Reduction of Aerodynamic Resistance of Heavy Vehicles and Effect on Fuel Economy

by J. J. du Buisson* and P. J. Erens**

Abstract

A number of commercially available aerodynamic drag reducing devices have been evaluated for different angles of incidence using wind tunnel model tests.

The information gained has been incorporated into a computer program simulating the operating conditions of a typical tractor and trailer combination. The program includes statistical information on wind strength and direction as well as terrain conditions for a typical long distance haul on South African roads as well as engine performance and gear ratio data.

In this way it was possible to make realistic assessments of the various configurations for reduction of aerodynamic drag with a view to possible fuel savings.

Nomenclature

a	– coefficient in rolling resistance equation	
A	– frontal area of tractor-trailer combination	[m^2]
b	– coefficient in rolling resistance equation	[s/m]
C_d	– vehicle drag coefficient	
C_d'	– wind averaged drag coefficient	
c_t	– turning stiffness factor	[N/radian]
c_v	– transmission viscous loss factor	
e	– transmission system efficiency	
F_a	– wind drag	[N]
F_c	– turning drag	[N]
F_g	– gravitational drag (positive or negative)	[N]
F_n	– nett accelerating force	[N]
F_r	– rolling resistance	[N]
g	– gravitational acceleration	[m/s^2]
k_s	– altitude correction factor	
M	– vehicle mass	[kg]
n	– number of chosen wind directions	
P_e	– engine power	[kW]
P_m	– maximum engine power	[kW]
$p(V_w)$	– wind velocity probability function	
$p(\varphi)$	– wind direction probability function	
$p(V_w;\varphi)$	– characteristic wind function	
r_c	– radius of curvature of vehicle path	[m]
V	– relative wind velocity	[m/s]
V_m	– maximum wind velocity	[m/s]
V_T	– vehicle road velocity	[m/s]
V_w	– wind velocity	[m/s]
W_e	– effective vehicle mass	[kg]
z_o	– altitude above sea level	[m]
φ	– wind direction angle	[deg]
ψ	– relative air velocity angle	[deg]
θ	– road incline angle	[deg]
ρ	– air density	[kg/m^3]

Introduction

The overall efficiency of operation of a heavy duty truck used for long distance haulage is determine by a number of factors including specific fuel consumption, rolling resistance, transmission system efficiency, type of terrain traversed, aerodynamic drag and driver characteristics. Ever increasing maintenance and fuel costs make it essential for the transport operator to make use of technological developments to save on expenditure wherever possible in order to remain competitive.

One of the areas where some savings can be achieved is in the reduction of aerodynamic drag. However, claims made in the past in regard to the magnitude of the savings which can be achieved have often been highly exagerated. A simple calculation will show that aerodynamic drag of a truck at 80 km/h represents roughly 50% of the total drag indicating that large savings can be achieved in this area. However, taking into account the factors mentioned above a reduction in aerodynamic drag will not necessarily lead to a proportional reduction in fuel consumption.

The aim of the project described here was therefore to estimate the real reduction in fuel consumption which could be achieved using various commercially available drag reducing devices. This was done by measuring drag on a truck model fitted with these devices at various angles of wind incidence and using this information in a computer simulation program which takes into account the various factors mentioned above and which simulates the operating pattern of a typical tractor-trailer combination on a long distance haulage route.

Definition of wind drag and its contribution total drag

The aerodynamic drag on a road vehicle in the direction of travel is always defined in terms of the relative velocity of the air, the frontal area and a drag coefficient which is a function of the angle between the airstream and the vehicle axis, so that

$$F_a = \tfrac{1}{2}.\rho.V^2.A_f.C_d \qquad (1)$$

Since there is usually a wind present at an angle to the direction in which the vehicle is travelling V is the velocity of the air relative to it. V is therefore the vector sum of the vehicle and wind velocity (see Figure 1(a))

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An example of the variation of C_d as a function of wind angle is shown in figure 1 (b).

The other component of drag on a vehicle travelling on a level road is the rolling resistance which is given by an equation of the form

$$F_r = (a + b.V_t).M.g \tag{2}$$

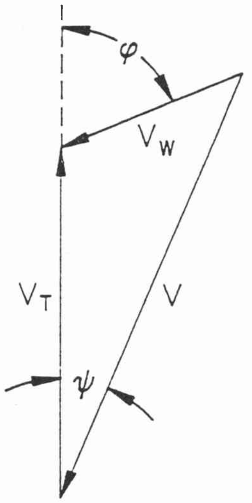

Figure 1(a) – Vector diagram illustrating relative wind velocity which is the vector sum of the vehicle and the wind velocities.

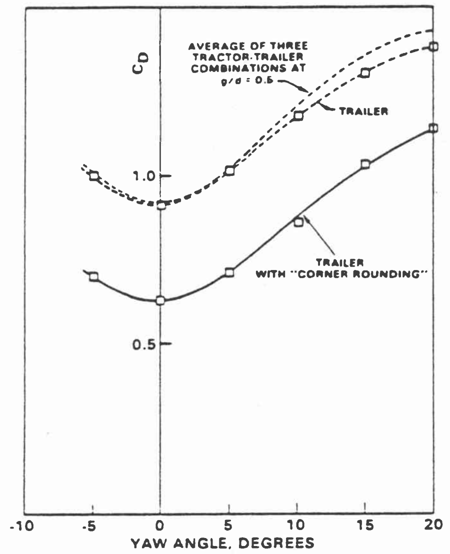

Figure 1(b) – Typical curve showing C_d as a function of yaw angle.

The total power required from the vehicle's engine will therefore be given by

$$P_e = (V_t/e).(F_a + F_r) \tag{3}$$

For a typical truck of 32 000 kg mass, $A = 9{,}5\ m^2$ with C_d of 0,95, $e = 0{,}86$, $a = 0{,}0061$ and $b = 0.000134$ the above value of aerodynamic drag being 50% of the total drag at roughly 80 km/h with no wind will be obtained. The power required from the engine can be calculated from equation (3) as 142,5 kW whereas the fuel consumption rate will depend on the engine speed and the specific fuel consumption under these conditions.

Using the above formulae and a given set of gear ratios together with engine performance curves one can easily be misled into calculating the fuel consumption for an average speed of say 75 km/h. This calculation is likely to give highly optimistic results as it ignores prevailing wind as well as incline conditions over the haulage route.

It therefore becomes clear that it is necessary to carry out an incremental calculation over the entire route in order to obtain a realistic value for fuel consumption. This can be done for a particular route or using statistical information in regard to a number of routes and wind conditions over the period of a year.

Model tests

The computer simulation program requires reliable information on the drag coefficients of the different devices available as a function of wind angle of incidence. For this reason wind-tunnel model tests were carried out on a 1/20th scale model of a well known tractor-trailer combination shown in Figure 2. The model is constructed so that drag reducing devices can be fitted to either the cab or trailer.

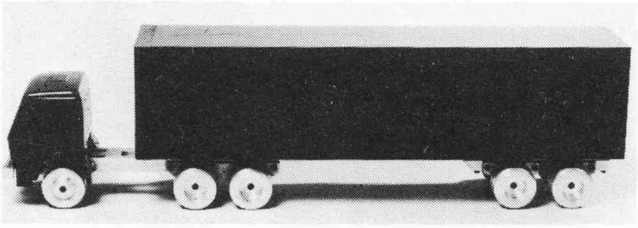

Figure 2 – 1/20th Scale tractor-trailer wind tunnel model.

The model tests were carried out in the University of Stellenbosch low speed wind tunnel which has a 1 × 1,4 meter cross section and a maximum test section velocity under the present configuration of about 80 m/s. The Reynolds number attainable with this velocity is in the order of a fifth of fullscale value which was not ideal but the results were found to be satisfactory. This could be shown by carrying out measurements at different air velocities and therefore Reynolds numbers. It was found that the drag coefficients determined varied little with Reynolds number. The only way to solve this problem is to use fullscale tests or at least a larger model which in this case was not possible.

The model mounted in the test section can be seen in Figure 3.

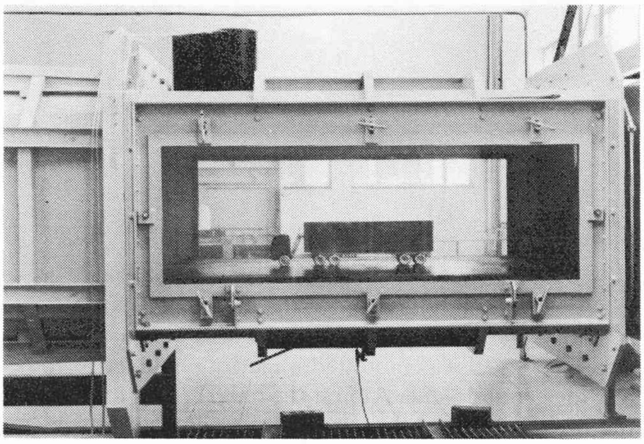

Figure 3 – Model in wind tunnel test section.

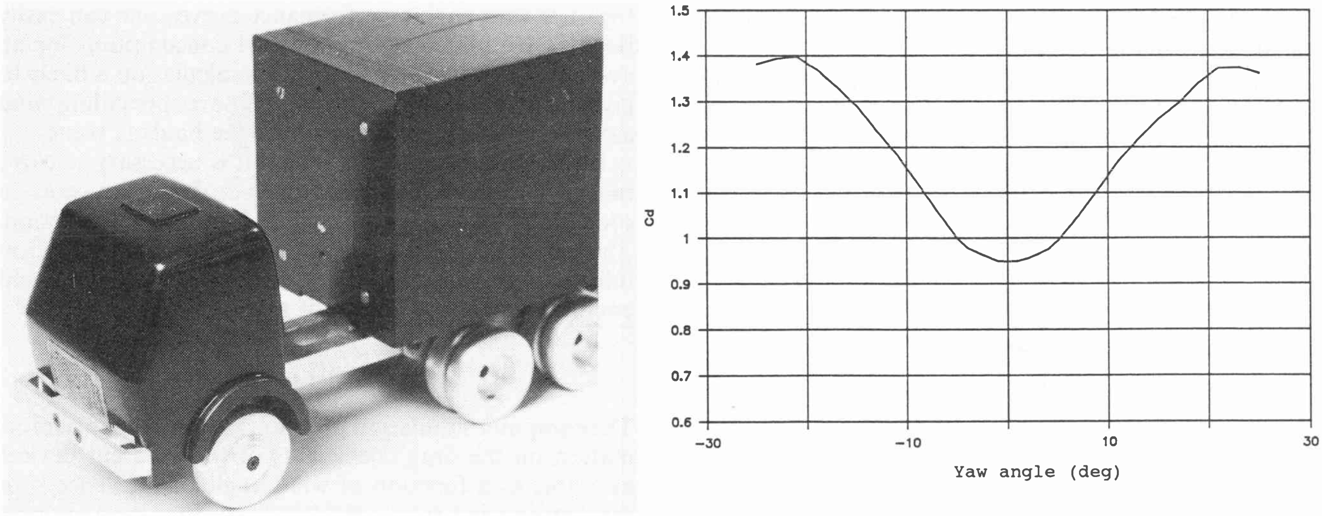

Figure 4(a) – Standard model configuration and drag coefficient as a function of yaw angle.

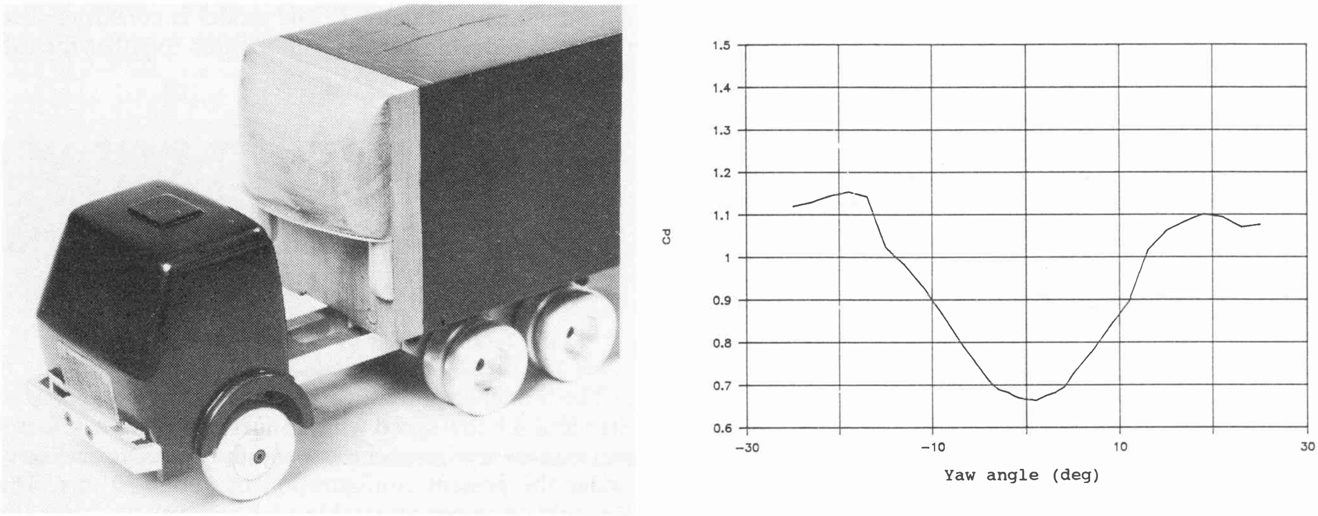

Figure 4(b) – Model with trailer-mounted device and drag coefficient curve.

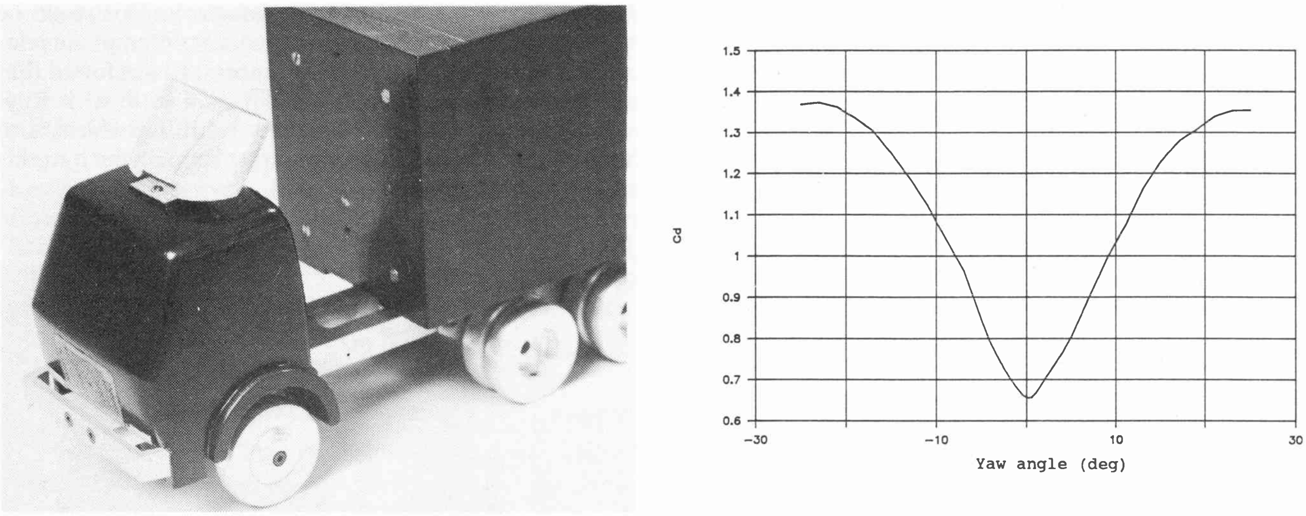

Figure 4(c) – Model with cab-mounted device and drag coefficient curve.

A fixed groundplane above the tunnel wall was used to minimise the boundary layer thickness beneath the model. The model was mounted on a drag balance fixed below the groundplane. At first the balance was attached to the tunnel wall but due to excessive vibration problems the mounting was later moved to the concrete floor below the tunnel. More detail on the testing method can be found in du Buisson [1].

Model tests were carried out on fifteen different drag reducing device configurations including the unmodified version which was considered to be the reference model. The drag as a function of wind incidence angle is shown in figures 4(a) through 4(c) for the reference model and two other configurations.

Drag reductions at zero incidence angle of up to 29% were obtained in some cases as can be seen in figure 5. The meaning of C_d' is explained below.

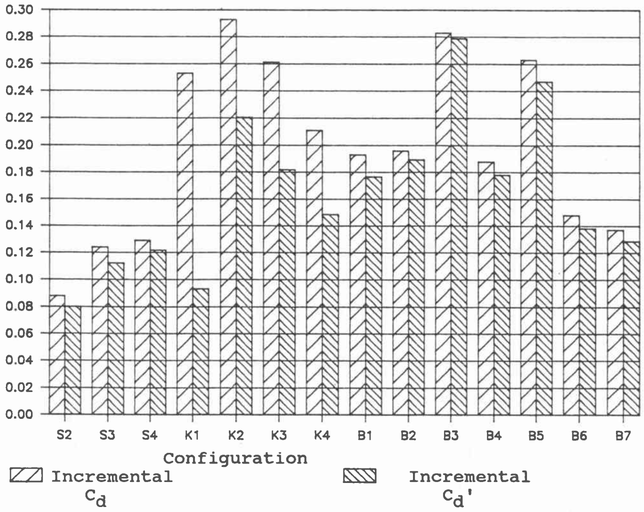

Figure 5 – Decrease of C_d and C_d' for various drag reducing devices relative to standard configuration.

Computer simulation of truck operation

A comprehensive simulation program to simulate the operating conditions of a tractor-trailer unit has to include the following:

(a) Drag coefficient as a function of relative wind direction.

(b) Correlations for rolling resistance as a function of vehicle mass and road velocity. If possible this should include effects of side forces due to wind and road curvature.

(c) Statistical information on wind strength and direction.

(d) Road profile information in terms of distance, gradient and altitude.

(e) Gear ratios and engine characteristics in terms of power, speed, torque and specific fuel consumption.

In the simulation program described here certain assumptions had to be made in terms of driver gear changing and travelling speed habits. This is a factor which contains the largest degree of uncertainty while it can drastically affect fuel consumption. However, it is expected that the driver will operate the truck within certain speed limits in terms of road speed and engine r.p.m.

For the purpose of the exercise described here a wind averaged drag coefficient, C_d', was determined using statistical information on wind direction and strength over the whole of South Africa and Namibia. It would however be possible to determine this value for a particular route, if required. The advantage of using this factor is that the wind drag is determined from this coefficient and the vehicles road velocity.

According to a method described by Ingram [2] the wind can be characterised by two probability functions representing wind strength and direction so that:

$$p(V_w;\varphi) = p(V_w).p(\varphi) \qquad (4)$$

The probability that the truck will expect wind from a particular direction, $p(\varphi)$, is fairly simple and equal to $1/n$ where n is the number of wind directions considered. This is due to the fact that in South Africa the probability of direction of travel was considered equal for all directions. However, in simulating a particular route such as Johannesburg to Cape Town this would have to be calculated using a statistically averaged direction of travel and wind information along the route.

Since the wind information is gathered ten meters above ground level a correction is also made regarding wind strength so that:

$$V_w = 0.82.V\ (10\ \text{meters}) \qquad (5)$$

The relative wind velocity is in turn given by:

$$V = V_t[1 + (V_w/V_t)^2 + 2.(V_w/V_t).\cos\varphi]^{\frac{1}{2}} \qquad (6)$$

Using the values of $p(V_w)$ and $C_d(\psi)$ obtained from the model tests C_d' can be calculated from:

$$C_d' = \int_0^{2\pi}\int_0^{V_m} C_d(\psi)\ [1 + (V_W/V_T{}^2 + 2.(V_W/V_T.\cos\psi].p(V_W,\varphi)\ dV_w.d\varphi \qquad (7)$$

The statistically determined drag of the vehicle at any speed is then given by:

$$F_a = \tfrac{1}{2}.\rho.V_t^2.A.C_d' \qquad (8)$$

As mentioned earlier the rolling resistance can be calculated using an equation such as (2). The first coefficient, a, can be further expressed as a constant plus other terms related to tyre pressure, temperature and road texture. For the purpose of the simulation the latter three factors were assumed constant under normal operating conditions so that "a" is essentially constant. The values of a and b given above in equation (2) are an average of values found in the literature.

In addition to the normal rolling resistance of the tyres an additional equation which accounts for the effect of turning can also be employed. Renouf [3] gives the following expression to account for this drag:

$$F_c = W^2.V_t^4/(r_c^2.n.c_t) \qquad (9)$$

with $c_t = 95552$ N/rad which represents a turning stiffness factor.

It was found that the magnitude of the above component is seldom more than 3% under open road conditions.

The effect of side wind force on rolling resistance can similarly be accounted for but was found to be so small that it was altogether ignored in the simulation.

The remaining two forces which influence the forward motion of the vehicle are the driving force exerted by the tyres and gravitational force given by:

$$F_g = W.g.\sin\theta \qquad (10)$$

The nett accelerating force is thus given by:

$$F_n = F_t - [F_r + F_a + F_g + F_c] \qquad (11)$$

The thrust force available at the wheels is deduced from the engine power and the various losses occurring between the engine and the driving wheels.

It can be shown that this force is given by:

$$F_t = (e_0 . k_s/V_t(.[P_e.(1 + c_v) - c_v.P_m] \qquad (12)$$

where e_0 = Transmission system efficiency at full load

The altitude correction factor is given by Smith [4] as:

$$k_s = [1 - 0{,}1312.z_0/1000] \qquad (13)$$

The acceleration of the vehicle can now be determined from:

$$A = F_n/W_e \qquad (14)$$

W_e, the effective mass represents the total mass of the vehicle plus a value related to the inertia of all the rotating components such as wheels, transmission components and engine. The method used to determine this value has been described by Smith [4].

Simply stated equation (14) is used to calculate the progress of a vehicle over a predetermined route taking into account the limits of road-speed and engine-speed which would normally be adhered to. It is thus determined at which instant gear changes will take place and during which periods constant speed will be maintained.

In the example calculations described below a 52 km route between Cape Town and Paarl was used. Figure 6 illustrates the road profile, while figures 7 through 10 demonstrate the respective velocities, gears selected, fuel consumptions and engine speeds obtained. The total fuel consumed over the route can be determined by integrating the fuel consumption rate curve over the entire route.

Simulations were carried out for a vehicle without drag reducing devices and with the devices tested in the wind tunnel. It was found that the devices illustrated in Figures 4(b) and 4(c) gave the greatest reductions in fuel consumption giving reductions of 7,3% and 6,0% respectively.

Discussion of results

From the graphs of drag coefficient illustrated it can be seen that the aerodynamic drag of a truck can easily be reduced by 25 to 30% which makes a reduction of 12,5 to 15% in the total drag theoretically possible. However, due to the various operating parameters involved the actual savings in fuel consumption are reduced percentagewise to about half the reduction in drag.

Clearly, if greater confidence in the simulation program is to be achieved it is necessary that road test measurements should be made as well in order to verify the results. This is no easy task since details of altitude and curvature for whole routes e.g. Johannesburg to Durban are required for the simulation while the measurements will only be meaningful if a large number of samples are

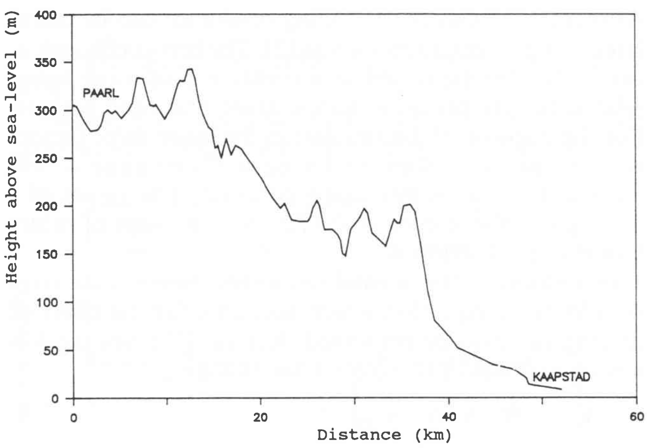

Figure 6 – Route profile used in simulation.

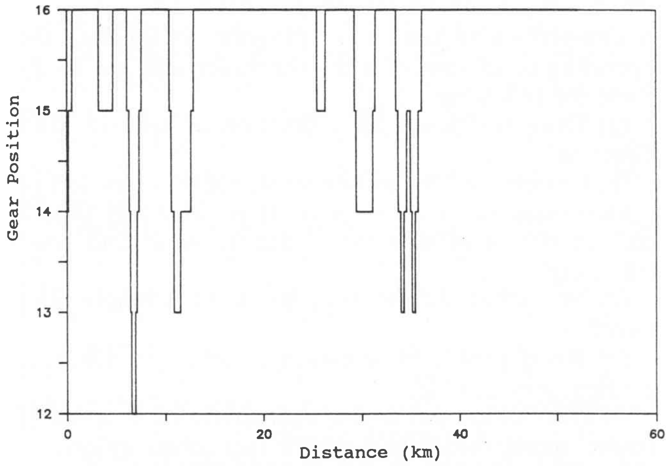

Figure 8 – Gear position along simulated route.

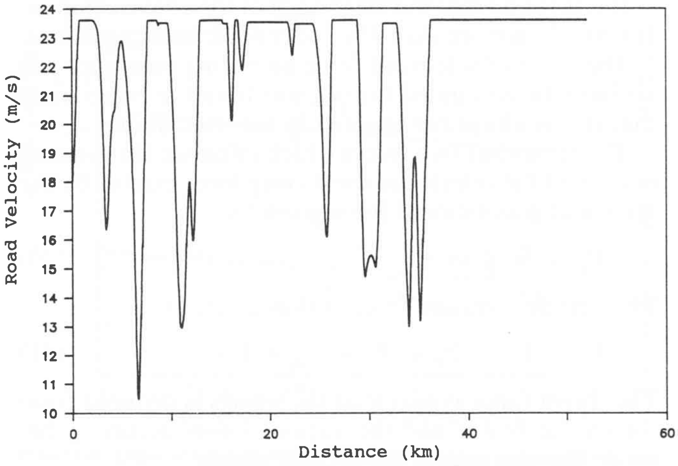

Figure 7 – Vehicle velocity with distance along route.

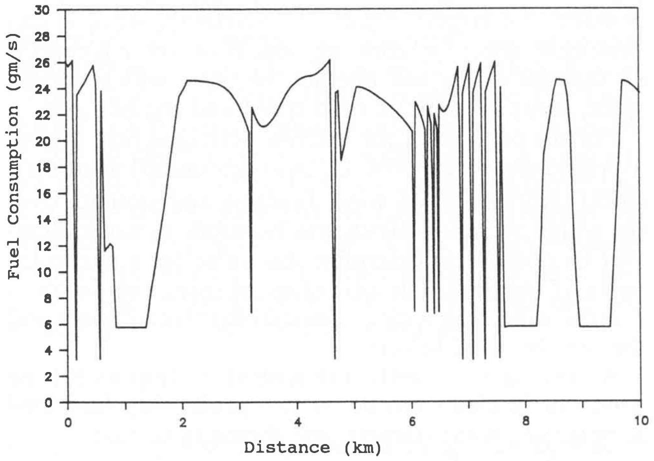

Figure 9 – Fuel consumption rate along simulated route.

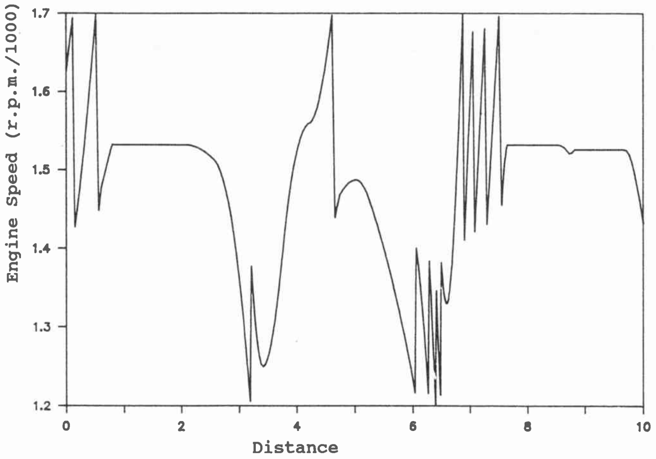

Figure 10 – Engine speed as function of distance along route.

considered. The program does however show that while drag reducing devices lead to fuel savings simple calculations tend to give optimistic estimates of the reduction in fuel consumption attainable.

Acknowledgements

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