Chapter 3

Performance of cars and light vans

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3.1 Introduction

Top speed and acceleration have long been of prime interest in vehicle performance. However, the drastic increase in fuel cost has now focused attention on fuel consumption, which has become one of the most important aspects to be considered in vehicle development. This chapter introduces the resistance to vehicle motion equation and explains the individual resistances and their effects on the various values which make up vehicle performance. Topics affecting fuel consumption will be dealt with explicitly. The performance examples are restricted to cars and light vans; trucks and buses will be covered in Chapter 8. Naturally, the same equations of resistance to motion apply to both passenger and commercial vehicles, but the significance of the individual resistances in the two groups differs sufficiently to justify separate treatment.

3.2 Resistance to vehicle motion

3.2.1 Equation of resistance to motion

The motion of a vehicle is resisted by the following forces:

Aerodynamic drag	W_D
Rolling resistance	W_R
Climbing resistance (gravitational)	W_C
Acceleration resistance (inertial)	W_A

The force on the driven wheels necessary for vehicle motion is therefore

$$Z = W_D + W_R + W_C + W_A \quad \text{N} \tag{3.1}$$

The engine performance necessary to overcome a vehicle's resistance to motion is

$$P = \frac{ZV}{\eta_G \eta_A} \times 10^{-3} \quad \text{kW} \tag{3.2}$$

where η_G and η_A represent the efficiencies of the gearbox and axle respectively, and V is the driving speed.

3.2.2 Analysis of the resistances to motion

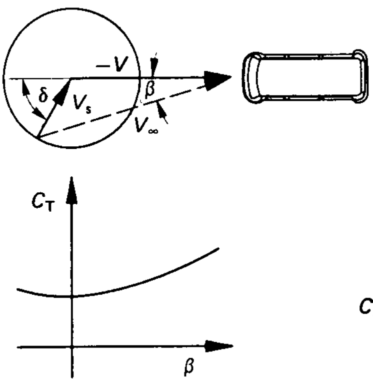
3.2.2.1 Aerodynamic drag

The aerodynamic drag results from flow characteristics and the aerodynamic data of the vehicle body

$$W_D = \frac{\rho}{2} V_\infty^2 c_T A \quad \text{N} \tag{3.3}$$

where the air density $\rho = 1.22\,\text{kg/m}^3$, the resultant air speed V_∞ m/s, the tangential force coefficient c_T (β) ($c_{T(\beta=0)} = c_D$) and the frontal area of the vehicle is A m^2.

The resultant air speed V_∞ is the sum of the vectors of the driving speed V and the wind speed V_S. The vector of the resulting airflow and driving direction determine the yaw angle β (Fig. 3.1).



Tangential force coefficient

$$C_T(\beta) = \frac{T(\beta)}{\frac{\rho}{2} V_\infty^2 A}$$

Average resistance coefficient

$$C_D^* = \frac{W^*}{\frac{\rho}{2} V^2 A}$$

$$C_D^* = \frac{1}{\pi}\int_0^\pi C_T(\beta)\left[\frac{V_\infty}{V}(\delta)\right]^2 d\delta$$

$$C_D^* = \frac{1}{\pi}\int_0^\pi C_T(\beta)\left[1 + \left(\frac{V_S}{V}\right)^2 + 2\,\frac{V_S}{V}\cos\delta\right] d\delta$$

Figure 3.1 Side wind averaged drag coefficient, after ref. 3.1

In the case of head- or tail-winds the following, based on Eqn 3.3, holds true:

$$W_D = \frac{\rho}{2}(V \pm V_S)^2 c_D A \tag{3.4}$$

Or under still-air conditions

$$W_D = \frac{\rho}{2} V^2 c_D A \tag{3.5}$$

This still-air condition seldom occurs, so a typical tangential force curve (Fig. 3.1), causing increased aerodynamic drag compared to the still-air condition, must be reckoned with. It is possible, for a defined vehicle usage and wind spectrum, to determine a mean aerodynamic drag coefficient, as described in Fig. 3.1. See also sections 4.2 and 8.4.3.

3.2.2.2 Rolling resistance

The rolling resistance of vehicle tyres is dependent on the load (normal force), tyre size and construction, tyre pressure and the axle geometry, i.e. caster and camber angles.

Assuming that the tyre manufacturer specifies tyre pressure in accordance with vehicle weight, and that for similar weight to tyre-pressure relationships similar rolling resistance results, the rolling resistance of a vehicle can be calculated with the formula

$$W_R = f_r G_N \tag{3.6}$$

where G_N is the normal force (newtons) due to the weight of the vehicle and f_r is the rolling resistance coefficient, which is dependent upon the type and size of the tyres. Figure 3.2 shows typical rolling resistance diagrams for radial and cross-ply tyres. The influence of caster and camber angles will be ignored.

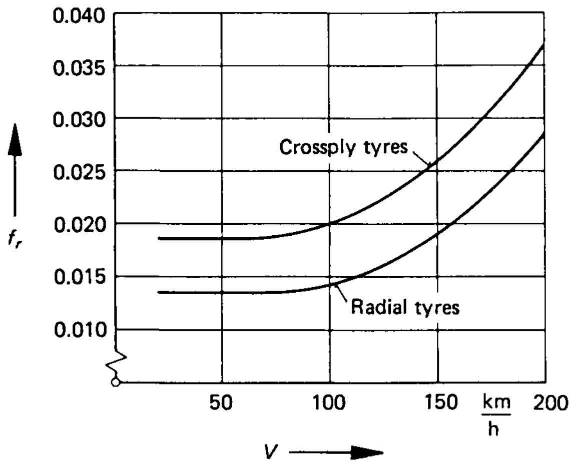


Figure 3.2 Rolling resistance coefficients for typical radial and crossply tyres

3.2.2.3 Climbing resistance

Climbing resistance may be expressed by

$$W_C = \sin\varphi \; G \; 9.81 \quad \text{N} \tag{3.7}$$

Climbing resistance is not taken into consideration for fuel consumption calculations, whether for steady speeds or for specified driving cycles. The reason is that representative altitude profiles for the respective driving cycles are unknown.

Investigations such as those by Schubert,[3.2] defining the frequency of ascents on specific freeways and cross-country routes (Fig. 3.3), are insufficient for general purposes.

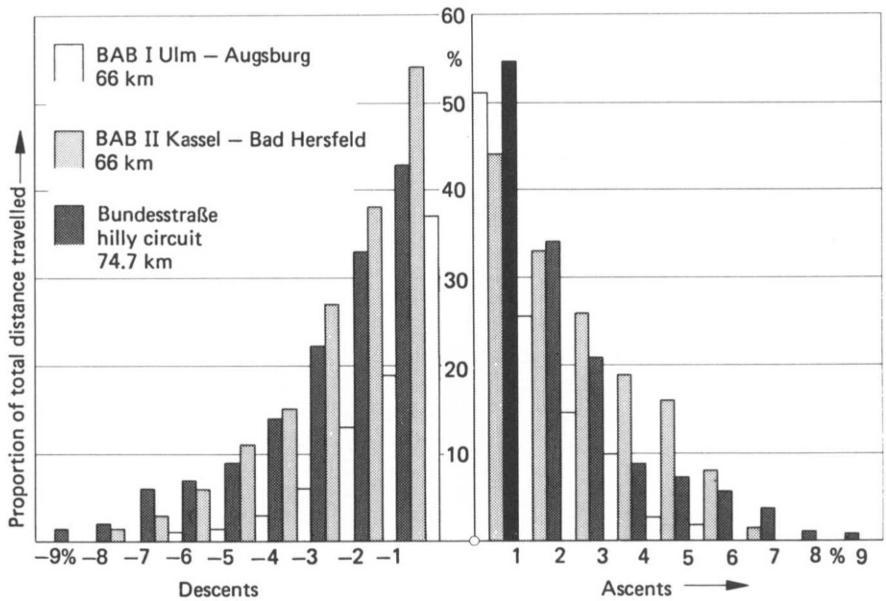


Figure 3.3 Distribution of ascents and descents on three observed routes (one direction), after ref. 3.2

3.2.2.4 Acceleration resistance

Resistance to acceleration may be expressed by

$$W_A = \dot{V}m(1 + \varepsilon_i)\ 9.81 \quad \text{N} \tag{3.8}$$

where the rotating masses in the various gears are taken into consideration by ε_i.

Representative ε_i values for cars, according to Bussien,[3.3] are as follows:

1st gear = 0.25
2nd gear = 0.15
3rd gear = 0.10
4th gear = 0.075

For exact calculations, the moments of inertia of all rotating parts relative to the vehicle mass must be established.

3.3 Performance

3.3.1 Motive force diagram

From the motive force diagram the speed-dependent propelling force at full engine load, the climbing ability and the necessary engine power in the various gears can be read off, see Fig. 3.4. The respective engine speeds are shown on a supplementary diagram (lower section of Fig. 3.4).

From the intersection of the driving resistance curve, consisting of rolling resistance and aerodynamic drag, and the propelling force at full

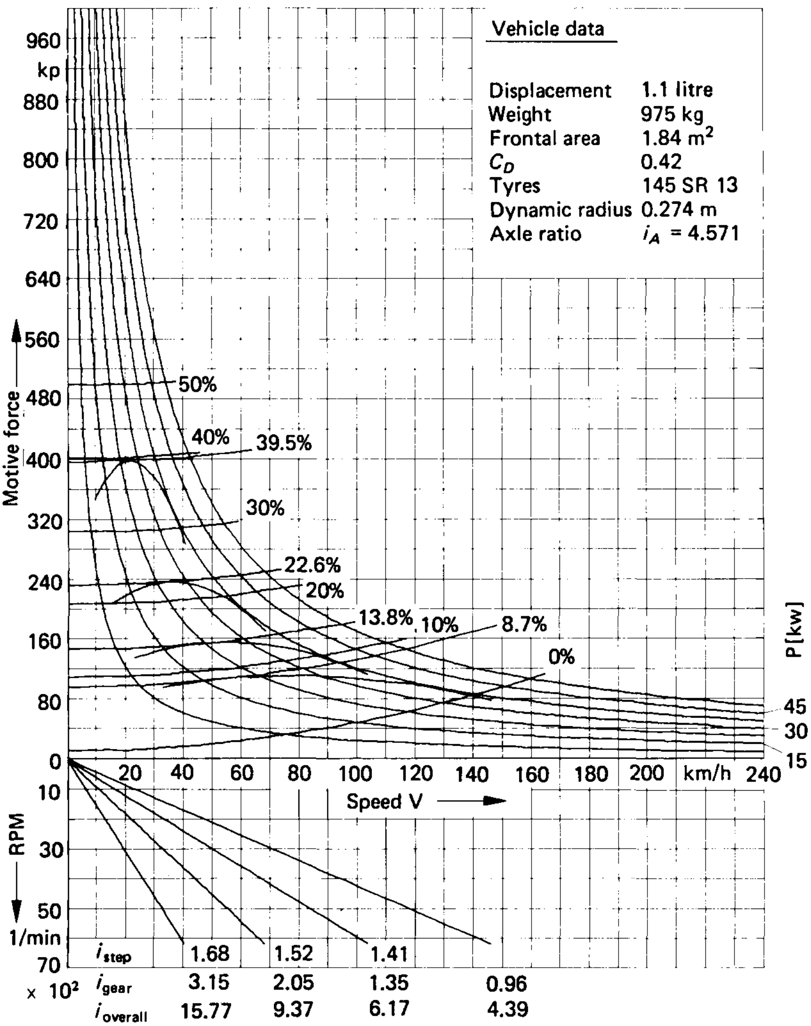


Figure 3.4 Typical motive force and speed diagram for a car

engine load in top gear, the top speed of the vehicle can be established. The propelling force remaining when the sum of rolling resistance and aerodynamic drag is deducted from available power for a given vehicle speed, can be used to climb hills and/or accelerate the vehicle in accordance with Eqn 3.1.

3.3.2 Acceleration time and elasticity

When one substitutes the above mentioned propelling force difference ΔZ for the acceleration resistance W_A in Eqn 3.8, it is possible to calculate instantaneous values for acceleration and driving speed:

$$\dot{V} = \frac{\Delta Z}{m(1 + \varepsilon_1) \times 9.81} \text{ m/s}^2 \tag{3.9}$$

$$V = 3.6 \int_0^t \dot{V} \mathrm{d}t \text{ km/h} = 2.24 \int_0^t \dot{V} \mathrm{d}t \text{ mile/h} \tag{3.10}$$

It is customary in Europe for acceleration to be quoted as the time necessary to accelerate from $V = 0$ to 100 km/h (62.5 mile/h), through the gears. The time necessary to accelerate from 40 km/h to 100 km/h (25 to 62.5 mile/h) in top gear is often used to express the elasticity of the vehicle, although other figures are also used.

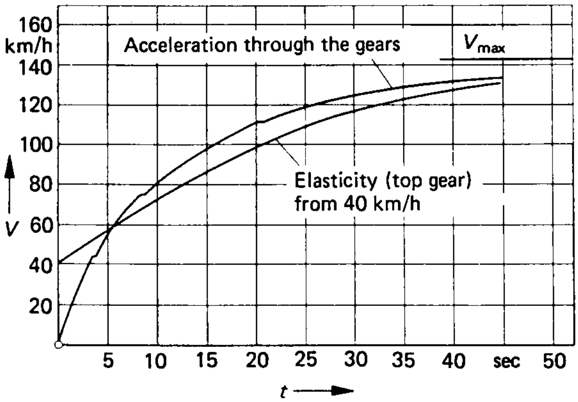


Figure 3.5 Acceleration and elasticity of a lower medium class vehicle

Figure 3.5 shows the resultant driving speed of the vehicle against time, the performance diagram and data for the vehicle being presented in Fig. 3.4.

Calculated acceleration sometimes fails to correlate with test results. This is partially due to the power output tolerance of production engines, which often vary from their published outputs by ±5 per cent. Additionally, these calculations use engine data which is measured on a static engine installation. During acceleration tests, rapid changes in rev/min occur, particularly in the lower gears.

3.4 The estimation of fuel consumption

3.4.1 Concept of estimation

During the development of a new vehicle, the various items affecting fuel economy must be evaluated in a cost benefit analysis. In order to do this, it is necessary to compute the fuel consumption in relation to all influential parameters, yet retaining the main customer requirements. The computer model can be based on a suitable predetermined driving cycle. It is helpful to use a specific fuel consumption map for an engine over a known range, although this data is generally available for steady speed engine operation only (on a test bed). The engine output while accelerating can be represented by the following supplementary computation.

3.4.2 The specific fuel consumption map

The fuel consumption of a vehicle is largely determined by the specific fuel consumption of its engine. It is dependent on the engine loading (torque, engine revolutions) and is illustrated in Fig. 3.6. The accuracy of fuel consumption computations is therefore only as good as the accuracy of the available specific fuel consumption map; it is therefore essential that a representative diagram for the engine is used for the calculations.

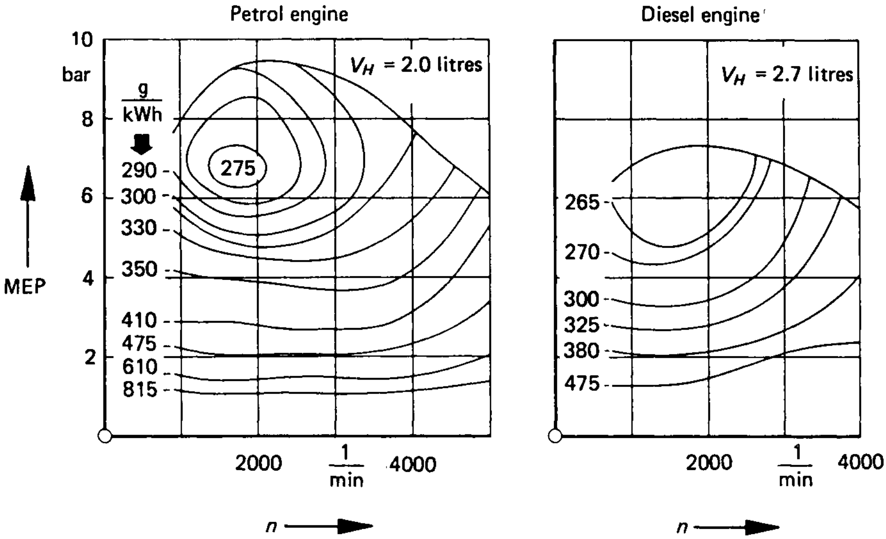


Figure 3.6 Specific Fuel Consumption (SFC) map calculated from standardized diagrams, after ref. 3.4

Often the specific fuel consumption diagram is not available at the time of calculation, particularly if theoretical variations of engine output are required. In this case hypothetical performance maps can be constructed by normalizing that for an existing engine. In effect, the mean effective pressure (MEP) and the engine revolutions n of this known diagram can be adjusted according to the ratio of the stated outputs of the engines. The hypothetical diagrams are therefore derived as follows:

$$\text{Specific fuel consumption, } b_s = b_{so} \tag{3.11}$$

$$\text{Mean effective pressure, MEP} = \text{MEP}_o \frac{P_{\text{stated}}}{P_{\text{stated o}}} \tag{3.12}$$

$$\text{Engine speed, } n = n_o \frac{n_{\text{stated}}}{n_{\text{stated o}}} \tag{3.13}$$

where the suffix 'o' refers to the values from the diagram of the existing representative engine.

Some general fuel consumption diagrams for petrol and diesel engines, derived from standard diagrams, are shown in Fig. 3.6.

The fuel consumption on idle and the amount of fuel injected by the accelerator pump are adapted to suit the various engines using the following formulae

$$\text{Idle consumption: } B_I = B_{IO}\,\frac{V_H}{V_{HO}} \tag{3.14}$$

$$\text{Accelerator pump amount: } B_A = B_{AO}\frac{V_H}{V_{HO}} \tag{3.15}$$

where V_H is the displacement volume of the engine.

3.4.3 Gear ratio matching

Most European cars have their gearing designed for an engine speed of 100 rpm above the stated peak power speed when the vehicle is travelling at top speed in top gear. Similarly, light vans are set up with engine speed 400 rpm above the stated peak power speed. The gear ratios must be

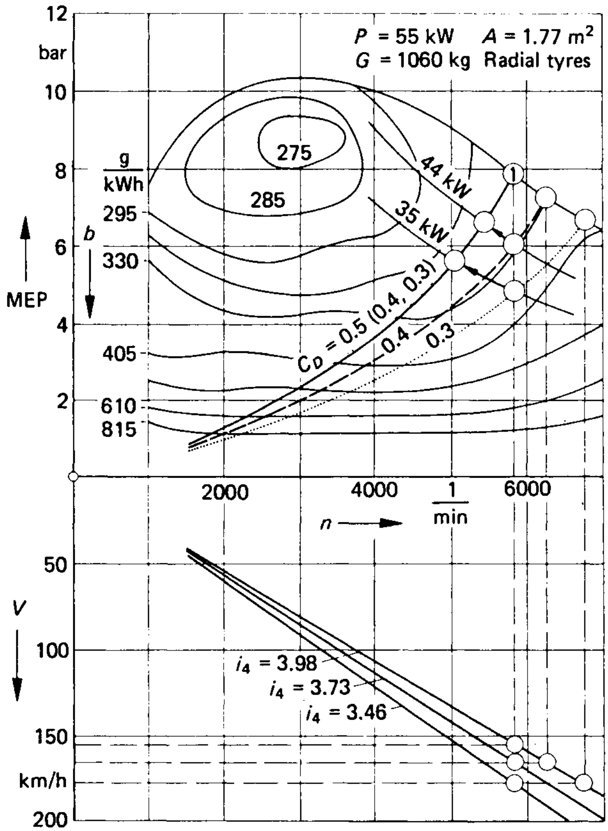


Figure 3.7 Driving resistance curves on a SFC map for various drag coefficients c_D, with and without gear ratio matching, after ref. 3.5

chosen so that acceleration and climbing capabilities meet the requirements and of course significant overlapping of adjacent gears must be ensured.

When aerodynamic drag is reduced for an existing car the gear ratio should be adjusted to suit the new top speed. Figure 3.7 shows the outcome when this ratio adjustment is not conducted. In order to illustrate the effect clearly, large steps in drag coefficient c_D from 0.5 to 0.4 to 0.3 are assumed.

For the same gear ratio of $i_4 = 3.98$ for the initial vehicle exhibiting $c_D = 0.5$ (solid line), the broken line and the dotted lines show $c_D = 0.4$ and 0.3 respectively. The effect of increased specific fuel consumption despite unchanged engine and vehicle speeds for a vehicle with reduced drag coefficient, i.e. less than 0.5, can be seen. This can mean that at lower partial load conditions, in spite of reduced performance requirements, higher fuel consumption can result. The new top speeds also lead to engine speeds which are above the maximum allowable.

When the gear ratios are changed in accordance with the previously mentioned criterion regarding the engine speed at maximum vehicle speed, the load points remain on a curve of constant power, which in the diagram (Fig. 3.7) is a hyperbolic function, back to the curve for $c_D = 0.5$.

This gear ratio modification means that the same specific fuel consumption for all three drag coefficients can be achieved for the full engine load condition. However, because the load points on the original curve, which are achieved by the gear ratio changes, lead to different driving speeds (see lower part of Fig. 3.7), a new spectrum (see Fig. 3.8) is

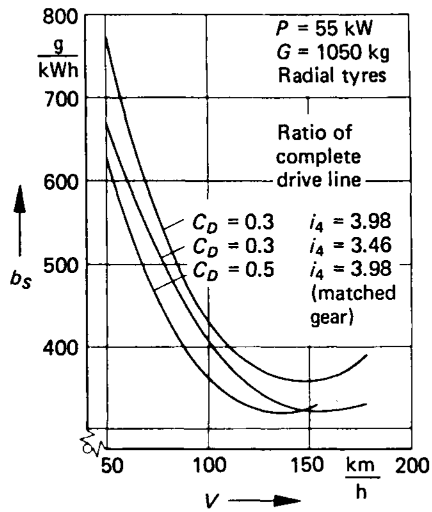


Figure 3.8 Specific fuel consumption at constant vehicle speed for a reduction of drag coefficient, with and without gear ratio matching.

obtained for the complete vehicle speed range. For clarity, only the curves for the original drag coefficient of 0.5 and the newly achieved 0.3 are shown.

As can be seen, with the help of gear ratio matching, the same specific fuel consumption under partial load can be achieved as for the original condition. In the lower partial load region specific consumption is slightly greater. Therefore, in spite of the reduced power requirement, higher fuel consumption occurs than for $c_D = 0.5$.

When driving cycles are taken into consideration the matching of the various gears is important. The following matching techniques are the most reasonable:

(a) Changing the axle ratio—whereby all gears are equally affected.
(b) Stepped matching rate for the various gears (1st gear unchanged, 2nd gear stepped to 33 per cent, 3rd gear to 57 per cent and 4th and 5th gears to 100 per cent, based on matching technique (a).
(c) Stepped matching rate for all the gears, with the vehicle weight and engine power also being taken into consideration.

The order in which the above possibilities are presented also indicates the order of practicability. The simplest to realize is possibility (a). It offers the best improvement with respect to fuel consumption, but has a negative effect in terms of acceleration in the lower gears. With an increase of 10 per cent in top speed, which results from approximately a 30 per cent reduction in aerodynamic drag, the acceleration time from 0 to 100 km/h (62.5 mile/h) would be approximately 10 per cent longer. Should this effect be unacceptable, gear matching according to method (b) is necessary.

For fuel-consumption calculations on vehicles which are also undergoing a weight reduction process along with aerodynamic drag reduction, method (c) must be used if acceleration characteristics are to be kept constant.

For first gear, a weight- or performance-based matching process must be effected to achieve acceleration characteristics similar to those of the original vehicle. A stepped matching of the second and third gears is then necessary. No weight- or performance-dependent matching is undertaken for fourth, or overdrive, therefore the overall process can be summarized as in Table 3.1.

Table 3.1 Gear matching methods

Gear	*1*	*2*	*3*	*4*	*5 (Overdrive)*
Aerodynamic drag-dependent matching (%)	0	33	67	100	100
Weight/performance-dependent matching (%)	100	67	33	0	0

When matching processes (b) and (c) are applied it must be ensured that sufficient overlap exists between the newly selected adjacent gears.

3.4.4 Driving cycles

When fuel consumption is discussed, one must differentiate between results measured during legally established artificial driving cycles, which take into account the requirements of various governmental standards (see Fig. 3.9) and the actual fuel consumption of a vehicle under daily use by its owner.

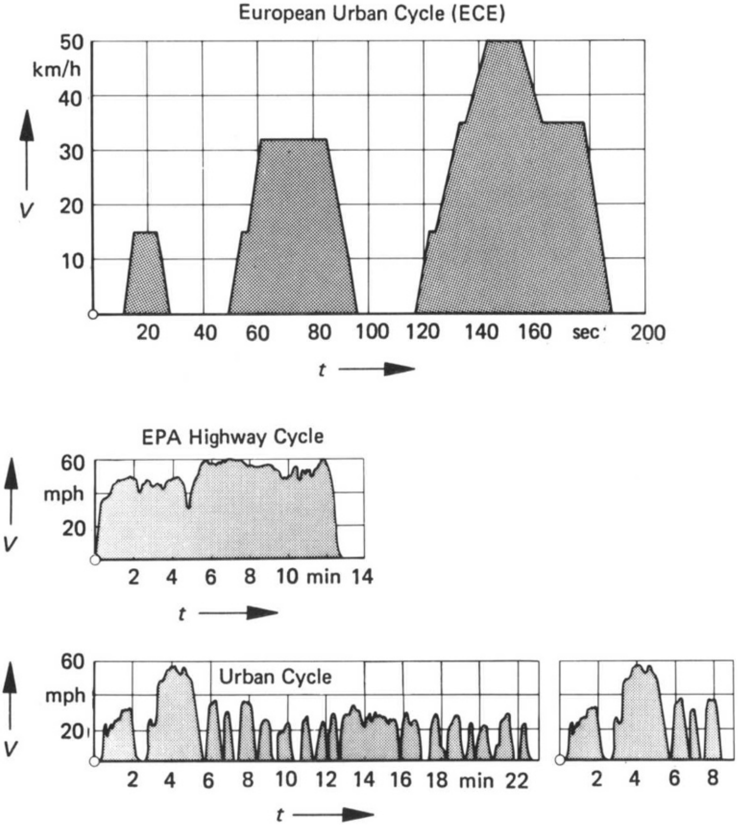


Figure 3.9 ECE and EPA driving schedules

H.-J. Emmelmann has published data[3.6] on the influence of various parameters on a driving cycle based on such normal vehicle usage. The cycle designated E'75 is based on 1975 traffic statistics for West Germany. However, because such a cycle does not represent general use, one is forced to adopt the obligatory legal driving cycles as a means of establishing the influence of aerodynamic drag and vehicle weight on fuel economy.

The so-called Euromix cycle is the recognized cycle for Europe, consisting of one-third city use consumption, measured in the ECE cycle (see Fig. 3.9), one-third at a constant 90 km/h (56 mile/h) and one-third at a constant 120 km/h (75 mile/h):

$$B_{\text{Euromix}} = \frac{1}{3}\,(B_{\text{ECE}} + B_{90} + B_{120}) \tag{3.16}$$

In the above formula B_{ECE} is obtained from dynamometer test bed tests for the middle value of certain load classes. For these, neither the actual weight of the car nor its aerodynamic drag is given any consideration, because the test was originally an emission test which used an artifical driving cycle (speed versus elapsed time).

Vehicle weight reduction only leads to a reduction in City Cycle (B_{ECE}) fuel consumption when it results in a change in test class to a lower level.

In the USA, fuel consumption is determined using a combination of test cycles based on the driving habits of that country. The 'Combined Fuel Economy' is established from the results of 55 per cent city traffic and 45 per cent highway traffic:

$$B_{comb} = 0.55\,B_{urb} + 0.45\,B_{high} \tag{3.17}$$

$$FE_{comb} = \frac{1}{0.55/FE_{urb} + 0.45/FE_{high}} \tag{3.18}$$

B_{urb} can also be established for a corresponding loading class on a brake dynamometer. The various loading classes are shown in Table 3.2.

Table 3.2 Kerb weight classes (kg)

ECE	*Sweden*	*USA-EPA*
Up to: 920	Up to: 941	Up to: 943.2
1150	1055	1056.8
1370	1168	1170.5
1600	1338	1340.9
	1565	1568.2

3.4.5 Gear change points

The points for changing up to the next gear are specified for the EPA Urban Cycle and the EPA Highway Cycle as follows:

1st to 2nd gear at V_D = 24.0 km/h (15 mile/h)
2nd to 3rd gear at V_D = 40.0 km/h (25 mile/h)
3rd to 4th gear at V_D = 64.0 km/h (40 mile/h)

Changing from 4th to 5th in suitably equipped vehicles can be conducted as follows:

4th to 5th gear at V_D = 73.6 km/h (46 mile/h)

The effect of this extra change point is that, for instance, during the City Cycle the percentage of time in 4th gear is reduced from 14.9 per cent of the complete cycle to only 1 per cent, with the remaining 13.9 per cent being driven in 5th gear. The gears remain engaged, during deceleration phases, until idle speed is reached.

The shift points for the ECE Cycle are shown in Table 3.3.

3.5 Fuel consumption and performance

3.5.1 Comparison of drag and rolling resistance

During constant speed driving on level road, the resistance to vehicle movement consists of both aerodynamic drag and rolling resistances, see

Table 3.3 Shift points for the ECE cycle

No.	*Operational condition*	*Test section*	*Acceleration* (m/s^2)	*Speed* (km/h)	*Time period of each: operational condition* (s)	*Time period of each: test section* (s)	*Total time* (s)	*Translation ratio to be used for mechanical gearboxes*
1	Idle	1			11	11	11	6 s PM + 5 s K_1[1]
2	Acceleration	2	1.04	0–15	4	4	15	1
3	Constant speed	3		15	8	8	23	1
4	Deceleration		−0.69	15–10	2		25	1
5	Deceleration with engine disengaged	4	−0.92	10–0	3	5	28	K_1
6	Idle	5			21	21	49	6 s PM + 5 s K_1[1]
7	Acceleration		0.83	0–15	5		54	1
8	Gear change	6			2	12	56	
9	Acceleration		0.94	15–32	5		61	2
10	Constant speed	7		32	24	24	85	2
11	Deceleration		−0.75	32–10	8		93	2
12	Deceleration with engine disengaged	8	−0.92	10–0	3	11	96	K_2
13	Idle	9			21	21	117	16 s PM + 5 s K_1
14	Acceleration		0.83	0–15	5		122	1
15	Gear change				2		124	
16	Acceleration	10	0.62	15–35	9	26	133	2
17	Gear change				2		135	
18	Acceleration		0.52	35–50	8		143	3
19	Constant speed	11		50	12	12	155	3
20	Deceleration	12	−0.52	50–35	8	8	163	3
21	Constant speed	13		35	13	13	176	3
22	Gear change				2		178	
23	Deceleration		−0.86	32–10	7	12	185	2
24	Deceleration with engine disengaged	14	−0.92	10–0	3		188	K_1
25	Idle	15			7	7	195	7 s PM

[1] PM = Idle with engine engaged
K_1, K_2 = 1st or 2nd gear with engine disengaged

Eqn 3.1. The proportions of these resistances as part of the total resistance vary with vehicle speed. This is illustrated in Fig. 3.10 from Hucho and Emmelmann.[3.7] The relationship between the aerodynamic drag W_D and the total resistance to movement, $W_D + W_R$, versus the driving speed V is shown; the parameter of the curves is the drag coefficient c_D. Cars, light

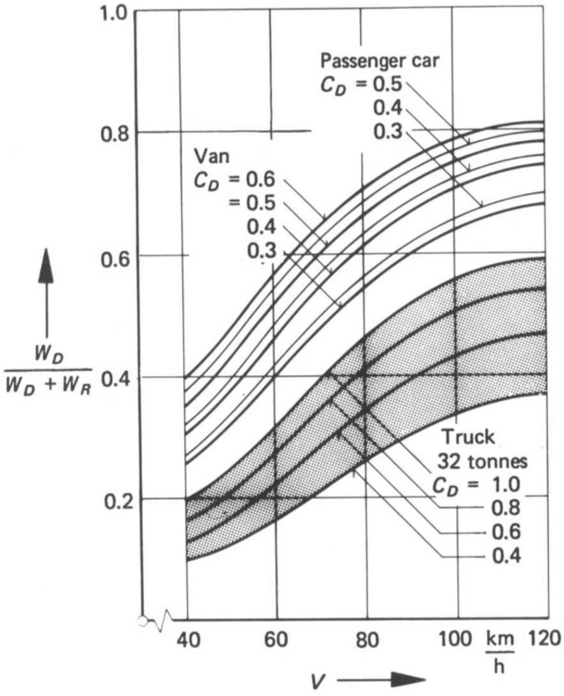


Figure 3.10 Proportion of the aerodynamic drag from the total external resistance for a car, van and truck, after ref. 3.4

vans and heavy trucks have been studied. The reduction of rolling resistance is seen to be most worthwhile for low speed (city) usage and for heavy vehicles, where the change from diagonal to radial tyres has provided an important means of reducing this rolling resistance.

3.5.2 Top speed

The top speed of a vehicle as a measure of its performance is only really meaningful in countries which do not have speed limits on their freeways (i.e. West Germany). Car magazines do of course still use this criterion in evaluating and comparing vehicles even though use of the vehicles at these speeds on public roads is only theoretical.

The top speed, despite its almost theoretical nature, and its use being hardly possible in normal traffic conditions, is a good guide to the fuel consumption characteristics of a vehicle. When two vehicles from the same weight class and having similar engine concepts are compared, the vehicle which exhibits the highest top speed will also present the lowest fuel consumption. In this respect theoretical top speed is also of interest in countries having speed restrictions, hence the interest by car magazines.

The interdependence between aerodynamic drag, top speed and engine power output is shown by the following equation, in which the vehicle weight is contained in the constant k.

$$V_{\max} = k \sqrt[3]{\left(\frac{N}{c_D A}\right)} \tag{3.19}$$

where N is measured in kilowatts.

For a medium-class vehicle, $k = 38.15$, which gives the result in kilometres per hour. Miles per hour values are obtained by using $k = 23.71$.

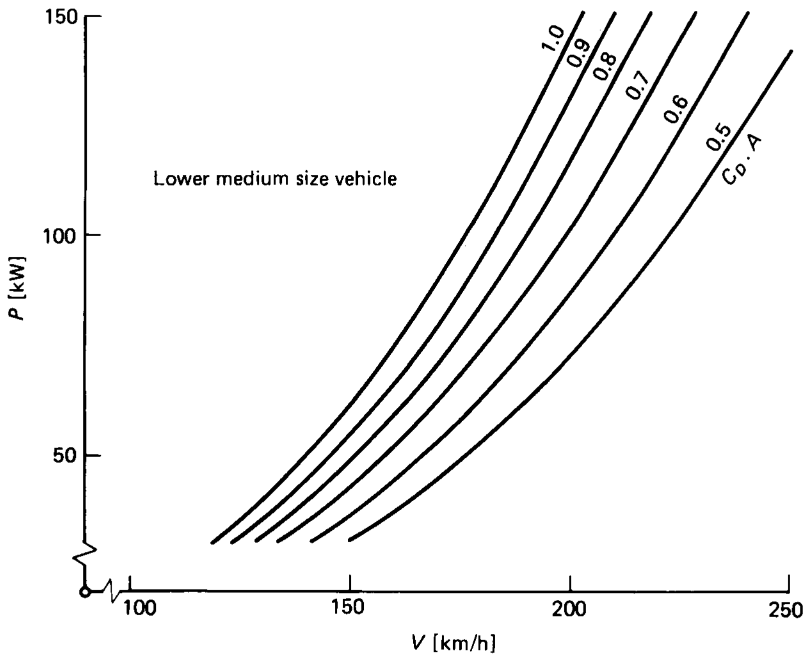


Figure 3.11 Influence of aerodynamic drag on top speed

Figure 3.11, which illustrates Eqn 3.19, suggests that aerodynamic drag reduction offers the possibility of reducing the drive train power output without loss of top speed performance. The fact that the resultant power output reduction worsens acceleration characteristics at lower speeds normally prevents its application as a means of fuel consumption reduction (see section 3.5.3).

3.5.3 Background to the fuel consumption discussion

The potential for the reduction of fuel consumption is heavily dependent on vehicle class, as is suggested by Fig. 3.10.

Reference 3.6 discusses the four main European vehicle classes, and what follows expands the discussion to extreme sub-compacts and full-size cars. The relative vehicle data are defined in Table 3.4. Light vans are dealt with in section 3.5.5 and heavy trucks in Chapter 8. The question to be answered is to what extent the various parameters that influence fuel consumption can be reduced.

Table 3.4 Definition of sub-compact and full-size cars

Vehicle	*G*	*P*
Sub-compact	900 kg (~2000 lb)	40 kW
Full-size car	1800 kg (~4000 lb)	150 kW

In section 3.5.2 it is shown that by reducing aerodynamic drag the power required can be reduced without significantly affecting top speed. However, when one compares the aerodynamic drag development of recent years (see Fig. 1.53) with the engine sizes being used, one comes to the conclusion that buyers have continued to purchase the same power units, priority being given to higher top speed. They could have turned to the next smaller engines in the vehicle programmes with a view to obtaining better fuel consumption, which has been shown by Janssen and Emmelmann[3.6]—a 10 per cent drive-power reduction could mean approximately 4 per cent lower fuel consumption. But since this drive-power reduction also reduces the acceleration characteristic of vehicles, which is considered of foremost importance at the present time, it appears unrealistic to discuss this possibility any further.

Continual efforts are still being made to reduce vehicle weight, often through the use of more expensive materials. This weight reduction has of course an influence on the fuel consumption, but an 'official' gain is rarely achieved. That is to say, only when testing on the brake dynamometer is brought into a lower vehicle test loading class is a reduced fuel consumption, of 0.3 to 0.4 litres/100 km in B_{urb}, or B_{ECE} in Europe, measurable.

As a result of attempts to reduce vehicle weight many conflicting design goals occur, such as noise level and mechanical strength (crash test). It is generally true to say that weight reduction, independent of this test class change aspect, rarely allows for greater than a 3 per cent consumption reduction, for which a weight reduction of more than 10 per cent is necessary.

In refs. 3.7 and 3.8 Emmelmann shows that the product of the aerodynamic drag coefficient c_D and the frontal area A can be reduced to a value of $c_DA \times 0.6\,\mathrm{m}^2$ (Fig. 3.12) for all vehicles, independent of vehicle class, when the available aerodynamic 'know how' is applied effectively. The GM-Opel Corsa SR is proof of this, being in autumn 1983 the first production vehicle to exhibit a value slightly lower than the above mentioned $c_DA = 0.6\,\mathrm{m}^2$. This has been achieved in spite of a vehicle length of only 3.60 m. It is also expected, within the foreseeable future,

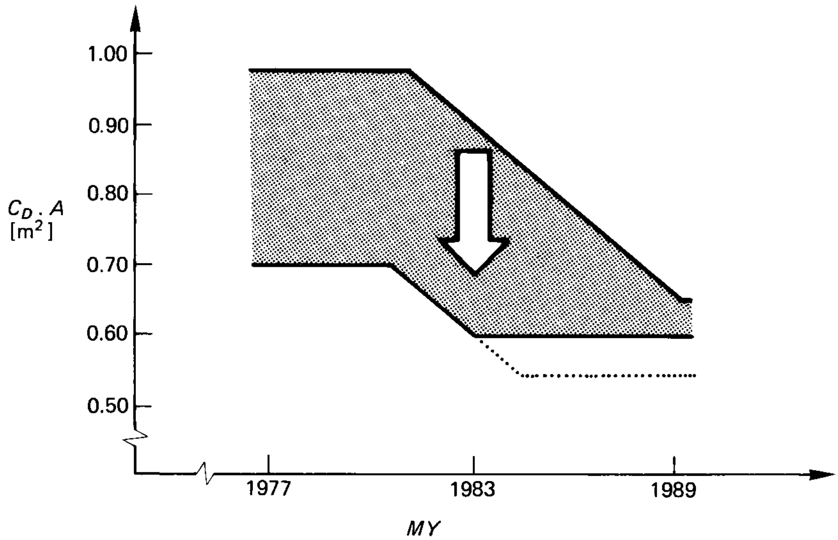


Figure 3.12 Trend of aerodynamic development, after ref. 3.8

that some particularly aerodynamically designed vehicles will exhibit drag areas $c_D A$ of approximately $0.55\,\mathrm{m}^2$.

This discussion shows that by far the biggest potential is offered by aerodynamic drag reduction. Since some current cars still exhibit values of $c_D A = 1.0\,\mathrm{m}^2$, it seems to be realistic to discuss aerodynamic drag reduction potentials of 40 per cent and more. Conversely, the discussed weight reduction potential of 10 per cent is still hard to achieve.

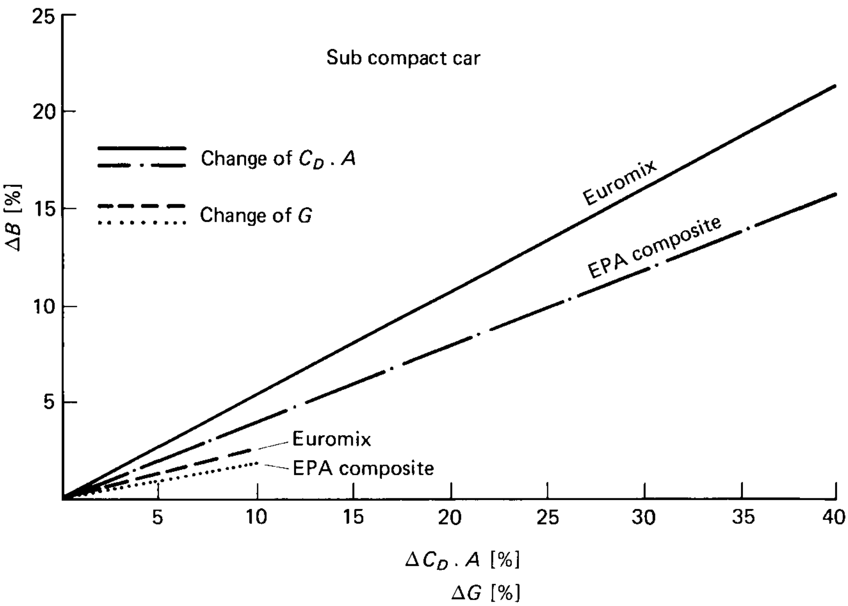


Figure 3.13 Impact of changes in aerodynamic drag and vehicle weight on fuel consumption for a sub-compact car

3.5.4 Impact of aerodynamic drag and weight on fuel consumption

The results presented in this section represent on-road driving for vehicle speeds corresponding to ECE and EPA driving schedules. The correlation between daily on-road use and ECE or EPA dynamometer tests is not addressed.

Figure 3.13 shows the impact of changes in aerodynamic drag and vehicle weight on fuel consumption for a sub-compact car. It is evident that a linear dependence exists for all considered driving schedules. Due to the constant velocity sections of the Euromix Cycle, the influence of the aerodynamic drag under this cycle is greater than for the EPA Composite.

Even so, the influence of a weight reduction is slightly higher for the Euromix Cycle compared to the EPA Composite Cycle because the ECE Urban Cycle includes some very steep velocity gradients (Fig. 3.9).

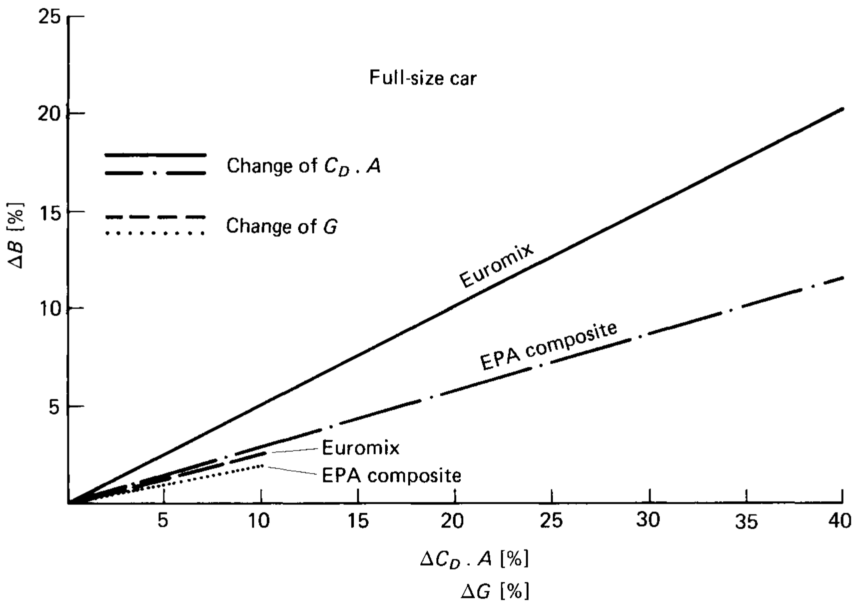


Figure 3.14 Impact of changes in aerodynamic drag and vehicle weight on fuel consumption for a full-size car

Similar results can be observed for the full-size car; see Fig. 3.14. In this case, the influence of reduction of the aerodynamic drag on the EPA Composite is less than for the sub-compact, while a reduction of weight leads to a larger percentage fuel consumption decrease than for sub-compacts.

Figures 3.13 and 3.14 clearly show that the potential for reducing fuel consumption by reducing aerodynamic drag is far greater than the potential by weight reduction, even though the scope available for reducing the two parameters is vastly different. Reduction of vehicle weight does of course influence acceleration and rolling resistance.

Detailed information concerning the impact of changes in aerodynamic drag and mass (weight) reduction for a percentage change in $c_D A$ has been elaborated by Sovran.[3.10] The analysis is based on the tractive energy

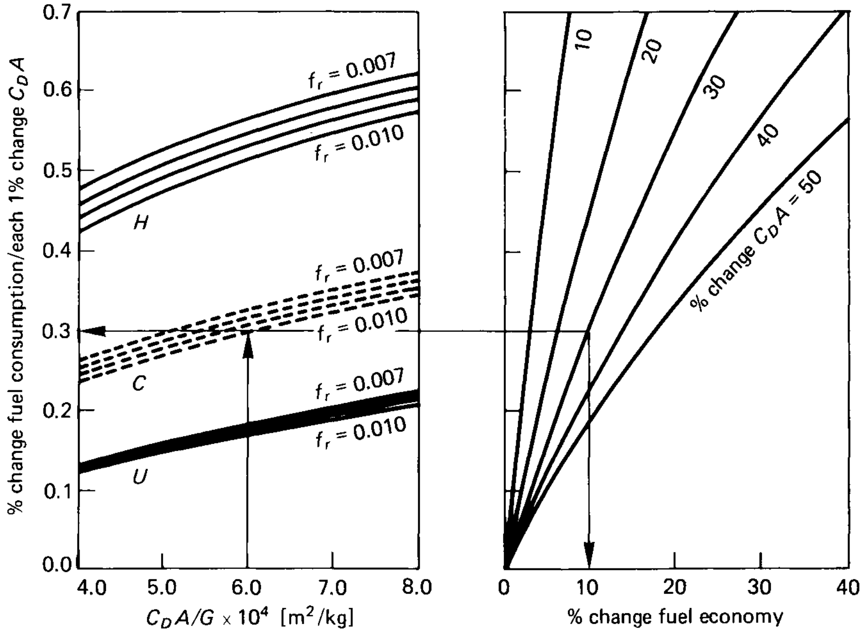


Figure 3.15 Impact of changes in aerodynamic drag on fuel consumption for the EPA schedules, after ref. 3.10

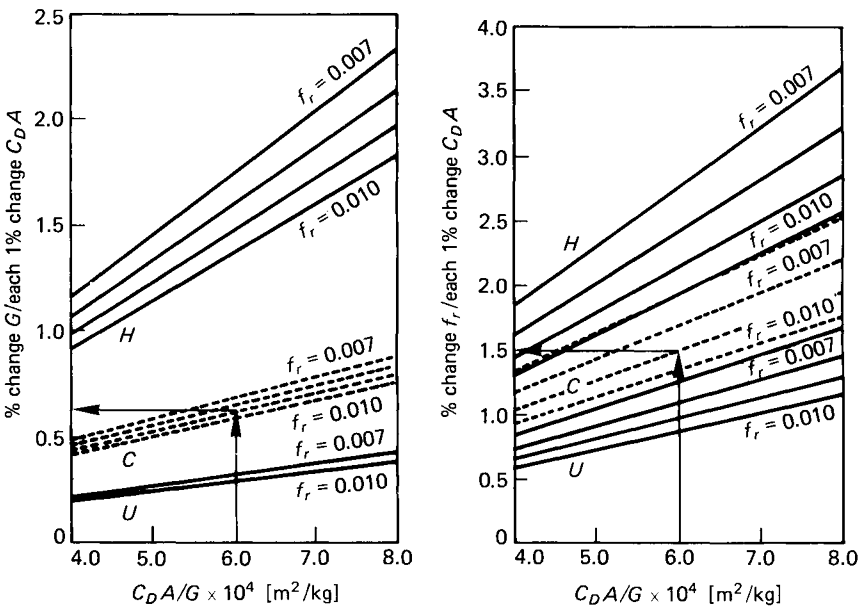


Figure 3.16 Mass and tyre coefficient reduction required for equivalent fuel saving for a 1 per cent change of aerodynamic drag, after ref. 3.10

required by vehicles to negotiate a driving schedule (see also Sovran and Bohn[3.11]).

Figure 3.15 shows the corresponding impact of changes in aerodynamic drag on fuel consumption for the three EPA Schedules, Highway (H), Urban (U) and Composite (C). With the help of the appropriate diagram the impact on fuel economy can be calculated. The control parameter of the curves in the left diagram is the rolling resistance (tyre) coefficient f_r. Figure 3.16 shows on the left the mass reduction and on the right the tyre coefficient reduction required to achieve the equivalent fuel saving of a 1 per cent change in aerodynamic drag. The working charts shown in Figs 3.15 and 3.16 are valid for any vehicle driving the EPA schedules.

3.5.5 Fuel consumption of light vans

The influence of aerodynamic drag on the fuel consumption of light vans has been studied by Hucho and Emmelmann.[3.4] Table 3.5 shows the essential data of the vehicles investigated. For comparison purposes both

Table 3.5 Data for fuel consumption calculations (after ref. 3.4)

Item		*Petrol engine*	*Diesel engine*
Frontal area		4 m²	
Kerb weight		1600 kg	
Gross vehicle weight		3000 kg	
Tyres		195 R 14 C	
Gearbox efficiency		0.9	
Power output		50 kW	
at rpm		4300 min^{-1}	3600 min^{-1}
Displacement volume		2000 cm^3	2700 cm^3
Original drive ratios	i_1	24.39	18.32
	i_2	12.31	9.24
	i_3	7.61	5.39
	i_4	4.87	3.67
Rolling resistance coefficient		0.0130 to 0.0145 according to speed	

petrol and diesel engined vehicles were studied. The fuel consumption of these vehicles with unchanged drive ratios over the speed range is shown in Fig. 3.17. The control parameter of the curves is the aerodynamic drag coefficient c_D, which is established in each case for the vehicles with half payload, this being the load condition used when measuring fuel consumption in accordance with DIN and the more recently adopted ECE standards.

These computed data correlate well with measured results, as was demonstrated by Hucho, Janssen and Schwarz[3.12] on passenger cars. The advantage of the diesel engine in offering low fuel consumption, especially under low engine loading, is clearly demonstrated.

The influence of aerodynamic drag on fuel consumption is considerable. For the petrol engine under constant speed conditions its effect is more evident than for the diesel. Adjusting the ratios of the gearbox would offer even better fuel economy potential along with drag reduction.

In order to evaluate the fuel consumption characteristics of light vans the

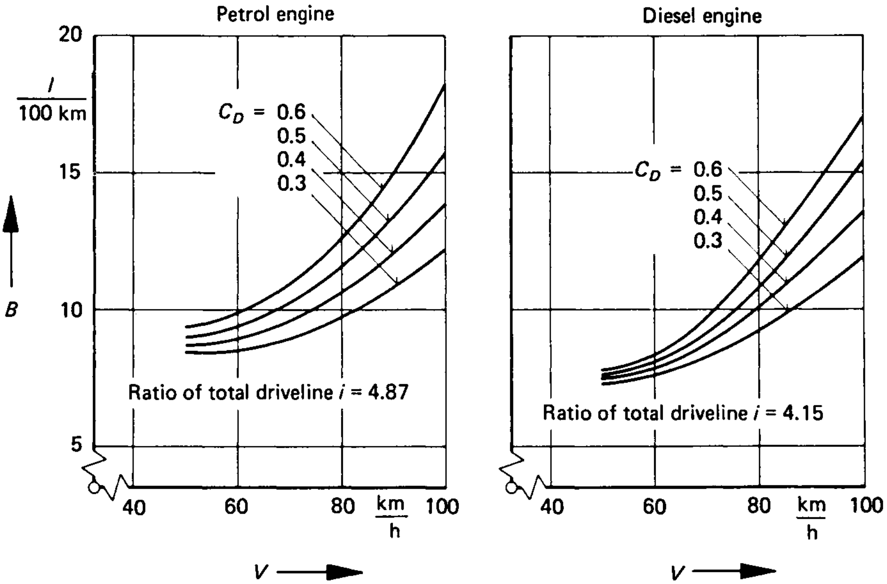


Figure 3.17 Impact of changes in aerodynamic drag on fuel consumption of a van, after ref. 3.4

EPA Combined Cycle, consisting of 45 per cent highway driving and 55 per cent city driving, was used. The reason for this choice is that light vans are driven as cars in similar traffic conditions, their use being centred around both town and local traffic environments. No freeway input was made as no information concerning this type of usage was available. For the same reason hill climbing was also not taken into consideration.

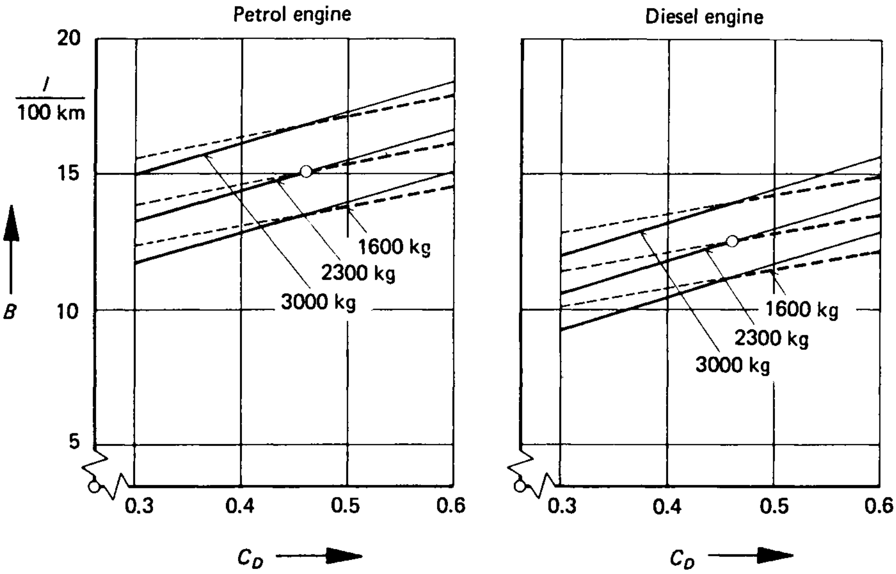


Figure 3.18 Fuel consumption of vans for the EPA Composite Cycle, after ref. 3.4

The fuel consumption for this cycle is shown in Fig. 3.18 as a function of the drag coefficient. Starting point for the study is the average drag coefficient of vans, 0.46, which was established by the measurement of 17 vans.[3.4] In this case the gearing is matched in order that the stated engine speed for maximum power output plus 400 rev/min is achieved when the vehicle is driven at top speed with half payload. If aerodynamic drag is reduced and the gear ratios are adjusted suitably, the continuous line shows the achievable fuel consumption.

The fuel saving is then approximately 50 per cent greater than that obtained with no matching (dotted line). Because light vans are essentially lower in performance than cars, when frontal area and weight are considered, their engines operate closer to the full load curve on the specific fuel consumption map during the driving cycle. As a result of this, more favourable specific fuel consumption is achieved, resulting in relatively low absolute consumption values. Because the increase in specific fuel consumption as a result of reduced load in this part of the diagram is less apparent than under partial load conditions, the effect of the increased specific consumption resulting from drag reduction is not so detrimental to the achieved absolute fuel consumption improvement. This leads to greater effectiveness of drag reduction in lowering fuel consumption on this type of vehicle than on passenger cars.

The effectiveness of the drag reduction on the fuel consumption B, for half payload, in the case where the gear ratios are matched is

$$\text{Petrol engine} \quad \frac{\Delta B}{B_o} = 0.40 \frac{\Delta c_D}{c_{Do}} \tag{3.20}$$

$$\text{Diesel engine} \quad \frac{\Delta B}{B_o} = 0.50 \frac{\Delta c_D}{c_{Do}} \tag{3.21}$$

where B_o and c_{Do} are values for the fuel consumption and drag coefficients before the drag improvement Δc_D. The effectiveness of the drag reduction is therefore greater for a van with a diesel engine than for one with a petrol engine. The reason is that the given driving cycle includes a considerable amount of engine operation in a specific consumption area (in the specific fuel consumption map) where the difference in consumption between petrol and diesel engines is considerable.

If, in the given example, the drag coefficient c_D was reduced from 0.46 to 0.30, a fuel consumption reduction for a petrol-engined vehicle of 14 per cent would be returned. For a diesel-engined variant, a reduction of 17 per cent would result.

3.6 Outlook

The continual depletion of crude oil supplies will certainly ensure gradual fuel price increases. Alternative fuels based on coal and alcohol will have little effect on the situation, but they may help to reduce the effect. Low fuel consumption will therefore be an important selling point.

Vehicles with favourable fuel consumption will be more expensive to manufacture than current vehicles. Further development of lighter vehicles will almost certainly lead to the use of more expensive materials. Aerodynamic requirements such as more rounded glass or flush window areas also add to product costs.

In order to decide which product price increases can be tolerated in the market place to obtain reduced fuel consumption, it is advisable to use a customer-oriented cost-benefit calculation. This ensures that only constructive measures that are economically viable in the eye of the customer are adopted.

3.7 Notation

A	frontal area
B	fuel consumption (l/100 km)
B_A	accelerator pump amount
B_I	idle consumption
G_N	normal force (weight)
P	engine power
V	vehicle speed
$\dot{V}$	acceleration
V_H	displacement volume
V_S	side-wind speed
V_∞	air speed
W_A	accleration resistance
W_C	climbing resistance
W_D	aerodynamic drag
W_R	rolling resistance
Z	tractive force
b_s	specific fuel consumption
c_D	drag coefficient
c_T	tangential force coefficient
f_r	rolling resistance coefficient
i	gear ratio
m	mass
n	engine speed
β	yaw angle
η_A	axle efficiency
η_G	gearbox efficiency
ρ	air density
φ	ascent angle