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FUEL ECONOMY AND PERFORMANCE OF FUTURE I.C. ENGINE VEHICLES

by

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This thesis is submitted to the University of Durham
for the Degree of **Master Of Science**

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ABSTRACT

In order to investigate the feasibility of the various proposed design improvements in motor vehicles, a flexible computer simulation package has been developed at the University of Durham. The package simulates motor vehicles by connecting a series of subroutines representing individual physical components which contain their own efficiency characteristics and together represent the desired total drive train.

It is shown that the replacement of the conventional four-speed manual gearbox by continuously variable transmission in present vehicles can lead to improvements of 7-23% in overall fuel economy.

Furthermore, it is demonstrated that by using present technologies improvements of 59-65% in overall fuel economy are possible by the turn of the century. It is also shown that these savings could be enhanced by a further 26% if the ideas at present incorporated in research vehicles are translated to on-road vehicles regardless of cost.

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LIST OF SYMBOLS

a	=	Engine angular acceleration.
A	=	Vehicle projected frontal area, m^2.
AC	=	Vehicle acceleration, m/s^2.
ACNEW	=	New estimate of vehicle acceleration rate, m/s^2.
ACOLD	=	Previous estimate of vehicle acceleration rate, ms.
AK	=	Final drive load dependent losses constant.
B	=	Ambient pressure, mm Hg.
CC	=	Engine cubic capacity, cm^3.
CD	=	Aerodynamic drag coefficient.
CR	=	Coefficient of rolling resistance.
CVCTRH	=	CVT highest gear ratio attainable at all power levels.
CVTCTR	=	CVT ratio.
CVTSCR	=	CVT Synchronous ratio.
Da	=	Density of air, Kg/m^3
D15	=	Density of air at 15°C, Kg/m^3.
E	=	Energy expenditure, J.
EFF	=	Estimated efficiency.
EFFCOP	=	Coupling efficiency.
EFFCVT	=	CVT efficiency.
EFFDR	=	Final drive efficiency.
EFFDT	=	Drive train efficiency.
EFFGB	=	Gearbox efficiency.
EFFMAX	=	CVT efficiency at maximum power.

LIST OF SYMBOLS Continued

Et	=	Estimated time for 0-60 mph W.O.T. acceleration, secs.
FDR	=	Final drive ratio.
G	=	Gradient.
Gear	=	Gear employed during present time interval.
GR	=	Gear ratio.
H	=	Time step.
I	=	Engine inertia, Kgm^2
K1,K2	=	Gearbox load dependent losses constants.
PD	=	Power expenditure on drag losses, w.
PDMAX	=	Final drive maximum input power, w.
PE	=	Engine developed power, w.
PEMAX	=	Maximum engine power, w.
PG	=	Power expenditure on gradient, w.
PL	=	On engine limiting power, w.
PLOSS	=	Power loss, w.
POWRCV	=	CVT input power, w.
POWRD	=	Final drive input power, w.
POWRG	=	Gearbox input power, w.
POWRW	=	Wheel power expenditure, w.
PWMAX	=	Maximum power at wheels, w.
PR	=	Power expenditure on rolling resistance, w.
RCV	=	CVT input speed, rev/min.
RD	=	Final drive input speed, rev/min.
RE	=	Engine speed, rev/min.
REMAX	=	Maximum engine speed, rev/min.

LIST OF SYMBOLS Continued

RG	=	Gearbox input speed, rev/min.
Ridle	=	Engine minimum idle speed, rev/min.
ROPT	=	Engine optimum speed, rev/min.
RW	=	Wheel rotational speed, rev/min.
t	=	Time.
TCV	=	Normalised CVT input torque.
TCVMAX	=	Maximum CVT input torque, Nm.
Te	=	Ambient temperature, °C.
TE	=	Tractive effort, N.
TEA	=	Tractive effort due to accelerative, N.
TED	=	Tractive effort due to drag losses, N.
TEG	=	Tractive effort due to gradient, N.
TER	=	Tractive effort due to rolling resistance, N.
TORQCV	=	CVT input torque, Nm.
TORQE	=	Engine torque, Nm.
TORQIN	=	Engine inertia torque loss, Nm.
TORQW	=	Torque at wheels, Nm.
U	=	Normalised vehicle speed.
VMAX	=	Maximum vehicle speed, m/s.
V	=	Vehicle speed, m/s.
VBGR	=	Transmatic initial reduction gear ratio.
VF	=	Vehicle final velocity, m/s.
VI	=	Vehicle initial velocity, m/s.
VN	=	Wind velocity, m/s.
V1	=	Vehicle initial velocity at time step, m/s.
V2	=	Vehicle final velocity at time step, m/s.

LIST OF SYMBOLS Continued

W	=	Vehicle mass, Kg.
Win	=	Effective vehicle weight increase due to wheel inertia, Kg.
WR	=	Wheel radius, m.
ΔRE	=	Change in engine speed in the time interval.
Δt	=	Time interval.

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CHAPTER ONE

INTRODUCTION

INTRODUCTION

The 1980s began with a nervous and tight oil market. The revolution in Iran and the curtailment of Iranian production in 1979-1980 spurred an increase in the price of oil by more than double to thirty dollars a barrel (1). These high prices, coming on top of the pricing revolution of 1973-1974, helped to create a world recession. These price increases also served to accelerate the search for oil outside OPEC. This, in line with the recession, caused a weakening of the oil market and a final collapse in February 1986 when oil prices tumbled by over fifteen dollars in just a few weeks (1). The position could easily be reversed, however, if it results in an upturn in the world economy and the abandonment of fuel-switching and energy conservation policies. Alternatively, a production agreement by the oil producing countries could lead to an upturn in the price of oil. Indeed such an agreement to cut back production was reached in OPEC member countries in August 1986, the effectiveness of which depends highly on whether it will be adhered to and the attitude of the non-OPEC oil producing countries.

The effect of 1973-1974 and 1979-1980 increases in oil prices on consumption in the EEC and UK is illustrated in

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Figs 1.1(2) and 1.2(3) respectively. Although the total petroleum consumption has decreased dramatically, the amount of petroleum used in transport has been relatively unchanged. It seems that the improvements in vehicle fleet fuel consumption has balanced out the increases in vehicle population.

1.1 Changes in Vehicle Population

Over the years there have been changes in vehicle and engine size. Fig 1.3(4) represents the number of cars of each class as a percentage of total cars registered in UK over the 1970-1984 period and shows the growing dominance of the cars of 1000-2000cc range. In fact, an estimate (5,6) for 1976 shows that almost 47% of the total petroleum energy used in road transport is consumed by cars of this range. This share is expected to have risen due to the increasing share of this range in the number of cars on the road. So this study examines possible fuel savings in vehicles in this range of engine size.

1.2 Fuel Economy Target

Fig 1.4(4) shows the vehicle population of the recent past and forecasts that, by the year 2000, car population is expected to rise between 22% and 50% relative to 1984. In order that consumption of petroleum in road transport is at least kept at the relatively constant levels of the past decade, the improvements in vehicle fleet petroleum consumption should balance out the increase in vehicle population which would require savings of about 22-50%. In order to obtain such fuel savings various methods have been proposed, and in order to investigate their feasibility some are reviewed here.

1.3 Supplementary Fuels

The primary fuel which forms a useful supplement to oil is natural gas. Natural gas reserves throughout the world are very large and there are several schemes to supply natural gas internationally, such as the Siberian pipe-line from the Soviet Union to Western Europe. Although it is relatively easy to convert existing engines to run on natural gas, it is an inconvenient fuel for road vehicles due to large difficulties in storage.

Natural gas, like coal, can also be processed into methanol, a liquified fuel which can be transported in tankers and is stable at ambient conditions. Methanol has an important advantage over other alternatives in that it can be blended with petrol by volume at up to 5% with few technical problems (7). There are proposals to increase the methanol content of motor fuels (7). The main drawbacks of such applications are its low energy density (half that of petrol) and its attraction for water which can have a corrosive effect on the material used in the fuel system.

Another proposed supplementary fuel is ethanol which is produced by fermentation of any crop rich in carbohydrates such as sugar cane, corn, etc. Productions of methanol and ethanol are very wasteful of energy, however, and the commercial justification doubtful. For ethanol, there is the added disadvantage that the source is essentially a food crop. In some places, like Brazil, where 95 vol % is now in use in fleet operation, there are strong indigenous and economic reasons, but for the rest of the world it holds relatively little possibility (7).

1.4 Alternative Engines

The increasing pressure for improved fuel economy has stimulated intensive research into alternative automotive engines, some of which are discussed here in order to determine whether they could replace the conventional engines in the near future.

1.4.1 Brayton Engine

The gas turbine with regenerative heating offers good thermal efficiency at high-load conditions, has multifuel capability and potentially low emission standards due to its continuous combustion process. It has a poor part-load efficiency, however, due to the parasitic losses within the engine which tend to increase relative to the power output as the size of the engine is reduced (8). Due to its advantages and its excellent power:weight ratio gas turbine can be attractive as a power unit for large trucks. For the range of vehicles considered here, however, its future is much less certain.

1.4.2 Stirling Engine

Modern Stirling Engines are closed cycle engines with an external heat supply and operate on a gas cycle with either helium or hydrogen as a working fluid. The external combustion process of the Stirling Engine can provide for good combustion, multifuel capability and low emission levels. The engine is quiet, has high thermal efficiency and good low-speed torque and therefore is well suited for automotive application. Since the Stirling Engine is an external combustion engine and heat has to be transferred from the combustion gasses to the working fluid via a heat exchanger, its further development depends on the successful development of an inexpensive and effective heat exchanger capable of containing hydrogen or helium at the high operating temperatures of the Stirling Engine. Further projections indicate that the Stirling Engine would be viable even later than the gas turbine (7).

1.4.3 Electric Vehicles

The biggest obstacle to employment of electric vehicles, at present, is the state of battery technology (7). In order to obtain a comparable performance with a conventional vehicle the electric car must await the development of a battery capable of achieving a much higher energy density. The more advanced nickel-zinc and sodium-sulphur batteries might provide the range required. The question is whether the batteries can be supplied at an acceptable cost and whether safety and cold weather problems could be overcome.

1.5 Improvements in Conventional Vehicle Design

It is evident from the arguments presented in the last section (1.4) that none of the alternative power units studied is presently able to achieve better performance than that of the conventional engines. In addition the conventional engines have the major advantage that the investment required to manufacture, service and fuel them has already been made. It is expected, therefore, that conventional petrol and diesel engines will remain the dominant automotive power unit, at least up to the next century (5,8). The improvements in vehicle design considered in this work are, therefore, in the four major areas of: improvements in conventional engines; better engine-transmission matching; improved engine control;

reduction of vehicle characteristics of drag, weight and rolling resistance. The fuel savings possible by improvements in fuel and lubricants are not considered here.

1.5.1 Improvements in Conventional Engines

Before embarking on a description of proposed improved spark ignnition (SI) engine designs, it may be useful to consider the efficiency of the 'Otto' cycle which is given by the expression:

$$\text{Efficiency} = 1 - (1/r)^{\gamma-1} \qquad (1.1)$$

where

r = compression ratio

and

γ = ratio of specific heats of the working fluid at constant pressure and constant temperature

Although this expression is not strictly valid in practical engines, the simplification which it introduces highlights the dependence of thermal efficiency on the compression ratio and on the γ of the working fluid. γ increases from 1.3 to 1.4 as the fuel/air mixture is weakened and as equation 1.1 shows this can have a desired effect on thermal efficiency. In practice, the highest compression ratio that can be reached is limited by the onset of knock, and due to the environmental pressures to remove the lead antiknock additives from gasoline, the octane number of gasoline is unlikely to increase much in future.

The improvements in SI engine design are therefore concentrated on alterations to the geometry of piston and cylinder head and changes in fuel distributions. Experimental engines embodying these principles exist such as the high compression Leanburn May Fireball (9), the two inlet valve Honda CVcc (10), and the direct fuel injection combustion systems developed by Ford (11) and Texaco (12). The most promising of these designs is the May Fireball with many of the world's motor manufacturers experimenting with its variations (8). Prediction of fuel savings available at the turn of the century due to improvements in SI engine design vary between 7-20% (5,7,13).

In a diesel engine the fuel is introduced into the cylinder only when combustion is required, and pre-ignition does not occur. Moreover, since fuel is injected at a controlled rate the problem of detonation as in SI engines does not arise. The compression ratio is therefore much higher in diesel engines, with the upper limit being fixed by the strength of cylinder, the bearings, etc. The design of a diesel engine, therefore, involves a compromise between high efficiency and low weight and cost. A comparison of the air-standard efficiencies of the Diesel and Otto cycle when made at appropriate compression ratios (Fig 1.5) suggests that the compression ignition (CI) engine will be the more efficient. This is born out in practice, and in general CI engines are more efficient than SI engines.

For passenger car applications where high specific power output is required, indirect injection engines are normally used. These engines can operate reasonably quietly over a wide range of speeds compared with the direct injection engines normally used in trucks and buses. The most efficient type of the indirect injection diesel engine is the form supercharged with open combustion chambers (14).

The diesel engine presents an alternative to SI engine and may take more of a share of the car market in the future. However, this increased share is anticipated to be limited by a shortage of middle distillation. Although diesel fuel can be obtained by cracking, this is a less energy efficient method. In addition the cost of high pressure fuel pumps and turbo blowers are such that fuel costs and mileages would have to be very high, in order to provide a pay-off in reduced fuel consumption (14).

Due to the above reasons, this work concentrates on the possible improvements in SI engines.

1.5.2 Improvements in Engine-Transmission Matching

A typical CI engine map for a European four-cylinder spark ignition engine is shown in Fig 1.6. The ideal line represents the locus of minimum specific fuel consumption points. For optimum fuel economy the engine has to run close to this line. A number of gear box designs are proposed to improve engine-transmission matching like multi-speed manual or automatic transmissions with over drive gears (15,16). In order to obtain optimum engine operation at all conditions of load and speed a continuously variable transmission (CVT) must be employed. Due to the large number of proposed CVT designs, these drives are described in Section 3.

Prediction of fuel savings, possible by better engine-transmission matching, vary between 3-10% (13) up to 50% (17).

1.5.3 Improvements in Engine Control

The control system is inseparable from the type of transmission and, in the case of employment of electronically controlled gear boxes and continuously variable transmissions, the direct linkage system between the accelerator and the engine is likely to be replaced by a closed loop control system where some driver choice is eliminated. The closed loop system could lead to optimisation of the spark timing and the mixture strength. Furthermore, fuel can be cut off during idling and overrun conditions each of which has already been achieved separately in vehicles like the Volkswagon Formel E concept with fuel off at idle (18) and Austin Rover Maestro (fuel off at overrun) with problems of engine warm-up solved by the Volkswagon control system where fuel cut-off at idle is only performed on a warm engine. It is also proposed that this concept can lead to fuel savings of up to 30% in urban driving (18).

1.5.4 Improvements in Vehicle Characteristics of Drag, Weight, and Rolling Resistance

In substitution of the present materials for lighter alternatives, consideration of trading fuel economy improvements does not only include material costs but fabrication costs and the amount of weight reduction achieved by the substitution. Manufacturing costs of a component is usually several times the bare material costs. Indeed the transition from metals to plastics which is occuring in many products is mainly due to superior fabrication costs (19,20). Recent predications for the near term are that the application of high strength steel, cast aluminium, plastics and magnesium in vehicles will grow resulting in weight savings due to material substitution of up to 20% (19,20). It should be noted that these predictions are based on material changes due to changing energy supply and prices and ignore factors like governmental pressure.

All major type manufacturers have been working on reducing tyre rolling resistance. Most work is concentrated on improved tread and carcase design (7). Higher inflation pressures also result in some benefits. These improvements may be restricted, however, by safety requirements.

Given that the seating attitude and position of car occupants will not change, notable reductions in frontal area are not likely. Further reductions in airdrag coefficient, however, are achievable by tighter controls on vehicle styling and better airflow management.

Predictions of the combined effect of improvements in drag, weight and rolling resistance on vehicle fuel economy before the turn of the century range from 6-8% (7) to 20% (21) in fuel savings.

The fuel saving potential of the described improvements in conventional vehicle design is evaluated in this work by using 'JANUS' (Durham University Vehicle Simulation Package) described in Section 2.

Figure 1.1 - EEC-10 Inland consumption of petroleum products.

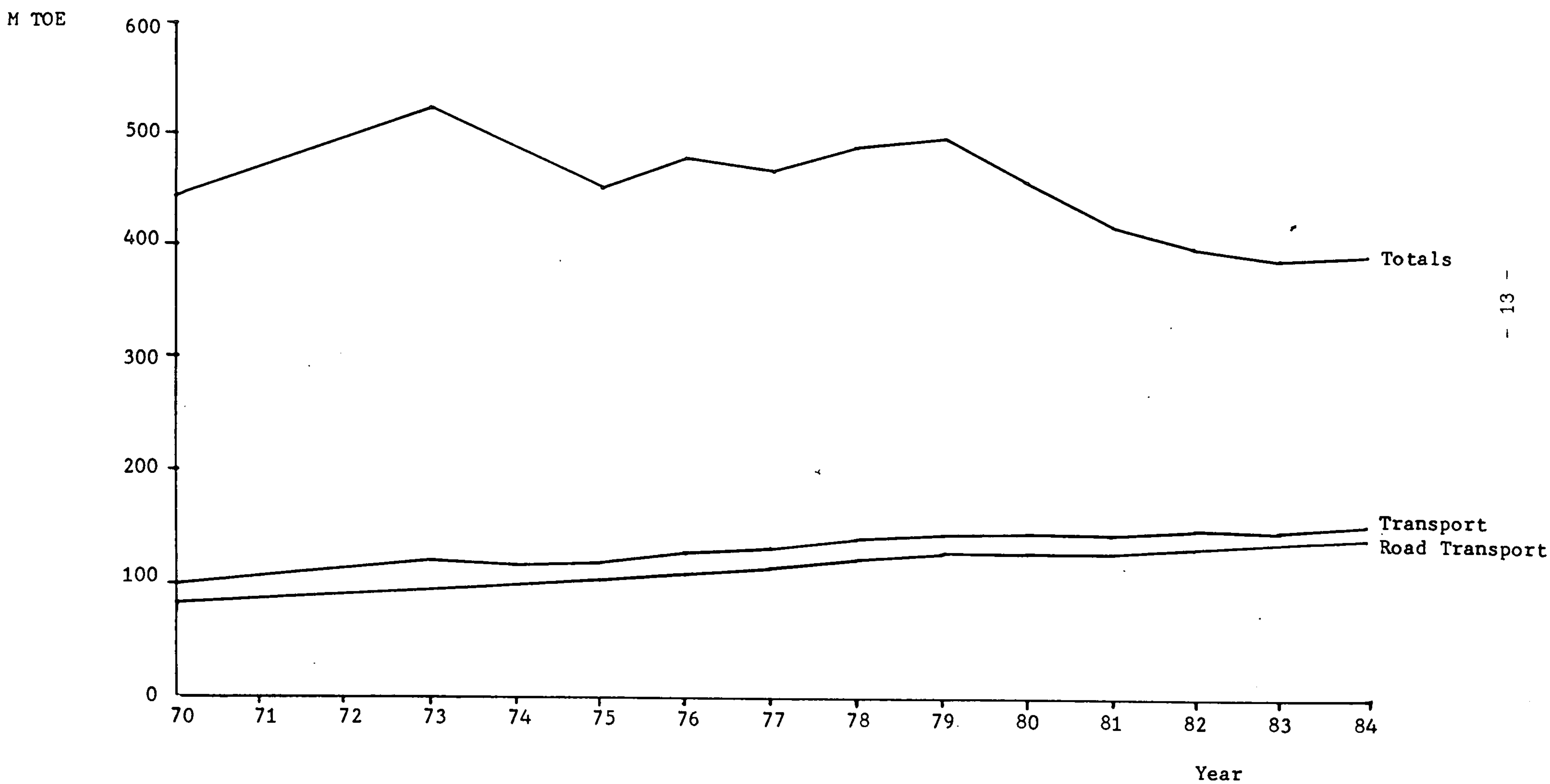


Figure 1.2 - U.K. Consumption of Petroleum .

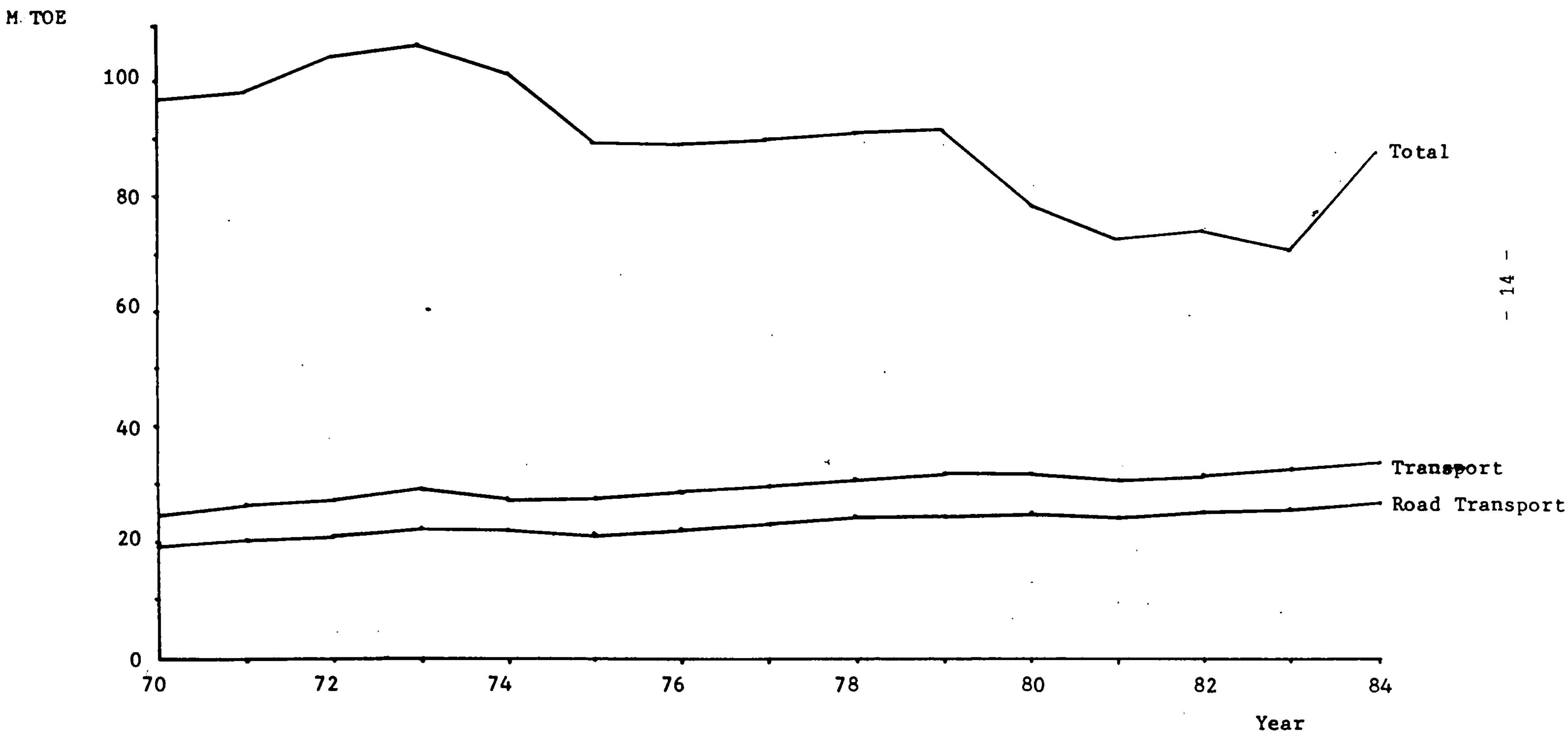


Figure 1.3 - Cars licenced in U.K. (1970-1984)

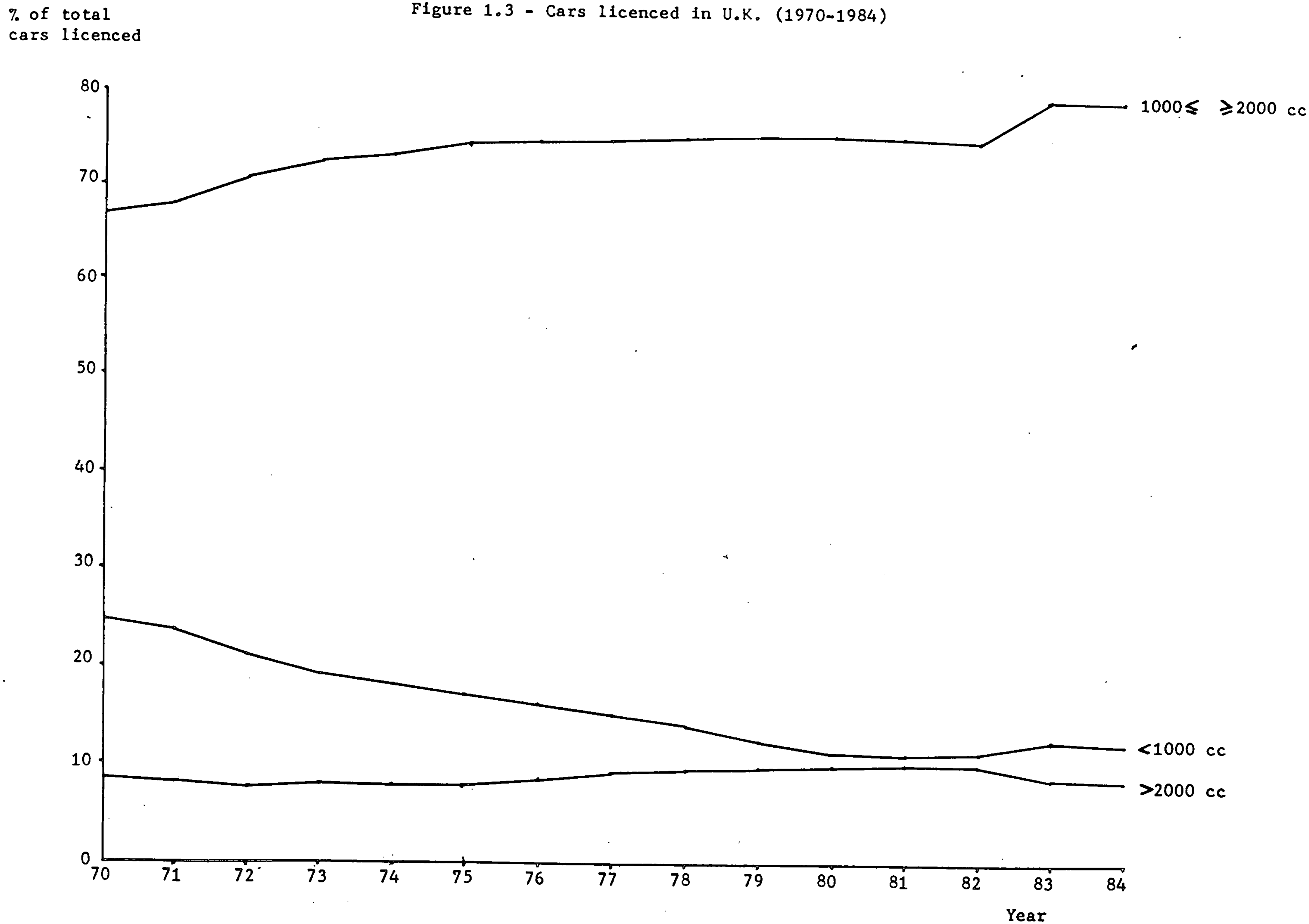


Figure 1.4 - Total number of cars licenced in U.K. and future projections.

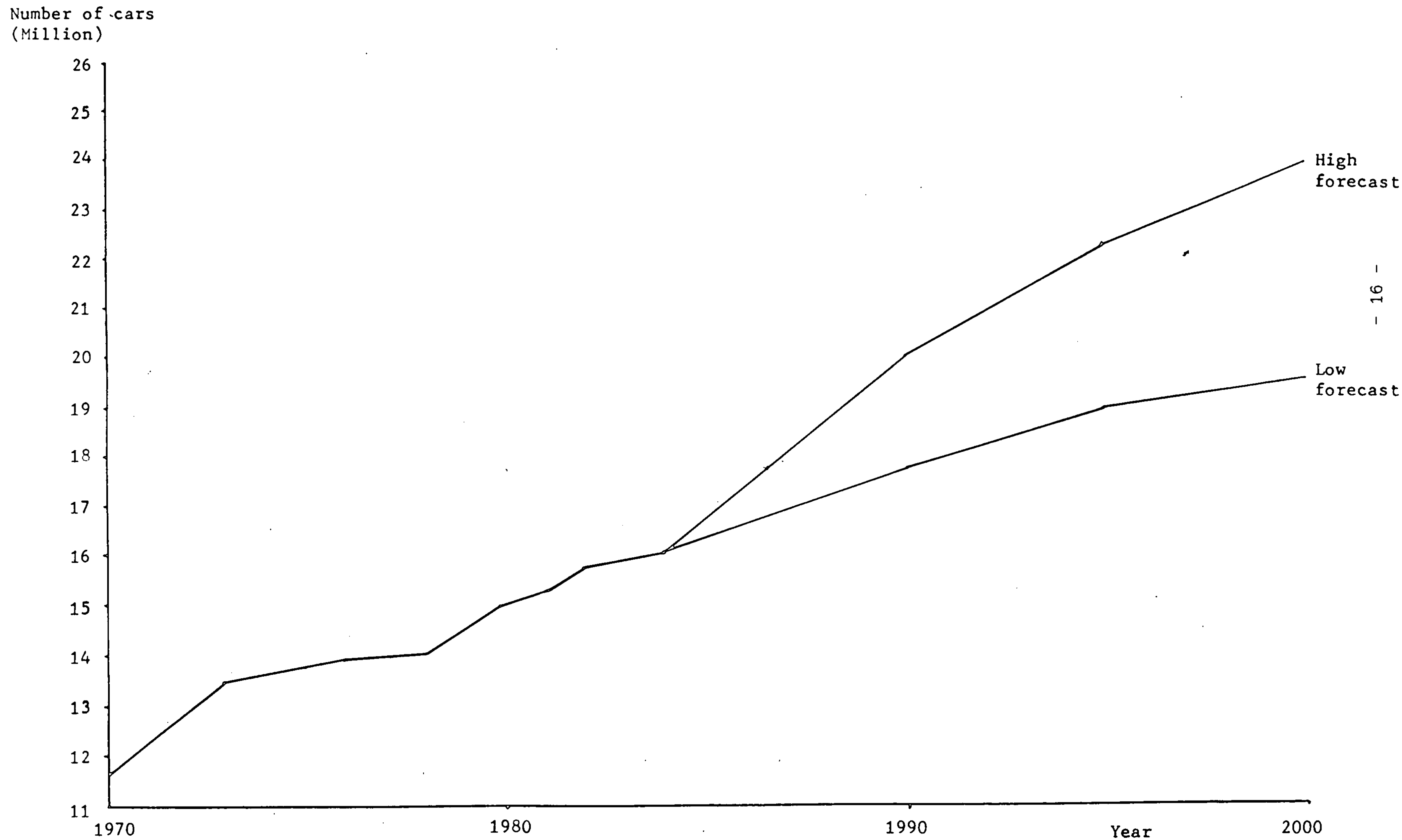
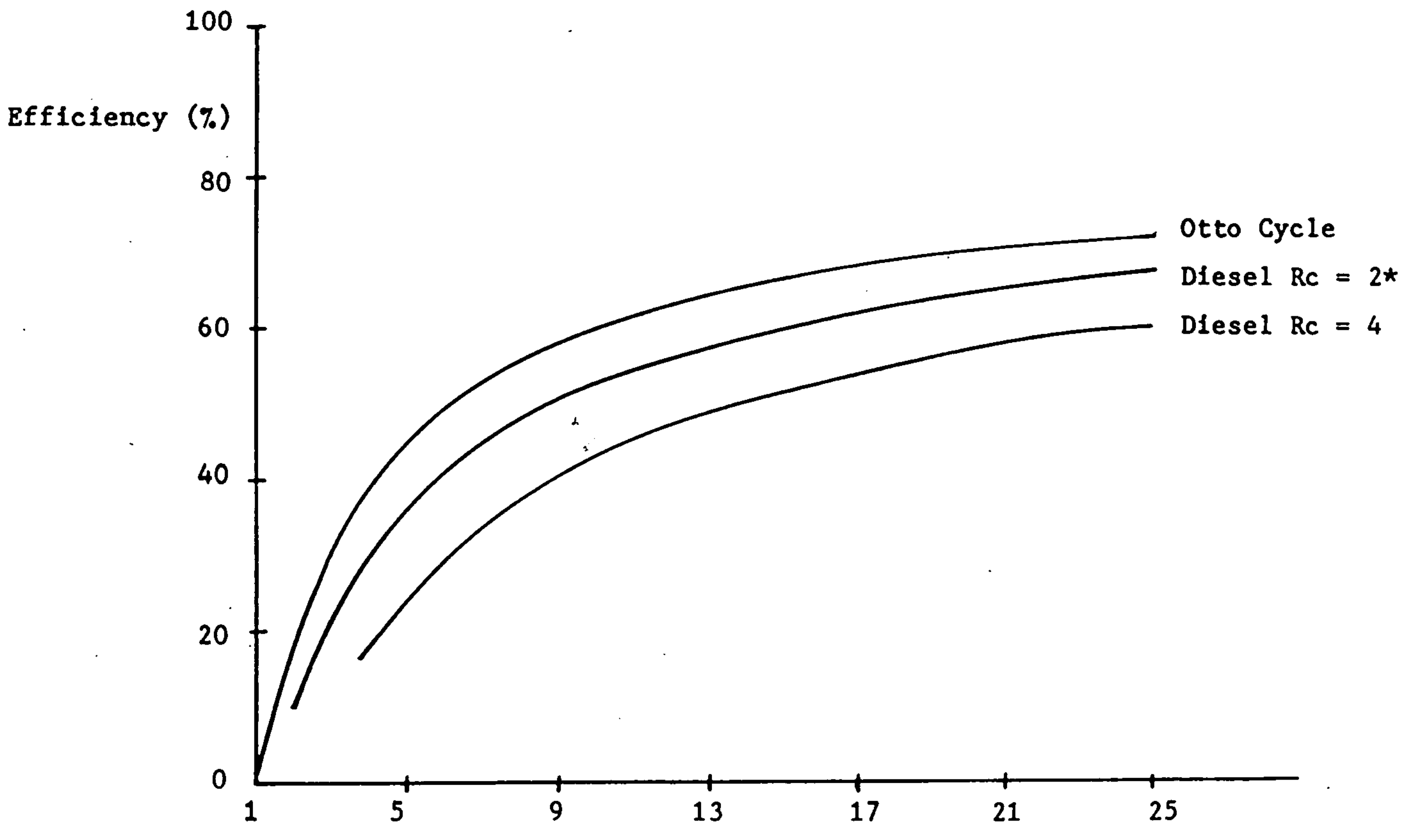
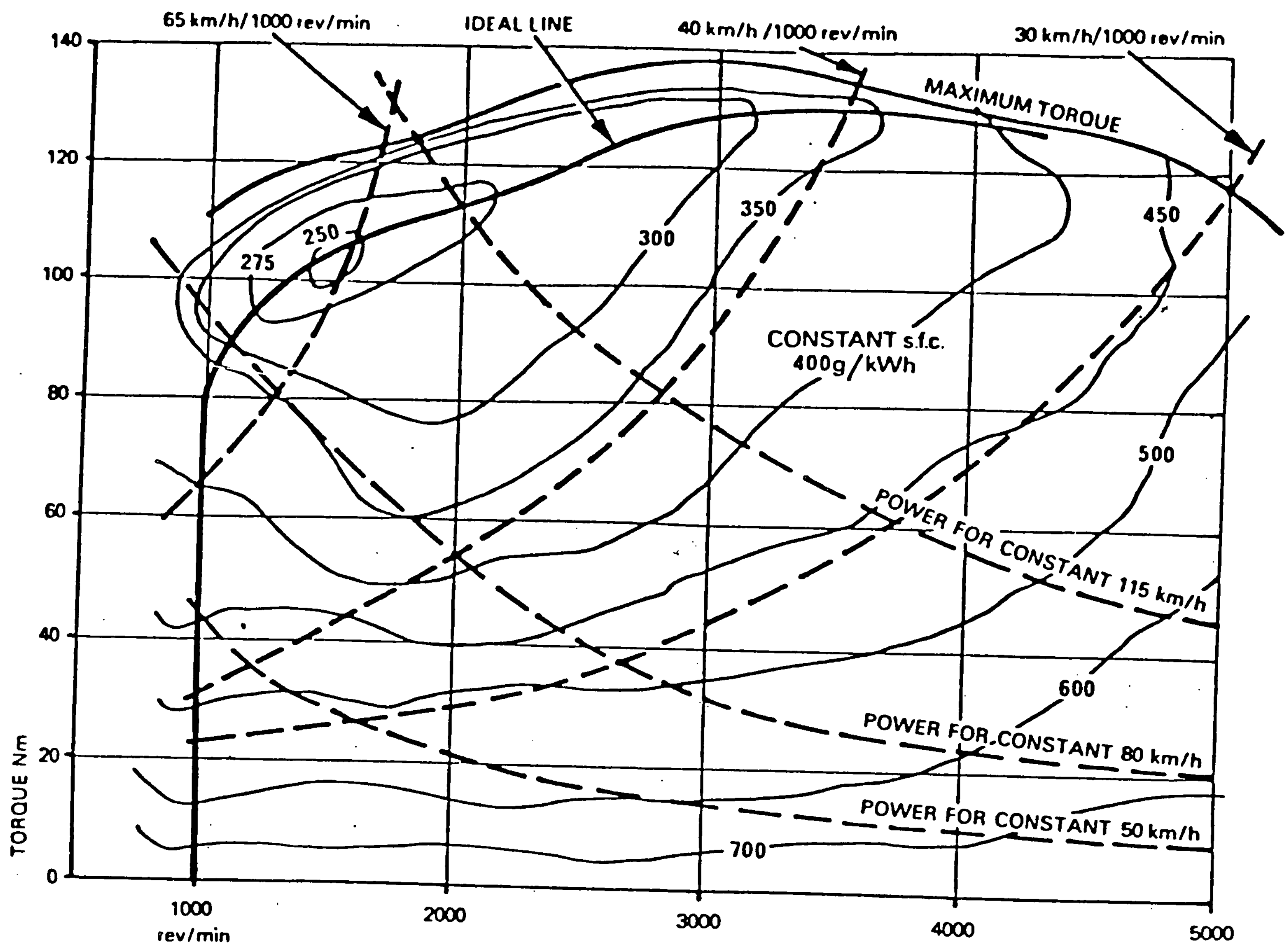


Figure 1.5 - A comparison of the air standard efficiency of the Otto and Diesel cycles .



* Rc = Cut Off Ratio .

Figure 1.6 - Engine map for a 1.85 litre Four Cylinder S.I. Engine.



2.1 Computer Modelling of Conventional Vehicles

Durham University simulation package 'JANUS' employs the approach shown in Fig 2.1, ie simulation is in an opposite direction to power flow. In order to obtain the flexibility required in simulation of different types of vehicles (eg electric, hybrids, etc) different subroutine blocks which represent individual vehicle components (and their efficiency variation at different conditions of load/speed) are assembled into a master program. In using this approach not only different types of vehicle could be simulated but also the degree of output could be extended by the user. The advantage of this type of structure can be illustrated by considering the layout of the internal combustion engined vehicle shown schematically in Fig 2.1 and the computer listing of the master program calling the software blocks used, in Fig 2.2. The names of the subroutines are selected to be similar to those of the components they represent to simplify program writing.

By employing a signalling system, at any particular period of the run, a certain section of the subroutine block is employed. The signals and their relevant operation are listed in Table 2.1 and the flow chart of this process is shown in Fig 2.3. In the first input section of the subroutine (eg transmission) most of the initial operating conditions could be specified. In order to evaluate other initial

CHAPTER TWO

COMPUTER MODELLING OF CONVENTIONAL VEHICLES

operating conditions (eg transmission weight), however, information from subroutines further down the block (Fig 2.2) may be required. Therefore after the first input section another pass round the blocks (ie second input section) is required in order to evaluate the remaining input data.

Having established the initial operating conditions, the simulation enters the first dynamic section, which determines the instantaneous requirements and conditions of each component, considered to be independent of other components. In some cases (like wide open throttle operation described in section 2.3.1), however, iterations around the block of subroutines might be necessary, in order to reach a steady state. Therefore the signal is kept in this mode until stability is achieved. The second dynamic section will then determine the total energy requirements and average efficiency of the component up to that time step. The process in the dynamic section is repeated for each time step, until the end of the driving cycle is reached.

Finally the last pass round the block of subroutines is the output section which provides the required information from each subroutine.

Once a simulation run has been completed, the controlling software allows further runs to be conducted with a different initial condition. The subroutine blocks used in simulation of internal combustion engined vehicles are described in the following sections.

2.2 'VC'

This subroutine represents the vehicle body specifications. The user can either specify vehicle weight, drag coefficient, frontal area and passenger/pay load weight data, or can select one of the 'standard' vehicles available for which the data is stored in the program.

In the former case the weight of the vehicle can be specified manually (ie input the body shell weight and each power train component weight as its subroutine is called) or alternatively the weight could be evaluated automatically by the program according to the vehicle power rating. It should be noted that when a component weight is added, the subsequent increase of body shell weight is also considered. The required weight propagation factor (WPF) can be either a user input or a standard ~~valve~~ value of 35% is used as the default level.

At the end of each simulation run this subroutine outputs vehicle body specification and if necessary weight of each component of the power train.

2.3 'DCYCLE' (Driving Cycle)

This subroutine represents the driving conditions which the simulated vehicle is expected to meet. The initial inputs to the subroutine are:

1. Type of driving cycle
2. Time step (secs)
3. Environmental conditions.

This program currently contains twelve urban driving cycles, a variable speed cruise cycle and a wide open throttle (WOT) acceleration test cycle. The time step value and environmental conditions could be defaulted to the values shown in Table 2.2.

Driving dynamic operation 'DCYCLE' determines average velocity and acceleration during the specified time step from the driving cycle data. At the end of each run this subroutine outputs the percentage of total time spent in each driving mode (ie acceleration, cruise, etc), total time, distance and average velocity of the vehicle over the driving cycle. The operation of WOT cycle, however, is different.

2.3.1 'WOT' (WOT Acceleration Test)

The WOT acceleration test simulation differs with the other driving cycles. The approach used is shown in Fig 2.4. The initial inputs to this subroutine are:

1. Initial and final velocity of the vehicle (MPH)

2. Estimated time to reach final velocity (secs).

By using the estimated time input and assuming linear acceleration, the initial acceleration rate is estimated. The velocity and acceleration rate are then supplied to the wheels subroutine and finally through the IC engine simulation, where the power and rotational speed requirements are checked against the full throttle line of the engine currently employed. If the power requirement is within 1% of the limiting (FI) power, the acceleration rate is recorded for that time step and from that the new initial velocity (for the next time step) is evaluated. If the required power does not correspond to the limiting power, however, a signal is sent back to the WOT subroutine to change the acceleration rate. In addition to the signal, the power developed (PE) and the limiting power (PL) values are also sent back. The new acceleration rate is evaluated as follows:

$$PE1 = PE - (PR + PG + PD) \qquad (2.1)$$

$$PL1 = PL - (PR + PG + PD) \qquad (2.2)$$

$$Acold = Ac \qquad (2.3)$$

$$Ac = Acold * PL1/PE1 \qquad (2.4)$$

$$Acnew = Ac + 0.8* (Ac - Acold) \qquad (2.5)$$

PL1 and PE1 refer to that part of limiting (PL) and developed power (PE) spent on acceleration, ie values of power spent on rolling resistance (PR), gradient (PG) and airdrag (PD) are deducted from the total.

This process (EQ 2.1 to 2.5) is repeated until the demanded power and the limiting power values correspond.

At the end of each simulation run 'WOT' outputs initial and final velocities and the minimum time required by the vehicle to attain the final speed.

2.3.2 Ability to Reduce Vehicle Acceleration Rate When Engine Limiting (F.T.) Power is Exceeded

If a small engined vehicle is driven over an urban cycle, during a certain part of the cycle the power or speed requirement might exceed the limiting power or maximum speed of the engine. In this case the vehicle is not able to follow the driving cycle as it stands. If this occurs a warning is flagged to the user's VDU screen and the cycle velocity and acceleration reduced until the maximum allowable engine ouput is attained. This way the vehicle will follow the performance it can achieve until another point is reached in the driving cycle which it is able to meet. The simulation process employed is similar to that of the WOT operation. Fig 2.5 shows the standard ECE-15 driving cycle and the cycle actually achieved by a 15 KW engined vehicle.

2.4 '<u>Wheels</u>'

The initial inputs to this subroutine are:

1. wheel radius (m)
2. effective vehicle weight increase due to wheel inertia (KG)'

The subroutine also receives vehicle data from 'VC' and driving conditions from 'DCYCLE' and evaluates the road power and energy requirements as follows:

1. Calculates air density

   $$Da(Kg/m^3) = D15 * \frac{288}{(273+Te)} * \frac{B}{760} \qquad (2.6)$$

   where D15 is the density of air at $15^{\circ}C = 1.226\ Kgm^3$

2. Evaluates the tractine effort (N)

   Due to air drag

   $$TED = 0.5 * Da * CD * A * (V+VW)^2 \qquad (2.7)$$

   Due to gradient

   $$TEG = G * W \qquad (2.8)$$

   Due to rolling resistance

   $$TER = CR * W \qquad (2.9)$$

   Due to acceleration

   $$TEA = Ac * (W + WINR) \qquad (2.10)$$

Wheel inertia is considered as an effective weight increase in acceleration and is therefore included in the accelerative tractive effort term. During normal operation the total tractine effort is therefore:

$$TE = TEA + TED + TER + TEG \qquad (2.11)$$

During overrun conditions, however, the total tractine effort available is:

TE = TEA - (TED + TER + TEG)

3. Wheel torque and rotational speed

TORQW = TE * WR (2.12)

RW(RPM) = 9.549 * V/WR (2.13)

4. And finally, wheel power and energy requirements

POWRW (W) = (2π/60) * TORQW * RW (2.14)

E = POWRW * H (2.15)

At the end of each run 'wheels' print-out total motoring energy requirements with its components (ie accelerative, rolling resistance, etc) over the driving cycle. A graph of wheels power requirement against time is also available.

2.5 'AXLE' (Final Drive)

The initial inputs to this subroutine are:

1. Final drive type
2. Final drive ratio.

The user can either select one of the six final drive types available, Table 2.3, or input a constant efficiency.

The principal dynamic inputs to this subroutine are wheels torque and rotational speed (TORQW,RW). As the final drive ratio is known then the final drive input speed can be evaluated.

RD = FDR * RW (2.16)

In order to determine final drive input power its efficiency has to be determined.

2.5.1 Final Drive Efficiency

If a constant efficiency final drive type has been selected then input power is evaluated as follows:

$$POWRD = POWRV/EFFDRV \qquad (2.17)$$

For spur gear or bevel-gear type final drive, however, the efficiency variation with load and speed has been considered and input power is determined by the iterative process of Fig 2.6 which employs equation 2.18 to evaluate power loss at the differential.

$$PLOSS\ (W) = AK * POWRD + 1.14 * 10^{-8} * RD^{2.1} \qquad (2.18)$$

where $AK = 0.02$ for spur gear type

$AK = 0.05$ for bevel gear type

The reasons for setting the churning loss index at 2.1 are discussed in Section 2.6.1.

Once the power loss has been determined, the axle power requirement is set to:

$$POWRD = POWRW + PLOSS \qquad (2.19)$$

The efficiency of spur gear type final drive calculated by equation 2.18 for a range of input power and speed is shown in Fig 2.7. At completion of each run 'AXLE' outputs the total energy requirements and the decelerative energy available at final drive and its average efficiency during both modes of operation.

2.6 'TRANS' (Transmission)

Initial inputs to 'TRANS' subroutine are as follows:

1. Type of transmission
2. Number of gears
3. The gear ratios
4. The speeds at which gear shifts are required or the choice of optimum gear shifting.

The types of transmissions available are manual gear box, automatic gear box, and continously variable transmissions. The latter are described in detail in Chapter 3.
The user can select from a two-speed to six-speed gear box. Six-speed transmissions seem a possible progression to five-speed gear boxes currently employed in some vehicles. The gear shifts sequence can either be specified or on default the program specified 'standard' shifts shown in Table 2.3 are employed. It should be noted that the first two shifts' velocities are the standard shift velocities of ECE 15 driving cycle (Fig 2.8). Different shift speeds for down shifting can also be specified.

In addition the user can ask for optimum gear shifting for fuel economy, operation of which is described in Section 2.6.2.

The principal dynamic input to 'TRANS' are axle input torque and rotational speed (TORQD, RD). Once the gear ratio is selected by one of the pre-described methods the transmission input speed can be determined as follows.

$$RG = RD * GR \qquad (2.20)$$

It should be noted that, as the time taken during the gear shift is small compared with the total cycle time (Fig 2.8), the simulation assumes instantaneous gear changes. In order to evaluate gear box input torque and power requirements, transmission efficiency has to be evaluated.

2.6.1 Gear Box Efficiency

Gear-box losses are either load dependent as gear tooth and bearing sliding losses or speed dependent such as churning and windage losses. The gear box power loss model is therefore a compromise between the load dependent and speed dependent losses.

$$PLOSS = K1 * K2 * POWRG + 1.14 * 10^{-8} * RG^{2.1} \qquad (2.20)$$

where K1 and K2 are constants

and K1 = 0.02 , K2 = 1.0 for manual gear box

K1 = 0.03 , K2 = 1.0 for automatic gear box.

The K2 valve for the fourth gear (direct drive) is, however, reduced to 0.5 for the manual gear box.

The ~~valve~~ value of the churning loss speed index has been assumed at 2.1 to obtain efficiency values typical of those cited in the literature (22,23). The efficiency (non-direct drive) of a manual gear box calculated by equation 2.20 is shown in Fig 2.9.

In order to evaluate the input power to the gear box the iterative process of figure 2.10 is used and once it has converged

$$POWRG = POWRD + PLOSS \qquad (2.21)$$

On completion of each simulation run, 'TRANS' outputs the total energy required and the decelerative energy available at the transmission and the average efficiency during each of these modes. The gear shift sequence over the driving cycle is also available on request.

2.6.2 'OPTGEAR' (Optimum-Gear)

As mentioned earlier the user can ask for optimum gear shifting. When 'OPTGEAR' is utilized the normal shift sequence of 'TRANS' is overriden and for each time step the optimum gear is selected. The criteria for selection of the optimum gear is the gear which would result in an engine speed closest to the speed on the minimum specific fuel consumption curve for that power level. Fig 2.11 shows the flow chart of 'OPTGEAR'. It receives transmission output torque and rotational speed requirements. The output power is divided by an estimate of transmission efficiency to give engine output power following which the ideal speed for that power level is determined. Then for first gear the resultant engine speed is evaluated and checked that it is within the range of engine operating speeds and that the required engine power is below the full throttle power at this speed. On failing any of the described checks this gear is not selected. This process is repeated for all the gears of the transmission and the gear resulting in an engine speed closest to the ideal speed is selected.

It should be noted that during deceleration this process is not employed and normal shift sequences of 'TRANS' is employed.

2.7 'COUPL'

The initial input to this subroutine is the type of coupling required. The types available are:

1. Friction clutch
2. Torque convertor with lock-up
3. Torque convertor.

Quite evidently, the reason for the lock-up is to eliminate, above a certain vehicle speed, the hydraulic losses resulting from the slippage of convertor rotating elements. Here, the non-lock-up slip is set at 5%. Fig 2.12 shows the variation of the efficiency model with speed ratio for the clutch and torque convertor.

'COUPL' determines whether the input rotational speed of the gear box is below the engine's minimum operating speed and if so sets engine speed to its minimum value. 'COUPL' also determines the required engine output power as follows.

$$POWRE = POWRG/EFFCOP \qquad (2.22)$$

At the end of each run it prints out the average efficiency of coupling for both normal and decelerative modes of operation.

2.8 'ICENG' (I C Engine)

The primary initial inputs to this subroutine are:

1. IC Engine type (ie selection of fuel map)
2. Engine maximum power (KW)
3. Engine maximum speed (RPM)
4. Minimum idle fuel consumption (litres/sec)
5. Engine inertia (KGM^2)
6. Whether fuel cut-off during idle and/or overrun is required.

The principal dynamic inputs to 'ICENG' are instantaneous engine speed and torque output requirements.

Engine Inertia Torque-Loss

As engine efficiency maps are obtained from steady-state load tests to obtain net engine torque during an acceleration interval, the engine inertia torque loss must be added algebraically to the output torque. As the engine inertia torque is proportional to engine angular acceleration, it is determined as follows:

$$\text{TORQIN (NM)} = I * a \qquad (2.23)$$

where a $\quad (M/S^2) \quad = \frac{2\Pi}{60} * \Delta RE/H \qquad (2.24)$

Therefore

$$\text{TORQIN (NM)} = 0.1047 * I * (RE-REold)/H \qquad (2.25)$$

where RE and REold represent engine speed at the present (n) and the previous (n-1) time steps. For the start of the cycle, the value of REold is evaluated by calculating the engine speed corresponding to the initial velocity.

The instantaneous simulation of gear change (Section 2.6) results in a step change in engine speed. In order to prevent translation of this step change into a high and false torque loss, during the time step in which gear shift occurs, engine inertia torque loss is neglected. This can be justified due to the usually small effect of engine inertia on fuel economy.

Once the inertia torque loss is evaluated, the torque and power developed by the engine are found as follows.

$$TE\ (NM) = TORQE + TORQIN \qquad (2.26)$$

and $$PE\ (W) = TE * RE * 2\pi/60 \qquad (2.27)$$

2.8.1 'CHECK'

This subroutine checks that the power developed by the engine does not exceed full throttle power at that particular speed. If so, a signal is sent to 'DCYCLE' subroutine to reduce the vehicle acceleration rate, and therefore final speed during the particular time step. This subroutine also checks that the engine speed demanded is below the engine maximum speed, failing which the simulation displays a warning and stops.

2.8.2 Engine Fuel Maps and 'SEARCH'

The fuel maps employed by JANUS are of the BMEP/Speed or Torque/Speed form. For computational purposes, the engine map data is expressed by storing the specific fuel consumption values in a matrix of up to 20 x 20 representing BMEP (or Normalised Torque) against normalised engine speed. The engine torque and speed are stored in the normalised form to permit the engine map to be used with different power ratings. It should be noted, however, that this should be done with extreme caution unless experimental data from a range of engine sizes is available. Fig 2.13 shows a typical

40 Kw engine stored in BMEP/speed format in JANUS. By employment of the 'SEARCH' subroutine the fuel consumption at any specific load and engine speed is evaluated by interpolating between the four specific fuel valves adjacent to this operating point.

At low engine torques, the operating point may lie below the first set of fuel consumption data stored in the map. It is then necessary to use as one the interpolation points fuel consumption at zero output torque and high engine speed. Such fuel consumption values are determined by assuming the idle fuel consumption to be increased in the ratio of operating speed to idle speed (24).

When the vehicle is stationary or decelerating, fuel consumption is set equal to the idle fuel consumption. In order to study the effect of fuel cut-off at idle or overrun conditions, the idle fuel consumption can be set to zero.

At the end of the cycle simulation, the total fuel used is calculated by adding up the fuel usage values of each time step.

2.8.3 Engine Usage Maps

Another feature of the 'ICENG' subroutine is that it records the amount of time spent at each specific region of power and speed requirement in a matrix equal in size to that of engine fuel map used. On completion of each run the engine usage map is printed-out which displays the percentage of total time spent at each region of engine map during the cycle. Figure 2.14 shows the 40 Kw engine of Fig ~~4.13~~ 2.13 and the fuel usage of a typical car of this power rating driven over the ECE-15 driving cycle.

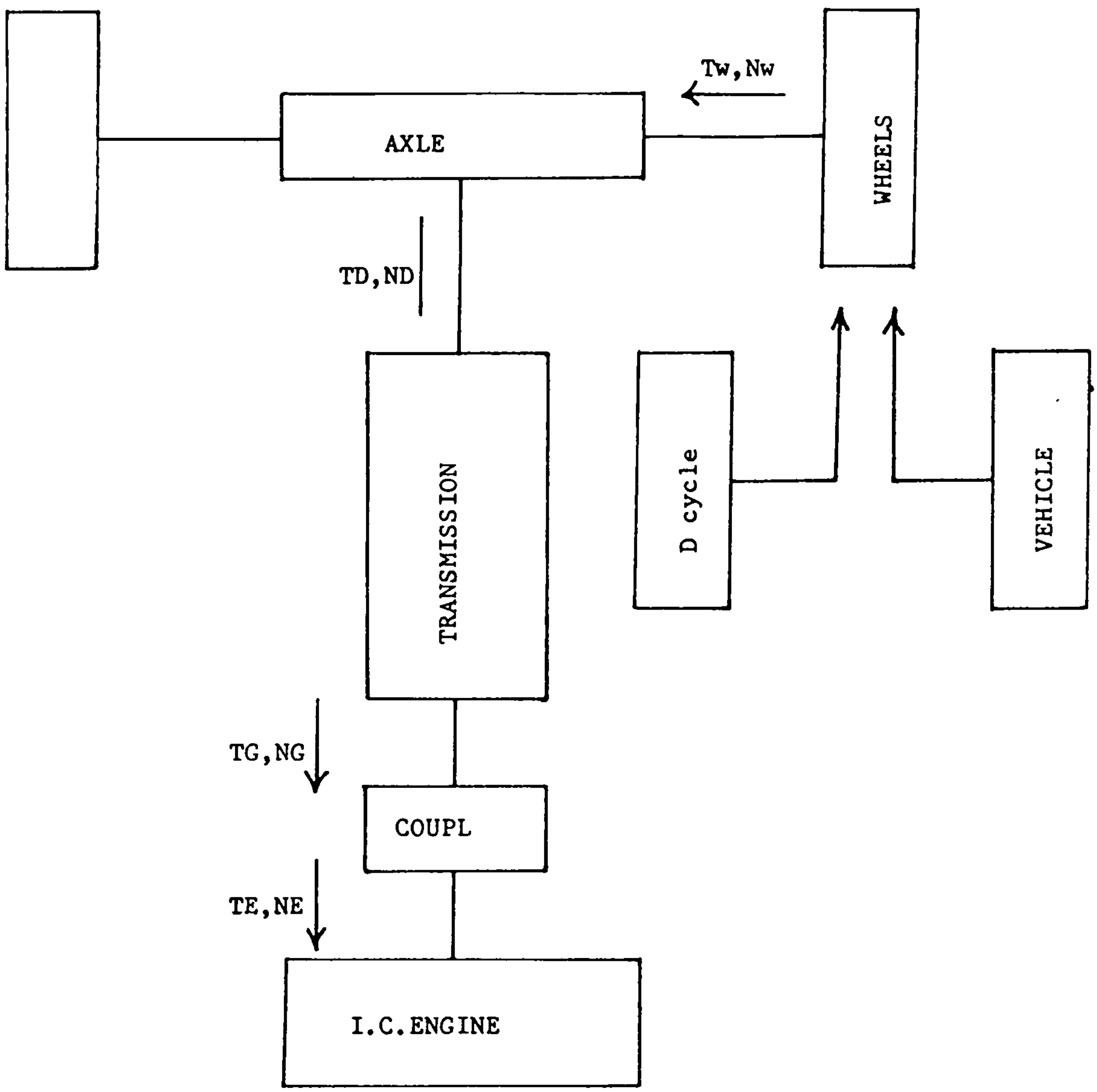


Figure 2.1 - Conventional vehicle simulation approach .

```
$BATCH
$INCLUDE 9,JANUSST.FTN/G
C
C I.C. ENGINED VEHICLE SIMULATION
C
C     *********** VEHICLE SUBROUTINE STRUCTURE ***********
      CALL VC(IFLAG)
      CALL DCYCLE(V,AC,IOPM,EFFDT,IFLAG)
      CALL WHEELS(V,AC,TORQW,RW,PB,IFLAG)
      CALL AXLE(TORQW,RW,TORQD,RD,PEMAX,REMAX,EFFDT,IFLAG)
      CALL TRANS(TORQD,RD,TORQG,RG,GR,EFFGB,GEAR,PEMAX,TRATE,
     *EFFDT,RMIN,REMAX,CVTP1,CVTP2,CVTS1,CVTS2,NCVT,PLIM1,RLIM1,
     *10,IFLAG)
     CALL COUPL(TORQG,RG,TORQE,RE,RMIN,REMAX,PB,EFFDT,IFLAG)
     CALL ICENG(TORQE,RE,PEMAX,TRATE,RMIN,REMAX,CVTP1,CVTS1,
    *PLIM1,RLIM1,NL,CC,PB,EFFDT,IFLAG)
C    ********** END OF VEHICLE SUBROUTINE ENTERIES *********
C
$INCLUDE 9,JANUSF.FTN/G
$BEND
```

Figure 2.2 - A list of 'JANUSIC' , the Master programme for simulation of I.C. engined vehicles.

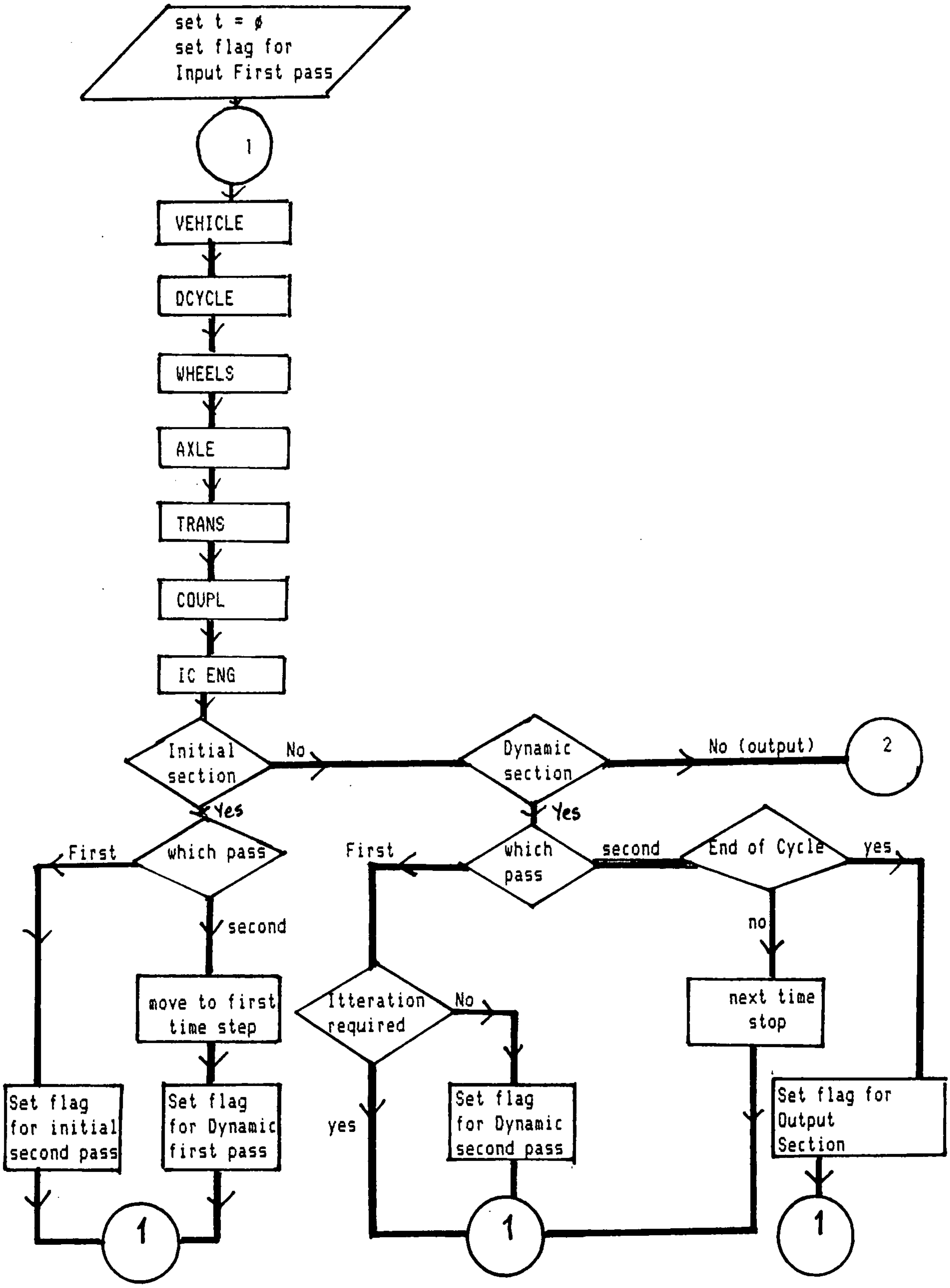


FIG 2.3 - Overall flow chart of the conventional vehicle design (continued on next page)

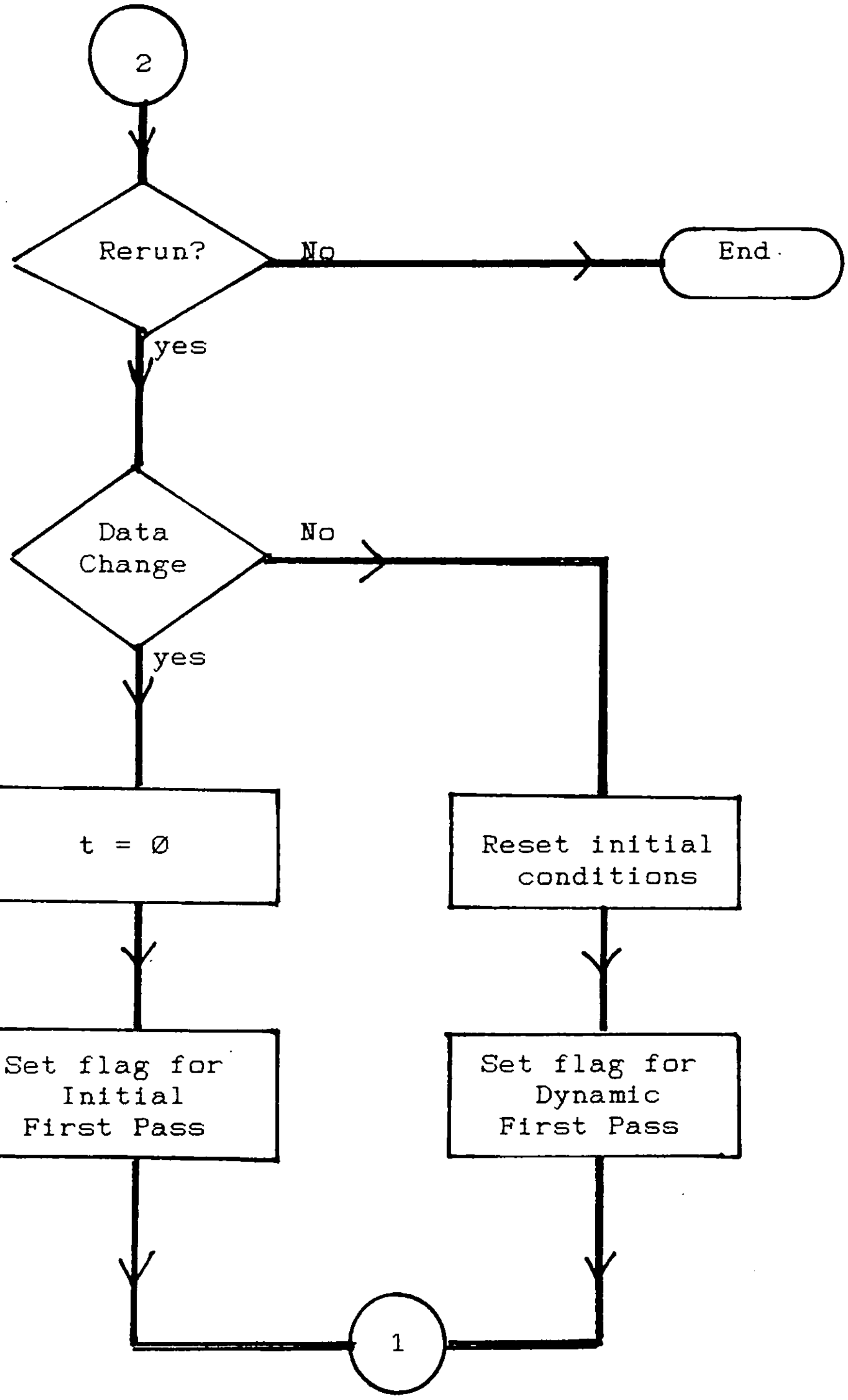


FIG 2.3 - continued.

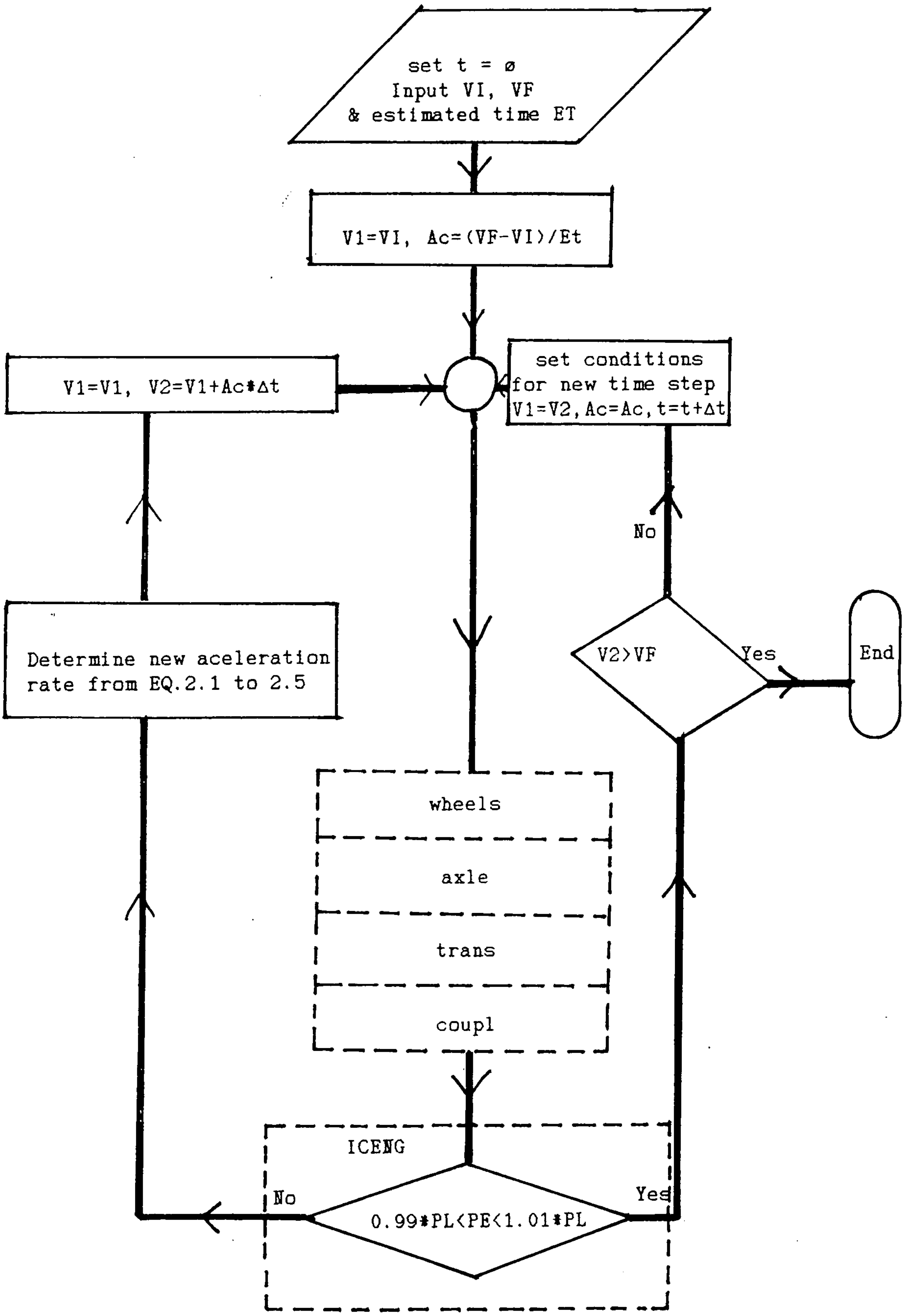
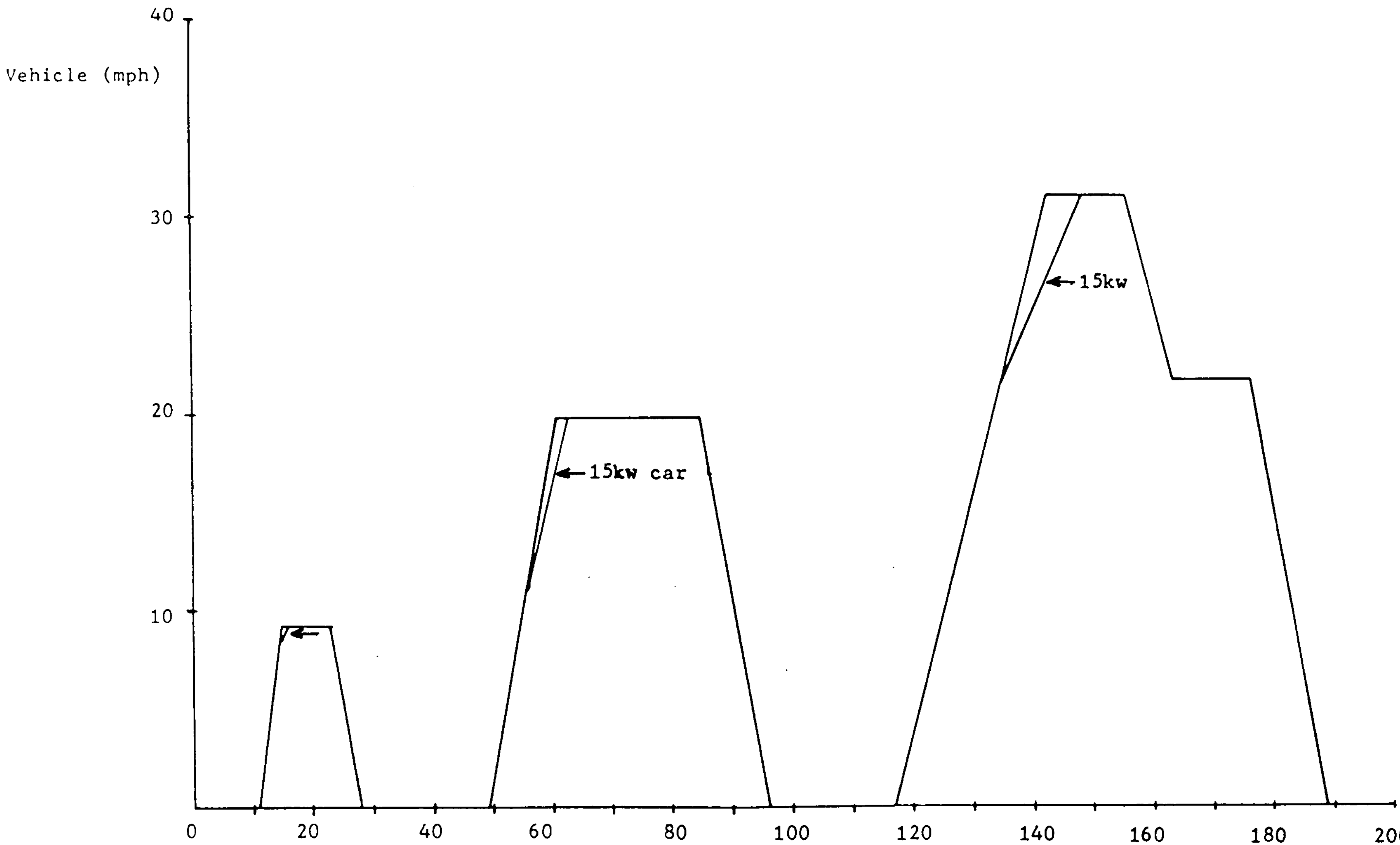


FIG 2.4 - Overall flow chart of WOT operation

Figure 2.5 - ECE 15 driving cycle and that met by a 15kw Small car .



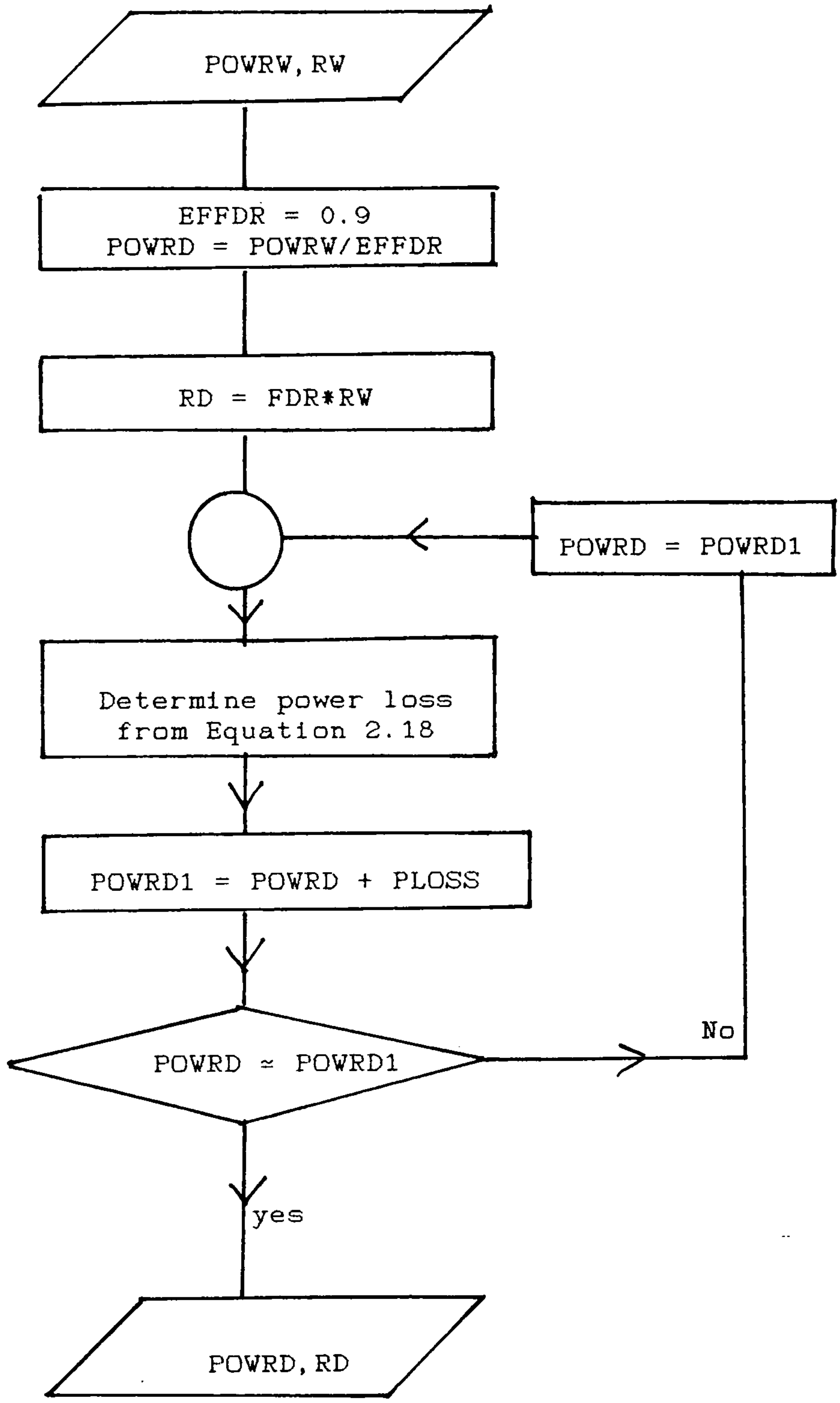


FIG 2.6 - Spur or bevel gear type final drive efficiency simulation process.

Figure 2.7 - Final Drive Efficiency (Spur - gear type) .

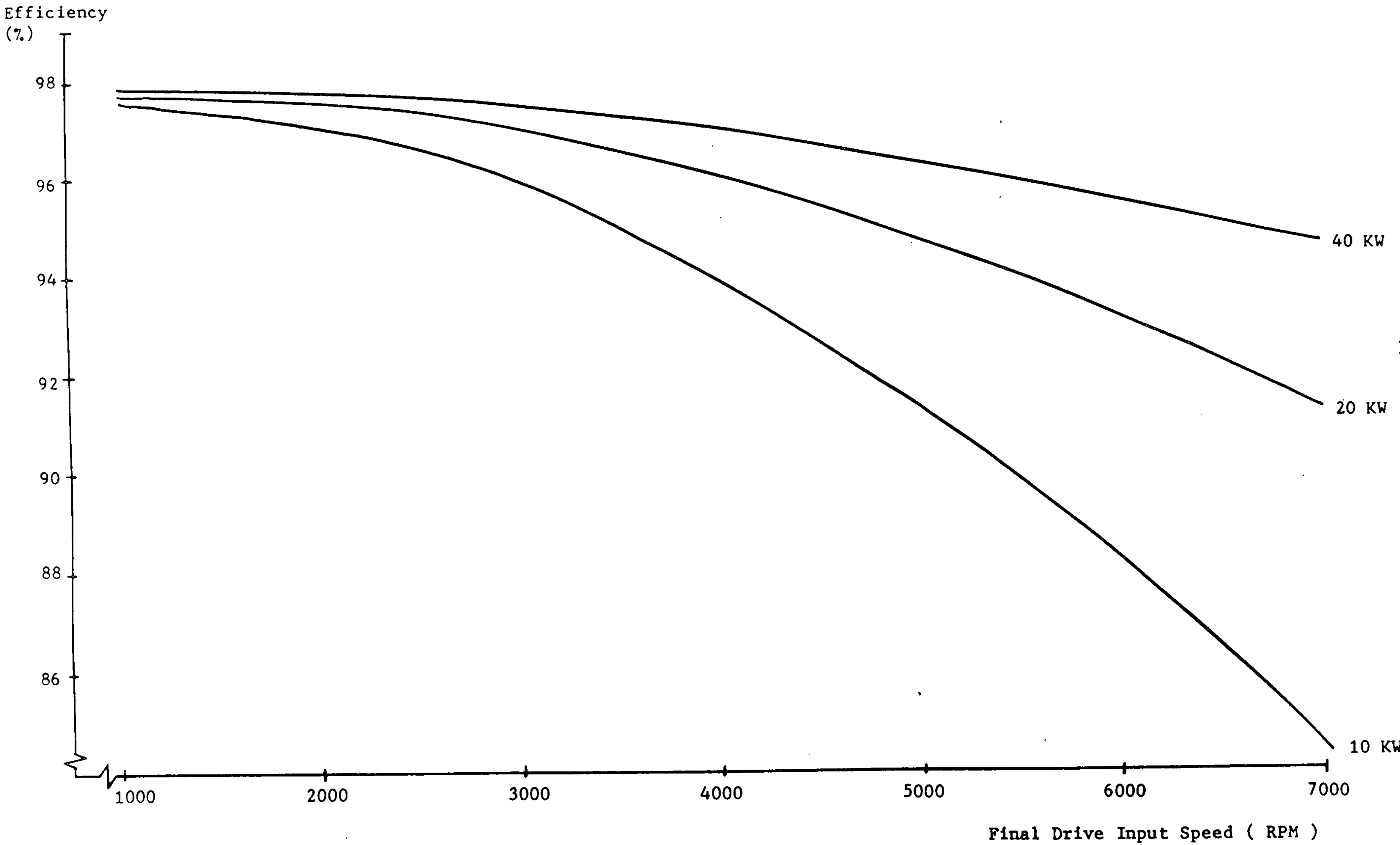


Figure 2.8 - ECE-15 Urban Driving Cycle.

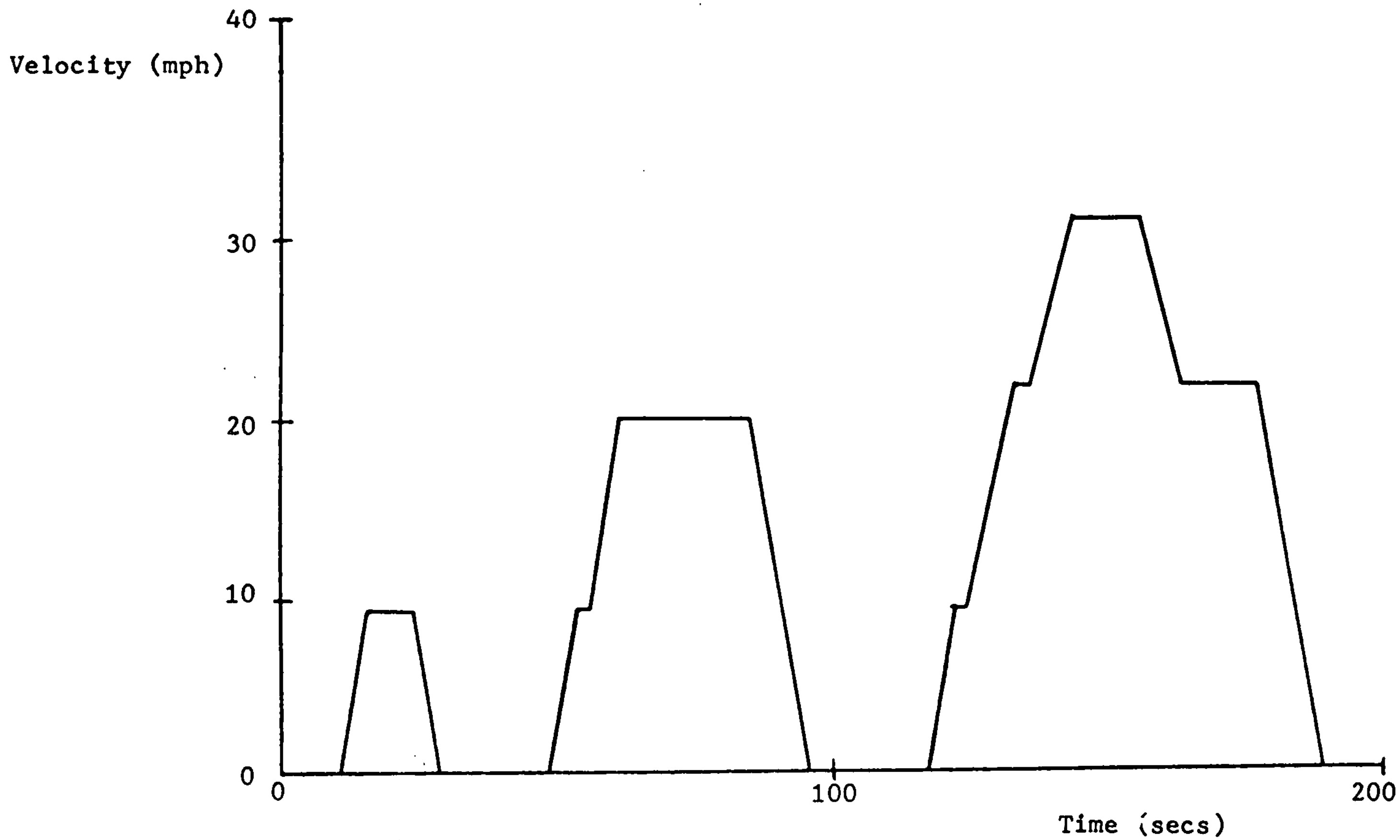
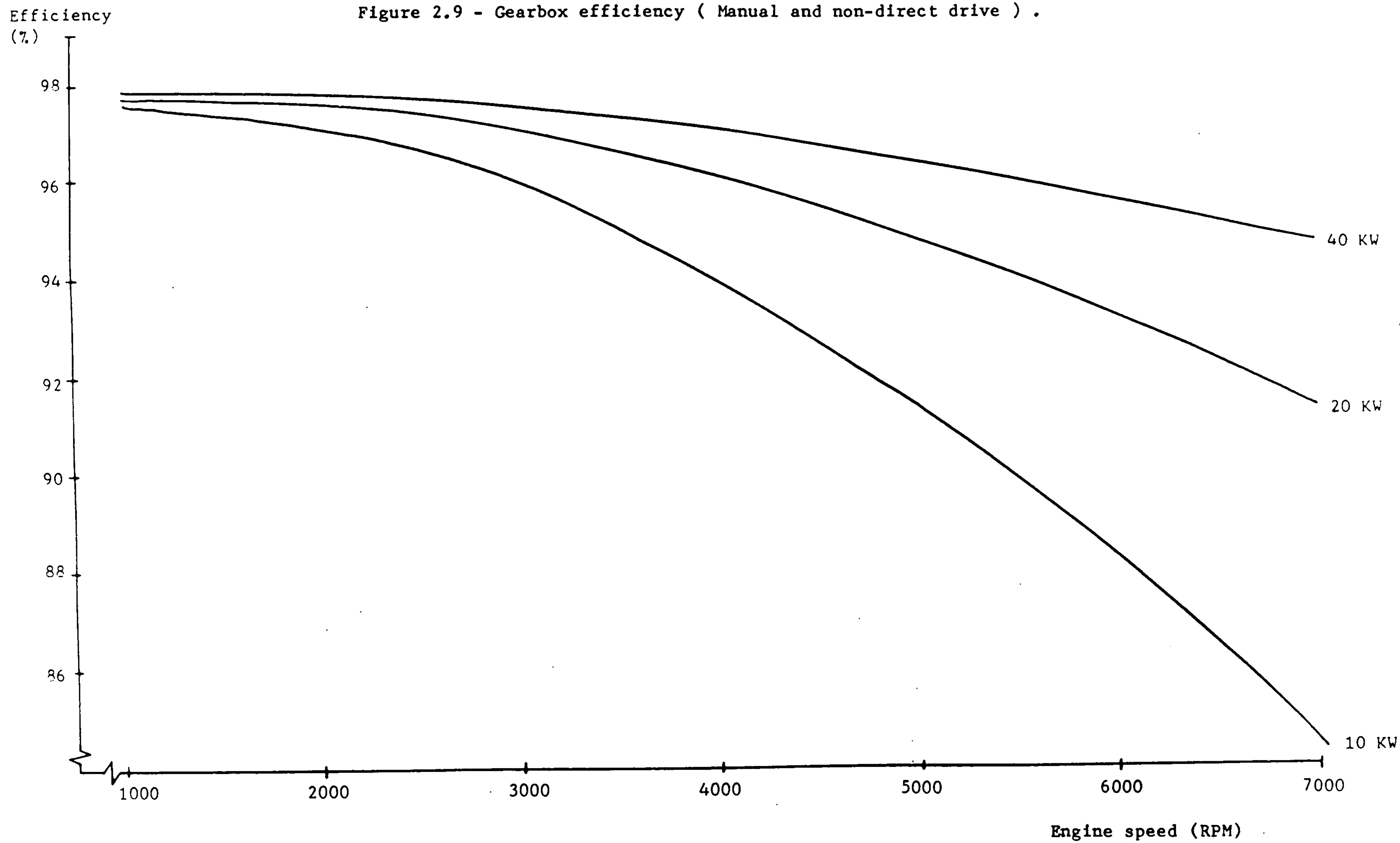


Figure 2.9 - Gearbox efficiency (Manual and non-direct drive) .



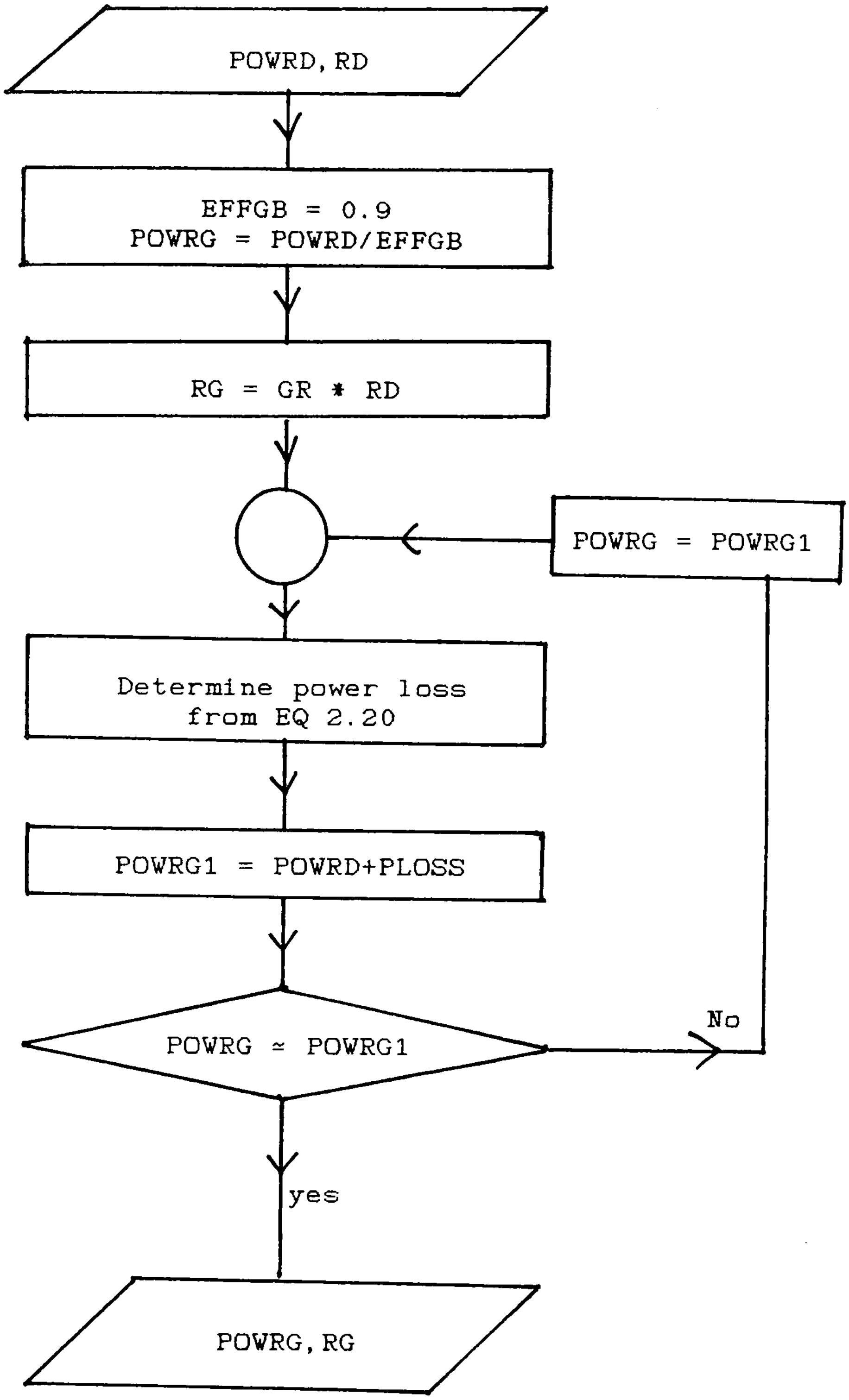


FIG 2.10 - Process employed to evaluate power loss in conventional gearbox by JANUS

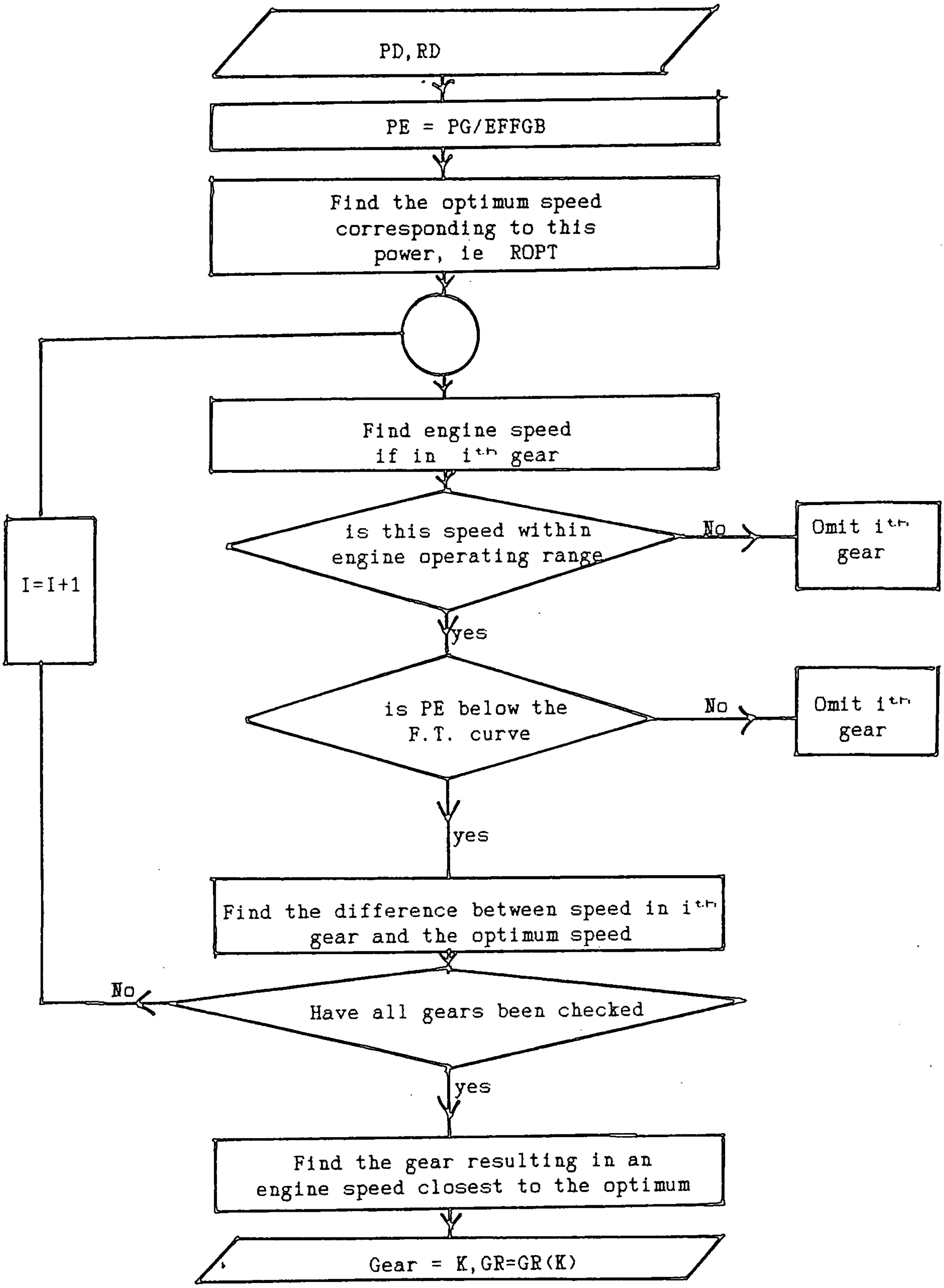
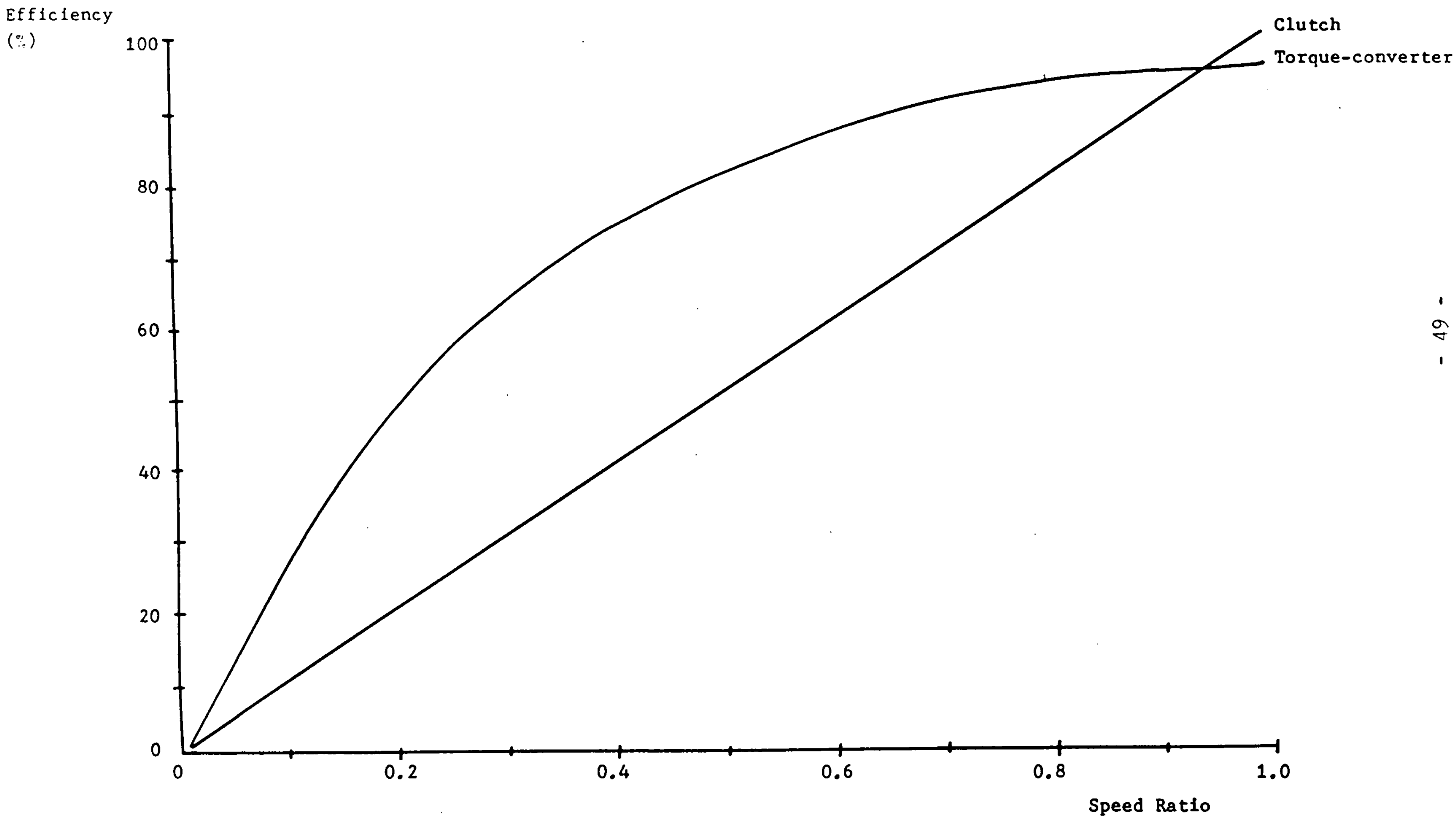


FIG 2.11 - Process employed to find the optimum gear in 'OPTGEAR'

Figure 2.12 - Coupling Efficiency .



I C ENGINE CHARACTERISTICS

I C ENGINE PERFORMANCE LIMIT CURVE

SPEED (RPM)	TORQUE (NM)	BMEP (PSI)	POWER (KW)
500.00	62.41	90.00	3.27
750.00	69.35	100.00	5.45
1000.00	69.35	100.00	7.26
1500.00	76.28	110.00	11.98
2000.00	83.22	120.00	17.43
2500.00	83.22	120.00	21.78
3000.00	90.15	130.00	28.32
3500.00	90.15	130.00	33.04
4000.00	83.22	120.00	34.85
4500.00	83.22	120.00	39.21
5000.00	76.28	110.00	39.93

I C ENGINE CVT OPERATING CURVE

SPEED (RPM)	BMEP (PSI)	POWER (KW)
1000.00	0.00	0.00
1000.00	80.00	5.82
1500.00	100.00	10.91
2000.00	100.00	14.55
3000.00	110.00	24.00
3500.00	120.00	30.55
4000.00	120.00	34.91
4500.00	120.00	39.27
5000.00	110.00	40.00

I C ENGINE PERFORMANCE MAP
X-SPEED (RPM/1000); Y-BMEP(PSI); SFC-PTS/HP-HR

130.00	0.00	0.00	0.00	0.50	0.49	0.49	0.50	0.51	0.50	0.50	0.52
120.00	0.00	0.55	0.54	0.50	0.49	0.49	0.49	0.49	0.50	0.50	0.52
110.00	0.57	0.55	0.54	0.50	0.48	0.48	0.48	0.49	0.50	0.50	0.52
100.00	0.57	0.55	0.54	0.50	0.48	0.48	0.48	0.49	0.50	0.50	0.52
90.00	0.57	0.56	0.54	0.51	0.49	0.49	0.48	0.50	0.51	0.51	0.53
80.00	0.59	0.57	0.55	0.52	0.50	0.51	0.51	0.51	0.52	0.53	0.54
70.00	0.62	0.60	0.57	0.54	0.53	0.53	0.53	0.53	0.54	0.55	0.56
60.00	0.66	0.64	0.61	0.57	0.56	0.56	0.55	0.56	0.57	0.58	0.60
50.00	0.83	0.70	0.66	0.62	0.60	0.60	0.60	0.61	0.62	0.63	0.65
40.00	1.04	0.88	0.75	0.69	0.67	0.66	0.65	0.66	0.66	0.69	0.70
30.00	1.30	1.08	0.85	0.82	0.78	0.77	0.78	0.78	0.80	0.81	0.84
20.00	1.55	1.38	1.20	1.02	1.00	0.98	1.00	1.00	1.00	1.00	1.00
10.00	1.83	1.68	1.54	1.36	1.34	1.30	1.30	1.30	1.30	1.30	1.30
	0.50	0.75	1.00	1.50	2.00	2.50	3.00	3.50	4.00	4.50	5.00

FIG 2.13 - A typical BMEP/speed 40kw engine map.

X-SPEED (RPM/1000); Y-BMEP(PSI); SFC-PTS/HP-HR

130.00	0.00	0.00	0.00	0.50	0.49	0.49	0.50	0.51	0.50	0.50	0.52
120.00	0.00	0.55	0.54	0.50	0.49	0.49	0.49	0.49	0.50	0.50	0.52
110.00	0.57	0.55	0.54	0.50	0.48	0.48	0.48	0.49	0.50	0.50	0.52
100.00	0.57	0.55	0.54	0.50	0.48	0.48	0.48	0.49	0.50	0.50	0.52
90.00	0.57	0.56	0.54	0.51	0.49	0.49	0.48	0.50	0.51	0.51	0.53
80.00	0.59	0.57	0.55	0.52	0.50	0.51	0.51	0.51	0.52	0.53	0.54
70.00	0.62	0.60	0.57	0.54	0.53	0.53	0.53	0.53	0.54	0.55	0.56
60.00	0.66	0.64	0.61	0.57	0.56	0.56	0.55	0.56	0.57	0.58	0.60
50.00	0.83	0.70	0.66	0.62	0.60	0.60	0.60	0.61	0.62	0.63	0.65
40.00	1.04	0.88	0.75	0.69	0.67	0.66	0.65	0.66	0.66	0.69	0.70
30.00	1.30	1.08	0.85	0.82	0.78	0.77	0.78	0.78	0.80	0.81	0.84
20.00	1.55	1.38	1.20	1.02	1.00	0.98	1.00	1.00	1.00	1.00	1.00
10.00	1.83	1.68	1.54	1.36	1.34	1.30	1.30	1.30	1.30	1.30	1.30
	0.50	0.75	1.00	1.50	2.00	2.50	3.00	3.50	4.00	4.50	5.00

IC ENGINE FUEL USAGE MAP

X-SPEED (RPM/1000); Y-BMEP(PSI); USAGE-%

130.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
120.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
110.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
100.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
90.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
80.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
70.00	0.0	0.0	0.0	0.0	2.6	1.0	0.0	0.0	0.0	0.0	0.0
60.00	0.0	0.0	0.0	0.5	2.6	0.5	0.0	0.0	0.0	0.0	0.0
50.00	0.0	0.0	0.0	2.0	1.5	1.5	0.0	0.0	0.0	0.0	0.0
40.00	0.0	0.0	1.0	2.0	0.5	0.0	0.0	0.0	0.0	0.0	0.0
30.00	0.0	0.0	1.5	1.0	1.0	0.0	0.0	0.0	0.0	0.0	0.0
20.00	0.0	0.0	2.0	0.0	7.1	7.1	0.0	0.0	0.0	0.0	0.0
10.00	0.0	0.0	0.0	0.0	3.6	11.7	0.0	0.0	0.0	0.0	0.0
	0.50	0.75	1.00	1.50	2.00	2.50	3.00	3.50	4.00	4.50	5.00

PERCENTAGE TIME AT ENGINE IDLE/OFF = 49.0
PERCENTAGE TIME COMP. BRAKING = 13.8

FIG 2.14 - Engine usage map for a small car driven over ECE-15 driving cycle

Signal (IFLAG)	Subroutine Operating Section
0	1st Input Section
1	2nd Input Section
2	1st Dynamic Section
3	2nd Dynamic Section
4	Output Section

TABLE 2.1 The signalling system used by JANUS for operational flexibility.

Condition	Standard Value
Ambient pressure (mm Hg)	760
Ambient temperature (C°)	15
Head wind velocity (mph)	0
Road gradient (%)	0
Time step (secs)	
Urban and cruise cycles	1
WOT acceleration tests	0.5

TABLE 2.2 JANUS standard time step and environmental conditions.

Final drive type	Efficiency (%)
Spur gear	85-95
Bevel gear	85-95
Chain drive	98.5
Toothed belt	95
VEE belt	92
FLAT belt	95

TABLE 2.3 Types of final drive available in 'JANUS' and their typical average cycle efficiency

Gear Shift	Shift velocity (mph)
1st ——— 2nd	10.0
2nd ——— 3rd	21.7
3rd ——— 4th	33.6
4th ——— 5th	44.7
5th ——— 6th	67.1

TABLE 2.4 The standard shift velocities of JANUS

C H A P T E R   T H R E E

THE CASE FOR CONTINUOUSLY VARIABLE TRANSMISSIONS (CVTs)

3.1 The Case For Continuously Variable Transmissions

The brake specific fuel consumption (BSFC) map of a typical gasoline engine is shown in Fig 3.1, where it may be seen that the most efficient operation of the engine is along the locus of its minimum BSFC points represented as the ideal line. This is only achieved in modern vehicles if engine speed is permitted to fluctuate relative to road speed. Optimum operation of an engine along this locus requires a range of road speeds to be available for each potential power demand. This is possible only by use of an efficient and wide range continuously variable transmission (25,26).

The power rating of the engine could also be reduced without a loss in accelerative performance, because the CVT enables the engine to run at peak power continuously rather than fluctuating with road speed. The smaller engine increases the engine load factor in urban driving, improving engine efficiency. Electric vehicles also benefit from a CVT to minimise start-up current and provide optimum power economy and regenerative braking. Fly-wheel/hybrid vehicles also require a CVT as the fly-wheel must be considered a fixed speed range power source for normal vehicle operation (27,28). Gas turbine automotive engines have an even greater need for a CVT. The single-shaft gas turbine requires a wide ratio CVT in order to prevent engine stall (29). The twin-shaft gas turbine also requires a CVT if part load fuel economy is to be improved to present automotive standards.

In addition to considerations of matching the existing characteristics of engines to vehicle requirements, there is the possibility of modifying engine characteristics in order to take full advantage of the CVT's flexibility. Assuming that the ratio spread of the CVT is broad enough, engine operating speed can be restricted to a more limited range than present and this may allow design improvements in such areas as turbo-charger matching, better carburettor characteristics and inlet and exhaust valve movement timings. Therefore, not only is it possible to use the CVT to operate along the locus of minimum BSFC points, but it may also be possible to rely on operating flexibility of transmissions, rather than that of engines, and thereby improve the efficiency values on the engine operating map itself.

3.2 Design Considerations

An acceptable CVT must be efficient, quiet, capable of adequate life, cost competitive and capable of automotive power. The CVT must also have a wide ratio range. At any given vehicle speed and for any required power level a certain gearing will provide maximum fuel economy for a given engine. In addition at any vehicle speed one transmission ratio will permit maximum acceleration with that engine. Therefore, ideally, a CVT should be able to provide both transmission ratios at any vehicle speed. This requires a wide-ratio range. Calculations (Section 6.1.1) based on the assumption that the CVT vehicle should have optimum operation whilst maintaining the same low speed acceleration and ability to overcome grades as the comparable conventional vehicle show that CVT gear ratio speeds of up to 10:1 are required. It should be noted that this value depends highly on the type of engine and the vehicle in which CVT is to be employed.

3.3 Basic Concepts and Principles of Continuously Variable Transmissions

Many automative CVT designs have been proposed and a large number have been built, either as prototypes or as production versions. The basic operation and characteristics of each are described here.

3.3.1 Hydrostatic Drives

A hydrostatic drive transmits power through the use of high pressure oil, typically at pressures up to 350 bar (30). The drive consists of a swash plate hydraulic pump and motor connected together by a couple of hydraulic lines with other hydraulic components such as check valves, relief valves and reservoir. The pump creates the hydraulic power and the motor converts the hydraulic power to mechanical power. The swash plate is a disc mounted obliquely on a rotating shaft (Fig 3.2) and around this axis the parallel cylinders of the pump are disposed. The rotating swash plate rubs against the slippers on the end of the plungers. Each piston thus functions to create a void which fills with fluid during one-half revolution and discharges this fluid during the remaining half revolution. The amount of fluid displaced per revolution is dependent on the stroke and therefore on the swash plate angle. The ideal speed ratio (based on zero leakage and incompressible fluid) for a hydrostatic unit is evaluated as follows (30).

$$\text{Speed Ratio} = \frac{\text{Output speed}}{\text{Input speed}} = \frac{\text{Pump displacement}}{\text{Motor displacement}}$$

For example, by fixing the motor at the full displacement of the pump, the speed ratio can be varied between -1 and +1 if the pump is fully reversible. Additional versatility could be gained in employing a motor with variable displacement. The function of pump and motor can also be reversed such that power is transmitted in the reverse direction.

Due to flow losses, good efficiency ratings are only achievable in a relatively narrow conversion range. Although hydrostatic CVTs are very adaptable and may have other advantages such as the possibility of eliminating the final drive differential by employment of one hydraulic motor for each drive wheel, major disadvantages such as size, weight, and above all their relatively poor efficiency have prevented their application in the range of vehicles considered here (29,30,31).

3.3.2 Hydrokinetic

The torque convertor is employed in nearly all modern automatic transmissions, poor efficiency characteristics have limited the useful ratio range to be less than adequate for a CVT (see Fig 2.12).

3.3.3 Electric CVT

The basic electric CVT consists of an electric motor, generator combination (Fig 3.2). The generator converts mechanical power to electrical power which is fed to the motor, which converts it back to mechanical power. Torque, voltage variations would be relatively easy to obtain by varying the generator field voltage and regenerative braking is easily achievable by reversing the roles of the two electric machines.

The main drawback of this simple electric transmission is its low efficiency. Even though electric machines can be very efficient at their best operating points, the efficiency will drop to unacceptable levels at speeds and loads far removed from these points. There is the further problem that size and weight of a pair of electric machines capable of automotive power will tend to be excessive.

3.3.4 Kinematic Linkages and Overrunning Clutches

A simple example of this type of CVT is the Zero-Max which is commercially available for light power industrial applications (31). Fig 3.4 illustrates the operating principle of a single linkage (four to eight of such linkages are normally used). The rotating shaft on the right has an eccentric for each linkage. This causes the power linkage to oscillate. By employment of overrunning clutches, a one directional and nearly uniform rotation of output shaft is achieved. The speed ratio is changed by moving point A of the control link.

Based on this basic principle, Power Matic Corp. are developing a more complex transmission for automotive application (17). The transmission employs a variable throw crank coupled to a number of hydraulic ratchets (type of overrunning clutch). The drive is proposed to provide an overall ratio spread of 12.5:1 at full load and an overall efficiency of about 92-96%. Insufficient experimental data is made available, however, to allow for an adequate evaluation.

3.3.5 CVT Employing Non-Circular Gears

An automotive CVT which employs non-circular gears is under development in Royal Military College of Canada (32). The transmission is proposed to operate as follows. The non-circular gears are designed to give velocity ratio profiles similar to those shown in Fig 3.5, where WA and WB are the ratios of the speeds of non-circular driven gears A and B relative to the speed of the non-circular driver gears, which rotate at constant speed. Gears A and B are connected to an asymmetrical differential, where their driver gears are mounted on concentric shafts with a controller that can alter their phase relationship. The whole unit (Fig 3.6) is then named a "Function Generator". The velocity ratio of output of this unit C is dependent on the phase relationship between the driver gears and whether B or A is leading, which are set by the controller (Fig 3.7). In either case, the phase shifts are constrained to that for which the constant speed portion is 100° long. In this manner, the output of the differential is said to be infinitely variable over the range $0 \leq WC \leq 2$. To achieve continuity, four such generators are spaced at 90° internal about a common set of driver gears and by using a "programmed clutching system" each generator is made a part of the power path only when a constant speed portion of the velocity profile is being generated.

The design is proposed to provide a kinematic ratio range of ∞:1 to 1:1 and an efficiency of 75-95% (theoretical estimates). There is, as yet, no experimental efficiency data available for an adequate edvaluation of this design.

3.3.6 Traction Drives

Traction drive is a term applied to CVT configurations which transmit power through the continuous action of lubricated rolling contacts. By changing the geometric relationship of the traction parts, the speed ratio is changed in a continuous manner. Traction drives have been available for many years (33), but were generally considered incapable of automotive surface polish and wear, and therefore a limited life and power capability. Recent developments in traction fluid properties (34) have led to various new and improved designs for automotive application, the basic concepts of which are discussed here.

The Cone and Ring Design: The basic concept is shown in Fig 3.8, where a ring is moved back and forth to vary the speed ratio. A more sophisticated transmission employing this basic principle and capable of automotive power, is being developed by Vadetec Corp. (34). Although theoretical estimates of its efficiency (80-90%) are promising and a ratio range of 0 to 1:1 is proposed, insufficient experimental data is made available for adequate evaluation of this design.

Toroidal Drives: The basic forms of toroidal drives are shown in Fig 3.9. The drive consists of a number of rollers which roll against a facing pair of toroidally surfaced discs. The speed ratio is changed by altering the roller angle so as to vary the distance of the rolling contacts from the disc centre lines and thus the relative rotational velocity of the dics. The discs are thrust together by an axial force (Fig 3.9) to provide the contact normal force needed to support the contact tangential force which transmits the torque from one disc via the rollers to the other disc. The resultant ratio spread of the basic configurations are 12:1 for the off-centre arrangement and 9:1 for on-centre design. More efficient CVTs using this basic concept and capable of automotive power have been developed which are discussed in Section 3.6.

3.3.7 Rubber V-Belt Drives

This type of CVT employs a V-belt to transmit power between two pulleys whose diameter can be varied (Fig 3.10). When both pulleys are made variable an overall ratio range of about 4:1 is typically achieved. The belt-drive was initially employed in very small vehicles such as DAF and Snowmobiles (31). Recent developments in belt design (35,36) have led to its employment in vehicles capable of over 50 Kw power, such as the Volvo 343, with transmission efficiencies in the range 80-88% (37).

3.3.8 Chain Belt Drives

A recent development of V-belt drives is the transmatic drive (38). Here the conical pulleys are joined by a metal belt transmitting torque in thrust instead of tension. The belt is composed of a number of wedge-shaped steel blocks which slide freely on a bunch of superimposed thin gauge steel bands. This arrangement guides the steel blocks in line into the pulley groove and leads them back on the slack side. Power is transmitted by the steel blocks contacting each other at their rocking edge. The belt is tensioned by hydraulic pressure forcing the sliding blocks against the fixed pulleys.

The transmission itself consists also of a centrifugal clutch and a reduction gear between the engine and the variable unit in order to limit the input speed to the variable unit. The whole transmission is enclosed in an aluminium casing and runs in oil. It is suggested that the transmission is no bigger than a standard manual gear box, weighs about 10% less than a conventional automatic transmission and that the basic unit could be used with power units between 1.0 and 2.3 litre in capacity, with the reduction gear and casing to match (38). The transmatic efficiency varies depending on the torque input and gear ratio and approaches efficiencies close to that of the manual gearbox. In addition the parallel but off-set input to ouput shaft is a useful feature for the front wheel drive cars. Due to these favourable features a number of motor manufacturers are currently considering its employment and one, FIAT, has announced its future employment in their UNO range (38).

3.4 The Power Split Principle

Certain continuously variable units (CVUs) offer a wide ratio spread but poor efficiency ratings, such as Hydrostatic Drives. Development of these transmissions has therefore been diverted to the use of Power Split Principle. Fig 3.11 shows the basic concept where only a part of the power is sent through the CVU, the remainder of power going through the more efficient mechanical path, the two components are then added in a differential at the output of the transmission. By sending a lower amount of power through the CVU rather than all the input power if it were used alone, the overall transmission efficiency is improved.

A schematic of a power split CVT which employs a planetary differential is shown in Fig 3.12. Note the gear ratios between the input shaft and the CVU input and also between the CVU output and the output shaft. In order to reduce the fraction of power going through the CVU, its torque needs to be reduced. But to satisfy the torque balance of the planetary differential, a relatively large gear ratio (W3/W4) is needed. As this gear ratio increases, the effect of a given change in CVU output speed on transmission output speed decreases, resulting in a lower ratio range.

This reduction could be compensated by having a gearbox in series with the CVT. The gearbox-CVT then becomes a wide ratio continuously variable system with the CVT filling in the ratio gaps between gears. This however reduces the overall efficiency and unless the efficiency increases gained by Power-Split arrangements is much greater, the added complexity is of no avail.

3.4.1 Power Recirculation Mode in Conventional Power Split Transmissions

Some power split transmissions (Section 3.6.1) operate in a power recirculative mode during a part of their operation usually vehicle start up. This notion is illustrated in Fig 3.13 which shows that a fraction of the power at the output is fed back via the CVU to the input shaft, this causes the power going through the mechanical path to be greater than input power. Recirculative power split mode is usually employed only at the low power operations to limit the amount of power recirculation which reduces transmission efficiency.

3.5 Regenerative (Inverse) Power Split Designs

Here, through the use of differential gearing, the overall ratio range is expanded at the expense of efficiency. The application of this prinicple is, therefore, attractive with CVUs which offer good efficiency characteristics but a limited ratio range such as toroidal traction drives. The concept is shown in Fig 3.14, where the differential sends a part of the power from the CVU back to the input shaft through the mechanical path. This causes the power going through the CVU to be greater than if it were used alone. The higher than input power going through the CVU produces a power loss greater than if the CVU were used alone. The reduction in efficiency may be quite small, however, with a good design and a small amount of power recirculated.

Fig 3.15 shows a schematic of a regenerative power split transmission. The arrangement is identical to that of Fig 3.12 but for an extra pair of gears between the mechanical path shaft and the input shaft, the purpose of which is to reverse the direction of rotation of the mechanical path shaft. The torque balance requirements of the planetary differential, however, constrain the torque on that shaft to have the same sign as that of Fig 3.12, resulting in power being taken away from the differential.

An example of the use of this principle is in the 'LOW' mode of operation of B L Perbury (Section 3.6.2) where the overall ratio range of the basic Perbury unit is increased to allow for vehicle start up.

3.6 CVTs Employing Power Split Arrangements

3.6.1 Hydromechanical Drives

As described in Section 3.3.1, hydrostatic drives provide wide ratio spread but have poor efficiency characteristics, due to this factor, research and development of this type of transmissions have been diverted to the use of power splitting arrangements.

Hydromechanical transmissions which use planetary gearing to achieve power splitting during a part or all of their operations have been developed by Sunstrand (39) and Orshanskey (40) Corps, the most efficient of which is the Orshanskey two-mode configuration shown schematically in Fig 3.16. The 'LOW' mode of operation is a recirculative power split. The 'HIGH' mode of operation is a power split designed to improve efficiency. The mode of operation is determined by a clutching arrangement at the output of the transmission. The transition between the two modes is only possible at a synchronous ratio. This is evident from Fig 3.17 which shows the efficiency characteristics of this transmission over a range of input power and speed. Due to weaknesses in such areas as weight, size, cost and high noise level, Orshanskey has commenced the development of a three-range hydromechanical transmission, the lowest mode being purely hydrostatic and the upper two modes of the split torque type. It is proposed that this design would result in a more efficient CVT (Fig 3.18) with a size and weight comparable to conventional automatic transmissions (41).

3.6.2 B. L. Purbury Toroidal Transmission

The basic CVU employed by this transmission is a double sided on-centre toroidal drive (Fig 3.19). The double sided design is selected to avoid the application of an axial load to thrust the discs together. In this design, the end discs are mounted on a common shaft which, in tension, reacts the applied load. The ratio range of the basic double sided unit is about 5:1. In order to increase the ratio spread and utilize the best efficiency over the whole operating range, a two mode synchronous arrangement is proposed (42). The 'LOW' mode is a regenerative power split configuration (Fig 3.20) which provides from about 5:1 reduction in reverse through zero output (neutral) to about 2.5:1 reduction. The 'HIGH' mode (with no power splitting) provides for speeds from about 2.5:1 reduction to 0.5:1 overdrive (Fig 3.21).

The efficiency characteristics of this transmission, which have been confirmed by tests, are shown in Fig 3.22. The effect of the different power flows on efficiency is evident. At low power levels, the 'LOW' mode of operation provides for efficiencies higher than that which would be achieved if the basic unit was used alone. This is due to the poor efficiency of the basic unit at these power levels, where recirculation of power increases the load factor on the CVU and therefore leads to improvements in efficiency.

In summary, the CVT combines a wide ratio spread with promising efficiency characteristics.

3.6.3 Cone Roller Toroidal Drive

The basic CVU is an off-centre toroidal drive with cone rollers (as opposed to the Perbury design which uses an on-centre toroidal drive with disc rollers). The roller is cone shaped in order to reduce spin losses and therefore surface wear (Fig 3.23). The cone angles is set so that spin goes to zero (tangent lines from traction contacts intersects at the drive centre line) at or near the ratio extremes where the contact forces are most severe (43). This arrangement has the effect of making the drive efficiency less sensitive to the position of rollers and therefore to variations in speed ratio. This can be readily seen by comparing the efficiency characteristics of the basic CRTD (Fig 3.24) with that of the pure Purbury drive ('HIGH' mode of operation in Fig 3.22). The basic CRTD has been developed by Exclematic Inc of USA (43), where a number of designs embodying the basic unit have been built and tested. The latest is a regenerative power split, a schematic of which is shown in Fig 3.25. The variable speed output from the CRTD unit rotates the ring gear of the planetary differential. A fraction of power is fed back to the input of the transmission via the sungear which rotates in the opposite direction to the ring gear. The planet carrier is the transmission output.

The regenerative power split design expands the ratio spread of the basic drive (from 6.48 reduction to 0.54 in overdrive) to around 100:1 to allow for vehicle start-up (43). Theoretically, the output speed can go to zero, however, an automatic clutch is engaged and disengaged at 0.1-0.2 mph to prevent excessive torque in the transmission. The overall efficiency characteristics of both the continuously variable unit and the complete regenerative transmission (these are based on computer predictions and are corrolated to test data) are shown in Fig 3.24. The CVT efficiency varies depending on the power level and gear ratio and approaches efficiencies close to that of the manual gear box.

3.7 CVT Control

The CVT needs to be controlled so that the output torque demanded is obtained with the engine operating at the most efficient condition that will produce that output torque. This implies that a specific CVT ratio is required for a given combination of desired output speed and torque. The control at the CVT is therefore a ratio control, but a control system like a microprocessor with transducers is also required to choose the proper CVT ratio continuously. Ratio control systems have already been developed (39,40) for different types of CVTs.

The engine operating schedule (ideal line) can be described by two of a number of parameters such as speed, torque, inlet manifold pressure or carburettor throttle angle. Engine speed and throttle angle are usually selected as being well behaved and sensitive functions of the ideal operating line. Additional inputs to the microprocesser are accelerator pedal position and CVT output speed.

Driving of such a car can be described as follows. The control system senses driver demand by means of accelerator pedal position which is translated into demand power through multiplication by road speed. The controller will then set the engine throttle position and CVT ratio based on the CVT output speed, the demand power and stored data.

3.8 CVTs Selected For Simulation

Table 3.1 summarises the most important characteristics of the CVTs reviewed. The CVTs selected for simulation based on their state of development and availability of experimental efficiency data are:

(a) Transmatic (chain belt drive)

(b) Orshanskey HMT

(c) B L Perbury drive

(d) CRTD

The hydrostatic and electric CVTs were not selected due to their major drawbacks described earlier. The non-circular gear CVT, the variable throw crank drive and the nutating traction drive were not selected due to lack of extensive experimental efficiency data. Finally the rubber V-belt drive was omitted as the more efficient transmatic was selected.

3.9 'CVT' (Continuously Variable Transmission) Simulation

This subroutine simulates the CVT and its controller. The initial inputs to the CVT simulation are as follows.

1. Type of CVT required

2. CVT deceleration strategy.

In addition to a constant efficiency CVT, there are four types of CVTs available.

There are two strategies for CVT deceleration:

1. Lock up with fixed gear ratio,
2. Deceleration operating locus,

the latter being a decelerative operating curve supplied to the controller. In addition to this curve the engine normal operating curve (ie locus of minimum specific fuel consumption points) is also supplied to this subroutine by the 'ICENG' subroutine.

The principal dynamic inputs to the CVT simulation are the torque and speed requirements of the differential (ie CVT output conditions).

In order to evaluate the CVT input conditions the CVT gear ratio and efficiency are required and yet for the evaluation of the latter the input conditions are needed (Fig 3.26). Following careful consideration in this work to obtain the best way of overcoming this simulative problem, the following methods were selected.

3.9.1 For Urban and Cruise Driving Cycles

The assumption made here is that the engine speed RE which is engine schedule speed for power produced by engine POWRE (Figs 3.26 & 3.27) equals to RCV which is the engine schedule speed corresponding to CVT input power POWRCV. In other words, as far as the CVT simulation is concerned no power loss due to inertia occurs in the engine. Based on this assumption the CVT input power is evaluated (the evaluation of efficiency is discussed later in Section 3.10).

$$\text{POWRCV} = \text{POWRD/EFFCVT} \qquad (3.1)$$

The CVT input speed (RE) is assumed to be equal to the engine schedule speed for this power level (RVC). These CVT input conditions (ie POWRCV, RVC) are then supplied to 'ICENG'.

The assumption of no engine inertia loss is obviously correct as far as cruising conditions are concerned, but may result in two types of simulation errors during accelerative conditions. These are:

1. In 'ICENG', the engine power produced is evaluated by addition of inertia power loss to CVT input power (Fig 3.27) and fuel consumption is evaluated for this power (POWRE) and RCV, instead of actual engine schedule speed RE and therefore a distance away from the optimum fuel consumption point.
2. As it will be discussed later (Section 3.10) the efficiency of some CVTs depends on the gear ratio demanded and therefore an error in CVT input speed means an error in gear ratio and subsequently in CVT efficiency.

The errors described are ~~negligable~~ negligible, however, when considering the relatively low acceleration rates associated with urban driving cycles. Furthermore, engine of a CVT vehicle will be operating at a constant speed at those lower power levels (see the engine map shown in Fig 3.1). In which case, RCV and RE will both be equal to this constant speed and the errors discussed non-existent.

During high acceleration rates associated with wide open throttle (WOT) driving, the engine inertia torque loss is considerable and the assumption made here would result in large simulation errors. Therefore a different approach for this operation is required which is discussed in the next section.

3.9.2 WOT Acceleration Operation

During WOT acceleration the CVT provides its lowest possible gearing till maximum engine power is attained after which by continuously varying CVT gear ratio the engine is kept at the speed corresponding to maximum power. Some types of CVT, however, by using regenerative power split principles, can provide gear ratios of up to 80:1 for vehicle start up (Section 3.6.3). If such gear ratios are used at higher power levels the power recirculated in the CVT would be greater than it can handle. Therefore a torque limit is usually set in CVT control (42) to restrict such high gear ratios to vehicle start up and low power levels. To simulate this effect here, during vehicle start up the CVT is allowed to provide these high gear ratios until engine idling speed is reached after which the gear ratio is limited to the highest gear ratio employable at all power levels namely 5:1 for B L Perbury and 4:1 for the HMT. Therefore in WOT acceleration simulation the CVT can use these high regenerative gear ratios till engine idling speed is achieved. After this, however, the CVT is locked into its lowest full load gear until maximum engine operating speed is obtained, after which the gear ratio is reduced continuously to keep the engine at this speed. The input conditions to the CVT are determined, therefore, as follows.

(a) Before engine idling speed is achieved

$$RE = Ridle \tag{3.2}$$

and

$$CVTGR = RE/RD \tag{3.3}$$

(b) After engine idling speed is attained till maximum engine speed is achieved

$$CVTGR = CVGRH \tag{3.4}$$

$$RE = CVTGR * RD \tag{3.5}$$

where CVGRH = the highest gear ratio possible at all power levels.

(c) Once maximum engine is attained and thereafter

$$RE = REMAX \qquad (3.6)$$

$$\text{and } CVTGR = RE/RD \qquad (3.3)$$

Once CVT input speed and gear ratio is determined its efficiency could be determined (Section 3.10) and so the CVT input power is evaluated by:

$$POWRCV = POWRD/EFFCVT \qquad (3.1)$$

During WOT acceleration the vehicle employing the CVT follows the engine optimum operating schedule (Fig 3.28) and not the full throttle line followed by conventional vehicles. This rule is followed in the WOT simulation of vehicle with CVT and is the only difference between this simulation and the WOT simulation of conventional vehicle shown in Fig 2.4.

3.10 CVT Efficiency Simulation

The objective here was to store the efficiency data in a flexible format so that it could be utilized for a range of vehicles.

3.10.1 B L Perbury Drive

At first the efficiency data of Fig 3.22 was stored in a matrix corresponding to percentage of maximum power against percentage of vehicle maximum road speed. This method did not, however, consider the variation in constant synchronous and maximum overdrive ratio lines possible by variation of final drive ratio or engine operating schedule of different vehicles (Figs 3.29 and 3.30 respectively). It was decided, therefore, that this method does not provide the required flexibility for simulation of a range of different engines and vehicles.

Following careful study as to the best method of storing the efficiency data it was concluded that to give the best accuracy and flexibility the data should be stored in a matrix corresponding to percentage of power to road wheels against CVT gear ratio. This required translation of axes of Fig 3.22 into CVT input and output speed which incorporated a number of problems. For transformation of X-axis into CVT output speed the final drive ratio and wheel radius of the test vehicle used in reference 42 were required, which were not available. In the case of the Y-axis the engine operating schedule was also not available and furthermore the axis refers to CVT output power and not input power which is represented by the operating schedule. The problem was solved as follows. On the constant gear ratio lines in Fig 3.22 the bottom sections (below about 9% power) refer to a constant CVT input speed. As this is the speed for low power levels it was assumed to be at 1000 rpm. As the synchronous ratio was known (2.5:1) the CVT output speed was found (400 rpm) and by correlating this speed to its respective linear value (Fig 3.31), the X-axis of Fig 3.22 can be translated into rotational speed. This results in a maximum overdrive ratio of about 0.45:1. Since CVT output speed and input speed (below 9% power level) are known and as the constant gear ratio lines follow the pattern of the engine operating curve and are therefore parallel, other constant gear ratio lines could be drawn as shown in Fig 3.31. The validity of the assumptions made is proven since the resulting 1:1 constant gear ratio line is the locus of maximum efficiency loops. The spin losses approach their minimum value at this ratio due to the geometry of the basic Perbury unit (33).

In order to achieve the flexibility required to enable variations of the synchronous ratio, further modifications were necessary. This was obtained by extending both the 'low' and 'high' regime efficiency maps beyond the range shown in Fig 3.22. The resultant maps for each regime are shown in Fig 3.32. Note that the synchronous ratio is limited to 3:1 which is an inherent limitation in the Perbury unit.

Also note that the maximum reduction ratio is 5:1. To obtain the CVT efficiency at greater reduction ratios (encountered, for example, when moving away from 'geared neutral'), a linear interpolation between the efficiency of the maximum reduction ratio and an assumed efficiency of zero at neutral takes place. This method is used for: all the CVTs simulated.

The process employed to find the efficiency of Perbury at any condition of load and speed is shown in Fig 3.33, where EFFMAX is the maximum efficiency of the Perbury at maximum power which is about 90% (see Fig 3.22) and CVTSCR is the synchronous ratio depending on which 'LOW' or 'HIGH' regime maps are searched for efficiency.

3.10.2 H M T

The efficiency data of this transmission shown in Fig 3.17 was translated into the desired format (ie normalised power versus gear ratio) by dividing by the rated power at 100 hp (44). The efficiency data at this power level, however, are not available and are assumed to follow the same pattern as the 60 hp curve but a lower efficiency, the difference is assumed to be twice the difference between 60 and 40 hp levels. For evaluation of HMT efficiency at higher gear ratios than that shown the efficiency curves of Fig 3.17 were extended in a fashion similar to Fig 3.18. The resultant efficiency map is shown in Fig 3.34 which corresponds to normalised power versus gear ratio. For evaluation of CVT efficiency at any condition of load and speed from this map, the same iterative process as employed for Perbury (Fig 3.33) is used.

3.10.3 C. R. T. D.

The available data was in the format shown in Fig 3.24 and due to unavailability of sufficient information the data could not be translated to a format corresponding to normalised power against gear ratio. However, unlike the Perbury, the efficiency data for each power level is continuous and relatively even over the range of vehicle velocities. It was decided, therefore, that the required flexibility could be achieved by storing the data in a matrix corresponding to normalised power and road speed. This was implemented by dividing the power levels and vehicle velocities by their maximum values (102 hp and 120 mph, Fig 3.24). The resultant normalised efficiency map is shown in Fig 3.35. In order to find the CVT efficiency at any condition of load and speed the rated power and road speed of the simulated vehicle are required. The former is supplied by 'ICENG' (Section 2.8) and the maximum speed of the vehicle is estimated as follows. Assuming engine power at maximum speed is the maximum power

$$PEMAX = PWMAX/EFFDT \qquad (3.8)$$

and assuming a drive train efficiency of 90% and zero accelerative or gradient work and from EQ 2.4 to 2.12

$$PEMAX = (CR * W + 0.5 * Da * A * VMAX^2) * VMAX/0.9 \qquad (3.9)$$

Also assuming standard environmental conditions (Table 2.1) the following relationship between maximum power and speed is arrived at

$$PEMAX = (CR * W + 0.613 * CD * A * VMAX^2) * VMAX/0.9 \qquad (3.10)$$

The maximum vehicle velocity is found by employing EQ 3.10 in the iterative process shown in Fig 3.36. The initial estimate of maximum velocity is made by assuming that the rolling resistance term in equation 3.10 is negligible and therefore

$$VMAX\ (m/s) = [(0.9 * PEMAX)/(0.613 * CD * A)]^{1/3} \quad (3.11)$$

Once the maximum vehicle speed and power are determined the CVT efficiency is found by the iterative process shown in Fig 3.37.

3.10.4 Transmatic (Chain Belt Drive)

The efficiency data available for this CVT was in the form of the overall drive train efficiency which varied according to the CVT input torque (Nm) versus the overall ratio of the drive train (ie including the final drive ratio). This data was transformed into a format corresponding to normalised torque against CVT gear ratio. The latter were found by assuming equal possible displacement for both pulleys.

In order to determine efficiency at any particular condition, the iterative process of Fig 3.38 is used where the torque is normalised with respect to the maximum torque available on the engine operating schedule (ie TCVMAX) and the continuously variable units gear ratio is found by dividing engine speed by Transmatic's initial reducation gear (VBGR) and the final drive speed. The efficiency found in this fashion corresponds to the overall efficiency of the drive train, which is transformed into CVT efficiency in division by final drive efficiency (EFFDRV). This process is repeated until the true efficiency of the drive is evaluated.

3.10.5 Constant Efficiency CVT

Here the power input to the CVT is found simply in dividing the CVT power output by the user specified constant efficiency.

3.11 Subsequent Changes in Other Subroutines

(a) 'TRANS'

Some changes had to be made, in order to enable this subroutine to overrun its normal operation (Section 2.6) when a CVT is employed.

(b) 'ICENG'

Along with each available engine map in 'ICENG' its corresponding optimum operating schedule was stored. Provisions were also made to enable change in the operating schedule by the user if a vehicle with CVT is not able to meet the optimum operating curve.

Figure 3.1 - Engine map for a 1.85 litre Four Cylinder s.i. Engine.

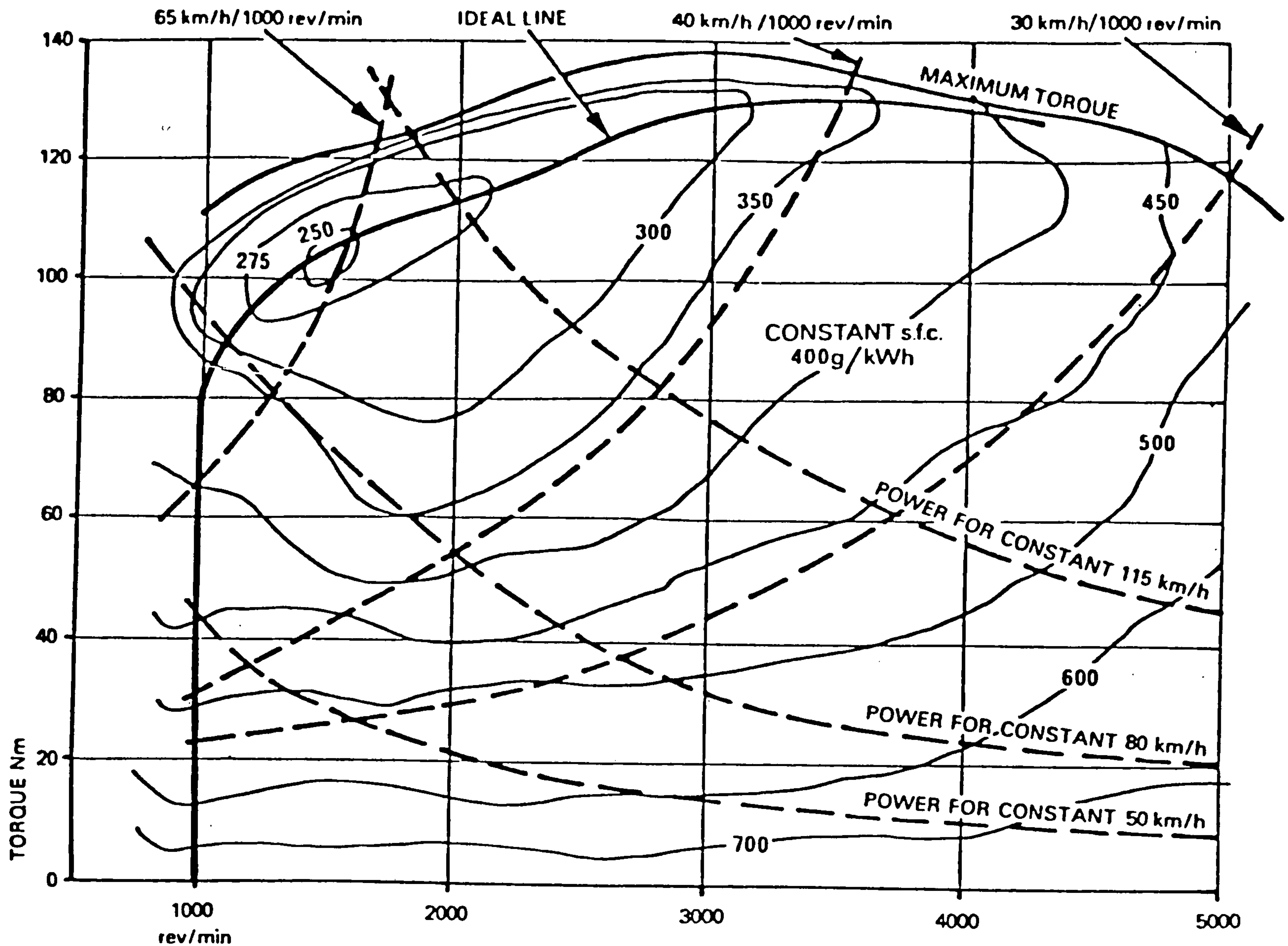
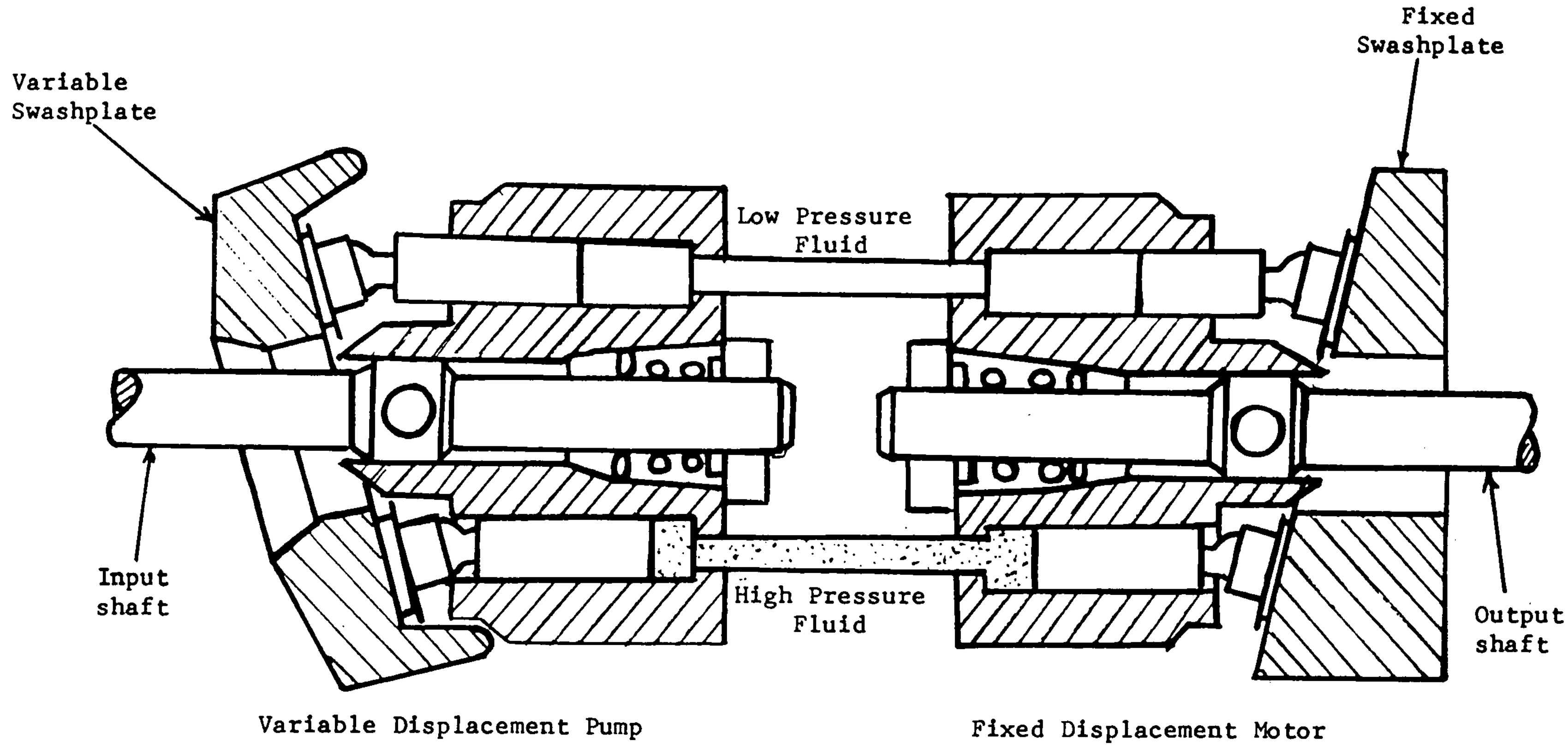


Figure 3.2 - A typical Hydrostatic Transmission .



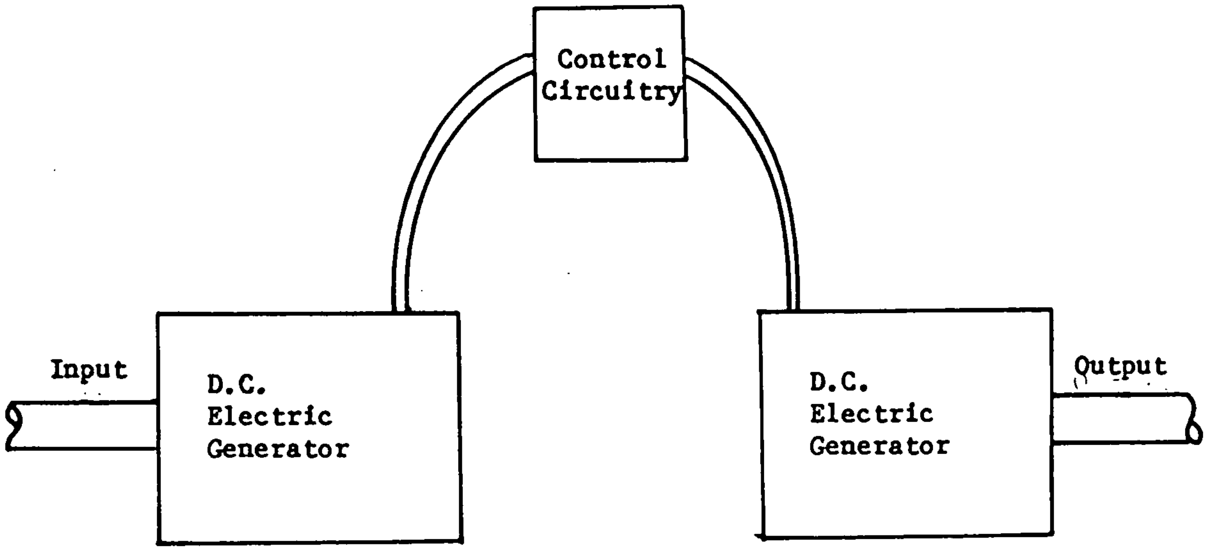


Figure 3.3 - Electric C.V.T.

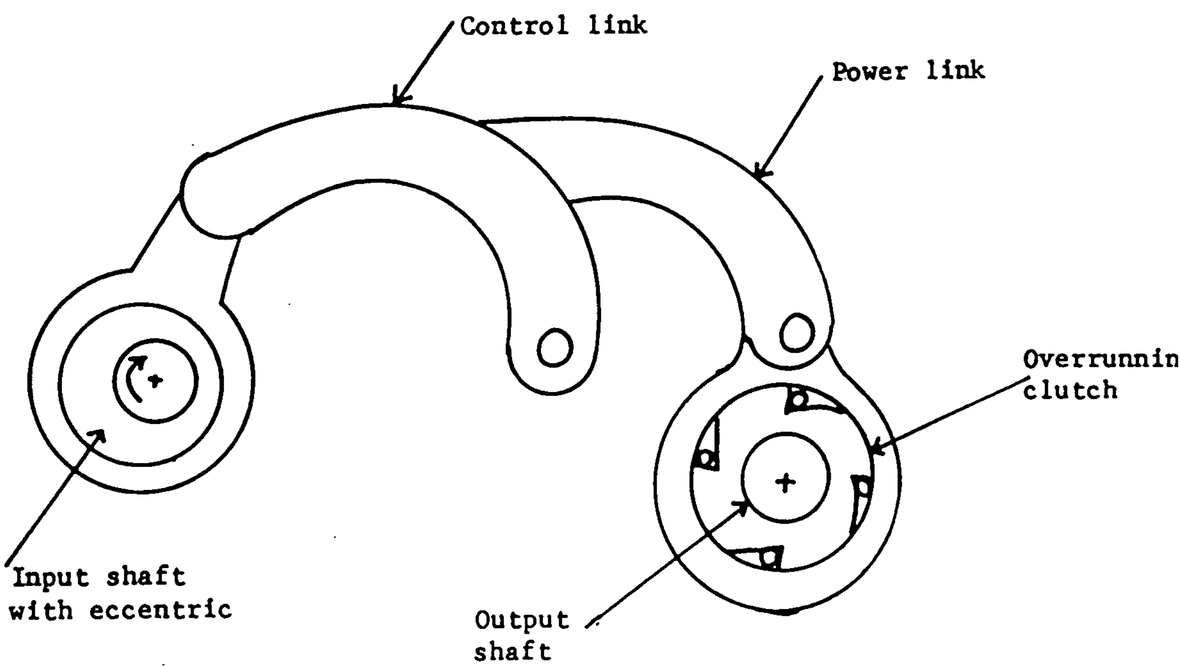


Figure 3.4 - One linkage of the Zero-Max Industrial C.V.T.

Figure 3.5 - Velocity profiles of non-circular gears .
(Source of data : Ref.32)

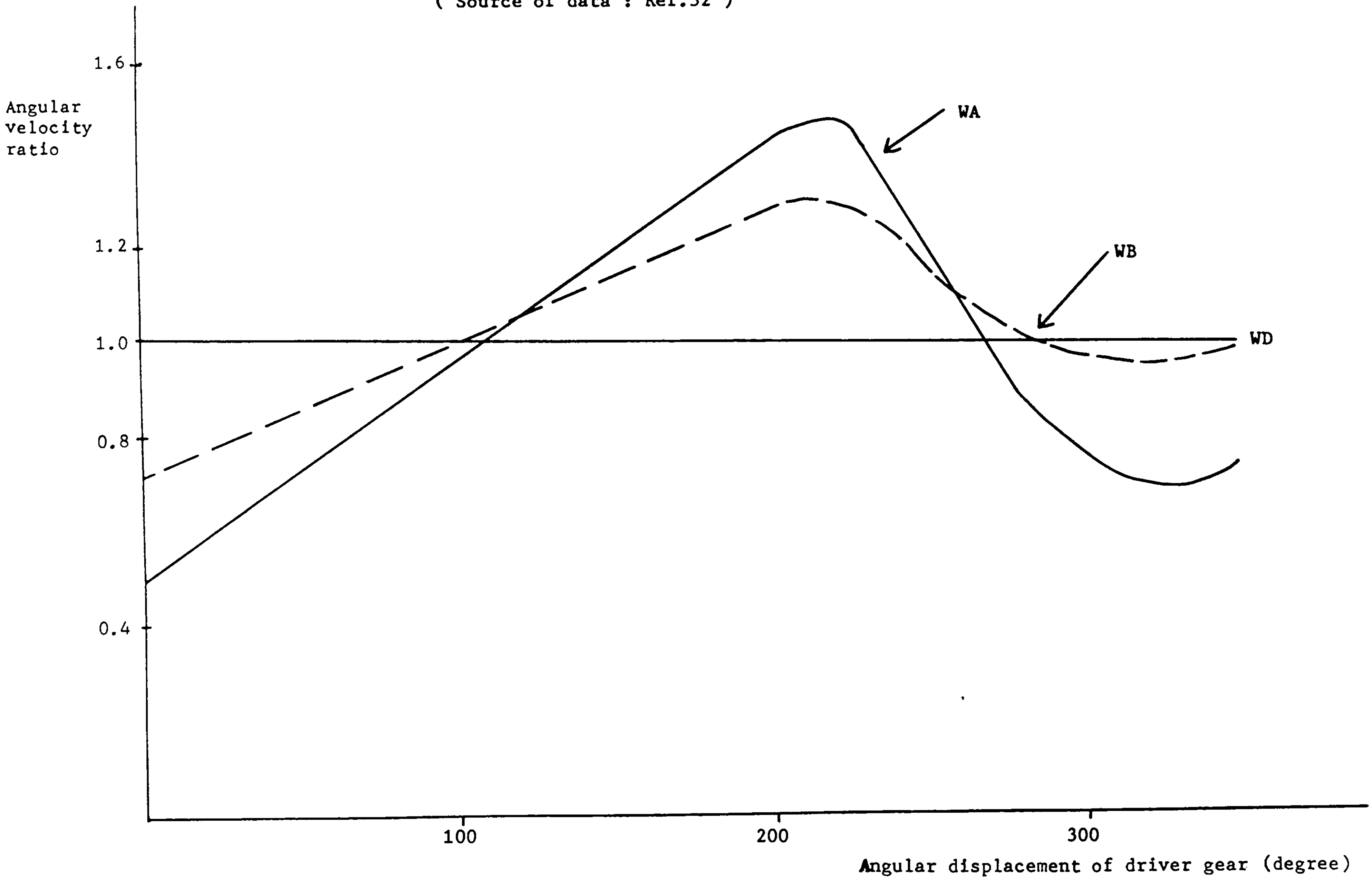


Figure 3.6 - " Function Generator " of the C.V.T. employing non-circular gears .
(Source : Ref.32)

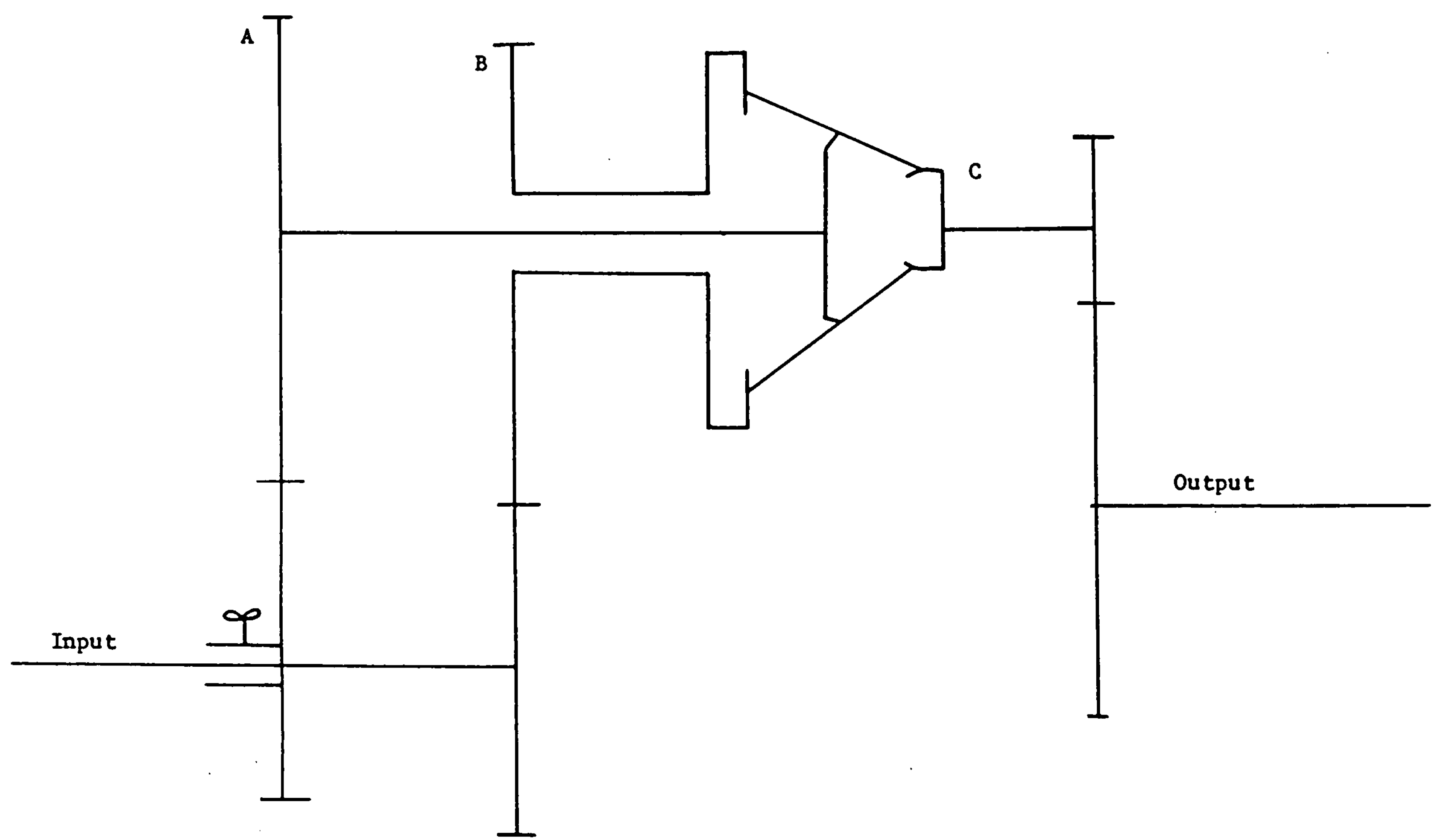
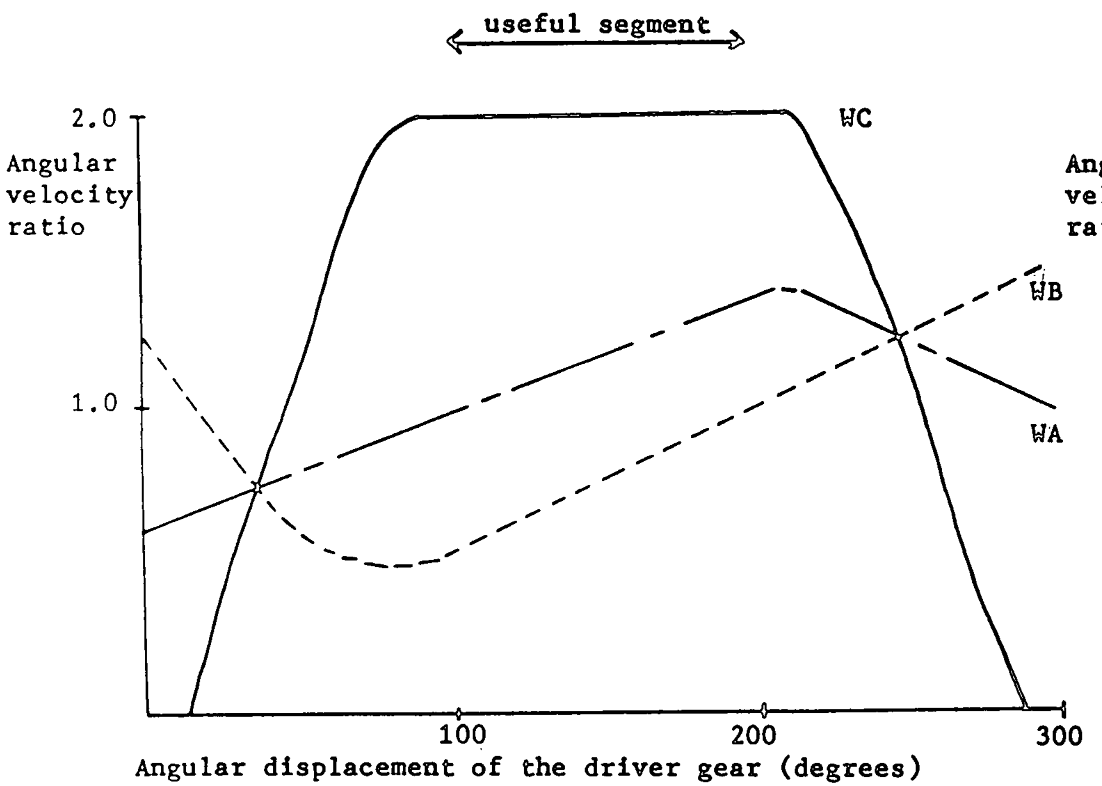
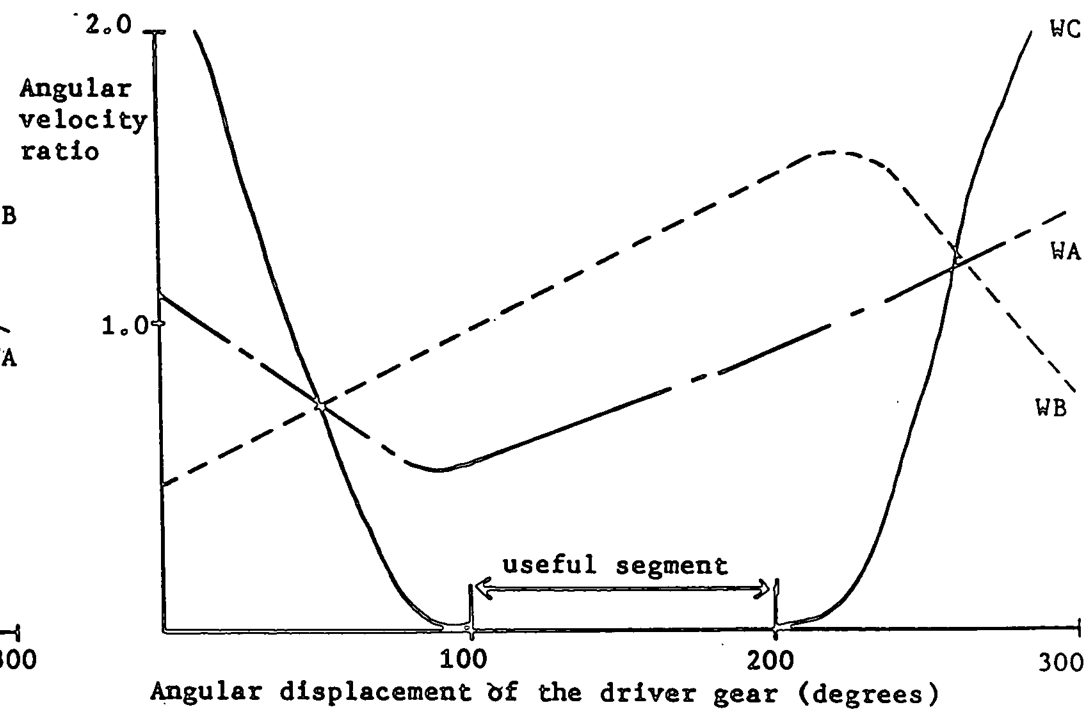


Figure 3.7 - Function generator output (wc)
(Source of data : Ref.32)



Gear B leading



Gear A leading

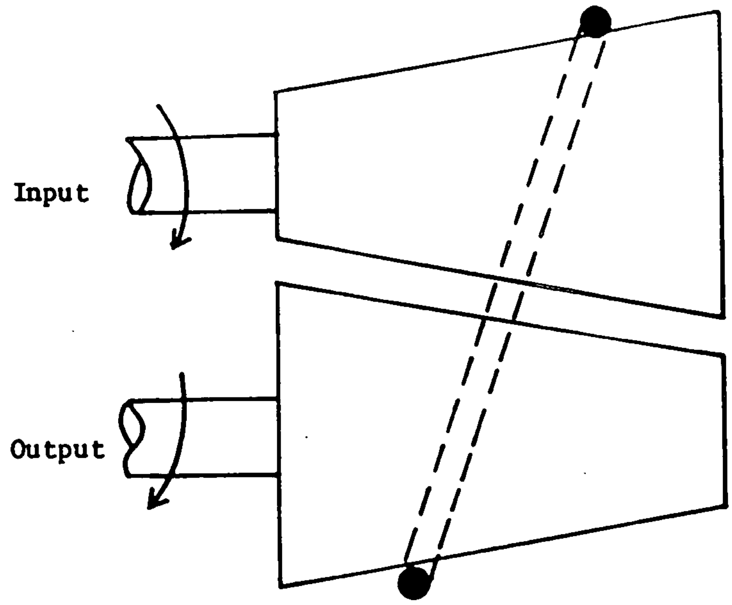
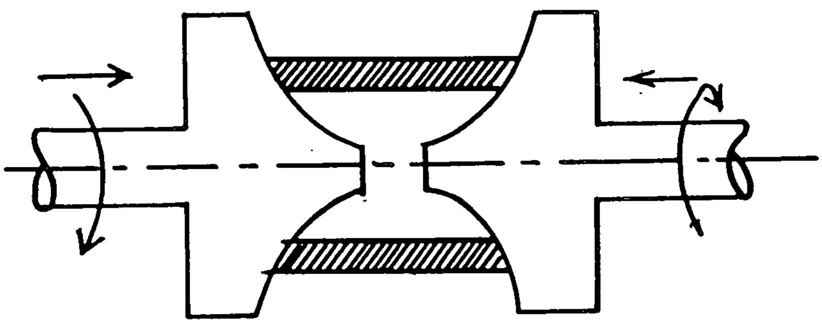
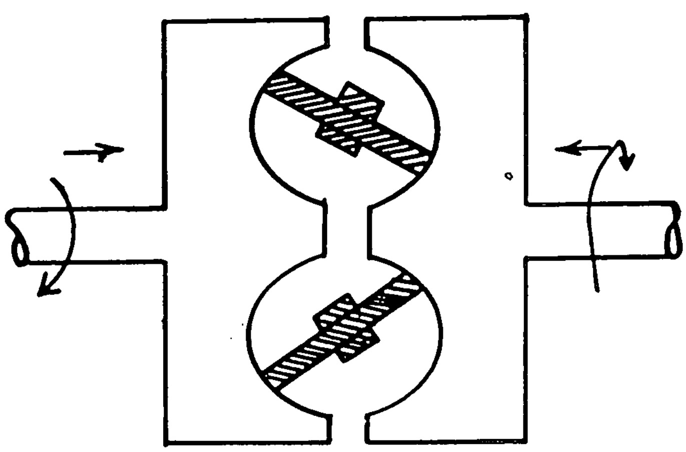


Figure 3.8 - The Cone and Ring Design .



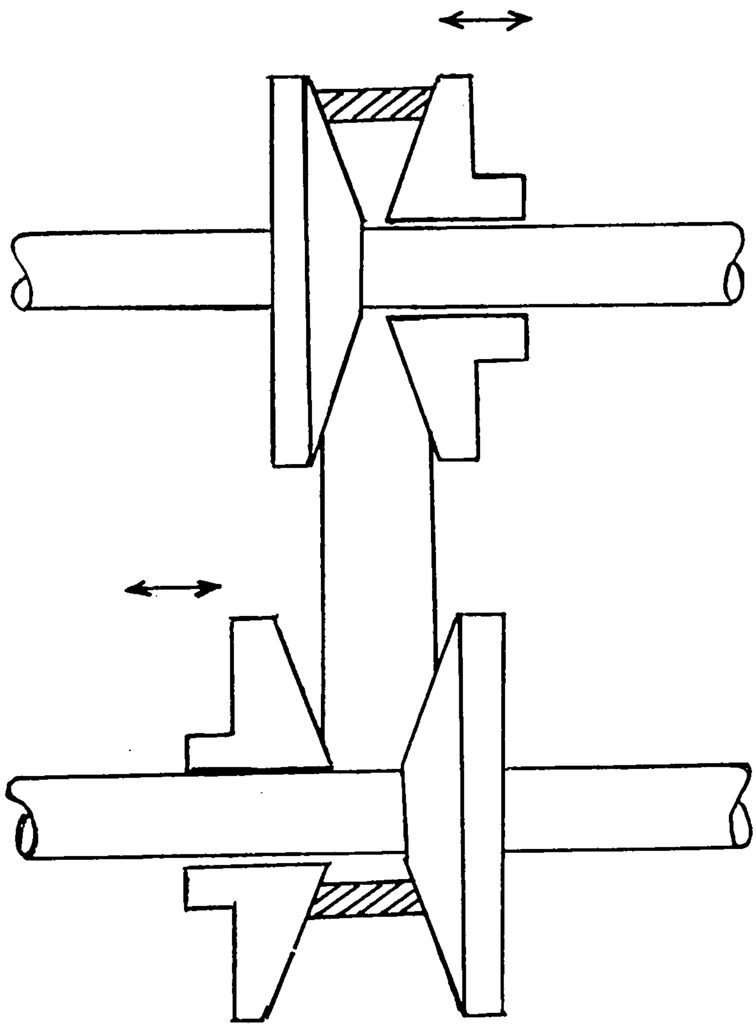
a) OFF - Centre Drive



b) On - Centre Drive

Figure 3.9 - Basic Toroidal Drives.

Figure 3.10 - Variable Diameter Pulley V-Belt Drive.



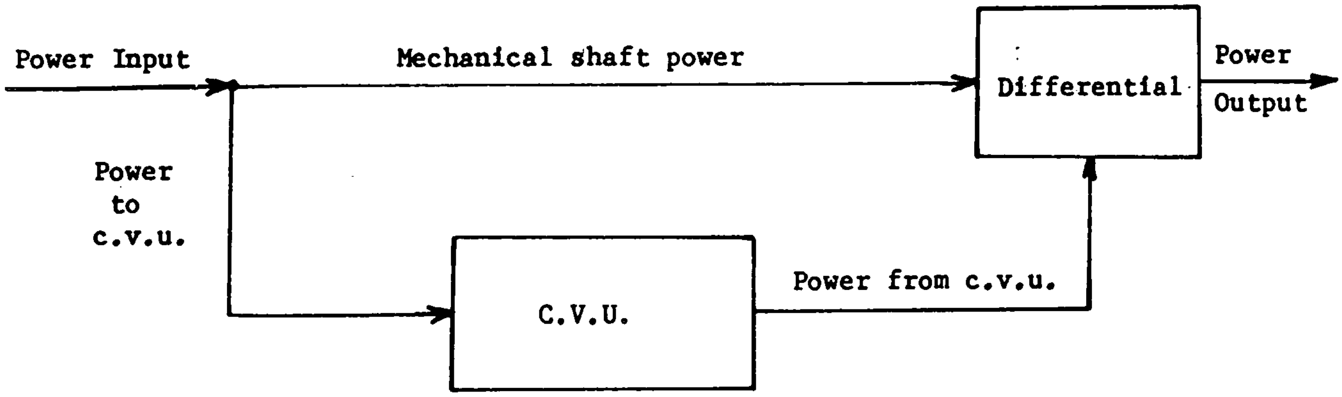


Figure 3.11 - The Power Split Principle .

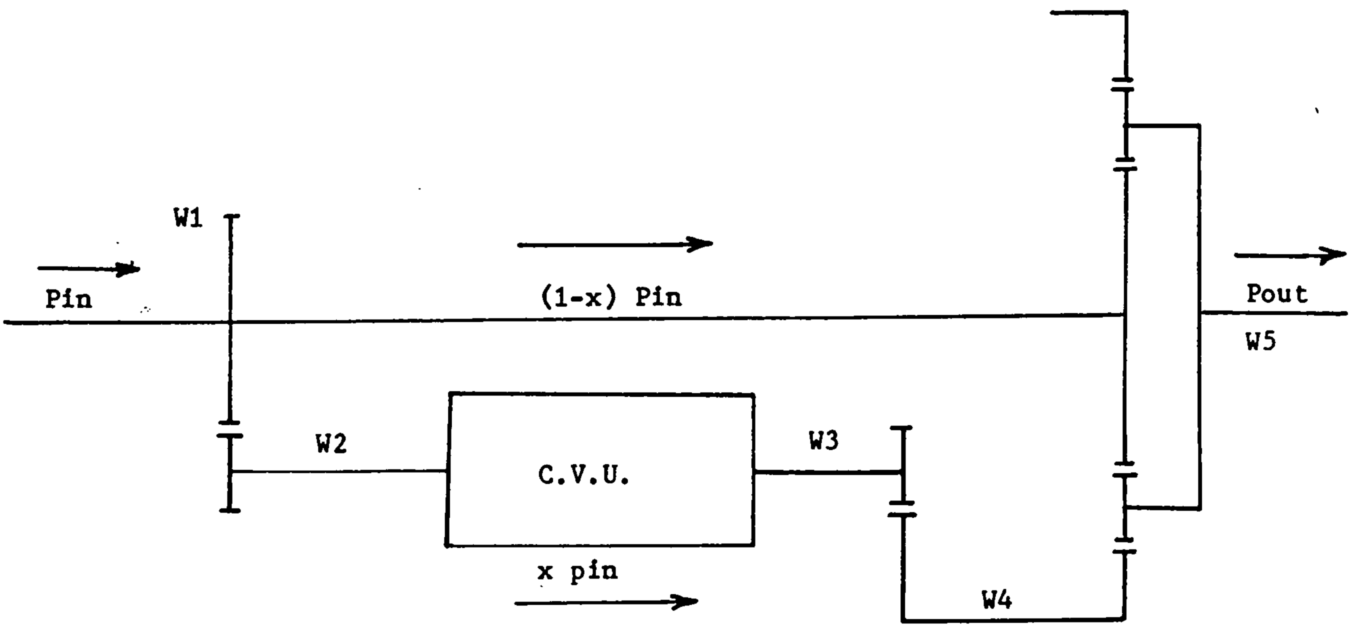


Figure 3.12 - Schematic of a Power Split C.V.T.

Figure 3.13 - Recirculative mode of operation of Conventional Power Split Drives .

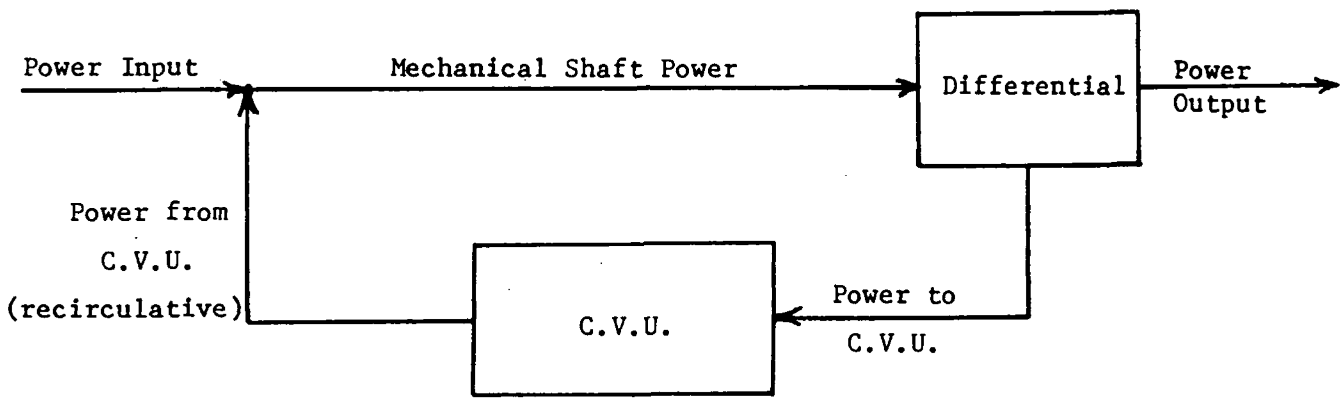


Figure 3.14 - Regenerative Power Split .

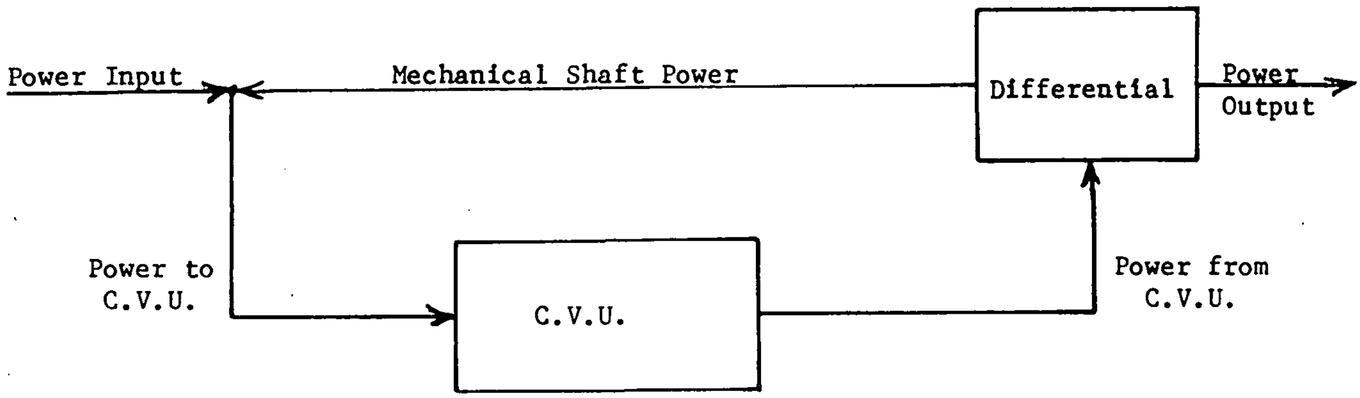
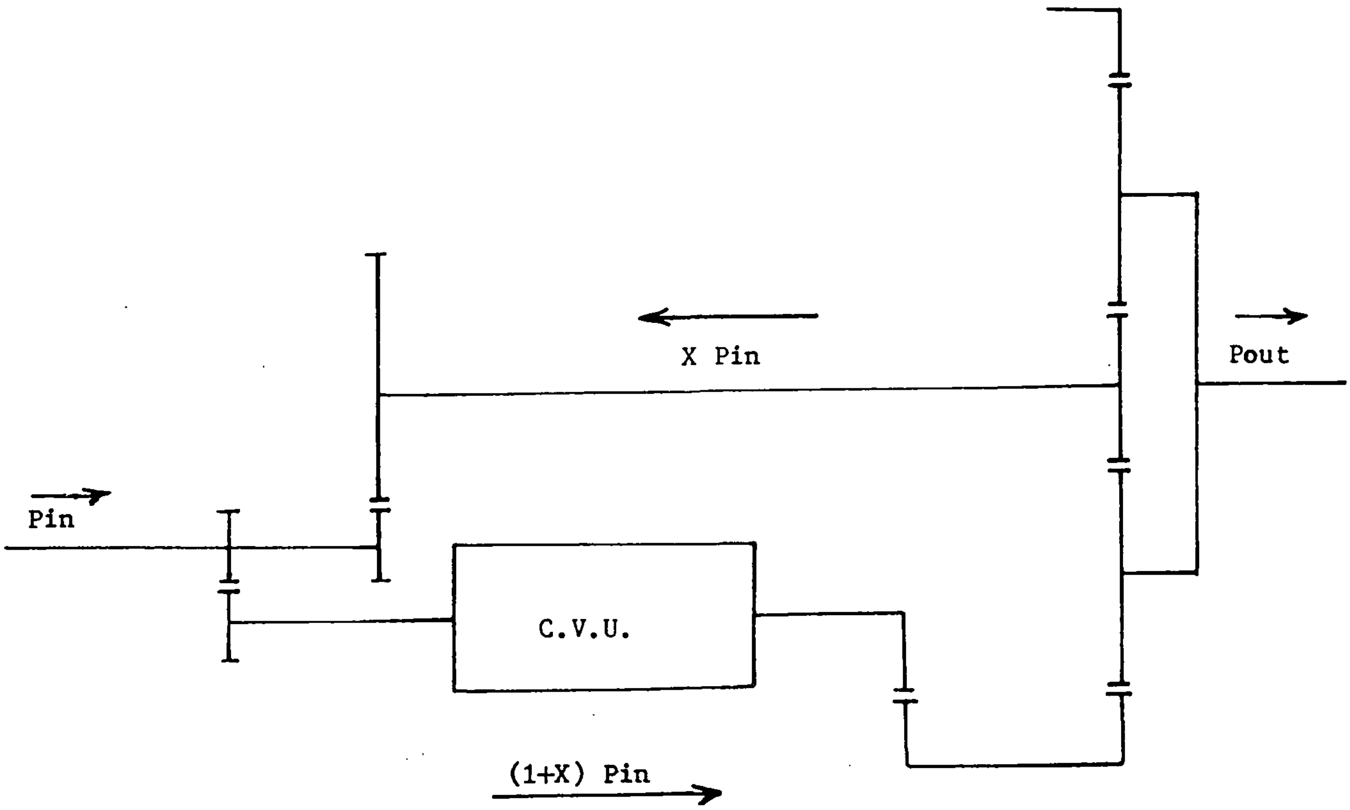
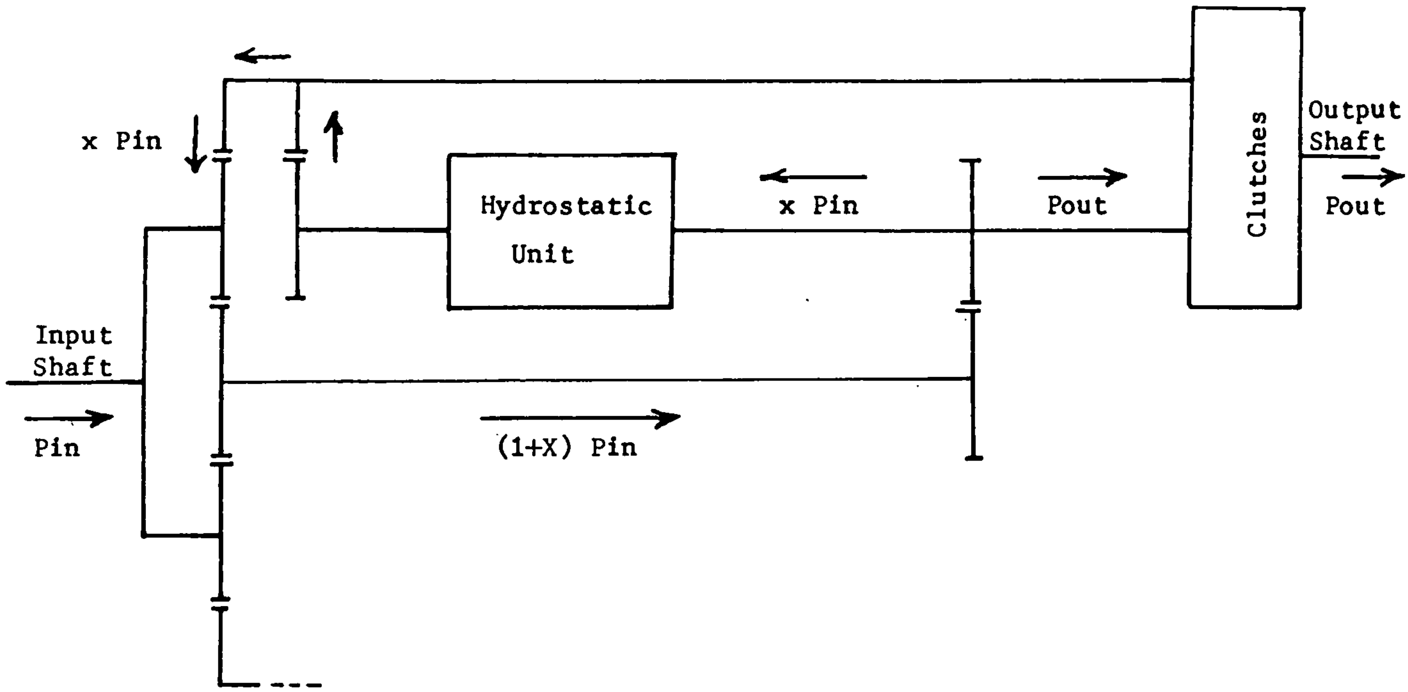


Figure 3.15 - Schematic of a Regenerative Power Split CVT .





a) " LOW " (Recirculative Power-split) mode of operation .

b) " HIGH " (Power-split) mode of operation .

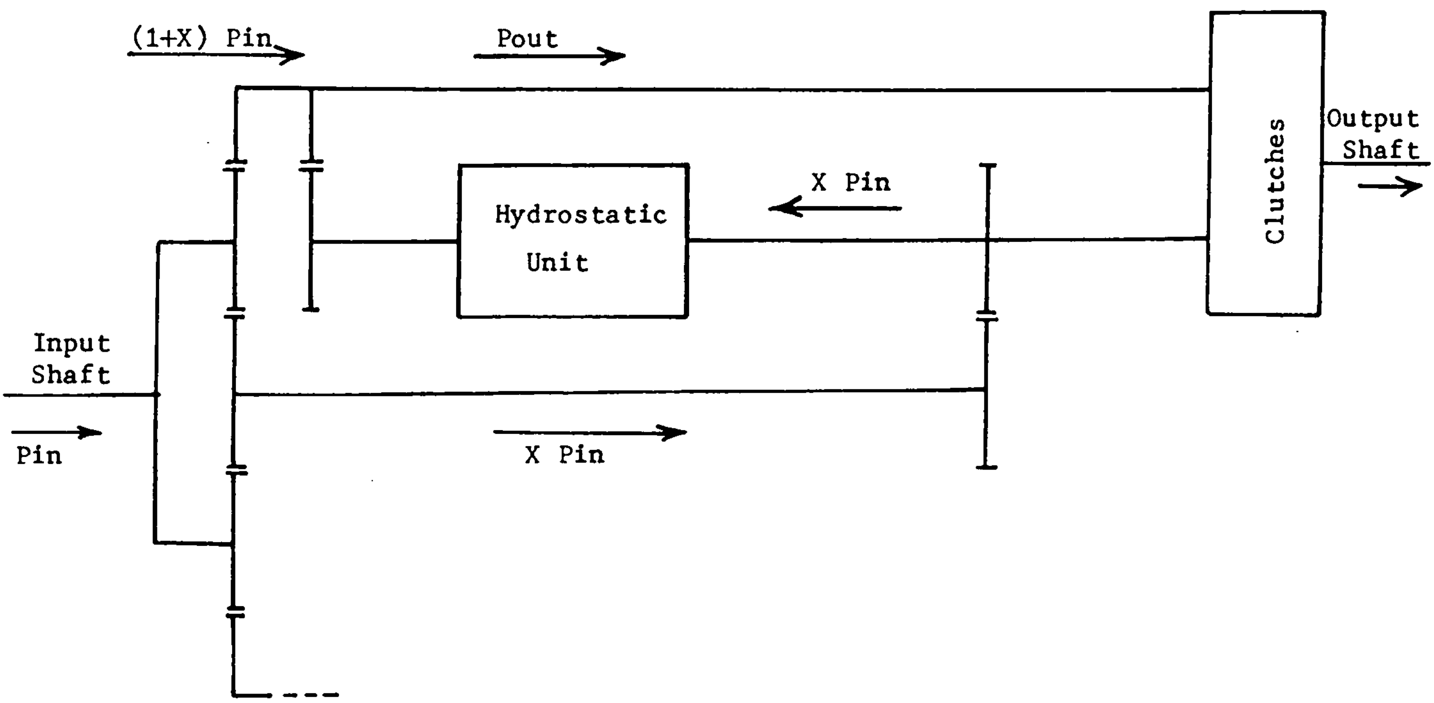


Figure 3.16 - A schematic of H.M.T. with Power Flows in High and Low modes of operation .

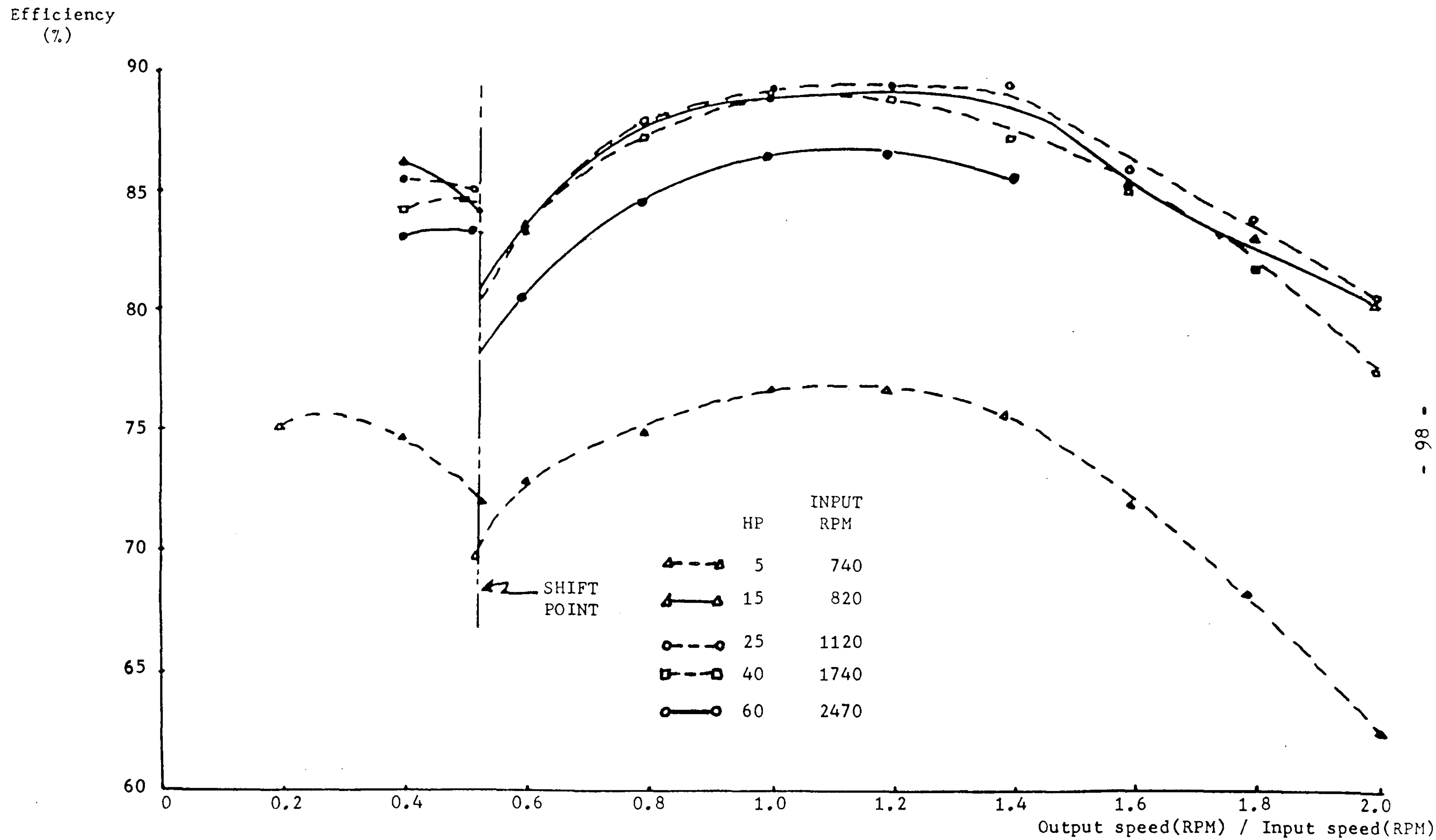


Figure 3.17 - Two-mode H.M.T. Efficiency , Test Data.
(Source : Ref.40)

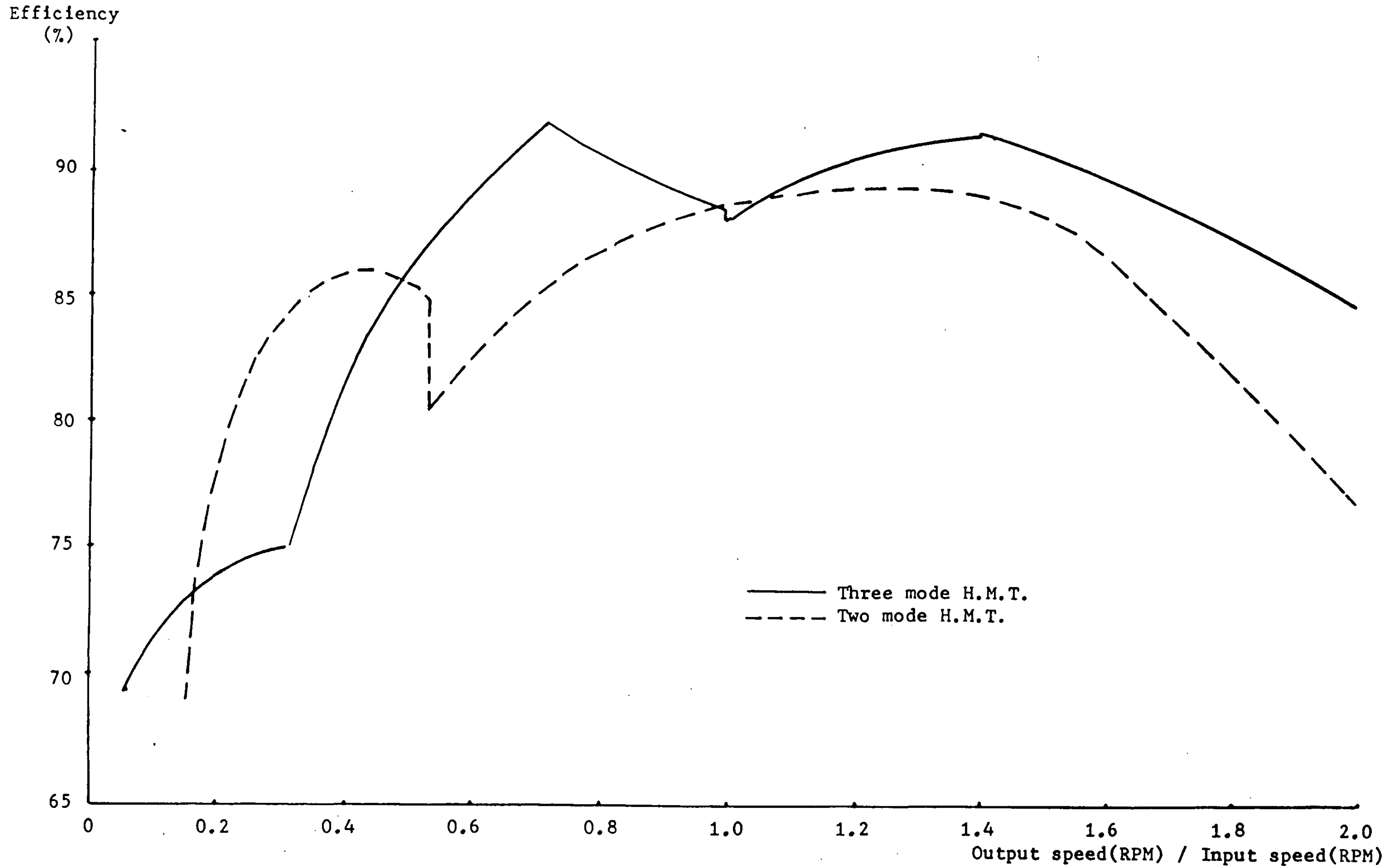
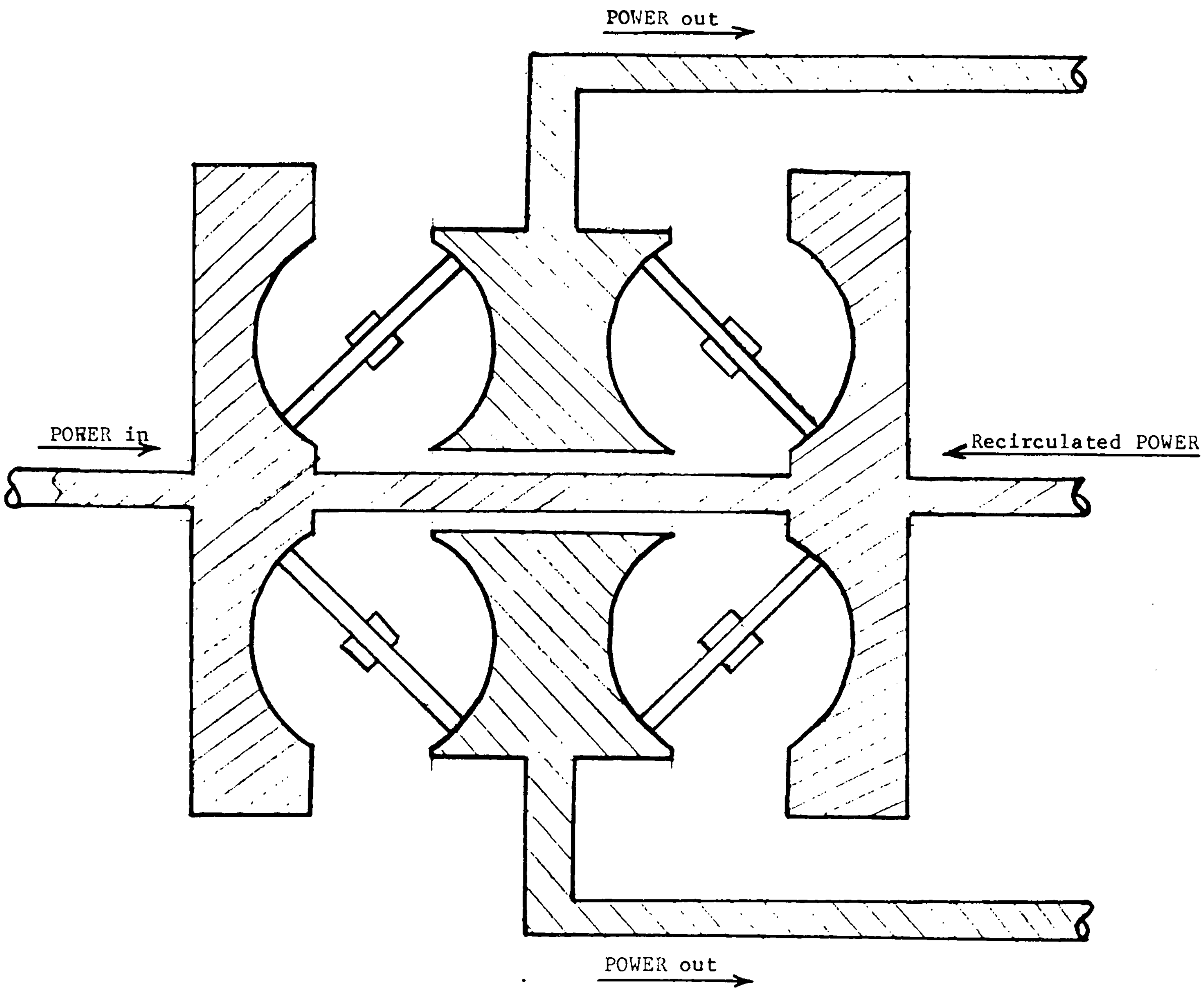


Figure 3.18 - Predicted Transmission Efficiency of the Two-mode H.M.T. vs. the Three-mode H.M.T. (Source : Ref.40).

Figure 3.19 - A schematic of the C.V.U. of B.L.Perbury Drive.



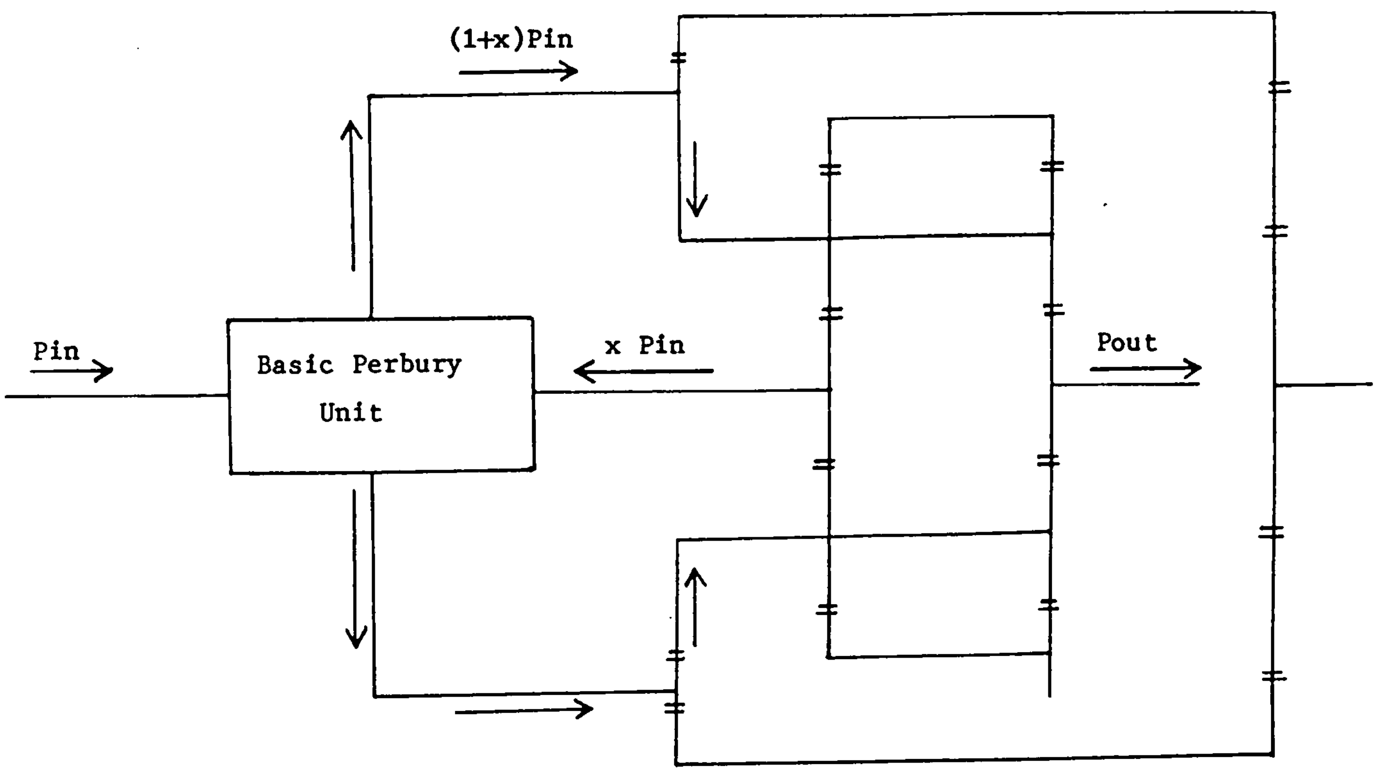


Figure 3.20 - A schematic of B.L. Perbury Drive with power flows in the " Low " regime (Recirculative power split) .

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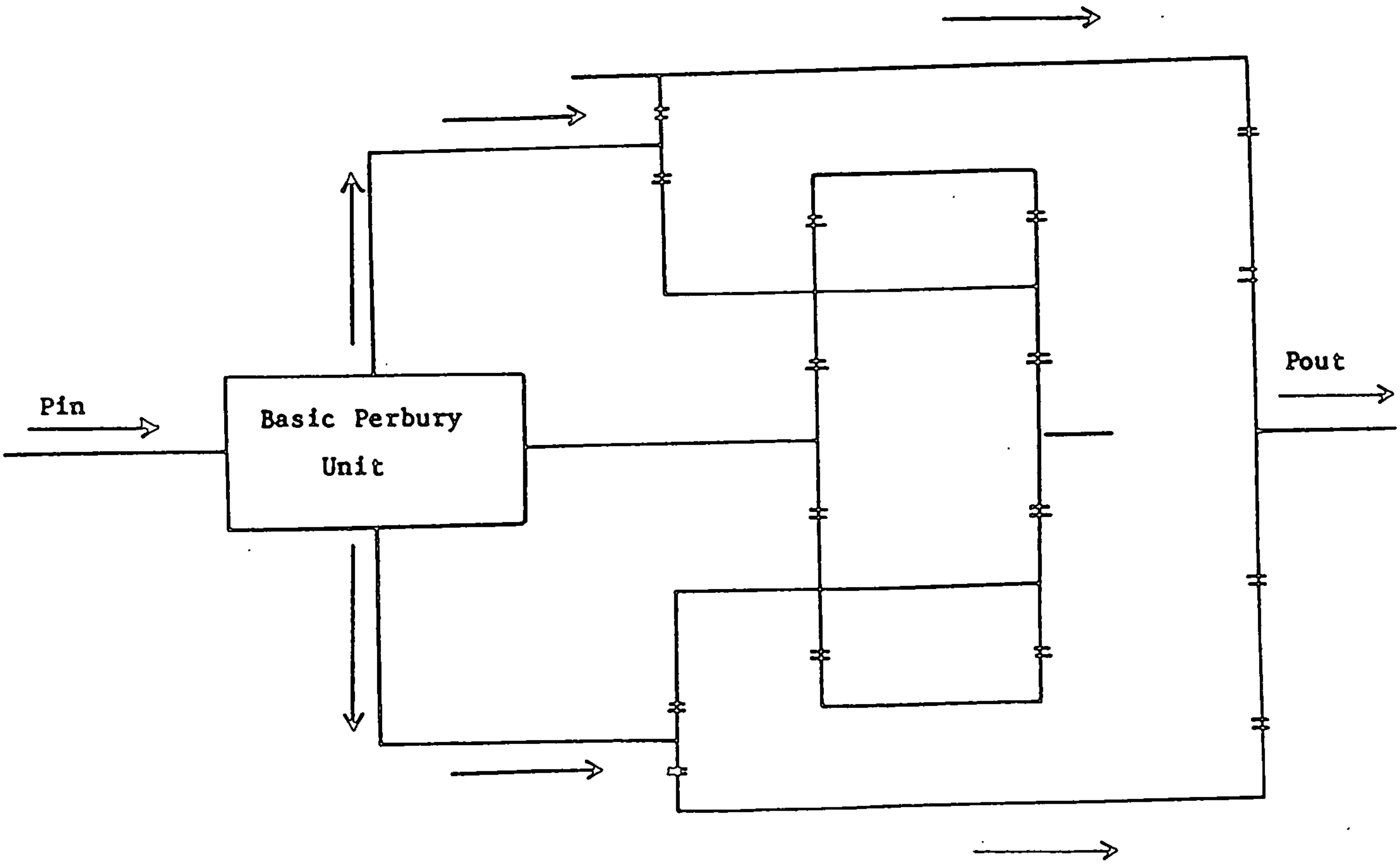
Figure 3.21 - A schematic of B.L. Perbury Drive with power flows in the " High " regime (no power-splitting).



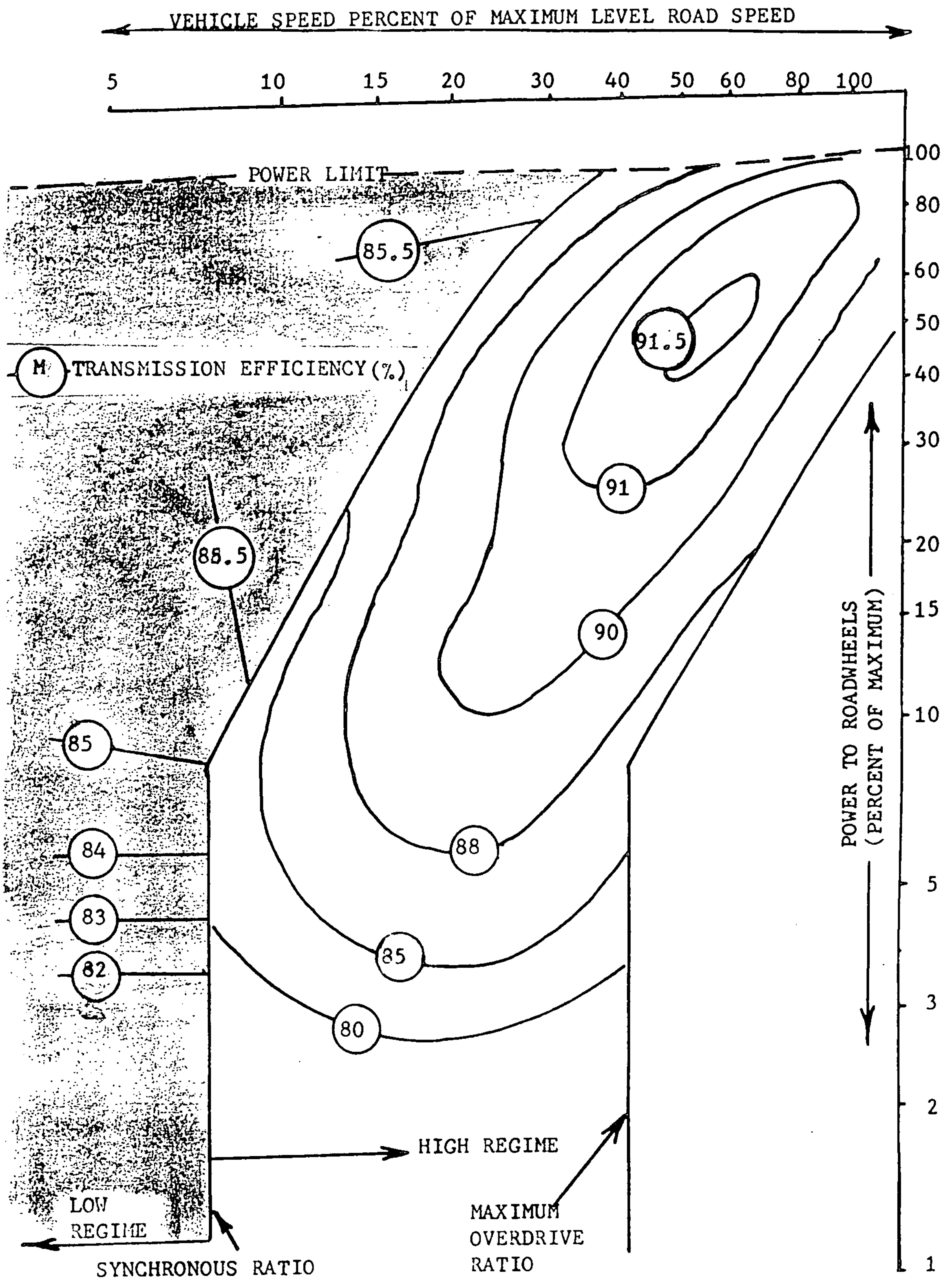


Figure 3.22 - Efficiency charactristics of B.L.Perbury Drive
(Source : Ref.42)

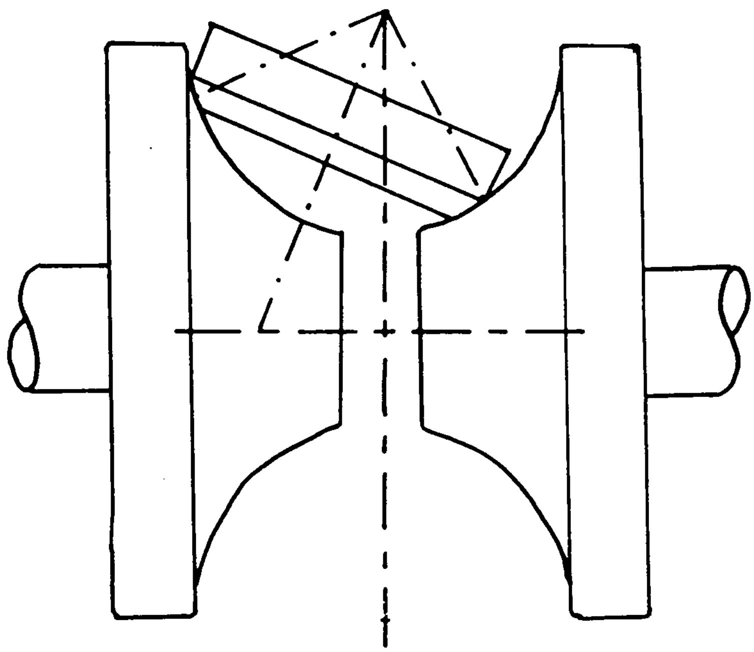


Figure 3.23 - The geometry of the basic C.R.T.D. unit.

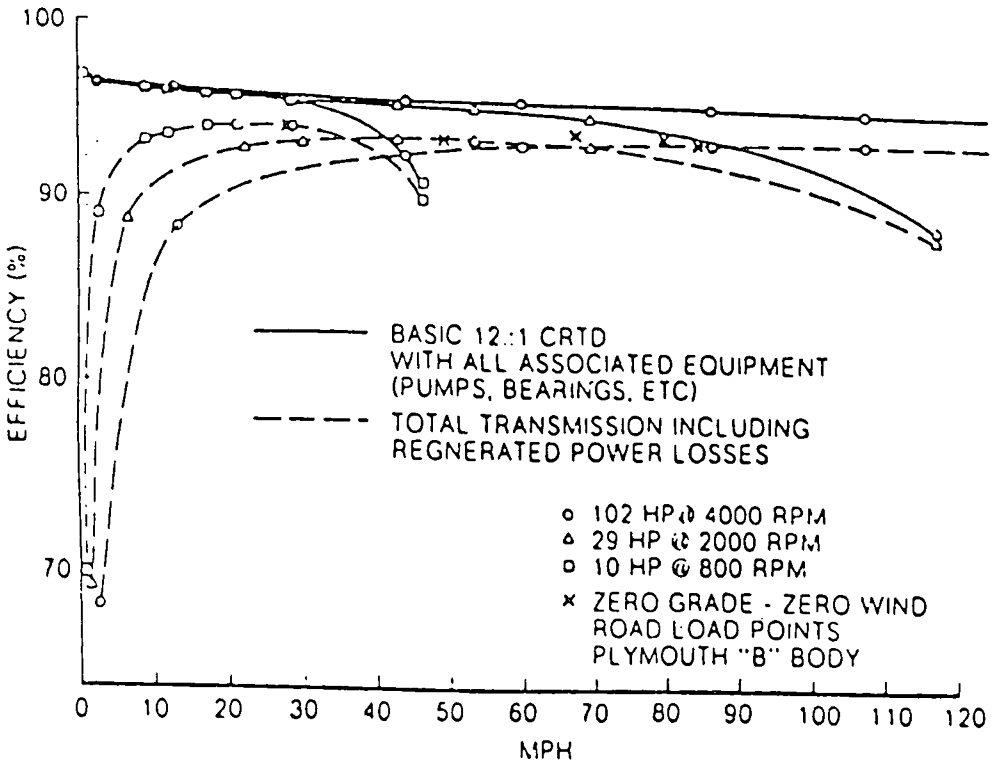


Figure 3.24 - The efficiency charactristics of C.R.T.D. (Source : Ref.43).

Figure 3.25 - A schematic of the Regenerative Power Split C.R.T.D.

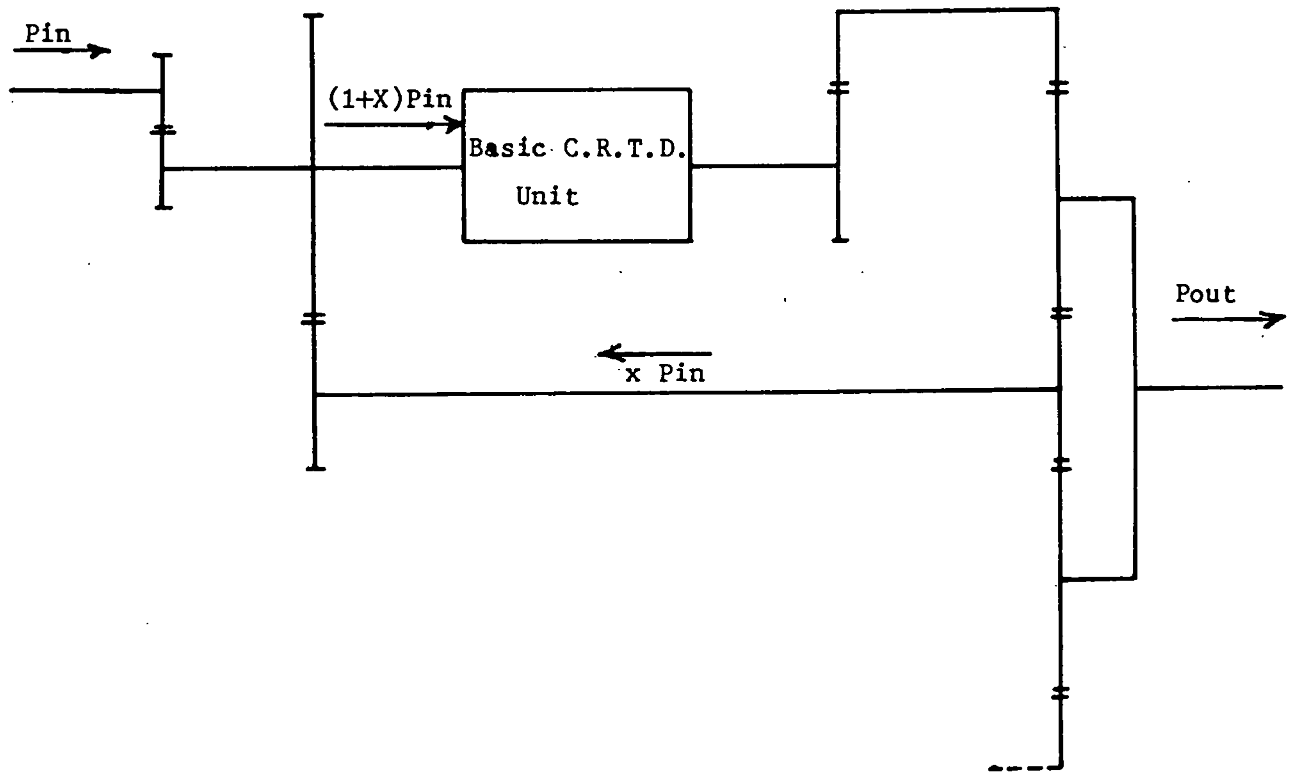


Figure 3.26 - Diagram of CVT Vehicle Simulation.

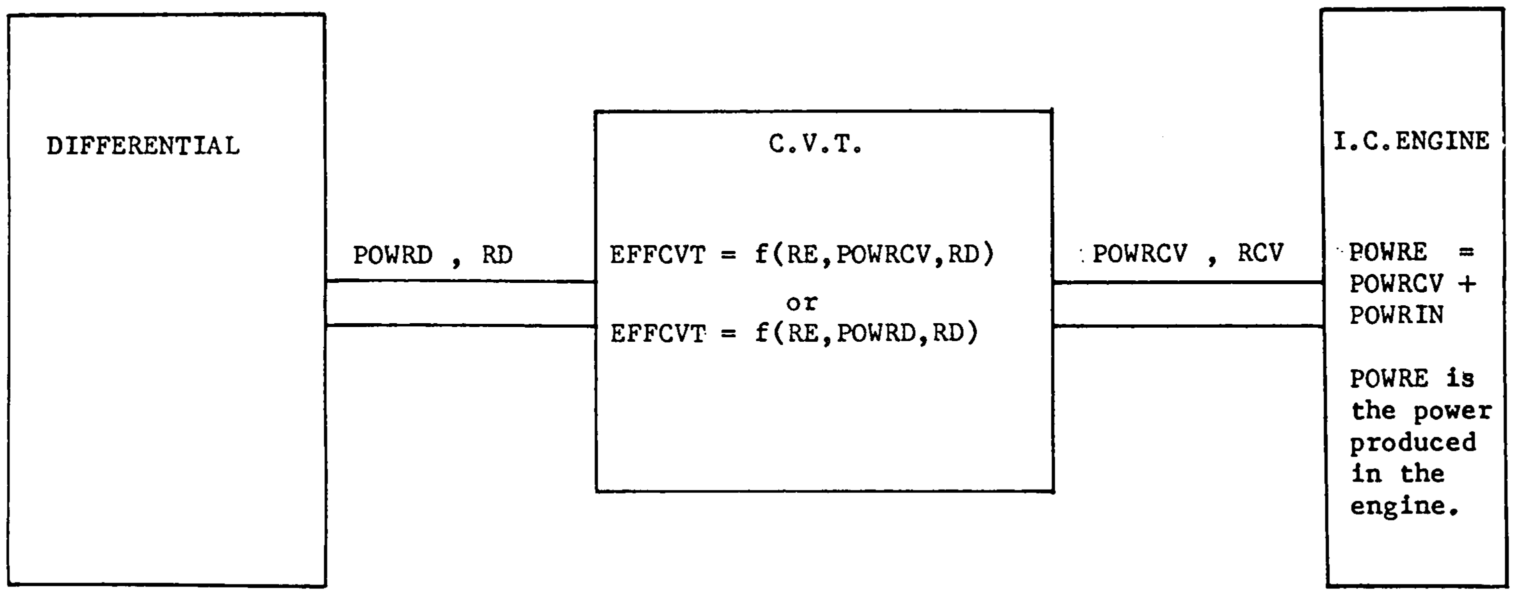
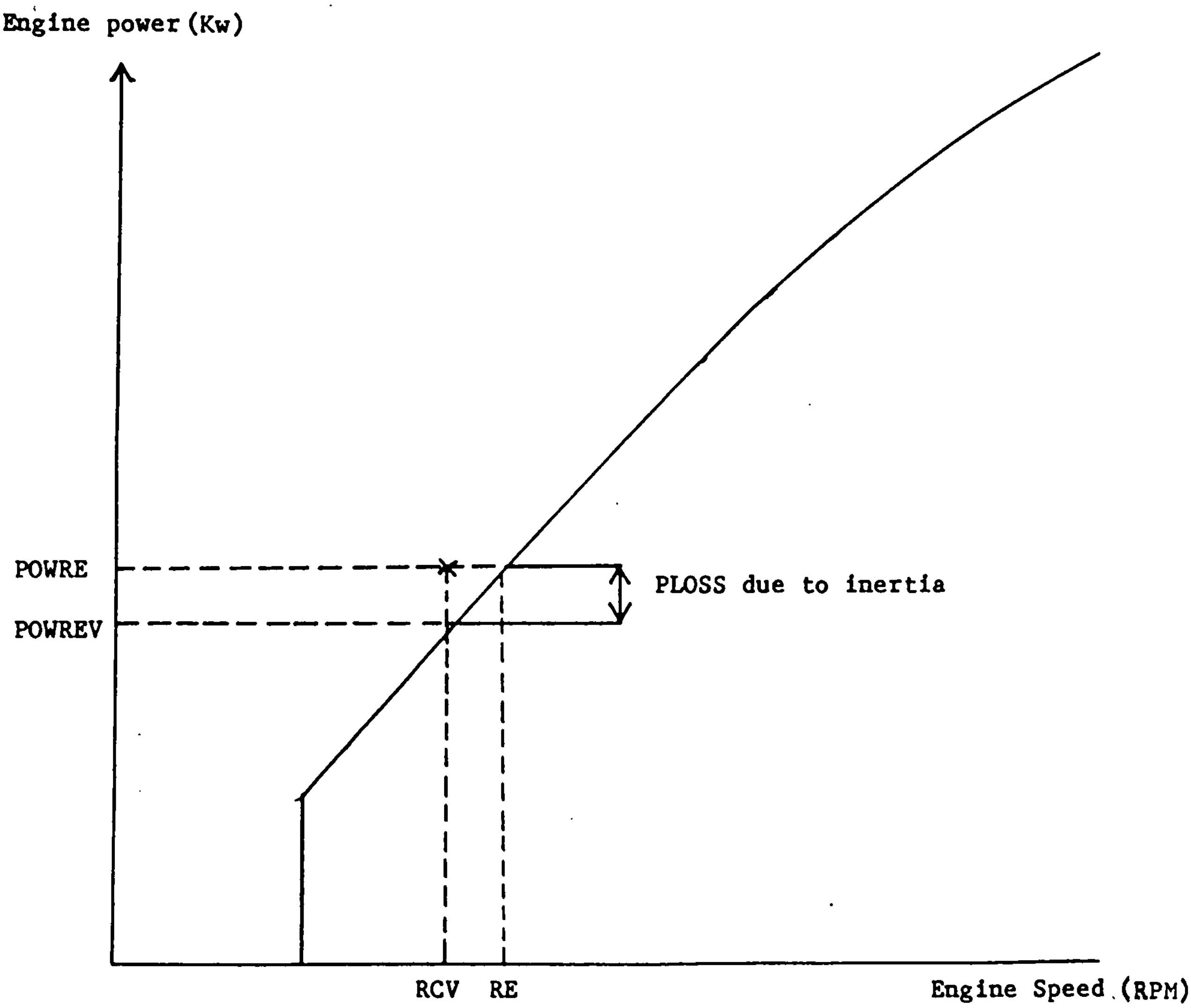


Figure 3.27 - Possible simulation error due to inertia .



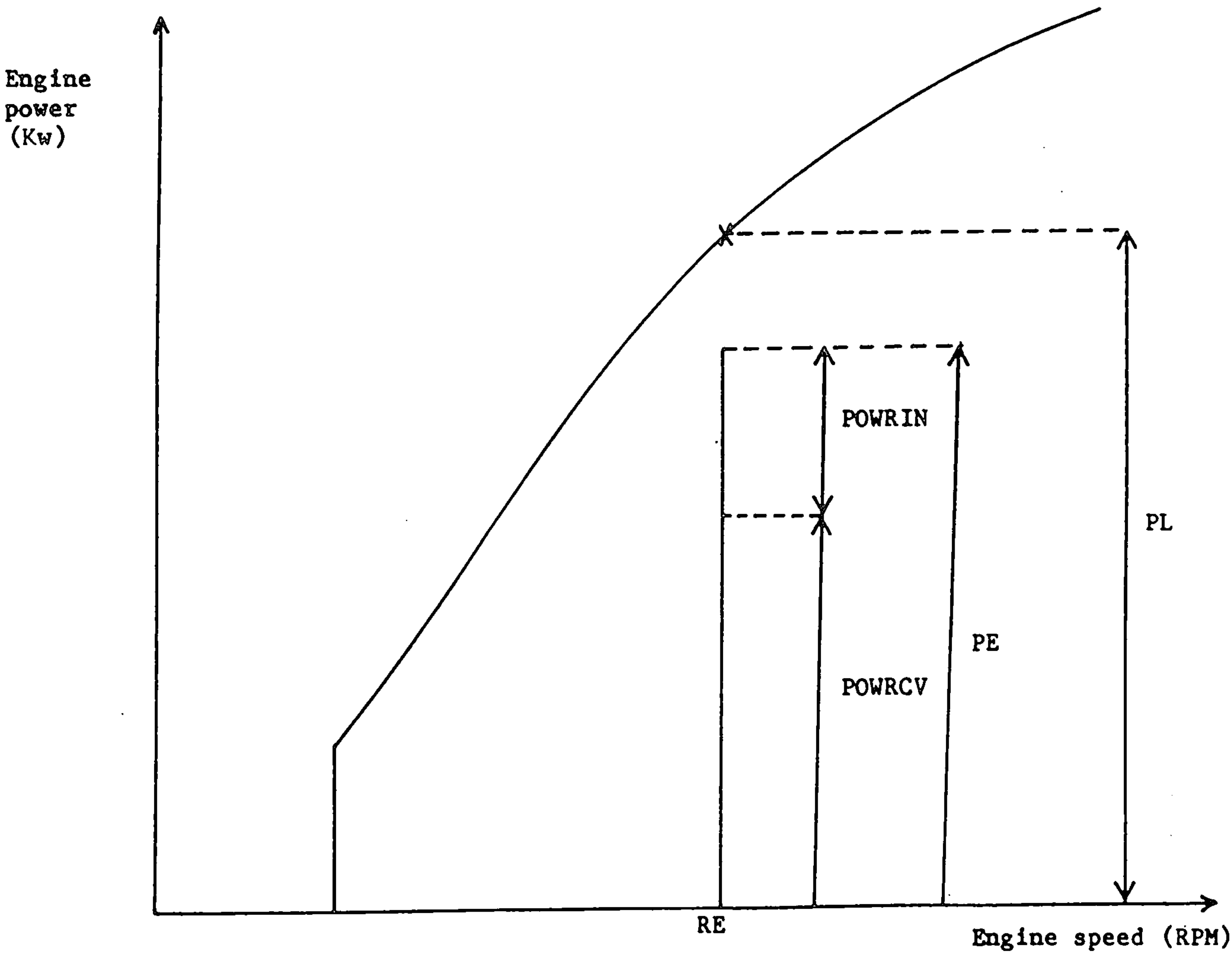


Figure 3.28 - C.V.T. control at W.O.T. acceleration .

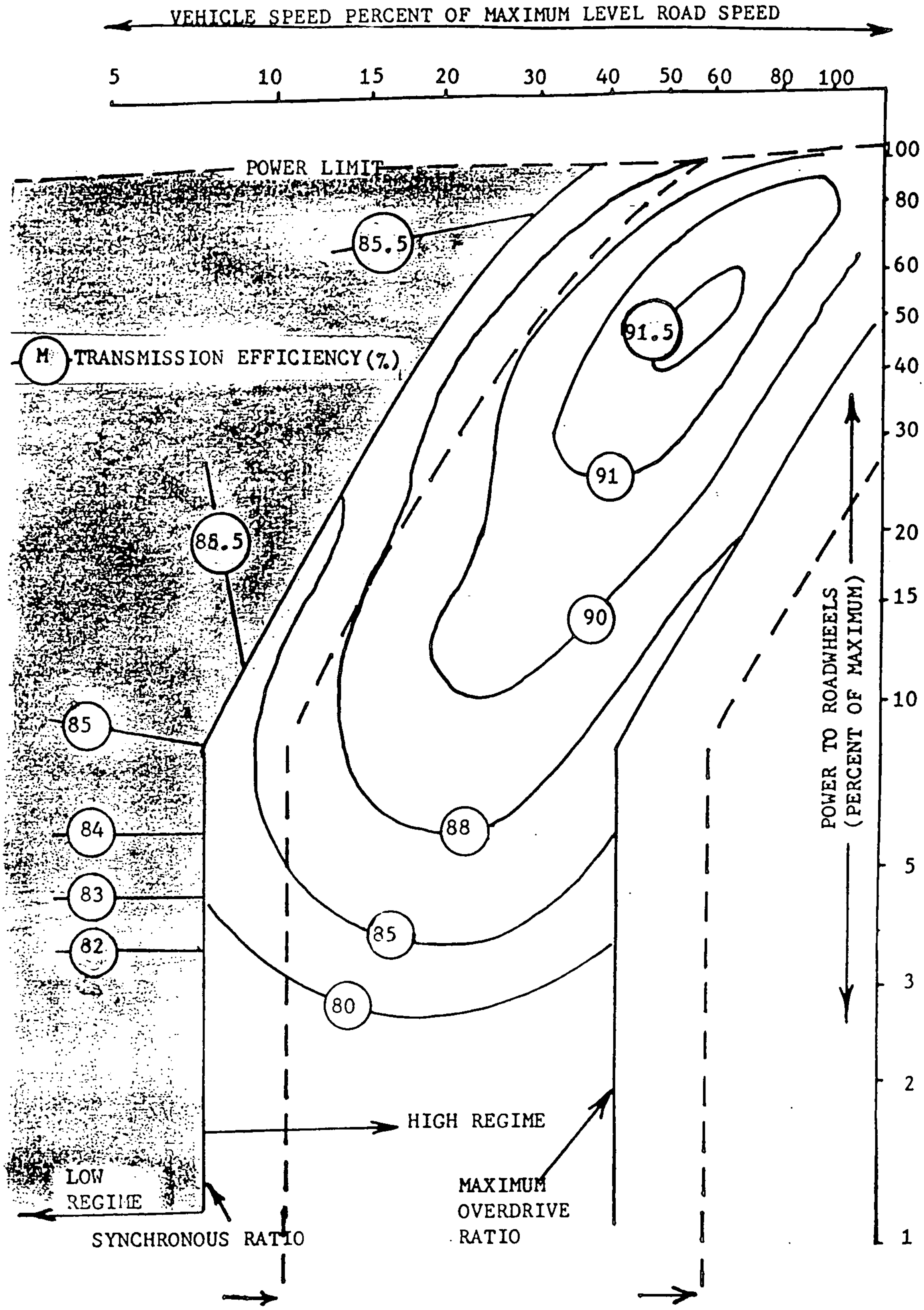


Figure 3.29 - Effect of a lower final drive ratio on the B.L.Perbury efficiency map stored in the present format.

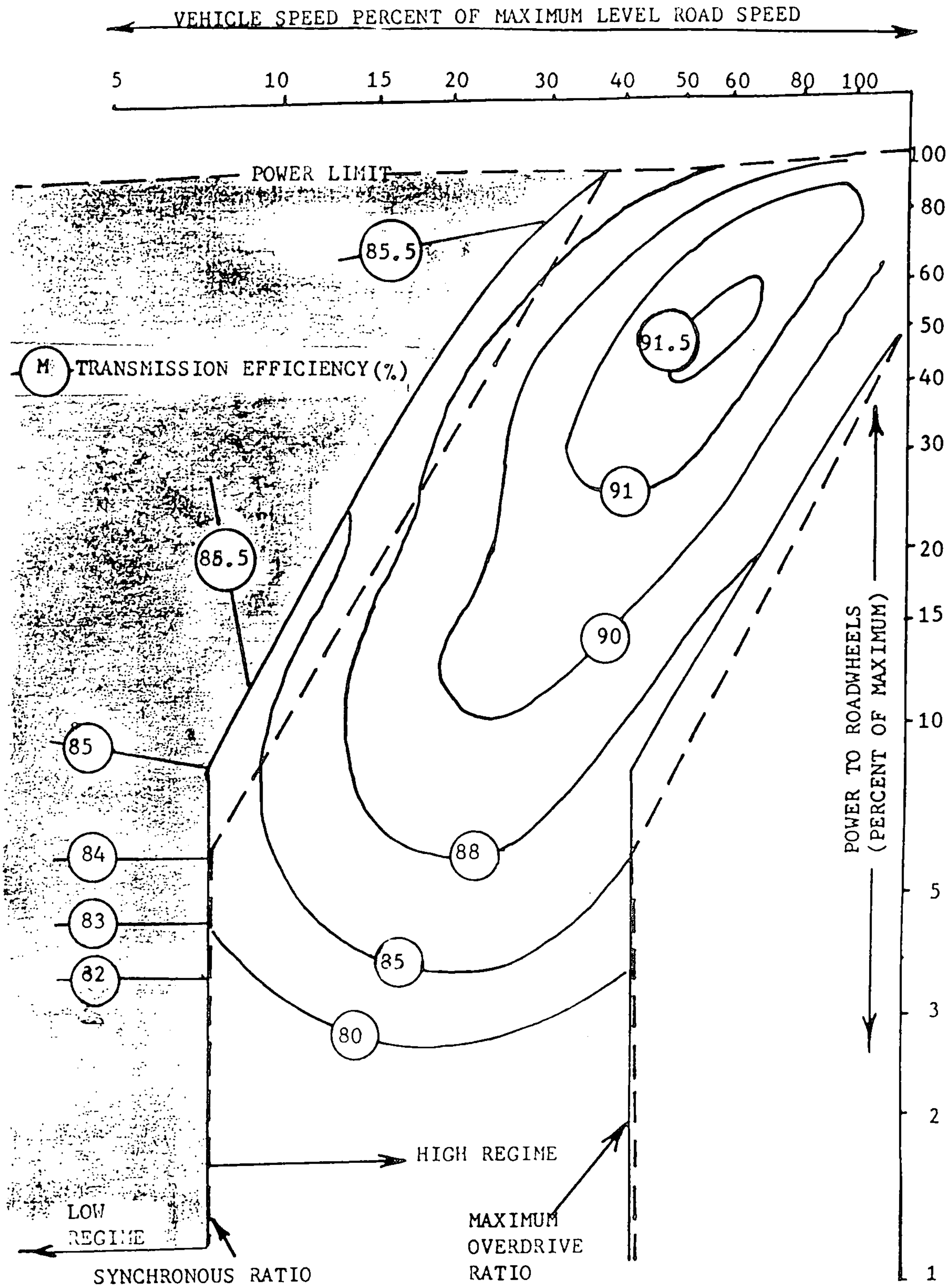


Figure 3.30 - Effect of a different engine operating scheduale on the perbury efficiency map stored in this format.

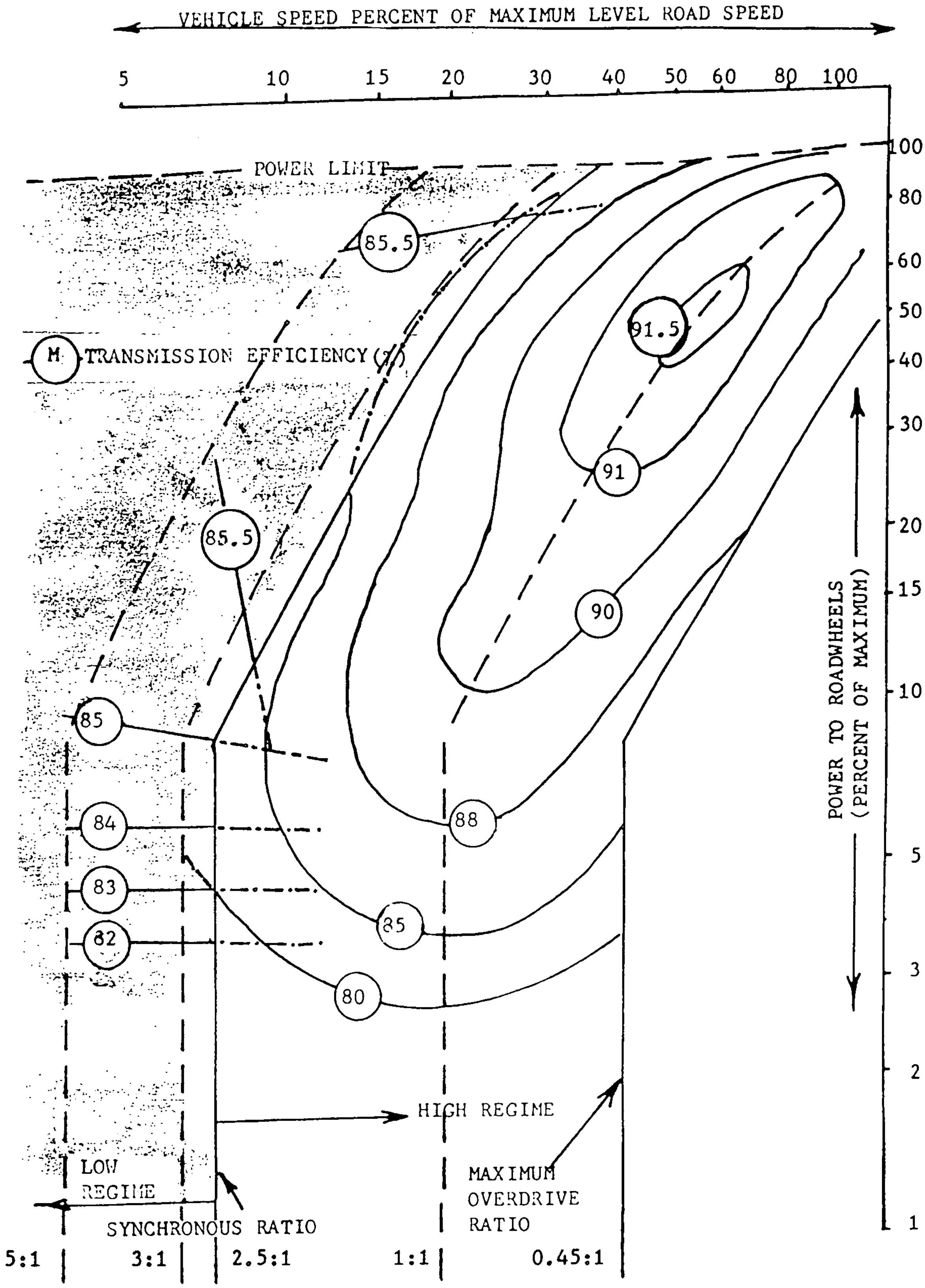


Figure 3.31 - Method used to convert perbury efficiency map from the present format to Gear ratio vs. % of maximum power.

PERBURY HIGH REGIME EFFICIENCY MAP

1.00	90.0	90.0	90.0	90.0	88.0	86.0	85.0	82.0
0.80	90.2	90.2	90.2	91.1	90.0	87.0	85.0	82.3
0.60	89.0	89.0	90.5	91.5	90.3	88.0	86.0	83.5
0.50	89.4	89.5	90.7	91.5	90.5	88.3	86.5	84.0
0.40	89.4	89.8	90.8	91.5	90.5	88.4	86.4	83.5
0.30	88.8	90.0	90.8	91.2	90.0	87.7	86.2	82.5
0.20	88.0	89.6	90.5	90.7	89.1	86.1	83.9	82.0
0.15	87.8	89.2	90.2	90.3	88.8	86.0	83.8	81.5
0.10	87.0	88.9	89.6	89.7	88.2	85.9	83.5	81.0
0.05	83.6	85.8	86.8	87.1	85.9	83.9	81.5	77.5
0.03	77.0	80.0	81.4	82.0	80.6	78.0	75.0	70.0
0.02	65.0	68.0	70.0	70.0	68.0	66.0	63.0	58.0
	0.45	0.60	0.75	1.00	1.50	2.00	2.50	3.00

NORMALISED POWER - GEAR RATIO

PERBURY LOW REGIME EFFICIENCY MAP

1.00	85.0	85.0	85.0	85.0	85.0	85.0	85.0
0.80	85.5	85.5	85.5	85.0	85.0	85.0	85.0
0.70	85.5	85.5	85.5	85.5	85.5	85.5	85.0
0.10	85.5	85.5	85.5	85.3	85.1	85.0	85.0
0.09	85.5	85.5	85.3	85.0	84.9	84.8	84.7
0.07	84.0	84.0	84.0	84.0	84.0	84.0	84.0
0.06	83.0	83.0	83.0	83.0	83.0	83.0	83.0
0.05	82.0	82.0	82.0	82.0	82.0	82.0	82.0
0.03	73.0	73.0	73.0	73.0	73.0	73.0	73.0
0.02	60.0	60.0	60.0	60.0	60.0	60.0	60.0
	1.00	1.50	2.00	2.50	3.00	3.75	5.00

NORMALISED POWER - GEAR RATIO

FIG 3.32 - B L Perbury efficiency map.

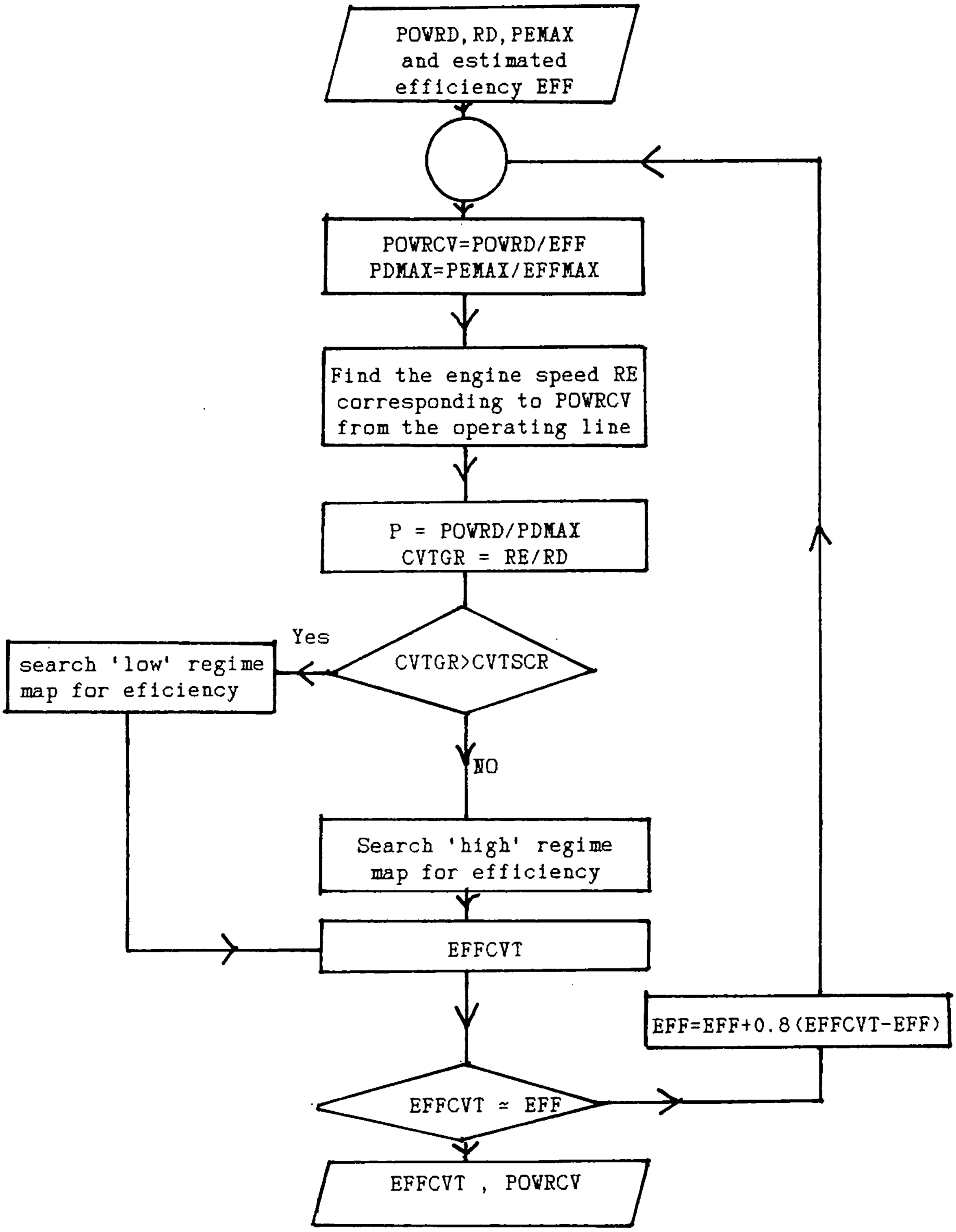


FIG 3.33 - Diagram of the process employed to find efficiency of the Perbury and the H.M.T.

H. M. T. HIGH REGIME EFFICIENCY MAP

1.00	69.0	76.0	79.0	81.0	82.0	81.0	79.0	74.4	74.0
0.60	74.0	80.0	83.2	85.6	86.6	86.4	84.5	80.4	78.3
0.40	77.3	82.9	85.2	87.6	88.8	89.0	87.0	83.4	80.4
0.25	80.4	83.5	86.3	89.0	89.4	89.3	87.9	83.4	80.4
0.15	80.2	82.5	85.4	88.6	89.0	89.0	87.4	83.4	80.5
0.05	62.5	67.9	72.1	75.4	76.5	76.5	75.2	72.5	70.0
	0.50	0.55	0.63	0.71	0.83	1.00	1.25	1.67	1.90

NORMALISED POWER - GEAR RATIO

H. M. T. LOW REGIME EFFICIENCY MAP

1.00	82.0	81.0	73.0	70.0	0.0
0.60	83.2	83.0	77.0	74.0	63.0
0.40	84.4	84.0	79.0	76.0	64.0
0.25	85.0	85.4	80.0	77.0	65.0
0.15	84.0	86.0	81.0	78.0	65.0
0.05	72.0	74.6	75.4	75.0	73.0
	1.90	2.50	4.00	5.00	7.00

NORMALISED POWER - GEAR RATIO

FIG 3.34 - Hydromechanical transmission efficiency map

LOW OPERATING BOUNDARY

NORMALISED VELOCITY & POWER RESPECTIVELY

0.005	0.098
0.010	0.284
0.020	1.000
0.385	1.000

HIGH OPERATING BOUNDARY

NORMALISED VELOCITY & POWER RESPECTIVELY

0.385	0.098
0.990	0.284
1.000	1.000

C R T D EFFICIENCY MAP
X-AXIS: NORMALISED VELOCITY: Y-AXIS: NORMALISED POWER

1,000	0,0	68,5	68,5	79,0	86,5	88,5	91,0	91,5	92,0	92,0	92,5	93,0	93,0	93,5	93,5	93,5
0,284	69,0	69,0	79,0	89,0	91,0	91,5	92,5	93,0	93,0	93,0	93,5	93,0	93,0	89,5	88,2	88,2
0,098	70,0	86,0	90,0	92,5	93,0	93,5	94,0	94,0	93,0	92,0	89,5	89,5	89,5	89,5	89,5	0,0
0,030	65,0	81,0	85,0	87,5	88,0	88,5	89,0	89,0	88,0	87,0	87,0	0,0	0,0	0,0	0,0	0,0
	0,005	0,010	0,020	0,052	0,085	0,100	0,175	0,230	0,300	0,333	0,385	0,500	0,583	0,900	0,990	1,000

FIG 3.35 - C. R. T. D. Efficiency map.

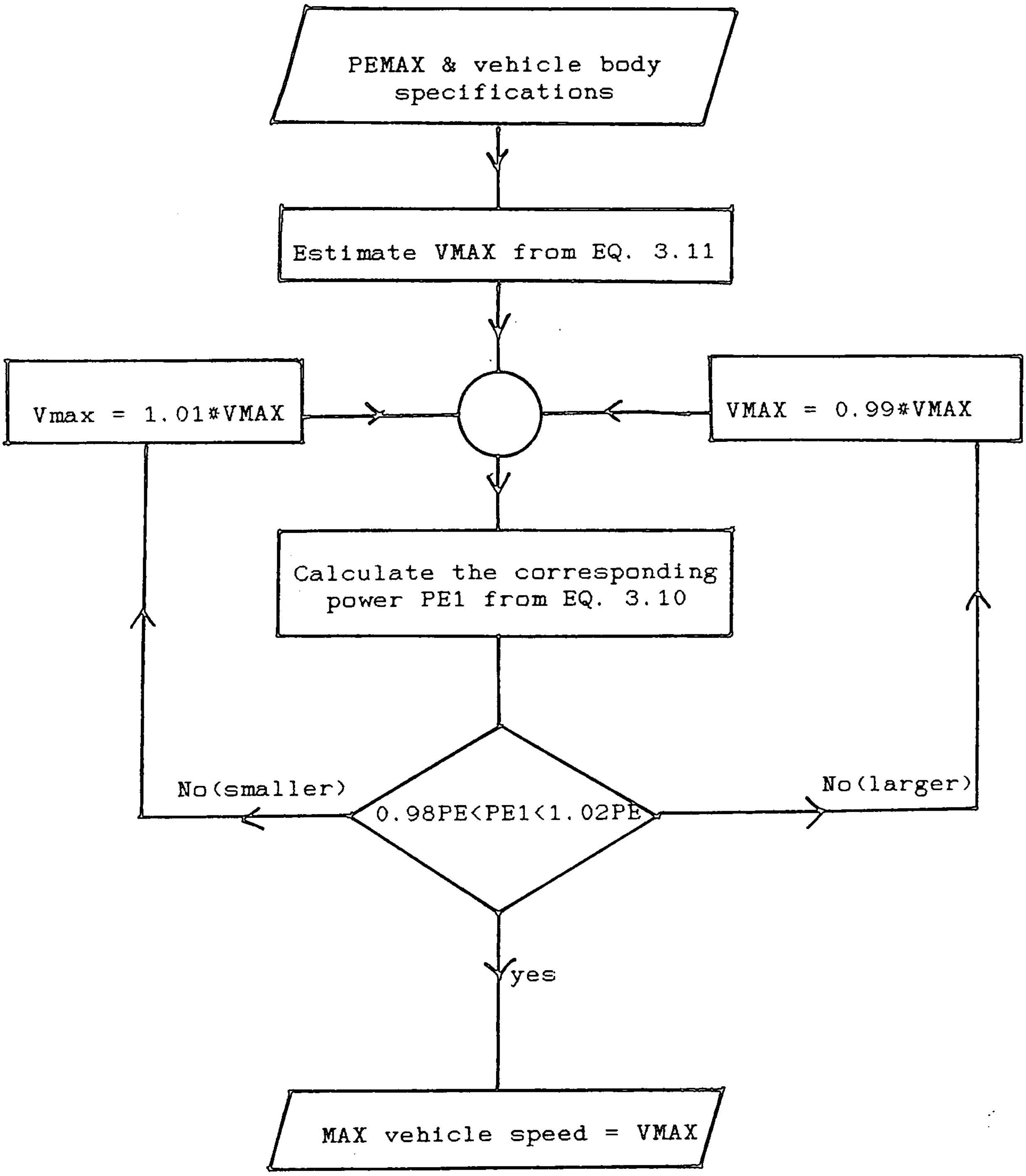


FIG 3.36 - The process used to evaluate the vehicle maximum speed required in determination of CRTD efficiency.

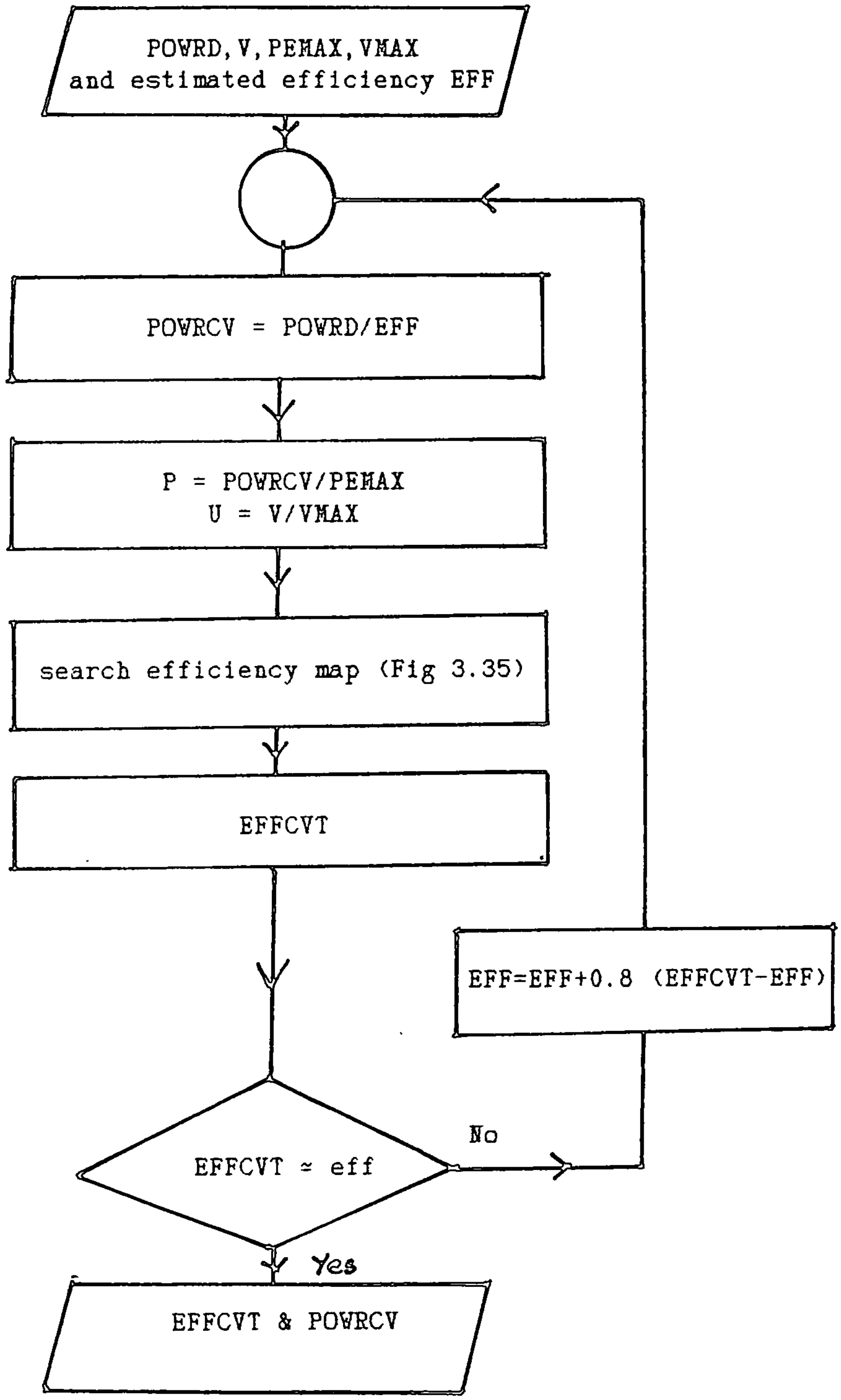


FIG 3.37 - Process used to evaluate CRTD efficiency at any condition.

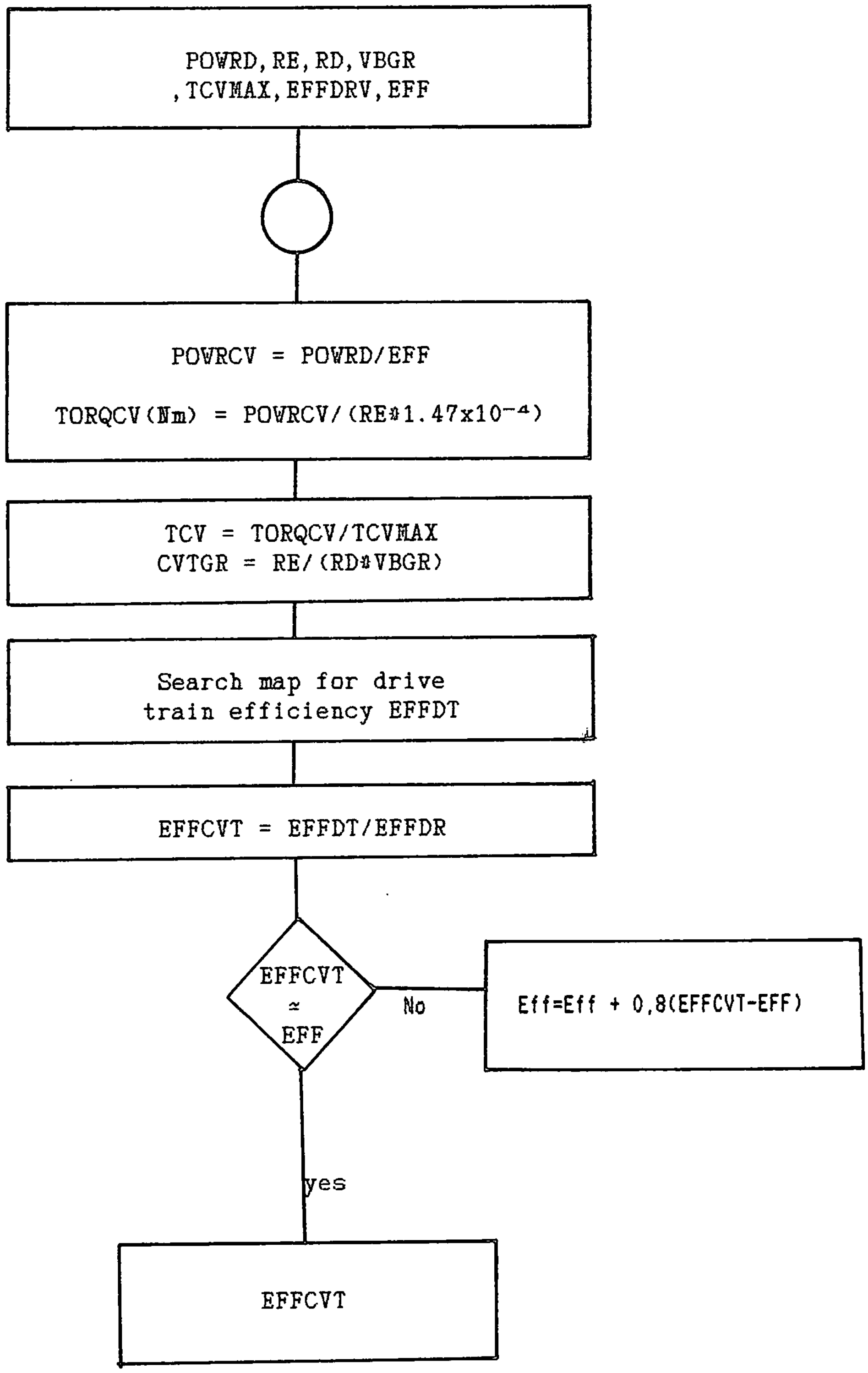


FIG 3.38 - The process employed to determine Transmatic efficiency at any condition.

C V T	Type of Experimental Efficiency Data Available	Performance Compared with Manual Gear Box			Overall gear ratio spread at Full Load
		Efficiency	Noise	Weight	
B.L.Perbury (42)	Extensive	Lower	Lower	Higher	10.0:1
C.R.T.D.(43)	Extensive	Proximate	Not Avail-able.	Not Avail-able.	12.5:1
Transmatic (38)	Extensive	Proximate	Lower	Higher	4.4:1
H.M.T.(41)	Extensive	Lower	Higher	Higher	8.0:1
Kinematic Linkages & Overrunning Clutches (31)	Poor	Higher	Not Avail-able.	Not Avail-able.	12.5:1
Nutating Traction Drive (34)	Poor	Lower	Not Avail-able.	Not Avail-able.	Not Available
Non-Circular Gears (32)	None Available	Lower	Not Avail-able.	Not Avail-able.	5.0:1
Electric	None Available	Not Available	Lower	Higher	Not Available
Rubber V-belt Drive (37)	Poor	Lower	Lower	Proxi--mate	4.0:1

TABLE 3.1 A summary of characteristics of the available types of CVT

C H A P T E R F O U R

PROGRAM VERIFICATION AND INPUT DATA

4 Program Verification And Input Data

The objective of this work is to examine the effect of various proposed design improvements (Section 1.5) by comupter simulation.

Before embarking on this task, however, it is necessary to establish the integrity of the simulation package. This is done by comparison of 'JANUS' computer predictions of vehicle performance with that of measured data.

4.1 Program Verification

The variables used as input data to the simulation package for two vehicles are shown in Table 4.1. The vehicles are selected due to availability of their physical parameters and engine maps.

Tables 4.2 and 4.3 compare the predicted fuel consumption values obtained by the simulation package with the measured data published by the Department of Transport (45). The predicted accelerative performances and maximum speeds are also compared with measured data (46). The predicted fuel consumption values are within 3% of the official data for cruise conditions and 10% for ECE 15 driving cycle. The larger differences in urban fuel economy are due to its critical dependence on the value of idle fuel consumption.

The predicted accelerative and maximum speed performance of Vehicle 1 are not as accurate as Vehicle 2 when compared with measured data. This can be explained as the effect of the different engine (than the official) used in the simulation, see Table 4.1. The proximity of the results of the simulation to measured data, arouses confidence in the simulation package for further predictive studies.

4.2 Input Data

The purpose of this work is to evaluate the effects of various proposed design improvements on the performance of present day vehicles. It is, therefore, necessary to establish the parameters considered as the norm for present vehicles. As this work concentrates on the vehicles of the 1000-2000cc range due to their growing dominance in the market (Section 1.1), the parameters selected (Table 4.4) are therefore of three vehicles considered as typical representatives of new vehicles in this range. This is also the case for the engine maps selected, shown in Figs 4.1, 4.2 & 4.3.

The effect of variations of environmental conditions on vehicle performance is also studied in this report. The environmental conditions considered as the norm, are the standard conditions of 'JANUS', shown in Table 2.1.

I. C. ENGINE CHARACTERISTICS

I. C. ENGINE PERFORMANCE LIMIT CURVE

SPEED (RPM)	TORQUE (NM)	BMEP (PSI)	POWER (KW)
750.00	61.19	110.00	4.80
996.00	66.75	120.00	6.96
1500.00	66.75	120.00	10.48
1998.00	72.31	130.00	15.13
2496.00	77.87	140.00	20.35
3000.00	77.87	140.00	24.46
3498.00	77.87	140.00	28.52
3996.00	72.31	130.00	30.25
4500.00	66.75	120.00	31.45
4998.00	66.75	120.00	34.93
5496.00	61.19	110.00	35.21
6000.00	55.62	100.00	34.94

I C ENGINE CVT OPERATING CURVE

SPEED (RPM)	BMEP (PSI)	POWER (KW)
996.00	0.00	0.00
996.00	90.00	5.23
1500.00	100.00	8.75
1998.00	110.00	12.82
3000.00	120.00	21.00
4998.00	120.00	34.99
5496.00	110.00	35.27
6000.00	100.00	35.00

I C ENGINE PERFORMANCE MAP

X-SPEED (rpm/1000); Y-BMEP (psi); SFC-PTS/HP-HR

140.00	0.53	0.53	0.53	0.53	0.53	0.53	0.51	0.51	0.51	0.51	0.51	0.51
130.00	0.51	0.51	0.51	0.51	0.49	0.49	0.49	0.50	0.50	0.50	0.50	0.50
120.00	0.57	0.57	0.50	0.48	0.48	0.48	0.49	0.49	0.50	0.53	0.53	0.53
110.00	0.57	0.54	0.49	0.48	0.48	0.48	0.49	0.50	0.50	0.52	0.55	0.55
100.00	0.56	0.53	0.50	0.49	0.48	0.49	0.50	0.51	0.51	0.52	0.53	0.58
90.00	0.56	0.54	0.51	0.50	0.49	0.49	0.50	0.52	0.53	0.53	0.54	0.55
80.00	0.57	0.55	0.53	0.51	0.50	0.50	0.51	0.53	0.54	0.55	0.56	0.57
70.00	0.57	0.56	0.54	0.53	0.53	0.52	0.53	0.55	0.57	0.58	0.59	0.60
60.00	0.59	0.58	0.56	0.55	0.55	0.55	0.56	0.58	0.60	0.63	0.63	0.65
50.00	0.63	0.60	0.60	0.59	0.58	0.59	0.60	0.63	0.65	0.68	0.70	0.70
40.00	0.70	0.68	0.65	0.65	0.65	0.65	0.65	0.68	0.70	0.75	0.80	0.80
30.00	0.80	0.80	0.75	0.75	0.75	0.75	0.80	0.80	0.85	0.90	0.95	1.00
20.00	1.00	1.00	1.00	0.95	0.95	1.00	1.00	1.05	1.10	1.20	1.30	1.30
10.00	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.70	1.70	1.70
	0.75	1.00	1.50	2.00	2.50	3.00	3.50	4.00	4.50	5.00	5.50	6.00

FIG 4.1 - Fuel map of the mini engine

I. C. ENGINE CHARACTERISTICS

I. C. ENGINE PERFORMANCE LIMIT CURVE

SPEED (RPM)	TORQUE (NM)	BMEP (PSI)	POWER (KW)
990.00	99.65	116.00	10.33
1485.00	106.52	124.00	16.56
1980.00	113.39	132.00	23.51
2475.00	117.69	137.00	30.50
3025.00	120.27	140.00	38.09
3520.00	120.27	140.00	44.32
4015.00	117.69	137.00	49.47
4510.00	109.96	128.00	51.92
5005.00	103.94	121.00	54.47
5500.00	95.35	111.00	54.91

I. C. ENGINE CVT OPERATING CURVE

SPEED (RPM)	BMEP (PSI)	POWER (KW)
990.00	0.00	0.00
990.00	105.00	9.36
1996.50	112.00	20.41
2497.00	119.00	26.77
3498.00	126.00	39.71
3998.50	133.00	47.91
4499.00	126.00	51.07
4999.50	119.00	53.60
5500.00	111.00	55.00

I. C. ENGINE PERFORMANCE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); SFC-PTS/HP-HR

140.00	0.49	0.49	0.49	0.49	0.47	0.48	0.49	0.49	0.49	0.49
133.00	0.50	0.50	0.50	0.47	0.46	0.47	0.48	0.48	0.48	0.48
126.00	0.55	0.55	0.48	0.44	0.44	0.45	0.46	0.51	0.51	0.51
119.00	0.55	0.52	0.46	0.45	0.45	0.46	0.47	0.49	0.51	0.51
112.00	0.48	0.49	0.47	0.45	0.46	0.46	0.47	0.49	0.50	0.53
105.00	0.42	0.48	0.47	0.46	0.47	0.46	0.47	0.49	0.50	0.53
98.00	0.44	0.47	0.47	0.47	0.47	0.46	0.47	0.50	0.50	0.53
91.00	0.46	0.48	0.47	0.48	0.48	0.48	0.48	0.50	0.51	0.54
84.00	0.50	0.48	0.48	0.49	0.49	0.49	0.49	0.51	0.51	0.57
77.00	0.50	0.50	0.49	0.50	0.51	0.51	0.51	0.51	0.53	0.59
70.00	0.50	0.52	0.51	0.50	0.52	0.52	0.53	0.54	0.56	0.56
63.00	0.53	0.53	0.52	0.54	0.55	0.55	0.56	0.56	0.58	0.58
56.00	0.55	0.57	0.57	0.58	0.57	0.58	0.58	0.60	0.60	0.60
49.00	0.63	0.60	0.62	0.61	0.60	0.61	0.62	0.64	0.62	0.64
42.00	0.66	0.64	0.66	0.64	0.63	0.64	0.65	0.69	0.69	0.67
35.00	0.68	0.67	0.74	0.73	0.72	0.73	0.74	0.80	0.76	0.80
28.00	0.82	0.82	0.82	0.81	0.80	0.82	0.84	0.90	0.90	0.90
21.00	1.01	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.20
14.00	1.72	1.29	1.33	1.31	1.30	1.31	1.35	1.48	1.51	1.72
	0.99	1.48	1.98	2.48	3.02	3.52	4.02	4.51	5.01	5.50

FIG 4.3 - The fuel map of the Medium engine

VEHICLE 1			VEHICLE 2	
Simulated	Official		Simulated	Official
910	910	Vehicle test weight(Kg)	1360	1360
0.4	0.4	Drag Coefficient	0.455	0.455
1.73	1.73	Frontal Area (m^2)	1.93	1.93
0.012	NA	Coeff. of Rolling Resistance	0.012	NA
0.25	0.25	Wheel Radius (m)	0.32	0.32
20	NA	Inertia as effective Weight (Kg)	28	NA
Spur Gear	Spur Gear	Final Drive Type	Spur Gear	Spur Gear
3.44:1	3.44:1	Ratio	3.72:1	3.72:1
Manual	Manual	Gear box Type	Manual	Manual
3.65:1	3.65:1	Ratios 1st	3.29:1	3.29.1
2.19:1	2.19:1	2nd	2.06:1	2.06:1
1.43:1	1.43:1	3rd	1.38:1	1.38:1
1.00:1	1.00:1	4th	1.00:1	1.00:1
1275	1275	Engine Capacity (cc)	2227	2227
9.4:1	9.4:1	Compression Ratio	9.0:1	9.0:1
40	45	Max. Power (KW)	83	83
5000	5650	Max. Speed (rpm)	5250	5250
0.21	NA	Idle Fuel Cons. (g/s)	0.24	NA
0.22	NA	Engine Inertia (Kgm^2)	0.40	NA

TABLE 4.1 - Design Data used for two vehicle validation

Vehicle 1	Fuel Economy (mpg) ECE-15	56 mph	75 mph	0-60 Acc. (Secs)	Max. Speed (mph)
Simulated	33.0	51.0	37.7	15.2	85.0
Experimental	32.8	51.2	37.9	13.4	90.0

TABLE 4.2 - Comparison of experimental and predicted data for vehicle 1.

Vehicle 2	Fuel Economy (mpg) ECE-15	56 mph	75 mph	0-60 Acc. (Secs)	Max. Speed (mph)
Simulated	20.3	35.1	27.3	12.2	100
Experimental	22.1	34.0	27.0	12.2	105

TABLE 4.3 - Comparison of experimental and predicted data for vehicle 2

Vehicle Parameters			Mini	Small	Medium
Test Weight (Kg)			800	900	1050
Drag Coefficient			0.40	0.38	0.35
Frontal Area (m^2)			1.73	1.80	1.95
Rolling Resistance Coefficient			0.012	0.012	0.012
Wheel Radius (m)			0.25	0.27	0.28
Wheel Inertia as effective weight (kg)			15	20	25
Final Drive	Type		Spur	Spur	Spur
	Ratio		3.65	3.44	3.14
Gear box	Type		Manual	Manual	Manual
	Ratio	1st	3.7	3.6	3.6
		2nd	2.2	2.0	2.0
		3rd	1.4	1.3	1.3
		4th	1.0	1.0	1.0
Engine Max. Power		(kw)	35	40	55
Engine Max. Speed		(rpm)	6000	5000	5500
Idle fuel consumption		(g/s)	0.12	0.14	0.16
Engine Inertia		(kgm^2)	0.15	0.22	0.30
Engine Capacity		(cc)	1000	1300	1600

TABLE 4.4 - Design Parameters of the three car types selected.

CHAPTER FIVE

AREAS FOR FURTHER IMPROVEMENTS IN CONVENTIONAL VEHICLES

5. Areas for Further Improvements in Conventional Vehicles

This chapter investigates the possible fuel savings by improvements in:

1. Vehicle characteristics of drag, mass and rolling resistance;
2. Improvements in engine transmission matching achieved by better gearing and gear shifting;
   (CVTs are discussed in Chapter 6).
3. Engine improvements;
4. Control improvements regarding fuel cut off at idle or overrun conditions.

In addition the effect of changes in other variables affecting fuel economy, such as environmental conditions and variations in engine size, are studied.

5.1 Effect of Variations in Environmental Conditions

The effect of a range of constant headwinds on the 'small' vehicle fuel economy is shown in Fig 5.1. Comparison of these effects on the fuel economy of the different driving cycles shows that the effect of the head wind on fuel consumption (in percentage terms) increases as the average cycle velocity increases. This is due to the increasing dominance of the air drag resistance term in the power expenditure at higher vehicle speeds (as shown in Table 5.1).

Within the normal variations of ambient conditions of pressure and temperature, no significant change in vehicle tractive effort was noted. This cannot be interpreted as no change in vehicle fuel economy, however, as neither the effects of the variations in ambient conditions on engine performance (fuel map) nor other factors such as cold engine operation are studied here.

The effect of gradient ($\pm$) on urban fuel economy of the Small car is shown in Fig 5.2. Although the relationship between gradient and vehicle energy expenditure (EQ 2.8 & 2.11) is linear, the positive gradient increases engine load factor in urban driving and the resultant improvement in engine efficiency reduces the penalty associated with the positive gradient. If urban driving over a hilly terrain is interpreted as driving the urban cycle up a certain gradient and then down the same gradient, this implies that overall fuel economy would improve in relation to that on a level road.

The effect of 2% gradient on steady state cruise fuel consumption is shown in Fig 5.3 which shows the reduction in fuel economy caused by the gradient, reducing as the cruising speed of the vehicle increases. This is due to the increasing share of the air drag component of power expenditure. This may be better illustrated by Fig 5.4 which shows the percentage increase in power expenditure due to 2% gradient reduce as the speed of the vehicle increases.

5.2 Effect of Vehicle Improvements on Fuel Economy

Once the drive power reaches the wheel, energy is absorbed in three ways: rolling resistance losses, air drag losses, and that expanded in acceleration. Table 5.1 shows the apportioning of the energy expenditure for the three vehicles.

5.2.1 Rolling Resistance

Rolling resistance losses account for about 33% of the energy consumed in overall (40%, 50%, 10%) driving. All major tyre manufacturers have been working on reducing tyre resistance. Most work is concentrated on improved tread and carcase design (7). Higher inflation pressures also result in some benefits. These improvements may be restricted, however, by safety requirements. Table 5.2 shows the percentage improvements in fuel economy obtainable by 10% reduction in rolling resistance and the biggest benefits are at lower vehicle speeds. Rolling resistance losses are also proportional to vehicle mass, a reduction of which will reduce these losses.

5.2.2 Vehicle mass

The important effect of weight on energy expenditure is shown in percentage terms in Table 5.1. Therefore great effort has been made by motor manufacturers to reduce vehicle mass by using lighter weight materials. The reduction in vehicle mass evolves not only around the primary savings resulting from direct substitution of a lighter material in place of a heavier one, but also the secondary savings resulting from lighter structural loads carried by the chassis, suspensions and a lighter power train.

Fig 5.5 shows the effect of reductions in mass on fuel economy and in order to study the effect of extra passengers and payload the range shown is extended beyond the test weight of the Small vehicle (900 Kg).

The largest benefits occur in urban driving and as the average speed of the driving cycles and therefore the effect of drag losses increases these improvements become smaller. The improvements in fuel economy obtainable by 10% reduction in mass of the three vehicles studied are shown in Table 5.3. It should be noted that these improvements refer only to weight savings achieved by material substitution applied in existing vehicles (ie no engine change).

When these primary weight savings are utilized, as described earlier further weight savings are to be had. As vehicle acceleration is inversely proportional to weight, the power rating of the new vehicle could be reduced without any loss in the accelerative performance. Fig 5.6 shows the amount to which power rating could be reduced without any loss in accelerative power of the 'small' vehicle. Table 5.4 shows the effect of 10% weight reduction applied in 'all new' vehicles with a lower power rating. The improvement over Table 5.3 results are due to the increased load factor of the smaller engine. It should be noted that the results of Table 5.4 are less accurate because the same engine map has been employed for the engine with the lower power rating. An important factor concerning material engineers' decision on material substitution is the amount of overall fuel savings which could be so achieved. Fig 5.7 shows the effect of weight reductions on both existing and new vehicles. The range considered for new vehicles extends beyond that of the existing vehicles due to the secondary weight savings possible for new vehicles. Recent estimates (46) indicate that an additional 60% to 90% of the primary weight savings is to be had if the planning horizon is sufficiently long-term to allow for complete redesign and manufacturing investment.

5.2.3 Air Drag

Around 50% of overall motoring energy requirements of the vehicles considered here could be attributed to drag losses as shown in Table 5.1. Given that the seating attitude and positions of car occupants will not change, notable reductions in frontal area are not likely. Further reductions in air drag coefficient, however, are achievable by tighter controls on vehicle styling and better airflow management. The effect of change in air drag coefficient on fuel economy is shown in Fig 5.8. This effect increases as the average vehicle speed and therefore the proportion of total energy expanded to overcome air drag increases. A reduction in air drag coefficient can also improve vehicle performance but the improvements in accelerative performance are very small. Table 5.5 shows that a 10% reduction in air drag coefficient leads to around 5% improvement in fuel economy at high cruising speeds.

5.3 Improvements in Engine-Transmission Matching

For best fuel economy, the engine should be geared to run as much of the time as possible at speeds and loading which allow peak thermal efficiency. Figs 5.9 and 5.10 show the effect of final drive ratio selection on vehicle fuel economy and accelerative performance. Clearly, the selection of gearing is a compromise between fuel economy and accelerative performance. For example by 10% reduction in final drive ratio fuel economy improvements of up to 5% are achieved. This is accompanied, however, by a 3% loss in 0-60 WOT acceleration time.

Instead of reducing the final drive ratio, some motor manufacturers have modified their gear box ratios to give top as an overdrive ratio. This improves high speed cruise fuel economy without any loss in low speed acceleration. But as the top gear is usually designed to attain the speed and torque conditions corresponding to maximum power, this modification of the top gear results in maximum power and therefore maximum speed being no longer attainable (Fig 5.11). Fig 5.12 shows the Medium vehicle fuel consumption while cruising in each gear, note the improvement in fuel economy as gearing increases. It also presents the case for the employment of five and six speed gear boxes. The resultant improvement in fuel economy should overcome any consumer resistance arising from loss of comfort associated with multiple gear shifting. In addition the speeds achievable in overdrive gears are well above the legal speed limits.

The urban (ECE 15) driving results presented so far have been obtained by using the standard gear shift speeds of ECE 15 driving cycle. The resultant engine operation map for the Medium vehicle is shown in Fig 5.13. Better fuel economies can be achieved, however, by better management of the gear box and therefore more efficient use of the engine. Table 5.6 shows the optimum gear shift velocities for each vehicle obtained by employment of the 'OPTGEAR' subroutine (Section 2.6.2). Employment of the optimum shift schedule results in the engine operating in more efficient regions (Fig 5.14) and therefore in better fuel economy (8-23%) as shown in Table 5.7. The large variation in the extent of these improvements is due to the varying engine sizes and the fixed gear shift speeds of the ECE-15 cycle. These results, however, refer to an ideal case where by correct usage of the gear box the driver optimizes the fuel consumption rates by approaching the ideal line (CVT operating schedule) as much as possible. But under the varying conditions of traffic the driver's attention is concentrated on exterior events and therefore cannot adequately deal with this additional problem. Optimum gear shifting can be achieved, however, by means of an automatic gear box which shifts gear relative to load and speed (ie electronically controlled).

Fuel economies obtainable by the various transmissions proposed here are compared in Table 5.8. Five speed manual gear boxes are already employed and a progression to a six speed box is possible. The automatic transmissions are assumed to have the same gear ratios as the manual, in order to isolate the efect of different gear ratios on fuel economy. It is also assumed that the automatic transmissions employ a torque convertor with lock-up in each gear. Four and five speed automatic transmissions which shift in relation to load and speed have been developed (16) but a six speed automatic must be considered an unlikely candidate, not least due to the state of development of continuously variable transmissions. The results of Table 5.8 shows that the fuel economies of the automatic vehicle are slightly lower than that which could be achieved by the manual transmission which is due to automatic's lower efficiency.

5.4 Fuel Cut-Off at Idle and Overrun Conditions

Fuel cut-off during idle or overrun conditions has already been achieved in some vehicles (Section 1.5.3) with the problem of engine warm-up and unburnt hydrocarbons solved by control systems which only allow fuel cut-off on a warm engine.

Table 5.9 shows the improvements in ECE 15 urban fuel consumption so achieved for the vehicles studied here. It should be noted that the savings would vary according to the driving cycle employed. For ECE 15 cycle, improvements of around 20% for fuel cut-off at idle and 10% for fuel cut-off at overrun conditions are ahcieved. Fuel savings of about 30%

are obtainable by fuel cut-off at both idle and overrun conditions. It should be noted that these effects are achieved by setting fuel consumption equal to zero at these conditions, and are therefore highly dependent on the value of fuel consumption.

5.5 Effect of Engine Size on Fuel Economy

Some motor manufacturers market their vehicles with a range of engine sizes in order to reduce the range of components used and so minimise manufacturing investment. Table 5.10 shows the effect on fuel economy of employing the three standard engines (with their repsective gear boxes) in the three standard vehicles. The urban fuel economy is the most dependent on engine size. This is because at such light load conditions the increased load factor of the smaller engines means operation in a more efficient region of the engine as shown in Fig 5.15. On the other hand, the larger engine has a greater reserve of power at a particular load condition and, therefore, a better accelerative performance (Fig 5.16).

5.6 Engine Improvements

Fig 5.17 shows the engine map for a modern three cylinder research engine. Comparison with the 'standard' engine (Fig 5.16) shows improvements in specific fuel consumption arising from improved cylinder size and lower friction losses which have also resulted in the relatively large areas of constant efficiency. The engine has also improved ignition at idle (FCI = 0.1 g/s as opposed to 1.2 g/s). Table 5.11 shows the fuel consumption available on employment of this engine in the Mini and Small vehicles. Results for the Medium vehicle are not presented due to the fact that the research engine characteristics are known only in the 30-50 Kw power range.

The estimated improvements in overall fuel economy are 9-13% approximately. These correspond to fuel savings cited in the Literature(13) at 7-15% due to the employment of better conventional engines.

5.7 Conclusions

1. Normal variations in ambient conditions have a negligible effect on vehicle tractive effort.

2. Although gradient and constant head wind have a linear relationship with power expenditure, their effect on fuel economy is non-linear.

3. 10% reduction in coefficient of rolling resistances or drag coefficient can lead to 1-3% improvement in urban fuel economy and 1-2% reduction in overall fuel consumption.

4. 10% reduction in mass leads to 3% reduction in urban fuel consumption and 2% improvement in urban fuel economy.

5. A reduction (10%) in final drive ratio could result in improvements (up to 5%) in fuel economy which is accompanied however by a loss of accelerative performance (up to 3% loss in 0-60 mph WOT acceleration time).

6. Addition of one or two overdrive gears to the four speed box can lead to improvements of up to 8% and 15% respectively.

7. Better gear box management (gear shifting) can lead to improvements of up to 23% in urban fuel economy.

8. By cutting fuel off at idle or at idle and overrun conditions, urban fuel economy improvements of up to 21% and 32% respectively could be achieved.

9. Employment of a more efficient 3-cylinder research engine leads to reductions of up to 15% in overall fuel consumption.

Figure 5.1 - Effect of constant headwind on Fuel Economy of the small car .

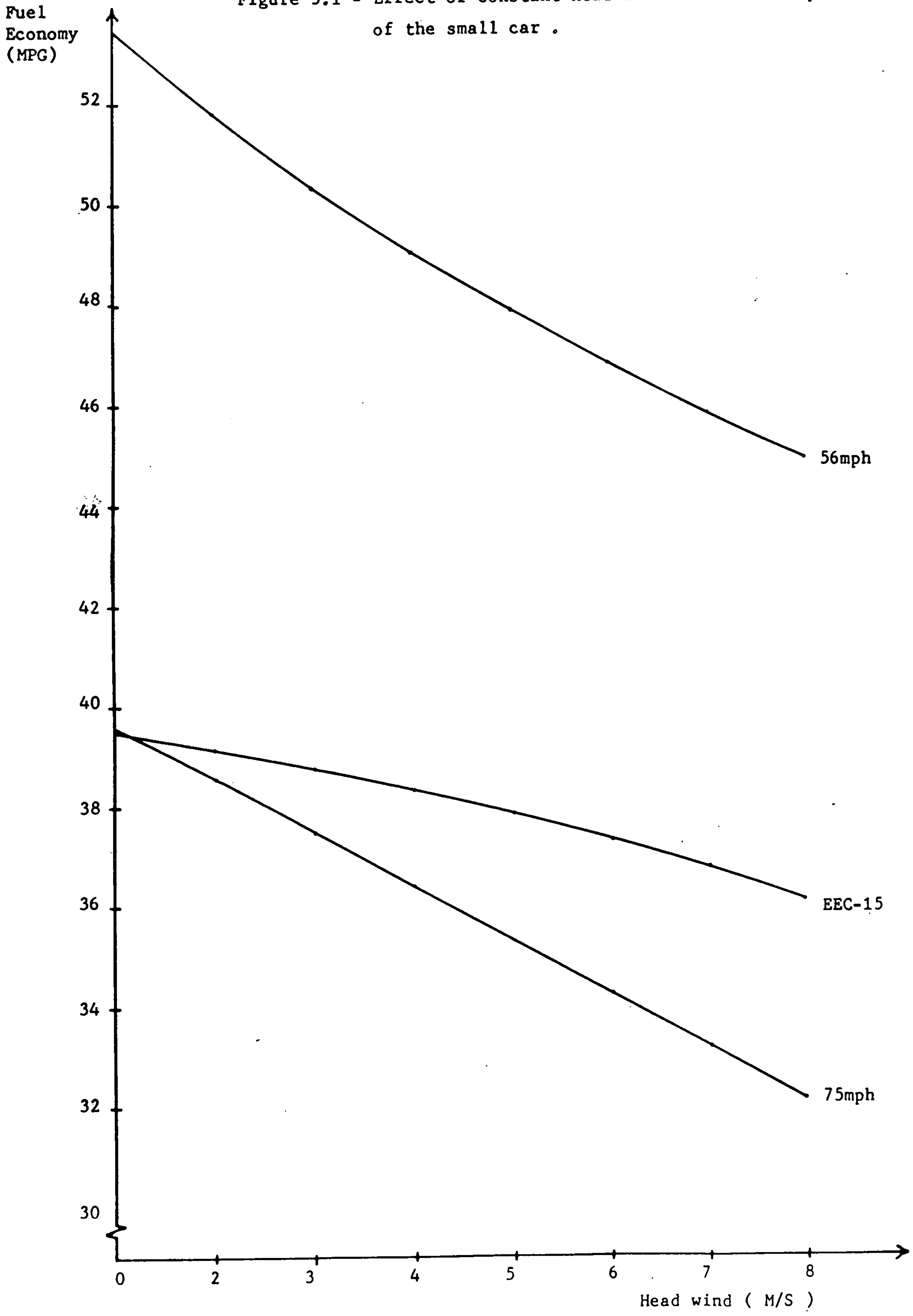


Figure 5.2 - Effect of Gradient on the Small car urban fuel economy .

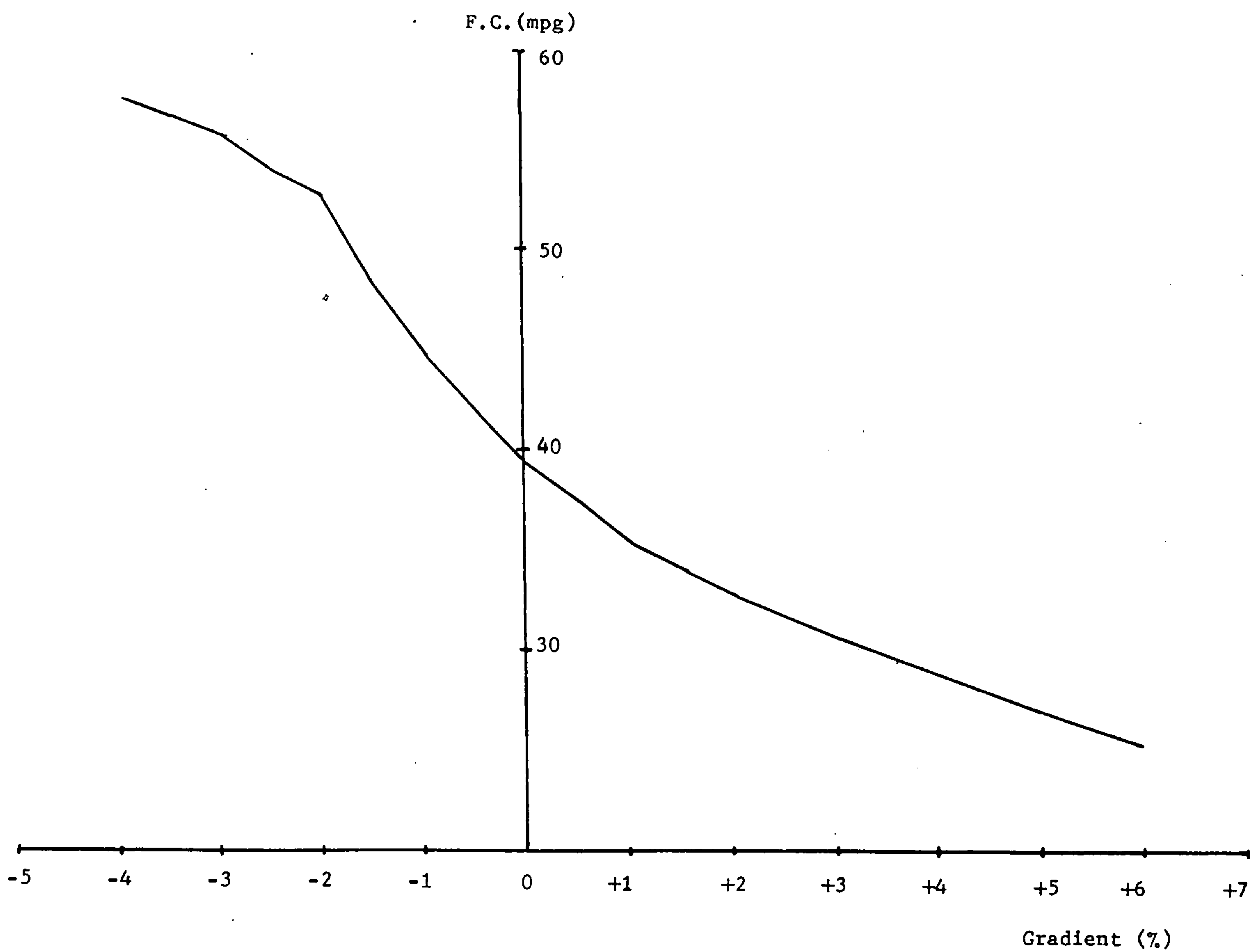


Figure 5.3 - Effect of gradient on steady state cruise fuel economy of Small car .

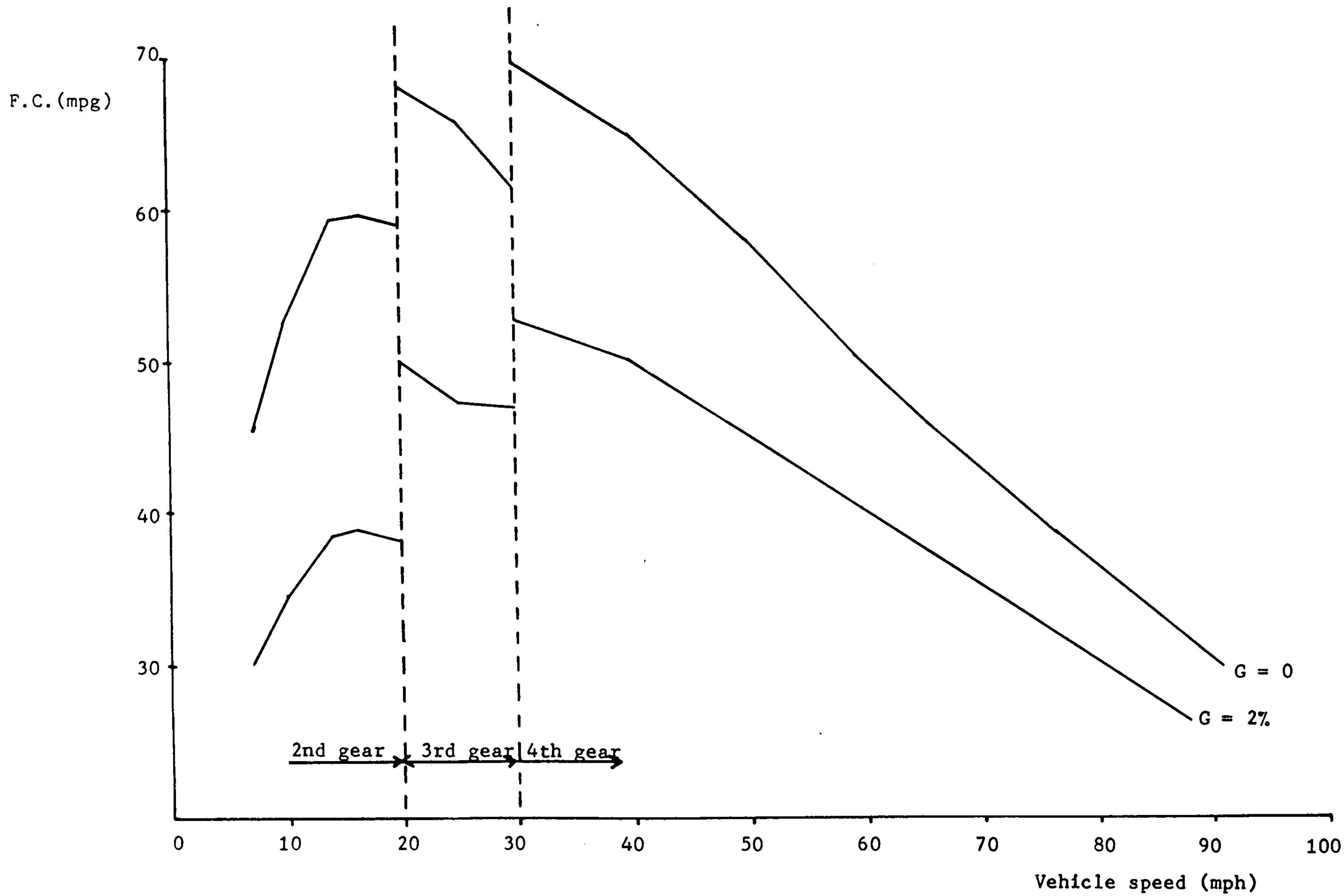


Figure 5.4 - Effect of 2% gradient on the steady state power expenditure of the small car .

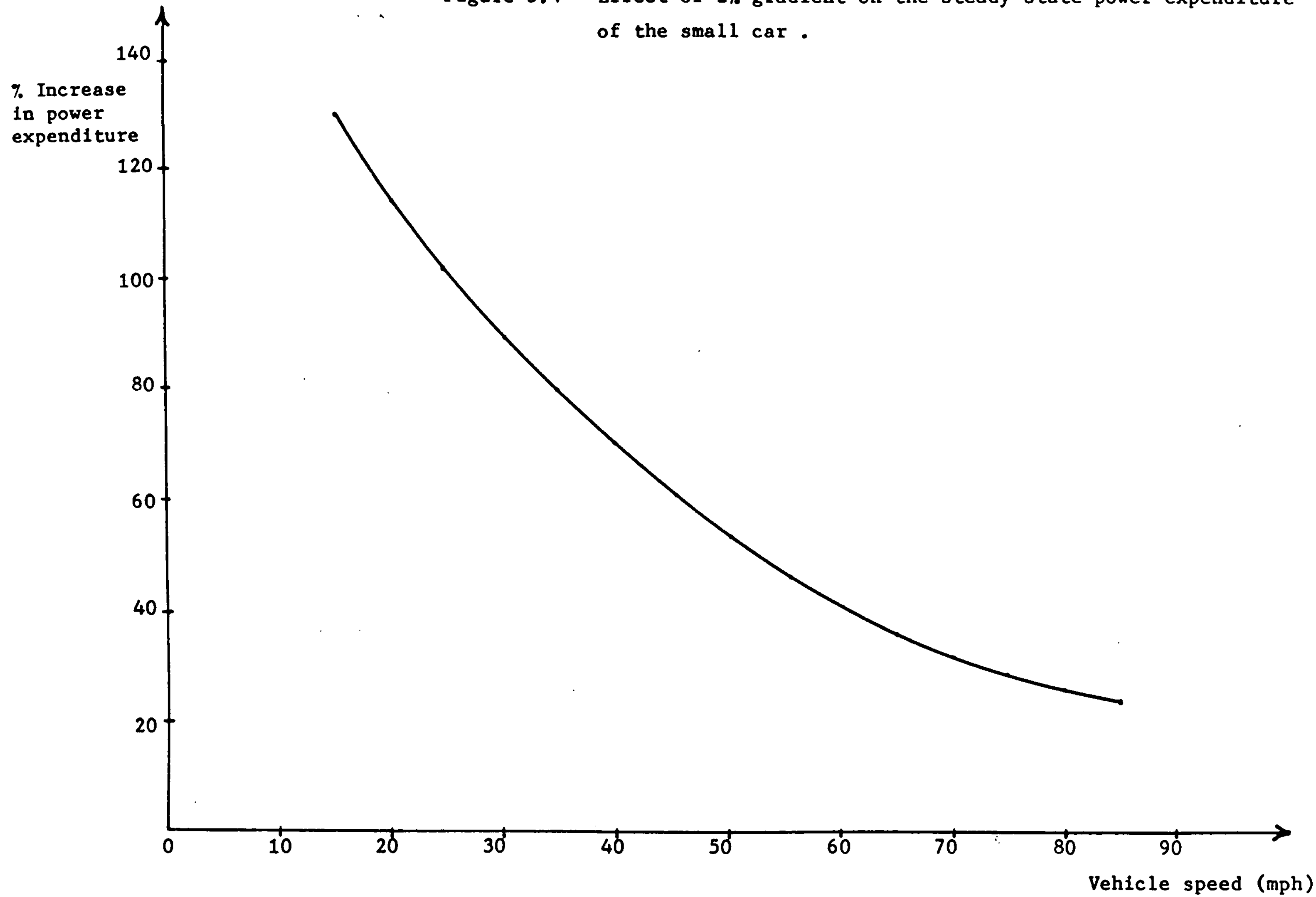


Figure 5.5 - Effect of variation of vehicle mass on fuel economy of Small car .

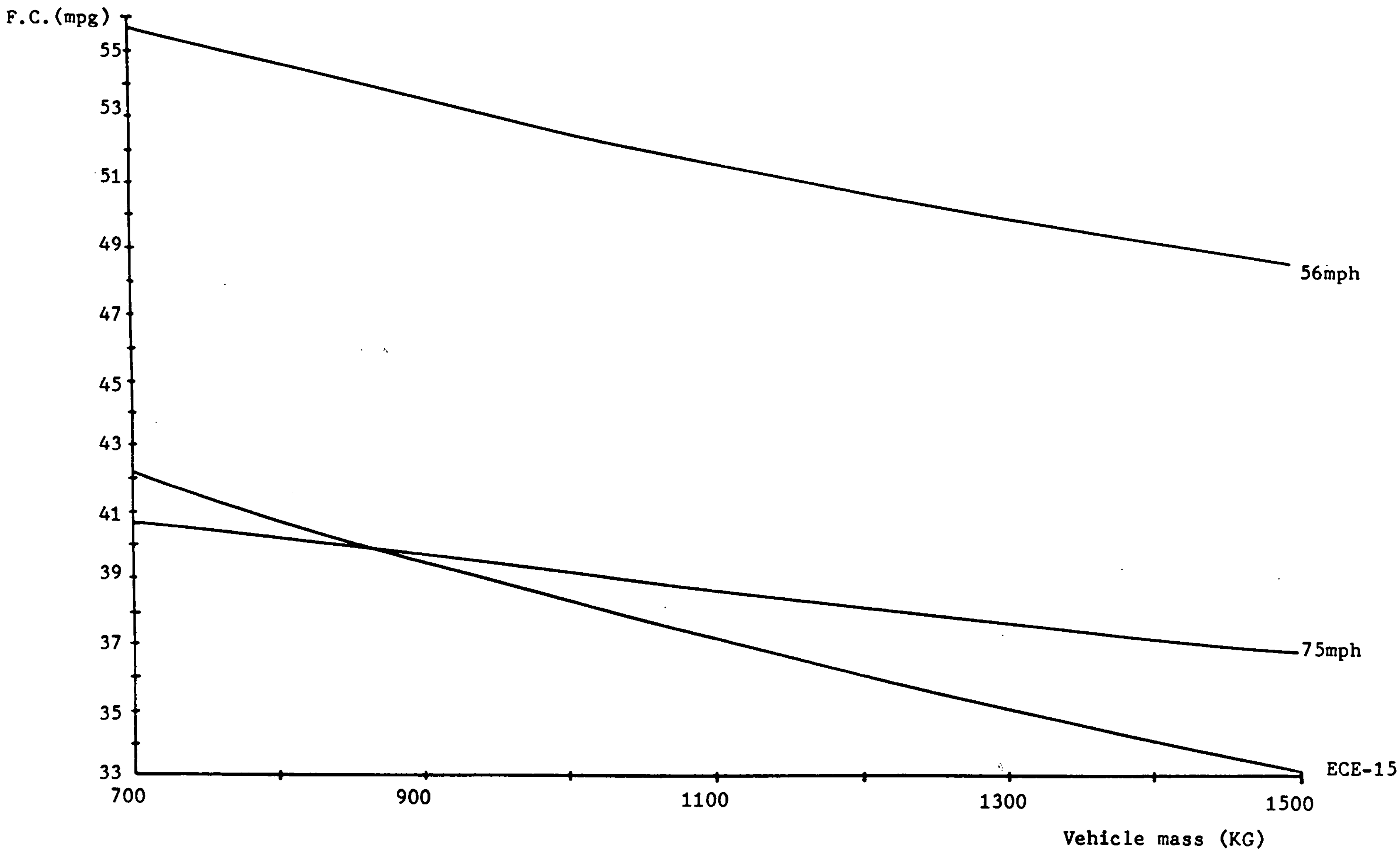


Figure 5.6 - The possible reduction in engine size available (with no loss in acceleration performance) due to weight reduction in the small car .

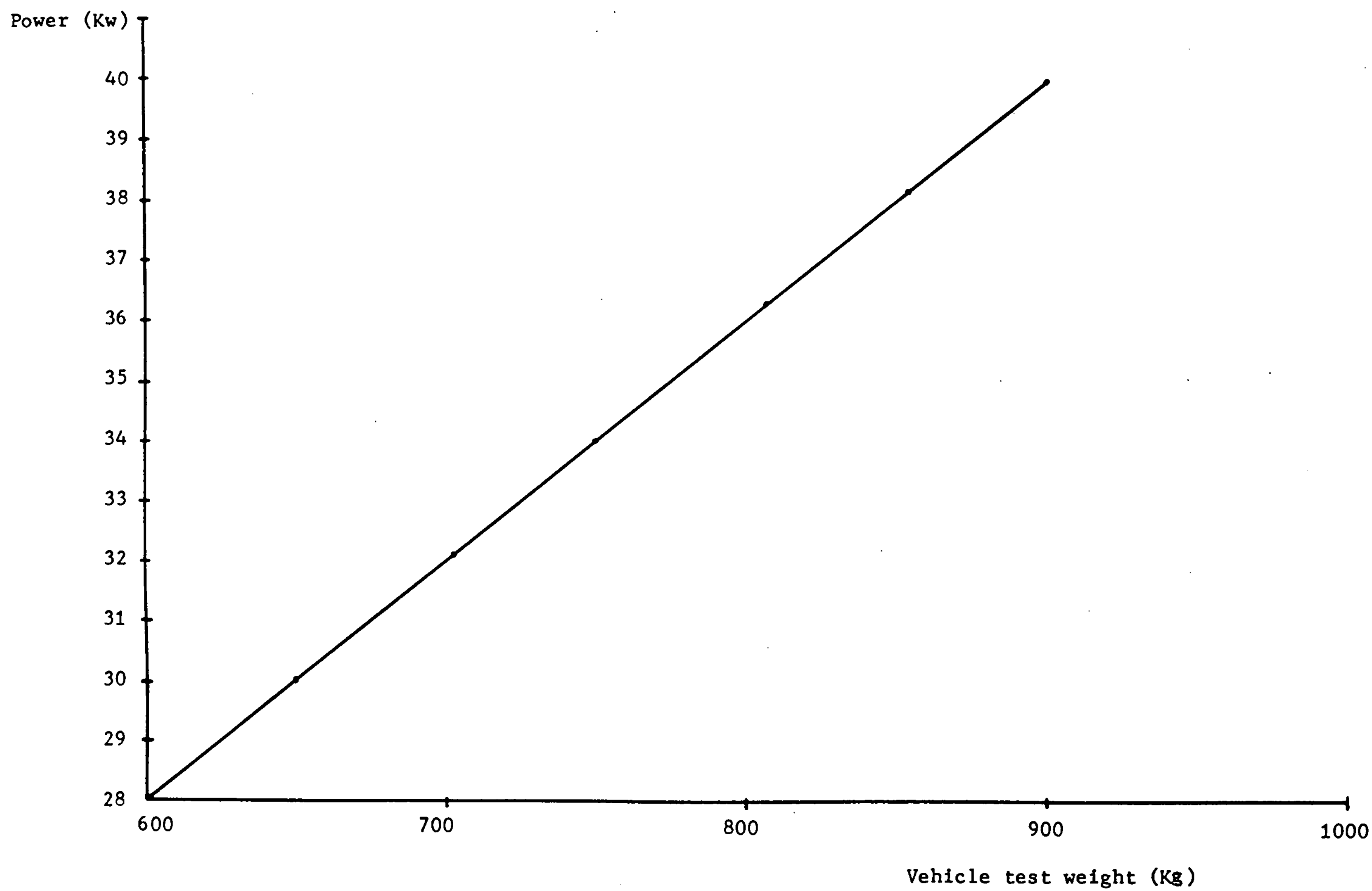


Figure 5.7 - Effect of weight reduction on overall fuel economy of the small car .

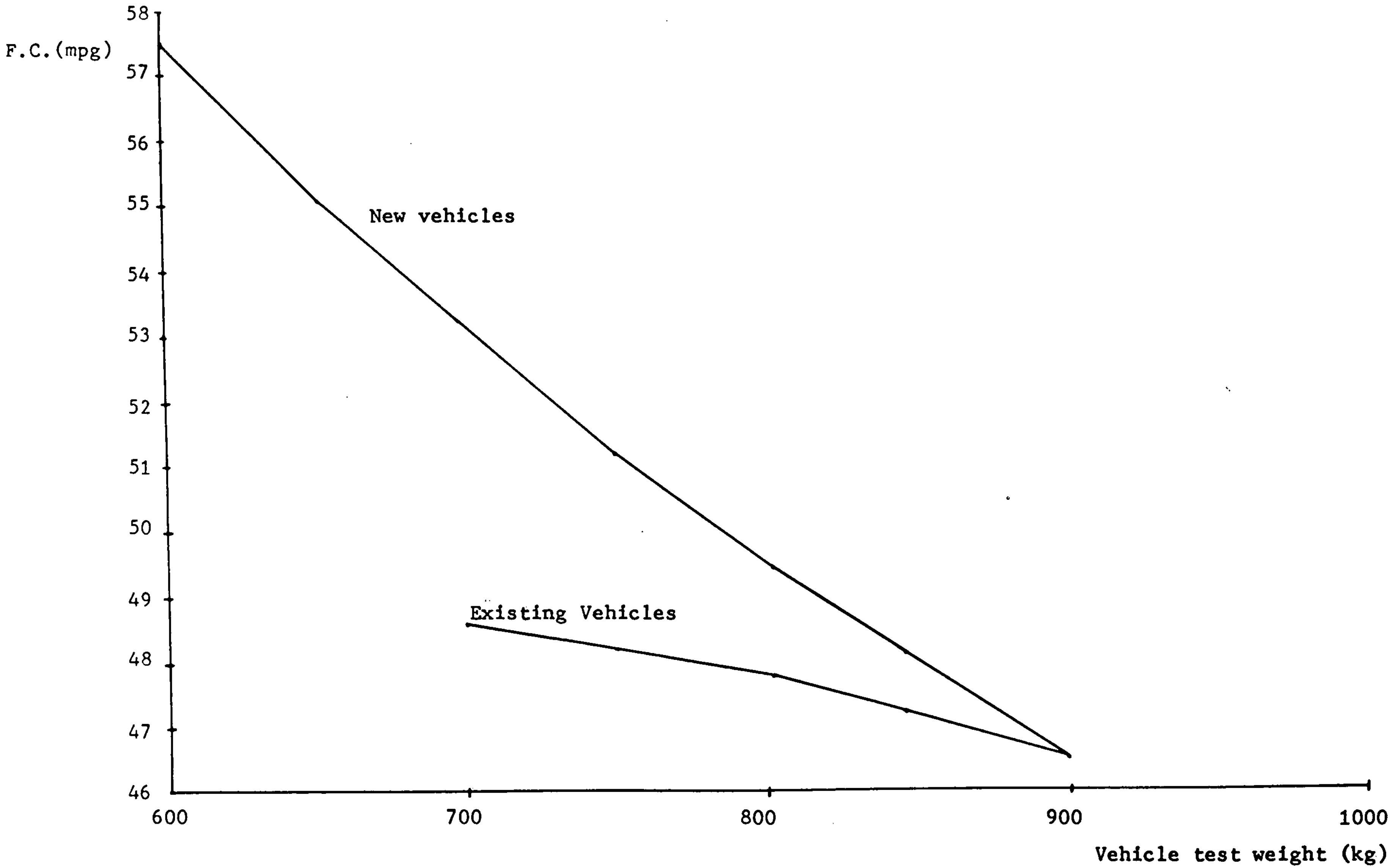


Figure 5.8 - Effect of air drag coefficient on fuel economy of Small car .

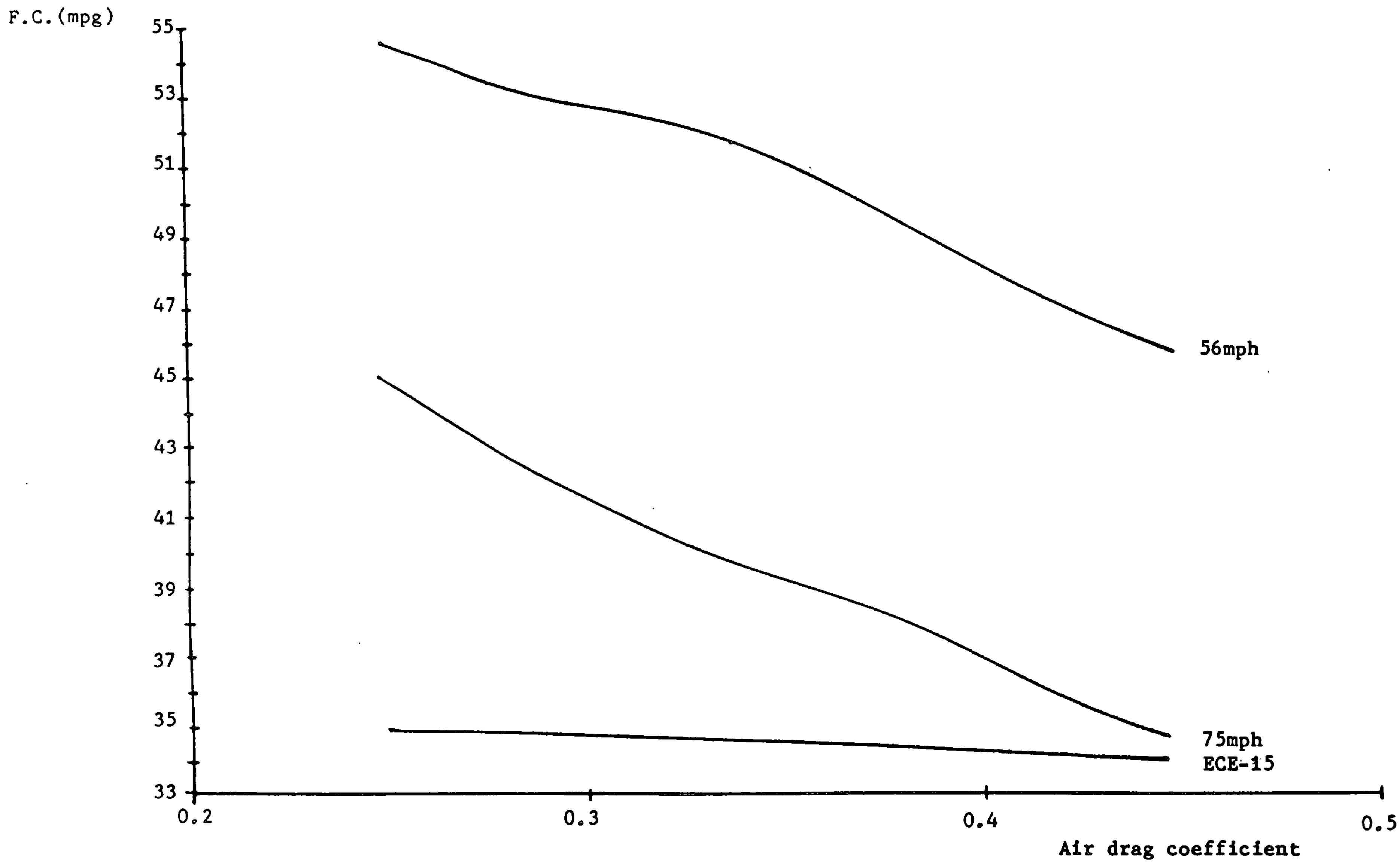


Figure 5.9 - Effect of variation of final drive ratio on the Medium car fuel economy .

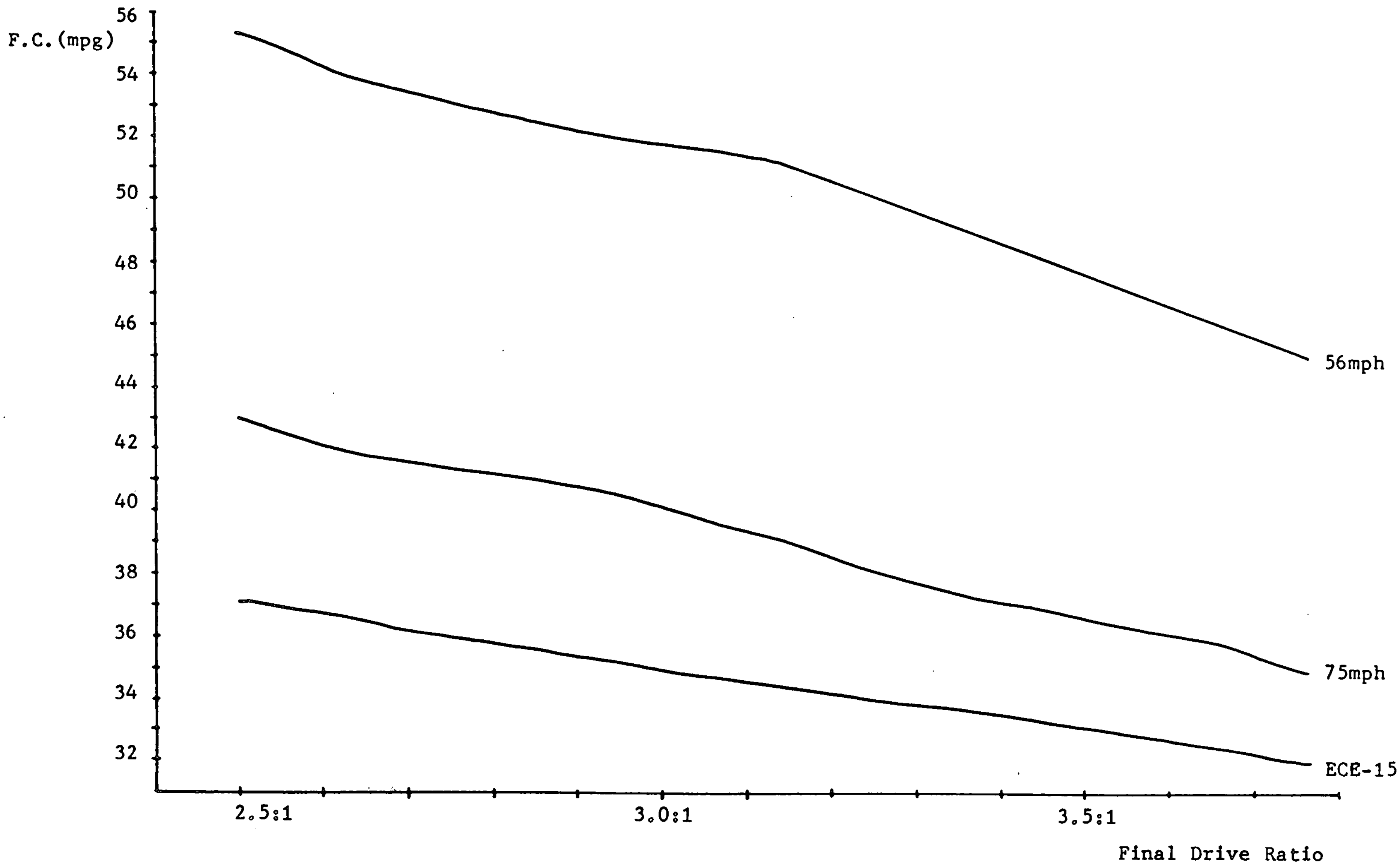


Figure 5.10 - Effect of final drive ratio on 0-60mph W.O.T. acceleration of the Medium car .

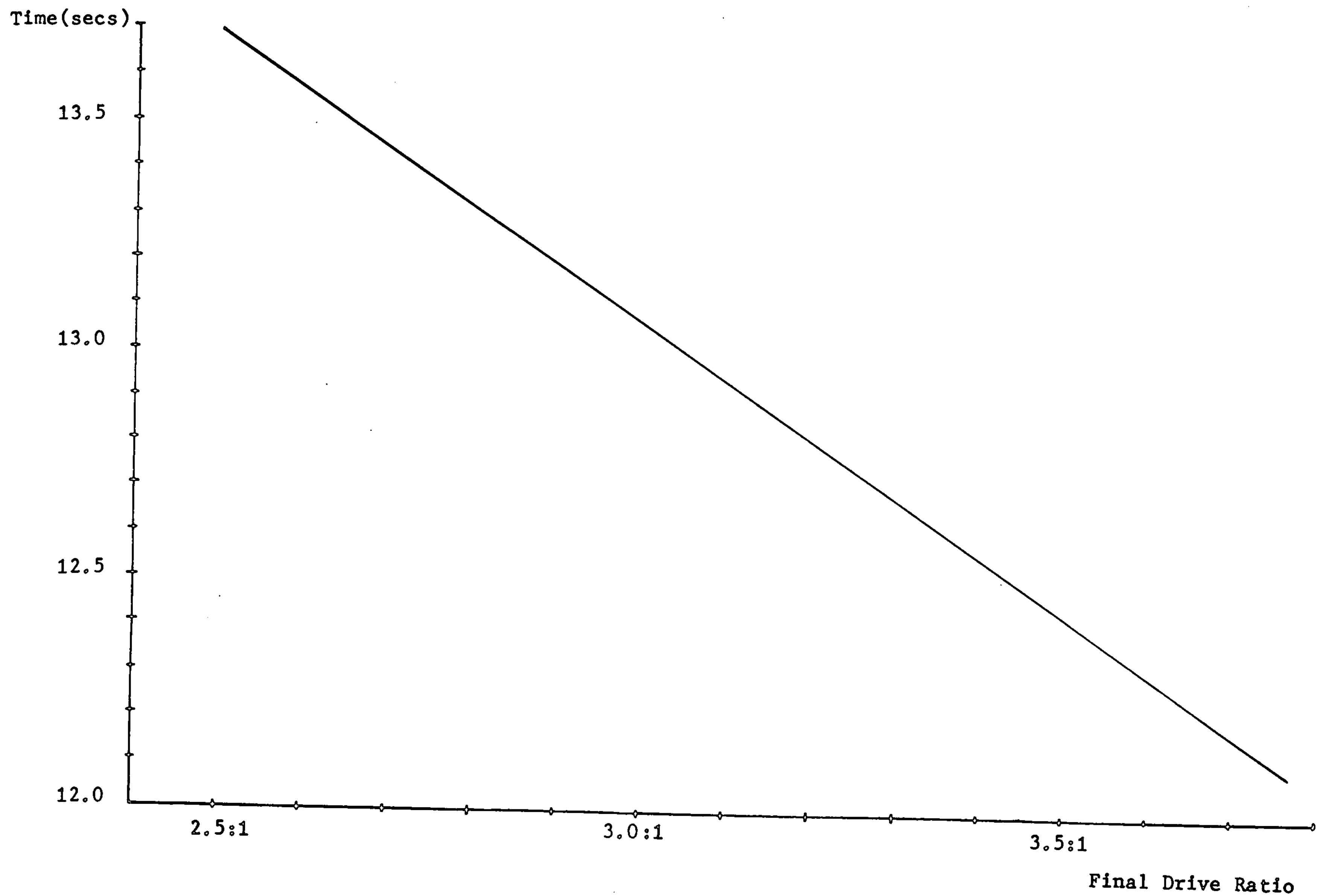


Figure 5.11 - Effect of overall gearing on the maximum speed of the Medium Vehicle .

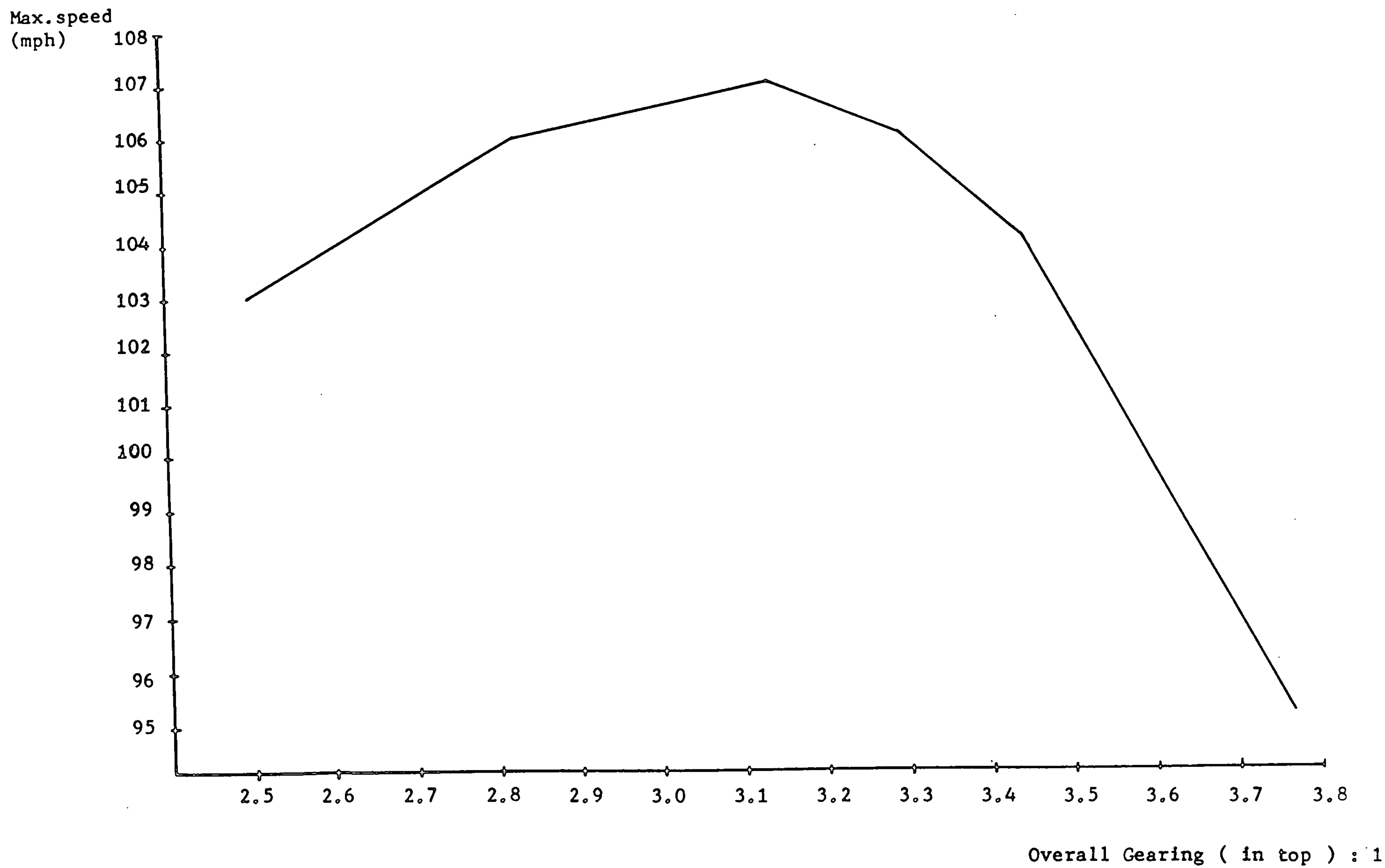
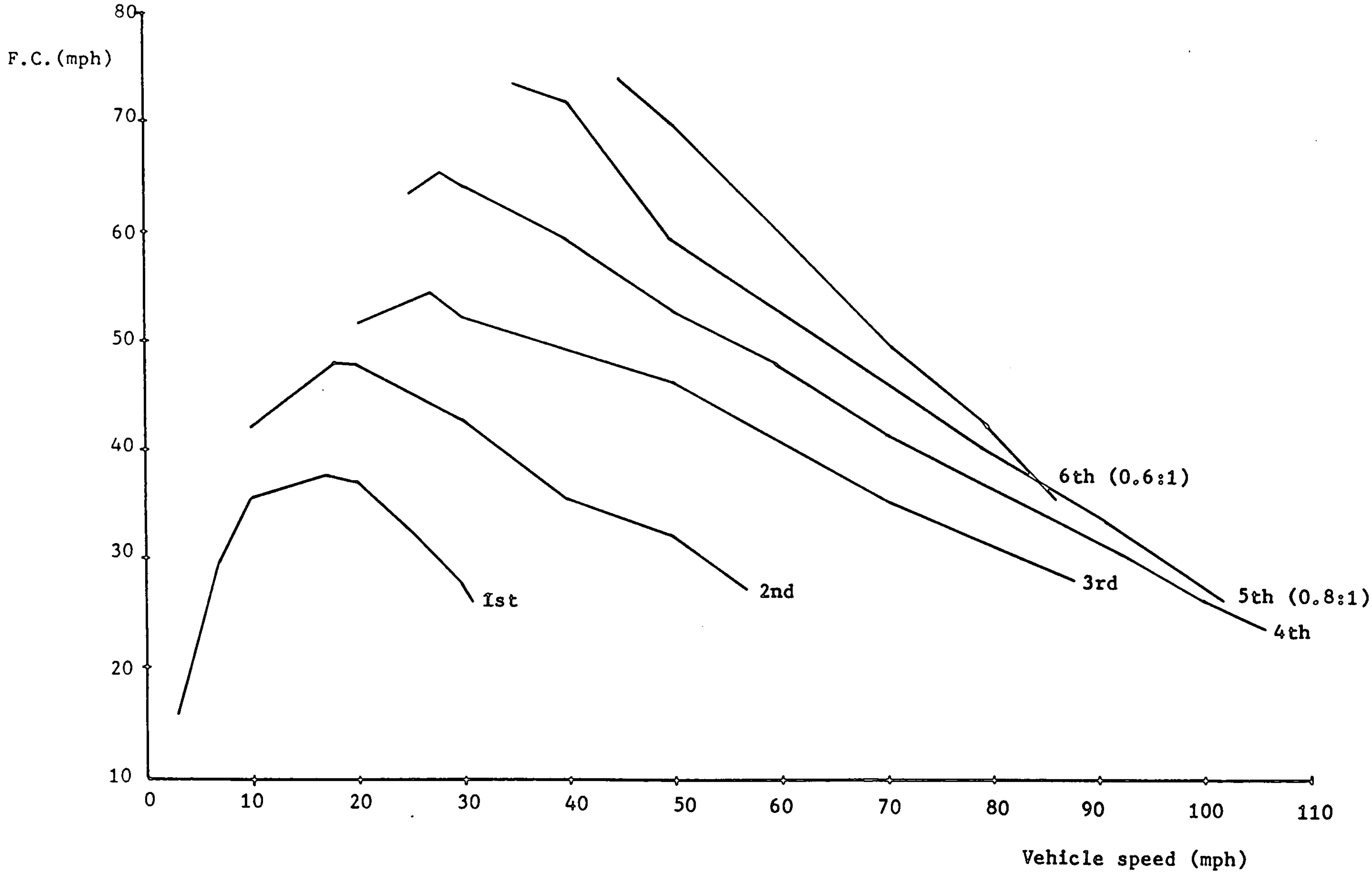


Figure 5.12 - Medium car cruise fuel consumption in each gear .



I. C. ENGINE PERFORMANCE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); SFC-PTS/HP-HR

140.00	0.49	0.49	0.49	0.49	0.47	0.48	0.49	0.49	0.49	0.49
133.00	0.50	0.50	0.50	0.47	0.46	0.47	0.48	0.48	0.48	0.48
126.00	0.55	0.55	0.48	0.44	0.44	0.45	0.46	0.51	0.51	0.51
119.00	0.55	0.52	0.46	0.45	0.45	0.46	0.47	0.49	0.51	0.51
112.00	0.48	0.49	0.47	0.45	0.46	0.46	0.47	0.49	0.50	0.53
105.00	0.42	0.48	0.47	0.46	0.47	0.46	0.47	0.49	0.50	0.53
98.00	0.44	0.47	0.47	0.47	0.47	0.46	0.47	0.50	0.50	0.53
91.00	0.46	0.48	0.47	0.48	0.48	0.48	0.48	0.50	0.51	0.54
84.00	0.50	0.48	0.48	0.49	0.49	0.49	0.49	0.51	0.51	0.57
77.00	0.50	0.50	0.49	0.50	0.51	0.51	0.51	0.51	0.53	0.59
70.00	0.50	0.52	0.51	0.50	0.52	0.52	0.53	0.54	0.56	0.56
63.00	0.53	0.53	0.52	0.54	0.55	0.55	0.56	0.56	0.58	0.58
56.00	0.55	0.57	0.57	0.58	0.57	0.58	0.58	0.60	0.60	0.60
49.00	0.63	0.60	0.62	0.61	0.60	0.61	0.62	0.64	0.62	0.64
42.00	0.66	0.64	0.66	0.64	0.63	0.64	0.65	0.69	0.69	0.67
35.00	0.68	0.67	0.74	0.73	0.72	0.73	0.74	0.80	0.76	0.80
28.00	0.82	0.82	0.82	0.81	0.80	0.82	0.84	0.90	0.90	0.90
21.00	1.01	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.20
14.00	1.72	1.29	1.33	1.31	1.30	1.31	1.35	1.48	1.51	1.72
	0.99	1.48	1.98	2.48	3.02	3.52	4.02	4.51	5.01	5.50

I. C. ENGINE FUEL USAGE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); USAGE - %

140.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
133.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
126.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
119.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
112.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
105.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
98.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
91.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
84.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
77.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
70.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
63.00	0.0	0.0	3.1	1.0	0.0	0.0	0.0	0.0	0.0	0.0
56.00	0.0	1.0	1.5	0.5	0.0	0.0	0.0	0.0	0.0	0.0
49.00	0.0	0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
42.00	0.0	3.1	2.6	1.0	0.0	0.0	0.0	0.0	0.0	0.0
35.00	0.0	1.5	0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0
28.00	0.0	2.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
21.00	0.0	2.6	0.0	6.6	0.0	0.0	0.0	0.0	0.0	0.0
14.00	0.0	6.6	4.1	12.2	0.0	0.0	0.0	0.0	0.0	0.0
	0.99	1.48	1.98	2.48	3.02	3.52	4.02	4.51	5.01	5.50

FIG 5.13 - Engine usage map of the Medium vehicle driven over ECE-15 cycle (standartd gear shifts)

I. C. ENGINE PERFORMANCE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); SFC-PTS/HP-HR

140.00	0.49	0.49	0.49	0.49	0.47	0.48	0.49	0.49	0.49	0.49
133.00	0.50	0.50	0.50	0.47	0.46	0.47	0.48	0.48	0.48	0.48
126.00	0.55	0.55	0.48	0.44	0.44	0.45	0.46	0.51	0.51	0.51
119.00	0.55	0.52	0.46	0.45	0.45	0.46	0.47	0.49	0.51	0.51
112.00	0.48	0.49	0.47	0.45	0.46	0.46	0.47	0.49	0.50	0.53
105.00	0.42	0.48	0.47	0.46	0.47	0.46	0.47	0.49	0.50	0.53
98.00	0.44	0.47	0.47	0.47	0.47	0.46	0.47	0.50	0.50	0.53
91.00	0.46	0.48	0.47	0.48	0.48	0.48	0.48	0.50	0.51	0.54
84.00	0.50	0.48	0.48	0.49	0.49	0.49	0.49	0.51	0.51	0.57
77.00	0.50	0.50	0.49	0.50	0.51	0.51	0.51	0.51	0.53	0.59
70.00	0.50	0.52	0.51	0.50	0.52	0.52	0.53	0.54	0.56	0.56
63.00	0.53	0.53	0.52	0.54	0.55	0.55	0.56	0.56	0.58	0.58
56.00	0.55	0.57	0.57	0.58	0.57	0.58	0.58	0.60	0.60	0.60
49.00	0.63	0.60	0.62	0.61	0.60	0.61	0.62	0.64	0.62	0.64
42.00	0.66	0.64	0.66	0.64	0.63	0.64	0.65	0.69	0.69	0.67
35.00	0.68	0.67	0.74	0.73	0.72	0.73	0.74	0.80	0.76	0.80
28.00	0.82	0.82	0.82	0.81	0.80	0.82	0.84	0.90	0.90	0.90
21.00	1.01	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.20
14.00	1.72	1.29	1.33	1.31	1.30	1.31	1.35	1.48	1.51	1.72
	0.99	1.48	1.98	2.48	3.02	3.52	4.02	4.51	5.01	5.50

I. C. ENGINE FUEL USAGE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); USAGE - %

140.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
133.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
126.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
119.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
112.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
105.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
98.00	0.0	0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
91.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
84.00	0.0	0.5	1.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
77.00	0.0	4.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
70.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
63.00	0.0	1.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
56.00	0.0	1.5	0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0
49.00	0.0	0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
42.00	0.0	3.1	0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0
35.00	0.0	1.5	0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0
28.00	0.0	2.6	0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0
21.00	0.0	21.4	6.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0
14.00	0.0	0.0	4.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	0.99	1.48	1.98	2.48	3.02	3.52	4.02	4.51	5.01	5.50

FIG 5.14 - Engine usage map of the Medium vehicle driven over ECE-15 cycle with optimum gear shifts.

I. C. ENGINE PERFORMANCE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); SFC-PTS/HP-HR

140.00	0.53	0.53	0.53	0.53	0.53	0.53	0.51	0.51	0.51	0.51	0.51	0.51
130.00	0.51	0.51	0.51	0.51	0.49	0.49	0.49	0.50	0.50	0.50	0.50	0.50
120.00	0.57	0.57	0.50	0.48	0.48	0.48	0.49	0.49	0.50	0.53	0.53	0.53
110.00	0.57	0.54	0.49	0.48	0.48	0.48	0.49	0.50	0.50	0.52	0.55	0.55
100.00	0.56	0.53	0.50	0.49	0.48	0.49	0.50	0.51	0.51	0.52	0.53	0.58
90.00	0.56	0.54	0.51	0.50	0.49	0.49	0.50	0.52	0.53	0.53	0.54	0.55
80.00	0.57	0.55	0.53	0.51	0.50	0.50	0.51	0.53	0.54	0.55	0.56	0.57
70.00	0.57	0.56	0.54	0.53	0.53	0.52	0.53	0.55	0.57	0.58	0.59	0.60
60.00	0.59	0.58	0.56	0.55	0.55	0.55	0.56	0.58	0.60	0.63	0.63	0.65
50.00	0.63	0.60	0.60	0.59	0.58	0.59	0.60	0.63	0.65	0.68	0.70	0.70
40.00	0.70	0.68	0.65	0.65	0.65	0.65	0.65	0.68	0.70	0.75	0.80	0.80
30.00	0.80	0.80	0.75	0.75	0.75	0.75	0.80	0.80	0.85	0.90	0.95	1.00
20.00	1.00	1.00	1.00	0.95	0.95	1.00	1.00	1.05	1.10	1.20	1.30	1.30
10.00	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.70	1.70	1.70
	0.75	1.00	1.50	2.00	2.50	3.00	3.50	4.00	4.50	5.00	5.50	6.00

I. C. ENGINE FUEL USAGE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); SFC-PTS/HP-HR

140.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
130.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
120.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
110.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
100.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
90.00.	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
80.00	0.0	0.0	0.0	3.1	0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0
70.00	0.0	0.0	0.5	1.5	0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0
60.00	0.0	0.0	0.5	1.5	1.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0
50.00	0.0	0.0	4.1	1.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
40.00	0.0	0.0	2.6	1.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
30.00	0.0	0.0	2.6	0.0	6.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0
20.00	0.0	0.0	0.0	7.1	12.2	0.0	0.0	0.0	0.0	0.0	0.0	0.0
10.00	0.0	0.0	0.0	3.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	0.75	1.00	1.50	2.00	2.50	3.00	3.50	4.00	4.50	5.00	5.50	6.00

PERCENTAGE TIME AT ENGINE IDLE/OFF = 49.0
PERCENTAGE TIME COMP. BRAKING = 13.8

I C ENGINE FUEL MAP USED-11

FIG 5.15 Mini engine (35kw) usage map by the Small car driven over ECE-15 cycle (standard shifts)

I. C. ENGINE PERFORMANCE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); SFC-PTS/HP-HR

140.00	0.49	0.49	0.49	0.49	0.47	0.48	0.49	0.49	0.49	0.49
133.00	0.50	0.50	0.50	0.47	0.46	0.47	0.48	0.48	0.48	0.48
126.00	0.55	0.55	0.48	0.44	0.44	0.45	0.46	0.51	0.51	0.51
119.00	0.55	0.52	0.46	0.45	0.45	0.46	0.47	0.49	0.51	0.51
112.00	0.48	0.49	0.47	0.45	0.46	0.46	0.47	0.49	0.50	0.53
105.00	0.42	0.48	0.47	0.46	0.47	0.46	0.47	0.49	0.50	0.53
98.00	0.44	0.47	0.47	0.47	0.47	0.46	0.47	0.50	0.50	0.53
91.00	0.46	0.48	0.47	0.48	0.48	0.48	0.48	0.50	0.51	0.54
84.00	0.50	0.48	0.48	0.49	0.49	0.49	0.49	0.51	0.51	0.57
77.00	0.50	0.50	0.49	0.50	0.51	0.51	0.51	0.51	0.53	0.59
70.00	0.50	0.52	0.51	0.50	0.52	0.52	0.53	0.54	0.56	0.56
63.00	0.53	0.53	0.52	0.54	0.55	0.55	0.56	0.56	0.58	0.58
56.00	0.55	0.57	0.57	0.58	0.57	0.58	0.58	0.60	0.60	0.60
49.00	0.63	0.60	0.62	0.61	0.60	0.61	0.62	0.64	0.62	0.64
42.00	0.66	0.64	0.66	0.64	0.63	0.64	0.65	0.69	0.69	0.67
35.00	0.68	0.67	0.74	0.73	0.72	0.73	0.74	0.80	0.76	0.80
28.00	0.82	0.82	0.82	0.81	0.80	0.82	0.84	0.90	0.90	0.90
21.00	1.01	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.20
14.00	1.72	1.29	1.33	1.31	1.30	1.31	1.35	1.48	1.51	1.72
	0.99	1.48	1.98	2.48	3.02	3.52	4.02	4.51	5.01	5.50

I. C. ENGINE FUEL USAGE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); USAGE - %

140.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
133.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
126.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
119.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
112.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
105.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
98.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
91.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
84.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
77.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
70.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
63.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
56.00	0.0	0.0	2.0	1.5	0.0	0.0	0.0	0.0	0.0	0.0
49.00	0.0	0.5	2.6	0.5	0.0	0.0	0.0	0.0	0.0	0.0
42.00	0.0	0.5	0.0	1.5	0.0	0.0	0.0	0.0	0.0	0.0
35.00	0.0	3.6	2.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0
28.00	0.0	1.5	1.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
21.00	0.0	3.6	0.0	6.6	0.0	0.0	0.0	0.0	0.0	0.0
14.00	0.0	0.0	10.7	12.2	0.0	0.0	0.0	0.0	0.0	0.0
	0.99	1.48	1.98	2.48	3.02	3.52	4.02	4.51	5.01	5.50

PERCENTAGE TIME AT ENGINE IDLE/OFF = 49.0
PERCENTAGE TIME COMP. BRAKING = 13.8

FIG 5.16 - Medium engine (55kw) usage map by the Small car driven over ECE-15 cycle (standard shifts).

I. C. ENGINE CHARACTERISTICS

I. C. ENGINE PERFORMANCE LIMIT CURVE

SPEED (RPM)	TORQUE (NM)	BMEP (PSI)	POWER (KW)
500.00	57.21	90.00	3.00
750.00	63.57	100.00	4.99
1000.00	69.93	110.00	7.32
1500.00	76.28	120.00	11.98
2000.00	82.64	130.00	17.30
2500.00	89.00	140.00	23.30
3000.00	89.00	140.00	27.95
3500.00	89.00	140.00	32.61
4000.00	89.00	140.00	37.27
4500.00	82.64	130.00	38.94
5000.00	76.28	120.00	39.93

I. C. ENGINE CVT OPERATING CURVE

SPEED (RPM)	BMEP (PSI)	POWER (KW)
10000.00	0.00	0.00
1000.00	80.00	5.33
1500.00	100.00	10.00
2000.00	110.00	14.67
3000.00	120.00	24.00
4500.00	130.00	39.00
5000.00	120.00	40.00

I. C. ENGINE PERFORMANCE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); SFC-PTS/HP-HR

140.00	0.00	0.00	0.00	0.46	0.44	0.44	0.44	0.44	0.43	0.42	0.44
130.00	0.00	0.00	0.47	0.46	0.44	0.44	0.43	0.43	0.42	0.42	0.44
120.00	0.00	0.49	0.47	0.46	0.44	0.44	0.42	0.42	0.42	0.42	0.44
110.00	0.55	0.49	0.47	0.46	0.44	0.44	0.43	0.43	0.43	0.43	0.44
100.00	0.55	0.49	0.48	0.46	0.44	0.44	0.44	0.44	0.44	0.44	0.45
90.00	0.55	0.49	0.49	0.46	0.45	0.45	-.45	0.45	0.45	0.45	0.45
80.00	0.55	0.51	0.50	0.48	0.47	0.46	0.46	0.46	0.46	0.46	0.47
70.00	0.57	0.55	0.53	0.50	0.49	0.48	0.47	0.47	0.47	0.47	0.48
60.00	0.60	0.57	0.56	0.54	0.52	0.51	0.50	0.49	0.49	0.49	0.52
50.00	0.70	0.60	0.59	0.58	0.56	0.55	0.54	0.54	0.54	0.55	0.58
40.00	0.80	0.72	0.67	0.62	0.60	0.59	0.59	0.59	0.59	0.64	0.74
30.00	1.07	0.88	0.78	0.73	0.71	0.72	0.73	0.75	0.77	0.80	0.80
20.00	1.37	1.18	1.00	0.92	0.90	0.95	1.00	0.98	0.98	0.98	0.98
10.00	1.97	1.80	1.50	1.30	1.30	1.42	1.42	1.42	1.42	1.42	1.42
	0.50	0.75	1.00	1.50	2.00	2.50	3.00	3.50	4.00	4.50	5.00

FIG 5.17 - The performance map of a 3-cylinder research engine.

Vehicle	Driving Cycle	Energy Expenditure (%) due to		
		Acceleration	Rolling Resistance	Air Drag
Mini	ECE-15	51.6	32.8	15.6
	56 mph	--	26.2	73.8
	75 mph	--	16.5	83.5
	Overall*	20.6	27.9	51.5
Small	ECE-15	52.7	33.4	13.9
	56 mph	--	28.7	71.3
	75 mph	--	18.4	81.6
	Overall	21.1	29.5	49.4
Medium	ECE-15	53.8	34.0	12.2
	56 mph	--	32.0	68.0
	75 mph	--	20.8	79.2
	Overall	21.5	31.7	46.8

* Overall (ECE-15 40%, 56 mph 50%, 75 mph 10%)

TABLE 5.1 - Apportioning of energy expenditure of the three standard vehicles.

Vehicle	ECE-15	56 mph	75 mph	Overall
Mini	1.3	1.6	0.9	1.4
Small	1.4	1.9	1.3	1.6
Medium	3.3	1.5	1.6	2.2

TABLE 5.2 - Improvement in fuel economy (%) obtainable by 10% reduction in rolling resistance coefficient.

Vehicle	ECE-15	56 mph	75 mph	Overall
Mini	2.6	1.6	0.7	1.9
Small	2.8	1.9	1.3	2.2
Medium	2.6	1.4	1.0	1.8

TABLE 5.3 - Effect of 10% reduction in weight on 'existing' vehicles fuel economy (%)

Vehicle	ECE-15	56 mph	75 mph
Mini	7	5	4
Small	7	5	4
Medium	7	5	6

TABLE 5.4 - Effect of 10% weight reduction on fuel economy (%) when applied in 'all new' vehicles.

Vehicle	ECE-15	56 mph	75 mph
Mini	1.3	1.6	0.9
Small	1.4	1.9	1.3
Medium	3.3	1.5	1.6

TABLE 5.5 - Improvement in fuel economy (%) obtainable by 10% reduction in Drag Coefficient.

Gear Shifts	Mini	Small	Medium	ECE-15 Standard Shifts
1st to 2nd	7.4	9.3	10.5	10.7
2nd to 3rd	11.5	14.2	16.1	21.7
3rd to 4th	16.0	18.4	20.9	-

TABLE 5.6 - Optimum and standard (ECE-15) gear shift velocities (mph) for the three vehicle studies.

Vehicle	Standard (ECE-15)		Optimum		% improvement in f.c.
	f.c. (mpg)	engine efficiency (%)	f.c. (mpg)	engine efficiency (%)	
Mini	41.8	12.5	51.3	14.0	22.7
Small	39.5	12.7	42.9	13.1	8.6
Medium	34.5	12.6	37.1	13.1	7.5

TABLE 5.7 - Urban fuel consumption and engine efficiency achieved by standard and optimum gear shifts.

Fuel economy (mpg) using ECE-15 gear shifts				Transmission Type		Fuel economy (mpg) using optimum gear shifts			
ECE15	56 mph cruise	75 mph cruise	Over -all f.c			ECE15	56 mph cruise	75 mph cruise	Over -all f.c
34.5	51.2	39.1	43.3	Manual	4-speed cruise in 1:1	37.1	51.2	39.1	44.4
	54.9	42.5	45.5		5-speed cruise in 0.8:1		54.9	42.5	46.5
	63.1	45.5	49.9		6-speed cruise in 0.6:1		63.1	45.5	50.9
34.1	50.4	38.7	42.7	Automatic	4-speed cruise in 1:1	36.8	50.4	38.7	43.8
	54.6	42.3	45.2		5-speed cruise in 0.8:1		54.6	42.3	46.3
	62.7	45.1	49.5		6-speed cruise in 0.6:1		62.7	45.1	50.6

TABLE 5.8 - Fuel economy, obtainable by various stepped transmission designs for the Medium car.

Vehicle	Normal Oper- -ation mpg	Fuel cut off at idle		Fuel cut off at idle and overrun	
		f.c.	% improvement	f.c.	% improvement
Mini	41.8	49.4	18.2	53.8	28.7
Small	39.5	47.4	20.0	52.1	31.9
Medium	34.5	41.7	20.8	45.3	31.3

TABLE 5.9 - Effect of fuel cut-off at idle and overrun conditions on urban fuel consumption. (All urban results refer to EEC-15 driving cycle and its standard gear-shifts)

Vehicle	Driving Cycle	Standard engine employed:		
		Mini (35 kw)	Small (40 kw)	Medium (55 kw)
Mini	EEC-15	41.8	37.6	32.1
	+ Idle	49.4	44.7	37.8
	+ Idle & overrun	53.4	48.8	41.2
	56 mph	55.5	50.2	43.5
	75 mph	38.7	36.8	34.0
Small	EEC-15	44.1	39.5	35.5
	+ Idle	53.1	47.4	41.1
	+ Idle & overrun	57.7	52.1	44.6
	56 mph	59.1	53.5	47.6
	75 mph	41.7	39.7	37.0
Medium	ECE-15	43.6	39.1	34.5
	+ Idle	52.4	46.8	41.7
	+ Idle & overrun	56.8	51.4	45.3
	56 mph	60.4	55.2	51.2
	75 mph	42.2	41.4	39.1

Table 5.10 - Effect of variations in engine size on fuel consumption.

Vehicle		Urban Driving (ECE-15) Normal	Idle*	Idle+**	Steady state criuse 56 mph	75 mph
Mini	Fuel Consumption (mpg)	46.1	53.0	57.3	61.6	45.9
	improvement in f.c. (%)	10.2	7.3	7.3	11.0	18.6
Small	Fuel Consumption (mpg)	45.5	53.1	58.1	61.5	46.4
	improvement in f.c. (%)	15.2	12.0	11.5	15.0	16.9

* Idle - Fuel cut off at idle

** Idle+ - Fuel cut off at idle and overrun

TABLE 5.11 - Fuel Consumption (mpg) available on employment of the research engine and the improvements (%) achieved over fuel economy of existing engines.

C H A P T E R   S I X

EVALUATION OF CVTs FOR PASSENGER CARS

6 Evaluation of CVTs For Passenger Cars

This chapter is to investigate whether presently available CVTs can improve the vehicle fuel economy by restricting the engine operation along the locus of its minimum BSFC points.

The possibility of reducing the vehicle power rating without a loss in accelerative performance is also investigated. This arises from CVT's ability to enable the engine to run at peak power rather than fluctuating with road speed as do the conventional vehicles.

6.1 CVT Vehicle Design and Performance (With Same Power Rating)

The assumptions made here are that all the transmissions to be tested have the same weight, occupy the same space as the manual gear box and that the CVT vehicle requires the same power as the conventional vehicle. The CVTs will replace the manual gear box, therefore, without any change in other vehicle data except the final drive ratio and the addition of an initial reduction ratio when necessary (ie for Transmatic).

6.1.1 CVT Vehicle Design Considerations

In order to enable the engine to operate along its optimum operating curve at every condition of load and speed, the CVT should be able to continuously provide gear ratios between the high gearing required for steady state cruising and the low gearing required for acceleration.

Table 6.1 shows the highest gearing required for each car to ensure that the engine operating schedule is met at every steady state cruise condition. These values are dependent on CVT efficiency which for this calculation was assumed at 85%. Also shown is the lowest gearing needed by these vehicles which is set equal to that of the comparable conventional vehicle (ie final drive ratio x first gear ratio) to give similar low speed acceleration and ability to overcome gradient. From the lowest and highest gearing requirements, the overall ratio spread needed for each car is evaluated and shown in Table 6.1.

Table 6.2 shows the gearing capabilities of the CVTs selected. Comparison of the overall ratio spreads of Figs 6.1 and 6.2 shows that the two traction drives can provide the required ratio spread. Although the HMT has a smaller overall ratio spread at full load than that required, its ratio spread at lower loads is wider (kinematic ratio range of 100:1) and should, therefore, be able to meet the gearing requirement of

Table 6.1. The Transmatic has a much smaller overall ratio spread than that required and although the engine operating schedule can be met, the vehicle accelerative performance may be reduced.

For each vehicle the final drive ratio required can be evaluated by dividing the highest gearing required by the highest gear of the employed CVT. It should be noted that the highest gearing required depends on the CVT efficiency and therefore differs for each type of CVT employed in the vehicle.

Table 6.3 shows the final drive ratios selected for each CVT vehicle to give the gearing requirements described. The initial reduction gears employed in between the engine and transmatic (to reduce speed input and therefore improve CVT efficiency) are also shown. These gears were selected so that the torque input to the Transmatic does not exceed the torque limit at 130 NM (as set by design in Ref 38) at any condition of load and speed. It should be noted that this torque limit can be set by the user if required.

6.1.2 Urban Fuel Economy

Tables 6.4 to 6.6 compare the transmission efficiencies and fuel economy of the conventional and CVT vehicles over the

ECE 15 driving cycle. The CVTs have a lower efficiency than the manual gear box and, therefore, the engine of the CVT car has to produce more power. Fuel economy improvements are achieved however due to the higher engine efficiencies obtained by operation along the engine's optimum operating curve (Fig 6.1) as opposed to engine operation dictated by the fixed gears of the conventional transmission (Fig 6.2).

It was shown earlier (Chapter 5) that the urban fuel economy of a conventional vehicle depends highly on the gear shift speeds. The CVT vehicle fuel economy is therefore compared with the best possible fuel consumption of the conventional vehicle and not that imposed by the ECE 15 standard gear shifts.

The B. L. Perbury offers transmission average cycle efficiencies in the range 83% to 85% and fuel economy improvements from 3.5% to 6.7% over the range of cars considered. Vehicles employing the cone roller toroidal drive achieve higher improvements in urban fuel economy (5.1% to 7.8%) than the Perbury vehicles due to their higher CVT efficiencies (90% to 93%).

The Orshanskey's HMT as expected has a poorer efficiency than the other selected CVTs and the fuel economy improvements achieved by its employment are therefore less (0.4% to 3.8%).

Although the Transmatic's average urban efficiency is slightly lower than the Perbury, it achieves better fuel economies. This is due to Transmatic's better efficiency at the very low power levels. The improvements in urban fuel economy achieved by employment of Transmatic range from 3.3% to 6.5%.

In conclusion, despite their lower efficiencies, employment of CVTs tested can lead to urban fuel economy improvements of up to 8% due to their more efficient use of the engine.

6.1.3 Steady State Cruise Fuel Consumption

Fig 6.3 shows a comparison of the mini car fuel economy when employing the conventional four speed gear box and transmatic CVT. In order to avoid the problem of gear shift points for the conventional gear box the results are given for a range of fixed speeds.

A similar comparison for the small vehicle when employing the manual four speed and Perbury is shown in Fig 6.4. It is evident that at any constant road speed, selection of a higher gear leads to an improvement in fuel economy due to the increased load factor. It should be noted, however, that a higher engine load factor leads to a lower accelerative ability. Fig 6.4 shows that the employment of Perbury leads to an improvement of about 12% to 30% in fuel economy for the range of speeds between 28-75 mph. These predictions of fuel economy improvements correspond to the measured improvements found in the literature (42).

Fig 6.5 compares the effect of employment of six speed gear box and the Perbury for the small car. The Perbury vehicle has a clear advantage over the speed range of 20-50 mph. At the higher road speeds, however, the employment of a six speed gear box leads to better fuel economy due to the gear box's superior efficiency.

The steady state fuel consumptions available in employment of the other CVTs are shown in Figs 6.6 and 6.7. The CVT vehicles' fuel consumption vary slightly depending on their transmission efficiency and the best fuel economies are obtained at speeds between 30-40 mph for the range of vehicles considered.

6.1.4 Overall Fuel Economy

Tables 6.7 to 6.9 compare the fuel economies of vehicles when employing the conventional four speed gear box and the selected CVTs. Also included are the results for five and six speed manual transmission.

The urban fuel economy is set equal for all the manual transmissions as their first four gears are identical and a shift beyond fourth gear during urban driving conditions must be considered unpractical.

The employment of CVT in the range of vehicles considered offers improvements in overall fuel economy between 6.7% to 23.2% over the best possible fuel economy with a conventional four speed gear box. This improvement reduces to 0.3% to 17.6% when compared with the five speed manual transmissions. Employment of the six speed gear box can result in better fuel economies than obtainable by the relatively less efficient CVTs like the HMT. Employment of the more efficient CVTs can, however, offer improvements of up to 7.5% or the six speed gear box.

In general, the smallest improvements in overall fuel economy are obtained by employment of the HMT and the highest in using the CRTD.

6.1.5 CVT Vehicle Performance

In WOT acceleration the engine speed of the conventional vehicle and therefore power available in engine and at road wheels fluctuate with road speed due to its fixed gearing transmission while vehicles employing a CVT can after a certain road speed supply peak power continuously. Therefore in employment of CVTs an improvement in accelerative performance may be achieved. Tables 6.10 to 6.12 compare the performance obtained in employment of the selected CVTs with the conventional vehicles over the range of vehicles considered here. Clearly, the employment of either of the traction drives improves vehicle accelerative performance. This is due to their relatively high efficiency and wide gear ratio range.

The vehicles employing the HMT have a poorer performance than the conventional manual gear box vehicles. This is due to HMT's poor efficiency at high power levels. The HMT vehicles performance found by simulation are comparable to that in literature (40) which claims an accelerative performance comparable to a vehicle with an automatic gear box.

The vehicles using the Transmatic transmission offer a poor accelerative performance which arises from this CVT's relatively narrow ratio spread.

Finally, as far as the maximum speed results are concerned, all the CVTs selected except the HMT can achieve similar speeds as that of the respective conventional vehicles.

6.2 Reduction in Engine Power Rating

It has been shown that the employment of the two traction drives can lead to improvements in vehicle acceleration. A reduction in engine power rating for the vehicles employing these CVTs could, therefore, be achieved without any reduction in vehicle accelerative performance. The smaller engine increases the engine load factor further improving engine efficiency in urban conditions. Tables 6.13 to 6.15 show the amount to which engine power rating could be reduced to obtain the same acceleration performance as the conventional vehicle. The reduction in engine power rating leads to a reduction in vehicle maximum speed as shown. If the reduction in maximum speed is considered unimportant and engines of lower power ratings are employed in the CVT vehicles, further improvements in vehicle fuel economy are obtained as shown in Tables 6.16 to 6.18. The results presented show that engine power rating could be reduced by a typical value of 11% which further improves the CVT vehicle fuel economy by a typical value of 2.5% compared with the conventional vehicle. This further improvement in fuel economy can be detected by comparing Figs 6.5 and 6.8. In conclusion the CVT vehicle with reduced power rating can achieve fuel economy improvements of around 18%, when compared with the conventional vehicles of the same accelerative performance.

6.3 Conclusions ~~(should this be 6.3?)~~

1. If present CVTs plus their control systems have the same weight as a conventional manual transmission, when employed with the same power rating, they can:

   a) improve urban (ECE 15) fuel economy up to 8% over the best obtainable by manual transmission;

   b) improve steady state cruise fuel economy over that obtained by a five speed manual transmission;

   c) improve steady state cruise fuel economy in the range 20-50 mph over that obtained by a six speed manual transmission;

   d) improve overall fuel economy by:

      7% to 23% over the best obtainable by four speed manual;

      0% to 18% over the best achieved by five speed manual;

      -5.1% to 7.5% over the best obtained by a six speed transmission;

   e) improve vehicle acceleration by about 11%.

2. The accelerative performance of a CVT vehicle is dependent on:

   a) CVT efficiency;

   b) CVT overall gear ratio range;

   c) engine maximum power and operating schedule.

3. When a traction drive is employed, engine power rating could be typically reduced by about 11% without any loss in accelerative performance. This is accompanied however by a reduction in maximum vehicle speed of up to 7%.

4. A vehicle employing a CVT and a lower power rating (same accelerative performance) can improve urban fuel consumption by 5-16% and the overall fuel economy by 12%-27% when compared with the best obtained by a four speed manual transmission.

I. C. ENGINE PERFORMANCE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); SFC-PTS/HP-HR

140.00	0.49	0.49	0.49	0.49	0.47	0.48	0.49	0.49	0.49	0.49
133.00	0.50	0.50	0.50	0.47	0.46	0.47	0.48	0.48	0.48	0.48
126.00	0.55	0.55	0.48	0.44	0.44	0.45	0.46	0.51	0.51	0.51
119.00	0.55	0.52	0.46	0.45	0.45	0.46	0.47	0.49	0.51	0.51
112.00	0.48	0.49	0.47	0.45	0.46	0.46	0.47	0.49	0.50	0.53
105.00	0.42	0.48	0.47	0.46	0.47	0.46	0.47	0.49	0.50	0.53
98.00	0.44	0.47	0.47	0.47	0.47	0.46	0.47	0.50	0.50	0.53
91.00	0.46	0.48	0.47	0.48	0.48	0.48	0.48	0.50	0.51	0.54
84.00	0.50	0.48	0.48	0.49	0.49	0.49	0.49	0.51	0.51	0.57
77.00	0.50	0.50	0.49	0.50	0.51	0.51	0.51	0.51	0.53	0.59
70.00	0.50	0.52	0.51	0.50	0.52	0.52	0.53	0.54	0.56	0.56
63.00	0.53	0.53	0.52	0.54	0.55	0.55	0.56	0.56	0.58	0.58
56.00	0.55	0.57	0.57	0.58	0.57	0.58	0.58	0.60	0.60	0.60
49.00	0.63	0.60	0.62	0.61	0.60	0.61	0.62	0.64	0.62	0.64
42.00	0.66	0.64	0.66	0.64	0.63	0.64	0.65	0.69	0.69	0.67
35.00	0.68	0.67	0.74	0.73	0.72	0.73	0.74	0.80	0.76	0.82
28.00	0.82	0.82	0.82	0.81	0.80	0.82	0.84	0.90	0.90	0.90
21.00	1.01	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.20
14.00	1.72	1.29	1.33	1.31	1.30	1.31	1.35	1.48	1.51	1.72
	0.99	1.48	1.98	2.48	3.02	3.52	4.02	4.51	5.01	5.50

I. C. ENGINE FUEL USAGE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); USAGE - %

140.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
133.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
126.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
119.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
112.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
105.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
98.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
91.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
84.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
77.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
70.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
63.00	0.0	0.0	3.1	1.0	0.0	0.0	0.0	0.0	0.0	0.0
56.00	0.0	1.0	1.5	0.5	0.0	0.0	0.0	0.0	0.0	0.0
49.00	0.0	0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
42.00	0.0	3.1	2.6	1.0	0.0	0.0	0.0	0.0	0.0	0.0
35.00	0.0	1.5	0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0
28.00	0.0	2.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
21.00	0.0	2.6	0.0	6.6	0.0	0.0	0.0	0.0	0.0	0.0
14.00	0.0	6.6	4.1	12.2	0.0	0.0	0.0	0.0	0.0	0.0
	0.99	1.48	1.98	2.48	3.02	3.52	4.02	4.51	5.01	5.50

FIG 6.1 - Engine usage map of the Medium car
driven over ECE-15 cycle (standard gear shifts)

I. C. ENGINE PERFORMANCE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); SFC-PTS/HP-HR

140.00	0.49	0.49	0.49	0.49	0.47	0.48	0.49	0.49	0.49	0.49
133.00	0.50	0.50	0.50	0.47	0.46	0.47	0.48	0.48	0.48	0.48
126.00	0.55	0.55	0.48	0.44	0.44	0.45	0.46	0.51	0.51	0.51
119.00	0.55	0.52	0.46	0.45	0.45	0.46	0.47	0.49	0.51	0.51
112.00	0.48	0.49	0.47	0.45	0.46	0.46	0.47	0.49	0.50	0.53
105.00	0.42	0.48	0.47	0.46	0.47	0.46	0.47	0.49	0.50	0.53
98.00	0.44	0.47	0.47	0.47	0.47	0.46	0.47	0.50	0.50	0.53
91.00	0.46	0.48	0.47	0.48	0.48	0.48	0.48	0.50	0.51	0.54
84.00	0.50	0.48	0.48	0.49	0.49	0.49	0.49	0.51	0.51	0.57
77.00	0.50	0.50	0.49	0.50	0.51	0.51	0.51	0.51	0.53	0.59
70.00	0.50	0.52	0.51	0.50	0.52	0.52	0.53	0.54	0.56	0.56
63.00	0.53	0.53	0.52	0.54	0.55	0.55	0.56	0.56	0.58	0.58
56.00	0.55	0.57	0.57	0.58	0.57	0.58	0.58	0.60	0.60	0.60
49.00	0.63	0.60	0.62	0.61	0.60	0.61	0.62	0.64	0.62	0.64
42.00	0.66	0.64	0.66	0.64	0.63	0.64	0.65	0.69	0.69	0.67
35.00	0.68	0.67	0.74	0.73	0.72	0.73	0.74	0.80	0.76	0.82
28.00	0.82	0.82	0.82	0.81	0.80	0.82	0.84	0.90	0.90	0.90
21.00	1.01	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.20
14.00	1.72	1.29	1.33	1.31	1.30	1.31	1.35	1.48	1.51	1.72
	0.99	1.48	1.98	2.48	3.02	3.52	4.02	4.51	5.01	5.50

I. C. ENGINE FUEL USAGE MAP

X-SPEED (RPM/1000); Y-BMEP (PSI); USAGE - %

140.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
133.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
126.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
119.00	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
112.00	0.0	2.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
105.00	0.0	1.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
98.00	0.0	0.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
91.00	0.0	1.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
84.00	0.0	1.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
77.00	0.0	1.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
70.00	0.0	1.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
63.00	0.0	1.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
56.00	0.0	1.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
49.00	0.0	1.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
42.00	0.0	8.7	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
35.00	0.0	1.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
28.00	0.0	2.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
21.00	0.0	20.4	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
14.00	0.0	6.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	0.99	1.48	1.98	2.48	3.02	3.52	4.02	4.51	5.01	5.50

PERCENTAGE TIME AT ENGINE IDLE/OFF = 49.0
PERCENTAGE TIME COMP. BRAKING = 18.4

FIG 6.2 - Engine usage map of the Medium car (employing the B L Perbury CVT) driven over ECE-15 cycle.

Figure 6.3 - Comparison of Mini Car cruise fuel consumption obtained by employment of Transmatic and 4-speed manual gear-box .

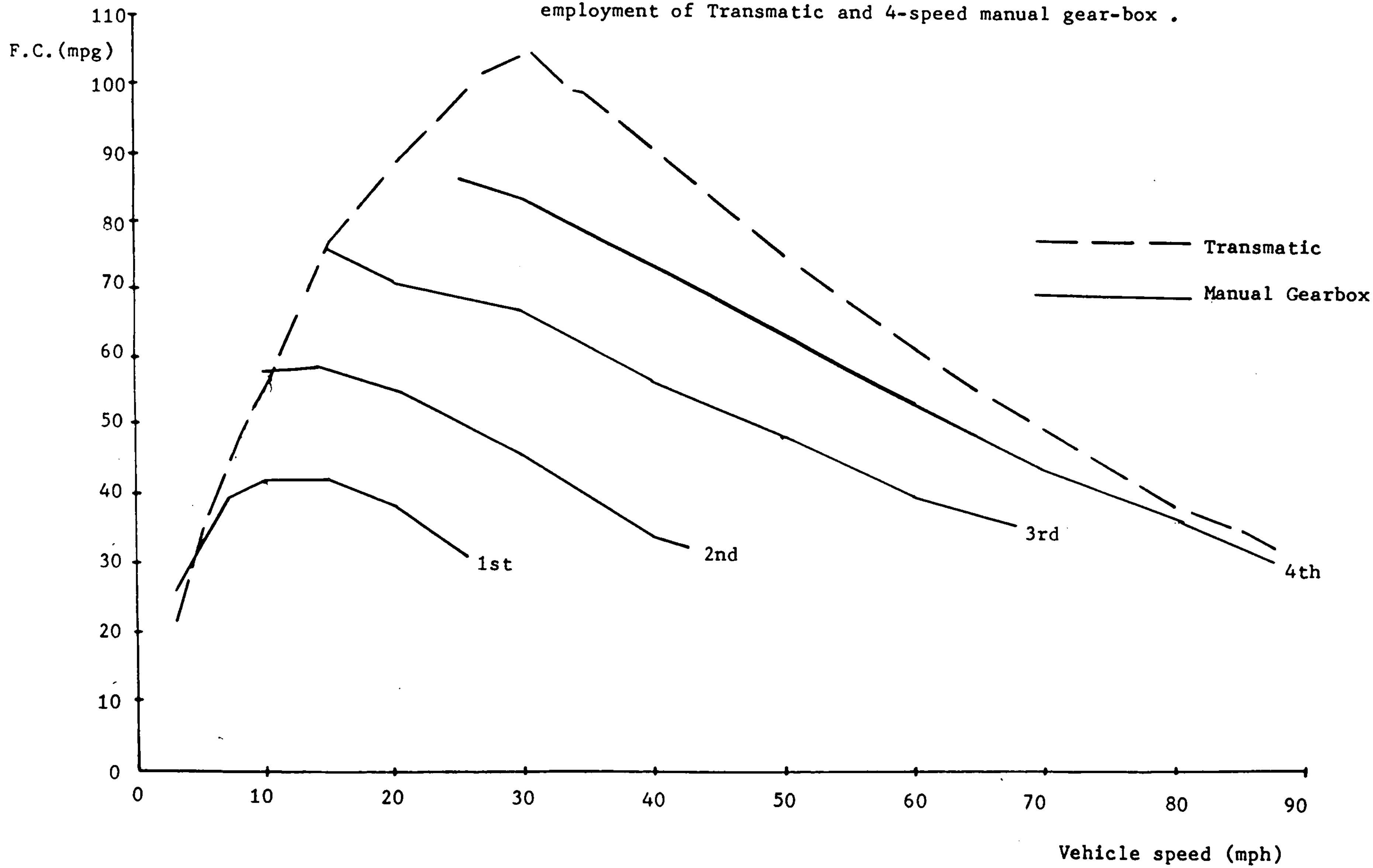


Figure 6.4 - Comparison of Small car cruise fuel consumption achieved by employment of Perbury or 4-speed manual gear-box .

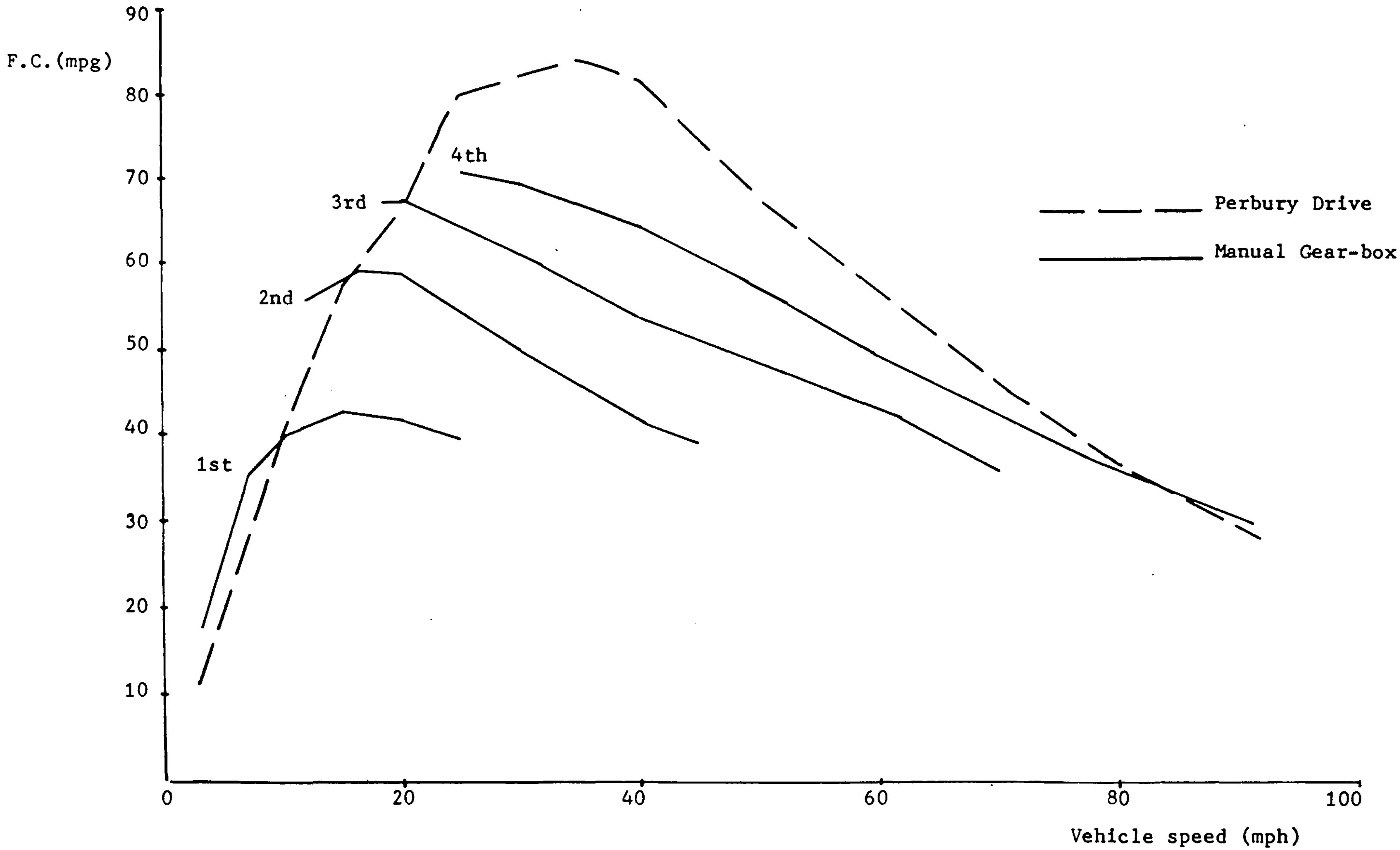


Figure 6.5 - Comparison of the fuel economy of the Small vehicle obtained by employing the Perbury and 6-speed manual gear-box .

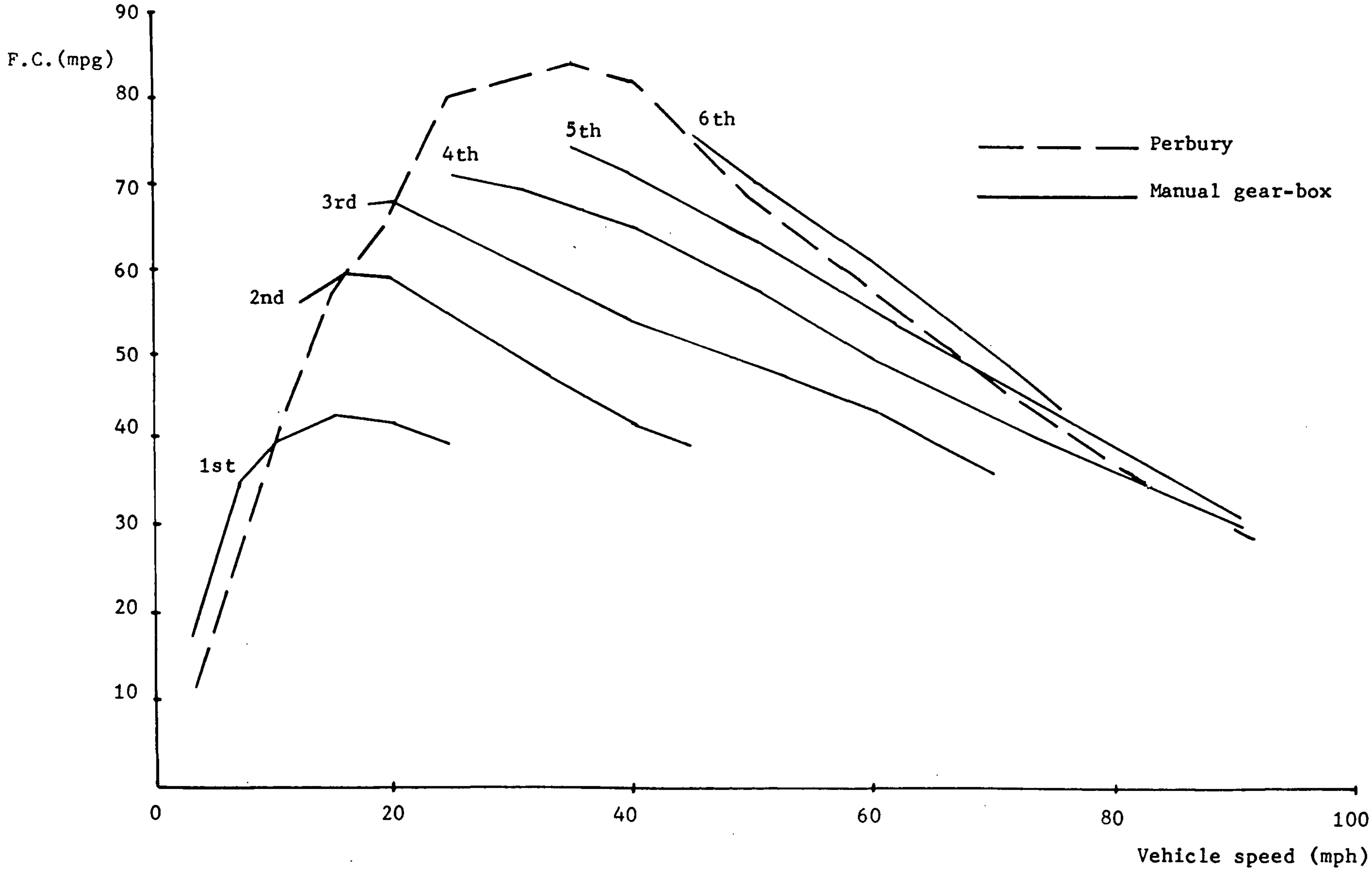


Figure 6.6 - A comparison of cruise fuel economies obtained by employment of C.R.T.D. and H.M.T. in the Small car .

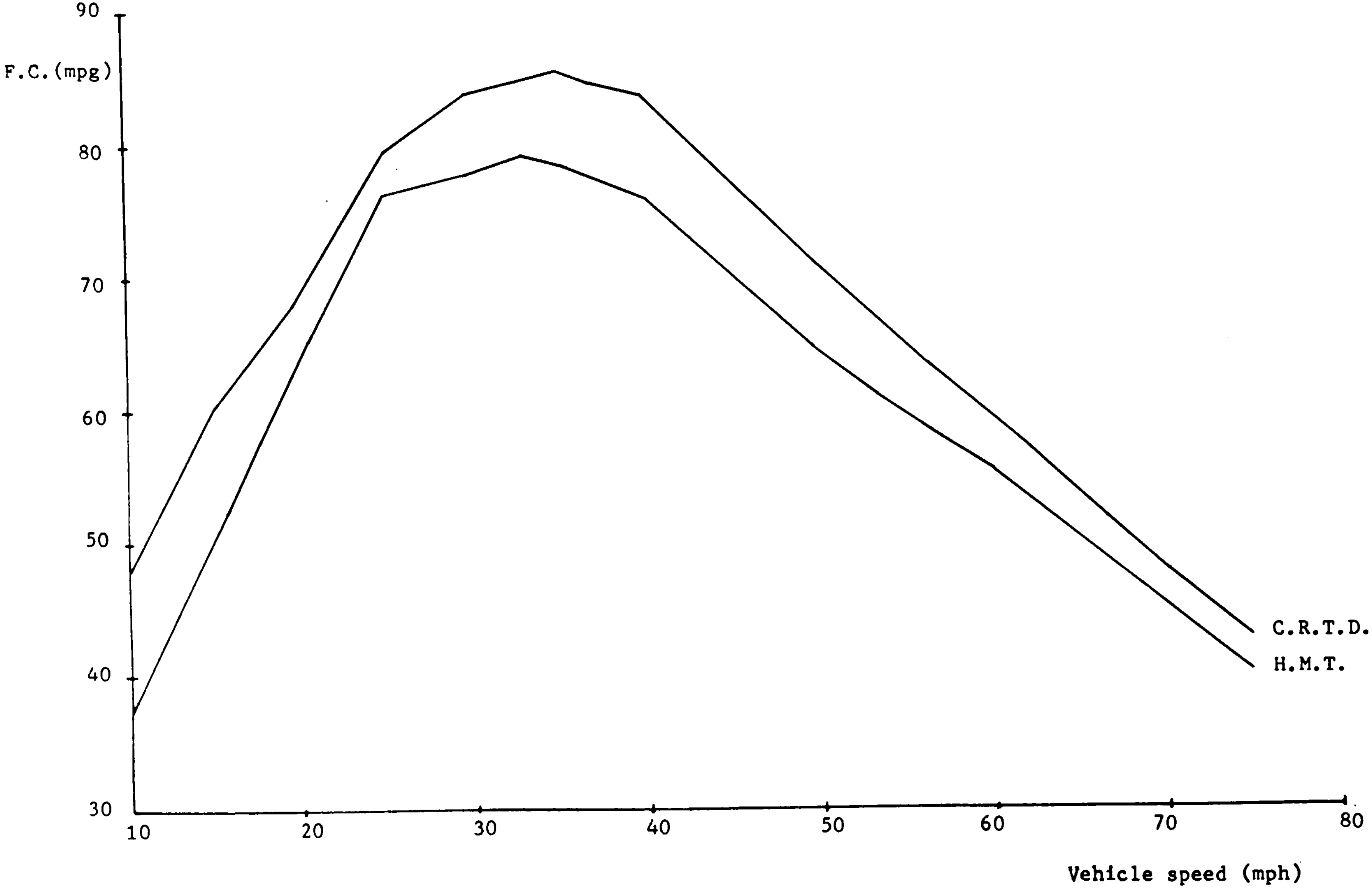


Figure 6.7 - Comparison of Small car cruise fuel economy obtained through the employment of Perbury drive and C.R.T.D.

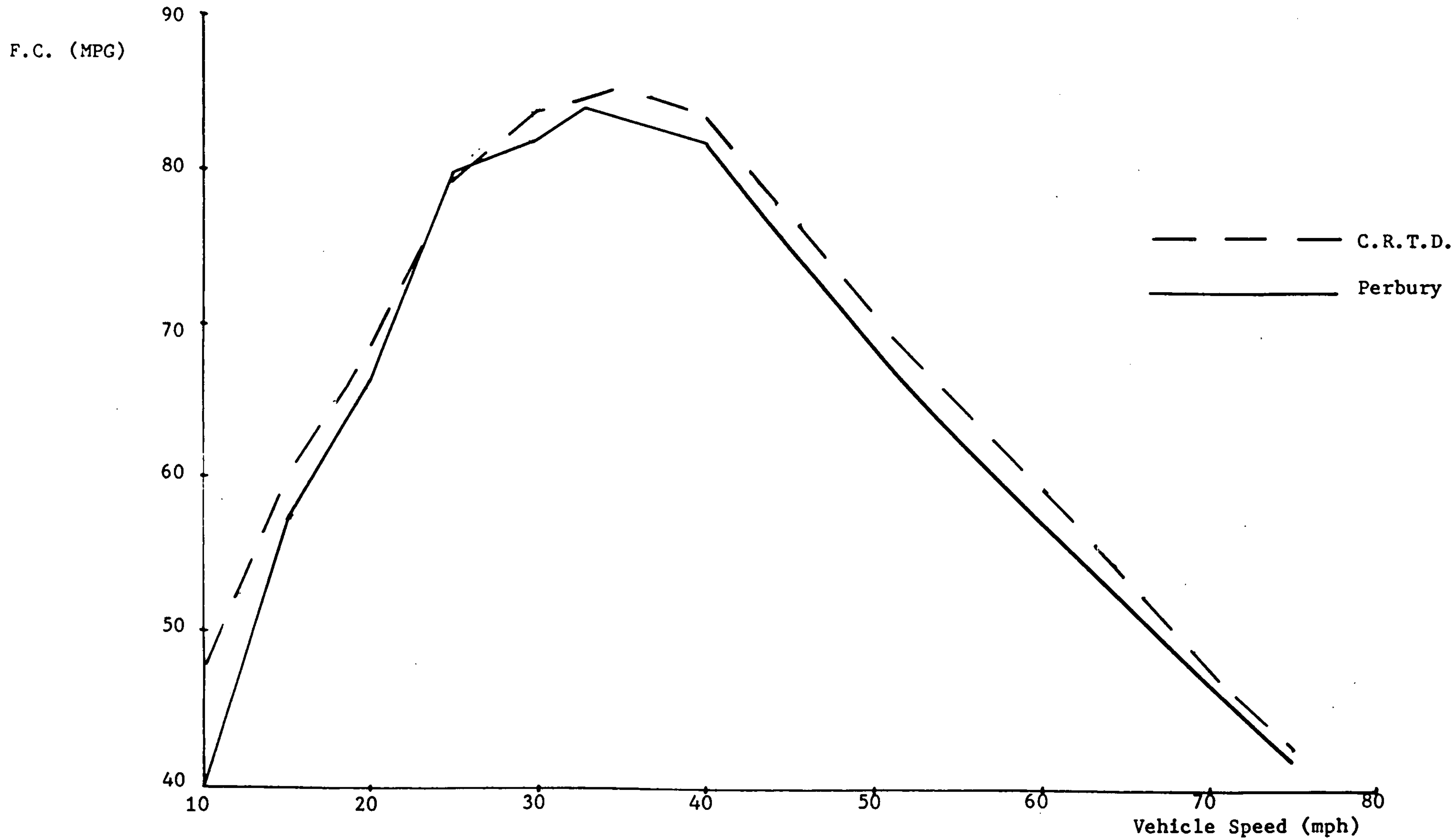
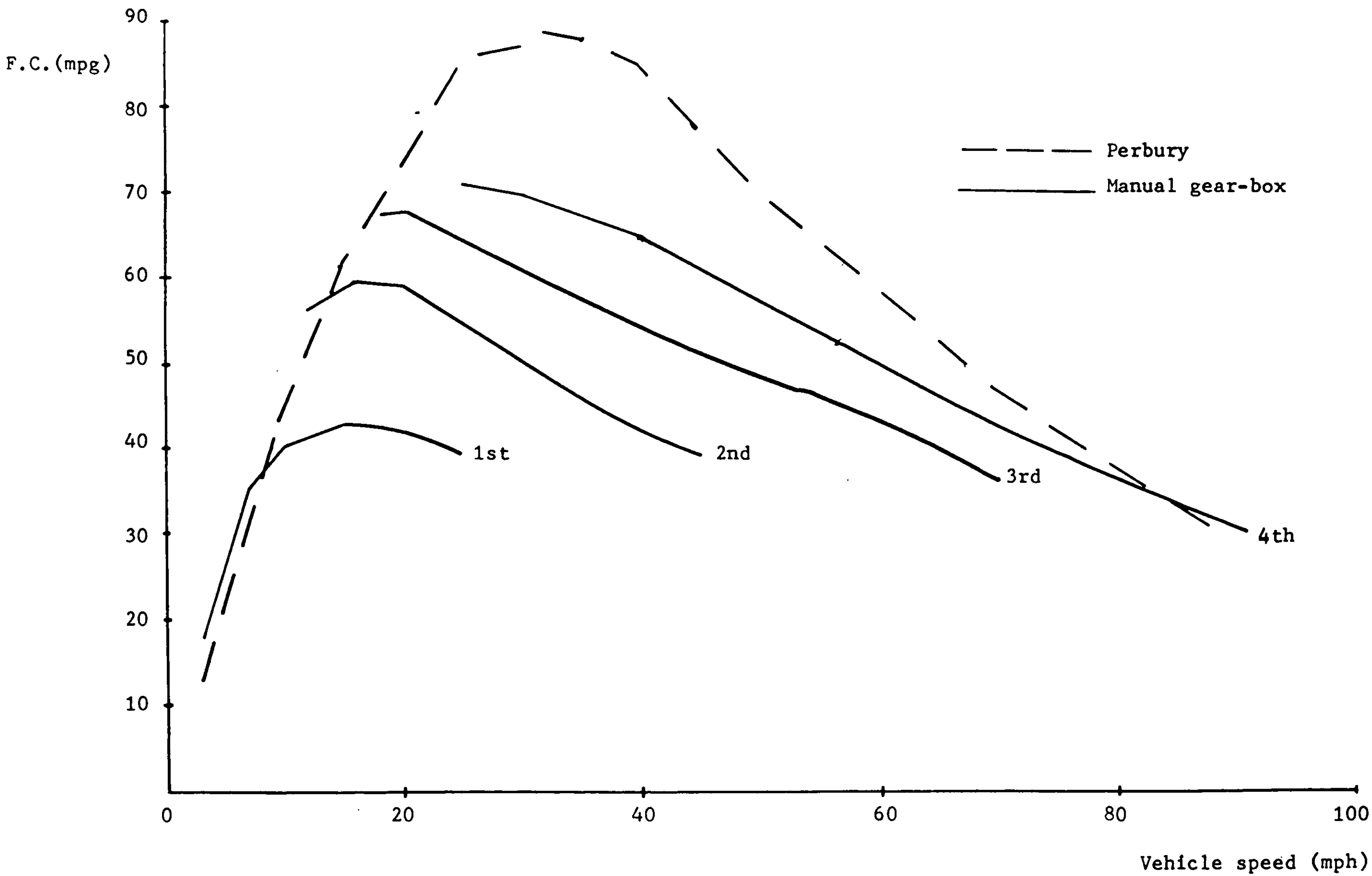


Figure 6.8 - Comparison of the fuel economy of the Conventional vehicle with the vehicle employing the Perbury with reduced power rating .



Vehicle	Mini	Small	Medium
Highest gearing needed	1.46:1	1.44:1	1.19:1
Lowest gearing needed (Final drive ratio x 1st gear ratio)	13.5:1	12.4:1	11.3:1
Overall ratio spread requirement	9.25:1	8.2:1	9.5:1

TABLE 6.1 - CVT vehicle gearing requirements

C V T	Perbury	C.R.T.D.	H.M.T.	Transmatic
Highest Gearing	0.45:1	0.54:1	0.50:1	0.48:1
Lowest Gearing	5.00:1	6.48:1	4.00:1	2.10:1
Overall ratio range available	11:1	12:1	8:1	4.4:1

TABLE 6.2 - Gearing capabilities of the selected CVTs at full power.

Vehicle	Mini	Small	Medium
Perbury	3.22	3.30	2.95
C.R.T.D.	2.85	2.82	2.35
H.M.T.	2.95	3.11	2.65
Transmatic*	1.5 (x 1.97)	1.97 (x 1.54)	2.34 (x 1.11)

* Figure in the parenthesis
is the initial reduction gear required by Transmatic.

TABLE 6.3 - Final drive ratios required for each CVT vehicle.

Transmission	Transmission efficiency (%)	Fuel Consumption (mpg)	% Improvements in fuel economy
Manual (ECE-15 shifts)	90.1	41.8	-18.5
Manual (optimum shifts)	94.8	51.3	-
Perbury	85.4	53.1	+ 3.5
C.R.T.D.	90.7	54.7	+ 6.6
H.M.T.	79.6	51.5	+ 0.4
Transmatic	83.7	53.0	+ 3.3

TABLE 6.4 - Mini vehicle urban fuel economy with different transmissions.

Transmission	Transmission efficiency (%)	Fuel Consumption (mpg)	% improvements in f.c.
Manual (ECE-15 shifts)	93.5	39.5	- 7.9
Manual (optimum shifts)	95.5	42.9	-
Perbury	85.1	44.9	+ 4.7
C.R.T.D.	90.7	45.1	45.1
H.M.T.	79.1	44.2	+ 3.0
Transmatic	83.3	44.8	+ 4.4

TABLE 6.5 - Small car urban fuel economy.

Transmission	Transmission efficiency (%)	Fuel Consumption (mpg)	% improvements in f.c.
Manual (ECE-15 shifts)	92.2	34.5	- 7.0
Manual (optimum shifts)	93.4	37.1	-
Perbury	83.2	39.6	+ 6.7
C.R.T.D.	90.0	40	+ 7.8
H.M.T.	76.2	38.5	+ 3.8
Transmatic	81.3	39.5	+ 6.5

TABLE 6.6 - Medium car urban fuel economy.

Transmission		Fuel consumption (mpg)				Improvements in overall f.c. (%)
		ECE-15	56 mph	75 mph	overall*	
MANUAL	4-speed cruise in 1:1	51.3** (41.8)	55.5	38.7	51.8	-
	5-speed cruise in 0.8:1		61.6	42.3	55.6	+ 7.3
	6-speed cruise in 0.6:1		67.1	44.4	58.5	+12.9
Perbury		53.1	64.6	41.9	57.7	+11.4
C.R.T.D.		54.7	67.2	43.0	59.8	+15.4
H.M.T.		51.5	62.5	39.5	55.8	+ 7.7
Transmatic		53.0	66.9	43.7	59.0	+13.9

* (40% ECE-15, 50% 56 mph, 10% 75 mph)

** Optimum gear shifts (standard gear shifts)

TABLE 6.7 - Comparison of 'Mini' car fuel economy (mpg) obtained by employment of the different types of transmissions.

Transmission		Fuel consumption (mpg)				Improvement in overall f.c. (%)
		ECE-15	56 mpg	75 mpg	overall*	
MANUAL	4-speed cruise in 1:1	42.9** (39.5)	53.5	39.7	47.9	-
	5-speed cruise in 0.8:1		58.2	43.4	50.6	+ 5.6
	6-speed cruise in 0.6:1		64.6	44.3	53.9	+12.5
Perbury		44.9	61.6	41.8	52.9	+10.4
C.R.T.D.		45.1	63.9	42.7	54.3	+13.4
H.M.T.		44.2	58.8	40.1	51.1	+ 6.7
Transmatic		44.9	63.7	43.4	54.2	+13.2

* (40% ECE-14, 50% 56 mph, 10% 75 mph.)
** Optimum gear shifts (standard gear shifts)

TABLE 6.8 - Comparison of 'small' car fuel economy obtained by different types of transmissions.

Transmission		Fuel Consumption (mpg) ECE-15	56 mph	75 mph	overall*	Improvements in overall f.c. (%)
MANUAL	4-speed cruise in 1:1	37.1** (34.5)	51.2	39.1	44.4	-
	5-speed cruise in 0.8:1		54.9	42.5	46.5	+ 4.7
	6-speed cruise in 0.6:1		63.1	45.5	50.9	+14.6
Perbury		39.6	66.6	43.0	53.4	+20.3
C.R.T.D.		40.0	68.7	43.6	54.7	+23.2
H.M.T.		38.5	59.8	41.8	49.5	+11.5
Transmatic		39.5	68.6	43.7	54.1	+21.8

* 40% ECE-15, 50% 56 mph, 10% 75 mph.
** optimum gear shifts (standard gear shifts).

TABLE 6.9 - Comparison of 'Medium' car fuel economy obtained in employment of different types of transmissions.

Performance	Conven- -tional	Perbury	CRTD	HMT	Trans- -matic
0-30 mph acceleration time (secs)	5.1	5.0	4.5	7.0	9.2
0-60 mph acceleration time (secs)	15.0	14.1	13.0	17.4	20.1
Maximum speed (mph)	91	88	88	83	88
Maximum speed at 5% gradient (mph)	70	72	72	69	72

TABLE 6.10 - Comparison of 'Mini' car performance when employing different types of transmissions.

Performance	Conven--tional	Perbury	CRTD	HMT	Trans--matic
0-30 mph acceleration time (secs)	5.5	4.5	4.6	6.2	9.4
0-60 mph aceleration time (secs)	15.7	14.0	12.9	16.5	19.4
Maximum speed (mph)	91	92	94	88	94
Maximum speed at 5% gradient (mph)	79	77	79	73	79

TABLE 6.11 - Comparison of 'Small' car performance when employing different types of transmissions.

Performance	Conven- -tional	Perbury	CRTD	HMT	Trans- -matic
0-30 mph acceleration time (secs)	4.9	4.8	4.5	6.1	8.7
0-60 mph acceleration time (secs)	12.9	12.1	11.2	14.5	19.2
Maximum speed (mph)	105	103	104	99	104
Maximum speed at 5% gradient (mph)	88	84	85	78	85

TABLE 6.12 - Comparison of 'Medium' car performance when employing different types of transmissions.

Vehicle	Engine Power Rating (kw)	Vehicle Weight (kg)	0-30 mph Min. acceler- -ation time (secs)	0-60 mph Min. acceler- -ation time (secs)	Maximum speed (mph)
Manual	35	800	5.1	15.0	91.0
Perbury	33	795	5.1	15.1	86.0
C.R.T.D.	31	790	5.1	14.9	86.0

TABLE 6.13 - Mini vehicles of comparable performance.

Vehicle	Engine Power Rating (kw)	Vehicle Weight (kg)	0-30 mph Min. acceler- -ation time (secs)	0-60 mph Min. acceler- -ation time (secs)	Maximum speed (mph)
Manual	40	900	5.5	15.7	91.0
Perbury	36	890	5.3	15.7	89
C.R.T.D.	33	880	5.4	15.6	88

TABLE 6.14 - Small vehicles of comparable performance.

Vehicle	Engine Power Rating (kw)	Vehicle Weight (kg)	0-30 mph Min. acceler- -ation time (secs)	0-60 mph Min. acceler- -ation time (secs)	Maximum speed (mph)
Manual	55	1050	4.9	12.9	105
Perbury	51	1040	4.9	13.0	100
C.R.T.D.	47	1030	4.8	12.9	98

TABLE 6.15 - Medium vehicles of comparable performance.

Transmission	ECE-15	56 mph cruise	75 mph cruise	Over-all f.c.	% improve-ments in overall f.c.
Manual (ECE-15 shifts)	41.8	55.5	38.7	48.3	- 6.8
Manual (optimum shifts)	51.3	55.5	38.7	51.8	-
Perbury	54.2	65.3	41.8	58.5	+12.9
C.R.T.D.	56.6	68.0	42.4	60.9	+17.6

TABLE 6.16 - Fuel economy (mpg) of Mini vehicles of similar performance

Transmission	ECE-15	56 mph cruise	75 mph cruise	Over-all f.c. (mpg)	% improve-ments in overall f.c.
Manual (ECE-15 shifts)	39.5	53.5	39.7	46.5	- 2.9
Manual (optimum shifts)	42.9	53.5	39.7	47.9	-
Perbury	45.0	63.1	41.8	53.7	+12.1
C.R.T.D.	47.8	66.3	42.5	56.5	+18.0

TABLE 6.17 - Fuel economy (mpg) of Small vehicles of similar performance.

	ECE-15	56 mph cruise	75 mph cruise	Over- -all f.c.	% improve- -ments in overall f.c.
Manual (ECE-15 shifts)	34.5	51.2	39.1	43.3	- 2.5
Manual (optimum shifts)	37.1	51.2	39.1	44.4	-
Perbury	41	67.2	43.3	54.3	+22.2
C.R.T.D.	43	69.3	44.7	56.3	+26.8

TABLE 6.18 - Fuel economy of Medium vehicles of similar performance.

C H A P T E R   S E V E N

VEHICLE FUEL ECONOMY BY THE YEAR 2000

7. Vehicle Fuel Economy by the Year 2000

In the last two chapters, the effect of possible improvements in each area of conventional vehicles are studied independently. This chapter attempts to predict the fuel economy of vehicles on the road by the year 2000, in incorporating the various improvements achieved in research vehicles and thought feasible by the turn of the century.

7.1 Vehicle Specification

Table 7.1 shows predicted design values for the range of vehicles considered. Vehicle improvements of at least 15% in vehicle total mass, 11% in drag coefficient and 17% in rolling resistance coefficient are thought possible. These improvements are well within the scope of the improvements already achieved in research vehicles (14, 47, 48, 49, 50). Feasibility of CVTs and their superiority over all other types of proposed transmissions has been shown in Chapter 6. It is believed that wide ratio range CVTs capable of efficiency characteristics at least equal to the B. L. Perbury will be employed in mass produced vehicles in the near future.

The vehicles employ a three cylinder (30-50 Kw) research engine (Section 5.6) with fuel cut-off during idle and overrun conditions. The engine power ratings are reduced relative to the present values as these vehicles employ a CVT and have improved vehicle characteristics. Table 7.2 shows that although the improved cars have lower power ratings they present a better performance than the present vehicles.

<u>Fuel Economy</u>

Figure 7.1 compares the cruise fuel consumptions predicted for a mini car at the turn of the century (Mini 2000) and those of the standard mini in each gear. Large improvements are to be gained, particularly at the 15-60 mph range.

The overall fuel saving obtained by design improvements in a number of areas of the vehicle, is not simply the algebraic sum of the percentage saving due to each component as these values are interelated. Tables 7.3 to 7.5 emphasise this point by first showing the fuel saving due to each improvement in isolation (ie no other vehicle data is changed) and then the fuel savings available if a number of these improvements are incorporated into a vehicle. Tables 7.3 to 7.5 also compare the fuel economy in MPG for the standard vehicle, with those predicted for such vehicles at the turn of the century. The largest savings are at urban driving conditions. Apart from when a CVT is employed all the urban figures shown have been obtained in using optimum gear shifts. Urban fuel savings of 80% to 90% are predicted which are mainly due to engine control improvements (ie fuel cut-off at idle and overrun). The overall fuel savings are in the range 59% to 65% which are higher than the high fuel economy target set at Section 1.

Table 7.6 compares the technical data of the Medium 2000 vehicle with two research vehicles of this range (47, 48). The performance of the three vehicles is shown in Table 7.7. The three vehicles have similar 0-60 mph acceleration and maximum speed performance. The urban fuel economy prediction for the Medium 2000 vehicle is higher than the two research vehicles as this vehicle has fuel cut-off at idle and overrun conditions. The urban fuel economy of the Medium 2000 without any fuel cut-off is, however, close to those achieved by the research vehicles, with the CVT compensating for its relatively higher weight and drag factor.

The steady state fuel consumption predictions for the Medium 2000 car are generally lower than that of the research vehicles which is due to its higher drag coefficient.

The overall fuel economy prediction for the Medium 2000 vehicle is slightly higher than the two research vehicles, which is due to its superior urban fuel consumption.

7.2 Advanced Vehicle

The vehicle specifications shown in Table 7.1 are moderate and if pressured the motor industry could readily embark on design of such vehicles. Comparison of Table 7.6 shows that better characteristics than those in Table 7.1 are attainable. This section attempts to estimate the type of fuel saving possible if costs are considered unimportant and the various parameters are taken to their technological limits which may be attained by the turn of the century. Table 7.8 shows the specification for such a vehicle which will be termed 'Advanced Mini'. Figs 7.2 and 7.3 show the steady state cruise fuel consumptions of the Mini 2000 and the Advanced Mini at various gradients. The gradients could also roughly represent constant rates of acceleration. Table 7.9 compares the figures in MPG for the two vehicles and suggests that further savings of around 26% in overall fuel economy is possible if the technological limits shown in Table 7.8 become feasible in the mass produced vehicles by the turn of the century.

7.3 Conclusions

1. Improvements of around 80% to 90% in urban fuel economy and 59% to 65% in overall fuel economy are predicted by the turn of the century.

2. The translation of ideas at present incorporated in research vehicles to on the road vehicles, regardless of cost, could lead to further fuel savings of up to 26% over Conclusion One.

Figure 7.1 - Comparison of cruise fuel economy of future Mini (2000) and the standard Mini in each gear .

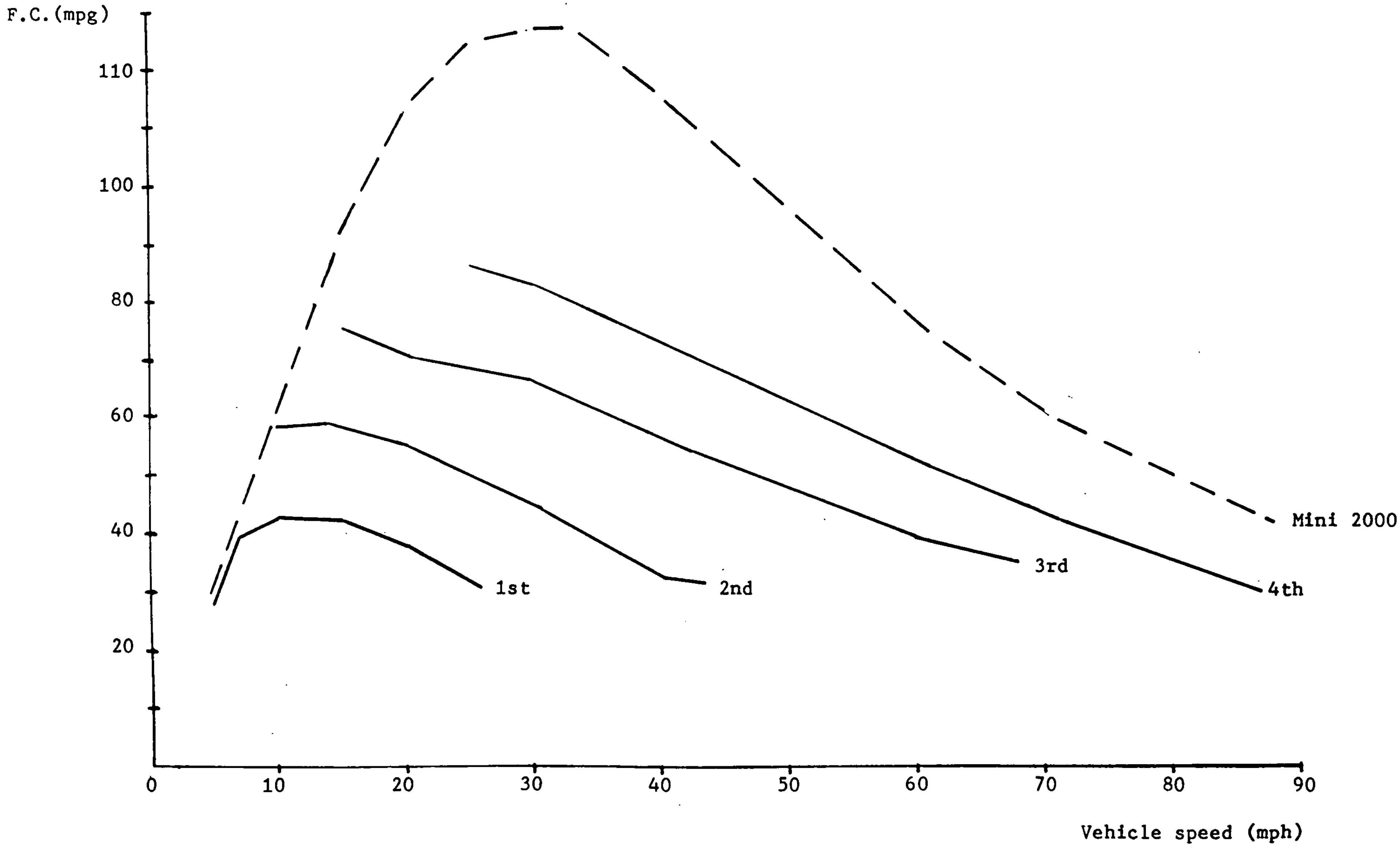


Figure 7.2 - Mini 2000 cruise fuel consumption at various gradients .

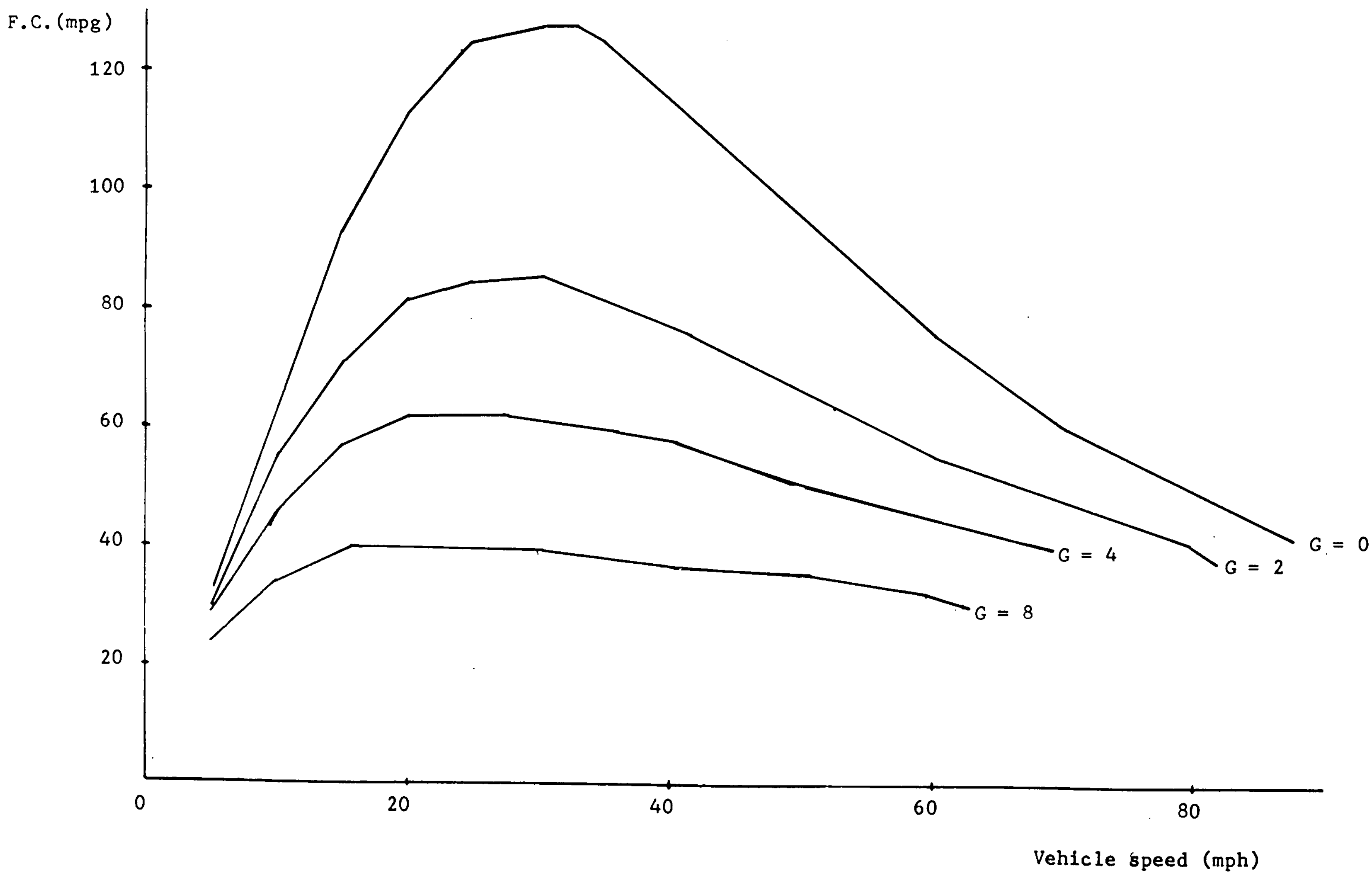
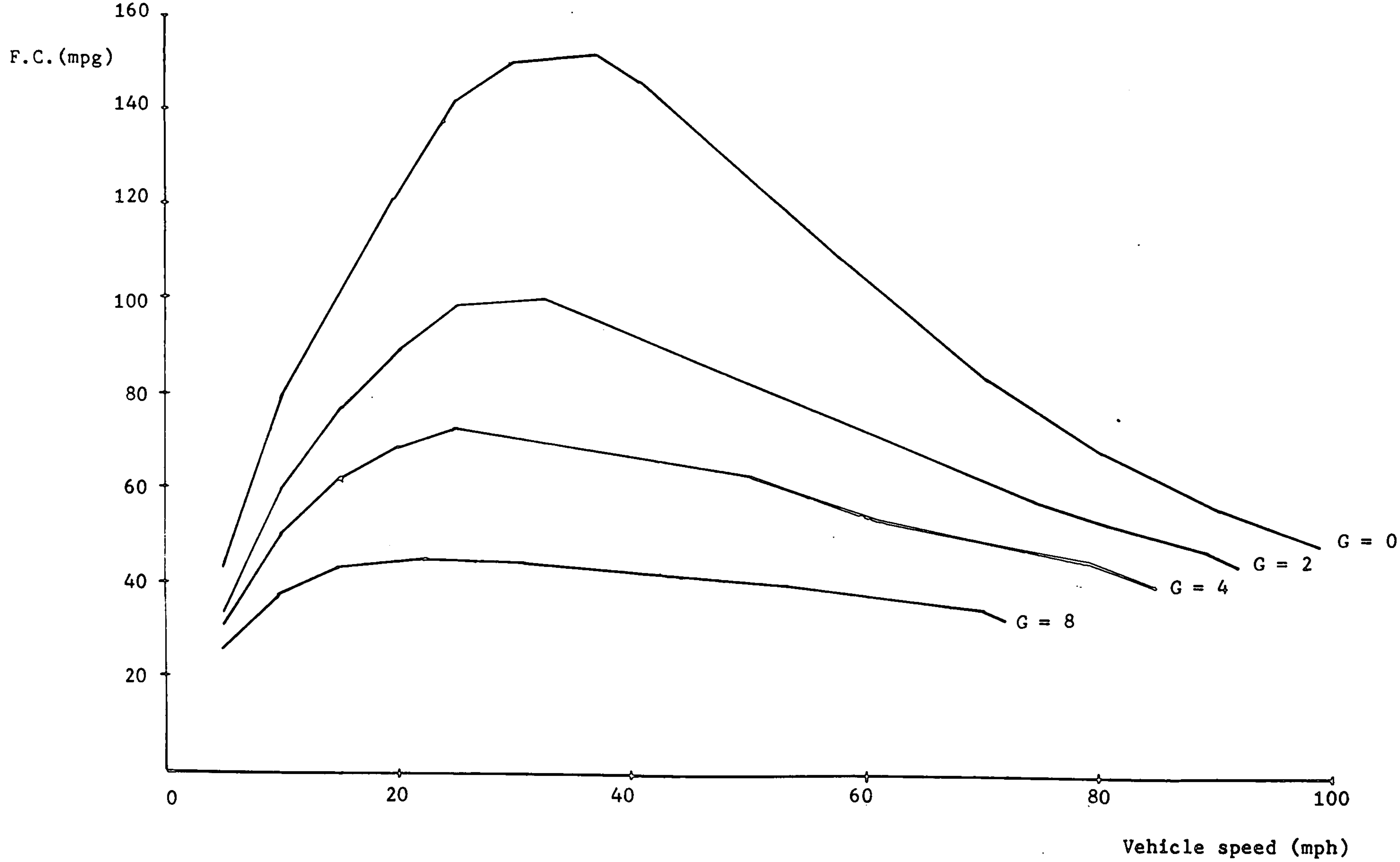


Figure 7.3 - Advanced mini car fuel consumption at various gradients .



Vehicle Characteristics	Mini 2000	Small 2000	Medium 2000
Vehicle weight (kg)	680	760	890
Drag coefficient	0.35	0.33	0.31
Frontal area (m^2)	1.73	1.8	1.95
Coefficient of rolling resistance	0.01	0.01	0.01
Wheel radius (m)	0.25	0.27	0.28
Wheel inertia as effective weight (kg)	10	15	20
Final drive ratio	3.32	3.40	3.51
CVT type	Perbury	Perbury	Perbury
Engine maximum power (kw)	30	35	45
Engine maximum speed (Rpm)	5000	5000	5000
Idle fuel consumption (g/s)	0.1	0.11	0.12
Engine inertia (kgm^2)	0.1	0.15	0.25

TABLE 7.1 - Predicted design characteristics of three car types by the year 2000.

Vehicle	WOT acceleration time (secs) 0-30 mph	0-60 mph	Max. Speed (mph)
Standard mini (35 kw)	5.1	15.0	91
Mini 2000 (30 kw)	4.7	13.8	88
Standard small (49 kw)	5.5	15.7	91
Small 2000 (35 kw)	4.6	13.2	93
Standard Medium (55 kw)	4.9	12.9	105
Medium 2000 (50 kw)	4.5	12.0	101

TABLE 7.2 - Comparison of performance of present vehicles and those thought feasible by the turn of century.

Design improvement	% Fuel saving possible			
	ECE-15*	56 mph	75 mph	overall
a) Vehicle improvements (reduction in W, CD & CR)	6	12	10	9
b) Engine transmission matching improvements (due to employment of B L Perbury)	4	16	8	10
c) Engine improvements (including the reduced power rating)	16	20	23	19
d) Engine & control improvements (fuel cut-off at idle and overrun)	36	20	23	27
e) Improvements obtained by incorporating a, b & c	29	50	45	41
f) Overall improvements (a, b and d)	80	50	45	62

Vehicle	Fuel consumption (mpg)			
	ECE-15*	56 mph	75 mph	overall
Standard Mini	51 (42)	56	39	52 (48)
Mini 2000 (f)	92	84	56	84

* optimum gear shifting during the urban driving cycle.

TABLE 7.3 - Areas for fuel saving and a comparison of mpg figures of standard Mini and those predicted for Mini car in the year 2000

	Design improvement	% Fuel saving possible			
		ECE-15*	56 mph	75 mph	overall
a)	Vehicle improvements (reduction in W, CD & CR)	5	7	10	7
b)	Engine transmission matching improvements (due to employment of B L Perbury)	5	15	5	10
c)	Engine improvements (including the reduced power rating)	26	20	20	22
d)	Engine & control improvements (fuel cut-off at idle & overrun)	67	20	20	39
e)	Improvements obtained by incorporating a, b & c	37	50	38	44
f)	Overall improvements (a, b and d)	90	50	38	65

Vehicle	Fuel consumption (mpg)			
	ECE-15*	56 mph	75 mph	overall
Standard Small	43 (40)	54	40	48 (47)
Small 2000 (f)	82	81	55	79

TABLE 7.4 - Areas for fuel saving and a comparison of mpg figures of standard Small and those predicted for a Small car in the year 2000.

	Design improvement	% Fuel saving possible			
		ECE-15*	56 mph	75 mph	overall
a)	Vehicle improvements (reduction in W CD & CR)	5	2	3	3
b)	Engine transmission matching improvements (due to employment of B L Perbury)	8	31	10	20
c)	Engine improvements (including the reduced power rating)	22	14	15	17
d)	Engine & control improvements (fuel cut-off at idle and overrun)	59	14	15	32
e)	Improvements obtained by incorporating a b & c	35	47	33	41
f)	Overall improvements (a b & d)	81	47	33	59

Vehicle	Fuel consumption (mpg)			
	ECE-15*	56 mph	75 mph	overall
Standard medium	37 (35)	51	39	44 (43)
Medium 2000 (f)	67	75	52	70

TABLE 7.5 - Areas for fuel saving and a comparison of mpg figures of standard, Medium and those predicted for a Medium car in the year 2000.

Vehicle	Medium 2000	BL ECV3 (14)	Volvo LCP 2000 (48)
Drag Coefficient	0.31	0.25	0.25 - 0.28
Frontal Area (m^2)	1.95	1.9	1.8
Test weight (kg)	890	664	707
Transmission type	Perbury	5-speed gear box	5-speed gear box
Power rating (kw)	50	54	NA (diesel)

TABLE 7.6 - A comparison of the technical data of the Medium 2000 car with two Research Vehicles (Ref. 14, 48)

Performance	Medium 2000	BL ECV3 (47)	Volvo LCP 2000 (48)
0-60 mph Acc. (secs)	12.0	11.00	11.0
Maximum speed (mph)	105	115	115
Urban (ECE-15) Fuel consumption (mpg) (with no fuel cut-off)	67 50	49	47
Steady state 56 mph fuel consumption (mpg)	75	81	66
Steady state 75 mph fuel consumption (mpg)	52	61	59
Overall fuel economy (ECE-15 - 40%, 56 mph 50% 75 mph 10%)	70	63	58

TABLE 7.7 - A comparison of the performance predictions of the Medium 2000 car with two Research Vehicles of its range.

Specification	Mini 2000	Advanced Mini
Vehicle weight (kg)	6	620
Drag coefficient	0.35	0.27
Frontal area (m^2)	1.73	1.65
Coefficient of rolling resistance	0.01	0.008
Wheel radius (m)	0.25	0.24
Wheel inertia as an effective weight (kg)	10	10
Final drive ratio	3.32	3.4
Final drive cycle efficiency (%)	(85-95)	95
CVT type	Perbury	92% constant ψ
Engine maximum power (kw)	30	30
Idle fuel consumption (pts/hr)	0.8	0.7

TABLE 7.8 - Specification for the Advanced vehicle

Vehicle	Fuel consumption (mpg)				% improvement in fuel economy	
	Urban	56 mph	75 mph	overall	Urban	overall
Mini	92	84	56	84	-	-
Advanced Mini	104	113	74	106	+13	+26

TABLE 7.9 - Predicted fuel economy of Mini 2000 and Advanced Mini, a comparison.

CHAPTER EIGHT

CONCLUSIONS

8. Conclusion

A flexible vehicle simulation package 'JANUS' has been developed to facilitate investigation into new concepts in road vehicle technology. The accuracy of the program and its component subroutines have been proved by extensive and detailed tests on a range of I.C. engined vehicles. These tests have produced simulation results comparable with the test data. The simulation results for steady state cruise results at speeds higher than 30 mph are the most accurate and usually within 3% of the test data. The results for low speed cruise are the least accurate which is due to the low load factor of the engine. At such low load levels component efficiency changes rapidly and efficiency measurements are least accurate. Although urban cycles contain some low speed cruise periods, they also include periods of acceleration. This increases the average cycle load factor and in general good agreement is attained between test and simulated results, usually within 10%. It should be noted that these results are heavily dependent on the idle fuel consumption.

Due to the absence of viable alternatives, this work concentrates on I.C. engines which are expected to remain the dominant power units, at least, up to the next century. The diesel engine as an alternative to S.I. engine may increase its share of the car market. This increase is likely to be limited however by a shortage of middle distillation. For improvements in S.I. engine design, increases on the highest compression ratio will be limited by the onset of knock. The octane number of gasoline is unlikely to increase much in the future due to pressures to remove the lead anti knock additives for environmental reasons. The improvements in S.I. design will, therefore, have to be obtained by changes in fuel distribution and alterations to the geometry of piston and cylinder head. The improved design employed in this work is a modern three-cylinder research engine with improved performance arising from optimum cylinder size, lower friction losses and improved ignition at idle. The simulation results suggest that the employment of this engine results in fuel savings in order of 11%.

The most efficient operation of an engine is along the locus of its minimum break specific fuel consumption (BSFC) points and effficient management of gear shifting can lead to the engine operating close to this locus. The simulation results suggest that in this way urban fuel consumption could be improved by 8-23% compared with the fuel economy obtained by using the ECE-15 gear shift schedule. The large variation in these improvements is due to different engine sizes. These results, however, refer to an ideal case where by correct usage of the gear box the driver optimises the fuel consumption rates by approaching the ideal line as much as possible. But under the varying conditions of traffic the driver attention is concentrated on exterior events and cannot adequately deal with this additional problem. This optimum gear shifting could be attained, however, by proposed automatic gear box designs with shift gears relative to load and speed. Another method of achieving better engine transmission matching at medium and high road speed is addition of extra overdrive gears. Five speed gear boxes are already employed in motor cars and a progression to a six speed gear box is possible. Simulation results suggest that improvements in order of 15% in overall fuel economy can be obtained in this manner. Such savings could overcome any consumer resistance arising from the loss of comfort associated with multiple gear shifting.

To ensure engine operation along its locus of minimum BSFC points at any condition of load and speed a wide ratio spread continuously variable transmission (CVT) must be employed. Although numerous CVT designs have been proposed, their introduction to on-the-road vehicles has not occurred. The major reasons for this have been, depending on the design, relatively poor efficiency characteristics, inadequate ratio spreads, high weight or exessive cost. The most promising of these designs with efficiency characteristics approaching that of a manual gear box are the Transmatic, B L Perbury, Cone Roller Toroidal Drive (CRTD) and Hydromechanical Drives. Simulation results suggest that employment of the above transmissions in vehicles can lead to improvements of 7-23% in overall fuel economy compared with the best attainable by conventional 4-speed vehicles. Furthermore, employment of a CVT with a wide ratio spread (such as the two traction drives) can result in a better accelerative performance as the CVT can maintain engine speed to that at which maximum power occurs, once this speed has been attained. If this improvement in vehicle acceleration is not desired, the engine power rating could be reduced to produce the same accelerative performance as that of a comparable conventional vehicle with a gear box. Simulation results suggest that such a vehicle will have a fuel economy 12-27% better than the comparable conventional vehicle.

In the employment of electronically controlled gear boxes and continuously variable transmission, the direct linkage system between the accerlator and the engine is likely to be replaced by a closed loop control system where some drive choice is eliminated. The closed loop system could lead to the optimisation of the spark timings and the mixture strength. In addition, fuel could be cut off during idling and overrun conditions. The simulation results show that the combined effect of such fuel cut-off is an overall fuel saving of around 12%. Clearly these improvements are highly dependent on present vehicles' idle fuel consumption.

The application of high strength steel, cast aluminimum, plastics and magnesium in vehicles is expected to grow resulting in further weight reduction. One effect of this will be a reduction in rolling resistance losses. Another avenue to reduce the rolling resistance losses is by improvements in tyre tread and carcase design; these improvements may be restricted, however, by safety requirements. Turning to air drag losses, given that the seating attitude and position of car occupants is unlikely to change, significant reductions in frontal area are not expected. Further reductions in air drag coefficient, however, are attainable by tighter controls on vehicle styling and better air flow management. Simulation results show that by combination of the best weight, air drag and rolling resistance characteristics of existing vehicles into one vehicle, fuel saving of 6-9% can be attained by the turn of the century.

The incorporation of the improvements in vehicle design discussed into on the road vehicles by the turn of the century results in fuel savings of 59-65%. These predictions surpass the fuel economy targets set at the beginning of this work and are in line with the present aims of the motor industry. The above savings could be further improved by 26% through the translation of the ideas at present incorporated in research vehicles to on the road vehicles by the year 2000.

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