed treatment was not be appropriate. Instead, a framework was implemented that uses aggregate level data for the energy used in performing each of the different types of road works modelled in HDM-4. For example, an average representative value of energy used per cubic metre of overlay can be specified. This value will then be multiplied by the total quantity of overlay performed on the road section to give the total energy used. It will also be necessary to distinguish between labour intensive works and mainly mechanised works.

C3.3 Energy balance framework

Based on the approach described in the previous sections, the following sections (ETSU, 1997) specify the calculations required to compare the energy consumption implications of alternative transport policy options. The outputs that are required from an energy balance analysis are:

- Total energy consumption;
- Total consumption of renewable and non-renewable energy (this is essentially a distinction between NMT energy use and all other energy use, except if biofuels are used);
- Total consumption of energy used nationally and energy used globally; and,
- Specific energy consumption.

These can be reported by vehicle types or aggregated by vehicle class.

The specific energy consumption is usually measured by the following indicators:

- Average energy use per kilometre by mode;
- Energy use per passenger kilometre for passenger transport modes; and,
- Energy use per tonne kilometre for freight transport modes.

C3.3.1 Outline of the Energy Balance Framework

The methodology for assessing the energy implications of transport policies splits into four main elements:

- Generation of the energy use characteristics for each vehicle type (both motorised and non-motorised);
- Incorporation of life-cycle effects;
- Calculation of total energy use; and,
- Generation of indicator results.

The average energy use factors are combined with the estimated total annual vehicle utilisation for each mode to give total annual energy use for the policy or measure being considered. These totals are then used to give a range of indicator results for comparative analysis.

ETSU (1997) present the overall computational logic for energy balance analysis in the form of pseudo code.

C3.3.2 Energy Use Factors for Motorised Vehicles

FUEL

Vehicle fuel use factors are split into factors for hot and cold operation. The fuel use per vehicle kilometre under 'hot' conditions comes as output from the RUE model of HDM-4.

The cold start fuel use is related to the basic hot fuel use, the ambient temperature and the average vehicle trip length. The level of cold start fuel use is related to the hot fuel use by factor CRAT, which is a function of the ambient temperature. This is expressed as follows:

$$FCOLD = CRAT\ FHOT \qquad \ldots(C3.2)$$

where FCOLD cold start fuel use in litres/1000 km
CRAT cold start ratio at a given ambient temperature
FHOT hot fuel in litres/1000 km

The hot fuel use is obtained from the equation:

$$FHOT = SFC/1000 \qquad \ldots(C3.3)$$

where SFC is the fuel consumption predicted by HDM-4 in litres/1000 km

The cold start ratios are for passenger cars for petrol and diesel technologies but are also applicable to two-wheelers and vans can be estimated from the following CORINAIR relationships (Eggleston, 1993).

$$CRAT = 1.47 - 0.009\ TEMP \quad \text{for petrol engine cars} \qquad \ldots(C3.4)$$

$$CRAT = 1.34 - 0.008\ TEMP \quad \text{for diesel engine cars} \quad \ldots(C3.5)$$

where TEMP is the average day temperature in degree celsius

Buses and trucks are usually considered to operate permanently under hot conditions, to a good approximation, since their average trip lengths are very high.

This fuel use under cold conditions only occurs in the initial stages of a journey. The proportion of any journey run cold is calculated as follows:

$$CRUN = \max [0, (0.698 - 0.051\ TL - (0.01051 - 0.000770\ TL)\ TEMP)] \quad \ldots(C3.6)$$

where CRUN is the proportion of the journey run under cold conditions
TL is the average trip length in km (default=15)

Taking into account the proportion of the journey run cold, the average vehicle fuel use per kilometre for the full trip can be calculated as follows:

$$FAVE = CRUN\ FCOLD + (1 - CRUN)\ FHOT \quad \ldots(C3.7)$$

where FAVE is the average vehicle fuel use in litres/km

The average vehicle fuel use, FAVE, is converted to energy use factor per kilometre by applying the energy content of fuel given in Table C3.2. Thus:

$$ENFUEL = FAVE\ FEC \quad \ldots(C3.8)$$

where ENFUEL is the annual average fuel energy consumption in MJ/km
FEC is the energy content of fuel type used in vehicle in MJ/litre (see Table C3.2)

LUBRICATING OIL

The annual average oil consumption in litres per 1000 km is converted to energy use factor using the energy conversion factor as follows:

$$ENOIL = OIL\ OEC/1000 \quad \ldots(C3.9)$$

where ENOIL is the annual average oil energy consumption in MJ/km
OIL is the annual average oil consumption in litres/1000 km
OEC is the energy content of lubricating oil in MJ/litre, (default = 47.7).

TYRE

The annual average number of equivalent new tyres consumed per 1000 km is converted to energy use factor as follows:

$$ENTYRE = TYRE\ TEC\ TWGT/1000 \quad \ldots(C3.10)$$

where ENTYRE is the annual average tyre energy consumption (in MJ/km)
TYRE is the annual average number of equivalent new tyres consumed per 1000 km
TWGT is the tyre weight in kg per tyre (see Table C3.3).
TEC energy content of tyre in MJ/kg (default = 32 MJ/kg).

VEHICLE REPAIR AND PARTS

The annual average parts consumption per 1000 km is converted to energy use factor as follows:

$$ENPART = PC\ ENVP/1000 \quad \ldots(C3.11)$$

where	ENPART	is the annual average parts energy consumption in MJ/km
	PC	is the parts consumption as a fraction of the replacement vehicle price
	ENVP	is the vehicle production energy use in MJ/km

and,

$$ENVP = \frac{ENVPROD}{LIFEKM} \quad \ldots(C3.12)$$

where	ENVPROD	is total energy used in the production of the vehicle in MJ (see Table C3.5).
	LIFEKM	is the vehicle service life in km

GLOBAL LIFE-CYCLE ENERGY USE FACTORS

The average annual life cycle energy use factor is given by:

$$EGLICY = ENFUEL + ENOIL + ENTYRE + ENPART + ENVP + (ENFUEL\ FP) \quad \ldots(C3.13)$$

where	EGLICY	is the annual average life cycle energy use factor in MJ/km
	FP	is the fuel production factor of the appropriate fuel type in MJ/MJ as shown in Table C3.4

The annual average global energy use per passenger-km is calculated as:

$$EGPAXKM = \frac{EGLICY}{PAX} \quad \ldots(C3.14)$$

where	EGPAXKM	is the annual average global energy use in MJ/passenger-km
	PAX	is the average number of passengers per vehicle

The annual average global energy per tonne-km is calculated as:

$$EGGDSKM = \frac{EGLICY}{PAYLD} \quad \ldots(C3.15)$$

where	EGGDSKM	is the annual average global energy use in MJ/tonne-km
	PAYLD	is the average payload per vehicle in tonnes

The average payload for each vehicle type k is calculated from the difference between the average operating weight and the tare weight as follows:

$$PAYLD = \max[0, (WGT\_OPER - WGT\_TARE)] \quad \ldots(C3.16)$$

NATIONAL LIFE-CYCLE ENERGY USE FACTORS

To incorporate certain life-cycle aspects into the energy use factors discussed earlier the following relationships should be used:

$$ENLICY = ENFUEL + ENOIL + ENTYRE + (PNP\ ENPART)$$

$$+ (PNV\ ENVP) + (PNF\ ENFUEL\ FP) \quad \ldots(C3.17)$$

where ENLICY is the annual average national life cycle energy use factor in MJ/km

PNP is the proportion of parts for vehicle produced within the country as a decimal

PNV is the proportion of vehicle produced within the country as a decimal.

PNF is the proportion of fuel type produced within the country as a decimal

The annual average national energy use per passenger-km is calculated as

$$ENPAXKM = \frac{ENLICY}{PAX} \quad \ldots(C3.18)$$

where ENPAXKM is the annual average national energy use per passenger-km in MJ/passenger-km

PAX is average number of passengers per vehicle

The annual average national energy per tonne-km is calculated as:

$$ENGDSKM = \frac{ENLICY}{PAYLD} \quad \ldots(C3.19)$$

where ENGDSKM is the annual average national energy use in MJ/tonne-km

C3.3.3 Energy Use Factors for an NMT

Energy use factors $EGLICY_k$, $ENLICY_k$ for each non-motorised transport mode k can be obtained as follows:

$$EGLICY = ENUSD\ x10^6 \quad \ldots(C3.20)$$

$$ENLICY = EGLICY \quad \ldots(C3.21)$$

where ENLICY is the annual average national life cycle energy use factor for NMT type in MJ/km

EGLICY is the annual average global life cycle energy use factor for NMT type in MJ/km

ENUSD is the average energy consumption for NMT type in Joules/km.

C3.3.4 Total Energy Use

TOTAL GLOBAL ENERGY USE

The annual global energy use for each vehicle type is calculated by multiplying the average energy use factor EGLICY by the total kilometres travelled by vehicle. Thus,

$$EGLOB = EGLICY\ VKM \qquad \ldots(C3.22)$$

where EGLOB is the annual global energy use for the vehicle type in MJ

VKM is the annual vehicle kilometres operated by the vehicle type in km

The annual total global energy use is then the sum of the energy use for each vehicle type (of both MT and NMT) plus the energy used for any road construction and maintenance in that analysis year:

$$EGTOT = \sum_{k=1}^{K} EGLOB_k + ENROAD \qquad \ldots(C3.23)$$

where EGTOT is the annual total global energy use in MJ

ENROAD is the energy used for any road construction and maintenance in the given analysis year in MJ

The energy used for any road construction and maintenance in the given analysis year is calculated from the equation:

$$ENROAD = \sum_{w=1}^{W} QTY_w\ WEF_w \qquad \ldots(C3.24)$$

where QTY_w is the amount of works type w.

WEF_w is the energy used for a unit amount of works type w in MJ/unit

The total global energy used over the analysis period for each investment option is given by the expression:

$$GLOENGY = \sum_{y=1}^{Y} EGTOT_y \qquad \ldots(C3.25)$$

where GLOENGY is the total global energy use over the analysis years y in MJ

TOTAL NATIONAL ENERGY CONSUMPTION

Energy use within a country is associated with vehicle use (both motorised and non-motorised) together with the energy use associated with any fuel, oil, vehicle and parts production that occurs within the country.

The annual national energy use for each vehicle type k is calculated by multiplying the average energy use factor ENLICY by the total kilometres travelled by vehicle. Thus,

$$ENAT = ENLICY\ VKM \qquad \ldots(C3.26)$$

where ENAT is the annual national energy use for the vehicle type in MJ

VKM is the annual vehicle kilometres operated by the vehicle type in km

The annual total national energy use is then the sum of the energy use for each vehicle type (of both MT and NMT) plus the energy used for any road construction and maintenance in that analysis year:

$$ENTOT = \sum_{k=1}^{K} ENAT_k + ENROAD \qquad \ldots(C3.27)$$

where ENTOT is the annual total national energy use in MJ
ENROAD is the energy used for any road construction and maintenance in the given analysis year in MJ

The total national energy used over the analysis period for each investment option is given by the expression:

$$NATENGY = \sum_{y=1}^{Y} ENTOT_y \qquad \ldots(C3.28)$$

where NATENGY is the total national energy use over the analysis years y in MJ

TOTAL RENEWABLE AND NON-RENEWABLE ENERGY CONSUMPTION

Assuming that renewable and non-renewable energy use can be split between energy use by NMT and MT respectively, then total renewable and non-renewable energy consumption is simply the total energy use for NMT modes and MT vehicles, respectively.

Thus, annual renewable energy is calculated as:

$$ERNW_i = \sum_{k \in NMT} EYi_k \qquad \ldots(C3.29)$$

where EYi is the annual energy use for the NMT vehicle type in MJ (i.e., $EGLOB_k$ or $ENAT_k$)
i is global (g) or national (n)

The total renewable energy (RNWTEi) used over the analysis period for each investment option is obtained by summing ERNWi over the years.

The annual non-renewable energy is calculated as:

$$ENONRW_i = \sum_{k \in MT} EYi_k \qquad \ldots(C3.30)$$

where EYi is the annual energy use for the MT vehicle type in MJ
i is global (g) or national (n)

The total non-renewable energy (NORNTEi) used over the analysis period for each investment option is obtained by summing ENONRWi over the years.

C3.3.5 Comparison of Investment Options

The true benefit of the assessment outlined above is in comparing results from before a transport measure or policy has been implemented (base case option n) with results after implementation (option m), so that the impact of the policy or measure can be seen.

The basic indicator of the performance of a measure is simply the difference between the base case results and the alternative case results:

$$\Delta ENERGYM_{(m-n)} = ENERGYM_n - ENERGYM_m \qquad \text{…(C3.31)}$$

The following indicators will be computed and implemented in the reports for each comparison of investment options m and n:

(i) Changes in the annual average global and national life-cycle energy use factors, $\Delta EGLICY_{k(m-n)}$ and $\Delta ENLICY_{k(m-n)}$, respectively, for vehicle type k, (in MJ/km)

(ii) Changes in the annual global and national energy use, $\Delta EGLOB_{k(m-n)}$ and $\Delta ENAT_{k(m-n)}$, respectively, for vehicle type k, (in MJ)

(iii) Changes in the annual average global and national energy use per passenger-km, $\Delta EGPAXKM_{k(m-n)}$ and $\Delta ENPAXKM_{k(m-n)}$, respectively, for vehicle type k, (in MJ/passenger-km)

(iv) Changes in the annual average global and national energy use per tonne-km, $\Delta EGGDSKM_{k(m-n)}$ and $\Delta ENGDSKM_{k(m-n)}$, respectively, for vehicle type k, (in MJ/tonne-km)

(v) Changes in the annual total global and national energy use, $\Delta EGTOT_{(m-n)}$ and $\Delta ENTOT_{(m-n)}$, respectively, (in MJ)

(vi) Changes in the total global and national energy use over the analysis period, $\Delta GLOENGY_{(m-n)}$ and $\Delta NATENGY_{(m-n)}$, respectively, (in MJ)

(vii) Changes in the annual total global and national renewable energy use, $\Delta ERNWg_{(m-n)}$ and $\Delta ERNWn_{(m-n)}$, respectively, (in MJ)

(viii) Changes in the annual total global and national non-renewable energy use, $\Delta ENONRNWg_{(m-n)}$ and $\Delta ENONRNWn_{(m-n)}$, respectively, (in MJ)

(ix) Changes in the total global and national renewable energy use over the analysis period, $\Delta RNWTEg_{(m-n)}$ and $\Delta RNWTEn_{(m-n)}$, respectively, (in MJ)

(x) Changes in the total global and national non-renewable energy use over the analysis period, $\Delta NORNTEg_{(m-n)}$ and $\Delta NORNTEn_{(m-n)}$, respectively, (in MJ)

D REFERENCES

AASHO (1952). *Road User Benefit Analyses for Highway Improvements.* American Association of State Highway Officials, Washington, D.C.

AASHTO (1977). *A Manual on User Benefit Analysis of Highway and Bus Transit Improvements.* American Association of State Highway Officials, Washington, D.C.

Abaynayaka, S.W., Howe, J.D.G.F., Jacobs, G.D. and Morosiuk, G. (1974) *A Study of Factors Affecting Vehicle Speeds on Rural Roads in Kenya*. Transport and Road Research Laboratory Supplementary Report 271, Department of the Environment, Crowthorne.

Abaynayaka, S.W., Morosiuk, G. and Hide, H. (1977). *The Effect of Vehicle and Road Characteristics on Commercial Vehicle Speeds in Ethiopia.* Transport and Road Research Laboratory Supplementary Report 271, Department of the Environment, Crowthorne.

Abelson, P.W. (1986). *The Economic Evaluation of Roads in Australia.* Australian Professional Publications, Sydney.

Agg, T.R. (1923). *The Economics of Highway Grades.* Iowa Engineering Experiment Station Bulletin 65. Iowa State College, Ames, Iowa.

Akcelik, R. (1990). *Calibrating Sidra*. ARRB Report ARR 180. Australian Road Research Board, Nunawading.

Akcelik, R. and Biggs, D.C. (1987). *Acceleration Profile Models for Vehicles in Road Traffic.* Transportation Science 21(1).

Akcelik, R., Biggs, D.C. and Lay, M. (1983). *Modelling Acceleration Profiles*. ARRB Internal Report AIR 390-3, Australian Road Research Board, Nunawading.

Amann, C.A. (1997). *The stretch for better passenger car fuel economy.* 1997 SAE Future Transportation Technology Conference.

American Association of State Highway Officials (1965). *A Policy on Geometric Design of Rural Highways.* AASHO, Washington, D.C.

An, F., Barth, M., Norbeck, F. and Ross, M. (1997). *Development of comprehensive modal emissions model operating under hot-stabilized conditions.* Transportation Research Record 1587, pp 52 – 62.

Anderson, S.D., Memmott, J.K. and Patil, S.S. (1992). *Road User Costs and the Influence of Pavement Type - A Perspective*. Texas Transportation Institute Research Project 0440, Austin, Texas.

Archilla, R. (1992). *The Effect of Long, Steep Downgrades on Two-lane Highway Traffic Operations.* M.Sc. Dissertation, Dept. of Civil Engineering, University of Calgary.

Armstrong, B.D. (1983). *The influences of Cool Engines on Car Fuel Consumption*. Dept. of the Environment, Transport and Road Research Loboratory Supplementary Report SR822, Crowthorne.

AUSTROADS (1989). *Guide to the Geometric Design of Rural Roads*. AUSTROADS, Sydney.

Autotech (1993). *Automobile Emissions and Combustion*. IMechE. p116

Bang, K. (1995). *Impact of Side Friction on Speed-Flow Relationships for Rural and Urban Highways.* Report to the ISOHDM Study, SWEROAD, Bandung, Indonesia.

Barnes, J. (1988). *A Speed Management Strategy for New Zealand Roads.* Proc. Road Traffic Safety Research Council Volume 2, Wellington.

Barnes, J. and Edgar, J. (1984). *Vehicle Speeds on Rural Roads.* Proc. 1984 IPENZ Conference 10(1), pp. 104-34, Auckland.

Barnes, J. and Edgar, J. (1987). *The Effect of the 1985 Rise in Open Road Speed Limits on Vehicle Speeds.* Proc. 1987 Roading Symposium, Transit N.Z., Wellington.

Barry, T.M. and Regan, J.A. (1978). *Traffic Noise Prediction Model*. FHWA Report No. FHWA-RD-77-108, Federal Highways Administration, Washington, D.C.

Bein, P. (1993). *Evaluation of State-of-the-art VOC Models.* Road & Transport Research 2(3), Australian Road Research Board, Nunawading.

Bennett, C.R. (1985). *A Highway Economic Appraisal Model for New Zealand*. Department of Civil Engineering Report, University of Auckland, Auckland.

Bennett, C.R. (1988). *Truck Speeds on Grades.* Working Paper RTRG 11-003, Roads and Transportation Research Group, Department of Civil Engineering, University of Auckland.

Bennett, C.R (1989a). *The New Zealand Vehicle Operating Cost Model*. RRU Bulletin 82, Transit New Zealand, Wellington.

Bennett, C.R. (1989b). *Motor Vehicle Depreciation in New Zealand.* RRU Occassional Paper, Transit New Zealand, Wellington.

Bennett, C.R (1993). *Revision of Project Evaluation Manual Speed Change Cycle Costs.* Report for Transit New Zealand, Wellington.

Bennett, C.R. (1994). *A Speed Prediction Model for Rural Two-lane Highways*. *School of Engineering Report*, Department of Civil Engineering, University of Auckland.

Bennett, C.R. (1995a). *Review of Experience Adapting the HDM Vehicle Operating Costs Model*. Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Bennett, C.R. (1995b). *Modelling Capital Costs in HDM-4*. Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Bennett, C.R. (1996a). *1995 HDM-4 Workshop on Road User Effects - Implementation of Recommendations.* Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Bennett, C.R. (1996b). *Modifications to the HDM-4 Parts Consumption Model*. Memo to the ISOHDM Secretariat, University of Birmingham.

Bennett, C.R. (1996c). *Modelling Capital Costs in HDM-4.* Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Bennett, C.R. (1996d). *1996 HDM-4 Workshop on Road User Effects - Briefing Paper*. Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Bennett, C.R. (1997) *HDM-4 Rolling Resistance Model*. Memo to the ISOHDM Study, University of Birmingham, U.K.

Bennett, C.R. (1998a). *Structured Testing of the HDM-4 Road User Effects Model for Australia*. Report to the Queensland Main Roads Department. HTC Infrastructure Management Ltd., Auckland.

Bennett, C.R. (1998b). *Modelling the Effects of Work Zones in HDM-4: Revised Methodology and Specifications*. Report ISOHDM 98/7/1 to the International Study of Highway Development and Management Tools, HTC Infrastructure Management Ltd., Auckland.

Bennett, C.R. (1998c). *Revision to HDM-4 Parts Consumption Model Parameters*. Report to the ISOHDM Study, HTC Infrastructure Management Ltd., Auckland.

Bennett, C.R. and Dunn, R.C.M. (1989). *A Note on the ARFCOM Fuel Model*. Australian Road Research 19(3) pp. 243-245.

Bennett, C.R. and Dunn, R.C.M. (1994). *An Evaluation of the AUSTROADS Horizontal Curve Design Standards for New Zealand.* Road and Transport Research Vol 3, pp. 54-63, Australian Road Research Board Ltd., Nunawading.

Bennett, C.R. and Greenwood, I.D. (1996a). *Modelling Non-Motorised Transport in HDM-4: Proposed Interim Methodology*. Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Bennett, C.R. and Greenwood, I.D. (1996b). *Proposed Methodology for Modelling Slow Moving Transport in HDM-4*. Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Bennett, C.R. and Paterson, W.D.O. (1999). *A Guide to Calibration and Adaptation.* HDM-4 Reference Series Volume 5, PIARC.

Bergman, W. and Crum, W.B. (1973). *New Concepts of Tyre Wear Measurement and Analysis*". Society of Automotive Engineers, SAE Transactions 730615b.

Bester, C.J. (1981). *Fuel Consumption of Highway Traffic*. Ph.D. Dissertation, University of Pretoria.

Biggs, D.C (1987). *Estimating Fuel Consumption of Light to Heavy Vehicles*. ARRB Internal Report AIR 454-1, Australian Road Research Board, Nunawading.

Biggs, D.C (1988). *ARFCOM - Models For Estimating Light to Heavy Vehicle Fuel Consumption*. Research Report ARR 152, Australian Road Research Board, Nunawading.

Biggs, D.C (1990). *Comparison of Observed Heavy Vehicle Fuel Consumption in Canada with Estimates Produced by ARFCOM*. Australian Road Research Board, Nunawading.

Biggs, D.C (1995). Correspondence with ISOHDM Study.

Biggs, D.C. and Akcelik, R. (1985). *Further Work on Modelling Car Fuel Consumption.* ARRB Internal Report AIR 390-10, Australian Road Research Board, Nunawading.

Bitumen and Tar Association and the Division of Roads and Transport and Technology (1989). *Economic Warrants for Surfacing Roads*, SURF Manual 7, Pretoria, 1989.

Bonney, R.S.P. and Stevens, N.F. (1967). *Vehicle Operating Costs on Bituminous, Gravel and Earth Roads in East and Central Africa*. Road Research Technical Paper No. 76, Ministry of Transport, London.

Bowyer, D.P., Akcelik, R. and Biggs, D.C. (1986). *Guide to Fuel Consumption Analyses for Urban Traffic Management.* Special Report 32, Australian Road Research Board, Nunawading.

Boyce, A.M., McDonald, M., Pearce, M.J., and Robinson, R. (1988). *A Review of Geometric Design and Standards for Rural Roads in Developing Countries. Transport and Road Research Laboratory Contractors Report 94,* Department of Transport, Crowthorne.

Brodin, A. and Carlsson, A. (1986). *The VTI Traffic Simulation Model*. VTI Report 321A, Linkoping.

BTCE (1996). *Costs of Reducing Greenhouse Gas Emissions From Australian Cars: An application of the BTCE CARMOD Model.* Working Paper 24, Bureau of Transport and Communications Economics, Canberra, Australia.

Bunce, J.A. and Tressider, J.O. (1967). *The Characteristics of Rural Roads in Jamaica a they Affect Traffic Speeds and Vehicle Operating Costs.* Institution of Civil Engineers, London.

Burke, C.E., Nagler, L.G, Lundstrom, L.C and Associates (1975). *Where Does All the Power Go?* SAE Transactions, Vol. 65.

Butler, B.C. (1984). *Use-Related Vehicle Depreciation*. Transportation Research Record 955, pp. 1-9, Transportation Research Board, Washington, D.C.

Carpenter, P. and Cenek, P.D. (1999). *Tyre Wear Modelling for HDM-4*. Central Laboratories Report 98-529474 to Transit NZ, Opus Central Laboratories, Lower Hutt.

Cenek, P.D. (1994). *Rolling Resistance Characteristics of New Zealand Roads*. Transit New Zealand Research Report PR3-001, Wellington.

Cenek, P.D. (1995). Correspondence with ISOHDM Study.

Cenek, P.D. (1999). Correspondence with ISOHDM Study.

Cenek, P.D. and Carpenter, P. (2000). *Correspondence with Authors*.

Cenek, P.D., Carpenter, P., Jamieson, N.J. and Stewart, P.F. (1996). *Investigation of the Relationship Between Frictiona nd Tyre Abrasion Characteristics of Road Surfaces*. Transit New Zealand Report PR3-0042, Opus Central Laboratories, Lower Hutt.

Cenek, P.D. and Jamieson, N.J. (1999). *Vehicle Operating Costs - Relationships with NAASRA Roughness*. Central Laboratories Report 99-529276.00, Opus Central Laboratories, Lower Hutt.

Cenek, P.D., Jamieson, N.J., Sutherland, A.J. and Swift, A. (1993). *Tyre Consumption Research Plan*. Transit New Zealand Research Report PR3-0042, Opus Central Laboratories, Lower Hutt.

Chesher, A.D. (1990). *Optimal Scrapping, Vehicle Operating Costs and the Benefits of Highway Improvements.* In J-P. Florens, M. Ivaldi, J-J. Laffont and F Laisney, eds. Microeconometrics, Surveys and Applications. Oxford: Basil Blackwell.

Chesher, A.D. (1995). *Optimal Scrapping With Varying Utilisation*. Paper to HDM-4 Workshop on Road User Effects, TRL, Crowthorne.

Chesher, A.D. and Harrison, R. (1987). *Vehicle Operating Costs: Evidence from Developing Countries.* World Bank Publications, Johns Hopkins Press.

Chesher, A.D., *et al*. (1980). *Vehicle Depreciation and Interest Costs: Some Evidence from Brazil.* Proc. World Conference on Transport Research, London.

Ching, P.Y. and Rooney, F.D. (1979a). *Truck Speeds on Grades in California.* Report FHWA-CA-TO-79-1, Federal Highways Administration, Washington, D.C.

Claffey, P. (1971). *Running costs of Motor Vehicles as Affected by Road Design and Traffic*. NHCRP Report 111, Highway Research Board, Washington, D.C.

Clean Cat (2000). *Clean Cat Diesel Engine Catalytic Converters.* http://clean-cat.com.

Cohen, Cohen, S., Cottinet, M. and Texier, P. (1994). *Road Operating Levels, Capacity and Level of Service.* Second International Symposium on Highway Capacity, 1994. Akcelik, R. and W. Reilly (ed.), Country Reports, pp. 57-66.

Collings, S. (1999). *Energy Balance Specifications.* Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Cooper, D.R.C., Jordan, P.G., and Young, J.C. (1980). *The Effect on Traffic Speeds of Resurfacing a Road.* TRRL Supplementary Report 571. Transport and Road Research Laboratory, Crowthorne.

Cox, B.E. (1983). *The Value of Travel Time*. RRU Occasional Paper. Transit New Zealand, Wellington.

Cox, J.B. (1994). *A Fleet Vehicle Operating Cost Model to More Accurately Determine Benefits of the Analysis Period.* Paper to the International Workshop on HDM-4, Kuala Lumpur. International Study of Highway Development and Management Tools, University of Birmingham.

Cox, J.B. (1995). *Models for Predicting Vehicle Operating Costs in Urban Conditions.* Report to AUSTROADS, ARUP Transportation Planning, Melbourne.

Cox, J.B. (1996a). *Models for Predicting Vehicle Operating Costs in Urban Areas.* Report to Austroads Project BS3.A.41. J.B. Cox and Arup Planning, Melbourne.

Cox, J.B. (1996b). *Effect of Road Surface condition on Vehicle Operating Costs in Australia: Literature Review and Fleet Database Analysis.* Report to AUSTROADS. Symonds Travers Morgan Pty Ltd., Melbourne.

CRRI (1982). *Road User Cost Study in India*. Central Road Research Institute, New Delhi.

CRRI (1985). *Traffic Simulation Modelling Study: Part 1 - Development of Simulation Models.* Central Road Research Institute, New Delhi.

Cundill, M. (1993). *RTIM3 Manual.* Transport Research Laboratory, Crowthorne.

Cundill, M. (1995a). *Proposed Relationships for Vehicle Maintenance and Repair Costs for Use in the HDM-4 Model*. Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Cundill, M. (1995b). *Recommended Relationships for Parts Modelling in HDM-4*. Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Curtayne, P.C., Visser, A.T., du Plessis, H.W. and Harrison, R. (1987). *Calibrating the Relationship Between Operating Costs of Buses and Road Roughness on Low-Volume Roads*. Transportation Research Record 1106, pp. 95-104, Transportation Research Board, Washington, D.C.

Daniels, C. (1974). *Vehicle Operating Costs in Transport Studies With Special Reference o the Work of the EIU in Africa*. Economist Intelligence Unit Technical Series No. 1, London.

de Weille, J. (1966). *Quantification of Road User Savings*. World Bank Staff Occasional Paper No. 2. The World Bank, Washington, D.C.

Dennis R. (1995). IT Transport, Personal Communication with ETSU.

Department of Trade and Industry (1996). *Digest of United Kingdom Energy Statistics, 1996*. HMSO, London.

Dhareshwar, A. (1983). *Bus and Truck Maintenance Parts Consumption: Preliminary Report on the Estimation of the Aggregate Relationship Based on Mechanistic Principles.* Office Memorandum 14 July 1983, the World Bank.

Diaz, J., Sergio R, and Gonzalez, S.M. (1983*). Flexible Models for Accidents on Chilean Roads*. Unpublished paper, June 1983.

DOT (1993). *COBA User's Manual*. Department of Transport, U.K.

DOT (1994). Cost-benefit Analysis of Rural Road Projects : Program CB-ROADS User's Guide. Department of Transport Pretoria.

Discount Converters Ltd. (2000). *Diesel Catalytic Converters*. http://www.discountconverter.com.

du Plessis, H.W. editor (1989). *An Investigation of Vehicle Operating Cost Relationships for use in South Africa.* In: Road Roughness Effects on Vehicle Operating Costs - Southern Africa Relations for Use in Economic Analyses and Road Management Systems. CSIR, Pretoria.

du Plessis, H.W. and Harrison, R. (1987). *Vehicle Tyre Wear as Affected by Road Condition*. Technical Note TM/23/87, National Institute for Road and Transport Research, CSIR, Pretoria..

du Plessis, H.W. and Meadows, J.F. (1990). *A Pilot Study to Determine the Operating Costs of Passenger Cars as Affected by Road Roughness.* CSIR Research Report DPVT-142, CSIR, Pretoria.

du Plessis, H.W., Morden, C.W., and Coetzee, C.H. (1989). *A Pilot Study to Determine the Effect of Road Surface Roughness on Vehicle Speeds.* In Road Roughness Effects on Vehicle Operating Costs: Southern Africa Relations for use in Economic Analyses and in Road Management Systems. South African Roads Board Project Report 88/070/3, CSIR, Pretoria.

du Plessis, H. and Schutte, I.C. (1991). *Road Roughness Effects on Vehicle Operating Costs: Southern Africa Relations for use in Economic Analyses and in Road Management Systems.* South African Roads Board Project Report 88/070/3, CSIR, Pretoria.

du Plessis, H.W., Visser, A.T. and Curtayne, P.C. (1990). *Fuel Consumption of Vehicles as Affected by Road-surface Characteristics.* Surface Characteristics of Roadways : International Research and Technologies, ASTM STP 1301, W.E. Meyer and J. Reichert, Eds., American Society for Testing and Materials, Philadelphia: 480-496.

Dudek, C.L. and Richards, M. (1981). *Traffic Capacity Through Work Zones on Urban Freeways.* Research Report 228-6, Texas Transportation Institute, College Station, Texas.

ECU: Commission of the European Communities Directorate Generale for Transport (1994). *Concerted Action 1.1 Cost-Benefit and Multi-criteria Analysis for New Road Construction, Final Report*. EURET, April 1994.EFRU (1997). *Vehicle Exhaust Gas*

Emissions – SMF Project 5034 Final Report. Energy and Fuels Research Unit, Auckland UniServices Report 6386.23.

EFRU (1997). *Vehicle Exhaust Gas Emissions - SMF Project 5034 Final Report*. Energy and Fuels Research Unit, Auckland UniServices Report 6386.23. University of Auckland, Auckland.

Eggleston, H.S., Gaudioso, D., Gorissen, N., Jourmard, R., Rijkeboer, R.C., Samaras, Z., and Zierock, K-H. (1993) *CORNAIR Working Group on Emission factors for Calculating 1990 Emissions from Road Traffic. Volume 1: Methodology and Emission Factors*. B4-3045 (91) 10PH, Commission of the European Communities, Brussels.

Elder, S.T. (1983). *Vehicle Fuel Consumption and Performance Test Methods*. Thesis, Department of Mechanical Engineering, University of Auckland, Auckland.

Elkins, G.E. and Semrau, J. (1988). *Development of Limiting Velocity Models for the Highway Performance Monitoring System.* Transportation Research Record 1195, pp. 138-140, Transportation Research Board, Washington, D.C.

Emmerson, J. (1970). *A Note on Speed-Road Curvature Relationships.* Traffic Engineering and Control 12(7), pp. 369.

EPA (1997). *Download from US Environmental Protection Agency web site*. http:www.epa.gov/OMSWWW/annem95.htm.

ETSU (1995). *Life-Cycle Analysis of Motor Fuel Emissions.* Final report to COST 319 Sub-group A4.C. ETSU Ref RYCA/18691001/Issue 1.

ETSU (1996). *Alternative Road Transport Fuels - A Preliminary Life-Cycle Study for the UK.* HMSO, London, March 1996.

ETSU (1997) *Emissions Modelling Framework for HDM-4.* Working Paper for Discussion with University of Birmingham and ODA. ETSU REF:RYCA 18825001/wp2/Issue 1. 4 November 1997.

Findlayson, A.M. and du Plessis, H.W. (1991). *Operating Costs of Medium to Heavy Trucks as Affected by Road Roughness*. CSIR Research Report PR/88/070/3, CSIR, Pretoria.

Ford, W.G. (1977). *Prediction of Vehicle Speeds on Roads in the Lake District.* TRRL Supplementary Report 230. Transport and Road Research Laboratory, Crowthorne.

Francher, P.S. and Winkler, C.B. (1984). *Retarders for Heavy Vehicles: Phase III Experimentation and Analysis; Performance, Brake Savings and Vehicle Stability.* US Department of Transportation, Report no. DOT HS 806672.

Galin, D. (1981). *Speeds on Two-Lane Rural Roads - A Multiple Regression Analysis.* Traffic Engineering and Control 22(8/9), pp. 453-460.

Gambard, J. and Louah, G. (1986). *Free Speed as a Function of Road Geometrical Characteristics.* Proc. P.I.R.A.C. Meeting, Seminar R.

Gardner, N.G. (1978). *The Effects of Reduced Speed Limits on Motor Vehicle Operating Speeds in Ontario.* Report to Ministry of Transportation and Communications, Downsview.

GEIPOT (1981). *Research on the Interrelationships Between Costs of Highway Construction Maintenance and Utilisation: Final Report on Brazil-UNDP Highway Research Project (12 Volumes).* Brasilia, Brazil.

Gelling, I.R. (1994). *Influence of Tread Polymer on Traction, Rolling Resistance, and Wear Properties of Tyres*. Vehicle-Road Interaction ASTM STP 1225, B.T. Kulakowski Ed., American Society for Testing and Materials, Philadelphia, 1994, pp. 107-118.

Good, M.C. (1978). *Road Curve Geometry and Driver Behaviour.* Australian Road Research Board Special Report 15, Australian Road Research Board, Nunawading.

Greenwood, I.D. (1996). *Fuel Consumption Model – Peng and Paccs*. Correspondence with the International Study of Highway Development and Management Tools. University of Birmingham.

Greenwood, I.D. (1997). *Predicting Tyre Diameter*. Memo to the International Study of Highway Development and Management Tools. University of Birmingham.

Greenwood, I.D. (1998). *Correspondence with the International Study of Highway Development and Management Tools.* University of Birmingham.

Greenwood, I.D. (1999). *Calibration of HDM-4 Congestion Model.* Report to Dessau-Sopran, Thailand Motorway Project, Bangkok.

Greenwood, I.D. (2000). *Modelling the Effects of Congestion on Fuel Consumption and Vehicle Emissions.* PhD Thesis, University of Auckland (draft).

Greenwood, I.D. and Bennett, C.R. (1995). *HDM-4 Fuel Consumption Modelling*. Preliminary Draft Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Greenwood, I.D., Bennett, C.R., and Rahman, A. (1995). *Effects of Pavement Maintenance on Road Users.* Preliminary Draft Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Gynnerstedt, G., Carlsson, A. and Westerlund, B (1977). *A Model for the Monte Carlo Simulation of Traffic Flow Along Two-Lane Single-Carriageway Rural Roads.* Swedish National Road and Traffic Research Institute Report 43, Swedish Road and Traffic Research Institute, Linköping.

Halcrow Fox (1982). Highway and Road Transport Consultancy Project - Volume 3: Vehicle Operating Cost Model Study. Draft Final Report to the Department of Communications, Jakarta.

Heavenrich, R.M., Murrell, J.D. and Hellman, H.K. (1991). *Light-duty Automotive Technology and Fuel Economy Trends Through 1991.* Control Technology and Applications Branch, EPA/AA/CTAB/91-02, US Envrironmental Protection Agency, Ann Arbor, Michigan.

Hall, F., Smith, W.S., Montgomery, F. and Brilon, W. (1994). *Comparison of Speed-Flow Relationships Based on Data from North America, the United Kingdom, and Germany.* Proc. Second International Symposium on Highway Capacity, 1994. Akcelik, R. (ed.), Volume 1, pp. 243-264.

Hammarström, U. and Karlsson, B. (1987). *VETO A Computer Program for Calculating Transport Costs as a Function of Road Standard.* VTI Report 501, Swedish Road and Traffic Institute, Linkoping.

Hammarström, U. and Karlsson, B. (1991). *VETO A Computer Program for Calculating Transport Costs as a Function of Road Standard.* English Translation of VTI Report 501, Swedish Road and Traffic Institute, Linkoping.

Hammarström and Henrikson (1994). *Reparationskostnader för bilar Kalibrering av Varldsbankens HDM-III-samband för svenska förhallanden*. VTI Meddelande Nr. 743, Swedish Road and Traffic Institute, Linköping.

Hammarström, U. (1994). *Description of VETO, EVA and HDM III With Regard to Vehicle Effects and Proposals for HDM-4.* Internal Report, VTI, Linköping.

Hammarström, U. (1999). *Vehicle Emission Deterioration Factors.* Personal communication via email Hammarström to Greenwood, 19 October 1999.

Hammarström, U. (2000). *Emission Predictions*. Personal communication via e-mail Hammarström to Greenwood, 9 March 2000.

Harrison, R. and Aziz, S. (1998), *HDM-4 Tyre Relationships.* Memorandum to ISOHDM Secretariat. Centre for Transportation Research, The University of Texas at Austin, USA.

Heywood, J. (1988). *Internal Combustion Engine Fundamentals.* McGraw-Hill International Press.

Heywood, J (1997). *Motor Vehicle Emissions Control: Past Achievements, Future Prospects.* 1997 George Stephenson Lecture Notes, Institute of Professional Engineers of New Zealand.

Hide, H. (1982). *Vehicle Operating Costs in the Caribbean: Results of a Survey of Vehicle Operators.* Transport and Road Research Laboratory Report 1031, Department of the Environment, Crowthorne.

Hide, H., Abaynayaka, S.W., Sayer, I., Wyatt, R.J. (1975). *The Kenya Road Transport Cost Study: Research on Vehicle Operating Costs.* Transport and Road Research Laboratory Report LR672, Department of the Environment, Crowthorne.

Hills, P. J. and Jones-Lee, M.W. (1981). *The Costs of Traffic Accidents and Evaluation of Accident Prevention in Developing Countries*. PTRC Summer Annual Meeting, University of Warwick, July 1981.

Hine, J.L. (1982). *Some Notes on an Optimisation Model of Vehicle Utilisation and Life and RTIM2.* Internal Report, Overseas Unit, Transport and Research Laboratory, Crowthorne.

Hine, J. (1994). *Pakistan Road Freight Industry: The Productivity and Time Use of Commercial Vehicles*. TRRL Research Report 333, Transport Research Laboratory, Crowthorne.

Hine, J.L. (1995). *Presentation to TRL Workshop on Road User Effects*. Transport Research Laboratory, Crowthorne.

Hine, J.L. (1996). *Briefing Paper on Vehicle Utilisation and Service Life for the December 1996 Workshop on HDM-4.* Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Hine, J. and Pangihutan, H. (1998). *Speed Profiles and Fuel Consumption: A Study of a Congested Road in Java*. Project Report PR/OSC/132/98, Transport Research Laboratory, Crowthorne.

HMSO (1988). *Calculation of Road Traffic Noise.* Department of Transport, Welsh Office.

Hoban, C.J. (1987). *Evaluating Traffic Capacity and Improvements to Road Geometry.* World Bank Technical Paper 74, The World Bank, Washington, D.C.

Hoban, C.J., Fawcett, G.J. and Robinson, G.K. (1985). *A Model For Simulating Traffic on Two-Lane Rural Roads: User Guide and Manual for TRARR Version 3.0.* Technical Manual ATM 10A, Australian Road Research Board, Nunawading.

Hoban, C.J., Reilly, W. and Archondo-Callao, R. (1994). *Economic Analysis of Road Projects with Congested Traffic.* World Bank Publications, Washington, D.C.

Hodges, J.W., Rolt, J. and Jones, T.E. (1975). *The Kenya Road Transport Cost Study: Research on Road Deterioration.* TRL Report LR 673, Department of the Environment, Crowthorne.

Hoff & Overgaard (1992). *Road User Cost Model.* Second Technical Advisory Services on Planning and Programming to the Directorate of Planning. Indonesia.

Hoff & Overgaard (1994). *Kabupaten Roads Economic Evaluation Model - Vehicle Operating Costs.* Draft Report. Directorate General of Highways, Ministry of Public Works, Government of Indonesia.

HTC (1999a). *Calibration of HDM Speed Prediction Model to Thailand*. Report DES/99/2 to Dessau-Sopran, Thailand Motorway Project. HTC Infrastructure Management Ltd., Auckland.

HTC (1999b). *Preparation of Revised Project Evaluation Manual VOC Tables*. Report TF/99/3. HTC Infrastructure Management Ltd., Auckland.

HTC (1999c). *Implementation of Predictive Modelling for Road Planning: Phase I Final Report of the NZ dTIMS Study*. HTC Infrastructure Management Ltd., Auckland.

Hughes, P.S. (1992). *A Strategy for Reducing Emissions of Greenhouse Gases from Personal Travel in Britain.* PhD Thesis Open University. 1992.

Ingram, K.C., (1978). *The Wind-averaged Drag Coefficient Applied to Heavy Goods Vehicles*. TRRL Supplementary Report 392, Crowthorne, Berkshire.

IRC (1993). *Manual on Economic Evaluation of Highway Projects in India.* IRC Special Publication 20, Indian Roads Congress, New Delhi.

ISOHDM (1995a). *Traffic Safety Position Paper*. Report to the International Study of Highway Development and Management Tools, University of Birmingham.

ISOHDM (1995b). *Proceedings of the International Workshop on HDM-4.* International Study of Highway Development and Mangement Tools, University of Birmingham.

ISOHDM (1997). *Specifications for NMT Modelling in HDM-4*. Internal Report. International Study of Highway Development and Management Tools, University of Birmingham, U.K.

ITE (1976). *Transportation and Traffic Engineering Handbook*. Prentice Hall, New Jersey.

Jackson, G.C. and Maze, T.H. (1983). *Vehicle Life-Cycle costing with Probabilistic Part Replacement and Repair Options*. Transportation Research Record 912, pp. 15-18, Transportation Research Board, Washington, D.C.

Jacobs, G D (1976). *A Study of Accident Rates on Rural Roads in Developing Countries*. TRRL Laboratory Report 732, Crowthorne, 1976.

Jamieson, N.J. and Cenek, P.D. (1999). *Effects of Pavement Construction on the Fuel Consumption of Trucks.* Proc. Options for Post-Millennium Pavements. NZ Institute of Highway Technology Symposium, Taupo.

Johansson, O. (1994). *International Study of Highway Development and Management Tools, Safety Analysis Framework I,II, III*. Report to the International Study of Highway Development and Management Tools, the University of Birmingham.

Jordaan, P. (1995). *Some Aspects of the HDM-III Evaluation of the Mbabane-Ngwenya Road in Swaziland.* Proc. International Workshop on HDM-4, University of Birmingham.

Kadiyali (1991). *Updating of Road User Cost Study.* Report to the Ministry of Surface Transport, L.R. Kadiyali and Associates Ltd., Delhi.

Kadiyali, L.R., Viswanathan, E., Janin, P.K., Bajpan, R.K. and Sharma, V.P.(1981). *A Comparative Study of Fuel Consumption With Radial vs Bias Nylon Tyres*. Road Research Paper 171, Central Road research Institute, Okhla, New Delhi, India.

Kadiyali, L.R. and Ummat, S.K. (1980). *Tyre Management in Road Transport*. Road Research Paper 168, Central Road Research Institute, New Delhi, India.

Karan, M.A., Haas, R., and Kher, R. (1976). *Effects of Pavement Roughness on Vehicle Speeds.* Transportation Research Record 602, pp. 122-127, TRB, Washington, D.C.

Kerali, H.G.R., Odoki, J.B., and Collings, S. (1997). *Energy Balance Framework for Road Transport Analysis.* Transportation Research Board, Washington DC, January 1998, Paper No. 980819.

Kerali, H.G.R., Odoki, J.B., and Wightman, D.C. (1994), *Vehicle Fleet Representation, Draft Working Paper*. International Study of Highway Development and Management Tools, Overseas Development Administration, Asian Development Bank, Swedish National Road Administration, and The World Bank.

Kerman, J.A., McDonald, M., and Mintsis, G.A. (1982). *Do Vehicles Slow Down on Bends? A Study Into Road Curvature, Driver Behaviour and Design.* Proc. Seminar K P.T.R.C. Meeting.

Knoroz, V.I and Shelukhin, A.S. (1964). *Test Data on the Rolling Resistance of Vehicle Tires on Roads with a Hard Surface*. Moscow Automotive Prom., Vehicle Research Institute, No 2, pp. 16-21.

Kolsrud, B (1985a). *Speeds in Rural Traffic. The Influence of Various Factors on Cars Speeds on Straight, Level Roads.* VTI Research Report 4(1), pp. 6-8, Swedish Road and Traffic Research Institute, Linköping.

Kolsrud, B, (1985b). *Speeds in Rural Traffic. Compliance With Displayed Speed Limits.* VTI Research Report 4(1), pp. 9-10, Swedish Road and Traffic Research Institute, Linköping.

Kosasih, D., Robinson, R., and Snell, J. (1987). *A Review of Some Recent Geometric Road Standards and Their Application to Developing Countries.* Transport and Road Research Laboratory Research Report 114, Department of Transport, Crowthorne.

Lamm, R., Hayward, J.C. and Cargin, J.G. (1986). *Comparison of Different Procedures for Evaluating Speed Consistency.* Transportation Research Record 1000, pp. 10-20, Transportation Research Board, Washington, D.C.

Le Maître, O, Süssner, M. and Zarak, C. (1998). *Evaluation of Tyre Wear Performance*. Society of Automotive Engineers Paper 980256.

Lea Associates (1988). *Saskatchewan Highways and Transportation Pavement Management Information System: PMIS Technical Manual Appendix I - Vehicle Operating Costs.* Lea Associates Ltd., Vancouver, B.C., Canada.

Lefeve, BA. (1953). *Speed Characteristics on Vertical Curves.* Proc. HRB, 32, pp. 395-413, Highway Research Board, Washington, D.C.

Leong, H.J.W. (1968). *The Distribution and Trend of Free Speeds on Two-Lane Two-Way Rural Highways in New South Wales.* Proc ARRB Conference 4(1)*,* pp. 791-814, Australian Research Board, Nunawading.

Lewis, C.A. (1995). *Energy Use in Bicycle and Animal-Drawn Transport in Developing Countries.* ETSU Working Paper 18400304. 1995.

Los, T.J. (1974). *Relationship Between Free Speed and Sight Distance on Two Way Roads.* BE. Thesis, Department of Civil Engineering, University of Queensland.

LTSA (2000). *New/used car registrations up for the year.* Media Release 11 January 2000. Land Transport Safety Authority, New Zealand.

Macdonald, R.A. (1987). *The Depreciation Component in Vehicle Operating Costs: A Review of Existing Methods of Estimation.* MSc (Eng) Dissertation, Department of Transportation and Highway Engineering, University of Birmingham.

Maibach, M., Peter, D. and Seiler, B. (1995). *Okoinventar Transporte.* Verlag Infras, Zurich.

Mannering, F.L. and Kilareski, W.P. (1990). *Principles of Highway Engineering and Traffic Analyses.* John Wiley and Sons, Brisbane.

Manning, F.L and Kilareski, W.P. (1990). *Principles of Highway Engineering and Traffic Analyses*. John Wiley and Sons, Brisbane.

McGuigan (1987). *An Examination of Relationships Between Road Accidents and Traffic flow*. Ph.D. Thesis, University of Newcastle upon Tyne, England, 1987.

McLean, J.R. (1974b). *Driver Behaviour on Curves - A Review.* Proc. ARRB Conference 7(5), pp. 129-147, Australian Road Research Board, Nunawading.

McLean, J.R. (1976). *Vehicle Speeds on High Standard Curves.* Proc. ARRB Conference 8(21), pp. 1-7, Australian Road Research Board, Nunawading.

McLean, J.R. (1978a). *Speeds on Curves - Preliminary Data Appraisal and Analysis.* Australian Road Research Board Internal Report AIR 200-1, Australian Road Research Board, Nunawading.

McLean, J.R. (1978b). *Speeds on Curves - Preliminary Data Appraisal and Analysis.* Australian Road Research Board Internal Report AIR 200-2, Australian Road Research Board, Nunawading.

McLean, J.R. (1978c). *Speeds on Curves: Regression Analysis.* Australian Road Research Board Internal Report AIR 200-3, Australian Road Research Board, Nunawading.

McLean, J.R. (1978d). *Speeds on Curves: Side Friction Factor Considerations.* Australian Road Research Board Internal Report AIR 200-4, Australian Road Research Board, Nunawading.

McLean, J.R. (1978e). *Observed Speed Distributions and Rural Road Traffic Operations.* Proc. ARRB Conference 9(5), pp. 235-44, Australian Road Research Board, Nunawading.

McLean, J.R. (1978f). *Review of the Design Speed Concept.* Australian Road Research 8(1), pp. 3-16, Australian Road Research Board.

McLean, J.R. (1979). *An Alternative to the Design Speed Concept for Low Speed Alignment Design.* Australian Road Research Board Internal Report AIR-200-5A, Australian Road Research Board, Nunawading.

McLean, J.R. (1981). *Driver Speed Behaviour and Rural Road Alignment Design.* Traffic Engineering & Control 22(4), pp. 208-211.

McLean, J.R. (1982a). *Two-Lane Road Traffic Flow and Capacity: Chapter 10 - Empirical Research*. Australian Road Research Board Internal Report AIR-359-6, Australian Road Research Board, Nunawading.

McLean, J.R. (1988). *Speeds, Friction Factors, and Alignment Design Standards.* Australian Road Research Board Report ARR 154, Australian Road Research Board, Nunawading.

McLean, J.R. (1989). *Two-Lane Highway Traffic Operations: Theory and Practice.* Gordon and Breach Science Publishers, Melbourne.

McLean, J.R. (1991). *Adapting the HDM-III Vehicle Speed Prediction Models for Australian Rural Highways.* Working Document TE 91/014, Australian Road Research Board, Nunawading.

McLean, J.R. (2000a). *Improved Prediction Models for Road Crash Savings – Rural*. Report RC 90122-1, ARRB Transport Research, Nunawading.

McLean, J.R. (2000b). *Improved Prediction Models for Road Crash Savings – Urban.* Report RC 90122-2, ARRB Transport Research, Nunawading.

McLean, J.R. and Chin-Lenn, R.L. (1977). *Speeds on Curves - Data Report.* Australian Road Research Board Internal Report AIR 200-1A, Australian Road Research Board, Nunawading.

Memmott, J.L. and Dudek, C.L. (1982). *A Model to Calculate the Road User Costs at Work Zones*. Research Report 292-1, Texas Transportation Institute, Texas A&M University, College Station.

Minderhoud, M.M., Botma, H., and Bovy, P.H.L. (1997). *Assessment of Roadway Capacity Estimation Methods*. Transportation Research Record 1572, pp. 59-67. Transportation Research Board, Washington, D.C.

Ministry of Transport (1980). *Motor Car Depreciation - A Study of Cars for Private Sale in Wellington.* Ministry of Transport Economics Division, Wellington.

Moore, D.F. (1975). *The Friction of Pneumatic Tyres*. University College, Dublin, Ireland, Elsevier Scientific Publishing Company.

Morosiuk, G. and Abaynayaka, S.W. (1982). *Vehicle Operating Costs in the Caribbean: An Experimental Study of Vehicle Performance.* TRRL Laboratory Report 1056. Transport and Road Research Laboratory, Crowthorne.

MOT (1984). Open Road Speed Limits. *Report of the Ministry of Transport to the Parliamentary Select Committee on Road Safety*, Wellington.

Moyer, R.A. and Winfrey, R. (1939). *Cost of Operating Rural Mail Carrier Motor Vehicles on Pavement, Gravel and Earth.* Iowa Engineering Experiment Station Bulletin 143. Iowa State College, Iowa.

MVA Consultancy, Institute of Transport Studies (Leeds), Transport Studies Unit (Oxford) (1987). *The Value of Travel Time Savings*. Policy Journals, Newbury, Berkshire, UK

Myers, T.T., Ashkenas, I.L. and Johnson, W.A. (1980).*Feasibility of a Grade Severity Rating System*. Report No. GHWA-RD-79-116.

NAASRA (1980). *Interim Guide to the Geometric Design of Rural Roads*. National Association of Australian State Road Authorities, Sydney.

Nakamura, M. (1994). *Research and Application of Highway Capacity in Japan.* Second International Symposium on Highway Capacity, 1994. Akcelik, R. and W. Reilly (ed.), Country Reports, pp. 103-112.

Nash, C.A. (1976). *The Replacement of Road Transport Vehicles.* Journal of Transport Economics and Policy, Vol X, No. 2, pp. 150-166.

NDLI (1995). *Modelling Road User Effects in HDM-4*, *Final Report*. Asian Development Bank Project RETA 5549. . International Study of Highway Development and Management Tools. N.D. Lea International, Vancouver.

NDLI (1997). *Gujarat Project Co-ordinating Consultancy: Calibration of Road User Effects*. N.D. Lea International, Vancouver, B.C., Canada.

Nicholson, A.J. and Wong, Y.D. (1992). *Risk Compensation and Driver Behaviour on Curves.* Proc. 1992 I.P.E.N.Z. Conference, Christchurch.

NITRR (1983). *Evaluation of Road User Costs: Technical Manual for RODES 2.* NITRR Manual P11, CSIR, Pretoria.

Nortström, O. and Andersson, O. (1995). *Swedish Research on Tyre Consumption*. Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Odoki, J.B. and Kerali, H.G.R. (1999a). *Revised Specifications for Modelling Non-Motorised Transport in HDM-4*. Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Odoki, J.B. and Kerali, H.G.R. (1999b). *Analytical Framework and Model Description.* HDM-4 Technical Reference Manual, Volume 4, PIARC, Paris.

Odoki, J.B. and Kerali, H.G.R. (1999c). *Energy Balance Specifications.* Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Odoki, J.B., (1997), *Traffic Characteristics Specifications for HDM-4*, *Draft Working Paper*. International Study of Highway Development and Management Tools., Overseas Development Administration, Asian Development Bank, Swedish National Road Administration, and The World Bank, 1997.

O'Flaherty, C.A. and Coombe, R.D. (1971a). *Speeds on Level Rural Roads, A Multivariate Approach: 1. Background to the Study.* Traffic Engineering and Control 13(1), pp. 20-21.

O'Flaherty, C.A. and Coombe, R.D. (1971b). *Speeds on Level Rural Roads, A Multivariate Approach: 2. The Modelling Process*. Traffic Engineering and Control 13(2), pp. 68-70.

O'Flaherty, C.A. and Coombe, R.D. (1971c). *Speeds on Level Rural Roads, A Multivariate Approach: 3. The Speed Models Produced*. Traffic Engineering and Control 13(3), pp. 108-111.

Olsen, P.L., Cleveland, D.E., Fancher, P.S., Kostyniuk, L.P., and Schneider, L.W. (1984). *Parameters Affecting Stopping Sight Distance.* NHCRP Report 270, Transportation Research Board, Washington, D.C.

Oppenlander, J.C. (1966). *Variables Influencing Spot-Speed Characteristics - Review of Literature.* Highway Research Board Special Report 89, Highway Research Board, Washington, D.C.

Opus-Beca (1998) *Vehicle Repair and Maintenance Costs: Literature Review and Operator Cost Survey*. Transfund New Zealand Research Report no. 118, Wellington.

Opus-TRL (1999). *Review of VOC-Pavement Roughness Relationships Contained in Transfund's Project Evaluation Manual*. Final Report 52927.00, Opus Central Laboratories, Lower Hutt.

Padeco Co. Ltd (1996). *Non-motorised Transport (NMT) Modelling in HDM-4*, *Draft Final Report (second Version)*. International Study of Highway Development and Management Tools. Transport Division, The World Bank, Washington, USA.

Padeco Co. Ltd (1997). *Recommended Short-Term Applications to Implement Non-motorised Transport (NMT) Modelling in HDM-4.* International Study of Highway Development and Management Tools. Transport Division, The World Bank, Washington, USA.

Papagiannakis, T. (1999). *On the Relationship Between Truck Operating Costs and Pavement Roughness*. Society of Automotive Engineers Paper 99C-73, SAE, Pennsylvania.

Parsley, L.L. and Robinson, R. (1982). *The TRRL Road Investment Model for Developing Countries (RTIM2).* TRRL Laboratory Report LR1057, Crowthorne.

Patel, T. (1996). *Power Station Failure Adds to Delhi's Woes.* New Scientist, 12 October 1996.

Paterson and Henein (1972). *Emission from Combustion Engines and Their Control.* Ann Arbor Science Publishers.

Paterson, W.D.O. and Watanatada, T. (1985). *Relationships Between Vehicle Speed, Ride Quality, and Road Roughness.* Measuring Road Roughness and its Effects on User Cost and Comfort. ASTM STP 884. American Society for Testing and Materials, Philadelphia, pp. 89-110.

Pienaar, W.J. (1984). *Olieverbruik van Padvoertuie in Suid-Afrika*. NITRR Report RT/5, National Institute of Transport and Road Research, CSIR, Pretoria.

Poelman, M.A. and Weir, R.P. (1992). *Vehicle Fatigue Induced by Road Surface Roughness*. Vehicle, Tire, Pavement Interface, ASTM STP 1164, J.J. Henry and J.C. Wambold, Eds., pp. 97-111. American Society for Testing and Materials, Philadelphia.

Polus, A., Borovsky, S., and Livneh, M. (1979). *Limited Sight Distance Effect on Speed.* ASCE Transportation Research Journal TE5, pp. 549-560.

Pursala, M. (1994). *Highway Capacity Research in Finland.* Second International Symposium on Highway Capacity, 1994. Akcelik, R. and W. Reilly (ed.), Country Reports, pp. 47-56.

Radelat, Dr. G. (1994). *Colombian Highway Capacity Manual Description*. Correspondence with the International Study of Highway Development and Management Tools, Kensington, Maryland, USA.

Replogle M (1992). *Non-Motorised Vehicles in Asian Cities.* World Bank Technical Paper 162. Washington DC, 1992.

Roberts, J. (1994). *The Modelling and Costing of Traffic Congestion and Their Application in the Analysis of the Economic Performance of Highway Cross-Sections Using HDM-III.* Proc. International Workshop on HDM-4. Bennett C. (ed.), pp. 213-256.

Robinson, R., Hide, H., Hodges, J.W., Rolt, J. and Abaynayaka, S.W. (1975). *A Road Transport Investment Model for Developing Countries.* TRL Report LR 675, Department of the Environment, Crowthorne.

Salusjärvi, M. (1981). *The Speed Limit Experiments on Public Roads in Finland.* Technical Research Centre of Finland Publications 7/81, Espoo.

Schallamach, A. (1981). *Tire Traction and Wear*. Mechanics of Tires, Ed. S.K. Clark. US Department of Transportation, National Highway Traffic Safety Administration, Washington, D.C.

Schutte, I.C. (1979). *Estimating the Costs of Maintenance and Depreciation as Elements of Vehicle Operating Costs*. NITRR Report RT/4/79, CSIR, Pretoria.

Schutte, I.C. (1981). *The Transportation Evaluation Process – National Cost Data*. NITRR Report RT/7/81, National Institute of Transport and Road Research, CSIR, Pretoria.

Schutte, I.C. (1994). *The Capital Cost of Vehicles in Project Appraisal: An Improved Module for CB-ROADS.* Project Report 92/469, South African Roads Board, Department of Transport, Pretoria.

Sheaffer, B., Blair, G. and Lassansk, G. (1998). *Two-Stroke Engines Technology and Emissions PT-69.* Society of Automotive Engineers.

Sime, M., Ashmore, S.C. and Alvi, S. (2000). *WesTrack Roughness, Fuel Consumption, and Maintenance Costs*. Technical Brief, Research Development and Technology, Turner-Fairbank Highway Research Centre, McLean, Virginia.

SNRA (1995). *ISOHDM Supplementary Technical Relationship Stud.* Draft Final Report, Swedish National Road Administration, November 1995.

Sovran, J. (1984). *The Effect of Ambient Wind on a Vehicle's Aerodynamic Work Requirement and Fuel Consumption*. Society Automotive Engineers (U.S). SAE Technical Paper No. 840298.

St John, A.D. and Kobett, D.R. (1978). *Grade Effects on Traffic Flow Stability and Capacity*. NHCRP Report 185, Transportation Research Board, Washington, D.C.

Starkey P (1993). *Animal-Powered Transport in Africa*. Appropriate Technology, Vol.20 No.1, June 1993 pp 9-10.

Sweroad (1997). *Indonesian Highway Capacity Manual (IHCM) February 1997*. Report to the Directorate General Bina Marga, Sweroad, Bandung.

Sweroad and Bina Karya (1994). *Work Package 3.2: Simulation Analysis*. Consulting Services for Indonesian Highway Capacity Manual. Internal Report, Bandung.

Symonds (1997). *Valuation of Travel Time Savings*. Transfund New Zealand Research Report 93. Symonds Travers Morgan (NZ) Ltd., Wellington.

Symonds Travers Morgan in Association with J.B. Cox (1996). *Effect of Road Surface Condition on Vehicle Operating Costs in Australia: Literature Review and Fleet Database Analysis*. Report to Austroads, Melbourne, Australia.

Taragin, A. (1954). *Driver Behaviour on Horizontal Curves.* Proc. of U.S. Highway Research Board, 33, pp. 446-66.

Tate, F. (1993). *An Investigation into Traffic Delay at Road Surfacing Works.* In: Extended Guidelines for Road Surfacing Selection based on an Analysis of Total Costs. Transit New Zealand Research Report PR3-0051, Wellington.

Thoreson, T. (1993). *Survey of Freight Operating Costs*. Report ARR 239, ARRB Transport Research, Vermont South, Australia.

Thoreson and Roper (1994) - Thoreson, T. and Roper, R. (1994). *Review of Vehicle Depreciation and Interest Cost Algorithms - 1st Draft.* Report for Austroads Project BS.3A.40, Australian Road Research Board, Nunawading.

Thoresen, T. and Roper, R. (1998). *Review of Road User Effect Estimation in a Prototype Version of HDM-4.* ARRB Report WD WD R98/001. ARRB Transport Research Ltd., Vermont South.

Tillman, R.N. (1983). *Tyre Management Plan*. Internal Report, Firestone Tyre Company, Wellington, New Zealand.

Tom, G.J.J. and Elcock, E.L. (1988). *Truck Speeds Along Downgrades in California.* FHWA Report FHWA/CA/TO-88/1, Federal Highways Administration, Washington, D.C.

Transfund (1994). *Project Evaluation Manual*. Transfund New Zealand, Wellington.

Transroute (1992). *Road Master Plan Project*. Final Report Appendix II: HDM Runs. Report to the Ministry of Communications Roads and Highways Dept., Dhaka.

TRB (1986). *Designing Safer Roads*. Transportation Research Board, Washington, D.C.

TRB (1992). *Highway Capacity Manual*. Transportation Research Board, Washington, D.C.

TRL (1995). *Predicting Changes in Accident Rates in Developing Countries Following Modifications in Road Design.* Report to the International Study of Highway Development and Management Tools, University of Birmingham.

Troutbeck, R.J. (1976). *Analysis of Free Speeds.* Proc. ARRB Conference 8(5), pp. 40-47, Australian Road Research Board, Nunawading.

Troutbeck, R.J., and Cowley, J.E. (1976). *Analysis of Mean Free Speeds on Australian Two-Lane Rural Highways.* ARR Report 42, Australian Road Research Board, Nunawading.

United Nations (1993). *Energy Efficiency in Transportation.* Alternatives for the Future. 1993.

van Aerde, M. and Yagar, S. (1981). *Capacity and Level of Service on Two-Lane Highways - Volume 3: Quantitative Estimates.* Ministry of Transportation and Communications Transportation Technology and Energy Branch Report, Toronto.

Visser, A.T., and Curtayne, P.C. (1985). *A Pilot Study on the Effect of Road Surface Properties on the Fuel Consumption of a Passenger Car*. NITRR Contract Report C/PAD/46.5, Pretoria.

Viswanathan, E. (1989). Regional Transferability of Brazil Speed and Fuel Models. *Highway Research Bulletin 39*, Indian Roads Congress, New Delhi.

VTI (1990). *VTI Annual Report*, p. 6. Swedish Road and Traffic Research Institute, Linköping.

Wahlgren, O. (1967). *The Dependence of Vehicle Speeds on Different Factors - particularly Road Geometry - On Two-Lane Highways in Finland.* Finland's Institute of Technology, Scientific Researches No. 22, Helsinki.

Walter, J.D. and Conant, F.S. (1974). *Energy Losses in Tires*. Tire Science and Technology, TSTCA, Volume 2, No 4, pp. 235-260.

Walton, C.M. and Lee, C.E. (1977). *Characteristics of Trucks Operating on Grades.* Transportation Research Record 601, pp. 23-30, Transportation Research Board, Washington, D.C.

Wardrop, J.G. (1952). *Some Theoretical Aspects of Road Traffic Research*. Proc. of the Institution of Civil Engineers, 2, pp. 325-362.

Watanatada, T. (1981). *Highway Design and Maintenance Standards Model (HDM) Model Description and User's Manual - Release II.* Transportation, Water and Telecommunications Department Report, the World Bank, Washington, D.C.

Watanatada, T. (1983). *Aggregate Parts Consumption Relationship Based on Mechanistic Principles.* Office Memorandum 4 May, 1983, the World Bank, Washington, D.C.

Watanatada, T. and Dhareshwar, A. (1986). *A Model for Predicting Free-Flow Speeds Based on Probabilistic Limiting Velocity Concepts: Theory and Estimation.* Transportation Research Record 1091, pp. 37-47, Transportation Research Board, Washington, D.C.

Watanatada T, Harral C G, Paterson W D O, Dhareshwar A M, Bhandari A, and Tsunokawa K, (1987a), *The Highway Design and Maintenance Standards Model, Volume 1 - Description*, The World Bank, John Hopkins University Press.

Watanatada, T., Dhareshwar, A., and Rezende-Lima, P.R.S. (1987b). *Vehicle Speeds and Operating Costs: Models for Road Planning and Management.* Johns Hopkins Press, Baltimore.

Winfrey, R. (1963). *Motor Vehicle Running Costs for Highway Economic Studies*. 3131 North Piedmont St., Arlington, Virginia.

Winfrey, R. (1969). *Economic Analysis for Highways*. International Textbook Company, Scranton, Pennsylvania.

Winfrey, R. and Zellner, C. (1971). *Summary and Evaluation of Economic Consequences of Highway Improvements.* NHCRP Report 122, National Academy of Sciences, Washington, D.C.

Wong, J.Y. (1993). *Theory of Ground Vehicles - 2nd Edition*. Wiley.

Wortman, R.H. (1965). *A Multivariate Analysis of Vehicular Speeds on Four-Lane Rural Highways.* Highway Research Record 42, pp. 1-18, Highway Research Board, Washington, D.C.

Wright, J.M and Tignor, S.C. (1965). *Relationship Between Gross Weights and Horsepowers of Commercial Vehicles Operating on Public Highways.* Society of Automotive Engineers Transactions, 73, pp. 469-77.

Yagar, S. (1981a). *Capacity and Level of Service on Two-Lane Highways - Volume 1.* Ministry of Transportation and Communications Transportation Technology and Energy Branch Report, Toronto.

Yagar, S. (1984). *Predicting Speeds for Rural 2-Lane Highways.* Transportation Research 18A(1), pp. 61-70.

Yagar, S. and van Aerde, M. (1983). *Geometric and Environmental Effects on Speeds of 2-Lane Highways in Ontario.* Transportation Research 17A (4), pp. 315-325.

Yuli, P. (1996). *Development of Speed and Fuel Consumption Models for Chinese Vehicles.* Ph.D. Dissertation, Dept. of Civil Engineering, University of Birmingham, Edgbaston.

Zaniewski, J., Butler, B.C., Cunningham, G., Elkins, G.E., Paggi, M.S. and Machemal, R. (1982). *Vehicle Operating costs, Fuel Consumption and Pavement Type and Condition Factors*. Technical Report Prepared for the US Department of Transportation, Federal Highways Administration.

Zukang, Y., Zhongren, W. and Zinghua, L. (1992). *Calibration of Vehicle Operating cost Model for Pavement Management System*. Selected Papers of Tongji University, 1992. Tongji University Press.

ANNEX B1.1: Predicting Superelevation

As described in the main text, it was deemed necessary to predict superelevation as a function of the curve radius, but in such a way that the value reflected driver speed behaviour.

An analysis was made of the side friction factor relationships from several sources. It was found that the one in the TRL Overseas Road Note 6 gave the most consistent trend of side friction versus design speed. This resulted in the equation (R^2 = 0.95; S.E. = 0.01):

$$f = \max(0.15, \min(0.33, 0.47 - 0.0054\ \text{DESSPD} + 2.21 \times 10^{-5}\ \text{DESSPD}^2))$$

(21.8) (-7.4) (4.0)

where DESSPD is the design speed in km/h

The ‘t’ statistics are given in parentheses beneath each coefficient. All were significant at 95% confidence.

A variety of analyses were done to predict the design speed as a function of different independent variables. The most reliable method was using the mean speed of passenger cars and speed data from Bennett (1994). This resulted in the following equation (R^2 = 0.72; S.E. = 6.6):

$$\text{DESSPD} = 17.0 + 0.88\ \text{SPDCAR}$$

(1.8) (7.4)

where SPDCAR is the mean speed of passenger cars in km/h

Equation B1.39 can be rewritten as:

$$e = \frac{\left(\frac{\text{SPDCAR}}{3.6}\right)^2}{R\ g} - f$$

Using the HDM speed prediction model we can predict SPDCAR as a function of the curve radius R. From this, the design speed DESSPD and then the side friction factor f. Thus, we establish a relationship between the superelevation and the curve radius which reflects the speed model.

When this was done it was found that the superelevation values were too high so the side friction factors were increased by 25 per cent. This resulted in much more reasonable predictions and the resulting equation is (R^2 = 0.98; S.E. = 0.01) was:

$$e = 0.45 - 0.68 \ln(R)$$

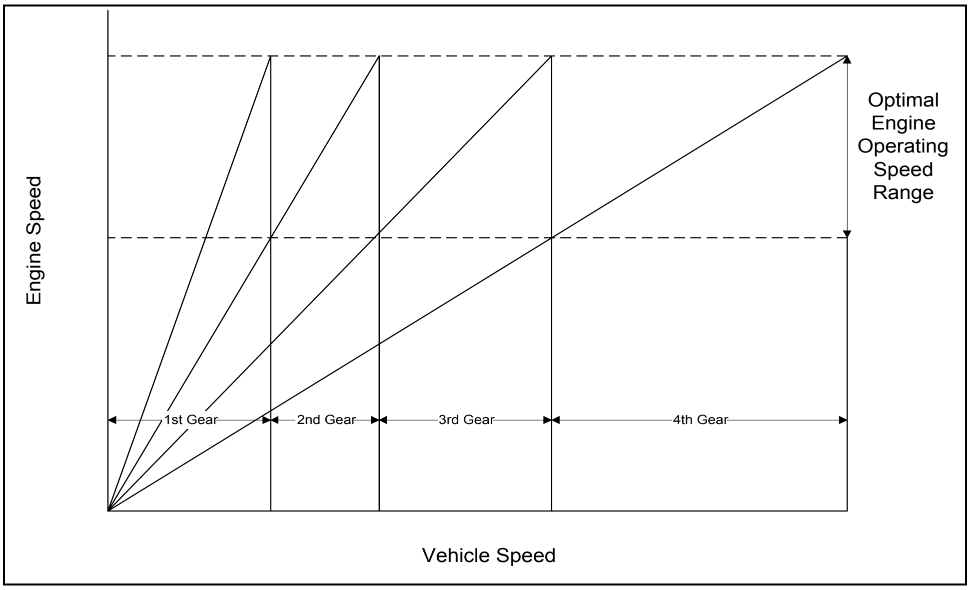
(35.6) (-28.9)

At low radii (< 160 m) this equation predicts superelevations greater than 0.10 m/m. Although in practice these will seldom, if ever, arise, this just reflects the predictions of the HDM speed model. Were these high radii not predicted, the curve-force predictions would be excessive when applied with the HDM speed model.

ANNEX B1.2: Simulating Engine Speed

Introduction

Wong (1993) indicates that vehicle manufacturers select gear ratios such that the driver can always operate the vehicle at or very near to the optimal engine power. This idealised gear selection is shown in Figure A for a four speed vehicle.



Source: Wong (1993)

Figure A: Manufacturers Idealised Gear Selection

In reality, drivers do not maintain the engine speed in the optimal range and hence there is overlap of the gears at the same speed. To reflect this a simulation model was developed which predicted driver gear selection and, thus, the engine speed and effective mass ratio (EMRAT).

Simulating Engine Speed

Figure B shows the input screen for the simulation program.

The important input data are:

- **Gear ratios:** This is the gearing for each of the gears in the vehicle, and the number of gears. Coupled with the **differential ratio**, this governs the engine speed for a given road speed.
- **Idle engine speed:** This is the minimum engine speed of the vehicle.
- **Wheel diameter**: This is used with the gear data to establish the engine speed.

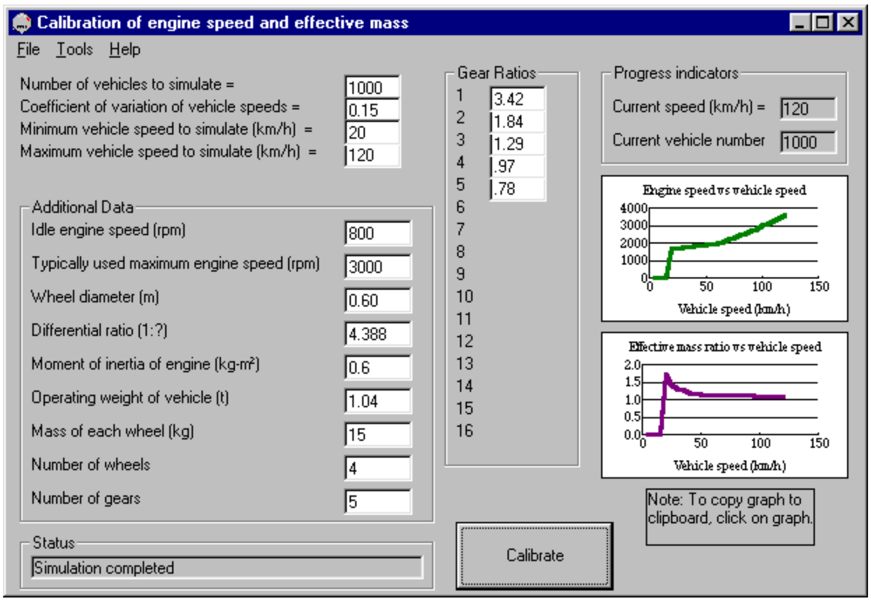


Figure B: Engine Speed Simulation Input Screen

These data are usually available from manufacturer's specifications. The wheel diameter is sometimes harder to obtain but it can be estimated from tyre size as follows.

Tyre sizes have a standard typology. The two most common are shown in Figure C along with a description of what each term means. The top typology is common with truck tyres and is based on the nominal section width being expressed in inches. The second is used with for all vehicles and has the nominal section width in mm along with the aspect ratio[1]. For clarity, the section width is separated from the aspect ratio by a slash (/)[2]. The discussion which follows is based on the second definition, *ie* a metric nominal section width. The imperial section width can be converted from inches to mm using the factor 1" = 25.4 mm.

Figure C: Standard Tyre Typology

1 The aspect ratio is the tyre's section height, which is the distance from the bead to the centre of tread, to the section width. An aspect ratio of 65 means that the tyre's section height is 65 per cent of the tyre's section width.

2 A less common typology uses the nominal width in inches and the aspect ratio separated by the slash, *eg* 14/80R20. Because of the much lower magnitude of the nominal width, it is readily apparent when this case arises and the data should be converted to mm.

The ply rating and load range usually only apply to heavy vehicle tyres so the standard typology reduces to:

xxx/yyRzz

where	xxx	is the nominal width of the tyre in mm
	yy	is the aspect ratio
	zz	is the rim size in inches

For light vehicles the aspect ratio of 82 is usually omitted so they are often specified, for example, as 175/R13 instead of 175/82R13. The same value can be assumed for heavy vehicles.

To calculate the tyre volume it is necessary to predict the tyre diameter, the tread depth, the tread width and the area of rubber vs grooves.

The tyre diameter in m is predicted as (Greenwood, 1997):

$$DIAM = 25.4\ zz + 2\ xxx\ yy/100$$

The model assumes that vehicles do not travel with an engine speed less than 15 percent higher than the idle speed (except for 1st gear), or more than 85 per cent of the engine speed at maximum power of the engine. The maximum speed a vehicle could travel at in a gear was further restricted by power requirements. This primarily affected the higher gears, as in the lower gears the maximum engine speed was the primary restriction. On the basis of data from the vehicle measurements, it was observed that drivers were not changing gears at high RPM levels. A maximum RPM was therefore put into the software to reflect this practice.

From these restrictions on engine speed, the range of speeds at which a vehicle could travel in any gear was determined based on the vehicles characteristics. Figure D illustrates the relationship between vehicle speed and engine speed for each of the gears in a 1995 Ford Laser passenger car.

As shown in Figure D, there is often a range of gears a driver can choose for a particular vehicle speed. For example at 30 km/h the Ford Laser could theoretically be in any of the 5 gears. It was therefore necessary to model this is in simulation. Over the range of speeds where a driver has a choice of gears, a straight line distribution of percentage of drivers in the higher gear was applied (see Figure E). Since the gear selection probably follows a Normal distribution an S-curve would have been more accurate, however the absence of any reliable data necessitated the use of the simplified function.

In the upper half of Figure E, the dashed horizontal lines indicate the restrictions imposed on engine speed for the simulation. By use of these restrictions, the assumed distribution of gear selection (represented by the dark solid line in the lower half of the figure) was obtained.

In the simulation, for any speed the gear was established by generating a random number and then applying it to the assumed function (*ie* Figure E). This approach was followed for the simple case, when there were only two gears, or for the complicated case when the drivers could select from several gears.

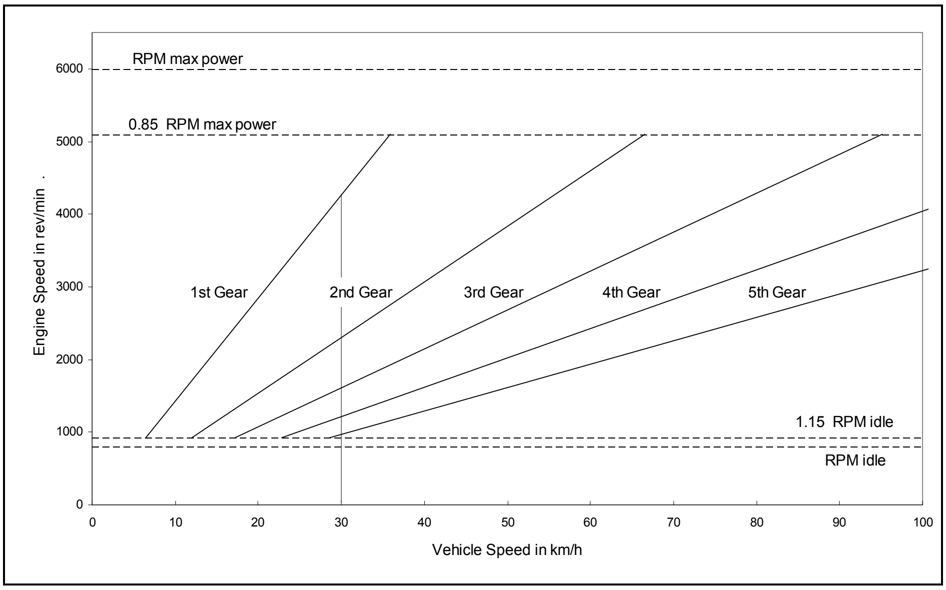


Figure D: Engine Speed Versus Vehicle Speed for Each Gear in a 1995 Ford Laser Passenger Car

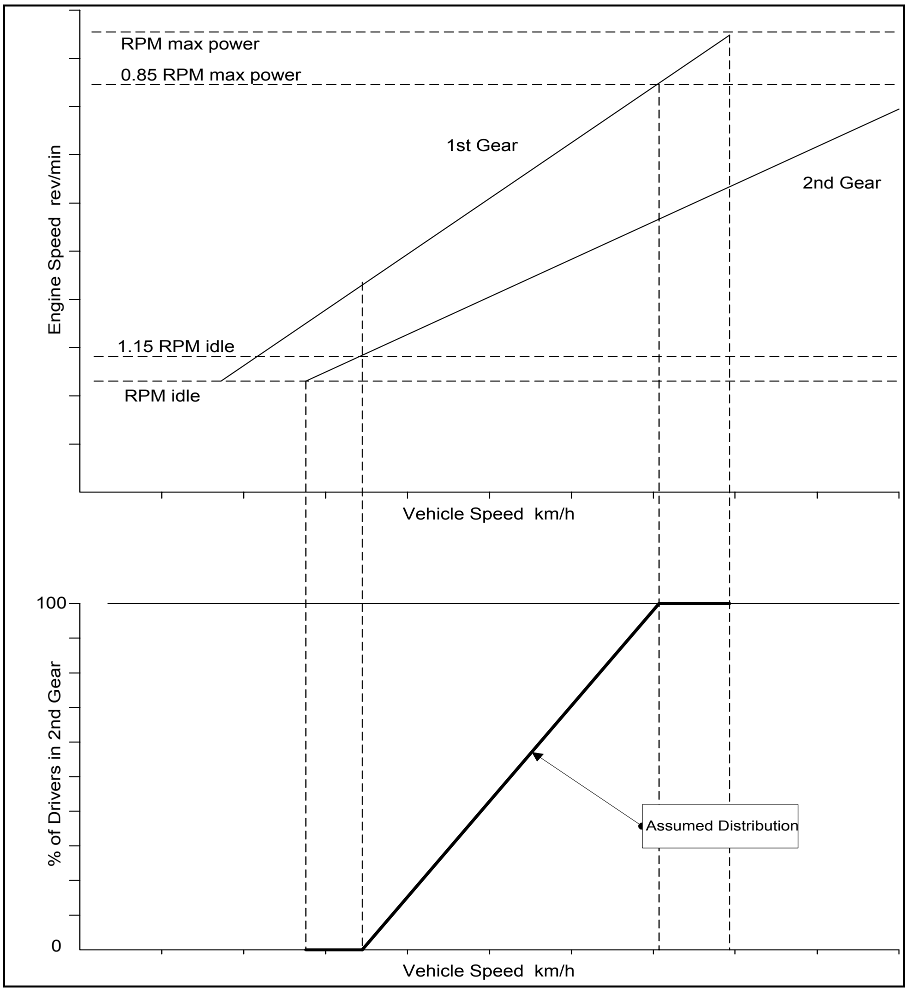


Figure E: Driver Gear Selection

How good is the simulation? Greenwood (1998) collected fuel consumption and engine speeds for passenger cars in Thailand. Figure F shows his raw engine speeds against his field data. This clearly shows the range of engine speeds possible at different road speeds and the inherent problems with establishing an engine speed model. The figure also shows the simulated engine speeds which follow the same trend as the field data.

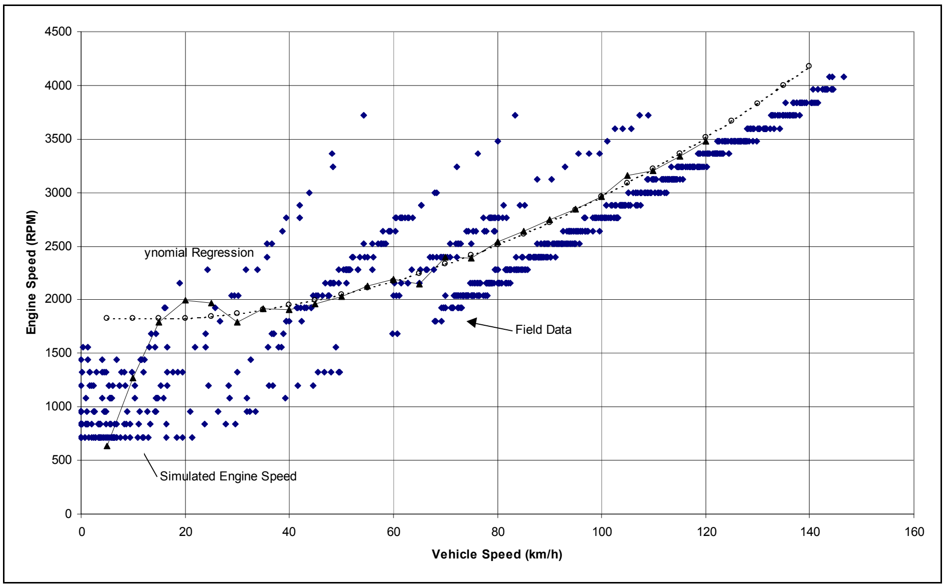


Figure F: Evaluation of Engine Speed Simulation for Passenger Cars

Simulation Results

Engine Speed

Figure G is an example of the engine speed simulation results for a number of passenger cars. NDLI (1995) recommended a three-zone model for predicting engine speed: below 20 km/h when the vehicle is in first gear; when the vehicle is in the top gear; and in between first and top gear. However, examination of the simulation results against actual engine speed data showed that it was just as appropriate to use a single polynomial equation over most of the speeds.

For simplicity, below 20 km/h one can use the 20 km/h engine speed. This does not have a major impact on the outcome when applying the model. This results in the following modified engine speed model which has been recommended for HDM-4:

$$RPM = a0 + a1\ SP + a2\ SP^2 + a3\ SP^3$$

$$SP = \max(20, S)$$

where RPM is the engine speed in revolutions per minute
S is the road speed in km/h

Using data for a range of representative vehicles the coefficients in Table A were established for the engine speed model. The values for buses are based on similar trucks since the it was found that the same basic gearings were often used in both.

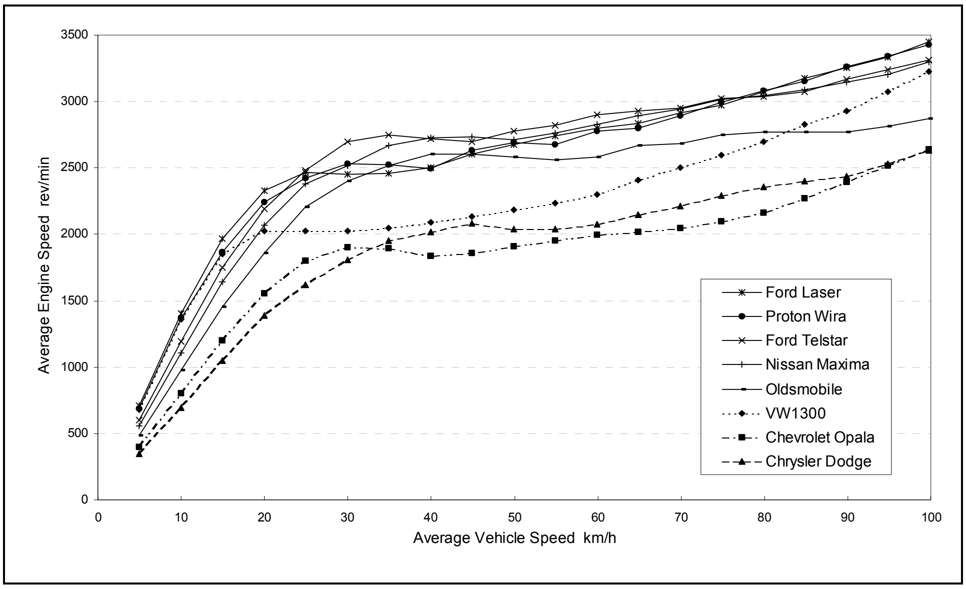


Figure G: Engine Speed Simulation - 1995 Ford Laser Passenger Car

Table A: Engine Speed Model Coefficients

Vehicle	RPM Model Coefficients			
	a0	a1	a2	a3
MC	-162	298.86	-4.6723	-0.0026
PC	1910	-12.311	0.2228	-0.0003
LT	2035	-20.036	0.3560	-0.0009
MT	1926	-32.352	0.7403	-0.0027
HT	1905	-12.988	0.2494	-0.0004
AT	1900	-10.178	0.1521	0.00004
MNB	2007	-18.648	0.3436	-0.0008
LB	2035	-20.036	0.3560	-0.0009
MB	1926	-32.352	0.7403	-0.0027
HB	1926	-32.352	0.7403	-0.0027

It should be noted that this model is different to that proposed by NDLI (1995) and incorporated into HDM-4 v 1.0. This model considered three separate zones: below 20 km/h; at high speeds where the top gear is used; and in between. The equations applied are:

RPM = a0 + 5.6 a1 + 31.36 a2 for $v \leq 5.6$ m/s (20 km/h)

RPM = for $5.6 < v \leq a3$ m/s

RPM = for $v > a3$ m/s

where a0 to a3 are constants

The parameter a3 indicates the velocity at which vehicles are in top gear. The slope of the continuous function at velocity a3 is related to top gear.

The engine speed RPM100 is calculated by substituting a3 into the third equation for v and 27.8 for a3.

Effective Mass

The effective mass analysis was done using vehicle characteristics were from Watanatada, *et al*. (1987a), Biggs (1988) and various motor vehicle manufacturers specifications. Using these data with the above methodology, the average values of EMRAT were calculated for each average vehicle speed.

The results for EMRAT for a Mitsubishi FV402 heavy truck, a Ford medium truck, Ford Telstar and Laser cars and a Mitsubishi Pajero 4WD are shown in Figure H. The results for other vehicles are presented in Greenwood and Bennett (1995).

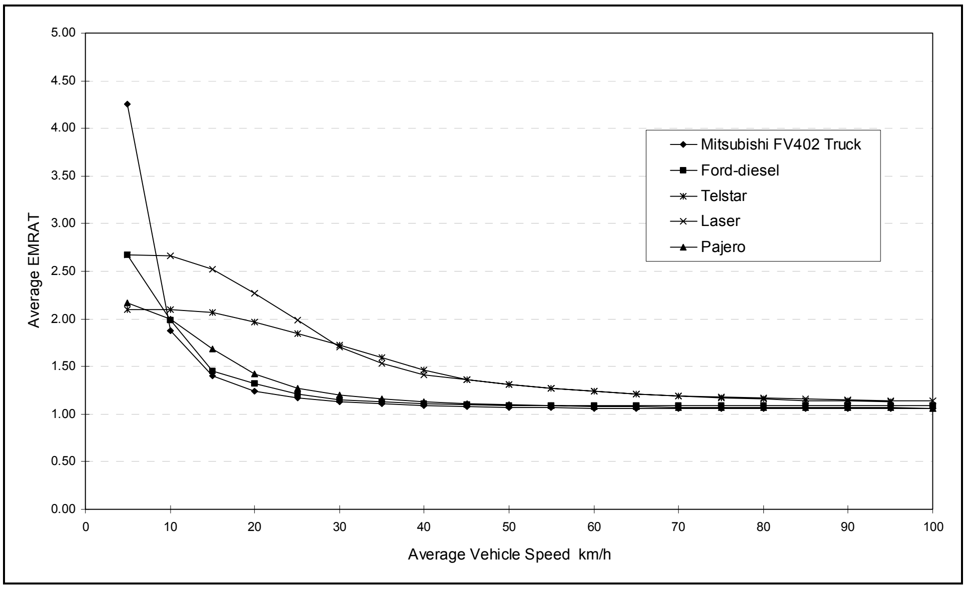


Figure H: Example of Effective Mass Ratio

The asymptotic nature of some vehicles at low speeds is due to the fact that they are in their top gear, and only that gear, at these speeds. By comparison, the heavy truck gearing is so high that even at low speeds there are still several gears to choose from.

ANNEX B2.1: Solving for Driving Power

The basic equation for calculating speed on a gradient is:

$$1000 \text{ Pd} = \text{z0 } v^3 + \text{z1 v}$$

$$\text{z0} = 0.5\ \rho \text{ CD CDmult AF} + \text{b13 CR1 CR2 FCLIM}$$

$$\text{z1} = \text{b11CR2 FCLIM Nw} + \text{b12 CR1 CR2 FCLIM M} + \text{M g GR}$$

This may be rearranged to:

$$f(v) = \text{a0 } v^3 + \text{a1 v} - 1000 \text{ Pd}$$

The solution of the cubic, f(v)=0, may be solved using Descartes' rule of signs. The methodology described below is taken from Watanatada, *et al.* (1987b).

The equation may be written as:

$$0 = v^3 + 3 \text{ z2 v} - 2 \text{ z3}$$

where,

$$\text{z2} = \frac{\text{z1}}{3 \text{ z0}}$$

$$\text{z3} = \frac{1000 \text{ Pd}}{2 \text{ z0}}$$

The solution depends upon the discriminant DT which is calculated as:

$$\text{DT} = \text{z2}^3 + \text{z3}^2$$

The solution to VDRIVE is then calculated from the magnitude of the discriminant as (Watanatada, *et al*., 1987b) :

DT > 0:

$$\text{VDRIVE} = \sqrt[3]{\max(0, \sqrt{\text{DT}} + \text{z3})} - \sqrt[3]{\max(0, \sqrt{\text{DT}} - \text{z3})}$$

DT ≤ 0:

$$\text{VDRIVE} = \max\left[\text{r} \cos(\text{z}), \text{r} \cos\left(\text{z} + \frac{2\pi}{3} \right), \text{r} \cos\left(\text{z} + \frac{4\pi}{3} \right) \right]$$

where,

$$\text{z} = \frac{1}{3} \arccos\left(\frac{-2 \text{ z3}}{\text{z2 r}} \right)$$

$$\text{r} = 2 \sqrt{-\text{z2}}$$

ANNEX B2.2: Examples of Roadside Friction Levels

XFRI = 0

XFRI = 0.95

XFRI = 0.90

XFRI = 0.80

XFRI = 0.70

XFRI = 0.60

ANNEX B7.1: Estimating Service Life

In HDM-4 the user needs to define the expected service life in kilometres for a vehicle operating on a smooth pavement. This value is then used to determine the effect of roughness on service life when using the **Optimal Life** technique. This expected service life is the distance at which it becomes appropriate to scrap the vehicle.

There are a number of different techniques available for calculating the service life and an overview of these may be found in Winfrey (1969). For a Level 1 calibration the ages of a sample of vehicles should be obtained, either from a small survey or by sampling advertisements of vehicles for sale. Daniels (1974) indicates that the service life will be double the mean age. This was also found to be the case in N.Z. (Bennett, 1985) where several different techniques were tested for estimating the service life.

This example of calculating the survival curve is based on the approach of Zaniewski, *et al*. (1982) and uses data from Bennett (1985).

Table A presents the results of a survey of vehicle ages. A sample consisting of the number of vehicles at different ages was obtained and expanded to represent the entire population. This expanded sample is presented in column (2) of Table A and column (3) contains the number of the vehicles of each age, which were originally registered.

Dividing the expanded sample in column (2) by the number of original registrations in column (3) gives a survival curve in column (4). Multiplying this column by 100 per cent gives the percentage of each age surviving.

Zaniewski, *et al*. (1982) recommends modifying this survival curve to improve the predicted service life. In theory, the highest value in column (4) represents a year when all of the registered vehicles are still in the population. As such, it is the sampling factor for the survey. Dividing the survival curve in column (4) by this sampling factor gives the survival ratios in column (5). These ratios are used in place of the values in column (4) for the survival curve[1].

The sum of column (5) is the area under the survival curve and this corresponds to the average service life. The data in Table A indicate that the average service life is 19 years.

In column (6) of Table A the average annual kilometreage for vehicles in each age range is given. To determine the average lifetime kilometreage the product of the survival ratio (column (5)) and the annual utilisation is calculated. These products are given in column (7). The sum of the values in column (7) represents the average lifetime kilometreage - in this example 187,202 km. The average annual kilometreage is given by the average lifetime kilometreage divided by the average service life. Using the earlier derived service life of 19 years results in an average annual kilometreage of 9,850 km/yr.

[1] For those years preceding the sampling factor the survival ratio is set to 1.0.

Table A: Example of establishing survival curve and lifetime utilisation

Vehicle age (1)	Number in expanded sample (2)	Original registrations (3)	Sample/original [(3)/(2)] (4)	Survival ratio (5)	Average annual utilisation (km/yr) (6)	Survival Ratio x Av. annual util [(5)*(6)] (7)
0- 1	65131	67149	0.97	1.00	11500	11500
1- 2	59261	61824	0.96	1.00	11500	11500
2- 3	71789	73527	0.98	1.00	11500	11500
3- 4	81653	83612	0.98	1.00	11500	11500
4- 5	91854	99213	0.93	1.00	11250	11250
5- 6	99569	102626	0.97	1.00	11000	11000
6- 7	89524	90302	0.99	1.00	10750	10750
7- 8	65413	74626	0.88	0.88	10500	9284
8- 9	68118	70426	0.97	0.98	10250	10000
9-10	53328	56438	0.94	0.95	10000	9531
10-11	46308	48808	0.95	0.96	9750	9331
11-12	50636	56484	0.90	0.90	9500	8590
12-13	53721	63768	0.84	0.85	9250	7860
13-14	54516	67189	0.81	0.82	9000	7366
14-15	52632	65742	0.80	0.81	8750	7066
15-16	44482	58347	0.76	0.77	8500	6536
16-17	27301	42680	0.64	0.65	8250	5323
17-18	24714	37663	0.66	0.66	8000	5295
18-19	18188	33859	0.54	0.54	7750	4199
19-20	13388	27214	0.49	0.50	7500	3722
20-21	13190	30985	0.43	0.43	7250	3113
21-22	15208	41323	0.37	0.37	7000	2599
22-23	11130	38980	0.29	0.29	6750	1944
23-24	12516	46237	0.27	0.27	6700	1829
24-25	8812	36610	0.24	0.24	6500	1578
25-26	2958	25724	0.12	0.12	6500	754
26-27	5645	37095	0.15	0.15	6500	998
27-28	2884	26595	0.11	0.11	6500	711
28-29	1378	20575	0.07	0.07	6500	439
> 29	8087	393744	0.02	0.02	6500	135

Source: Bennett (1985)

Notes: Sampling factor:

ANNEX B13.1: Analytical Analysis of Work Zone Effects

This annex presents details on the analytical analysis of work zone effects. The equations and methodology presented here are the basis for the work zone simulation model ROADWORK.

Queuing Effects

Queue Delay

There are three basic cases which arise with work zones. Under the first case the queuing is a function of the capacity, under the second case it is also influenced by the travel time through the work zone, while for the third case it is largely influenced by the duration of time the road is closed — the fourth case is a combination of this third case and any of the first two cases. The following describes how these are modelled for HDM-4.

Queue Delay - Case 1: Multi-lane Highway With One or Both Directions Affected

For multi-lane highways where opposing directions of traffic are not required to share a single lane through the work zone, the average delay to vehicles is calculated for one of two conditions.

(A) Arrival flow less than restricted capacity of work zone

Zero queue at start of the time period:

The delay in this case is based on the random arrival of vehicles and constant departures from the work zone. The departures are a function of the work zone capacity. The average number of vehicles in the queue during the interval i is given by:

$$VEHQUE_i = \frac{\left(ARRIVAL_i / CAPWZ\right)^2}{2\left(1 - ARRIVAL_i / CAPWZ\right)}$$

where $VEHQUE_i$ is the number of vehicles queued for interval i
$ARRIVAL_i$ is the arrival flow rate for interval i in pcse/h
CAPWZ is the capacity of the work zone in pcse/h

The average delay per vehicle in the queue is calculated from:

$$DELAYQUE_i = \frac{(ARRIVAL_i / CAPWZ)1800}{CAPWZ(1 - ARRIVAL_i / CAPWZ)}$$

where $DELAYQUE_i$ is the average delay per vehicle during the time interval i in s.

Finite queue at start of time period:

If the queue dissipates during the analysis interval the delay is then given by:

$$DQUE_i = \frac{VEHQUE_{i-1}^2}{2\left(CAPWZ_i - ARRIVAL_i\right)} 60ANALINT$$

where $DQUE_i$ is the average total delay while a queue exists during interval i in s

The delay per vehicle is calculated as:

$$DELAYQUE_i = \frac{DQUE_i\,60}{CAPWZ_i\,ANALINT}$$

where $DELAYQUE_i$ is the delay per vehicle for interval i in s/veh
ANALINT is the analysis interval in minutes

(B) Arrival flow temporarily exceeds the restricted capacity of work zone

This situation is based on constant arrivals of vehicles and constant departures from the work zone. The methodology adopted is that used by Memmott and Dudek (1982). The number of vehicles in the queue is given by:

$$VEHQUE_i = VEHQUE_{i-1} + (ARRIVAL_i - CAPWZ_i)\ ANALINT/60$$

where $VEHQUE_{i-1}$ is the number of vehicles queued at the end of analysis interval i-1 (note: $VEHQUE_0 = 0$)

The average delay in the queue is calculated as:

$$DQUE_i = \frac{(VEHQUE_{i-1} + VEHQUE_i)}{2} 60ANALINT$$

<u>Queue Delay - Case 2: Two-lane Highway With Both Directions Affected</u>

For two-lane highways the average delay to vehicles are calculated using a method similar to the N.Z. method suggested by Tate (1993) and outlined in Greenwood, *et al.* (1995). It is analogous to a traffic signal problem and is therefore analysed on this basis.

These effects are calculated as:

$$IG = \frac{WZLEN}{VELWZ}$$

where IG is the travel (or intergreen) time in s
WZLEN is the length of the work zone in m
VELWZ is the average speed of travel through the work zone in m/s

The work zone cycle time, WZCYCLE, can be input to HDM-4 based on the behaviour of the flagmen and the site conditions, or calculated as given below. For HDM-4, a maximum default value for WZCYCLE of 600 s (or 10 min) is recommended.

$$WZCYCLE = \frac{3(IG + LOSTIME) + 5}{1 - \left(\frac{ARRIVAL1 + ARRIVAL2}{CAPWZ}\right)}$$

$$\text{EFFG}=\left[\text{WZCYCLE}-2\left(\text{IG}+\text{LOSTIME}\right)\right]\frac{\text{ARRIVAL}}{\text{ARRIVAL1+ARRIVAL2}}$$

$$\text{GCRATIO} = \frac{\text{EFFG}}{\text{WZCYCLE}}$$

where EFFG is the effective green time in s
WZCYCLE is the work zone cycle length in s
LOSTIME is the lost time due to the change over from one flow direction to the other in s
GCRATIO is the ratio of the effective green time to cycle length
ARRIVAL1 is the arrival flow from direction 1 in pcse/hr
ARRIVAL2 is the arrival flow from direction 2 in pcse/hr

For HDM-4, a minimum default value for LOSTIME of 2 s per changeover is recommended. The average delay per vehicle can be expressed as follows:

$$\text{DELAYQUE} = \text{DELAYUN} + \text{DELAYIN}$$

where DELAYQUE is the total delay in the queue during interval i in s/veh
DELAYUN is the uniform (non-platooned) delay in s/veh
DELAYIN is the incremental delay in s/veh

The incremental delay, DELAYIN, is due to the arrival flow being random rather than uniform and due to the number of vehicles left in the queue at the end of each changeover.

The uniform delay, DELAYUN, for direction can be calculated from:

$$\text{DELAYUN}=\frac{\text{WZCYCLE}(1-\text{GCRATIO})^2}{1-\left(\frac{\text{ARRIVAL}}{\text{CAPWZ}}\right)}\frac{\text{ARRIVAL}}{2}$$

Based on Akcelik (1990),

if X < Xo then DELAYIN = 0

else DELAYIN > 0

where X is the ratio of arrivals to capacity
Xo is the critical ratio of arrivals to capacity
DELAYIN is the incremental delay in s/veh

$$\text{X} = \frac{\text{ARRIVALWZCYCLE}}{\text{CAPWZEFFG}}$$

$$\text{Xo} = 0.67 + \frac{\text{WZCAPEFFG}}{3600}$$

For the case where X > Xo, then DELAYIN can be calculated from:

$$\text{DELAYIN} = \frac{\text{ANALINT}}{240}\left((X-1)+\sqrt{(X-1)^2+\frac{720(X<Xo)}{\text{ARRIVALANALINT}}}\right)\text{ARRIVAL}$$

The average number of the vehicles in the queue can be calculated from one of the following:

$$\text{VEHQUE}\frac{\text{DELAYUN}}{\text{WZCYCLE}} \qquad \text{for } X < Xo$$

$$\text{VEHQUE}=\frac{\text{DELAYUN}}{\text{WZCYCLE}}+\frac{\text{DELAYIN}}{X} \qquad \text{for } X \geq Xo$$

<u>Queue Delay - Case 3: Multi-lane Highway, Two-Lane Highway or One-Lane Highway Temporarily Closed</u>

This applies to multi-lane and two-lane highways where one or both directions of vehicle flow are completely stopped for a period time.

Under constant flow conditions the additional time taken to clear the queued vehicles after the road is opened is given by:

$$\text{TIMCLQ} = \frac{\text{CLOSEINTARRIVAL}}{\text{CAPWZ}-\text{ARRIVAL}}$$

where TIMCLQ is the time taken for all the vehicles queued to clear in s
CLOSEINT is the time the road is closed to vehicles in s

The total delay can be calculated from:

$$\text{TOTDELAY=CLOSEINTTIMCLQ}\frac{\text{ARRIVAL}}{120}$$

where TOTDELAY is the total delay in veh-s

The average delay over the period CLOSEINT + TIMCLQ is given by:

DELAYQUE = 30 CLOSEINT

The average number of vehicles queued during CLOSEINT + TIMCLQ is given by:

$$\text{VEHQUE=}\frac{\text{CLOSEINTARRIVAL}}{1800}$$

<u>Queue Delay - Case 4: Special Combination</u>

This is a combination of Case 3 plus either Case 1 or 2.

Time Spent Moving Through the Queue

The time spent by vehicles moving at capacity in the queue is given by:

$$\mathrm{TIMEQUE} = \frac{\mathrm{VEHQUE3600}}{\mathrm{CAPWZ}}$$

where TIMEQUE is the time spent by vehicles in the queue in s/veh

The above TIMEQUE can also be calculated from the following:

$$\mathrm{TIMEQUE} = \mathrm{QUELEN} / \mathrm{VELQUE}$$

$$\mathrm{QUELEN}_i = \frac{\mathrm{VEHQUE}_i\,\mathrm{VEHSPC}}{\mathrm{NLAPP}}$$

where QUELEN_i is the length of queue in m
VEHQUE_i is the number of vehicles queued during the time interval i
VEHSPC is the average vehicle space in m
NLAPP is the number of open lanes approaching the site available for queues to build up in

$$\mathrm{VELQUE} = \frac{\mathrm{VEHSPCCAPWZ}}{\mathrm{NLAPP3600}}$$

where VELQUE is the speed of vehicles travelling in the queue in m/s
VEHSPC is the average vehicle space in m

The average vehicle space can be calculated by assigning an average space for a passenger car, and then utilising the values given in Table B13.2 to ascertain an average spacing for the traffic stream.

Travel Time and Vehicle Operating Costs

The travel time costs will be calculated based on the time spent moving through the queue and the time spent stopped.

The travel time cost per vehicle is given by:

$$\mathrm{TTQCST} = (\mathrm{TIMEQUE}+\mathrm{DELAY})\mathrm{VALTIME}/3600$$

where TTQCST is the travel time queue cost in unit cost/veh
VALTIME is the value of time in unit cost/veh/h

The steady state vehicle operating costs while passing through the queue will be based on the average speed through the queue.

The additional costs due to congestion will be calculated from the HDM-4 congestion model at a relative density of 1.0.

The costs while stationary will be based on the idling rate of fuel consumption as given in Chapter B4.

Network Analysis

For Cases 1 and 2, the following equations will give appropriate values suitable for network analysis.

Unless stated otherwise, the terms have the same meanings as described earlier.

Case 1: Multi-lane Highway with One or Both Directions Affected

$$VEHQUE_i = \frac{\left(ARRIVAL_i / CAPWZ\right)^2}{2\left(1 - ARRIVAL_i / CAPWZ\right)} \qquad ARRIVAL_I < CAPWZ$$

$$DELAYQUE_i = \frac{(ARRIVAL_i / CAPWZ)1800}{CAPWZ(1 - ARRIVAL_i / CAPWZ)}$$

Note: The above assumes random arrivals and constant departures - hence it can only be used when:

$$ARRIVAL_i / CAPWZ \leq \text{about } 0.9$$

Case 2: Two-lane Highway with Both Directions Affected

For each direction, where ARRIVAL < 0.9 CAPWZ, then:

$$VEHQUE1 = \frac{(1 - GCRATIO1)^2}{1 - \left(\frac{ARRIVAL1}{CAPWZ}\right)} \frac{ARRIVAL1}{2}$$

As an approximation:

$$DELAYQUE_i = \frac{(WZCYCLE + 2IG)^2}{8WZCYCLE\left(1 - \frac{ARRIVAL1}{CAPWZ}\right)}$$

where the following approximations are used,

$$WZCYCLE = \frac{3IG}{\left(1 - \frac{2ARRIVAL1}{CAPWZ}\right)}$$

$$EEFG1 = (WZCYCLE - 2\ IG) / 2$$

$$ARRIVAL1 = ARRIVAL2$$

Similar to project analysis, it is recommended that the above be used only for:

$$WZCYCLE \leq \text{about } 600 \text{ s } (10 \text{ min})$$

ANNEX B.13.2: Calculating Work Zone Effects

Vehicles Queued

A-1. Queue Length

The queue length is calculated from the queue size and the vehicle spacings:

QUEUE_LEN = QUEUE VEH_SPACE/NUM_LANES

where	QUEUE_LEN	is the average queue length in m
	QUEUE	is the average queue size [WORKZONE Output]
	VEH_SPACE	is the average vehicle spacing in m
	NUM_LANES	is the number of lanes

B-1. Additional Time Through Queue

The additional time through the queue is calculated as:

ADD_TIME_TQ = 3.6 QUEUE_LEN (1/SPD_Q - 1/SPD_APP)

where	ADD_TIME_TQ	is the additional time due to travel through the queue in s
	SPD_Q	is the speed through the queue in km/h
	SPD_APP	is the approach speed in km/h

This value is also included in the WORKZONE output matrix so it can also be interpolated if the matrix is used, but there is no real benefit to doing this.

B-2. Additional Time Through Work Zone

The additional time through the work zone is calculated as:

ADD_TIME_WZ = 3.6 LWZ (1/SPD_WZ - 1/SPD_APP)

where ADD_TIME_WZ is the additional time due to travel through the work zone in s

This value is also included in the WORKZONE output matrix so could also be interpolated if the matrix is used as input.

B-3. Time Stationary

The time stationary (TIME_STAT) is included in the WORKZONE output matrix and is calculated directly from those data.

If the regression approach is used which has developed a model for the total time stationary, it is given as:

TIME_STAT = max(0, DELAY - ADD_TIME_Q - ADD_TIME_WZ)

where TIME_STAT is the time stationary in s

B-4. Queued Vehicles - Additional Time Due to Speed Changes

The vehicles queued have three speed changes:

- Approach Speed (SPD_APP) - Queue Speed (0)
- Queue Speed - Work Zone Speed (SPD_WZ)
- Work Zone Speed - Approach Speed

The additional time equations are presented in the main body of this report.

B-4a. Additional Time to Decelerate from Approach Speed to a Stop

The final speed is 0 so the equation reduces to:

$$ADD\_TIME\_Qd = SPD\_APP/[a0 + a1\ SPD\_APP + a2\ sqrt(SPD\_APP)]$$

where ADD_TIME_Qd is the additional time to decelerate from the approach speed to a stop in s
a0 to a2 are regression coefficients

B-4b. Additional Time to Accelerate from Stop to Work Zone Speed

Here, the initial speed is 0 and the final speed is the work zone speed:

$$ADD\_TIME\_Qa1 = SPD\_WZ/[a0 + a1\ SPD\_WZ + a2\ sqrt(SPD\_WZ)]$$

where ADD_TIMEQa1 is the additional time to accelerate from the queue to the work zone in s

B-4c. Additional Time to Accelerate from Work Zone to Approach Speed

The equation is:

$$ADD\_TIME\_Qa2 = (SPD\_APP-SPD\_WZ)/[a0 + a1\ SPD\_APP + a2\ sqrt(SPD\_APP - SPD\_WZ)]$$

where ADD_TIME_Qa2 is the additional time to accelerate from the work zone to the approach speed in s

B-4d. Additional Time for Those Vehicles Forced to Queue

The total additional time for those vehicles forced to queue is:

$$ADD\_TIME\_Q = TIME\_STAT + ADD\_TIME\_TQ + ADD\_TIME\_Qd + ADD\_TIME\_Qa1 + ADD\_TIME\_Qa2 + ADD\_TIME\_WZ$$

C-1. Fuel Consumed While Stationary

The fuel consumed while stationary is the product of the time stationary and the idle fuel rate:

$$FUEL\_STAT = TIME\_STAT\ FUEL\_IDLE$$

where FUEL_STAT is the fuel consumed while stationary in mL
FUEL_IDLE is the idle fuel rate in mL/s

C-2. Additional Fuel While Moving Through Queue

The fuel while moving through the queue is calculated with the HDM-4 fuel model at the queue speed:

$$AD\_FUEL\_TQ = QUEUE\_LEN\ (SFC\_SQU - SFC\_SAPP)/1000$$

where SFC_SQU is the fuel consumption in L/1000 km at the queue speed
SFC_SAPP is the fuel consumption in L/1000 km at the approach speed
AD_FUEL_TQ is the additional fuel consumed while to move through the queue in mL

The fuel consumption in L/1000 km corresponds to mL/km so the factor of 1000 is used to adjust it to the per m basis which is used to define the queue length.

C-3. Additional Fuel to Travel Through Work Zone

The vehicles travel through the work zone at a different speed to the approach speed, and thus has a different fuel consumption. The fuel consumption is calculated as:

$$ADD\_FUELWZ = LWZ\ (SFC\_SWZ - SFC\_SAPP)/1000$$

where LWZ is the work zone length (m)
SFC_SWZ is the fuel consumption in L/1000 km at the work zone speed
SFC_SAPP is the fuel consumption in L/1000 km at the approach speed

C-4. Queued Vehicles - Additional Fuel Due to Speed Changes

The vehicles queued have three speed changes:

- Approach Speed (SPD_APP) - Queue Speed (0)
- Queue Speed - Work Zone Speed (SPD_WZ)
- Work Zone Speed - Approach Speed

The additional fuel equations are given in the main body of this report.

C-4a. Additional Fuel to Decelerate to Stop

The deceleration is from an initial speed (SPD_APP) to a final speed of 0 so the equation is:

$$ADD\_FUEL\_Qd = a0 + a1\ SPD\_APP + a2\ SPD\_APP\ \mathrm{sqrt}(SPD\_APP)$$

where SPD_APP is the approach speed in km/h
ADD_FUEL_Qd is the additional fuel to decelerate from the approach speed to a stop in mL
a0 to a2 are regression coefficients

C-4b. Additional Fuel to Accelerate from Stop to Work Zone Speed

Here, the initial speed is 0 and the final speed is the work zone speed so the equation is:

$$ADD\_FUEL\_Qa1 = a0 + a1\ SPD\_WZ + a2\ SPD\_WZ\ sqrt(SPD\_WZ)$$

where	SPD_WZ	is the work zone speed in km/h
	ADD_FUEL_Qa1	is the additional fuel to accelerate from the queue to the work zone in mL

C-4c. Additional Fuel to Accelerate from Work Zone Speed to Approach Speed

The equation is:

$$ADD\_FUEL\_Qa2 = a0 + a1\ (SPD\_APP - SPD\_WZ) + a2\ SPD\_APP\ sqrt(SPD\_APP - SPD\_WZ)$$

where	ADD_FUEL_Qa2	is the additional fuel to accelerate from the work zone to the approach speed in mL

C-4d. Additional Fuel to Those Vehicles Forced To Queue

The total additional fuel to those forced to queue is:

$$ADD\_FUEL\_Q = ADD\_FUEL\_Qd + FUEL\_STAT + ADD\_FUEL\_TQ + ADD\_FUEL\_Qa1 + ADD\_FUEL\_Qa2 + ADD\_FUEL\_WZ$$

Vehicles Not Queued

Vehicles which are not queued have two speed change cycles:

- Approach Speed - Work Zone Speed
- Work Zone Speed - Approach Speed

There are also the work zone effects.

A. Additional Time Due to Speed Changes

A-1. Additional Time to Decelerate from Approach Speed to Work Zone Speed

$$ADD\_TIME\_Nd = (SPD\_APP - SPD\_WZ)/[a0 + a1\ SPD\_APP + a2\ sqrt(SPD\_APP - SPD\_WZ)]$$

where	ADD_TIME_Nd	is the additional time to decelerate from the approach speed to a stop in s

A-2. Additional Time to Accelerate from Work Zone to Approach Speed

$$ADD\_TIME\_Na = (SPD\_APP - SPD\_WZ)/[a0 + a1\ SPD\_APP + a2\ sqrt(SPD\_APP - SPD\_WZ)]$$

where ADD_TIME_Na is the additional time to accelerate from the work zone to the approach speed in s

A-3. Additional Time Due to Work Zones

The additional time due to work zones is the same as that for queued vehicles:

$$ADD\_TIME\_WZ = 3.6\ LWZ\ (1/SPD\_WZ - 1/SPD\_APP)$$

A-4. Total Additional Time for Vehicles Not Queuing

$$ADD\_TIME\_NQ = ADD\_TIME\_Nd + ADD\_TIME\_Na + ADD\_TIME\_WZ$$

B. Additional Fuel Consumption Due to Speed Changes

B-1. Additional Fuel to Decelerate to Work Zone Speed

$$ADD\_FUEL\_Nd = a0 + a1\ (SPD\_APP - SPD\_WZ) + a2\ SPD\_APP\ sqrt(SPD\_APP - SPD\_WZ)$$

where ADD_FUEL_Nd is the additional fuel to decelerate from the approach speed to a work zone speed in mL

B-2. Additional Fuel to Accelerate from Work Zone Speed to Approach Speed

$$ADD\_FUEL\_Na = a0 + a1\ (SPD\_APP - SPD\_WZ) + a2\ SPD\_APP\ sqrt(SPD\_APP - SPD\_WZ)$$

where ADD_FUEL_Na is the additional fuel to accelerate from the work zone to the approach speed in mL

B-3. Additional Fuel to Travel Through Work Zone

The vehicles travel through the work zone at a different speed to the approach speed, and thus has a different fuel consumption. The fuel consumption is calculated as:

$$ADD\_FUEL\_WZ = LWZ\ (SFC\_SWZ - SFC\_SAPP)/1000$$

where LWZ is the work zone length in m
SFC_SWZ is the fuel consumption in L/1000 km at the work zone speed
SFC_SAPP is the fuel consumption in L/1000 km at the approach speed

B-4. Additional Fuel to Vehicles Not Queuing

The total additional fuel is:

$$ADD\_FUEL\_NQ = ADD\_FUEL\_Nd + ADD\_FUEL\_Na + ADD\_FUEL+WZ$$

where ADD_FUEL_NQ is the total additional fuel consumption to vehicles not queued

Total Work Zone Effects

The total additional time and additional fuel is calculated as:

$$ADD\_FUEL = AADT\ [ADD\_FUEL\_NQ\ (100 - PCT\_QUE) + ADD\_FUEL\_Q\ PCT\_QUE]/100$$

$$ADD\_TIME = AADT\ [ADD\_TIME\_NQ\ (100 - PCT\_QUE) + ADD\_TIME\_Q\ PCT\_QUE]/100$$

LIST OF TERMS

α	is the idle fuel consumption in mL/s
αs	is the site parameter depending on site conditions
β	is a speed parameter derived from the Brazil data
μ	is the coefficient of friction
σ	is a speed parameter derived from the Brazil data
π	is pi constant (3.14)
θ	is the angle of incline in radians
ϕ	is the angle of view in radians
χ	is the direction of the wind relative to the direction of travel
ξ	is the fuel-to-power efficiency factor in mL/kW/s
ξb	is the base engine efficiency in mL/kW/s
ρ	is the mass density of air in kg/m^3
Φ	is the slip angle
λ	is the tyre slip
λ_c	is the circumferential slip
λ_l	is the lateral slip
ψ	is the yaw angle (*ie* the apparent direction of the wind)
ψc	is the yaw angle in degrees at which CDmult is a maximum
θa	is the acceleration model time ratio
ε_i	is the maximum catalyst efficiency for emissions ei
ψn	is the function of segment adjustment *ie* adjustment for finite length roadways
Δs	is the attenuation in dB provided by shielding
σa	is the total acceleration noise in m/s^2
σaal	is the noise due to the road alignment in m/s^2
σadr	is the noise due to natural variations in the driver's speed in m/s^2
$\sigma airi$	is the noise due to roughness in m/s^2
$\sigma irimax$	is the maximum acceleration noise due to roughness (default = 0.30 m/s^2)
$\sigma anmt$	is the noise due to speed limits in m/s^2
σasf	is the noise due to roadside friction in m/s^2
σat	is the noise due to traffic interactions in m/s^2
σatm	is the maximum traffic noise in m/s^2
$\sigma xfrimax$	is the maximum acceleration noise due to side friction (default = 0.20 m/s^2)
$\sigma xnmtmax$	is the maximum acceleration noise due to non-motorised transport (default = 0.40 m/s^2)
a	is the acceleration in m/s^2
a(t)	is the acceleration at time t in m/s^2
a,b,c	are model parameters
a0 to a8	are model coefficients or regression parameters
AADT	is the total traffic entering the intersection or using the section in veh/day
aav	is the average acceleration in m/s^2
ACCCST	is the cost of accelerating from the final speed back to the initial speed
ACCYR	is the number of accidents per year
a_{CO}	is a constant = g_{CO}/g_{fuel}
a_{CO2}	is a fuel dependent model parameter representing the ratio of hydrogen to carbon atoms in the fuel
AD	is the annual depreciation as a percentage of the current vehicle price
AD_FUEL_TQ	is the additional fuel consumed while to move through the queue in mL
ADD_FUEL_Na	is the additional fuel to accelerate from the work zone to the approach speed in mL
ADD_FUEL_Nd	is the additional fuel to decelerate from the approach speed to a work zone speed in mL
ADD_FUEL_NQ	is the total additional fuel consumption to vehicles not queued
ADD_FUEL_Qa1	is the additional fuel to accelerate from the queue to the work zone in mL
ADD_FUEL_Qa2	is the additional fuel to accelerate from the work zone to the approach speed in mL
ADD_FUEL_Qd	is the additional fuel to decelerate from the approach speed to a stop in mL
ADD_TIME_Na	is the additional time to accelerate from the work zone to the approach speed in s
ADD_TIME_Nd	is the additional time to decelerate from the approach speed to a stop in s
ADD_TIME_Qa2	is the additional time to accelerate from the work zone to the approach speed in s
ADD_TIME_Qd	is the additional time to decelerate from the approach speed to a stop in s
ADD_TIME_TQ	is the additional time due to travel through the queue in s
ADD_TIME_WZ	is the additional time due to travel through the work zone in s
ADD_TIMEQa1	is the additional time to accelerate from the queue to the work zone in s
ADDCST	is the additional cost due to the speed change cycle

adfuela	is the additional fuel to accelerate in mL
adfueld	is the additional fuel to decelerate in mL
adtima	is the additional travel time due to acceleration in s
AF	is the projected frontal area of the vehicle in m^2
AFtrl	is the projected frontal area of the trailer in m^2
AFveh	is the projected frontal area of the towing vehicle in m^2
AGE	is the vehicle age in years
a_{HC}	is a constant = g_{HC}/g_{fuel}
AINV	is the average interest charge in per cent
AKM	is the average annual utilisation in km/yr
AKM0	is the user defined average annual utilisation in km
AKMVi	is the annual kilometreage of vehicles with age i
ALT	is the altitude above sea level in m
am	is the maximum acceleration rate in m/s^2
AMAXRI	is the roughness at which the maximum acceleration noise arises (default = 20 IRI m/km)
ANALINT	is the analysis interval in minutes
a_{NOx}	is a constant = g_{NOx}/g_{fuel}
a_{Pb}	is a constant = g_{Pb}/g_{fuel}
ar	is a polynomial acceleration model parameter
$ARRIVAL_i$	is the arrival flow rate for interval i in pcse/h
ARRIVAL1	is the arrival flow from direction 1 in pcse/hr
ARRIVAL2	is the arrival flow from direction 2 in pcse/hr
ARRIVALS	is the number of arriving vehicles in the hour for the direction
ARVMAX	is the maximum average rectified velocity in mm/s
a_{SO2}	is a constant = g_{SO2}/g_{fuel}
A_T	is the representative air temperature coefficient
atan	is the arc tan in radians
A_W	is the representative wet road coefficient
b	is a function of the tyre's dimensions and mechanical properties
b11, b12, b13	are rolling resistance parameters
BD	is the braking distance in m
beng	is the speed dependent engine drag parameter
BI	is the roughness in BI mm/km
b_i	is the stoichiometric CPF coefficient
C0tc	is the tread wear rate constant in dm^3/1000 km
CAGV	is the average hourly value of cargo holding time in cost/h
CAPCRT	is the annual depreciation and interest costs in cost/veh/yr
CAPCST	is the total capital cost in cost/1000 km
CAPWZ	is the capacity of the work zone in pcse/h
CARGC	is the cargo holding cost for NMT in cost/km
CARGO	is the cargo cost in cost/h
CARGOH	is the annual number of cargo holding hours per 1000 veh-km
CD	is the aerodynamic drag coefficient
CD(ψ)	is the wind averaged CD
CDmult	is the CD multiplier = CD(ψ)/CD
CDveh	is the truck aerodynamic drag coefficient without the trailer
CDvtrl	is the truck and trailer aerodynamic drag coefficient
ceng	is the speed independent engine drag parameter
CFT	is the circumferential force on the tyre in N
CGL	is the critical gradient length in km
CH	is the number of hours per crew member per 1000 veh-km
CKM	is the cumulative kilometreage in km calculated as 0.5 AKM LIFE
CLOSEINT	is the time the road is closed to vehicles in s
Cm	is an exponent
Co	is the static coefficient of rolling resistance
CPCON	is the elasticity of congestion fuel on parts (default=0.1)
CPF_i	is the Catalyst Pass Fraction for emission i
CR	is the coefficient of rolling resistance
CR1	is a rolling resistance tyre factor
CR2	is a rolling resistance surface factor
CRWC	is the crew cost in cost/km
CRWV	is the average crew wages in cost/h
CST_LABOUR	is the hourly cost of maintenance labour
CTIME	is the time cost per passenger in cost/h
CTNEW	is the cost of a new tyre

CTRET	is the average cost of a retreaded tyre
CTYRE	is the total tyre cost per 1000 km
CURVE	is the horizontal curvature in degrees/km
Cv	is the dynamic coefficient of rolling resistance in $1/(m/s)^{Cm}$
CW1	is the width below which VDESMIN applies
CW2	is the width where VDES2 applies
D	is the density in veh/km
d'	shortest distance from effective source point
d0 to d6	are the depreciation rates for year vehicles of age i
dBDIST	is the correction for distance
dBGR	is the correction due to gradients due to gradients in dB(A)
dBSECT	is the correction for section length
dBSPEED	is the correction due to speed in dB(A)
DECCST	is the cost of decelerating from an initial to final speed
DEF	is the Benkelman Beam rebound deflection in mm
DELAYIN	is the incremental delay in s/veh
$DELAYQUE_i$	is the average delay per vehicle during the time interval i in s.
DELAYQUE	is the total delay in the queue during interval i in s/veh
$DELAYQUE_i$	is the delay per vehicle for interval i in s/veh
DELAYUN	is the uniform (non-platooned) delay in s/veh
DENG	is the engine capacity in litres
DEP	is the depreciation cost as a fraction of the replacement vehicle price, less tyres
DEP(t)	is the depreciation cost at time t
DEPANFAC	is the annual depreciation factor as a fraction of the new vehicle price per year
DEPCST	is the depreciation cost per km or per 1000 km
DEPFAC	is the vehicle depreciation factor as a fraction of the new vehicle price per km
DEPPCT	is the vehicle depreciation in per cent in year t
DESSPD	is the design speed in km/h
dFUEL	is the additional fuel due to congestion as a fraction
DIS	is the perpendicular distance, in m from the centerline of the traffic lane to the observer
DISNEW	is the distance travelled in 1000 km when the tyre is new
DISRET	is the distance travelled in 1000 km by a retreaded tyre
DIST	is the distance travelled in m (= 1000)
DISTCHNG	is the distance between oil changes in 1000 km
DISTDEP	is the distance based depreciation adjusted for surface type in cost/km
DISTOT	is the total distance in 1000 km travelled by a tyre carcass
Do	is the reference distance at which the emission levels are measured
dPC	is the change in parts consumption as a decimal
$DQUE_i$	is the average total delay while a queue exists during interval i in s
DR	is depreciation rate per excess dB (A)
dRI	is the range of roughness over which the change in parts arises
Dw	is the diameter of the wheels in m
e	is the superelevation in m/m
EALC	is the accessory load constant in kW
ECFLC	is the cooling fan constant
edt	is the drivetrain efficiency as a decimal
EEV	is the number of vehicles entering and exiting roadside premises in veh/h
EFFG	is the effective green time in s
ehp	is the proportionate decrease in efficiency at high output power
ELIFE	is the effective vehicle life in years
EMRAT	is the effective mass ratio
ENC	is the energy cost in cost/km
ENFAC	is an enforcement factor (default = 1.10)
ENUSD	is the average energy consumption of NMT in Joules/km
$ENUSD_d$	is the downhill energy consumption in Joules/km
$ENUSD_u$	is the uphill energy consumption in Joules/km
EOE_{CO}	is the engine-out CO emissions in g/km
EOE_{HC}	is the engine-out HC emissions in g/km
EOE_i	is the engine out emission in g/km for emission i
EOE_{NOx}	is the engine-out NOx emissions in g/km
EOE_{Pb}	is the engine-out Pb emissions in g/km
EOE_{SO2}	is the engine-out SO_2 emissions in g/km
EQNT	is the number of equivalent new tyres consumed per 1000 km
EVU	is the elasticity of utilisation
EXPOSINT	is the accident exposure at intersections in million veh
EXPOSURE	is the annual exposure to accidents

EXPSSEC	is the accident exposure between intersections in million veh-km
F	is the fall in m/km
f	is the side friction factor
Fa	is the aerodynamic force opposing motion in N
FALL	is the fall of the road in m/km
Fbr	is the braking force required in N
Fc	is the curvature resistance in N
FC	is the fuel consumption (including congestion effects) in g/km or L/1000 km
FCCONG	is the congested fuel consumption in mL/km
FCLIM	is a climatic factor related to the percentage of driving done in snow and rain
FCmin	is the minimum fuel consumption in mL/s
Fcr	is the curvature force in N
Ff	is the side friction force in N
Fg	is the gradient force in N
Fi	is the inertial resistance in N
FLEET	is the proportion of vehicles susceptible to fleet reduction effects
FLV	is a factor for local effects, vehicle type, *etc*.
FNC	is a factor representing the variation in circumferential forces
FNL	is a factor representing the variation in lateral forces
FR	is property frontage width in meters
Fr	is the rolling resistance in N
FR_{NOx}	is a the fuel threshold below which NOx emissions are very low in g/s
Frtrl	is the rolling resistance of the trailer
Frveh	is the rolling resistance of the towing vehicle
Ftr	is the tractive force applied to the wheels in N
FUEL_IDLE	is the idle fuel rate in mL/s
FUEL_STAT	is the fuel consumed while stationary in mL
g	is the acceleration due to gravity in m/s^2
GCRATIO	is the ratio of the effective green time to cycle length
$g_{emissions}/g_{fuel}$	is the ratio of engine-out emissions per gram of fuel consumed for emission i
GR	is the gradient in per cent
H	is the total gear reduction
h	is the proportionate increase in CD at angle ψc
ha	is the average headway for the hour in s
HAV	is the number of hours vehicle available for use in h/yr
hm	is the minimum headway for a following vehicle in s
HRD	is the number of hours the vehicle is driven per year
HRD0	is the baseline number of hours driven in h/yr
HRWK0	is the average annual utilisation in hr/yr
HRWK0	is the average number of NMT working hours per year
HRWKVi	is the annual working hours of vehicles with age i
hw_i	is the randomly generated headway in s
hwy	is the average headway in s/veh
i	is the emission type (*ie* HC, CO, NOx, SO_2, Pb, PM or CO_2)
Ie	is the moment of inertia of the engine in kg-m^2
IFC	is the instantaneous fuel consumption in mL/s
IG	is the travel (or intergreen) time in s
INT	is the interest cost as a fraction of the replacement vehicle price
INTANFAC	is the annual interest factor as a fraction of the new vehicle price per year
INTCST	is the interest cost per km
INTFAC	is the interest factor as a fraction of the depreciable vehicle price per km
ir	is the interest rate as a decimal
IRI	is the roughness in IRI m/km
irm	is the monthly interest rate as a decimal
Iw	is the moment of inertia of the wheels in kg-m^2
k	is the traffic density in veh/km
K0	is a calibration factor reflecting pavement properties
K0	is the slip energy coefficient in dm^3/MJ
K0pc, K0lh	is the rotation calibration factor (default = 1)
K1pc, K1lh	is the translation calibration factor (default = 0)
kc	is the circumferential stiffness of the tyre
Kcr2	is a calibration factor
Kcs	is a calibration coefficient
Kef	is an energy efficiency factor to account for energy used to overcome other forces opposing motion (default=1.1).
kjam	is the jam density in veh/km

kl	is the lateral stiffness of the tyre
KMCST	is the total maintenance and capital costs on a per kilometre basis
kp	is the parts model age exponent
kPea	is a calibration factor (default = 1)
kpfac	is the age exponent calibration factor (default=1.0)
KTEMP	is the temperature correction factor for tyre consumption
	is the reference energy mean emission level of the ith class of vehicles
LCOST	is the labour cost per 1000 km
Leq (hi)	is the hourly equivalent sound level of the ith class of vehicles
LFT	is the lateral force on the tyre in N
LH	is the number of labour hours
LIFE	is the vehicle service life in years
LIFE0	is the baseline (*ie* user specified) average vehicle life in years
LIFEFAC	is the vehicle service life factor
LIFEKM	is the lifetime utilisation in km
LIFEKM0	is the baseline (*ie* user specified) service life in km
LIFEKMPCT	is the lifetime kilometreage as percentage of baseline life
LOSTIME	is the lost time due to the change over from one flow direction to the other in s
LWZ	is the work zone length (m)
M	is the vehicle mass
m	is a polynomial acceleration model parameter
M’	is the effective mass in kg
m0 to m2	are simplifications of the HDM-III parts and labour model terms
MAINT	are the maintenance and repair costs
MassFuel	is the mass of fuel in g/mL
MAXVLIFE	is the estimated maximum vehicle life with minimal or no use
MCSTCKM	is the maintenance and repair costs at cumulative kilometreage CKM
MDF_i	is the maximum deterioration factor for emission i (default = 10)
Me	is the inertial mass of the engine and drivetrain in kg
MinFuel	is the minimum fuel consumption in mL/s
Mtrl	is the trailer operating mass in kg
Mveh	is the vehicle operating mass in kg
Mw	is the inertial mass of the wheels in kg
NFT	is the normal force on the tyre in N
Ni	is the number of vehicles in the ith class passing a specified point in a one-hour period
NL	is the noise level for all sections in dB(A)
NLAPP	is the number of open lanes approaching the site available for queues to build up in
NLi	is the noise level for section I in dB(A)
NR	is the number of retreads for the tyre carcass
NR0	is the base number of retreads for very smooth, tangent roads
NRC	is the percentage of new tyres sold that are retreads as a decimal
NSCST	is the noise cost in cost/lane-m.
nseg	is the number of sections
NTP	is the new tyre price
NTRIPS	is the number of trips per year
NTV	is the number of tyres per vehicle
NUM_LANES	is the number of lanes
NUM_WHEELS	is the number of wheels on the vehicle
NVP	is the replacement vehicle price
NVPLT	is the replacement vehicle price less tyres
Nw	is the number of wheels
Nwtrl	is the number of wheels on the trailer
Nwveh	is the number of wheels on the vehicle
OA	is the total annual overhead cost
OC	is the oil consumption in L/1000 km
OCKM	is the overhead cost per km
OHC	is the overhead cost in cost/yr
OIL	is the oil consumption in L/1000 km
OILCAP	is the engine oil capacity in L
OILCONT	is the oil loss due to contamination in L/1000 km
OILOPER	is the oil loss due to operation in L/1000 km
OL	is the optimal year for scrapping
OPC	is the opportunity cost of the cargo as a decimal
OVHD	is the overhead cost (excluding pedestrians) in cost/km
Paccs	is the power required to power engine accessories in kW

Paccs_a0, Pacss_a1	are parameter values in the accessories power model
PARTS	is the parts consumption as a fraction of the replacement vehicle price
PAX	is the number of passengers (non-crew occupants) in the vehicle
PAXC	is the passenger time cost for NMT in cost/veh-km
PC	is the parts consumption as a percentage of the new vehicle price per 1000 km
PCHC	is the purchase cost of the NMT
PCOST	is the parts cost per 1000 km
PCTCGT	is the fraction of vehicles whose cargo will benefit from time savings
PCTDS	is the percentage of driving done on snow covered roads
PCTDW	is the percentage of driving done on wet roads
PCTHCV	is the percentage of heavy vehicles in the traffic stream
PctPeng	is the percentage of Pengaccs ascribed to engine drag (default = 80%)
PCTPRV	is the percentage of private vehicles
PCTVi	is the percentage of vehicles of age i in the fleet
PCTVi	is the percentage of vehicles of age i in the fleet
PCTWK	is the percentage of passengers on work-purpose journey
Pd	is the used driving power delivered to the wheels in kW
PED	is the pedestrian flow in ped/h
Peng	is the power required to overcome internal engine drag in kW
PLIMIT	is the posted speed limit in km/h
Pmax	is the maximum engine power in kW
PN	is the current vehicle value as a percentage of the price of a new vehicle
PNH	is the annual number of non-working passenger-hours per 1000 veh-km
PP	is the percentage of vehicle use on private trips
PRD	is the perception/reaction distance in m
Prop_Pb	is the proportion of lead emitted (default = 0.75)
PSV	is the number of vehicle stops and parking manoeuvres in events/h
Ptot	are the total vehicle power requirements in kW
Ptot	is the total power required in kW
Ptr	is the power required to overcome tractive forces in kW
PV	is property value in cost/property
PVC	is the present value of the total cost
PWH	is the annual number of working passenger-hours per 1000 veh-km
Q	is the flow in veh/h
q	is a polynomial acceleration model parameter
$QUELEN_i$	is the length of queue in m
QUEUE	is the average queue size [WORKZONE Output]
QUEUE_LEN	is the average queue length in m
r	is the rolling radius of the tyres in m
r_i	is the deterioration factor for emission i in %/year
R	is the radius of curvature in m
RAND	is a random number between 0 and 1
RELDEN	is the relative density
RF	is the rise and fall in m/km
rg	is the radius of gyration of the tyre in m
r_{HC}	is a constant to account for incomplete combustion in g/s
RI	is the adjusted roughness in IRI m/km
RI_{adj}	is the adjusted road roughness in IRI m/km
RImin	is the minimum adjusted roughness (default = 3 IRI m/km)
RISE	is the rise of the road in m/km
RL	is the route length in km
RMC	is the repair/maintenance cost in cost/km
RPM	is the engine speed in revolutions per minute
RPM100	is the engine speed when travelling at 100 km/h in rpm
RPMIdle	is the idle engine speed in rpm
RREC	is the ratio of the cost of retreads to new tyres (CTRET/CTNEW)
RS	is the rise in m/km
RTWR	is the life of a retreaded tyre relative to a new tyre as a decimal ($\leq$ 1.0)
RUN(t)	are the running costs at time t
RVPLT	is the residual vehicle price less tyres
RVPLTPCT	is the residual vehicle price in per cent.
S	is the average vehicle speed of the vehicle class in km/h.
S0	is the baseline average speed in km/h
SECTLEN	is the length of the section in km
SFC	is the fuel consumption in L/1000 km

SFCL	is the side friction class where 0 = Very Low; 1 = Low, 2 = Medium, 3 = High, 4 = Very High.
SFC_SAPP	is the fuel consumption in L/1000 km at the approach speed
SFC_SQU	is the fuel consumption in L/1000 km at the queue speed
SFC_SWZ	is the fuel consumption in L/1000 km at the work zone speed
SFF	is a roadside friction factor (0 – 5)
SFI	is the free speed under ideal conditions
Si	is the average speed of the ith class of vehicles in km/h
SMV	is the number of slow-moving vehicles in veh/h
SPD_APP	is the approach speed in km/h
SPD_Q	is the speed through the queue in km/h
SPD_WZ	is the work zone speed in km/h
SPDCAR	is the mean speed of passenger cars in km/h
SPEED	is the vehicle road speed in km/h
SQ	is the traffic-influenced speed in km/h
t	is the time in s
T	is the time period over which the Leq is computed (usually 1 hour)
T0	is the reference air temperature in °C
ta	is the time to accelerate in s
TAIR	is the temperature of the air in °C
td	is the time to decelerate in s
TD	is the amount of time spent driving on the trip in h/trip
TDPINT	is the marginal time depreciation and interest cost in cost/h
Tdsp	is the texture depth in mm from the sand patch method
TE	is the tyre energy in MNm/1000 km
TFT	is the total tangential force which is the vectorial sum of the mutually perpendicular lateral and circumferential components in N
TIMCLQ	is the time taken for all the vehicles queued to clear in s
TIME_STAT	is the time stationary in s
TIMEDEP	is the time based depreciation cost as fraction of the replacement vehicle price per year
TIMEQUE	is the time spent by vehicles in the queue in s/veh
TMC	is the time cost of NMT in cost/km
TMW	is the total mass of the wheels in kg
TN	is the amount of time spent on non-driving activities as part of the round trip tour, such as loading, unloading, refueling, layovers, *etc*., in h/trip
TOC	is the time and operating cost of NMT per vehicle-km
TOTDELAY	is the total delay in veh-s
TPE_i	are the Tailpipe Emissions in g/km for emission i
TRPM	is the load governed maximum engine speed in rev/min
Tsv	is the tyre slip velocity defined as the vectoral difference between the travel and circumferential velocities of the wheel
TSEC	is the time taken to traverse the section in s
TT	is the total travel time for the trip in h/trip
TTKKM	is the travel time in h/1000 km
TTQCST	is the travel time queue cost in unit cost/veh
TWN	is the tread wear per 1000 km as a decimal for new tyres
TWR	is the tread wear per 1000 km as a decimal for retreaded tyres
TWT	is the total tread wear in dm^3/1000 km
TWT_c	is the tread wear resulting from circumferential forces in dm^3/1000 km
TWT_l	is the tread wear resulting from lateral forces in dm^3/1000 km
ΔTWT	is the change in tread wear
UCEN	is a user-defined unit cost of energy used by NMT in cost/Joule.
UNICST	is the cost of travelling the same distance at the original speed
v'	is the average speed in m/s calculated as the section length divided by the time
v	is the vehicle velocity in m/s
V_i^y	is the number of vehicles of age i in analysis year y
v0	is the velocity at the entrance to a section in m/s
v1	is the velocity at the end of the section in m/s
VALCAR	is the value of the cargo
VALTIME	is the value of time in unit cost/veh/h
Vapp	is the approach speed in m/s
VBRAKE	is the limiting braking speed on negative gradients in m/s
VCR	is the volume-to-capacity ratio
VCURVE	is the limiting curve speed in m/s
VDES	is the width influenced desired speed in m/s

VDES2	is the minimum desires speed on two lane roads in m/s
VDESIR	is the limiting desired speed in m/s
VDESIR0	is the desired speed in m/s in the absence of speed limits
VDESMIN	is the minimum desired speed on single lane roads in m/s
VDESR	is the desired speed on the road under examination
VDIFF	is the maximum difference in vehicle speeds due to width reductions
VDRIVE	is the limiting driving speed in m/s
VEH_SPACE	is the average vehicle spacing in m
$VEHQUE_i$	is the number of vehicles queued during the time interval i
$VEHQUE_{i-1}$	is the number of vehicles queued at the end of analysis interval i-1
VEHSPC	is the average vehicle space in m
VEHVAL	is the value of the vehicle at time t
velfin	is the final velocity in m/s
velinit	is the initial velocity in m/s
VELQUE	is the speed of vehicles travelling in the queue in m/s
VELWZ	is the average speed of travel through the work zone in m/s
VGRAD	is the speed limited by gradient in m/s
VOC	is the operating cost of the NMT in cost/km
vr	is the speed of the vehicle relative to the wind in m/s
Vra	is the velocity of the wheel relative to the axle
VROUGH	is the limiting roughness speed in m/s
VSS	is the steady state speed in m/s
Vw	is the wind velocity in m/s
W0	is the reference wet road percentage
WET	is the representative percentage for wet roads
WRFACT	is a width reduction factor which is a function of the effective width of the carriageway
WZCYCLE	is the work zone cycle length in s
WZLEN	is the length of the work zone in m
X	is the ratio of arrivals to capacity
XMT	is a speed reduction factor due to motorised traffic and roadside activities, allowable range: min 0.4 to max 1.0 (default = 1.0)
Xo	is the critical ratio of arrivals to capacity
xxx	is the nominal width of the tyre in mm
yy	is the aspect ratio
zz	is the rim size in inches