MODELLING EMISSIONS FROM CARS

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ABSTRACT:

Exhaust emissions, HC, CO and NO_x *are identified as being more variable than fuel consumption, and therefore more difficult to model. The causes of variability in both source and measurement are described. The emissions rates and their coefficients of variation are depicted on joint axes of power and speed which allow explanation of the terms in 5 models of the lumped parameter type. Four cars have been selected to test the models. The performance of the models in 'closed loop' tests tends to improve with increasing complexity. Generally, emissions are predicted with 2 to 5 times the errors experienced in the prediction of fuel consumption with a given level of model.*

INTRODUCTION

Australian research has been at the forefront of the methodology of forecasting the exhaust emissions from motor vehicle in traffic, and more recently leading the way in the prediction of vehicle fuel consumption.

The early work in the simulation of exhaust emissions (Watson 1973 and Kinselman et al 1974), subdivided the vehicles' behaviour in two discrete ways, which, whilst making predictive computation extremely cumbersome, was founded on the fundamental nature of the problem. Firstly, the emission process was divided into the exhaust flow rate and the concentration of each of the pollutant species, namely hydrocarbons (or unburnt fuel) HC, carbon monoxide CO and nitrogen oxides NO_x, in the exhaust flow. Secondly, the driving pattern was subdivided into modes (sometimes called elements) comprising acceleration, cruise, deceleration and idle.

The exhaust flow rate was shown to relate principally to engine capacity and vehicle mass as well as the implied power described in terms of velocity and acceleration. These concepts still form the basis of recent models (Kent, Post & Tomlin, 1982).

This early work (Kinselman et al 1974) although capable of being applied at the microscale was only used in a gross way - in providing the travel speed related correction factors to the standard drive cycle emissions and was employed in the computation of the mobile emissions source in the estimation of inventories for urban air-sheds (see for example U.S. Environmental Protection Agency 1978). The method is still employed today. The emissions rates measured according to the U.S. 1975 emissions test procedure on a chassis dynamometer and the Los Angeles LA-4 driving pattern (Federal Register, 1973) are multiplied by factors according to estimated average travel speed of vehicles within prescribed (usually elements of an othoganal grid) area of the city. The factors are computed in quite fine detail, varying with emission type, year of vehicle manufacture and distance accumulation* and then integrated for the fleet on the basis of licence plate related information.

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*The year of manufacture is needed because of the change in emissions standards almost on an annual basis up until 1981 and the deterioration in exhaust emissions control systems with age, maintenance and tampering.

Scrutiny of the early formulations revealed inconsistencies when the models were employed for microscale (vehicle manoeuvre) related analysis. For example the lack of coherence in the formulae employed led to unacceptable inconsistencies, such as sudden transitions in emissions rates as a vehicle changed from mild acceleration to cruise, and the arbitrary decision needed to differentiate the acceleration rates at the cut off between acceleration, cruise and deceleration (0.5., 1 or 1.5, miles/h/s) and the interval for which a mode should be deemed to exist before it was treated separately.

THE SOURCES OF EMISSIONS VARIABILITY

Then, as now fundamental problems exist. The exhaust concentrations of HC, NO_x and CO vary significantly from engine cycle to engine cycle because of combustion cycle by-cycle variations. Further, much more significantly than fuel consumption, emissions are dependent upon the air-fuel ratio supplied by the carburettor and much less on engine power output. Fig. 1 shows typical curves of dependence on air-fuel ratio.

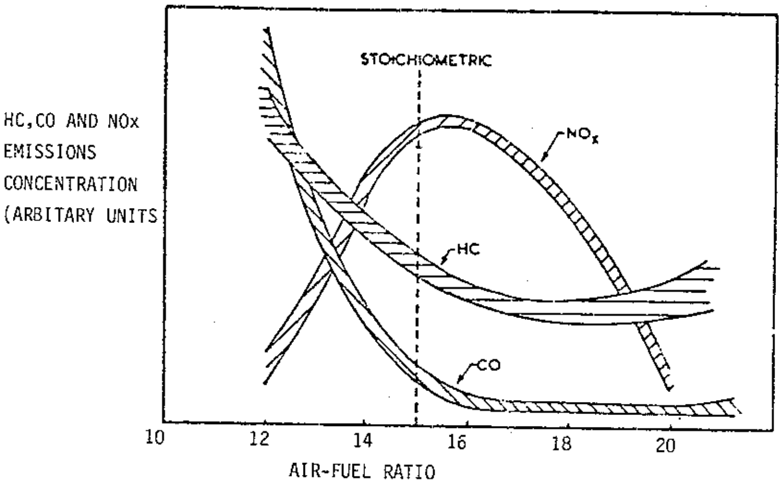

Fig. 1. Effect of air fuel ratio on exhaust emissions' concentration. The general trends are valid for all power outputs.

When the accelerator pedal of the car is depressed or released, the associated movement of the engine's throttle causes a change in the pressure in the intake manifold. Coupled with this pressure change is the removal or growth in the liquid fuel layer which pervades the walls of the inlet manifold, in quasi-steady equilibrium with the vapor air stream flowing through it.

The richening or leaning of the mixture causes large perturbations in exhaust emissions. Particularly in urban driving, variations in throttle position and manifold pressure occur almost continuously and thus there are significant contributions from transient effects to exhaust emissions.

Gear shifting provides a further cause of variability. At a given speed and power demand, a driver may have a choice of 2 or 3 gears at which the car can operate. Although shift points are mandated in the test cycles, particularly in cruise conditions there will often be 'overlapping' gear use. Plainly the lower numerical gear means higher engine speeds, increased friction and other losses which in turn affect emissions.

Further, contributions to emissions variability exists in the measurement process. From the traffic study viewpoint, emissions should be related to vehicle motion (velocity, acceleration etc.) However, in the measurement of instantaneous emissions on a vehicle dynamometer, the analysers are remote from the engine. Fig. 2 shows a typical measuring system and the time delay (δt) components in the flow of gas to the analysers. The delays in the Constant Volume sampling system (CVS) of Fig. 2, used in emissions testing to Australian Design Rules, are variable in the exhaust pipe (δt_1) and coupling pipe (δt_2). These residence times vary from about 0.35 at maximum power to several seconds at idle. Plainly these delays must be allowed for in relating vehicle motion to emissions measured. Even when the equipment is designed for minimum delay, 6 to 10 seconds lapse between production and measurement.

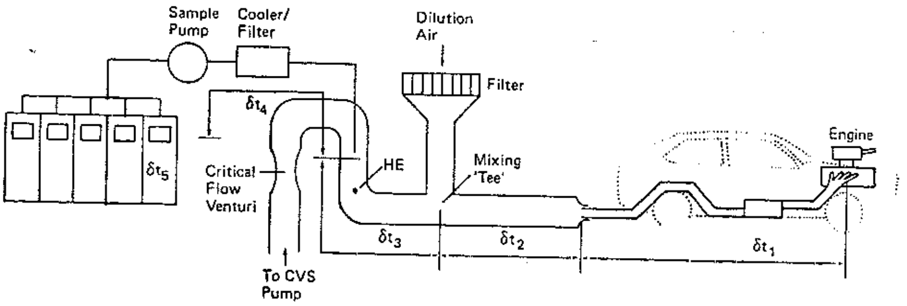

Fig. 2. Delays (δt) in the vehicle exhaust and the constant volume sampling system when used for continuous exhaust analyses.

The method of accounting for these delays and other problems such as mixing of the gases in mufflers and so on have recently been described (Watson et al 1985).

OBJECT

The object of this paper is to describe a test program designed to gather fuel and emissions data, the magnitude and variability of fuel consumption and exhaust emissions in typical driving situations, and how well simple models can describe the measured data.

TEST PROCEDURES

The results used in this paper were obtained as part of a survey of 40 cars (Watson et al 1985). Each vehicle was tested to driving patterns which embodied standard driving cycles (AS2077 city or ADR27A and ADR37) based on Los Angeles driving and cycles which we have developed, Melbourne peak (Watson, Milkins and Braunsteins, 1982) and Melbourne cold start (Lansell, Watson and Milkins, 1983). These latter cycles correspond quite closely to driving patterns measured elsewhere in Australia (Lansell and Watson, 1984). Each of these tests cycles contains between 8 and 18 microtrips. The Australian cycles involve considerably higher power demands than the U.S. city cycles and also compare favourably with driving in 5 U.S. cities (Wasielewski and Evans and Chang, 1980). The frequency of power demand for our Ford Cortina 4.1 litre test car over each of the cycles is given in Fig. 3. Note the vertical scale difference between the upper and lower two figures.

Other tests were also performed including steady speed cruising over the range from idle to 120 km/h in 10 km/h increments, acceleration tests at wide open throttle from rest to 110 km/h and coast down tests under closed throttle drive/4th gear (automatic/manual) and in neutral. The first coast downs provide data on the deceleration rate variation with speed above which the throttle must be opened. The second coast downs allow the determination of the vehicles aerodynamic and rolling resistance coefficients.

The above tests were performed on the chassis dynamometer with loads set to the vehicle's measured (weighbridge) inertia and to road drag determined in a newly developed procedure which allows replication on the dynamometer of steady speed fuel consumption to within ± 2 mL/min(Watson et al, 1985). A typical result for the GM Holden's Commodore, studied later in the paper is given in Fig. 4.

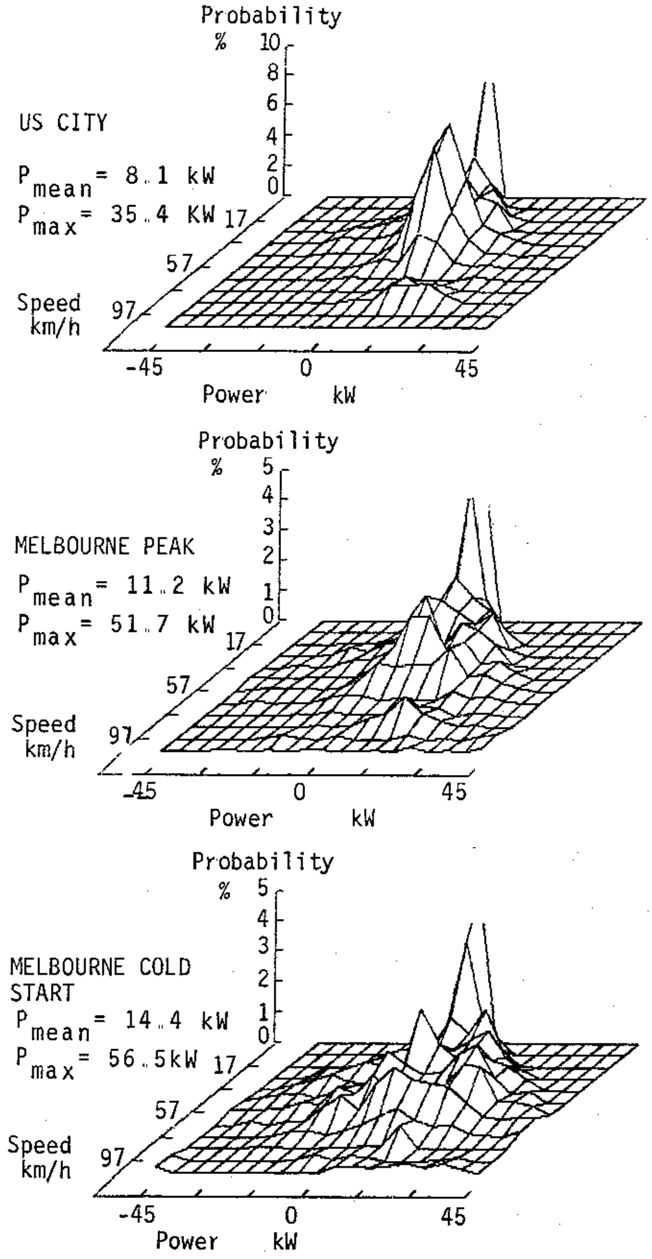

Fig. 3. Probability of power demand for the University of Melbourne's test car (4.1L Ford Cortina of 1585 kg mass) for the ADR27A or U.S. city cycle and the Melbourne peak and cold start driving cycles.

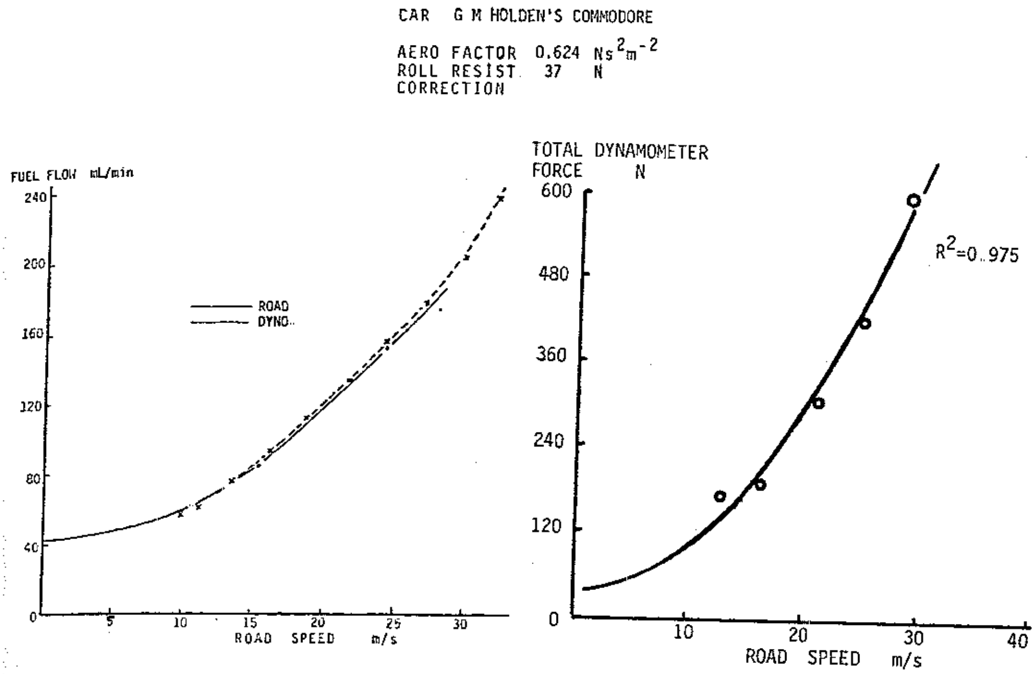

Fig. 4. Dynamometer tractive force versus true vehicle speed determined from on-road tests at a range of fixed throttle positions (Watson et al 1985) The **left** graphs display on-road and dynamometer fuel consumption using the determined rolling resistance correction factor R_c and aerodynamic factor b_2.

During all the dynamometer tests CO_2, CO, HC and NO_x emissions are logged at ½ second intervals, together with additional information such as vehicle and engine speeds, dynamometer torque and volumetric fuel flow to the carburettor.

The test results analysed here involve only hot start tests, so that changes due to 'warm up' of engine and vehicle components following a cold start have not been considered. Fuel flows analysed are those obtained from the exhaust analysis using the 'carbon balance' method which avoids the uncertainties, at the short time scale, of the fluctuations in fuel flow into the carburettor owing to the buffering effect of the float bowl. However, when integrated, rates measured by each method agree very closely as shown in Fig. 5.

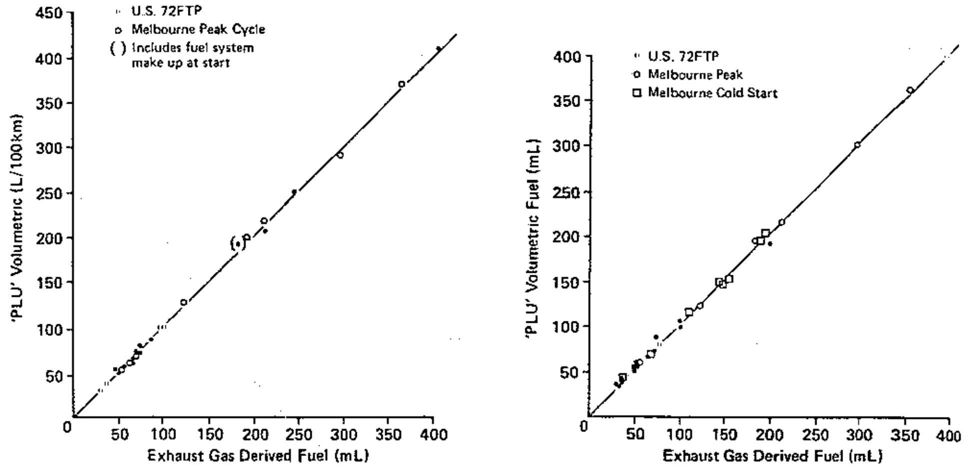

Fig. 5. Comparison of volumetric and exhaust emission derived fuel quantities integrated over microtrips for 3.3L GMH Commodore (left) and Ford Cortina (right) U.S. 72 FTP = ADR27A Solid line represents identity.

THE EQUATIONS FOR THE MODELS

Here five forms of lumped-parameter* type models for emissions are evaluated. These form the bases for the evaluation of nearly 30 models, which in various ways deviate from those now described (Holyoake 1985).

Naturally, the first 3 models have evolved from the early equations applied by Watson (1980). But the work of Bowyer, Akcelik and Biggs (1984) show that they are all quite closely related. Coefficients and correlation coefficient squared for each model are given in table 1.

Model 1

Is the original PKE-v_r model (Watson 1982) transformed into a fuel or emissions rate equation. If both fuel and emissions rates are represented by f (ignoring subscripts for HC, CO etc.)

$$f_t = \left[A\ v + B\ v^3 + (D\ v\ \underline{PKE} + E\ \underline{PKE})\ M + F\right]_{v>o} + f_{i\ v=o} \qquad (M1)$$

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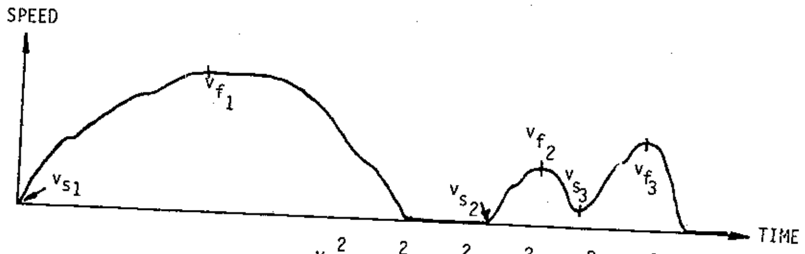
* Lumped parameter indicates that several physical features are joined together in models which retain physical significance viz. (engine efficiency x transmission efficiency x rolling resistance) as one co-efficient.

where A is the coefficient related to rolling resistance etc.
B is the coefficient related to aerodynamic drag etc.
M is the vehicle mass
D and E are constants related to the acceleration work term PKE defined in Fig. 6.
F is the rate related to engine friction
f_i is the idle rate

Calibration : The model is calibrated in 3 steps.

1. Determination of the idle rate
2. From steady speed tests, regression to find A, B and F.
3. D and E from regression of the incremental rates over the steady speed values over hot start drive cycles at ½ second intervals or alternatively for values integrated over microtrips.



$$PKE = \frac{v_{f_1}^2 - v_{s_1}^2 + v_{f_2}^2 - v_{s_2}^2 + v_{f_3}^2 - v_{s_3}^2}{D}$$

Fig. 6. The PKE (positive acceleration kinetic energy per distance) term explained.

Model 2

It is clear from examination of Model 1 that when applied instantaneously it will tend to :

* over-estimate deceleration fuel/emissions because it reverts to the steady speed cruise rates, whereas deceleration rates are typically less than cruise.
* Will underestimate acceleration rates to compensate for the over-estimated deceleration rates.

To combat this problem it was proposed that when the throttle was closed the fuel/emissions rate assumes the idle value. This is quite justified for fuel since sonic (constant) flow prevails in the carburettor. It is a poor assumption for emissions as will be seen. Integration of the PKE term continues only to closed throttle, when the deceleration rate is higher than coasting, where inertial forces equal aerodynamic forces because of engine braking.

Thus we have :

$$f_t = \left[A\ v + B\ v^3 + (D\ v\ PKE + E\ PKE)\ M + F\right]_o + f_{i_c} \quad \text{(M2)}$$

where the subscripts o stand for throttle open and c for throttle closed.

Calibration: is as for M1 except that the incremental fuel flow values are found only for throttle open.

The throttle open acceleration boundary is described as

$$a_c = f(v) \quad (1)$$

and determined from routinely performed tests in the program, as described previously. Fig. 7 shows a comparison of coast down and closed throttle decelerations. Extra fuel and emissions are consumed/produced in providing energy to overcome engine braking at speeds in excess of 5 to 7 m/s.

The reader is reminded that at decelerations more negative than throttle open, the vehicle brakes will have to be applied.

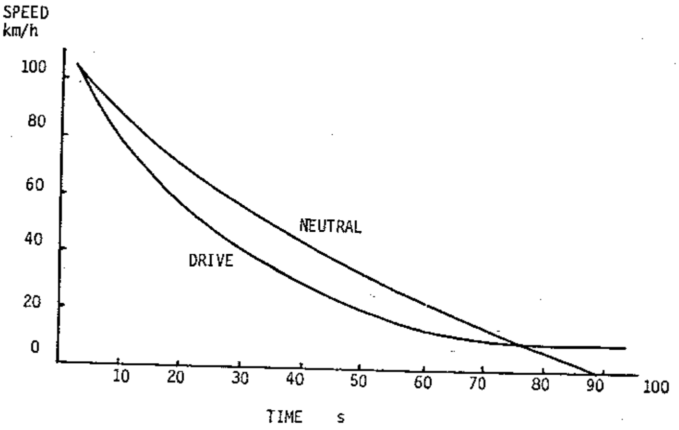


Fig. 7. Speed-time profiles of coast downs for GMH Commodore in neutral and drive. The throttle is closed for both tests

Model 3

It is a true kinetic energy model, in that during mild deceleration with throttle open, allowance is made for the dissipation of kinetic energy against aerodynamic and rolling resistance forces.

$$f_t = \left[A\ v + B\ v^3 + (D\ \underline{KE} + E\ v\ \underline{KE})\ M + F\right]_o + f_{i_c} \quad \text{(M3)}$$

Where $\underline{KE}$ is the kinetic energy.

Calibration : Is as for model M2. It should be noted that the KE term can become negative under mild deceleration conditions when the throttle is open.

Model 4

Is the well known Kent et al (1982) power demand model.

$$f_t = \alpha + \beta(P)_{P>o} \qquad \text{(M4)}$$

where $$P = b_1v + b_2v^3 + M\frac{dv}{dt}v \qquad \text{(2)}$$

β is the regression coefficient for drive cycle fuel rates at ½s intervals.

b_1 and b_2 are coefficients in the power equation related to rolling resistance and aerodynamic drag.

Calibration: The idle fuel rate α is determined from separate measurements. The coefficients b_1 and b_2 have to be determined from coast down tests with the vehicle in neutral.

Unfortunately during these tests there is extra friction due to the turning of the axle final drive gears, and this is included in b_1. With knowledge of P from equation 2 the coefficient β may be obtained from regression forced through the origin of the ½ second fuel rate or emissions measurement for all positive powers.

Table 1 : Table of coefficients and correlation coefficient squared

MODEL 1

	A	B	D	E	F	r2
HC	0.00000	0.46347E-06	0.15083E-06	0.88645E-06	0.174511E-02	0.825
CO	-0.24400E-03	0.13317E-04	0.39934E-05	0.10281E-03	-0.659040E-01	0.648
NOX	0.84659E-03	-0.82291E-07	0.11162E-05	-0.43675E-05	-0.819300E-03	0.742
FUEL	0.31860E-01	0.94761E-04	0.20985E-04	0.18159E-03	0.615610E-01	0.792

MODEL 2

	A	B	D	E	F	r2
HC	0.00000	0.46347E-06	0.17411E-06	-0.19947E-07	0.105714E-02	0.7237
CO	-0.24900E-03	0.13317E-04	0.82306E-05	0.30534E-04	0.758860E-04	0.7340
NOX	0.84659E-03	-0.82291E-07	0.12321E-05	-0.56118E-05	0.702000E-04	0.6446
FUEL	0.31860E-01	0.94761E-04	0.39037E-04	0.69016E-04	0.629860E-01	0.8591

MODEL 3

	A	B	D	E	F	r2
HC	0.00000	0.46347E-06	0.23704E-06	-0.52019E-06	0.126291E-02	0.7490
CO	0.24900E-03	0.13317E-04	0.11209E-04	0.10868E-04	-0.394760E-01	0.7188
NOX	0.84659E-02	-0.82291E-07	0.13079E-05	-0.68246E-05	0.249790E-03	0.4805
FUEL	0.31860E-01	0.94761E-04	0.55629E-04	-0.57754E-04	0.618210E-00	0.9055

MODEL 4

	α	β	b1	b2
HC	0.00168	0.23274E-06	0.27216E+03	0.56683
CO	0.00080	0.55320E-05	0.27216E+03	0.56683
NOX	0.00139	0.15417E-03	0.27216E+03	0.56683
FUEL	0.76000	0.94714E-04	0.27216E+03	0.56683

MODEL 5

	α	b1β	b2β	β1	β2	r2
HC	0.00108	-0.80515E-04	0.70147E-06	0.28107E-03	-0.93799E-04	0.525
CO	0.00080	-0.94448E-03	0.65800E-05	0.63696E-02	0.62020E-03	0.268
NOX	0.00139	0.61962E-03	0.52031E-07	0.26333E-02	-0.14592E-02	0.523
FUEL	0.76000	0.24942E-01	0.10451E-03	0.80397E-01	-0.13624E-03	0.736

Model 5

Is an extension of Model 5 by Bowyer, Akcelik and Biggs (1984) including an additional term - the energy - acceleration efficiency paramater β_2, thus

$$f_t = \alpha + \beta_1 \ (b_1 + b_2 V^2 + M \frac{dv}{dt}) \underset{>o}{v} + \beta_2 M(\frac{dv}{dt})^2 \underset{>o}{v} \qquad (M5)$$

Calibration: As for M4 except that the second regression coefficient β_2 is included. And the steady speed fuel consumption equation may be used for the first 3 terms, ie.

$$(f_t)_{a = o} = \alpha + c_1 v + c_2 v^3 \qquad (3)$$

where c_1 and c_2 are found by regression, and $a = dv/dt$ (refer to P.19).

Direct quantification of the rolling resistance and aerodynamic coefficients b_1 and b_2 is avoided by this method.

The reader will probably have observed that all 5 models are part of a family with minor variations in assumptions and calibration methods.

Note that :

$$M \underline{KE} = M \frac{\Delta v^2}{2} = M \int_{t_1}^{t_2} a \ v \ dt \approx \int_{t_1}^{t_2} P_i \ dt \qquad (4)$$

where P_i is the power to accelerate the vehicle

MODEL VARIATIONS

All the above models can include a gradient term. This has been omitted for simplicity. Plainly kinetic energy and potential energy changes can be equated, so that the coefficient for gradient does not have to be separately determined. Equation 1 for throttle closure will need inclusion of the gradient in the acceleration term.

Several variations of the models are possible, leading to 27 equations in Holyoake's (1985) evaluation. For example the mean running velocity v_r can be used as :

$$\overline{v}_r{}^3 \approx \Sigma v^3 \qquad (5)$$

Only the average speed and stop times are then needed to obtain v_r for M1 and the peak and minimum speeds for the $\underline{PKE}$. Another option is to include only one $\underline{KE}$ or $\underline{PKE}$ term.

For Model 4, improved regression can be obtained by determining α and β simultaneously, but then α no longer represents the idle fuel rate.

These variations and others are presently excluded to fit the presentation within these confines.

DATA BASE

For brevity, examination of the data available has been confined to four cars : a GM Holden's (GMH) Kingswood with a 4.2L V8 engine and automatic transmission, a GMH Commodore with 3.3L 6 cylinder engine and automatic transmission, our test car used in many previous experiments, the Ford Cortina Wagon with the 4.1L 6 cylinder engine and automatic transmission and a Honda Civic with a 1.3L 4 cylinder engine and 5 speed manual transmission.

The test data involve the hot start driving of the ADR27A and MPC cycles. The data have been represented graphically in Figs. 8 and 9 for the Ford Cortina and Honda Civic.

This new way of expressing fuel/emissions rate data is based on the form of equation (M4). In the speed direction we would expect the surface to vary as a cubic equation in speed, but linearly in the low power domain for $P > 0$. These trends are best observed in the fuel surfaces for both cars. It can be seen that for $P < 0$ the fuel rate is essentially constant, except at higher speeds where fuel is needed to overcome engine braking; more noticeable in the Honda with a manual transmission.

At high speeds and powers there are some 'holes' in the surface where no data points exist. The fuel rates, especially for the Honda, show substantial increases at high speed. There is also a tendency, at very high powers, which are only used at the highest speeds, for the slope of the fuel rate with respect to power to increase. This justifies the presence of the β_2 term in M5, and the second terms, allowing for a change of slope df_t/dP at higher speeds, in M1, M2 and M3 i.e. **MEPKE** and MEKE.

For fuel rates the coefficient of variation* in the right side graphs are seen to be very small, generally less than 0.2 over nearly all the speed-power surface.

Although the emission graphs have a tendency to have the same basic shape as the fuel data, as expected since exhaust flow rate is essentially proportional to fuel rate, there exists usually higher values of C.O.V.

* C.O.V. is defined as standard deviation/mean

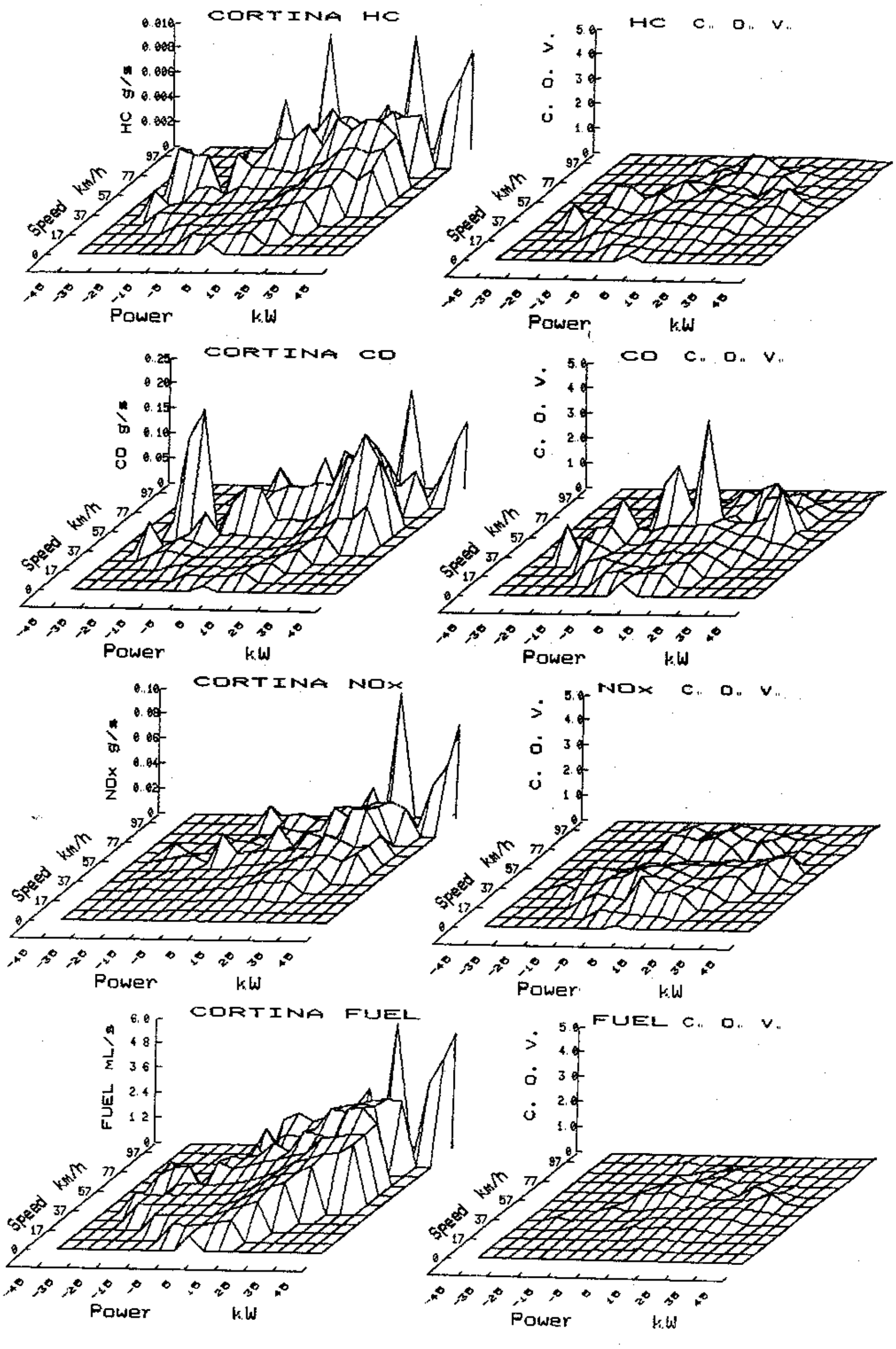


Fig. 8. Emissions surfaces and their coefficients of variation on speed-power axes for Ford Cortina.

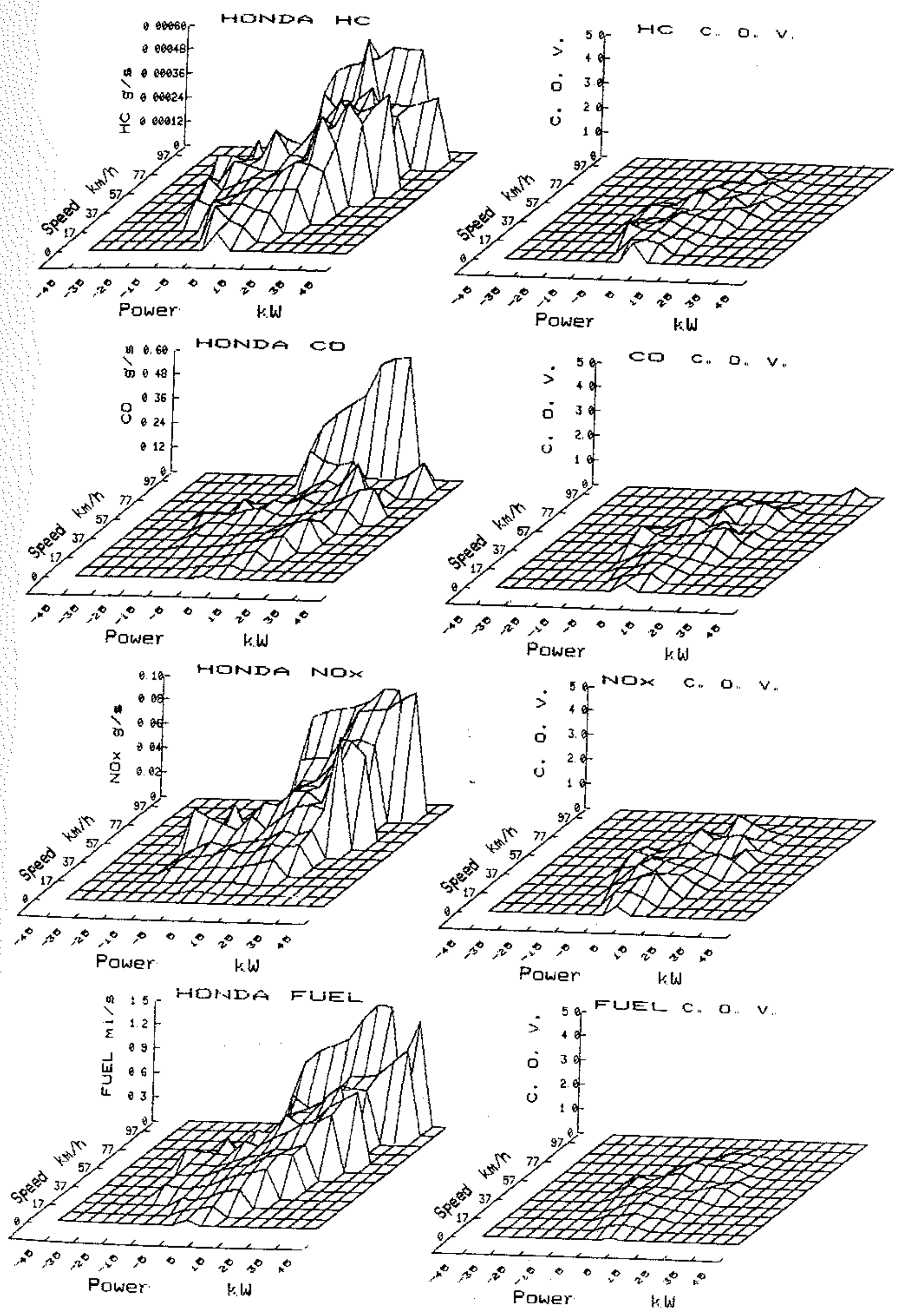


Fig. 9. Emissions surfaces and their coefficients of variation on speed-power axes for Honda Civic

The variability is the result of the reasons given in the introduction, and the underlying assumption of the time (or previous history) independence of the data presentation and the models. It has been suggested by Kent, Post and Tomlin (1982) that the power demand model is unsatisfactory for describing CO emissions. C.O.V.'s are large for the Cortina, as high as 3.0, but the actual CO emission levels are very low for this car, less than for Honda over much of the surface. The variability is especially bad under very light throttle engine operation, ie. near steady (low power) moderate speed driving, when the progression jets in the carburettor can cause shifts in air-fuel ratios.

The high peak under deceleration is clearly at variance with the concept of constant rates for closed throttle operation embodied in all the models.

The graphical presentation used for the data has been extended to demonstrate the relative contribution of each node on the mesh. It can be seen in Fig. 10 that the just mentioned CO peak contributes 0.7% and therefore is relatively unimportant.

The prediction of emissions surfaces and their variance can be improved by the use of much more complex models (see for example Watson et al 1983) called transient engine mapping models which are beyond the scope of application to traffic management problems.

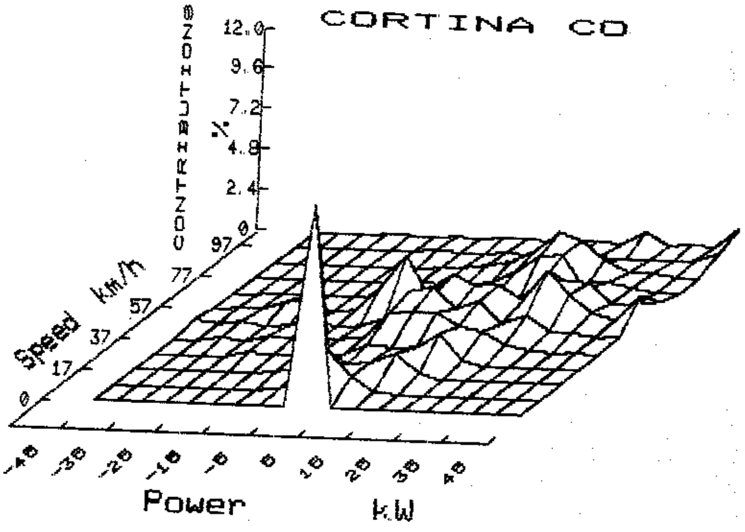


Fig. 10. Contribution of each node of mesh to total CO emission over the test cycles.

STEADY SPEED EMISSIONS

The bases of all 5 models can be the steady speed emissions or fuel function. This is especially beneficial since, as part of the test program, the fuel rate obtained on the dynamometer is validated against road data (refer to Fig. 7). A second advantage is in providing confidence that the cruise speed rates will be well founded when needed in network simulation tasks, examples of which appear in Bowyer, Akcelik and Biggs (1984).

There are however problems with behaviour of the emissions surfaces for some cars, particularly those using engine fuel and ignition management as prime controls to meet emissions standards. An example is the 6 cylinder Holden engine prior to the current model. Fig. 11 depicts the steady speed relations for the Holden Commodore examined in this paper. It is clear that the three term, third order equation (based on the physics of vehicle motion) has difficulty in fitting closely the HC and CO emission curve and produces negative emissions for the NO_x rates. Clearly higher order polynominal expressions in V can improve the regression and have been employed where necessary.

RESULTS

Analysis, here has been confined to 'closed loop' - answering how well do the models explain the data on which they were based, ie. Hot start ADR27A and MPC tests. Open loop on MCC, and modal (acceleration, deceleration etc.) analyses have been performed (Holyoake 1985). Most usually the open loop and closed loop tests agree quite closely. To minimise the volume of material, the former tests only are demonstrated.

All the results are treated on a per microtrip basis (ie. one start to the next start). The microtrip times vary just over 20s up to 4 minutes.

The cumulative residuals, or sum of the microtrip errors in percent, e is defined as

$$e = \frac{\Sigma(\text{error per microtrip}) \text{ x } 100}{\Sigma \text{ microtrip fuel/emission (mL/g)}}$$

The unsigned error $|e|$ is defined as

$$|e| = \frac{\Sigma \text{ abs (error per microtrip) x } 100}{\Sigma \text{ microtrip fuel/emission}}$$

The coefficient of variation in the error

$$\text{COV} = \frac{\sigma \text{ microtrips x } 100}{\text{mean fuel/emission for all microtrips}}$$

The results for the four cars are given in Table 1.

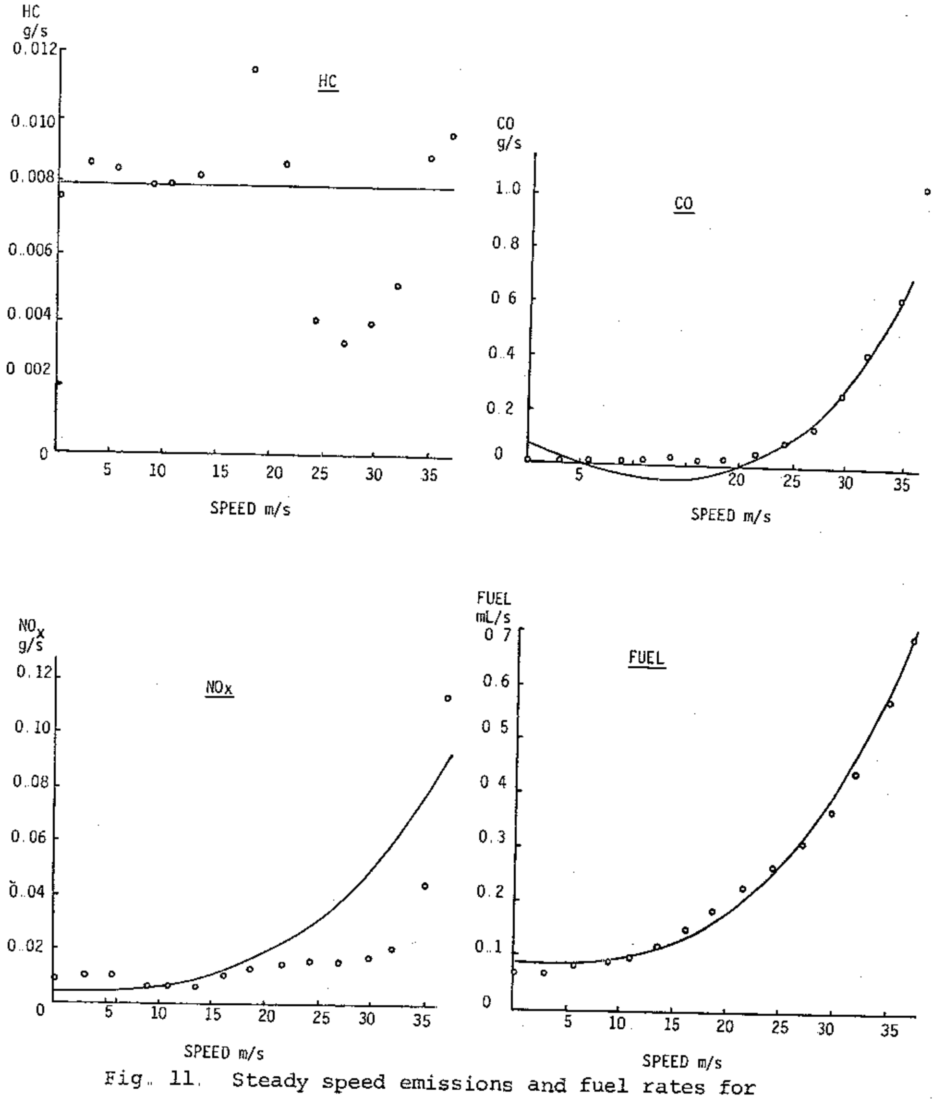


Fig. 11. Steady speed emissions and fuel rates for GMH Commodore.

It should be apparent that the sum of the microtrip errors compares the total error to the total fuel used. The unsigned residual (error) $|e|$ is therefore a more meaningful criterion. For fuel rates, the performance of the PKE and KE models (M1, M2 and M3) are similar with the KE model producing average errors of just over 3%. Model M4 can be made to produce zero e by free regression of the idle flow rate, but then gives higher $|e|$, and of course the fuel rate while stopped, is then not predicted accurately.

TABLE 2 - ERRORS IN MICROTRIP EMISSIONS AND FUEL CONSUMPTION

		1978 FORD CORTINA, 4.1L AUTO					1976 HOLDEN KINGSWOOD, 4.2L AUTO			
MODEL		HC	CO	NO_x	FUEL		HC	CO	NO_x	FUEL
M1	e %	5.7	26.1	6.7	2.0	e %	8.8	4.05	4.6	3.7
	|e|%	9.8	42.0	16.8	3.1	|e|%	14.8	15.1	13.9	4.9
	COV%	13.7	51.0	27.6	3.1	COV%	22.7	43.5	21.6	4.9
M2	e %	1.3	30.5	4.1	1.9	e %	14.8	2.7	4.9	3.7
	|e|%	9.4	54.4	15.4	3.1	|e|%	16.9	17.2	13.7	4.4
	COV%	18.0	54.7	25	3.2	COV%	25.4	48.5	22.9	4.4
M3	e %	2.4	47.5	6.7	2.6	e %	17.2	5.8	6.1	4.2
	|e|%	9.8	49.2	18.8	3.1	|e|%	19.8	28.6	13.3	5.0
	COV%	18.0	62.9	36	2.0	COV%	27.8	4.6	23.6	3.22
M4	e %	10.4	23.0	19.2	8.3	e %	14.3	-12.4	7.2	17.0
	|e|%	19.1	49.4	20.6	8.5	|e|%	24.1	43.8	35.7	17.0
	COV%	36.0	74.3	31	14.7	COV%	38.7	7.9	75.8	13.6
M5	e %	10.6	33.1	25.2	6.7	e %	16.6	- 6.6	-58	13.7
	|e|%	17.6	43.6	27.0	6.7	|e|%	19.1	65.3	67	13.7
	COV%	23.4	69.9	44.9	4.6	COV%	38.3	9.96	78.9	10.2
		1981 HONDA CIVIC 1.3L 5 SPEED MANUAL					1983 HOLDEN COMMODORE 3.3L AUTO			
M1	e %	5.4	4.1	- 2.1	0.5	e %	5.3	24.7	4.7	3.0
	|e|%	9.1	15.1	7.3	5.0	|e|%	17.9	40.9	10.4	4.1
	COV%	12.7	43.5	13.9	6.6	COV%	33.7	81.0	16.2	4.5
M2	e %	7.1	2.7	- 0.3	1.2	e %	8.2	33.4	6.6	0.57
	|e|%	12.7	17.2	11.9	5.9	|e|%	20.3	45.7	10.6	3.0
	COV%	15.2	48.5	20.1	8.1	COV%	37.8	75.8	14.4	4.2
M3	e %	7.6	3.2	- 2.1	2.7	e %	0.5	-15.9	2.4	1.3
	|e|%	13.5	17.6	7.8	3.9	|e|%	2.7	54.1	11.4	2.6
	COV%	15.7	49.5	13.4	4.6	COV%	38.2	79.7	22.9	3.8
M4	e %	40	20.9	22.5	17.9	e %	6.1	4.09	16.9	7.4
	|e|%	63	26.3	23.4	17.9	|e|%	29.2	67.2	24.9	7.4
	COV%	43.3	46.5	41.8	10.8	COV%	47.2	114.3	44.9	9.3
M5	e %	39	16.5	8.5	20.4	e %	7.0	15.3	18.9	12.8
	|e|%	59	36.7	13.2	20.4	|e|%	27.2	73.6	25.3	12.8
	COV%	22.9	52.4	15.0	13.5	COV%	42.3	116.2	34.6	11.6

Note half second data values are used to calibrate the models.

The M5 model incorporates acceleration squared terms which become most significant at highest power outputs. By including into the data file, high power-operation, obtained during wide open throttle tests, the acceleration squared term becomes more significant and the driving cycle microtrip results are considerably improved. This may be observed in Table 3 for the Honda Civic, which should be compared with the data at the foot of Table 3.

TABLE 3 - RESIDUALS FROM MODEL M5 WITH MODIFIED DATA INPUT

	HC	CO	NO_x	FUEL
e %	14.4	4.8	2.2	5.9
$\|e\|$%	14.5	16.2	3.7	6.0
COV%	13.5	17.2	3.5	4.8

Examination of Table 2 indicates that emissions predictions for all models are worse than those for the prediction of fuel quantities. This is because of the increased variability in the data, which is sourced in the vehicle rather than in the data itself, since, for example, aggregate forms of the KE Model (M3) produce almost identical results to the instantaneous form employed here (Holyoake, 1985).

Fig. 12 depicts the regions in which the M5 model works least well for the Cortina CO emissions. The error surface shows particularly the problems associated with the high CO peak during deceleration, and the general level of under prediction during deceleration. A considerably larger data base is needed before it can be stated that variations of the basic fuel consumption derived models are necessary.

For example, relaxing the necessity for the term in M5 being >0 is a simple variation that could be useful.

CONCLUSIONS

The simple 'closed loop' tests (model prediction of calibration data) employed here, have demonstrated that predictions of exhaust emissions for microtrips are much less accurate (two to five times) than fuel consumption. This is true for all 5 models tested when the criteria are cumulative microtrip error, the unsigned error sum and the coefficient of variation of the errors.

Although the equations employed in the models are founded on the physics of the vehicle motion, the formation of emissions are dependent on transient effects such as air-fuel ratio shifts which cause increased variability, displayed in coefficient of variation surfaces (in Figs. 8 and 9). It would appear that this variability can only be accommodated in much more complex models which account for the transient nature of the emissions formation. However, transient engine mapping models, which can account for the non-temporal sources of variability, are too complex to aggregate for the vehicle fleet, as they treat vehicles individually, and therefore likely to be of use only in specialised microscale traffic management studies.

The final quantification of the probable errors in emissions predictions of the lumped parameter type models studied here, must await the evaluation of the data for the fleet of over 40 vehicles which we have already tested.

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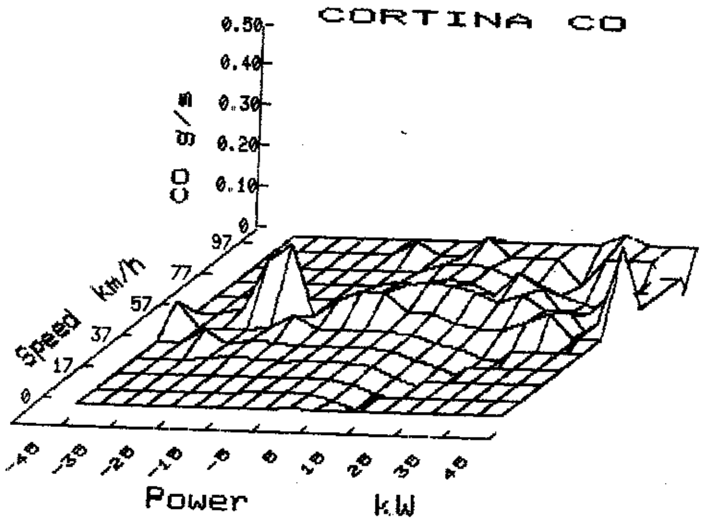

Fig 12 Emissions error surface between measured data and M5 model for CO and Ford Cortina car.

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