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DIESEL ENGINE MODELLING UNDER STEADY AND TRANSIENT CONDITIONS USING A TRANSPUTER BASED CONCURRENT COMPUTER

by

Jun Qiao B.Eng

A Doctoral Thesis submitted in partial fulfilment of the requirements for the award
of Doctor of Philosophy of Loughborough University of Technology

June 1990

Supervisor: Professor J C Dent, PhD, CEng
Department of Mechanical Engineering

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TO MY PARENTS

FOR THEIR LOVE AND SUPPORT

SYNOPSIS

The work presented describes the development of an interactive model to simulate a direct injection diesel engine under both steady and transient conditions, based on the application of concurrent process computing methods. Starting with the modelling of the engine under steady operating conditions, in which induction, injection, air entrainment, fuel air mixing, combustion, emission and mechanical friction processes are considered, the fuel pump, governor, engine crankshaft and external load dynamics are also investigated to model the transient behaviour of the engine and its associated load. Employing a two zone combustion model enables detailed performance and exhaust emissions predictions to be produced with economic use of computing time. The modelling work has been developed on a concurrent computer consisting of four INMOS T800 transputers which have been linked together and installed on an IBM PC-XT host. The sequential model is written in FORTRAN 77 and can be implemented on all machines equipped with standard FORTRAN 77 compilers. A description of the task division running concurrently using harnesses developed in the transputer operating system language OCCAM is also provided. The parallel model implemented on three transputers is just about double the speed of the same model running on one transputer, and is about three times faster than the medium speed mainframe, Honeywell multics. Furthermore, on the concurrent computer, calculation and graphic presentation can also be carried out in parallel which cannot be done on any sequential machines. Experiments were undertaken to measure various performance and emission data to validate the model. Comparison of modelling results with engine experiments is in general good.

ACKNOWLEDGEMENTS

The author wishes to express his gratitude to the following:

Professor J.C. Dent for his untiring guidance and supervision throughout this work.

The British Council and Chinese government for the scholarship support to carry out this investigation.

Ford motor company for providing experimental data to test the model.

Mr David Johnston at the Rutherford Appleton laboratory for assistance in the development of the OCCAM Harness.

Messrs. S Taylor, S Line and all the members of the technical staff of the Mechanical Engineering Department for their assistance.

All friends who have helped in different ways throughout the project. In particular thanks given to Messrs. C Garner, S Robinson, X D Chen, G J Lu, H Sindano and Miss N Wipawin for their inspiration and help during this work.

NOTATION

A	Frontal area of vehicle	(m ²)
B	Cylinder bore	(m)
CN	Fuel cetane number	(-)
D	Duration	(-)
D _b	Bearing diameter	(mm)
d'	Equivalent injector nozzle diameter	(m)
E	Activation energy	(J/kg mole)
f	Vehicle acceleration	(m/s ²)
g	Acceleration of gravity	(m/s ²)
h _i	Heat transfer coefficient	(kW/m ² /K)
H _o	Lower calorific value of the fuel	(kJ/kg)
I _E	Engine inertia	(kg m ²)
I _s	System inertia referred to crankshaft radius	(kg m ²)
k	Turbulence kinetic energy	(kJ)
	Integral length scale of turbulence	(m)
L	Stroke	(m)
L _A	Acceleration torque	(Nm)
L _b	Bearing length	(mm)
L _c	Connecting rod length	(m)
L _E	Engine torque	(Nm)
L _T	Tractive torque	(Nm)
L _v	Maximum valve lift	(mm)
m	Mass	(kg)
m.	Mass flow rate	(kg/s)

n	Radical nuclei number density	(particles/m ³)
n_b	Back axle gear ratio	(-)
	Number of bearings	(-)
n_c	Number of cylinders	(-)
n_g	Transmission ratio	(-)
n_v	Number of valves	(-)
N	Soot particle number density	(particles/m ³)
N_A	Wheel speed	(rpm)
N_C	Input speed to transmission	(rpm)
N_E	Engine speed	(rpm)
N_S	Propeller shaft speed	(rpm)
p_a	Atmospheric pressure	(kpa)
p_i	Intake manifold pressure	(kpa)
P	Cylinder pressure	(pa)
P_{fu}	Partial pressure of fuel	(pa)
P_{O_2}	Partial pressure of oxygen	(pa)
Q	Fuel injection rate	(m ³ /s)
Q_{rel}	Heat release rate	(m ³ /s)
Q_{tr}	Heat transfer rate	(m ³ /s)
r	Wheel radius	(m)
r_c	Compression ratio	(-)
r_e	Exhaust valve diameter / bore	(-)
r_i	Intake valve diameter / bore	(-)
r_E	Transmission equivalent radius	(m)
R	Gas constant	(J/kg K)
R_a	Air resistance	(N)
R_g	Road grade resistance	(N)
R_r	Rolling resistance	(N)
R_t	Total tractive resistance	(N)

R_u	Universal gas constant	(J/kg mole K)
s	Stoichiometric air/fuel ratio	(-)
S	Spray penetration	(m)
S_p	Mean piston speed	(m/s)
S_r	Swirl ratio	(-)
S_w	Wall jet penetration	(m)
t	Time	(s)
Δt	Time interval	(s)
T	Temperature	(K)
u	Internal energy	(J/kg)
U	Velocity	(m/s)
V	Volume	(m ³)
V_{ci}	Clearance Volume	(m ³)
V_f	Volumetric fuel delivery per stroke	(m ³)
V_m	Vehicle speed	(km/s)
V_s	Swept Volume	(m ³)
W	Vehicle mass	(kg)
x,y	Cartesian Coordinates	(-)
ε	Turbulence eddy dissipation rate	(m ² /s)
θ	Crank position ,Road slope	(-)
ω	Angular velocity	(rad/s)
ρ	Density	(kg/m ³)
τ	Mixing time	(s)
ϕ	Equivalence ratio	(-)
β	Mode of burning proportionality factor	(-)

SUBSCRIPTS

The following subscripts pertain to:

- Time mean value
- * Fine structure
- a Air
- ac Air consumption during combustion
- ae Air entrained into the spray
- asr Air in the surrounding
- ch Charge mean condition
- cb Combustion
- evo Exhaust valve opening
- f Fuel
- g Bulk gas stream
- h Thermal properties
- inj Injection
- jet Fuel spray jet
- o oxidation
- o2 oxygen
- s soot
- st stoichiometric condition
- vb fuel vapour burnt
- w wall

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CHAPTER 1

LITERATURE SURVEY

1.1 INTRODUCTION

The intense development of the diesel engine has required fundamental understanding of its component processes under both steady and transient operating conditions, and the use of computer simulation is a useful tool in this understanding.

In the past, computer simulation has been limited by hardware costs and the architecture of the computer, which followed the classical Von Neumann principle of sequential data flow through a single central processing unit (CPU). The advent of cheap computers with multiple processors has enabled concurrent or parallel processing methods of computation to be explored as a method of reducing computational time for simulation.

The present work describes the development of an interactive model to simulate a direct injection diesel engine under both steady and transient conditions, based on the application of concurrent process computing methods. The work is aimed at predicting diesel engines performance and emission with economic use of computing time. The steady state combustion model is based on the work reported in several publications (Dent and Mehta (1), Dent et al (2), Kyriakides et al (3), Sindano (4)). The fuel pump, governor, engine-load dynamics are investigated to model the transient behaviour of the drive train.

1.2 DIESEL ENGINE PERFORMANCE AND EMISSIONS

The diesel engine has served as a power source for heavy duty applications such as electrical generation, marine propulsion and land traction because of its fuel economy advantage. Due to the rise in the cost of petroleum and increasing demands for a clean environment, there has been considerable development in the application of diesel engines to light and medium load conditions such as passenger cars over the past two decades (Hofbauer and Sator (5), Monaghan (6), Cichoki and Cartellieri (7)).

Satisfactory operation of diesel engines depends on proper control of the fuel injection and air motion. The ideal engine system should have a high efficiency, high output, low levels of noise and exhaust emissions. To some extent these requirements are in conflict with each other, for example, engine output is directly limited by the exhaust smoke levels (Stone (8)).

Diesel engine combustion systems in general commercial use today can be divided into two main classes, namely, direct injection (open chamber) and indirect injection (divided chamber). The light duty indirect injection diesel engines have lower levels of noise and exhaust emissions (6), compared to equivalent direct injection diesels. However the high compression ratios required by the indirect injection engines to start adequately when cold, result in high thermal stress. Also the increased heat transfer and delayed heat release (due to transfer of burning gas to the cylinder) in the indirect injection engine result in loss in thermal efficiency, about 10% lower than the direct injection counterpart (6).

1.2.1 THE DIESEL ENGINE COMBUSTION PROCESS

Combustion in a diesel engine can be considered to commence at the dynamic injection point and to consist of the following stages, illustrated in figure (1.1) (Haddad and Watson (9)):

(i) Ignition delay, 1-2. After injection there is initially no apparent deviation from the unfired cycle. During this period the fuel is breaking up into droplets and vaporised. Mixing of the vaporised fuel and air will take place and hence an ignitable prepared mixture is ready to burn after the ignition lag has elapsed.

(ii) Rapid or uncontrolled combustion, 2-3. This period is characterised by a very rapid rise in pressure caused by ignition of the fuel/air mixture prepared during the ignition delay period.

(iii) Controlled combustion, 3-4. The chemical reaction rate is governed by the rate of preparation of fresh fuel/air mixture, which is dependent on purely physical processes.

(iv) Final combustion, 4-5. The rate of combustion is controlled by diffusion until all the fuel or air is utilised.

The length of the delay period (the first stage, 1-2) is very important since it affects the development of the processes following ignition and hence the rate of pressure rise, maximum of cylinder pressure, noise levels, etc.

The first appearance of a flame following ignition is of the premixed type, and the peak rate of heat release is normally associated with the short period of uncontrolled burning (the second stage, 2-3). During this period the flame spreads to enclose the fuel sprays which subsequently burn as a turbulent diffusion flame (the third and final stages, 3-5). So, to achieve adequate combustion, it is essential to mix the fuel and air rapidly since the diesel combustion is mainly diffusion controlled. In large engines used in industrial and marine applications, this is achieved by high pressure injection of fuel (injection pressures of the order of 1000 bar are quite common (8)). For small high speed engines used in automotive applications, however, the mixing by the fuel injection is not enough, hence the air motion is used to enhance mixing. The optimisation of combustion chamber design to produce optimum air movement over a wide range of operating conditions is essential in this type of engine.

Since diesel combustion is mainly controlled by the diffusion burning, which is characterised by carbon particle radiation, it is inevitable to form carbon particles in a diesel engine. Research activity aimed at improving diesel smoke emission, has therefore been directed at either improving combustion in the cylinder by ways such as microprocessor based engine management systems (Gobburu (10), Tomisawa (11)) or burning the carbon particles off in the exhaust traps (MacDonald and Simon (12)). Chemical kinetics shows that the formation of NO_x increases very strongly with increasing flame temperature. The high temperature caused by the heat release pattern of diesel combustion assists the formation of NO_x and the highest concentration of NO_x should be for slightly rich mixtures which have the highest flame temperature (8). However, NO_x formation is also influenced by the concentrations of oxygen and nitrogen. Kaneko et al (13) have shown that the peak NO_x concentration takes place where the product of oxygen and nitrogen concentrations becomes the maximum value. The introducing of EGR (exhaust gas recirculation) can reduce the NO_x emission effectively since it lowers both flame temperature and oxygen concentration.

1.2.2 TRANSIENT RESPONSE OF DIESEL ENGINES

The transient response of an engine is determined by the inertia of the engine rotating parts and the load, and by the difference between the instantaneous torque developed by the engine and that applied by the load.

Typical examples of engines operating under transient conditions are given below:

Engine Acceleration

Vehicle engines are subject to frequent accelerations, hence their response to changes in desired engine speed is important - driveability. For natural aspirated engines, this is a less severe problem, but exhaust smoke limitations are imposed on the engine.

A typical example is accelerating a traction engine. When the control lever on the fuel pump is moved to increase fuelling, the governor moves the control spool so that more fuel is delivered to the cylinders. This increase in fuel is normally not matched by a proportionate increase in air flow to the cylinders which is directly controlled by engine speed. In severe cases this results in poor combustion, a slow response (bad driveability of a vehicle) and dense smoke emission.

Satisfactory operation of diesel engines under transient conditions depends on proper control of the fuel delivery and air induction. A good engine/governor system should have fast speed response (good driveability) and a low level of exhaust emissions. To a certain extent these requirements are conflicting with each other and figure (1.2) illustrates the trade-off between the driveability and smoke emission for an engine accelerating against a steady load, with and without fuel controller (Watson (14)).

Sudden Load Changes Imposed on Engine

A similar situation is the sudden application of load to a speed governed engine. A typical example is an engine driving an earth moving vehicle which experiences load applications as part of its normal duty. When the load is applied, the engine speed will drop. In response to this, the governor will deliver more fuel to the cylinders. Again, the increase in air will not match the increase in fuel, resulting in a long recovery time and dense exhaust smoke.

Over the last two decades, the importance of a clean environment with a low level of pollutants has been increasingly recognised and this has resulted in the control of diesel exhaust emissions. In some developed countries legislation has been introduced to limit the emissions from diesel engine powered vehicles. For example, in 1980, the U.S. Federal Government introduced legislation to limit the emissions of light-duty diesel cars (15).

The Emission Standards of both US Federal Government and California for passenger cars from 1975 to 1985 are shown in Table 1.1 and it can be noted that

the regulatory requirements are tightening. This indication will increase research and development activity directed at making the existing engines or design of new engines to meet the stringent emission standards of the 1990's.

Table 1.1 Emission Standards for Passenger Cars from 1975 to 1985

Model Year	-----Federal-----				-----California-----			
	HC	CO	NOx	Evap	HC	CO	NOx	Evap
1975-76	1.5	15	3.1	2	0.9	9.0	2.0	2
1977	1.5	15	2.0	2	0.41	9.0	1.5	2
1978-79	1.5	15	2.0	6	0.41	9.0	1.5	6
1980	0.41	7.0	2.0	6	0.41(b)	9.0	1.0(c)	2
1981	0.41	3.4(d)	1.0(e)	2	A	0.41(b)	3.4	1.0(c)
					B	0.41(b)	7.0	0.7
1982	0.41	3.4(d)	1.0(e)	2	A	0.41(b)	7.0	0.4(c)
					B	0.41(b)	7.0	0.7
1983	0.41	3.4	1.0(e)	2	0.41(b)	7.0	0.4(f)	2
1984	0.41	3.4	1.0(e)	2	0.41(b)	7.0	0.4(f)	2
1985	0.41	3.4	1.0	2	0.41(b)	7.0	0.4(f)	2

(a) Standards apply to the FTP applicable at the time and are expressed in grams/mile for the exhaust emissions and grams/test for evaporative emissions.

(b) .39 gm/mi standard for hydrocarbons other than methane if methane is actually measured.

(c) 1.5 gm/mi allowed with 100,000 mile durability.

(d) Waiver up to 7.0 gm/mi possible.

(e) Waiver up to 1.5 gm/mi possible for diesel or innovative technology.

(f) 1.0 gm/mi allowed with 100,000 mile durability. There is also a 0.7 gm/mi option which requires acceptance of recall liability for seven years or 75,000 miles.

1.3 COMPUTER MODELLING OF DIESEL ENGINES

The major feature that governs the complexity of the processes occurring in a diesel engine is that all processes are unsteady. The modelling of the unsteady phenomenon is normally approached by assuming quasi-steady processes.

1.3.1 IMPORTANCE OF MODELLING

The development of a new design of engine can be time-consuming, laborious and expensive. If an analytical method is available for calculating performance and emission, development time and cost can be significantly reduced. Suitable models can also be used to perform parametric studies and hence to help optimise engine design and operating parameters such as injection and valve timings, matching the fuel injection to air swirl process, etc..

In an overview of engine modelling, Myers (16) attributes the increasingly important role of mathematical models in engine development to two main reasons:

- (i) The increasingly severe constraints of high fuel economy and low emissions have made engine optimisation a lengthy, expensive process.
- (ii) High speed computers have made practical the detailed cycle analysis currently being investigated.

Over the past three decades a large amount of work has been undertaken dealing with diesel engine modelling (1-4, 12, 14, 16-45, 50, 70-74) and gained much success in understanding the complex processes occurring during the engine cycle. However the description of diesel engines under transient conditions seems incomplete, especially the emissions under such conditions. One reason for this is that modelling diesel engines under transient conditions involves enormous computational time, even when employing the simplest of combustion models based on the 'filling and emptying' concept.

1.3.2 DIESEL ENGINE COMBUSTION MODELLING

The combustion model is one of the key components in the simulation of the complete diesel engine operating cycle. The various models available in the literature are discussed in detail below.

1.3.2.1 Types of Combustion Model

Heywood (17) and Mehta (18) both give reviews on various models for this purpose. There are many ways of classifying diesel combustion models and this has been illustrated by Mehta (18) in figure (1.3). Based on the approach used, the models available can be broadly classified into two categories:

- (i) Thermodynamic or Phenomenological
- (ii) Multi-dimensional

The usefulness, accuracy and running costs of these models vary tremendously.

Phenomenological Models

These models are also referred to as thermodynamic models as they are structured around the thermodynamic analysis of engine cylinder contents and empirical or semi-empirical process descriptions.

Depending on the process description they are classified as zero-dimensional or quasi-dimensional models. For both zero-dimensional and quasi-dimensional modelling the analysis of the induction, compression and exhaust are basically the same. The flow into/out of the engine cylinder during the induction /exhaust processes is handled using quasi-steady one dimensional flow equations. Mass flows past valves and other restrictions are modelled by the equations for isentropic adiabatic flow through an orifice. A discharge coefficient is used to relate the effective area for the particular constriction to the ideal area for the isentropic flow. System pressures upstream and downstream of the restriction are used for the pressure ratio in these calculations.

In the zero-dimensional models, the heat release rate (or fuel burning rate) is defined by an empirical expression which is obtained by matching certain mathematical function with the engine experimental heat release data. In a quasi-dimensional model an attempt is made to predict the rate of burning from more

fundamental physical quantities such as, fuel injection, atomisation and droplet distribution, air entrainment, turbulent mixing, heat transfer, etc.

Multi-Dimensional Models

Multi-dimensional models have, in principle, the capability of predicting details of fluid flow within the engine cylinder and the propagation rate and geometrical shape of the flame. They require detailed finite difference formulations in which the engine combustion chamber geometry is represented by a large number of cells. Mass, momentum and energy conservation are applied to the cylinder charge and the injected fuel, along with appropriate descriptions of the turbulence processes, chemical processes, boundary layer processes for each of the cells. A solution of the algorithms with appropriate boundary and initial conditions yields detailed information about the combustion process such as spatial and temporal distributions of velocity, temperature, and species concentration.

Multi-dimensional models have an advantage over phenomenological models in that they yield detailed information concerning influences of engine geometry on flow. Since finite cell representation of geometry is involved, the amount of iterative calculations and memory storage for cell data must be large. Fast CPUs and large memory are required, hence they are used for combustion chamber modelling rather than complete engine modelling. For parametric studies of the effects of changes in design and operating variables on engine performance, efficiency and emissions, the phenomenological models are better suited due to their simplicity and comparatively smaller computational requirements and costs.

1.3.2.2 DISCUSSION OF SELECTED COMBUSTION MODELS REPORTED IN THE LITERATURE

As mentioned in the last section, multi-dimensional models, such as the ones reported by Amsden et al (19), Watkins et al (20) and Errera et al (21), are useful where details of engine geometry and flow are essential. However, because of the heavy cost, these models are less suited for engine diagnostic, parametric and

engine transient studies. So the following discussion is focused on a number of selected phenomenological models available in the literature, which are considered relevant to the study here.

Detailed discussion of the various phenomenological combustion models has been provided by Mehta (18) and only a brief overview is outlined here with focus on the models which form the basis of the present work.

Zero-dimensional Models

The early phenomenological models (Lyn (22), Whitehouse and Way (23), Borman (24), Shipinski et al (25), Woschni and Anisits (26)) were zero-dimensional in description. They were mainly concerned with basic cycle thermodynamics for evaluating performance as a function of engine operating conditions. These models were based on the assumptions such as the homogeneous mixture, thermodynamic equilibrium and absence of flow characterisation. One of the major drawbacks of such models is that satisfactory mixing is always assumed. Therefore, the situation of poor mixing hence poor combustion and pollutant formation cannot be evaluated by these models.

In these models the overall effects of the combustion are represented by the apparent heat release rate (or fuel burning rate), as termed by Borman (24), which is defined by an empirical expression obtained by matching certain mathematical function with the engine experimental heat release data.

Both Shipinski et al (25) and Woschni and Anisits (26) based their heat release correlations on the Wiebe function (27):

$$FB = 1 - \exp [-K_2(t)^{(K_1+1)}] \quad (1.1)$$

where FB = fraction of fuel burnt/total injected per cycle per cylinder

t = time from ignition, non-dimensionalised over the total time allowed for combustion [= $\Delta t/\Delta t_{com}$]

K_1 = shape factor for the curve

K_2 = combustion efficiency coefficient.

They also attempted to link the shape factor, K_1 , to the engine operating conditions. However, their correlations were not satisfactory over a wide range of engine speed and load.

Diesel combustion has a two-part characteristic, broadly related to the premixed and diffusion burning phases, as shown in figure (1.1). Watson et al (28) suggested that a curve of this type would be produced by combining two functions, with a phase proportionality factor:

$$FB = \beta f_1(t) + (1-\beta)f_2(t) \quad (1.2)$$

where $f_1(t)$ = 'premixed' burning function

$f_2(t)$ = 'diffusion' burning function

β = phase proportionality factor of premixed to total burning.

The diffusion burning function [$f_2(t)$] is based on the Wiebe function, but [$f_1(t)$] is an additional term:

$$f_1(t) = 1 - (1 - t^{K_1})^{K_2} \quad (1.3)$$

$$f_2(t) = 1 - \exp(-K_3 t^{K_4}) \quad (1.4)$$

The constants K_1 , K_2 , K_3 , K_4 and β are then correlated to the major controlling variables, engine speed, load and ignition delay for a particular engine.

Watson's model has been quite successful in evaluating engine performance over wide range of operating conditions.

Quasi-dimensional Models

Later phenomenological models were quasi-dimensional types (Shahed et al (29, 30), Hiroyasu and Kadota (31), Meguerdichian and Watson (32)). They dealt in greater detail with the heterogeneity of the mixture, and the flow characteristics of the fluid inside the engine cylinder, which are essential for the prediction of pollutant formation. The detailed description of fuel spray and air swirl motion brought one- or two-dimensionality into the models by introducing the concept such as multi-zone division of the fuel spray. However assumptions were adopted to simplify the calculation, hence to make the models cost effective. The following is a list of the major assumptions used by such models:

- (1) Conical configuration of the fuel spray
- (2) Calculation of air entrainment based on spray penetration
- (3) Droplet evaporation based on isolated droplet behaviour
- (4) Characterisation of turbulent fuel-air mixing
- (5) Solid body rotation for describing swirl motion
- (6) Calculation of burning rate in terms of local air fuel mixture conditions.

Since a fairly detailed study of the fluid mechanics and thermodynamics is involved, the comprehensive nature of these model enables investigations to be performed. Examples are :

- (i) To predict cylinder pressure for component stress calculation.
- (ii) To predict component temperature and hence thermal loading.
- (iii) To predict the exhaust emissions and hence evaluation of different methods to reduce these emissions.

Over the past few years, three models (1, 3, 4) have been developed at Loughborough University of Technology to simulate the steady state combustion and emission for Quiescent, Swirl and Wall-wetting engines respectively. All these models were based on the quasi-dimensional approach due to its competitiveness, in term of computational time, to produce accurate diagnostic and parametric

results of performance over the full load range of an engine. The general approach of these models is to calculate the fuel injection and air entrainment, and then distribute the fuel injected and air entrained into small individual zones within a fuel spray, according to the velocity and temperature profiles respectively. Stoichiometric combustion is postulated in these zones to obtain the fuel burning rate. The simultaneous solution of the energy equation, the equation of state and the burning rate equation in the burning zones yields instantaneous pressure, temperature and volume values. Although still lacking the accuracy to correctly predict magnitudes of the exhaust emissions, they do predict the general trends favourably. Figures (1.4)- (1.6) show the typical engine performance and emission results from these models respectively.

1.3.3 MODELLING OF ENGINES UNDER TRANSIENT CONDITIONS

1.3.3.1 Types of Transient Model

Any dynamic system can be placed in one of two categories - linear or non-linear. Clearly there are many sources of non-linearity in diesel engines, however the mathematical models available from the literature use different methods to simulate the engine under transient conditions. Watson and Marzouk (40) have classified these models into three categories:

- i. Linear
- ii. Quasi-linear
- iii. Non-linear

A brief discussion of the usefulness and accuracy of these three types of models is provided below and a detailed survey of various transient models reported in the literature will be conducted in the next section.

Linear Models

The early dynamic models (Welbourn et al (33), Bowns (34)) belong to this category. They are based on either continuous control or sampled data concepts and have been used to study the stability of engine governors. Compared with the continuous control method, the sampled data approach is closer to the discontinuous nature of the diesel engine operations, but the assumption of a constant sampling interval has imposed a major handicap in this approach. These models should be limited to small perturbations in engine speed. Thus they may be only used in speed governing of engines and associated stability studies.

Quasi-linear Models

Quasi-linear models, such as the ones reported in references (Ledger and Wamsley (35), Mencik et al (36), Tsai and Goyal (37), Rackmil et al (38)), are based on steady state characteristics of the engine. In essence, these models link steady state experimental data representing engine thermo and fluid dynamics with dynamic models of the mechanical components. These models have the advantage of simplicity and short run time, such that real time simulation may be possible and are useful for control system development purposes. The major drawback of such models is their heavy reliance on experimental data, particularly at over-rich air-fuel ratio conditions. Therefore, this type of model tends to be valid for a relatively narrow operating range.

Non-linear Models

The non-linear models (Winterbone et al (39), Watson and Marzouk (40), Goyal (41), Wallace and Way (42), Watson and Janota (43), Watson (44), Jones et al (45)) are based on engine cycle simulation, which considers the thermo and fluid dynamics of engine operation under both steady state and transient conditions. Equations for the conservation of mass and energy are solved assuming quasi-steady processes for very short time intervals. The models are, in principle, capable of accurate performance prediction under both steady and transient

conditions and are widely used by the diesel engine manufacturers for engine design, research and development. One of the factors limiting the application of such models is the excessive computer processing time required to obtain an acceptable solution.

1.3.3.2 Description of Various Transient Models from Literature

A list of useful dynamic models available from the literature is given in Table 1.2.

In a very early attempt, Welbourn et al (33) applied continuous control theory to the problem of engine governing and their work gave an insight into the principles involved and made possible the design and selection of governors for diesel engines. Welbourn et al regarded the nature of engine operation as a series of pulses (injection events). However, in their theoretical approach they dealt with the discrete nature of the pulses by incorporating an arbitrary time delay involved in the engine-governor loop. This method has limitations as it does not recognise the fact that the engine can only be controlled at the instant of injection and no matter how sophisticated the governing system may be, it is inactive between the injection events.

To account for the pulsed nature of the engine, Bowns (34) has used the sampled data control theory in his work. Compared with the continuous control methods, the sampled data approach gives a more accurate representation of the dynamic lags in the engine and therefore allows a more accurate assessment of the stability of any engine-governor combination to be conducted. However the assumption of a constant sampling interval adopted in the work has limited the usage of this approach (permissible variation of engine speed is less than 10 percent).

The quasi-linear model of Ledger and Walmsley (35) is developed on the basis of steady-state characteristics of the engine. In the model, the engine brake torque is empirically related to the fuel/cycle as:

$$\text{brake torque} \propto \text{bmep} = a * \text{fuel/cycle} - bN_E$$

(if trapped air/fuel ≥ 19)

$= 1/19 * \text{air/cycle}$

(if trapped air/fuel < 19) (1.5)

where N_E = Engine speed

a,b = Empirical constants

The governor was represented by its transfer function which was obtained from analysing test data. For the fuel pump, Ledger and Walmsley assumed that the steady state fuel output was a linear function of the rack movement and was independent of engine speed. Furthermore, the engine airflow and temperature rise of the working medium are linked with engine speed and trapped air-fuel ratio empirically. This model was developed for a hybrid computer and constitutes a further step towards the dynamic simulation of diesel engines. However, since a large amount of experimental data is involved, this model is valid only for the narrow operating range for which it was developed.

The work of Mencik et al (36) is focused on the simulation of wide-open throttle vehicle performance. Special consideration is given to the initial stages of the simulation including the buildup of engine speed and torque from the idle conditions for automatic transmission vehicles and the clutch engagement process for manual transmission vehicles. The engine thermo and fluid dynamics is represented by the engine torque output experimentally obtained as a function of engine speed. This torque, combined with the load torque, determines the acceleration of the engine. Since the engine torque output is normally obtained from steady state engine dynamometer tests, all transient effects on engine torque are ignored.

The quasi-linear dynamic model of Tsai and Goyal (37) is developed for control system analysis and design. Linearised equations are obtained for the governor, the combustion process and the engine, taking into account the variation of the essential parameters over a wide range of operating conditions (from 600 to 2400

RPM). The model uses as its basis the steady state characteristics of the engine at the beginning and end of a change in input or load demand. The changes from one state to another are characterised by linear transfer functions of either constant or variable parameters. The engine combustion process is represented by the steady state fuel-torque curves at different engine speeds, obtained from the standard engine test data. The change in engine torque follows that of injected fuel after a time delay which varies in length by the time between successive engine firings. Tsai and Goyal have shown that for four-stroke cycle, six cylinder high speed engine, this delay (τ) can be approximated by a first order lag element

$$\tau = 1/(0.65*s*t_c+1) \quad (1.6)$$

where s = the Laplace variable

t_c = time constant = $20/N_E$ (sec)

N_E = engine speed (rpm)

This model can achieve real time simulation (about 1 second for 1 second real engine running time) and has been applied to the engine control analysis and design over a much wider range of operating conditions (from 600 to 2400 RPM), compared with the model by Bowns (permissible variation of engine speed is less than 10 percent).

The recent quasi-linear model of Rackmil et al (38) has been developed for a locomotive diesel engine and electrohydraulic governor. In the model, a regression formulation based on engine dynamometer data for an experimental engine is used to calculate engine brake torque output, averaged over all cylinders. The regression represents the steady state engine brake torque as a function of rack position and engine speed. Figure (1.7) shows the normalised brake torque calculated from the regression over the full operating range of the engine (from 200 to 900 RPM and from zero fuelling to high levels of fuelling). Corrections are applied to the calculated steady state torque to account for dynamic deviations of in-cylinder trapped air-fuel ratio from the steady state value. This can take into account some transient effects on engine torque. The delay in torque produced in response to a change in fuelling level is accounted in three parts:

- (1) time from rack position change until fuel is injected into any cylinder,
- (2) time for fuel to burn in a given cylinder and produce a torque moment at the crank throw,
- (3) time for the new torque level to be produced in sufficient number of cylinders.

The regression of the engine fuelling and brake torque involves a large amount of experimental data, which makes such a model useful only in investigating the overall responses of the engine and governor where detailed thermo and fluid dynamics is not essential.

Non-linear models were initiated in Britain by the work of Winterbone et al (39), Watson and Marzouk (40) and in America by that of Goyal (41) in the late 1970s. All these investigators based their work on the 'filling and emptying' concept. This concept is a numerical integration, on a time stepping basis, of the coupled first order differential equations which result from mass and energy conservation principles. The dynamics of the governor and engine are evaluated from momentum conservation.

In the model of Winterbone et al (39), the engine torque output is obtained from an ideal cycle representation of the engine closed period, with an allowance for heat transfer from the cylinder gases during the expansion stroke. The peak cylinder pressure and a diagram factor were both provided as functions of engine speed and load. They found that the idealised form of the combustion period resulted in an overestimate of power produced. To correct this overestimate, they used an empirically obtained diagram factor which was a function of load, i.e.

$$\xi_D = 0.86 - (bmep - 4)/100 \quad (1.7)$$

They further assumed that friction was proportional to engine speed so that the net output of the engine could be evaluated from the ideal imep as:

$$bmep = (imep - K_{fr} N_E) \xi_D \quad (1.8)$$

Compared with the quasi-linear models (35-38), the model of Winterbone et al requires less empirical data because of the more fundamental representation of the engine system by the step by step 'filling and emptying' method. However the ideal cycle representation of the engine closed period has limited the usage of the model to applications such as transfer function evaluation.

To get a better account for the overall effects of the combustion. Watson and Marzouk (40), and Goyal (41) both employed a semi-empirical heat release rate approach in their models to simulate turbocharged diesel engines. Watson and Marzouk (40) postulated two algebraic expressions (equations (1.3) and (1.4)) to compute the heat release rate, and correlated them to an extensive set of results for a wide range of operating conditions. Their approach reproduced the two distinct phases of the burning characteristics of a diesel engine and formed the basis of many subsequent dynamic models (43, 44).

Goyal (41) calculated the heat release rate using Wiebe's function (equation (1.1)) and correlated the combustion efficient and shape constants to the air-fuel ratio, pressure and temperature at the beginning of combustion. The rest of his engine thermodynamic simulation was based on the work by Borman (24). Both Watson and Marzouk (40), and Goyal (41) have shown that their models could predict the overall engine responses and the effects of certain design changes on these responses with reasonable accuracy.

Arguing that simulation programs useful for industry should be compact and cost effective, Wallace and Way (42) replaced the conventional detailed step by step procedure by discrete calculation steps based on a simplified engine cycle, as illustrated in figure (1.8), in which compression (1-2) is assumed adiabatic, combustion is calculated by heat release model with heat transfer to coolant and expansion is again assumed adiabatic. Combustion is calculated in three discrete steps, corresponding to (a) constant volume, (b) constant pressure, (c) a transition period between constant pressure and adiabatic expansion. Effective combustion before 5° BTDC is ignored. Constant volume combustion (2-3) is assumed to apply

between 5° BTDC and 5° ATDC. The proportion of fuel burnt during this period is obtained from a simplified form of the Whitehouse-Way (23) formulation:

$$dm_f/d\theta = K (m_{f,inj} - m_f) \quad (1.9)$$

where m_f = mass of fuel prepared
 $m_{f,inj}$ = mass of fuel injected
 K = rate constant for fuel preparation

Wallace and Way have found that their model gave accuracies a little inferior to those of full step-by-step programs such as reported in references (40, 41), while run times were of the order of one tenth of the latter. Figure (1.9) gives the comparison of the predicted and experimental engine speed responses from the models of Watson and Marzouk (40), Goyal (41) and Wallace and Way (42). From this figure, it can be seen that the model by Goyal most accurately predicts the engine speed response (within 16%), while the models by Watson and marzouk, and Wallace and Way give accuracy 19% and 25% respectively.

The exhaust smoke during the transient operation of a diesel engine was first addressed by Watson and Janota (43), where they empirically correlated the smoke opacity to engine speed (N_E), overall equivalence ratio (ϕ) and the premixed burning fraction (β , relating premixed and diffusion burning in the combustion correlation, section 1.3.2.2). The correlation they obtained has the following form:

$$S/S_{ref} = a [N_E/N_{Eref}]^b [\phi/\phi_{ref}]^c \exp[d(\beta/\beta_{ref})] \quad (1.10)$$

where a-d are empirical constants and the subscript ref denotes to the reference values. In the later work by Watson (44), he correlated equation (1.10) to two single cylinder engines and gave the values of a-d as

$$a = 0.177 \quad b = 0.54 \quad c = 3.73 \quad d = 1.734$$

For the multicylinder engine, the optimum index on equivalence ratio was higher ($c=6.62$).

Equation (1.10) has not found much use because of the empirical nature and the constants a-d tend to vary from engine to engine. In the present study, equation (1.10) with the values of the constants suggested by Watson was tested with experimental data supplied by FORD motor company and the comparison is given in figure (1.10). It can be noted from the figure that big discrepancies exist, indicating the invalidity of the values of a-d mentioned above.

The recent work of Jones et al (45) has reported a multi-processor model to simulate diesel engines under both steady and transient conditions for condition monitoring applications. The uniqueness of their work is the application of the parallel processing hardware (multi- processors), which brings a dramatic reduction of computation time compared to supermini and mainframe computers, and is very compact and relatively inexpensive. The mathematical model is based on the filling and emptying method. In order to permit parallel processing, the engine model must be divided into a number of computational tasks which should be relatively independent and can be performed concurrently. According to Jones et al, the task division is vitally important since it determines the time spent by the processors in communicating with one another, and the structure of the program itself. In concurrent situations, however, there are no general rules for the task division and it has to be treated in an *ad hoc* way.

In the work by Jones et al, a shared memory multiprocessing computer consisting of an array of 32 bit Motorola 68020 processors is used to allow concurrent solution of the equations resulted from the mass, energy and momentum conservation principles and the language used for the program modules is BCPL.

A summary of the salient features of the quasi- and non- linear models described above are provided in Table 1.3.

1.4 DISCUSSION OF THE LITERATURE STUDY

With the ever-increasing demands for a clean environment, combustion behaviour and pollutant formation during transient excursions of an engine need to be investigated. However there is no fundamental modelling of exhaust emissions during transient periods from the literature. One reason for this is that the modelling of diesel engines under transient conditions involves enormous computational time. Even employing the single zone 'filling and emptying' method, the acceptable solution time is likely to be in the order of 3 - 30 CPU hours on all but the most powerful mainframe computers for a 5 seconds transient period of the engine. The problems of including detailed description of fuel jet air entrainment and turbulent mixing processes, which are essential in the prediction of the exhaust emissions are obvious.

Approximations and simplifications must be introduced to simulate a complex device such as a diesel engine. Some of the simplifications are made because of a lack of fundamental understanding, but others simply due to the practical limits of computing cost. The application of concurrent process computing methods can reduce the computing time, by performing independent tasks within the simulation at the same time, on separate processors, instead of reducing computing time at the cost of accuracy.

In terms of the methods adopted in the simulation, the models can be generally classified into the following categories:

- (i) Sequential model - based on sequential processing methods (1-4,12, 14, 16-18, 22-44,50,70-74)
- (ii) Concurrentmodel- based on parallel processing methods, such as the pioneering work of Jones et al (45).

The concurrent models have the potential to offer more complexity with relatively shorter computational time. This has been discussed by Winterbone (46) in the chairman's introduction to the conference entitled "Computers in Engine Technology".

In conclusion, there is a need for fundamental understanding of the performance, especially exhaust emissions for an engine under transient conditions. To achieve this, detailed but inexpensive models based on the concurrent processing methods are required.

1.5 OUTLINE OF THE PROJECT OBJECTIVES

The present work is aimed at providing an interactive model to simulate a direct injection diesel engine under both steady and transient conditions, based on the application of concurrent process computing methods. Starting with the modelling of the induction, injection, air entrainment, fuel air mixing, combustion and emission processes in a naturally aspirated diesel engine at steady conditions, based on the work reported in early publications (1, 2, 3, 4), the friction, fuel pump, governor and load dynamics will be investigated to study the drive train transient behaviour. The modelling work will be developed on a concurrent computer consisting of four INMOS T800 transputers. The sequential model will be written in FORTRAN 77 and intended to be implemented on all machines equipped with standard FORTRAN 77 compilers. The sequential code will be divided and run concurrently by using OCCAM as a harness. Experiments will be carried out to measure various performance and emission data to validate the model.

Table 1.2 List of Dynamic Models of Diesel Engine

Author(s)		Year	Purpose	Concept Used
1.	Wellbourn Roberts Fuller (33)	1959	Stability of governor	Continuous control theory (Linear)
2.	Bowns (34)	1971	Dynamic characteristics of engines	Sampled data concept (Linear)
3.	Ledger Walmsley(35)	1971	Performance	Steady-state characteristics (Quasi-linear)
4.	Mencik Tobler Blumberg(36)	1978	Wide-open throttle performance	Steady-state characteristics (Quasi-linear)
5.	Tsai Goyal (37)	1986	Control analysis and design	Steady-state characteristics (Quasi-linear)
6.	Rackmil Blumberg et al (38)	1988	Performance of locomotive engine	Steady-state characteristics (Quasi-linear)
7.	Winterbone Thiruarooran Wellstead(39)	1977	Transfer function evaluation	Filling and emptying (Non-linear)
8.	Watson Marzouk (40)	1977	Performance	Filling and emptying (Non-linear)
9.	Goyal (41)	1978	Performance	Filling and emptying (Non-linear)
10.	Wallace Way (42)	1980	Performance	Filling and emptying (Non-linear)
11.	Watson	1980	Performance and smoke emission	Filling and emptying (Non-linear)
12.	Janota (43)			
13.	Watson (44)	1984	Control system development	Filling and emptying (Non-linear)
	Jones Charlton Daniels (45)	1987	Condition monitoring	Filling and emptying (Non-linear, parallel)

Table 1.3 Summary of the Process Description in the Non-linear Models

Process	Winterbone Thiruarooran Wellstead (39)	Watson Marzouk (40)	Goyal (41)	Wallace Way (42)	Watson Jonota (43)	Watson (44)	Jones Charlton Daniels (45)
Engine thermo and fluid dynamics	Filling and emptying	Filling and emptying	Filling and emptying	Filling and emptying	Filling and emptying	Filling and emptying	Filling and emptying
Exhaust emission	Not accounted	Not accounted	Not accounted	Not accounted	Simple empirical correlation for soot emission	Simple empirical correlation for soot emission	Not accounted
Friction	Assuming friction is proportional to engine speed	Use Chen and Flynn's equation	Use Chen and Flynn's equation	N	Use Chen and Flynn's equation	Use Chen and Flynn's equation	N
Engine load dynamics	Y/N	Input load torque as a function of time	Input load torque in a tabular form	N	Input load torque as a function of time	Input load torque as a function of time	N

Table 1.3 Summary of the Process Description in the Non-linear Models

Governor dynamics	Rate of movement of rack position proportional to difference between instantaneous and steady droops	A second order differential equation based on momentum conservation	N	N	A second order differential equation based on momentum conservation	A second order differential equation based on momentum conservation	N
Pump	N	Steady-state fuel pump characteristics	Steady-state fuel pump characteristics	N	Steady-state fuel pump characteristics	Steady-state fuel pump characteristics	N

Figure 1.1 Typical Indicator Diagrams of Diesel Combustion (9)

Figure 1.2 Acceleration of a Turbocharged Diesel Engine Against a Steady Load Showing the Tradeoff between Response and Exhaust Smoke (Experimental Data)

(12)

Figure 1.3 Classification of Diesel Engine Models (18)

Figure 1.4 Comparison of Experimental Cylinder Pressure with the Prediction from the Model of Dent and Mehta for a Quiescent Engine (1)

Cylinder pressure diagram

Fuel mass fractions

Soot formation and oxidation concentrations

Figure 10: Nitric oxide concentrations

Figure 1.5 Typical Results from the Model of Kyriakides et al (3)

Figure 1.6 Comparison of Experimental Cylinder Pressure with the Prediction from the Model of Sindano (with Different Injection Timing (4))

Figure 1.7 Normalised Brake Torque Versus Engine Speed and Rack for the Experimental Engine (38)

Figure 1.8 Simplified Engine Cycle Used by Wallace and Way (42)

Figure 1.9 Comparison of the Predicted and Experimental Engine Speed Responses from the Models of Watson and Marzouk (40), Goyal (41) and Wallace and Way (42)

(a) Watson and Marzouk's

(c) Wallace and Way's

(b) Goyal's

Figure 1.9 Comparison of the Predicted and Experimental Engine Speed Responses from the Models of Watson and Marzouk (40), Goyal (41) and Wallace and Way (42)

Figure1.10 Comparison of Exhaust Smoke Emission between Experimental and Predicted by Watson's Correlation (44)

CHAPTER 2

BASIS OF THE PROPOSED MODEL

2.1 INTRODUCTION

In this chapter, a theoretical analysis of the direct injection diesel engine is presented, forming a basis of a dynamic model developed to simulate the performance and emission of the engine under both steady and transient conditions. The model accounts for the time discrete behaviour of the diesel engine, which is governed by the intermittent fuel injection and combustion processes and results in successive torque pulse at the crankshaft.

In order to derive the mathematical simulation, the engine has been subdivided into seven interactive component systems which can be modelled individually, engine induction-exhaust, combustion, exhaust emissions, fuel supply, friction loss, governor and load dynamics. From the control standpoint, the basic configuration of the engine system can be presented in the form of a closed control loop, as shown in figure (2.1). The variable to be controlled is the engine speed, and the governor is the controller. The throttle position (Th) represents the demanded engine speed, and the actual engine speed (NE) is measured and fed back to the controller for comparison with the demanded. The governor senses the difference between the actual engine speed and the throttle demand and controls the engine by a proportional movement of the control spool. According to this movement and the instantaneous engine speed, the quantity of fuel per engine cycle delivered from the pump is determined, and the time of injection of the fuel. The fuel injected into the cylinder is then mixed with the air induced into the fuel sprays and the mixture is burnt to produce a torque pulse at the crankshaft. The net torque, after deducting friction loss, is available to overcome the instantaneous inertia of the engine parts

and whatever load is externally applied to the engine, gear box, driveshaft, back axle and driving wheels of a vehicle say.

The modelling of the component systems in the control loop is presented in the following sections.

2.2 PHYSICAL BASIS OF THE COMPONENT SYSTEM MODELLING

The modelling of the engine at steady state conditions is based on the work of Dent and Mehta (1), Kyriakides et al (3) and Sindano (4). All these investigators based their models on the quasi-dimensional (multi-zone) approach, in which the developing fuel sprays within the engine cylinder were considered to be divided into a number of sectors over which the fuel and air entrained into the sprays were distributed, and combustion and pollutant formation in these sectors considered. This type of phenomenological model has been shown to accurately represent the steady state combustion events in the cylinder during an engine cycle (figures (1.4)-(1.6)).

However, because of the heavy demands of the computing time of this type of model, due to the sectorisation of the fuel spray, it is inappropriate for the simulation of an engine during a transient process. As an example, it takes about 7 minutes (CPU time) to run the combustion model by Kyriakides et al (3) from the inlet valve closure to the end of combustion on a conventional mainframe, such as Honeywell Multics. To simulate an engine under a transient condition, say for 10 seconds, requires about 200 engine cycles to be analysed (if the engine is running at medium speed). This will take about 23 hours to run a transient model based on the phenomenological combustion model outlined above. Therefore, in the present study, the burning zone is not sectorised, but a simple two zone (burning fuel spray/unburnt surroundings) model has been adopted, which has been found to

reduce the computing time by a factor of ten, without a significant penalty in the trend of the smoke and NO_x emission predictions, since the modelling of these emissions in the existing models (3) and (4) are in essence of the global type.

In the present study, in common with Watson (44), it is assumed that all cylinders in a multicylinder engine behave identically at the same point of their thermodynamic cycle. Hence, modelling needs only be undertaken on a single cylinder.

In order to make transient predictions, four further submodels are required, i.e. friction loss, governor, fuel delivery and engine-load dynamics.

2.2.1 ENGINE COMBUSTION PROCESS

To facilitate the analysis, the cylinder is divided into two regions, jet and surrounding, as illustrated in figure (2.2). When injection starts, the air in the surrounding is transferred into the fuel jet zone by an entrainment mechanism. At the end of injection, turbulent mixing dominates the air entrainment process. The air transferred into the jet is mixed with the fuel and combustion occurs after ignition delay.

The treatment of the fuel spray as a single zone reduced computing time dramatically however, the fuel burning rate cannot be predicted accurately due to the lack of information such as the distributions of the fuel and air concentrations in the burning region. In the present work, an apparent fuel burning law approach based on the work of Watson et al (28) is adopted to account for the overall effects of the combustion. The heat transfer to the cylinder surface is according to Woschni (47) and is averaged over the whole combustion volume. The energy conservation equations are applied to the two zones :

i) Jet Fluids:

$$\begin{aligned}
 [m_{air} C_{vair} + m_{combp} C_{vcombp} + m_{vap} C_{vvap} + m_{liq} C_{pli}q] dT_j + p dv_j = \\
 (U_{liq0} - U_{vap}) dm_{vap} + (h_{air, sr} - U_{air, jet}) dm_{air} + (U_{liq0} - U_{liq, jet}) dm_{liq} + \\
 dQ_{rel} + dQ_{tr, jet}
 \end{aligned}
 \tag{2.1}$$

ii) Surroundings:

$$m_{air, sr} C_{vair} dT - p dv_j = (h_{air, sr} - U_{air, sr}) dm_{air, sr} - p dv_p + dQ_{tr, sr}
 \tag{2.2}$$

Alongside these two energy conservation equations, the equations of state are also applied to the two zones:

i) Jet Fluids:

$$\begin{aligned}
 (R_{combp} m_{combp} + R_{air} m_{air} + R_{vap} m_{vap}) dT_j - p dv_j - v_j dp = \\
 -(R_{air} dm_{air} + R_{vap} dm_{vap} + R_{combp} dm_{combp}) T_j
 \end{aligned}
 \tag{2.3}$$

ii) Surrounding:

$$dm_{air, sr}/m_{air, sr} + dT/T = dv_{sr}/v + dp/p
 \tag{2.4}$$

Equations (2.1) to (2.4) form an energy matrix which contains four variables concerning changes in jet and surrounding conditions in the cylinder, the changes in the cylinder pressure (dp), jet and surrounding temperatures (dt_j and dt_a) and jet volume(dv_j). Gaussian elimination is used to solve this matrix to determine these four variables. The conditions inside the cylinder are therefore determined at each time step. The first two variables, (dp) and (dt_j) are then input to the emissions model to predict the exhaust smoke and NO_x.

Apart from abolishing the concept of sectorisation, the characterisations of the fundamental processes such as the fuel evaporation, entrainment and mixing, ignition delay and fuel burning are also changed to enable the prediction of the transient operations. These changes are discussed below.

2.2.1.1 Fuel Evaporation

For the fuel evaporation process an isolated droplet model has been assumed and calculation of vaporisation rate of droplets ($\dot{m}''$) follows Spalding (48):

$$\dot{m}'' = g^* \ln(1+B) \quad (2.5)$$

where g^* is the convective mass transfer conductance and B is mass transfer number.

From Reynolds Analogy, g^* may be defined by the following ratio:

$$g^* = h/C_p \quad (2.6)$$

where h is the heat transfer coefficient and C_p the specific heat.

Assuming Lewis number equal to unity, g^* can be obtained from Nusselt number (Nu) as:

$$g^* = h/C_p = k Nu/(d_p C_p) \quad (2.7)$$

where k is the thermal conductivity and d_p the mean droplet diameter.

Simmons (49) has conducted a large number of tests on fuel atomisation and found that for more than 95% of the data points, the mean droplet diameter (d_p) is 1.2 times of Sauter Mean Diameter (SMD), which can be evaluated from the correlation of Knight (50):

$$\text{SMD} = 8.0 P^{-0.458} Q^{0.209} v_f^{0.215} \quad (2.8)$$

Kamimoto and Matsuoka (51) have shown that the correlation of Ranz and Marshall (52) gave reasonable trends in evaluating the spray behaviour based on an isolated droplet assumption. This correlation gives Nusselt number (Nu) in terms of Reynolds (Re) and Prandtl numbers (Pr) as:

$$\text{Nu} = 2 + 0.6 \text{Re}^{1/2} \text{Pr}^{1/3} \quad (2.9)$$

where Re is based on the mean droplet diameter and Prandtl number is taken equal to 0.7.

When ignition starts, Nu is taken equal to 2, following Mehta(18).

The value of mass transfer number B is obtained from conservation laws:

* For mass conservation, the transfer number is represented by:

$$B_m = (m_g - m_s) / (m_s - 1) \quad (2.10)$$

* For energy conservation, the transfer number is:

$$B_h = C_p (T_g - T_s) / L \quad (2.11)$$

where m and T represent the fuel vapour mass fraction and temperature respectively. Subscript g and s denote to the air vapour mixture surrounding the droplets and droplet surface respectively. L is the specific enthalpy of vaporisation of the fuel and C_p the thermal capacity of air-vapour mixture.

Assuming that pressure gradient in the direction of flow is negligible, m_g has a value of zero. By assuming a perfect gas mixture, the mole fraction of the fuel vapour is equal to the partial pressure fraction (P_s/P), thus the fuel vapour mass fraction m_s is expressed as:

$$m_s = ((P_s/P) W_f) / W \quad (2.12)$$

where W is the equivalent molecular weight of the mixture determined by:

$$W = (P_s/P) W_f + [1 - (P_s/P)]W_a \quad (2.13)$$

where W_a and W_f are the molecular weights of air and fuel respectively. P is the total pressure in the bulk stream. Substituting equation (2.13) in (2.12) results in:

$$m_s = P_s / [\gamma P + (1-\gamma)P_s] \quad (2.14)$$

where γ is the ratio of W_a/W_f .

The saturation vapour pressure on the droplet surface (P_s) is evaluated from Antoine's relationship which has the following form:

$$\text{Log}_{10} P_s = K_1 + K_2 / (T_s + K_3) \quad (2.15)$$

where T_s is the droplet surface temperature and $K_1 - K_3$ are constants. In common with Sindano (4), the following values applied to n-Dodecane are used to approximate diesel fuel:

$$K_1 = 7.0$$

$$K_2 = -1625.93$$

$$K_3 = -92.9$$

The evaluation of T_s is based on an iteration between equation (2.10) and (2.11) to reach a common solution due to intersection of B_m and B_h .

Further implementation of the fuel evaporation routine described here will be discussed later.

2.2.1.2 The Transportation of Air into the Burning Zone

During injection, air entrainment is calculated from the equation used by Escudier (53) to compute the change in air entrainment rate with distance along the trajectory (dm_a/ds) of a burning jet in a cross flow of air as:

$$dm_a/ds = 2\pi r_j (\rho_m \rho_a)^{1/2} [\alpha U_m - V_t] + \lambda |V_n| \quad (2.16)$$

where

r_j is the local radius of the jet; ρ_m and ρ_a the densities of the jet and surrounding air respectively; U_m is the local velocity on the jet trajectory, V_t and V_n are the tangential and normal components of the cross-flow to the trajectory, α and λ are experimental constants. Dent et al (2) have found that good agreement with cylinder pressure data could be obtained by using the values of α and λ postulated by Sinnamon et al (54) and substitution of ρ_m for ρ_a in equation (2.16).

The expression for change in jet centreline trajectory obtained by Sinnamon et al (54), combined with the correlation of Chiu et al (55) to evaluate the change in jet penetration due to swirl, determines the instantaneous position of the jet tip relative to the nozzle orifice. The criterion of Skifstad (56) was then used to determine whether the jet impingement occurs. If so, the wall jet model developed by Kyriakides et al (3) is called to evaluate the entrainment in the wall jet.

At the end of injection period, the jet momentum due to fuel injection will be zero and the mixing process will be controlled by the turbulent kinetic energy dissipation in the fuel jets. Dent and Mehta (1) relate the mass transfer rate of air into the burning products to the turbulent mixing rate as:

$$-dm_a/dt = C_1 (m_a / \tau) \quad (2.17)$$

where m_a is the mass of air available in the surrounding

τ is the turbulent mixing time

C_1 is a constant and evaluated by equating the time mean entrainment rate over the whole injection period to the right hand side of equation (2.17).

In the high swirl diesel engine, the turbulent mixing rate is considered to consist of two components: one due to the injection process and the other due to the existence of the swirl. Considering that the turbulent mixing rate was controlled by the energy input to the fuel jet during the injection process, Dent (57) relates the first component to the engine operating conditions as :

$$1 / \tau_{inj} = C_1 N^{1/2} [N_E / \theta_{ip}]^{3/2} [V_{fi} / (d_o^2 n_h)] \quad (2.18)$$

For the second one, Dent et al (2) considered a solid body forced vortex air swirl and expressed this rate as :

$$1 / \tau_{swirl} = [(\omega^2 d^2) / (32 \nu)]^{1/2} \quad (2.19)$$

The two rates are considered to act in parallel

$$R = R_{inj} R_{swirl} / (R_{inj} + R_{swirl}) \quad (2.20)$$

From equation (2.20), it can be seen that it is the slower mixing rate which controls the mixing process. Comparing the two rates suggests that R_{swirl} is the slower.

2.2.1.3 Ignition Delay

In developing the previous combustion model, Kyriakides et al (3) used experimental ignition delay data directly as input to the model. This approach not only limits the usefulness of the model but it is not practical under transient simulation.

Dent and Mehta (1) have conducted an extensive study of the available correlations of ignition delay and found that the empirical correlation of Hardenberg and Hase (58) was most suitable for quiescent engines. This correlation gives the ignition

delay (in time) in terms of charge temperature T (kelvins) and pressure p (bars) during the delay as:

$$ID (s) = [0.06 / N_E + 0.0012 L] \exp [E_A (1 / (R T C R^c)^{-1} - 1 / 17190) + (21.1 / (p C R^c - 12.4))^{0.63}] \quad (2.21)$$

where N_E is the engine speed (rpm)

L is stroke

E_A is the apparent activation energy (J/mole)

$$= 618,840 / (CN + 25)$$

CN = the fuel cetane number

A comparison between the calculated ignition delay data using Hardenberg and Hase's expression and the experimental ones listed in Table 2.1 is shown in figure (2.3). It is evident that the agreement is not satisfactory (Kyriakides has found the same but he avoid this problem by using experimental ignition delay data directly as input to the model).

From equation (2.21) it can be noted that when the engine speed is low, the increase in engine speed will result in a reduction in ignition delay. However when the engine speed is high ($N_E > 2000$ rpm), $0.06/N_E$ is significantly lower than $0.0012 L$ ($L = 90.54$ mm), the increase in engine speed would have little effect on ignition delay. In reality, however, when the engine speed goes up, both air temperature and cylinder pressure increase, resulting in a reduction in ignition delay.

In the present study, a term $(1000/N)^{0.093}$ related to the engine speed is included in equation (2.21), in an attempt to account for the effect of engine speed on the ignition delay. The predicted ignition delay data using the modified expression is again compared with the experimental ones. The agreement, as shown in figure (2.4), is improved. The modified correlation is further tested with the AVL engine experimental data. Figure (2.5) gives the predicted results from the correlation of

Hardenberg and Hase and the modified one, compared with the engine experimental data. It can be seen from the figure that the modified correlation predicted the ignition delay closer to the engine experimental data.

2.2.1.4 Fuel Burning Rate

The overall effects of the combustion can be represented by the fuel burning rate. The Wiebe function (27) is the most widely used equation to express the cumulative fuel burnt in terms of dimensionless time:

$$FB = 1 - \exp [-K_2 t^{(K_1+1)}] \quad (2.22)$$

Differentiating equation (2.22) results in the fuel burning rate function :

$$FBR = (K_1 + 1) K_2 t^{K_1} \exp [-K_2 t^{(K_1 + 1)}] \quad (2.23)$$

where FB = fraction of fuel burnt/total injected per cycle per cylinder

FBR = non-dimensional fuel burning rate [=d(FB)/dt]

t = time from ignition, non-dimensionalised over the total time allowed for combustion [=Δt/Δt_{com}]

K₁ = shape factor for the curve

K₂ = combustion efficiency coefficient.

Initially Wiebe (27) derived equation (2.23) for homogeneous mixtures, a condition that may be true for a spark ignition engine.

Shipinski et al (25) correlated the heat release rate with the rate of injection by fitting equation (2.22) to experimentally determined heat release diagrams, over a range of engine test conditions on a high-speed swirl-type direct-injection engine. They also attempted to link the shape factor, K₁, to the overall air-fuel ratio and the fraction of the total fuel injected during the delay period. However, their

correlation was not satisfactory over a wide range of engine speed and load, and they concluded that the Wiebe function was not sufficiently flexible to account for the heat release pattern in a diesel engine which is characterised by two distinct phases.

Based on equation (2.22), Woschni and Anisits (26) assumed that one apparent heat release diagram would always be available and based their correlation on the change in apparent heat release rate relative to this known condition. The change in K_1 was related to changes in ignition delay (ID), engine speed (N_E), pressure (P_z) and temperature (T_z) at the beginning of compression as:

$$K_1 = K_{1ref} [ID_{ref}/ID]^a [P_z/P_{zref}] [T_{zref}/T_z] [N_{Eref}/N_E]^b \quad (2.24)$$

where a and b are empirical constants. The subscript ref refers to datum conditions at which the heat release rate is known.

Equations (2.24), incorporated with equation (2.23), is then used to predict the heat release rate at other conditions.

Unlike Shipinski's correlation, Woschni and Anisits' does not require the fuel injection rate as data and is therefore simpler to use. However, the actual heat release shape defined by the Wiebe function differs markedly in character to that measured on most types of diesel engine.

It is well established that the burning characteristics of an open-chamber diesel engine consists of two distinct phases: a premixed burning phase which occurs during the early stages of combustion and is characterised by relatively high heat release rates and short duration, and a diffusion burning phase which occupies most of the combustion period and is characterised by relatively low heat release rates. Wiebe function (equation (2.23)) cannot represent this two phase heat release pattern accurately. To quantify the fuel burnt by either mechanism, several investigators have proposed two functions based on Wiebe function. Watson et al

(28), for example, postulated two algebraic expressions, one for premixed burning and the other for diffusion controlled burning as:

$$M_p(t) = 1 - (1 - t^{C_{p1}})^{C_{p2}} \quad (\text{Cumulative})$$

$$\text{or } M_p'(t) = C_{p1} C_{p2} t^{(C_{p1} - 1)} (1 - t^{C_{p1}})^{(C_{p2} - 1)} \quad (\text{Rate}) \quad (2.25)$$

$$M_d(t) = 1 - \exp [-C_{d1} t^{C_{d2}}] \quad (\text{Cumulative})$$

$$\text{or } M_d'(t) = C_{d1} C_{d2} t^{(C_{d1} - 1)} \exp [-C_{d1} t^{C_{d2}}] \quad (\text{Rate}) \quad (2.26)$$

Where $M_p(t)$ and $M_d(t)$ are pre-mixed and diffusion fuel burning rates respectively.

C_{p1} , C_{p2} , C_{d1} and C_{d2} are shape factors.

The diffusion burning function [$M_d(t)$] is based on the Wiebe function, but the pre-mixed burning function [$M_p(t)$] is an additional term. Furthermore, a 'mode of burning factor', β , was also introduced by the above workers (28). This expresses the cumulative fuel burnt by pre-mixed burning as a fraction of the total fuel injected,

$$\beta = M_p(t)/M_t(t) \quad (2.27)$$

Consequently the total fuel burning rate ($M_t'(t)$) can be expressed as:

$$M_t'(t) = \beta M_p'(t) + (1-\beta) M_d'(t) \quad (2.28)$$

The constants C_{p1} , C_{p2} , C_{d1} , C_{d2} and β are then correlated to the major operating variables, i.e. engine speed, load and ignition delay for a particular engine. For two different types of engine, Watson et al (28) found the correlations for C_{p1} and C_{p2} were the same.

Miyamoto et al (59) also reported a similar approach to calculate the heat release rate. Their correlation is based on two Wiebe functions and has the following form:

$$\begin{aligned} dQ / d\theta = & 6.9 Q_p/\theta_p (M_p +1) (\theta/\theta_p)^{M_p} \exp [-6.9 (\theta/\theta_p)^{(M_p +1)}] + \\ & 6.9 Q_d/\theta_d (M_d+1) (\theta/\theta_d)^{M_d} \exp [-6.9 (\theta/\theta_d)^{(M_d +1)}] \end{aligned} \quad (2.29)$$

where the subscripts, p and d, refer to the pre-mixed and diffusive combustion parts, respectively. Q_p and Q_d are the amounts of heat release. θ_p and θ_d are the durations of heat release, and M_p and M_d are shape factors.

The six adjustable parameters M_p , M_d , Q_p , Q_d , θ_p , θ_d are then correlated to the major controlling variables, i.e. injection rate, ignition delay and load for a particular engine.

Since the correlations of Watson et al (28) and Miyamoto et al (59) are very similar and both work for a wide range of operating conditions, any one of them can be incorporated into a transient model. The work of Watson et al has been adopted in the present work mainly because there are five parameters to be adjusted while six in the correlation of Miyamoto et al. Therefore, in the present studies, equations (2.25), (2.26) and (2.28) are used to calculate the fuel burning rate. The five constants C_{p1} , C_{p2} , C_{d1} , C_{d2} and are correlated to engine speed, load and ignition delay for a 2.5 L high speed direct injection diesel engine, details of which can be found in in Appendix A. Extensive trials were carried out and the following correlations were found to give the best agreement between the predicted and the experimental cylinder pressures over a wide range of operating conditions (engine speed 1000-4000 rpm, A/F 24-33), with the correlations of C_{p1} and C_{p2} the same as suggested by Watson et al:

$$C_{p1} = 2 + 1.25 * 10^{-8} (\tau_{ig} N_E)^{2.4} \quad (2.30)$$

$$C_{p2} = 5000 \quad (2.31)$$

$$C_{d1} = 79.705 F_{tr}^{1.484} \quad (2.32)$$

$$C_{d2} = 0.581/F_{tr}^{0.739} \quad (2.33)$$

$$\beta = 1 - 1.0569 F_{tr}^{0.489}/\tau_{ig}^{0.55} \quad (2.34)$$

2.2.1.5 Heat Release Rate

For lean mixture ($\phi < 1$), and in the absence of dissociation effects, the heat release rate is proportional to the fuel burning rate (since thermodynamic equilibrium is assumed in the cylinder):

$$dQ_{rel} = dm_{vb} H_o \quad (2.35)$$

In the case of richer fuel mixtures, the above linear relationship is considered to hold until either dissociation starts or the charge equivalence ratio exceeded unity. Beyond the stoichiometric fuel/air ratio, any increase in the amount of reacting fuel would lead to a reduction in the cumulative heat release.

When injection starts, the fuel evaporation routine is called to evaluate the fuel evaporation rate and then air entrainment routine is executed to calculate the air in the surrounding entrained into the fuel jet zone and mixed with the fuel vapour. At the end of injection, turbulent mixing routine is called to evaluate the air-fuel mixing process. Combustion routine is called after ignition delay to determine the fuel and air burning rates (stoichiometric combustion is assumed) and heat release rate. Then energy balance routine is executed to determine the four variables concerning changes in jet and surrounding conditions in the cylinder, the changes in the cylinder pressure (dp), jet and surrounding temperatures (dt_j and dt_a) and jet volume(dv_j) at each time step. The first two variables, (dp) and (dt_j), together with the concentrations of combustion products and fuel vapour left in the jet, and

instantaneous air/fuel ratio (obtained from the air entrained into the jet deducting the air burnt and the fuel evaporated deducting the fuel burnt) are then input to the emission routines, which will be discussed in the next section, to predict the exhaust emissions of smoke and NO_x.

2.2.2 EXHAUST EMISSIONS

The details of the modelling of the NO_x and exhaust smoke emissions can be found in reference (1), (3) and (4). In the present study, a faster algorithm is incorporated in the calculation of the equilibrium of the combustion products, while several modifications have been made on the existing exhaust smoke emission modelling.

2.2.2.1 Modelling for NO_x Emission

Many researchers have shown that the NO formation is a non-equilibrium process and can be evaluated on the basis of the famous Zel'dovich mechanism (can predict NO emissions within 10% of those actually observed (9)). This is described by the following equations:

In order to obtain an approximate equation for the rate of the kinetic mechanism, the following assumptions are made:

1. The C-O-H system is in equilibrium and is not affected by dissociations of N₂
2. N atoms are assumed to change concentration by a quasi-steady process.

It has been illustrated by Kyriakides and et al (3) that the NO formation rate can be expressed by the following equation, based on the above two assumptions:

$$1/V \, d/dT([NO] V) = 2 (1-\alpha^2) R_1/(1 + \alpha K) \quad (2.38)$$

where $K = R_1/R_2$

$$R_1 = k_1 [N]_{eq} [NO]_{eq}$$

$$R_2 = k_2 [N]_{eq} [O_2]_{eq}$$

k_1 and k_2 are the forward rate constants for reactions represented by expressions (2.36) and (2.37) respectively. α is the ratio of NO concentration to its equilibrium concentration and the subscript eq refers to the equilibrium concentration of the corresponding species. The following section describes the evaluation of $[NO]_{eq}$, $[N]_{eq}$ and $[O_2]_{eq}$.

Solution for Equilibrium Compositions

In the existing model (4), eleven species are assumed in the equilibrium of the combustion products and the chemical reaction is approximated by the following expression:

The appropriate balanced equation may be written as :

$$\begin{aligned} y[C_n H_m + (n + m/4)(O_2 + 3.76 N_2)] = & x_1 H + x_2 O + x_3 N + x_4 H_2 + x_5 OH + \\ & x_6 CO + x_7 NO + x_8 O_2 + x_9 H_2O + \\ & x_{10} CO_2 + x_{11} N_2 \end{aligned} \quad (2.40)$$

Considering the atom mass balance for C, O, H, N and considering the equilibrium of the following reactions:

Eleven equations are obtained to determine the eleven variables x_1 - x_{11} . Newton-Raphson's method is employed to solve this eleven equation system.

In the work by Meintjes and Morgan (60), the Newton-Raphson and Gauss-Seidel methods are compared and it is concluded that the first method is slower but reliable, whilst the second one is fast but can be unreliable. Meintjes and Morgan (60) suggested a hierarchical algorithm to combine the speed of Gauss-Seidel and the reliability of Newton-Raphson, this procedure was incorporated in the present work. The jet temperature obtained from energy balance and instantaneous air/fuel ratio (obtained from the air entrained into the jet deducting the air burnt and the fuel evaporated deducting the fuel burnt) are input from the combustion submodel

discussed in section (2.2.1). The NO_x routine is called to execute when ignition starts.

2.2.2.2 Exhaust Smoke Modelling

The exhaust smoke emission model used in the present studies is discussed in detail by Sindano (4) and only a brief overview is outlined here. Tesner et al (61) suggested that soot formation occurs in three separate stages, viz; spontaneous formation of radical nuclei - which are the pre-cursors of soot formation, the formation of soot particles around the nuclei, and the destruction of the nuclei. Formation of radical nuclei was given by the following expression:

$$dn/dt = n_0 + (f-g) n - g_0 n N \quad (2.48)$$

where n_0 represents spontaneous rate for the formation of radical, defined by the following equation:

$$n_0 = A_0 C_{fu} \exp [- E/(RT)] \quad (2.49)$$

and the formation of the soot particles is :

$$dN/dt = (a - b N) n \quad (2.50)$$

The existing soot oxidation model (4) combines the expression postulated by Nagle and Strickland-Constable (62), which accounts for the soot oxidation due to O₂ and the one by Fenimore and Jones (63) to account for the soot oxidation due to OH radicals, i.e.:

$$dm_s/dt = - (60/p_s) (m_s/ds) [m_{O_2}'' + m_{OH}'] \quad (g/cm^2 s) \quad (2.51)$$

where m_{O_2}'' is the reaction rate on soot surfaces due to oxygen, calculated as:

$$m_{O_2}'' = 12 [(K_p P_{O_2}) / (1 + K_2 P_{O_2}) x + P_{O_2} (1 - x)] \quad (2.52)$$

and $m_{OH''}$ is the similar rate due to the OH radical and calculated as:

$$m_{OH''} = 1.63 \cdot 10^4 \alpha (P_{O_2})^{1/4} (P_{H_2O})^{1/2} / T^{1/2} \exp [- 37800/(RT)] \quad (2.53)$$

It has been demonstrated by Mehta (18) after Magnussen (64) that turbulence effects can be accounted by multiplying all the kinetic terms by the rate of transfer of mass between the bulk fluid and the fine structure of turbulence, thus equations (2.48), (2.50) and (2.51) become :

$$dn/dt = (n_o - g_o n N) \chi + (f - g) N \quad (2.54)$$

$$dN/dt = (a - b N) \chi n \quad (2.55)$$

$$dm_s/dt = - (60/\rho_s) (m_s/ds) (m_{O_2''} + m_{OH''}) \chi \quad (2.56)$$

respectively,

where χ is the rate of transfer of mass between the bulk fluid and the fine structure of turbulence

$$= r^* m$$

r^* is the mass fraction occupied by fine structures,

m is the fraction of fine structures reacting.

Magnussen (64) has related χ to the concentration of combustion products and fuel as:

$$\chi = r^* m = m_{pr}/(m_{pr} + 4.45 m_{fu}) \quad (2.57)$$

where m_{pr} and m_{fu} are the concentrations of combustion products and fuel vapour left in the jet respectively.

The formation and oxidation models discussed above have been incorporated in the combustion model and it was found that the predicted net smoke emission is unacceptably high (5.7 Bosch compared with 3.91 Bosch of the experimental data). The accumulated soot formation and oxidation are plotted in figure (2.6) (f) (the typical performance results predicted by the combustion model is given in figure (2.6) (a-e)). It can be seen from the figure that the possible reason for the overestimated smoke emission is the underestimated soot oxidation rate.

Magnussen (64) has developed an eddy-dissipation mechanism for soot oxidation which assumes that oxidation of soot and fuel are mutually related and are controlled by local mixing in eddies of the Kolmogorov scale. Thus,

$$dm_s/dt = (dm_f/dt) (m_s/m_{fu}) \quad (2.58)$$

where dm_f/dt is the fuel burning rate and m_s the concentration of soot.

Magnussen (64) also considered the nuclei combustion rate and related it to the fuel burning rate as :

$$dn_s/dt = (dm_f/dt) (n_s/m_{fu}) \quad (2.59)$$

where n_s is the concentrations of nuclei.

The accumulated soot formation and oxidation due to Magnussen (64) are again plotted in figure (2.6) (g) and it can be noted that Magnussen's eddy-dissipation mechanism predicts oxidation rates much higher than the purely kinetic scheme (Nagle et al (62) and Fenimore et al (63)). This confirms the findings by Ahmad et al (65).

The present smoke emission model adopts the soot formation mechanism proposed by Tesner et al (61) to describe soot formation. Although Magnussen (64) considered a turbulent eddy mixing model for combustion and events in the fine structure, the close relationship between the fuel combustion and soot combustion indicated in equation (2.58) is considered generally applicable, no matter how combustion rate is determined. Thus equations (2.58) and (2.59) form the basis of the soot oxidation in the present work. The concentrations of combustion products and fuel vapour left in the jet, the jet temperature and cylinder pressure obtained

from energy balance and instantaneous air/fuel ratio are input from the combustion submodel discussed in section (2.2.1). The smoke routine is called to execute when ignition starts.

2.2.3 THE EXHAUST AND INDUCTION PROCESSES

Figure (2.7) shows a schematic diagram of the physical processes of exhaust and induction. A fundamental model by Sherman and Blumberg (66) and used by Sindano (4) which simulates the changes in cylinder pressure, temperature and mass flow during these processes is adopted in the present study. Based on the open system First law energy balance on the control volume consisting of the cylinder working volume, the following equation can be written, in terms of the change in internal energy:

$$dU = dQ - dW + h_i dm_i + h_e dm_e \quad (2.60)$$

where dm_i and dm_e are flows through the intake and exhaust manifolds, respectively, and h_i and h_e the respective enthalpies. dQ is the heat transfer to the cylinder walls, and dW the work term. Sherman and Blumberg have shown that by combining with the perfect gas law, the following differential equation to calculate the change in cylinder pressure can be written:

$$dP_c = 1/V [- \nu P_c dV + (\gamma - 1) dQ + R_r (T_i dm_i + T_e dm_e)] \quad (2.61)$$

Considering mass conservations in the cylinder, Horlock and Woods (67) have derived the following equation for the change in charge temperature:

$$dT_c/T_c = dP_c/P_c + dV_c/V_c - (R_c T_c/P_c V_c) (m_i + m_e) \quad (2.62)$$

Mass flow rate through the inlet and exhaust valves is approximated by isentropic compressible flow equations as applicable to flow through nozzles, and modified by

a discharge coefficient (Quasi-steady flow conditions are assumed at each time step):

$$dm/dt = C_d A_v P_u F (2/(R T_u))^{1/2} \quad (2.63)$$

where:

$$F = \{ [\gamma/(\gamma-1)] [P/P_u]^{2/\gamma} [1 - (P/P_u)((\gamma-1)/\gamma)] \}^{1/2} \quad (\text{for subsonic flow})$$

$$= \{ [\gamma/(\gamma+1)] [2/(\gamma+1)]^{(2/(\gamma-1))} \}^{1/2} \quad (\text{for sonic flow - choked})$$

P_u = upstream pressure

P = downstream pressure

C_d = discharge coefficient

A_v = minimum flow area at the valve

T_u = upstream temperature

For the inlet/exhaust flow, the upstream pressure is the inlet manifold pressure/cylinder pressure and the downstream one is the cylinder pressure/exhaust manifold pressure. If the inlet manifold pressure is lower than the cylinder pressure or the cylinder pressure is lower than the exhaust manifold pressure, backflow will occur (from cylinder to inlet manifold or from exhaust manifold to cylinder). In this situation, a negative flow rate is obtained.

At steady state conditions, cycle convergence can be obtained by successive iterations between this model and the combustion model through the entire cycle using the computed cylinder charge mass as criteria. Cycle convergence occurs normally within two to five iterations, dependent on the initial estimation of the thermodynamic conditions of the starting point. Under transient conditions, the iteration is carried out only for the first cycle. After the first cycle is converged, the following cycles will always have the right starting points.

2.2.4 ENGINE FRICTION LOSSES

One of the first comprehensive analyses of engine friction was undertaken by Gish et al (68). Each cylinder of a four-cylinder gasoline engine was instrumented with balanced disc pressure transducers and an ingenious electronic technique was used for obtaining accurate pressure-volume measurements in each cylinder to determine the indicated power. The brake power during actual operation of the engine was simultaneously measured as well. Gish et al got the true friction as the difference between the indicated power of all cylinders and the measured brake power. The correlation they obtained relates the fmep (friction mean effective pressure) to peak pressure as:

$$\text{fmep} = 1.0 + 0.0125 p_{\max} \text{ (bar)} \quad (2.64)$$

Since equation (2.64) was obtained at only one speed (1600 rpm), it is not possible to evaluate the effect of the speed on friction. Bearing in mind this weakness, Chen and Flynn (69), Winterbone and Tennant (70) both investigated the friction losses and related their experimental data to the engine speed as well as the peak cylinder pressure for their own engines.

The correlations of Chen and Flynn (69) has been used widely by many investigators including Ledger and Walmsley (35), Watson and Marzouk (40) and Goyal (41). The work was conducted on a single cylinder engine capable of withstanding peak pressures of 200 bars at imep levels of up to 18 bar at 3200 rpm. The engine was constructed in two variants and was fitted with chain driven balancer shafts. One variant ER-2 had separately driven accessories. The correlations they got for each engine are:

$$\text{fmep} = 0.138 + 0.005 p_{\max} + 0.164 S_p \text{ (bar)} \quad \text{for ER-2} \quad (2.65)$$

and

$$\text{fmep} = 1.172 + 0.01 p_{\max} + 0.164 S_p \text{ (bar)} \quad \text{for ER-1} \quad (2.66)$$

where S_p = the mean piston speed (m/s)

Since there is a large difference in the coefficients between the two engines, Winterbone and Tennant (70) argue that this undermines confidence in extrapolating equations (2.65) and (2.66) to a vastly different multicylinder engine. So they approach the problem of the friction losses by retaining the Chen and Flynn structure but justify the coefficients empirically for their Leyland 520 engine and the equation they got was:

$$f_{mep} = 0.061 + 0.294 N_E/1000 + 0.016 p_{max} \text{ (bar)} \quad (2.67)$$

A recent work by Patton et al (71) was based on a combination of lubrication theory and empirical results, including predictions of rubbing losses from the crankshaft, reciprocating, and valvetrain components, auxiliary losses from engine accessories, and pumping losses from the intake and exhaust systems. The comprehensive formula postulated by them relates the total friction losses to basic engine design and operating parameters as :

$$\begin{aligned} f_{mep} = & 1.22 * 10^{-5} (D_b/(B^2 S n_c)) + 3.03 * 10^{-4} (N_E D_b^3 L_b n_b)/(B^2 S n_c) + \\ & 1.35 * 10^{-10} (D_b^2 N_E^2 n_b)/n_c + 2.94 * 10^2 S_p/B + 4.06 * 10^4 * \\ & (1 + 1000/N_E)/B^2 + 3.03 * 10^{-4} (N_E D_b^3 L_b n_b)/(B^2 S n_c) + \\ & 6.89 P_i/P_a [0.088 r_c + 0.182 r_c (1.33 - K S_p)] + C_{ff} (1 + 1000/N_E) * \\ & n_v/(S n_c) + C_{rf} (N_E n_v)/(S n_c) + C_{oh} (L_v^{1.5} N_E^{0.5} n_v)/(B S n_c) + \\ & C_{om} (1 + 1000/N_E) L_v n_v/(S n_c) + 6.23 + 5.22 * 10^{-3} N_E - \\ & 1.79 * 10^{-7} N_E^2 + P_a - P_i + 4.12 * 10^{-3} (P_i/P_a)^2 (S_p^2/(n_v^2 r_i^4)) + \\ & 0.178 (P_i S_p/P_a)^2 + 4.12 * 10^{-3} (P_i/P_a)^2 (S_p^2/(n_v^2 r_e^4)) \end{aligned} \quad (2.68)$$

where K is equal to 2.38×10^{-2} , and C_{ff} , C_{rf} , C_{oh} and C_{om} are constants for valvetrain mechanism terms.

Although the work of Gish et al gave an insight into the problem of true engine friction, they have not taken the effects of engine speed into account, while the coefficients in Chen and Flynn type correlations tend to vary from engine to engine. Furthermore, it is not possible to assess the effects of basic engine design parameters on levels of friction from equations (2.64)-(2.67). Therefore, in this study, equation (2.68) by Patton et al (71) was adopted to represent the total engine friction losses, allowing for the investigation of the effects of the basic engine design and operating parameters on the total friction losses.

2.2.5 GOVERNOR DYNAMICS

An engine governor is a speed sensitive device which automatically controls the speed of an engine by adjusting the fuel pump delivery. In performing its duty the governor fulfils two functions (72):

1. It serves as a speed indicator to measure the engine speed
2. it acts as a power mechanism to actuate the fuel control whenever a speed change occurs.

Thus the governor and the engine form a closed loop to which speed and load changes are applied. The two types of widely used governors are constant speed and variable speed. A constant speed governor is used on engines such as the ones that drive electrical generators to maintain the engine at a single speed for any load level, whilst the variable speed type is used mainly in vehicle engines to control the speed over a wide range of load demands.

2.2.5.1 Speed Sensing Device

The flyweight mechanism shown in figure (2.8) is a speed sensing device used in almost any mechanical governor. The basic principle is that a force is required to compel a mass to follow a circular path. This force is proportional to the square of the speed of rotation. The flyweights are normally located symmetrically around the

governor shaft, which is linked to the engine crankshaft. When the engine is running, the rotating flyweights produce a centrifugal force opposed and balanced by a spring through the governor sleeve. A rise in engine speed increases the centrifugal force and the flyweights move outwards, pushing the governor sleeve and causing an increase in the spring force, which balances the governor sleeve in a new position. The movement in the governor sleeve drives the governor control spool towards its minimum travel, which decreases the amount of fuel delivered per cycle.

2.2.5.2 Engine Speed Governing

The governor investigated is a variable speed type, shown in figure (2.9) (73). For analytical purpose, the spring-controlled governor is redrawn for convenience in figure(2.10). As described by Morrison and Crossland (74), the transfer function relates the change in the displacement of the spring loaded governor sleeve of mass M to the change in angular speed of the governor shaft ω_g . It could be assumed that when the arms holding the balls are vertical the compression of the spring, which has a stiffness K , is X_0 . It could be further assumed that there is some viscous damping acting at the governor sleeve, with a damping constant f . If due to a small change, ω in the speed the small displacement of the governor sleeve at some subsequent time is x the governor arms will be displaced by a small angle θ , shown in an exaggerated form in figure(2.10), where

$$\sin\theta = x/h \approx \theta \quad (2.69)$$

Considering moments about the bell crank pivot at O , ignoring the moment due to gravity, and assuming there are four balls, the following equation could be written down:

$$\begin{array}{cccccc}
 -K (X_0 + x) h L_1/L_2 & = & 4 m v^2 \theta'' & + & M h x'' & + & h f x' - 2 v m (\omega_g + \omega)^2 (r + v \theta) \\
 \downarrow & & \downarrow & & \downarrow & & \downarrow \\
 \text{Moment due to} & & \text{Inertia} & & \text{Inertia} & & \text{Damping} & & \text{Moment to provide} \\
 \text{spring force} & & \text{torque of} & & \text{torque of} & & \text{torque} & & \text{centripetal force} \\
 & & \text{the balls} & & \text{governor} & & & & \\
 & & & & \text{sleeve mass} & & & &
 \end{array}$$

(2.70)

substituting for θ from equation (2.69), this becomes:

$$L_1/L_2 K (X_0 + x) h = 4 m v^2/h x'' + M h x'' + h f x' - 4 m v (\omega_g + \omega)^2 (r + v x/h)$$

(2.71)

Neglecting second-order small terms, this reduced to:

$$(4 m v^2/h + Mh) x'' + h f x' + K L_1/L_2 (x + X_0) - 4 m v^2 \omega_g^2 x /h = 4 m v \omega_g^2 r$$

(2.72)

Equation (2.72) is rearranged as:

$$x'' + C_1 x' + C_2 (x + X_0) - C_3 x \omega_g^2 = C_4 \omega_g^2$$

(2.73)

where X_0 is the governor control lever position

x is the governor sleeve position

C_1 - C_4 are constants for the system and related to the governor system parameters as:

$$C_1 = h^2 f / (4 m v^2 + M h^2)$$

(2.74)

$$C_2 = K L_1/L_2 h / (4 m v^2 + M h^2) \quad (2.75)$$

$$C_3 = 4 m v^2 / (4 m v^2 + M h^2) \quad (2.76)$$

$$C_4 = 4 m v r h / (4 m v^2 + M h^2) \quad (2.77)$$

(1). For a constant speed governor, $X_0=0$

(2). In the case of variable speed governor:

(i). For steady state governing,

$$4 m v r \omega_g^2 = K X_0 h L_1/L_2 \quad (2.78)$$

Equation (2.78) indicates how the engine speed can be adjusted by simply varying the initial position of the governor sleeve X_0 .

(ii). Governing under transient conditions,

a. In the case of a sudden movement of the governor control lever, the required governor sleeve position X_r is quickly achieved. From equation (2.73), the corresponding change in the governor sleeve position x can be predicted.

b. Another example is a sudden application of load. In this case, X_0 is constant, the decrease in engine speed caused by the applied load changes the governor sleeve position x , which can again be predicted by equation (2.73).

By considering the linear speed of the governor sleeve position ($u = \dot{x}$) as an intermediary variable, the second order differential equation (2.73) for describing the governor dynamics is resolved into two first order equations as:

$$u' + C_1 u + C_2 (x + X_0) - C_3 x \omega_g^2 = C_4 \omega_g^2 \quad (2.79)$$

$$u = x' \quad (2.80)$$

And equations (2.79) and (2.80) can be solved by the Runge-kutta fourth order algorithm to get the governor sleeve position x . From figure (2.8) it can be seen that the governor control spool position x_c is related to the governor sleeve position x as:

$$x_c = x L_3 / L_2 \quad (2.81)$$

The shaft of the governor sleeve is connected with the camshaft through gears as:

$$\omega_g = \omega_c n_c / n_g \quad (2.82)$$

In the four stroke engine, the camshaft speed is half of the crank shaft speed, i.e.

$$\omega_c = \omega_e / 2 = \pi N_E \quad (2.83)$$

Details of the measurements of the governor system parameters can be found in Chapter 6.

2.2.6 FUEL INJECTION SYSTEM

The fuel injection system is the heart of an engine and consists of the fuel pump, delivery valve, fuel pipes and injectors. The fuel injection system is required to deliver the precise quantity of fuel according to engine load and speed, and to inject the fuel at the correct instant. For the present study, a detailed analysis of the fuel pump and injection system, taking into account wave action in the fuel pipes, such as the ones by Becchi (75) and Wylie et al (76), was considered to be inappropriate. The computational time for the solution of the detailed analysis is excessively longer. Furthermore, a highly accurate prediction of the injection rate diagram preceding each firing is not essential, since the combustion is accounted for by the apparent fuel burning rate (section 2.2.1.3).

2.2.6.1 Fuel Delivery process

The fuel pump in the present investigation is Bosch VE type distributor pump, shown schematically in figure (2.11)(73). It can be noted from the figure that the distributor pump contains the following functional groups:

- (1) high speed pump with distributor
- (2) mechanical governor
- (3) Hydraulic timing mechanism
- (4) vane-type supply pump
- (5) shutoff mechanism

A fuel injection pump's delivery of fuel is a dynamic process, made up of several stroke phases. The pressure necessary for injection into the engine cylinders is generated by the plunger. The distributor-plunger motion phases shown in figure (2.12) illustrate the metering of fuel to an engine cylinder. It can be seen from the figure that the quantity of fuel delivered to an engine cylinder is dependent on both fuel pump control spool position (rack position) and pump-cavity pressure. Since the pump-cavity pressure is set to be proportional to engine speed through the pressure control valve, the fuel quantity of fuel delivered is determined by the engine speed as well as fuel pump control spool position.

To evaluate fuel flow under transient conditions, Ledger et al (77) first attempted an experimental relationship between the quantity of fuel injected and pump rack position (based on steady state fuelling rate measurement at different pump rack positions), regardless the effects of engine speed. Marzouk (78) also based his transient fuelling rate on the steady state fuel pump characteristics but he included the effects of engine speed variation. Keeping the pump rack position constant, Marzouk got the relationship between the fuelling rate and the engine speed by varying load levels, then keeping the engine speed constant, he varied the pump rack position to measure the steady state fuelling rate. The present work followed Marzouk's method to characterise the fuel pump.

The fuel pump steady state characteristics was obtained from the overall fuel consumption measurements which were carried out by using two Carrier venturi meters, one in in-flow and the other in the return flow from the injector spill. The quantity of fuel per stroke per cylinder were measured at different engine speed (1000-3000 rpm) and governor control spool position (25-100 %) (section 6.2). The experimental data were then fitted using the Taylor Series, as shown in figure (2.13), resulting in the fuel pump characteristics in the following form:

$$\begin{aligned}
 p_{cs} = 25\% \quad M_f &= a_1 + a_2*N + a_3*N^2 + a_4*N^3 \\
 p_{cs} = 50\% \quad M_f &= b_1 + b_2*N + b_3*N^2 + b_4*N^3 \\
 p_{cs} = 75\% \quad M_f &= c_1 + c_2*N + c_3*N^2 + c_4*N^3 \\
 p_{cs} = 100\% \quad M_f &= d_1 + d_2*N + d_3*N^2 + d_4*N^3 + d_5*N^4
 \end{aligned} \tag{2.84}$$

where p_{cs} = governor control spool position

M_f = fuelling rate (mm³/cylinder/cycle)

a_1 - d_5 = constants

For any control spool position other than 25%, 50%, 75% and 100%, the fuelling rate is obtained by interpolated between the values of two ends (e.g. for $p_{cs}=31\%$, M_f is interpolated from the values for 25% and 50% respectively).

2.2.6.2 Fuel Injection Process

At steady state condition, following Sindano (4), the simulation of fuel injection may be achieved in either of the following ways:

- 1) Providing an injection pressure diagram obtained experimentally on an engine with identical dimensions, and under identical load conditions.
- 2) Making use of the volume of fuel injected per stroke to construct a trapezoidal injection pressure diagram with a mean injection pressure calculated from the equation by Dent (57):

$$\Delta P_{inj} = 2 \rho l / (\pi C_d) [6 N_E Q / (n d_o^2 \theta_{ip})]^2 \quad (2.85)$$

where C_d = Injector nozzle discharge coefficient

θ_{ip} = Injection duration in degrees crank angle

Q = Volume of fuel injected ($\text{mm}^3/\text{stroke}$)

Under transient condition, the experimental injection pressure diagram would not be available, therefore the second method is used. From the calculated injection pressure diagram, the model calculates the velocity of the jet at the nozzle. In the existing model (3), the injection parameters such as the start of injection and injection duration for each operating condition are part of the input and read from a data file. This would not be realistic under transient running.

The distributor pump (figure (2.11)) in the present study employs a hydraulic timing-advance device at its underside and at right angles to the pump's longitudinal axis. The operation of the timing-advance device is schematically shown in figure (2.14). The timing piston is held in its initial position (when engine is at rest) by a preloaded spring. During operation, the piston side opposite to the spring is under the same pressure as the fuel pressure in the pump cavity, which increases with increasing engine speed and quantity of fuel. When engine speed reaches approximately 300 rpm, fuel pressure reaches a value sufficient to overcome the spring preload and move the piston, as shown in figure (2.14), to the left. This axial piston motion is transmitted via the sliding block and pin to the roller ring, which rotates in bearings. This alters the arrangement of cam plate and roller ring so that the rotating cam plate is lifted at an earlier point in time by the rollers. The rollers with their ring are thus turned by a specific angle relative to cam plate and distributor plunger. So it is clear that the start of injection is closely related to the quantity of fuel injected and engine speed. Two expressions have been correlated to the amount of fuel and engine speed based on the experimental data in Table 2.1, for calculating the start of injection and injection duration respectively which have the following forms:

$$(S_{io} - S_{io_{ref}}) / S_{io_{ref}} = a_1 (Q/Q_{ref})^{b_1} [(N_E - N_{E_{ref}})/N_{E_{ref}}]^{c_1} \quad (2.86)$$

$$(D_{inj} - D_{inj_{ref}}) / D_{inj_{ref}} = a_2 (Q/Q_{ref})^{b_2} [(N_E - N_{E_{ref}})/N_{E_{ref}}]^{c_2} \quad (2.87)$$

where Q is quantity of fuel injected

N_E is engine speed

a_1 - c_1 , a_2 - c_2 are empirical constants

ref denotes to the reference values.

2.2.7 LOAD DYNAMICS

The vehicle drive train is shown in figure (2.15). When the clutch is engaged, $N_C = N_E$ and the vehicle speed V_m is related to the engine speed N_E as:

$$V_m = 2 \pi * r * N_E / (n_b * n_g) \quad (2.88)$$

where n_b and n_g are the engaged ratio and the back axle ratio respectively. Following Winterbone et al (79), the engine power is equal to the tractive power and acceleration power, i.e.

$$2 \pi L_E N_E = 2 \pi L_T N_C + 2 \pi L_A N_C \quad (2.89)$$

The tractive torque L_T is obtained from the tractive resistance, R_t of the vehicle. According to Taborek (80), the tractive resistance consists of mainly air resistance, rolling resistance and road grade resistance, i.e.

$$R_t = R_a + R_r + R_g \quad (2.90)$$

where

<1> R_a is the air resistance given by

$$R_a = k A V^2 \quad (2.91)$$

with k a constant.

<2> R_r is the rolling resistance. This includes the tyre rolling resistance and deformation, the friction in the wheel bearing and any other friction elements which are a function of the motion of the vehicle.

$$R_r = W (a + bV_m) \quad (2.92)$$

where a , b are constants which have to be determined for each case as function of the tyre and road condition.

<3> R_g is the road grade resistance.

$$R_g = W_g \sin \theta \quad (2.93)$$

Taking all the resistances together, the total tractive resistance is:

$$R_t = k AV^2 + W_g \sin \theta + W (a+bV_m) \quad (2.94)$$

The values of a , b and k used by Winterbone et al (79) are adopted in the present study since the their truck is very similar to the vehicle under present investigation (FORD TRANSIT minibus).

The tractive torque L_T is then determined:

$$L_T = R_t \cdot r_E \quad (2.95)$$

where r_E is the transmission equivalent radius and is related to the engaged ratio n_g , the back axle ratio n_b , and the driving wheel radius r , given by

$$r_E = [1/n_b] \cdot [1/n_g] \cdot r \quad (2.96)$$

The acceleration torque L_A is related to the drive train acceleration α and the system inertia I_S by the expression:

$$L_A = I_S \alpha = I_S \cdot d\omega_e/dt \quad (2.97)$$

therefore

$$dN_C/dt = 1/2 \pi * L_A/I_S \quad (2.98)$$

The system inertia I_S is referred to the crankshaft radius r , by the expression

$$I_S = W r^2 [1/n_b]^2 * [1/n_g]^2 + I_E \quad (2.99)$$

where I_E is the engine inertia referred to the crankshaft.

The engine torque is evaluated from the engine net power output as follow:

$$L_E = 7120.834/1.34/N_E \text{ Power} \quad (2.100)$$

where Power is the engine net power output

$$= (\text{imep} - \text{fmep}) * V_s * N_E / (120 * 10^{-2}) * n_c$$

imep = indicated mean effective pressure

fmep = friction mean effective pressure

V_s = cylinder swept volume

n_c = number of cylinders

From equation (2.89), the acceleration torque, thus the acceleration of the engine can be calculated. The vehicle acceleration f can be obtained from the following expression:

$$f = 2 \pi * r * dN_C/dt / (n_b * n_g) \quad (2.101)$$

The vehicle modelled in the present work is a FORD TRANSIT minibus, details of which are listed in Appendix A.

Table 2.1 The Experimental Data Provided By Ford Motor Company

	1000 RPM			2000 RPM			3000 RPM			4000 RPM		
	A	B	C	D	E	F	G	H	I	J	K	L
START OF INJECTION (BTDC)	3.5	7.3	0.9	3.7	6.1	0.5	7.5	10.5	5	11.3	13.1	8.5
INJECTION DURATION (°CA)	8.1	8.6	8.3	13.9	14	14	20	20.3	19	24.5	24.6	24.4
IGNITION DELAY (°CA)	6.1	6	6.6	6.8	6.3	7.7	8.7	9	9.8	10.2	10.7	10.6
POWER (KW)	11.9	11.9	11.5	29.2	29.4	27.9	44.3	44.5	43.2	50.8	51.1	49.2
FUEL DELIVERY (MM ³ /STR)	28.1	28.6	28.2	34.1	34	33.5	35.5	35.5	35.2	34.1	33.9	33.7
EXHAUST TEMPERATURE (°C)	374	355	388	519	504	539	600.	580	602	631	622	640

Figure 2.1 Schematic Diagram of the Engine - Governor Control Loop

Figure2.2 Conceptual Structure of the Cylinder Charge during Injection

Figure 2.3 Comparison of Ignition Delay between Experimental and Predicted from Hardenberg and Hase's Correlation

Figure 2.4 Comparison of Ignition Delay between Experimental and Predicted from Modified Hardenberg and Hase's Correlation

Figure 2.5 Comparison of Ignition Delay between Hardenberg and Hase's Correlation and the Modified Expression Based on the AVL Experimental Data

Figure 2.6 Typical Performance and Exhaust Smoke Emission Results Predicted by the Model.

Figure 2.7 Schematic of Heat and Mass Flow Processes in Induction/Exhaust Model
(66)

Figure 2.8 Schematic of the Flyweight Mechanism

Figure 2.9 The Variable Speed Governor in a Bosch VE Type of Pump (73)

Figure 2.10 Schematic of a Mechanical Governor

4 *Functional groups of the distributor pump.*

1 Vane-type supply pump

Supplies fuel from tank to injection-pump cavity.

2 High-pressure pump with distributor
Produces injection pressure, moves and distributes fuel to cylinders.

3 Mechanical governor

Controls engine speed, varies fuel delivery over control range.

4 Electromagnetic shutoff valve

Interrupts fuel delivery to stop engine.

5 Injection-timing unit

Adjusts beginning of injection according to engine speed.

Figure 2.11 Schematic of Bosch VE Type Fuel Injection Pump (73)

15 Strokes and delivery phases.

Entry of fuel.

With the plunger at bottom dead center, fuel flows through inlet passage (2) and metering slit (3) into the high-pressure chamber (4).

Fuel delivery.

Plunger motion toward top dead center closes the inlet passage and pressurizes fuel in the high-pressure chamber (5). Further rotary motion of the plunger causes a distributor slit (6) to open the outlet bore (7) to a cylinder.

End of delivery.

Fuel delivery ceases when the control spool (8) opens the cutoff bore (9).

Entry of fuel.

As the plunger returns to bottom dead center, its rotary-reciprocating motion closes the cutoff bore and the high-pressure chamber fills again.

Figure 2.12 Schematic of the Fuel Delivery Process (73)

Throttle opening 25%

Throttle opening 50%

Throttle opening 75%

Throttle opening 100%

Figure 2.13 The Experimental and Derived Polynomial Pump Characteristics

Figure 2.14 Schematic of the Operation of Timing-Advance Device (73)

Figure 2.15 Schematic of the Drive Train

CHAPTER 3

TRANSPUTER AND OCCAM

3.1 INTRODUCTION

As Jones et al have pointed out (45), the problem with the conventional computer model to simulate an engine under transient conditions is that it consumes enormous computer processing time to obtain an acceptable solution. Because of the heavy computer requirement, detailed diesel engine simulations have been restricted to application as a research tool and design office aid.

Attempts have been made to increase the speed of diesel engine simulation, usually by making many simplifying assumptions. Wallace and Winkler (81) for example, achieved a fast-running model at the cost of accuracy by adopting modified air-standard cycles. The question is whether the mathematical model can be kept in reasonable details but run faster by other computational methods. Certainly a supercomputer is a solution, but the high cost make it not very practical.

All current micro-computers follow the basic Von Neumann architecture, in which a central processor is connected via a data bus to a memory unit. Programmed instructions are passed to the processor via the bus, executed and the results transferred via the bus to memory - the execution of events occurs sequentially.

The runtime of a computer program could be reduced by performing different, but relatively independent tasks within the program at the same time, by using a number of processors - concurrent or parallel processing. Over the last decade, there have been dramatic developments in the parallel processing field (Fathi and

Krieger (82), Schendel (83), Evans (84) and Wait (85)). One of these is the advent of the transputer which has made possible inexpensive concurrent processing (Walker (86)).

3.2 TRANSPUTER SYSTEM

The transputer is a high performance single chip computer - figure (3.1), with link interfaces which enable a number of transputers to be readily assembled into networks of general topology - figure (3.2) shows typical examples. Each link consists of a pair of wires allowing serial communication between transputers - the output of one transputer being connected to the input of another.

3.2.1 THE BASIS OF THE TRANSPUTER SYSTEM

The treatment of parallel processing in transputer systems is based on a model of computation call CSP - Communicating Sequential Processes proposed by Hoare of Oxford University in 1978 (87). In this model, a computing system is a collection of concurrent active sequential processes which can only communicate with each other over channels. Only two processes may be connected by a channel, which can only carry messages in one direction: if communication in both directions between two processes is required, two channels must be used. Channel communication must be synchronised. (i.e. a process wanting to send a message over a channel is always forced to wait until the receiving process reads the message).

A process in this model, is just a "black box" connected to the outside world only by its channels (figure (3.1)). A process could be a bit of hardware or a software module; in particular it may also be another complex system, itself consisting of a number of communicating processes.

The transputer has been designed to be a hardware realisation of the CSP model. Each transputer processor has four serial links, to connect it with other transputers.

Each link has two hardware channels, one in each direction. The hardware channels behave exactly like the abstract channels discussed above; they provide synchronised, unidirectional communication.

Arbitrary networks of transputers can be constructed simply by connecting their links together with ordinary wires, the only limitation being that each processor cannot be directly connected to more than four others.

The transputer is based on the VLSI (very large scale integration) technology and can be programmed efficiently in most high level languages like C, FORTRAN and PASCAL. A number of transputers can construct a MIMD (multiple instruction and multiple data) computer.

3.2.2 THE TYPES OF TRANSPUTER HARDWARE

The typical transputer hardware modules are T414 and T800 transputers. Detailed information on these two processors can be found in reference (88). The main differences between them are that T800 has a floating point operation unit and has 4 kbyte on-chip RAM instead of 2 kbyte RAM on T414.

For intensive floating point arithmetic the Inmos T800-20 chip is used. This is a high performance 32 bit Reduced Instruction Set processor operating at 20MHz with a 64 bit floating point unit capable of a sustained output of 1.5 million floating point operations per second (Mflops). This chip is one of the fastest 32 bit microprocessors on the market (89).

3.3 OCCAM PROGRAMMING LANGUAGE

OCCAM is the first programming language to be devised explicitly for parallelism, which provides a formal description that specifies the transputer chip, and is used for programming it. OCCAM is designed to handle the mixed sequential and

concurrent nature of real world processes and is able to provide automatic communication and synchronisation between concurrent processes.

Underlying formal semantics derived from CSP model, OCCAM has a close relationship with the transputer and is designed to fully exploit its architecture. OCCAM is executed more or less directly on the transputer and is the "assembly language of the transputer".

3.3.1 THE NOVEL FEATURES OF THE OCCAM LANGUAGE

OCCAM is the first language to be based upon the concept of parallel, in addition to sequential execution, and to provide automatic communication and synchronisation between concurrent processes. As well as defining variables for storing values in the same way as the sequential languages (FORTRAN, C and PASCAL), OCCAM uses channels for communicating values. Channels look in many ways like variables, except that rather than assigning a value to them for storage, we output to them or input from them. The value output by one process is the input to another process, the channel behaving like a pipe joining the two processes. A channel here, just like a hardware channel mentioned in section 3.2.1, can only join two processes. A channel provides a one-way synchronised communication, so two channels would be needed for a two way communication.

A channel can pass values either between two processes running on the same computer, or between two processes running on different computers. In the first case the channel would in fact be just a location in memory, rather like a variable. In the second case the channel could represent a real hardware link, such as a transputer link or other serial communication line. Both cases are represented identically in an OCCAM program (Pountain (90)).

Communication over self-synchronising channels is a novel and powerful feature of OCCAM.

3.3.2 OCCAM USED AS HARNESS

A separately compiled program written in one of the high level languages such as C, FORTRAN and PASCAL can execute as a single process in an environment of OCCAM channels. This allows the program to run on a single transputer, or several such programs, possibly written in different languages, to run on several transputers communicating via INMOS links, or several programs to run on a single transputer within a harness of OCCAM channels.

Two levels of access to the OCCAM channels are provided:

- (1) Calls to the standard language input/output system are translated into a simple protocol on the channels. The user provides an OCCAM program to interpret this protocol according to the system requirements of the application. For host execution, a standard interface to the transputer development system is provided.
- (2) A lower level of access provides language extension or a library for direct access to the channels, providing an OCCAM-like input/output system.

In the present study, the modules written in FORTRAN to simulate the processes occurring in the diesel engine are implemented concurrently by using OCCAM as a harness. The OCCAM harnesses are developed within the OCCAM toolset (91) which provides the two levels of access to the OCCAM channels mentioned above. The details of the development of the OCCAM harnesses and the implementation of the FORTRAN programs in parallel are discussed in Chapter 4.

3.3.3 THE PRIMITIVE PROCESSES OF THE OCCAM LANGUAGE

Following reference (92), OCCAM programs are built from combination of three kinds of primitive process:

- (1) Assignment process, which changes the value of a variable, just as it would in most conventional languages.

(2) Input process, which inputs a value from a channel into a variable.

(3) Output process, which outputs a value to a channel.

Conventional sequential programs can be expressed in OCCAM using variables and assignments, combined to form sequential constructs, while concurrent programmes can be expressed in OCCAM using channels, inputs and outputs, which are combined to form parallel and alternative constructs.

3.4 PROGRAMMING OF TRANSPUTERS USING OCCAM

OCCAM can be used to develop parallel or concurrent programs on an individual transputer or a network of transputers. When used with an individual transputer, the transputer shares its time between the concurrent processes and channel communication is implemented by moving data within memory. When used to program a network of transputers, each transputer executes the process allocated to it; channel communication between OCCAM processes on different transputers is implemented directly by the transputer links. Hence, the same OCCAM program can be implemented on a single transputer or a network of transputers. Therefore, development of computer code may be undertaken on a single transputer, but evaluation carried out on a network of transputers. Part of this evaluation will be:

- * Confirmation of load balancing estimates made during development - the ideal is a uniform level of activity across the network throughout the computational cycle.

- * Optimisation of the number of transputers for a given task. If the number of processors is too large the computational task per processor is small, so the overall computational time is controlled by the time taken for higher inter-transputer communication.

* Validation of results against known solutions.

In programming a network of transputers using OCCAM, either directly or as a harness, Care should be taken to the following problems:

(1) Load balancing between transputers. The execution of a concurrent program terminates when the last task is performed. Unevenly distributing tasks can cause some transputers to be idle.

(2) Communication/calculation balance within a transputer.

(3) Deadlock is a common problem associated with concurrent programming and it can be caused by a process terminating incorrectly without informing connected processes. When mapping the program onto a network of transputers, care must be taken to make sure that all the communication has a right sequence of sending and receiving, and all the subprograms are terminated properly to avoid deadlock.

Figure 3.1 (a) Schematic of the Transputer Chip

Figure 3.1 (b) Architecture of the Transputer Chip

Figure 3.2 (a) Square Array of Nine Transputers

Figure 3.2 (b) Array of Four Transputers to Produce a Larger Transputer

CHAPTER 4

MODELLING OF DIESEL ENGINE USING A TRANSPUTER BASED CONCURRENT COMPUTER

4.1 INTRODUCTION

In this chapter, the structure of the concurrent computer and the software distribution are provided with a detailed description of the implementation of the model in parallel.

4.2 THE STRUCTURE OF THE CONCURRENT COMPUTER

The concurrent computer used in this study consists of an IBM PC-XT host and a Quintek fast4 board (93) which consists of four INMOS T800-20 transputers with a megabyte of local RAM each, and an interface between the INMOS architecture and the host computer's bus. The host interface provides the access to the transputer environment needed by software running on the IBM PC host, using an INMOS link for data transfer.

The computer hardware lay out is shown schematically in figure (4.1). The logical structure of this hardware layout is vertical organisation, i.e. a single master processor (IBM PC-XT host) with multi-slave processors (transputers T0-T3). The links are pre-wired into a square with link 2 of each transputer connected to link 3 of the next transputer. A pre-wired connector is used to attach the host PC data bus

to link 0 of transputer T0. Furthermore, the spare links 1 are also connected in a cross: T0 to T2 and T1 to T3. In transputer terminology, T0 is the root transputer and the rest are targets. The target transputers can communicate with the host PC only through the root transputer. The links are high speed, bi-directional serial communication lines and have data transfer speed of 10 Mbits/second.

4.3 THE MODEL DISTRIBUTION

The flow chart of component processes in the model is shown in figure (4.2), with the names of the major routines listed below the corresponding processes. Each subroutine has been developed and tested individually, to assess their time requirement. It was found that the combustion, induction-exhaust, entrainment and emission submodels account for the major part of the computation time. So these submodels are allocated to the three target transputers, whilst the fuel delivery, friction losses, governor and load dynamics are implemented in the root transputer, which also handles all communication with the target transputers and the host. The general program development, all the supervisory functions and graphics presentations are conducted in the host processor. This arrangement, illustrated in figure (4.3), makes the tasks implemented on different processors relatively balanced.

It is quite clear that some of the processes occurring inside the cylinder are time dependent. Taking a four stroke diesel engine as an example, when the inlet valve is opened, the air is induced into the cylinder. After the piston passes the bottom dead centre, the inlet valve is closed and the air is compressed. The fuel is then injected and the induced air is entrained into the fuel spray and the mixture is formed. Ignition occurs near the top dead centre and combustion carries on for a certain period, then the exhaust valve is opened near the end of combustion. While the piston moves up and gas is expelled, the pollutants are exhausted from the cylinder. Although the tasks simulating the time dependent processes mentioned above are allocated to different transputers, they are not implemented at exactly the

same time. To retain the sequence and benefit from concurrency, pipeline implementation is carried out. For example, the combustion and emission submodels are distributed into target transputer T1 and T3 respectively. However, at each time step, the emission calculation cannot continue until such data as cylinder pressure and flame temperature are determined in the combustion submodel. For pipeline implementation, communications between the combustion and emission submodels running on transputer T1 and T3 respectively occur at each time step, so that after the combustion calculation is carried out at the given time step, transputer T1 passes information necessary (e.g. cylinder pressure, flame temperature, equivalence ratio) to the emission submodel on T3 transputer and continues to conduct combustion calculation at the next time step, while the emission submodel receives the information and run at the previous time step. Although this results in high communication levels, pipelining is still much faster than the conventional sequential loop. Figure (4.4) shows the comparison of the pipelining and the conventional sequential loop.

4.4 CONCURRENT IMPLEMENTATION OF THE MODEL

The FORTRAN programs are compiled with the 3L FORTRAN compiler (94) and linked with the FORTRAN run time library and OCCAM harnesses. A configuration program is developed in OCCAM to allocate the software channels used by the OCCAM harnesses to the hardware links and to distribute the resulting OCCAM modules (FORTRAN program and OCCAM harness) to different transputers. The file server then loads the OCCAM modules and starts the execution of the modules concurrently on a network of transputers.

4.4.1 THE DEVELOPMENT OF THE OCCAM HARNESSES AND THE CONFIGURATION PROGRAM

Both the OCCAM harnesses and the configuration program were developed using the INMOS OCCAM toolset (91) which is designed to enable users to create

OCCAM harnesses for non-OCCAM programs. The OCCAM toolset consists of :

- * an OCCAM compiler capable of compiling OCCAM programs for T414 and T800 transputers.
- * a linker.
- * a configurer for configuring the software channels used by the OCCAM harnesses to the hardware links and the resulting OCCAM modules to different transputers.
- * a file server/loader for loading and serving transputer boards from the host PC.
- * OCCAM libraries for file server access, real and extended arithmetic etc.

The OCCAM compiler recognises a directive, **#IMPORT**, which specifies a non-OCCAM program. The **#IMPORT** directive looks like this:

#IMPORT "programname (parameters) = wspace"

where:

programname is the name of the non-OCCAM program. This name must be unique within an OCCAM harness.

parameters is the parameter list for the call and wspace is the number of workspace slots required for the call (currently always 1).

In the parameter list, the memory required for the non-OCCAM program should be specified with care. Allocating too much memory to the non-OCCAM program leads to not enough space left for the utilities such as compiler and configurer. In the present study, 200 kbytes of memory is allocated to each FORTRAN program.

Channels are passed into non-OCCAM programs via two integer vectors of pointers, one for the input channels, the other for the output channels. The pointers are set up using OCCAM predefined procedures **LOAD.INPUT.CHANNEL** and **LOAD.OUTPUT.CHANNEL**. The libraries for the non-OCCAM language being used will provide access to these channels with input/output primitives equivalent to OCCAM's input (?) and output (!).

To enable the OCCAM compiler to access the extended arithmetic library, the following DOS command

```
set extlibs = c:\toolset\extlibs.bin
```

should be added to **autoexec.bat** so that the name **EXTLIBS** is set every time the IBM host is switched on.

A configuration program is used to allocate different OCCAM modules to different transputers. The hardware channels used for inter-processor communications are also explicitly defined in this program. The software channels declared in the OCCAM harnesses are also assigned to the hardware channels by using the **PLACE** statements. The configurer is used to configure this program so that the file server can load it to distribute the OCCAM modules to a network of transputers.

The file server provides two functions. The first is to load a program to a transputer board. The second is to provide a program running on a transputer board with access to the host file system, the host's screen and keyboard, as well as other host functions. The file server runs on the host computer in parallel with the programs running on the transputer board. Communication with the file server is via a pair of channels, one going to and the other coming from the file server. These channels are connected to link 0 of the root transputer. Communications with the file server from any target transputer has to be routed via the root transputer. Since communication with the file server is through a pair of OCCAM channels, only one OCCAM process or only one non-OCCAM program can communicate with the

server at any one time. If more than one program need to communicate with the file server in parallel, then a multiplexer process has to be used, i.e. they communicate sequentially.

A non-OCCAM program should have two input channels and two output channels reserved for use by the run time libraries. These channels must be mapped onto single input and output channels to be sent to the file server. A special multiplexer process is required for this function, and this process, called **io.handler**, is supplied by the OCCAM board support library. This process must be called in parallel with the non-OCCAM program called from the OCCAM harness.

The actual OCCAM harnesses and configuration program are listed in Appendix B.

4.4.2 THE 3L FORTRAN COMPILER AND THE RUN TIME LIBRARY

The FORTRAN programs are compiled with the 3L FORTRAN compiler (94). The run time library, supported by the compiler, for the transputer contains a number of extra subroutines which are not part of standard FORTRAN 77. These routines are described as follow:

(1) Timer routine

The run time library contains a subroutine, **ICLOCK**, which sets its (INTEGER) argument to the elapsed time in seconds, according to the host system clock. It can be used to calculates the running time of a particular module. In fact, the major routines shown in figure (4.2) to simulate the component processes in the engine has been tested individually with this subroutine and it was found that the combustion, induction-exhaust, entrainment and emission submodels accounted for the major part of the computation time.

(2) Channel communication routines

The subroutines below are provided by the run time library to allow FORTRAN programs to send and receive messages across OCCAM channels.

SUBROUTINE CHANINMESSAGE (ICHANNEL, BUFFER, NBYTES)

SUBROUTINE CHANOUTMESSAGE (ICHANNEL, BUFFER, NBYTES)

These are the basic procedures supplied to perform input/output to OCCAM channels. **ICHANNEL** is an index into one of the arrays of channels provided as parameters to the FORTRAN program: i.e. these procedures perform the primitive send and receive message operations on the **ICHANNEL**th input or output channel as appropriate. **BUFFER** can be any FORTRAN object, and **NBYTE** is an integer object or expression determining the size of the message.

These subroutines enable FORTRAN programs to communicate with the OCCAM harnesses executing concurrently with them.

(3) Standalone run time library

The standalone run time library contains only essential initialisation code and the channel input/output functions. A program linked with the standalone library cannot use standard input/output function like **READ**, etc.

If the only run time library functions a FORTRAN program needs are the channel input/output functions, the standalone run time library can be used to reduce the size of the resulting executable program.

4.4.3 FORTRAN PROGRAMS COMMUNICATING WITH EACH OTHER

Taking two FORTRAN programs as an example, the procedures are as follow:

(1) The first FORTRAN program running on the target transputer T1 calls the subroutine **CHANOUTMESSAGE** in the run time library to output a message to its OCCAM harness through a channel predefined in the run time library.

(2) The OCCAM harness on T1 receives the message from the predefined channel and sends it to the root transputer T0 through the link between T1 and T0.

(3) The OCCAM harness on T0 receives the message from the link and outputs it to the input channel for the second FORTRAN program.

(4) The second FORTRAN program on T0 then calls the subroutine **CHANINMESSAGE** to get the message from the input channel predefined in the run time library.

It can be noted that $2n+1$ programs are required to run n FORTRAN programs in parallel within the OCCAM toolset, including n FORTRAN programs, n OCCAM harnesses and one configuration program to allocate the software channels used by the OCCAM harnesses to the hardware links (such as the one between T0 and T1 in the above example) and to load the appropriate codes to different transputers.

4.4.4 THE PRE-PROCESSING AND EXECUTION

The pre-processing can be undertaken only on the root transputer and necessitates :

(1) Compiling the FORTRAN programs and OCCAM harnesses. The FORTRAN programs are compiled with the options /T8A and /S. The first option makes the compiler generate code to work round a floating-point firmware bug in Rev A of the T800-20 processor which affects integer to real conversions. It should only be used with the t8f or f77 commands. The second option makes the compiler allocate scalar variables in static storage, so that they keep their values between calls to a subroutine. By default this is not the case, and each time a subroutine is called any scalar variables which are not explicitly initialised by assignments or DATA statements have undefined values (94).

The OCCAM compiler views OCCAM source as belonging to one of three types:

- * a program main body
- * a separate compilation procedure
- * a library

Therefore, the OCCAM harnesses need to be compiled with the option /m. This option indicates that a main body or configuration level separate compilation procedure is being compiled (91). The option /T8 is also used so that the compiled code will work on a T800 processor.

(2) The compiled FORTRAN programs are linked with the OCCAM harnesses and FORTRAN run-time library by the linker so that each FORTRAN program is treated as one OCCAM process. Two options need to be used. The first, /m informs the linker that OCCAM and non-OCCAM programs are to be linked together. The second option /c should be used when linking code for a network of transputers.

The FORTRAN program that is to be run on the root transputer is linked with the normal run-time library so that it can use the **READ** and **WRITE** statements to conduct the standard input and output. The FORTRAN programs to be run on the target transputers, are linked with the standalone run-time library with which the input and output are carried out only through predefined channels.

(3) The configuration program is configured so that the software channels used by the OCCAM harnesses for data transfer are allocated to the transputer hardware links.

The file server then run under MS-DOS operation system on the host processor (master) to distribute the OCCAM modules and start the execution of individual modules on different transputers. The specify flag option of the server, -:o should be set to 1 if a large program is to be executed.

The commands used to compile, link, configure and run the concurrent model is given in Appendix C.

4.5 DISCUSSION

The transputer has made practical concurrent process computing methods explored to simulate a complicated device such as a diesel engine. Application of this method is still in its early stages. At the moment, the implementation of application software very much depends on the configuration of the hardware (this is the reason that the configuration program is required to allocate the software channels used by the OCCAM harnesses to the hardware links). One of the advantages of using OCCAM harnesses is that with different transputer networks, Only the OCCAM harnesses and configuration program need to be changed to handle the different communication scheme.

Figure 4.1 Schematic of the hardware layout

Figure 4.2(a) FLOW CHART OF THE TRANSIENT MODEL

Figure 4.2(b) Flow chart of the Engine Cycle Model

Figure 4.3 Organisation of the model tasks onto the computer hardware

Figure 4.4 Computing time comparison of conventional do loop with pipelining

CHAPTER 5

STRUCTURAL DISCUSSION OF THE COMPUTER PROGRAM

5.1 INTRODUCTION

This chapter is divided into three major sections to discuss the structure of the simulation program. The first section outlines the program modules. The second deals with the detailed structure of the program, giving a brief description of its essential subroutines. The input data requirements and the resulting output are discussed in the final section.

5.2 FLOW CHART AND PROGRAM MODULES

The code simulates the physical processes in the engine described in Chapter 2 was developed in standard FORTRAN 77. The description of various processes is carried out in separate subroutines, forming a modular structure in which each module can be developed independently and updated easily. Figure (5.1) shows a flow chart of the parallel computer program.

There are two modes of operation, i.e. the engine at steady operating conditions and under transient operating conditions. Under transient conditions, two schemes can be setup:

(1) Engine acceleration/deceleration, the operation starts when the governor control lever is suddenly moved. (The engine is in free acceleration when the external load is set to zero).

(2) Sudden load acceptance, the operation starts when the external load is suddenly applied/reduced.

The Program starts by reading an option from the keyboard which determines whether to load new engine specifications or using the ones already in file. If it is the first case, the new engine specifications can be input into file by following the instructions given by the computer. If it is the second case, the program proceeds to read the engine specifications and load conditions from a data file. Then the program requires an input from the keyboard to determine whether to simulate the engine at steady state conditions or under transient operating conditions.

If running at steady operating conditions, the cycle simulation begins by setting the crank angle equal to its initial value and by setting all sums accumulated over the cycle equal to zero. A test is then made to see if the inlet valve is closed. If it is the case, the program proceeds to perform the motoring calculation during the compression period, while checking whether the fuel injection has started. When the fuel injection starts, the injected fuel quantity and spray penetration are calculated. The relative position of the jet tip with respect to the piston geometry is computed to determine whether the jet impingement occurs. The air entrainment and fuel evaporation are then evaluated. When ignition occurs, the combustion and heat release calculations are performed. The calculation of heat transfer is carried out next. Then comes the energy balance. After the exhaust valve is opened, the exhaust and induction processes are computed. Cycle convergence can be obtained by successive iterations between the exhaust-induction model and the combustion model through the entire cycle using the absolute and fractional changes in the computed residual fraction as criteria.

When the cycle has converged, the program repeats the cycle simulation one more time. During this final calculation, the friction losses are evaluated to obtain the

net work and torque output. Furthermore, the soot and NO_x emissions are computed after the cylinder pressure and flame temperature are evaluated from the energy balance at each time step (typically 1° CA). The net soot and NO_x formed at each crank angle are integrated throughout the closed period. At the opening of the exhaust valve, the integrated quantities are corrected to normal temperature and pressure (NTP) and the resulting values are displayed on the screen to show the level of exhaust emissions in the engine while operating under the conditions at which the simulation has been carried out.

If running under transient operating conditions, the program asks whether to simulate the engine acceleration or load acceptance. In the first scheme, the movement of the governor control lever is the stimulus, while in the second scheme, the change in the engine conditions is caused by the suddenly applied load.

The first cycle is simulated at steady operating condition and iteration is carried out for this cycle. The program then calculates the governor control-spool movement according to the change in engine speed and governor control lever movement. From the governor control-spool movement and instantaneous engine speed, the fuelling is determined from the fuel pump characteristics and input to the cycle simulation discussed above. The acceleration of the engine is determined by instantaneous torque developed by the engine less that applied by the load and system inertia.

5.3 DETAILS OF ESSENTIAL SUBROUTINES

There are thirty major subroutines, supported by others, which have been distributed into four programs, namely combustion, entrainment, emission and graphics programs. The four programs can be implemented concurrently and communicate with each other on a four transputer network by using OCCAM as harnesses, the reason for this division and the detailed implementation can be found in Chapter 4.

The details of the essential subroutines and their distribution into these four programs are discussed in this section.

5.3.1 SUBROUTINES IN GRAPHICS PROGRAM

The graphics program is distributed to the root transputer and it handles all the communications between the target transputers and host processor. It also contains the subroutines necessary for the transient simulation and instantaneous graphics presentation. The essential subroutines in the program are organised in different groups listed as follow:

1. INPUT - input the engine specifications and operating conditions
2. ICLOCK - checks the running time
3. COMM - communicates with the other transputers and the host processor
4. TRAN - simulates the processes during the engine transient operations
5. OUTPUT - output results to file
6. GRAPH - plots the results during the implementation.

INPUT Routine

The following is a list of subroutines concerned with the data input before simulation:

- (1) SPEC - provides interactive data preparation and is called only when the new engine specifications and running conditions need to be loaded.
- (2) INPUT - reads data from files already created by the operator using subroutine SPEC and prints a brief description of the engine specifications and running conditions on the screen.

ICLOCK Routine

This is a run time library routine and called to obtain the instantaneous running time.

COMM Routine

In the COMM routine, the variables listed in Table 5.1 are read from an input data file at the beginning of the execution, then passed to the three target transputers. During the execution, the COMM routine input the results from the target transputers.

TRAN Routine

Transient operations are dealt with in the following subroutines:

- (1) FRICTION - calculates the friction losses in terms of FMEP (friction mean effective pressure) from the equation of Patton et al (71) so that the net torque developed by the engine can be evaluated.
- (2) GOVERNOR - predicts the governor control-spool movement and needs as input the governor control lever position and the instantaneous engine speed.
- (3) PUMP - characterises the fuel pump and determines the fuelling rate from the engine speed and governor control-spool position.
- (4) LOAD - calculates the load torque and determines the engine acceleration/deceleration.

OUTPUT Routine

The output of the results is carried out in three subroutines:

(1) BOSCH - converts the net soot formed to BOSCH units by interpolating from the values in Table 5.2.

(2) SOUTPUT - writes the values of the important variables at steady state operations.

(3) TOUTPUT - writes the values of the variables during transient operations.

GRAPH Routine

The following subroutines can provide the instantaneous graphics presentation during the execution :

(1) WINDOW - divides the screen into a number of windows for an organised results presentation.

(2) PLOT - plots the results in graphics form.

5.3.2 SUBROUTINES IN COMBUSTION PROGRAM

The combustion program is distributed to the first target transputer and simulates the major processes in an engine cycle. Several subroutines are involved in the simulation, among which the following are the most significant:

1.COMM - communicates with the other transputers, this includes two parts: (1) receiving data from other transputers (2) sending data to other transputers.

2. PREINJ - performs the calculations before injection

3. INICAL - initialises all the variables at the start of injection

4. INJSIM - simulates the injection process

5. EVAP - calculates the fuel evaporation rate

6. COMBUST - computes the fuel burning rate, heat release rate, heat transfer, mass and energy balance

7. MIX - simulates the turbulent mixing process

8. EXIN - simulates the gas exchange process.

The interact of these routines can be found in figure (4.2).

COMM Routine

In the COMM routine, the variables listed in Table 5.1 are input from the graphics program in the root transputer at the beginning of the execution. During injection, the COMM routine input the values of the jet tip velocity and entrainment rate from the entrainment program in the second target transputer at every degree crank angle.

When combustion occurs, the following variables are output to both the graphics program in the root transputer and the emission program in the third target transputer on every degree crank angle basis:

- Position of the crank angle
- Cylinder pressure
- Temperature of the burning region
- Volume of the burning region
- Air entrainment rate
- Air consumption rate
- Density of the burning mixture
- Accumulated fuel evaporated
- Fuel burning rate
- Accumulated fuel burnt
- Accumulated combustion products
- Mixing rate

During exhaust and induction processes, the predicted cylinder pressure is output to the graphics program to be displayed on the screen. When the cycle is completed, the IMEP (indicated mean effective pressure) is sent back to the graphics program.

PREINJ Routine

This routine carries out motoring computations for the period from the closure of the inlet valve to the start of injection.

INICAL Routine

When the injection starts, subroutine INICAL is executed to calculate the initial volume, pressure, temperature and mass of the charge at the start of injection. It also initialises volume and mass of fuel in the jet and assigns value of zero to all indicators and counters.

INJSIM Routine

When fuel injection starts, this routine is called to compute the following variables over the discretised time step (Δt):

- Injection velocity at the nozzle
- Quantity of fuel injected over the interval Δt
- Sauter Mean Diameter of droplets
- Instantaneous engine motoring conditions
- Cylinder trapped mass.

EVAP Routine

In EVAP routine, the fuel evaporation rate is calculated with and without combustion, using mass transfer considerations. The mixture formation is also considered in this routine.

COMBUSTION Routine

Combustion is dealt with in the following subroutines:

(1) BURRATE - calculates the fuel burning rate and heat release rate.

(2) WOSCHNI- evaluates the heat transfer between the cylinder contents and cylinder wall. The heat transfer coefficient is calculated from the equation of Woschni (47).

(3) MASSB - balances all the masses.

(4) ENEBAL - organises and solves the energy matrix discussed in Chapter 2, to provide increments of the cylinder pressure, the burning mixture and surrounding charge temperatures and the volume of the burning zone.

MIX Routine

After the end of injection, the jet is characterised by the turbulent mixing which is accounted for in the MIX routine. The evaporation and mixture formation are also evaluated in this routine.

EXIN Routine

The EXIN routine is executed when the exhaust valve is opened. The cylinder charge mass, pressure and temperature during the gas exchange period are evaluated from the theory discussed in section 2.2.3.

5.3.3 SUBROUTINES IN ENTRAINMENT PROGRAM

The entrainment program is allocated to the second target transputer and used to calculate the air entrainment into the free jet and wall jet (if jet impingement occurs). The essential subroutines involved in the calculations for air entrainment are:

1. COMM - communicates with the other transputers
2. FREJET - calculates the entrainment in the free jet

3. WALJET - calculates the entrainment in the wall jet.

COMM Routine

In the COMM routine, the variables listed in Table 5.1 are input from the graphics program in the root transputer at the beginning of the execution. When fuel injection starts, the air entrainment is calculated in the FREJET and WALJET routines at every degree crank angle. The COMM routine then passes the values of the jet tip velocity and entrainment rate to the combustion program in the first target transputer.

FREJET Routine

The change in air entrainment rate with distance along the jet trajectory is computed from equation (2.16). The equation for change in jet centreline trajectory by Sinnamon et al (54), combined with the correlation of Chiu et al (55) to obtain the change in jet penetration due to swirl, determines the instantaneous jet tip position. The criterion of Skifstad (56) is then used to determine whether the jet impinges the piston wall.

WALJET Routine

The equations by Spalding (95) are used to evaluate the air entrainment into the wall jet region.

5.3.4 SUBROUTINES IN EMISSION PROGRAM

The emission program is run in the third target transputer and used to calculate the exhaust emissions (soot and NO_x). It contains the following three major routines:

1. COMM - communicates with the other transputers
2. NO_x - calculates the NO_x emission

3. SMOKE - calculates the net soot emission according to the theory discussed in section 2.2.2.2.

COMM Routine

In the COMM routine, the twelve variables mentioned in section 5.3.2 are input from the combustion program in the first target transputer on every degree crank angle basis when combustion occurs:

After the soot and NO_x emissions are computed, the COMM routine passes the net values of the soot and NO_x emissions back to the graphics program in the root transputer.

SMOKE Routine

There are two subroutines to calculate the smoke emission:

- (1) SOOT - computes the soot formation from the equations of Tesner et al (61).
- (2) SMOX - calculates the soot oxidation from the equations of Magnussen (64).

NO_x Routine

The evaluation of the NO_x emission is carried out in two subroutines:

- (1) ECP - calculates the equilibrium composition of combustion products at the pressure and temperature pertaining to the burning region.
- (2) NITRO - solves the kinetic rate differential equation for NO formation using a 4th-Order Runge-Kutta algorithm.

5.4 MODEL INPUT AND OUTPUT

The input data requirements and the resulting output are discussed in this section.

5.4.1 MODEL INPUT REQUIREMENTS

The model is detailed and therefore requires as input data basic engine geometry. The model contains an interactive data preparation module that helps the operator input data in a suitable form for the model. This can be used to create a completely new data set or modify an existing data file. Alternatively data files can be created with the operative's own routines and submitted to model as ready to run input data files. The following input data is required:

1. Interactively from keyboard

(1) An option to determine whether to input different type of engine and different geometric specification:

(i) if $n_{spec} = 1$ loading new engine dimensions. Reference (4) gives a detailed terminal dialogue to input new engine specification and etc.

(ii) if $n_{spec} \neq 1$ using the engine dimensions and load conditions already on file.

(2) Name of the file containing the data already created, if n_{spec} is not set to 1.

(3) Model operational mode

(i) if $mode = 0$ steady operating condition

(ii) if $mode = 1$ transient operating condition

(4) EGR level.

2. Input from file

(1) Initial engine speed

(2) Initial load

The initial load can be input in the following two ways

- (i) in the form of Air/Fuel ratio
- (ii) fuel injected per cylinder per cycle, so that Air/Fuel can be calculated
- (3) Ambient pressure and temperature
- (4) Valve timing
- (5) Injection timing and duration
- (6) Required running time for the transient condition.

All calculations are carried out using SI units. To make the model more cost effective, the discrete calculation steps procedure used by Watson (44) is applied to the following processes as:

- (1) Compression 2° CA
- (2) Combustion and expansion 1° CA
- (3) Exhaust and induction 5° CA

The typical data file required as input to the model is listed in Table 5.1.

5.4.2 MODEL OUTPUT

At steady operating conditions, the instantaneous values corresponding to each crank position are written to file during the running of the program. The following is a list of these parameters:

- (1) Cylinder pressure
- (2) Temperature of the burning zone
- (3) Mean charge temperature
- (4) Fuel injection rate
- (5) Cumulative quantity of fuel injected
- (6) Fuel mass fractions:
 - Evaporated
 - Burned
- (7) Fuel burning rates:
 - Premixed

- Diffusion
- Total
- (8) Heat release rate
- (9) Heat flux between the cylinder contents and the walls
- (10) Air entrainment rate
- (11) Equilibrium concentration of NO_x
- (12) Kinetically controlled NO_x formation
- (13) Instantaneous soot concentrations:
 - Formation
 - Oxidation

Under transient operating conditions, the values of the above thirteen parameters can be written into file if required. Furthermore, the following results at the end of each cycle are also output to file:

- (1) Engine speed response
- (2) Soot emission
- (3) NO_x emission
- (4) Governor control spool movement
- (5) Fuelling rate
- (6) Vehicle speed (if the engine is running in a vehicle)
- (7) Friction mean effective pressure
- (8) Brake mean effective pressure
- (9) Instantaneous power developed by the engine.

If run sequentially on the Honeywell Multics, the output data mentioned above can be plotted using the existing graphics package, Tella-graph. If run in parallel on transputers, a TSR (terminate and stay resident) graphics routine has been developed to interrupt the host processor and plot the data on the EGA graphics card. While the target transputers perform the calculations for the current time step, the root transputer can send the results from the previous time step back to the host processor which displays the results on the screen, resulting in the calculation and plotting in parallel.

The instantaneous values of cylinder pressures are displayed in graphics form at successive crank angle over a cycle. At the end of each cycle, the movement of the governor control spool, the fuelling rate and the resulting engine speed are also presented in graphics form.

Four windows are opened on the screen and the cylinder pressure, governor control-spool travel, the fuelling rate and the engine speed response are plotted within each window. The graphics presentation can be any of the sixteen colours of the EGA (Enhanced Graphics Adapter) card. Apart from the graphics presentation, there is also a window for numerical output, including the instantaneous running time, the cycle number, the soot and NO_x emission levels, the start of injection and the vehicle speed.

5.5 SUMMARY

The general structure of the program is sufficiently flexible to enable adaptation to different engine types and configuration. The sequential code is developed in standard FORTRAN 77. No special library routines are required to run the model, making it portable. The parallel code has been developed for two, three and four transputer networks. The modular structure of the program makes it very easy to develop the parallel code to run on different number of transputers. Different hardware configuration and communication schemes are handled in the OCCAM harnesses.

A Menu driven user interface has been developed to drive and support the model. Various options have been built into the package so that the model can be run sequentially on the IBM PC-XT or a single transputer, or run in parallel on two, three, or four transputers, with various graphics presentations.

The package also provides the options to run the model to simulate the engine on a test bed or in a minibus, and run the model with different communication schemes.

Table 5.1 Input Data for Baseline Condition (SET A)

VARIABLE (VALUE)				
INITIAL ENGINE SPEED 1000 (RPM)	STROKE 0.09054 (M)	BORE 0.09367 (M)	COMPRESSION RATIO 19.1	CONNECTING ROD LENGTH 0.154 (M)
COMBUSTION CHAMBER HIGHT 17 (MM)	COMBUSTION CHMABER WIDTH 47.828 (MM)	VOLUME OF BOWL $0.257 \cdot 10^{-4} \text{ (M}^3\text{)}$	PISTON SURFACE AREA $0.415 \cdot 10^{-5} \text{ (M}^2\text{)}$	CLEARANCE HIGHT $0.127 \cdot 10^{-2} \text{ (M)}$
SWIRL RATIO 2.06	NUMBER OF NOZZLE HOLES 4	DIAMETER OF NOZZLE HOLE 0.23 (MM)	DISCHARGE COEFFICIENT 0.45	
START OF INJECTION 3.5 ($^{\circ}\text{CA BTDC}$)	INJECTION DURATION 8.1 ($^{\circ}\text{CA}$)	INLET VALVE CLOSURE 141 ($^{\circ}\text{CA BTDC}$)	EXHAUST VALVE OPENING 126 ($^{\circ}\text{CA ATDC}$)	
VOLUMETRIC EFFICIENCY 0.85	AMBIENT PRESSURE 1.03 (BAR)	AMBIENT TEMPERATURE 298.6 ($^{\circ}\text{C}$)	VOLUME OF FUEL INJECTED $28.1 \cdot 10^{-9} \text{ (M}^3\text{)}$	

Table 5.2 Coverision from Mass Concentration to BOSCH units

SOLID CONTENT (gm/cubic metre)	BOSCH UNITS	SOLID CONTENT (gm/cubic metre)	BOSCH UNITS
0.028	1.0	0.279	4.0
0.033	1.1	0.295	4.1
0.037	1.2	0.311	4.2
0.041	1.3	0.329	4.3
0.046	1.4	0.347	4.4
0.051	1.5	0.366	4.5
0.056	1.6	0.386	4.6
0.062	1.7	0.407	4.7
0.068	1.8	0.429	4.8
0.074	1.9	0.450	4.9
0.079	2.0	0.474	5.0
0.086	2.1	0.499	5.1
0.092	2.2	0.525	5.2
0.099	2.3	0.551	5.3
0.107	2.4	0.579	5.4
0.115	2.5	0.607	5.5
0.123	2.6	0.636	5.6
0.131	2.7	0.666	5.7
0.139	2.8	0.698	5.8
0.147	2.9	0.730	5.9
0.156	3.0	0.763	6.0
0.166	3.1	0.796	6.1
0.175	3.2	0.832	6.2
0.186	3.3	0.869	6.3
0.197	3.4	0.906	6.4
0.209	3.5	0.943	6.5
0.222	3.6	0.980	6.6
0.236	3.7	1.019	6.7
0.249	3.8	1.058	6.8
0.264	3.9	1.097	6.9
0.279	4.0	1.136	7.0

Figure 5.1 Flow Chart of the Parallel Model

CHAPTER 6

EXPERIMENTAL TECHNIQUES

6.1 INTRODUCTION

This chapter provides a description of the experimental programme carried out during the present work, including the tests conducted at steady and transient operating conditions. Details of the instrumentation and the microcomputer based data acquisition and analysis are also presented.

The work reported here is divided into three sections. The first section deals with the steady state measurements while the second discusses the measurements conducted under transient operating conditions. Microcomputer based data acquisition and analysis of the experimental results are presented in the third section.

The experiments were carried out on a FORD 2.5 L swirl chamber direct injection diesel engine available in the Internal Combustion engine laboratory of the department of Mechanical Engineering, Loughborough University of Technology. This is a four cylinder four stroke high speed engine. The fuel injection system comprised a BOSCH distributor VE type pump with four injectors connected by high pressure pipes of equal length. A flyweight-spring type mechanical governor was an integral part of the fuel pump assembly. The engine test bed, the major instrumentation and the microcomputer based data acquisition system are shown in figure (6.1), (6.2) and (6.3) respectively.

The engine was loaded by a Froude type G hydraulic dynamometer capable of a maximum BHP (Brake Horse Power) of 200 at 2550 rpm. Details of the experimental engine are given in Appendix A.

6.2 STEADY STATE MEASUREMENTS

Extensive experimental data obtained at steady state on an identical engine to the one mentioned above have been supplied by the FORD motor company. These include the cylinder pressures, fuel line pressures and other data listed in Table 2.1. The conditions at which these data were taken are given in Table 6.1.

In the process of simulating the governor dynamics (section 2.2.5), the governor system parameters such as dimension, governor spring stiffness etc. need to be evaluated. Furthermore, the steady state characteristics of the fuel pump needs to be measured to determine the fuelling rate. Measurements of these two sets of data are discussed in detail in this section.

6.2.1 MEASUREMENTS OF THE GOVERNOR COMPONENTS

The governor under investigation was a variable speed type in a BOSCH VE type fuel pump and the mathematical modelling of the governor has been conducted in section 2.2.5. In order to evaluate the four governor system constants C_1 to C_4 in equation (2.73), the following variables were measured after the fuel pump was disassembled:

1. Masses of the governor flyweight and sleeve, m and M
2. Dimensions of the governor, r , v , h , L_1 , L_2 and L_3
3. Stiffness of the spring, K
4. Ratio of governor gear and camshaft gear
5. Maximum travel of governor control spool

The measured values of the above eleven variables are listed in Table 6.2.

6.2.2 FUEL PUMP CHARACTERISTICS MEASUREMENTS

The fuelling per cycle per cylinder was evaluated from the overall fuel consumption measurements which were carried out by using two Carrier venturi meters, one in in-flow and the other in the return flow from the injector spill. An Ultra Violet type 2000 Galvanometer Recorder (U.V. recorder) was used. Calibration of the two venturi meters is as follow:

1. The two meters were connected sequentially in the in-flow line (just for the calibration purpose) and the signals from the two meters were recorded on the U.V. recorder and calibrated with the actual fuel flow rate obtained from the time for 210cc fuel passed through one section of the in- flow line.
2. A FORTRAN program called **linefit** was written based on the least square method to fit the calibration data and the resulting calibration factors for the two venturi meters with the experimental data are shown in figure (6.4).

The quantity of fuel per stroke per cylinder were measured at different engine speed (1000-3000 rpm) and governor control spool position (25-100%). The experimental data were then fitted using the Taylor Series, as shown in figure (6.5), resulting in the fuel pump characteristics in the form of equation (2.84).

6.3 TRANSIENT MEASUREMENTS

In order to validate the prediction from the computer model, the following performance and emission data were measured under transient operating conditions:

1. Cylinder pressure
2. Exhaust gas temperature

3. Engine speed response
4. Exhaust smoke emission

The over-all responses of the external variables, i.e. exhaust gas temperature, engine speed and exhaust smoke emission under transient were measured for periods up to 150 engine cycles. In order to validate the computed values that vary during the cycle, the cylinder pressure histories in cylinder no. 1 were recorded for the first 50 engine cycles.

The measurements of fuel flow during transient operations were also attempted but not very successfully. Because of the small diameter in the venturi meters, depressurisation occurred when increased amount of fuel was required to pass through the two meters rapidly and a large amount of air was sucked into both in-flow and outflow. Consequently the results obtained on the U.V. recorder would not represent the actual fuelling when the engine was operated normally without the two venturi meters on the fuel lines.

Three types of transient tests have been carried out, namely,

1. engine free acceleration
2. engine acceleration with load taken up due to throttle demand
3. sudden applied load at constant governor setting.

6.3.1 CYLINDER PRESSURE MEASUREMENTS

The cylinder pressure was measured using a Kistler Type 601A piezo-electric pressure transducer capable of a maximum pressure 1000 bars and amplified using a Kistler Type 5001 charge amplifier.

Calibration of Pressure Transducer

Calibration of the pressure transducer and charge amplifier was conducted with reference pressures supplied with a dead weight tester, and the charge amplifier

output monitored with a digital voltmeter. The photograph of the calibration assembly is given in figure (6.6).

The pressure transducer was calibrated from 10 bar to 100 bars in steps of 10 bar. The correlation between the applied pressure and the output voltage was found to be linear within the range under consideration. The FORTRAN program **linefit** was then used to fit the calibration data and the resulting calibration factor with the experimental data is shown in figure (6.7).

A long time constant was used on the charge amplifier during the calibration but was switched on a short time constant during operation with engine pressure data.

6.3.2 EXHAUST TEMPERATURE MEASUREMENTS

The transient exhaust temperature was measured using a thermocouple which was inserted in the exhaust manifold and the thermocouple signal was amplified using an output signal multiplier (OSM). This thermocouple is a fast response chromel/alumel type and its dynamic response time is less than 0.1 second.

Calibration of Thermocouple

Calibration of the thermocouple and OSM was carried out with reference temperatures measured by a digital thermometer and the OSM output monitored with a Nicolet Storage oscilloscope.

The thermocouple was calibrated from 296⁰ C to 361⁰ C. The linearity between the measured temperature and the voltage was fairly good within the range under consideration. The FORTRAN program **linefit** was used to fit the calibration data and the resulting calibration factor with the experimental data is shown in figure (6.8).

The maximum time for one measurement was in the range from 15 to 20 seconds. Measuring for longer times, resulted in the smoke particles flowing in the exhaust

manifold sticking to the thermocouple, making it less sensitive to the temperature change.

6.3.3 ENGINE SPEED MEASUREMENTS

Engine speed was measured by two devices:

1. The shaft was fitted with a 180 tooth gear wheel and an electronic counter was used to count the Top Dead Centre (TDC) signal to get rpm signal
2. A frequency to voltage converter (FVC) was designed to convert the rpm signal to voltage.

Calibration of FVC

Calibration of the FVC was carried out with reference engine speeds measured by the FVC and the FVC output monitored with a Nicolet Storage oscilloscope.

The FVC was calibrated from 729 rpm to 2892 rpm. The correlation between the engine speed measured and the output voltage was found to be linear within the range under consideration. The FORTRAN program `linefit` was then used to fit the calibration data and the resulting calibration factor with the experimental data is shown in figure (6.9).

6.3.4 EXHAUST SMOKE MEASUREMENTS

During transient tests, the exhaust smoke responds so fast that any cumulative type smokemeter was considered not appropriate. In the present work, the exhaust smoke emission was measured by using a laser and a photomultiplier tube (PMT). The laser intensity is alternated by the flow of smoke particles, the larger the amount of particles the less the laser intensity, so that the signal from the PMT can be used to represent smoke emission level.

6.3.4.1 Description of the Experimental Rig

A small pipe was joined to the main exhaust manifold and a control valve was installed in this pipe. Two windows were positioned in the pipe near the joint and a laser beam was passed through the windows. When the control valve was opened, part of the exhaust gas flowed into this pipe from the main manifold and reduced the intensity of the laser beam. A PMT was aligned to record the decreased intensity of the laser beam from the windows. The output signal from the PMT was then amplified by an output signal multiplier (OSM).

Three problems have been encountered with the measurements of transient soot emission using this method:

1. When the control valve was opened, the soot particles flowed into the pipe. Some of the particles stuck onto the windows and obscured the laser source very quickly. So, after a very short time the PMT signal would not represent the actual soot concentration. To solve this problem, compressed air was made to flow on the inside surface of the windows thus blowing away the soot particles. It was found that to keep the window clean for sufficient time (3-5 seconds), before blocking became significant, the air pressure would be in the range of 1.5 to 2 bars.
2. When the compressed air was supplied, the majority of it went into the pipe and diluted the soot concentration. This brought a problem in calibration. At steady state, a Hartridge smokemeter installed in the downstream of the pipe was used to calibrate the PMT signal. Since the soot flow was diluted, the calibration was not reliable. To solve this problem, two walls were built inside the rig and two holes were drilled in them to let the laser beam go through. Since the holes were small (3.5 mm in diameter at the moment), the majority of the compressed air would go out of the rig instead of flowing into the pipe. It was found that with air pressure of 2 bars, dilution of the soot particles was 5.7%, while with pressure of 1.5 bars, the dilution could reduce to only 1.4%.

3. Building of walls inside the rig did prevent the dilution of soot flow. However it made vibration a severe problem. When the engine was running, the rig was vibrating with the engine. The laser beam would not always pass through the holes on the inner walls, effectively being shuttered, so that the PMT would produce a vibrating signal which was much bigger than the signal representing the soot concentration. Some signal variation on the oscilloscope could still be observed even when the control valve was closed (there was no soot flow). This problem was overcome by making the laser source vibrate with the rig so that there would be no relative movement between the laser beam and the rig. This was achieved by adopting two fibre optics, one connecting the laser source to a wall, the other connecting the other wall to the PMT. This reduced signal variation was observed on the oscilloscope when the control valve was closed and the engine run at varied speed (the rig vibrated at varying frequency).

The schematic and photograph of the final experimental rig for smoke measurements are given in figure (6.10) and (6.11) respectively.

Calibration of PMT

Calibration of the PMT and OSM was conducted at steady state, with reference soot emission levels measured by a Hartridge smokemeter and the OSM output monitored with a Nicolet Storage oscilloscope.

The PMT was calibrated from 17 Hartridge units to 67 Hartridge units. Within the above range, the relationship between the measured soot level and the voltage was considered to be fairly linearised. The FORTRAN program **linefit** was used to fit the calibration data and the resulting calibration factor with the experimental data is shown in figure (6.12).

6.4 DATA ACQUISITION AND ANALYSIS

As mentioned in section 6.3, Three types of transient tests have been conducted. For the first two types, the governor control lever was used as a trigger and for the last one, the load lever was used as a trigger to start the data acquisition.

The schematic of the engine test bed and data acquisition system, based on a VICTOR microcomputer is presented in figure (6.13).

6.4.1 MICROCOMPUTER BASED DATA ACQUISITION

The digital computer used for the on line real time acquisition of the data was a VICTOR VPC III 286 micro-computer (IBM-PC compatible). It is enhanced to provide two different clock rates - 6 MHz and 8MHz. It has a 32-bit word length and 640 kbytes memory. The computer stores the data using DMA (Direct Memory Access) mode and the analog signal is converted to the digital signal through an A/D converter.

6.4.1.1 Acquisition of Cylinder Pressure

The pressure signal was digitised in synchronisation with engine crank angle at an interval of two degree. The conversion procedure took the following steps:

$\Downarrow$
 Integer

where the integer is a value between 0 - 4096, corresponding to an input voltage to the analog/digital converter of 0 - 10 volts.

The computer data acquisition software was designed to store and digitise the cylinder pressure for the first 50 consecutive cycles when the transient running was triggered.

6.4.1.2 Acquisition of Exhaust Temperature

After the transient running was triggered, the exhaust temperature data was stored for 7 seconds (about 150 cycles when an engine run at medium speed) and digitised by the A/D converter at a rate of 100 data points per second. The conversion for the temperature signal took the following steps:

Exhaust Temperature
 $\Downarrow$
 $\Downarrow$
 Thermocouple--OSM
 $\Downarrow$
 $\Downarrow$
 A/D Converter
 $\Downarrow$
 $\Downarrow$
 Integer

6.4.1.3 Acquisition of Engine Speed

After the transient running was triggered, the engine speed data was also recorded for 7 seconds and digitised at a rate of 100 data points per second. The following listed the procedure for the conversion of the engine speed signal:

6.4.1.4 Acquisition of Soot Emission

After the transient running was triggered, the computer records the soot concentration data for 7 seconds and digitised it at a rate of 100 data points per second. The conversion of the soot concentration signal was as follow:

6.4.2 ANALYSIS OF EXPERIMENTAL RESULTS

Preliminary tests were carried out and the original pressure, exhaust temperature, engine speed and exhaust smoke data at the second type of transient tests were stored on a Nicolet Storage oscilloscope and photographed. Figures (6.14)-(6.17) show these photographs.

The results indicate that the cylinder pressure, smoke emission and engine speed rise rapidly with the sudden movement of the governor setting, whilst the exhaust gas temperature builds up in a slower manner.

6.4.2.1 Analysis of Cylinder Pressure Data

The piezo-electric transducer measures relative rather than absolute pressures. The absolute values of each integer obtained in the data acquisition process was directly related to the absolute cylinder pressure. The integer obtained from the A/D converter was multiplied by an A/D factor (5.12/2048 for the converter used in this work) and then multiplied by the pressure transducer calibration factor (obtained from the dead weight tests) to obtain the relative pressure in bars. The absolute pressure at each crank angle was then obtained by adding a constant amount to the relative cylinder pressure at each crank angle. The amount to be added was determined by assuming that the cylinder pressure at Bottom Dead Centre (BDC) of intake was equal to the mean intake manifold absolute pressure. This method was following the one adopted by Lancaster (96). The difference between the mean intake manifold pressure and the relative pressure at BDC of intake was then added to each of the 360 points making up one complete cycle.

The pressure data thus obtained for cycle 1 to 46 at an interval of 9 cycles are presented in figures (6.18)-(6.20), each corresponding to one of the three types of tests.

6.4.2.2 Analysis of Exhaust Temperature Data

The integer converted from the exhaust temperature signal by the A/D converter was multiplied by the A/D factor and then multiplied by the thermocouple calibration factor to obtain the temperature in $^{\circ}\text{C}$. Since the signal multiplied by the OSM was inverted, the resulting data had to be inverted back to get the actual temperature response.

The resulting temperatures for the three types of tests are shown in figures (6.21)-(6.23).

6.4.2.3 Analysis of Engine Speed Data

The integer obtained from the engine speed signal was multiplied by the A/D factor and then multiplied by the FVC calibration factor to obtain the engine speed responses in rpm.

Figures (6.24)-(6.26) give the resulting engine speed responses for the three types of tests.

6.4.2.4 Analysis of Soot Emission Data

The conversion of the integer mentioned in section 6.4.1.4 back to the soot concentration took the following steps:

1. The integer was multiplied by the A/D factor and then the PMT calibration factor to obtain the soot concentration in Hartridge units
2. Since the PMT signal was multiplied by the OSM, the Soot concentration was inverted

3. The soot concentration was then converted from Hartridge units to Bosch units for the convenience of comparison with the predicted results. A FORTRAN program was written to carry out this conversion based on the values in Table 6.3.

The soot concentrations obtained from the above procedures for the three types of tests are presented in figures (6.27)-(6.29).

6.5 CONCLUSION

The cylinder pressure, engine speed and exhaust temperature have been quite successfully measured.

The exhaust smoke measurements demonstrated the feasibility of the method, but further development work is needed for more accurate measurements, such as optimisation of the diameters of the walls.

The results (figures (6.18)-(6.29)) form a basis used to validate the predictions from the model and this will be conducted in chapter 7.

Table 6.1 The Operating Conditions of the Experimental Data

	A	B	C	D	E	F	G	H	I	J	K	L
ENGINE SPEED (RPM)	1000	1000	1000	2000	2000	2000	3000	3000	3000	4000	4000	4000
AIR/FUEL RATIO	32.94	32.82	32.76	25.56	25.66	26.35	24.40	24.45	25.55	26.67	26.76	27.05
INJECTION TIMING (°CA BTDC)	3.5	7.3	0.9	3.7	6.1	0.5	7.5	10.5	5.0	11.3	13.1	8.5
INJECTION DURATION (°CA)	8.1	8.6	8.3	13.9	14.0	14.0	20.0	20.3	19.0	24.5	24.6	24.4
IGNITION DELAY (°CA)	6.1	6.0	6.6	6.8	6.3	7.7	8.7	9.0	9.8	10.2	10.7	10.6
FUEL INJECTED (MM ³ /STR/CYL)	28.1	28.6	28.2	34.1	34.0	33.5	35.5	35.5	35.2	34.1	33.9	33.7

Table 6.2 Measured Data of the Governor

VARIABLE	VALUE
MASS OF FLYWEIGHT m (g)	10.4
MASS OF GOVERNOR SLEEVE M (g)	11.9
STIFFNESS OF SPRING K (KN/M)	2.2727
RATIO OF GOVERNOR GEAR AND CAMSHAFT GEAR	37/23
MAXIMUM OF GOVERNOR C-S TRAVEL (MM)	9.47
DIMENSION OF GOVERNOR (MM) v h r L1 L2 L3	 11.5 13.1 16.2 47.5 23.5 12.0

Table 6.3 Conversion from Hartridge Unit to Bosch Unit

SMOKEMETER UNITS			
HARTRIDGE	BOSCH	HARTRIDGE	BOSCH
10	1.10	50	3.91
1	1.20	1	3.96
2	1.30	2	4.01
3	1.40	3	4.07
4	1.48	4	4.12
5	1.57	5	4.17
6	1.67	6	4.22
7	1.75	7	4.27
8	1.84	8	4.32
9	1.93	9	4.37
20	2.02	60	4.42
1	2.10	1	4.47
2	2.18	2	4.52
3	2.26	3	4.57
4	2.34	4	4.62
5	2.42	5	4.67
6	2.50	6	4.72
7	2.57	7	4.76
8	2.64	8	4.81
9	2.71	9	4.86
30	2.77	70	4.91
1	2.83	1	4.96
2	2.89	2	5.01
3	2.96	3	5.07
4	3.02	4	5.12
5	3.08	5	5.17
6	3.14	6	5.23
7	3.20	7	5.28
8	3.26	8	5.34
9	3.31	9	5.40
40	3.37	80	5.45
1	3.42	1	5.51
2	3.47	2	5.57
3	3.53	3	5.65
4	3.59	4	5.72
5	3.64	5	5.80
6	3.70	6	5.87
7	3.75	7	5.95
8	3.80	8	6.04
9	3.86	9	6.13
50	3.91	90	6.22

Figure 6.1 Engine Test Bed

Figure 6.2 Instrumentation of the Experiments

Figure 6.3 Microcomputer Based Data Acquisition System

Figure 6.4 Calibration of the Two Venturi Meters

Throttle opening 25%

Throttle opening 50%

Throttle opening 75%

Throttle opening 100%

Figure 6.5 Curve Fitting of Fuel Pump Characteristics by Taylor Series

Figure 6.6 Rig for Pressure Transducer Calibration

Figure 6.7 Calibration of the Pressure Transducer

Figure 6.8 Calibration of the Thermocouple

Figure 6.9 Calibration of the Frequency to Voltage Converter

1. laser source
2. optical fiber
3. adapter
4. cap
5. compressed air
6. main body of the rig
7. air passby pipe
8. inner wall
9. soot particles
- 10 photo-multiplier

Figure 6.10 Schematic of the rig

Figure 6.11 Rig for Exhaust smoke Measurement

Figure 6.12 Calibration of the Soot Concentration

Figure 6.13 Schematic diagram of the engine test bed and computer based data acquisition system

(a)

(b)

Figure 6.14 Experimental Cylinder pressures (Engine Acceleration with Load Taken up due to the Throttle Demand, Shown on the Oscilloscope)

(a)

(b)

Figure 6.15 Experimental Exhaust Temperature (Engine Operating Condition Is the Same with Figure 6.14)

(a)

(b)

Figure 6.16 Experimental Engine Speed Response (Engine Operating Condition Is the Same with Figure 6.14)

(a)

(b)

0.11V

1.87s

Figure 6.17 Experimental Exhaust Smoke Emission (Engine Operating Condition Is the Same with Figure 6.14) (a-b)

(c)

(d)

0.33V

2.44s

Figure 6.17 Experimental Exhaust Smoke Emission (Engine Operating Condition Is the Same with Figure 6.14) (c-d)

Figure 6.18 Experimental Engine Cylinder Pressure (Type I)

Figure 6.19 Experimental Engine Cylinder Pressure (Type II)

Figure 6.20 Experimental Engine Cylinder Pressure (Type III)

Figure 6.21 Experimental Exhaust Temperature (Type I)

Figure 6.22 Experimental Exhaust Temperature (Type II)

Figure 6.23 Experimental Exhaust Temperature (Type III)

Figure 6.24 Experimental Engine Speed Responses (Type I)

Figure 6.25 Experimental Engine Speed Responses (Type II)

Figure 6.26 Experimental Engine Speed Responses (Type III)

Figure 6.27 Experimental Exhaust Smoke Emission (Type I)

Figure 6.28 Experimental Exhaust Smoke Emission (Type II)

Figure 6.29 Experimental Exhaust Smoke Emission (Type III)

CHAPTER 7

EVALUATION OF SIMULATION RESULTS

7.1 INTRODUCTION

In this chapter, the predicted results are compared with the engine experimental data thus demonstrating the validity of the simulation program.

The work is presented in three sections, namely:

1. Evaluation of the predictions at steady state with the aid of experimental data
2. Validation of the performance and emission predictions under transient operating conditions against the experimental data discussed in section 6.4
3. Discussion of the results.

7.2 RESULTS AT STEADY OPERATING CONDITIONS

The model was first tested at steady state in which the main input data were engine speed and fuelling (or Air/Fuel ratio). In these situations, the governor, fuel pump and engine-load dynamics models were disabled.

An extensive series of program runs have been carried out in order to assess the ability of the simulation to predict the steady state performance and emission of a diesel engine. Initially the model was executed at those running conditions at which experimental data are available. These conditions may be found in Table 6.1 and covers a wide operating range (engine speed 1000-4000 rpm, A/F 24-29).

The main outputs from the model at steady operating conditions were outlined in section 5.4.2 and these results are stored on a data file ready to be presented in graphical form and used for diagnostic analysis. The cylinder pressure may be plotted against crank angle, from the closure of the inlet valve to the opening of the exhaust valve while all the other outputs can be plotted against crank angle from the start of injection to the opening of the exhaust valve.

7.2.1 CYLINDER PRESSURE

The comparison of measured cylinder pressure diagrams taken from a four cylinder engine (details given in Appendix A) with model predictions for various operating conditions is presented in figures (7.1)-(7.12). It may be seen that agreement between prediction and experiments is in general good over a wide operating range (engine speed 1000-4000 rpm, A/F 24-29). Since pressure in the cylinder is one of the most fundamental variables and provides a preliminary picture of engine performance, the agreement shown in figures (7.1)-(7.12) gives an idea of the accuracy of the model running at steady operating conditions.

7.2.2 OTHER PREDICTED RESULTS AT BASELINE CONDITION

The other predicted results are rate and accumulated heat release, jet and charge temperatures, premixed, diffusion and total fuel burning rates, mass fractions of fuel injected, evaporated and burnt, air entrainment and fuel injection rates, heat flux, NO_x equilibrium and kinetic concentrations and smoke formation and oxidation. Since the nature of the results is similar at all the operating conditions, plots of these are presented only at baseline condition (Set A) in figures (7.13)-(7.20). Details of the baseline condition are as follow:

Engine speed = 1000 rpm
 Air/fuel ratio = 28.94
 Injection timing = -3.5° CA
 Injection Duration = 8.1° CA
 Fuel injected = $28.1 \text{ mm}^3/\text{STROKE}$

For conditions other than baseline only the essential variables are investigated and presented in parametric form in section 7.2.4.

Air entrainment rate and fuel injection rate are given in figure (7.17) and it can be noticed that in the early stage of injection, the air entrainment rate is very much dependent on the fuel injection rate. This is consistent with the findings of other worker (4). Wall jet occurs only for a short time near the end of injection, and when it happens the air is entrained at a slower rate than the one in the free jet region.

It can be observed from figures (7.13) and (7.15) that the two mechanisms, premixed and diffusion controlled burnings discussed in section 2.2.1 reproduce the fundamental nature of the diesel engine combustion; i.e. an initial peak followed by gradual decay in the heat release rate. The jet and charge temperatures, heat flux and NO_x formation all follow this trend.

The predicted NO_x and smoke emissions are shown in figures (7.19) and (7.20) respectively. Chemical kinetics shows that the formation of NO_x is mainly controlled by kinetic rate which is highly dependent on flame temperature. From figure (7.19) it can be seen that the highest concentration of NO_x takes place around 15° CA where the flame temperature comes to a maximum value (figure (7.14)). Also in agreement with the findings of other investigators (97, 98), nitric oxide was formed very rapidly, reaching a peak value in less than 20° CA. Raine (99) has shown that the soot oxidation occurs mainly during the high temperature period and this finding can also be noted from figure (7.20).

7.2.3 PARAMETRIC VARIATIONS

Effects of the changes in the following engine operating conditions are investigated:

1. Effect of injection timing
2. Effect of engine speed

7.2.3.1 Effects of Injection Timing

Advanced injection timing increases the ignition delay and hence leads to a higher peak pressure. This effect is illustrated by the model in figure (7.21) (a). The influence of timing on the burning zone temperature is given in figure (7.21) (b). With retarded injection timing, heat release occurs later in the expansion stroke, leading to a lower temperature in the cylinder.

Figures (7.21) (c) and (d) illustrate the effects of injection scheduling on exhaust emissions. It is well established that the exhaust NO_x emission is very much dependent on temperature (100) and (3). The higher temperature, associated with advanced timing, causes higher NO_x level. This effect is demonstrated by the model predictions shown in figure (7.21) (c).

As discussed by Mehta (4), the exhaust smoke emission is very much dependent on the temperature. The lower temperature with retarded condition causes poor oxidation, therefore higher smoke emission. Figure (7.21) (d) shows that the model is capable of predicting the right trend but not the exact values. Taking the global nature of the smoke emission model into account, the inaccuracy in the predicted values is considered acceptable.

7.2.3.2 Effects of Engine Speed

Higher engine speed results in high level of air movement and therefore improves air/fuel mixing, leading to higher cylinder pressure and temperature. This effect is illustrated by the model in figure (7.22) (a) and (b).

Effects of engine speed on exhaust emissions are also investigated. The higher temperature with increased engine speed causes higher NO_x level and a reduction of smoke level and these trends have been observed in references (101) and (102). The model has accurately predicted these trends and the results are shown in figure (7.22) (c) and (d).

7.3 RESULTS AT TRANSIENT CONDITIONS

As mentioned in Chapter 6, the following types of transient tests have been carried out:

1. engine free acceleration
2. engine acceleration with load taken up due to throttle demand
3. sudden applied load at constant governor setting.

The predicted and experimental results obtained at these three basic types of tests are presented in this section.

7.3.1 ENGINE FREE ACCELERATION

The engine accelerates freely when the external load is set to zero and the system inertia is sum of the engine inertia and flywheel inertia.

The measured and predicted pressure diagrams for cycle 1 to 46 at an interval of 9 are presented in figures (7.23) and (7.24) respectively. From the above figures, it can be seen that the predicted and measured cylinder pressure follow the same trend, i.e. when the transient is triggered, the peak pressures rise rapidly in the early stage and increase slowly after about 30 cycles.

The predicted and measured engine speed response show excellent agreement in figure (7.25). This indicates that the engine acceleration, therefore the power

developed by the engine is evaluated accurately by the model. Without external load, the engine responds to the sudden movement of the governor setting very quickly. From figure (7.25), it can be seen that both experimental and predicted speed, show that it takes about 2 seconds for engine speed to increase from 1000 rpm to 2400 rpm.

Figure (7.26) shows the measured and predicted responses of the exhaust temperatures. Although the trends of the predicted and experimental are similar, the values are quite different. This difference may be due to the fact that the predicted value is the temperature of the gas just coming out of the cylinder, while the experimental is the temperature of the gas measured in downstream of the exhaust manifold (about one fourth of a meter away from the cylinders). The hot gas is cooled down when travelling from cylinders to the point being measured. From the energy conservation, the following equation can be written:

$$M_g C_p (T_1 - T_2) = h A ((T_1 + T_2)/2 - T_a) \quad (7.1)$$

where M_g is the mass of exhaust gases

C_p is the specific heat

h is the heat transfer coefficient

A is the heat transfer area

T_1 , T_2 and T_a are temperatures of upstream, down stream and atmospheric respectively.

When the measured down stream temperature (T_2) is 290°C , the upstream temperature (T_1) calculated from equation (7.1) is about 378°C , taking approximated values for the other parameters ($M_g = 7.1 \times 10^{-4}$ kg, $C_p = 10810$ J/kgK, $h = 47$ J/m²K, $T_a = 25^\circ\text{C}$ and $A = 0.047$ m²). From figure (7.26), it can be seen that the upstream temperature (T_1) predicted from the model is about 360°C when the measured down stream temperature (T_2) is 290°C . So taking the measurement position into account, the predicted exhaust temperature from the model is quite satisfactory.

The measured and predicted exhaust smoke are presented in figure (7.27) and it can be seen that although both the predicted and experimental soot concentration show a sudden increase after the transient is triggered, there are three differences between them.

1. The difference in the maximum values may be due to the overestimation by prediction. The possible reason for this is the inaccuracy by ignoring the spatial effects in the smoke model. This may be caused by the treatment of the whole burning jet as one zone. In the future development, multi-zone combustion models can be considered and the smoke formation can be calculated locally in each zone to improve the accuracy of the smoke emission.

2. The delayed rising of the experimental soot concentrations. One of the assumptions made in the modelling is that all cylinders in a multicylinder engine behave identically at the same point of their thermodynamic cycle. However, the pressure wave in the real engine fuel line will cause a delay in the actual response of the engine. Therefore a delay in the experimental soot concentration. Figure (7.28) shows that the measured smoke concentration is shifted 1.5 seconds to the left, good agreement is obtained as far as the start of overshooting is concerned.

3. The difference in values between the measured and predicted after the running returns to steady state again. Effects of engine speed on exhaust smoke emission at steady state are discussed in section 7.2.3.2. The higher temperature with increased engine speed results in a significantly higher soot oxidation rate, therefore a reduction of net smoke emission level. It is quite clear from figure (7.28) that the smoke level after transient (at 2400 RPM) is lower than the one before the transient operation has started (at 1000 RPM). Although the application of compressed air blows away the soot particles, it can keep the window clean for only 3 to 4 seconds before the blocking becomes significant. After 3 to 4 seconds, the running reaches to steady state and the measured values at this period overestimates the soot emission.

Plots of other variables predicted by the model are presented in figures (7.29)-(7.36). The predicted movement of the governor control-spool (CS) is shown in figure (7.29). It can be noticed that it takes about half a second for the CS to travel to the maximum when the governor control lever is suddenly pushed to the maximum. This is due to the system inertia and damping. The predicted fuelling rate is given in figure (7.30) and a similar trend to the governor CS movement can be observed. When the transient running is triggered by the sudden movement of the governor control lever, the increased fuelling results in higher temperature in the cylinders, leading to an overshooting in NO_x emission. This is demonstrated by the model in figure (7.31). The friction loss is plotted in figure (7.32) and it can be seen that as the engine speed rises from 1000 rpm to 2400 rpm, the FMEP increased from 1.52 to 1.6 bars. From figure (7.33), it can be seen that the start of injection increases from 3.5 CA° to 4.9 CA° within about one second and then decreases to 4 CA°, while the injection duration rises steadily until the engine comes to steady again. A steady increase in the air flow rate is observed in figure (7.35) and this is considered to be the right trend since the air flow rate more or less follows the engine speed response. Figure (7.36) shows that there is an overshoot in the engine overall equivalence ratio, indicating that the fuelling rate increases faster than the air flow rate.

7.3.2 ENGINE ACCELERATION WITH A MINIBUS

Data of a FORD TRANSIT minibus, taken from reference (103), are used to calculate the load dynamics. The experimental data used for comparison are the ones taken from the second type of test, i.e. engine acceleration with varied load.

The comparisons between the measured and predicted cylinder pressures, engine speeds, exhaust temperatures and smoke emissions are given in figures (7.37)-(7.41) respectively. It can be seen from these figures that the salient features of the comparisons are very similar to the ones discussed in section 7.3.1.

Figure (7.39) shows the engine response to a combined increase in load and demand speed. Since the change in demand speed can be executed faster than load,

the over-all response of engine speed is similar to the first type of test, shown in figure (7.25). This confirms the findings by Marzouk (78).

Compared with the first type of tests, engine acceleration with varied load produces a slower engine response. It takes about 3 seconds for the engine speed to increase from 1000 rpm to 2000 rpm, compared with 2 seconds from 1000 rpm to 2400 rpm in the first type of test. The percentage overshoot in the exhaust temperature is also smaller in the former case. This also applies to the peak pressure and the work by Marzouk (78) shows similar trends. Although the peak smoke emission is higher with increased load, the percentage overshooting tends to be lower than the one with zero load.

Plots of other variables predicted for the acceleration of an engine with a FORD TRANSIT minibus are presented in figures (7.42)-(7.49). The predicted movement of the governor CS is shown in figure (7.42) and it can be noticed that it takes a longer time for the CS to travel to the maximum compared with the one shown in figure (7.29). The predicted fuelling rate given in figure (7.43) shows smaller percentage increase compared with the one illustrated in figure (7.30). This trend is also observed in friction losses, shown in figure (7.45). The effects of transient on diesel engine exhaust emissions have been experimentally examined by Callahan et al (104) and their results show that both particulate and NO_x increase with accelerations of the engine. This finding further confirms the trends shown in figure (7.31) and (7.44). The start of injection, injection duration, air flow rate and overall equivalence ratio are shown in figures (7.46)-(7.49) respectively and the trends indicated by these figures are very similar to the ones shown in figures (7.33)-(7.36).

7.3.3 ENGINE SUDDEN LOAD ACCEPTANCE

This transient operation is triggered by the sudden movement of the load lever. Since a hydraulic dynamometer is used in the present work, it takes several seconds for the external load to increase from zero to the final value, dependent on the

desired level. The following results are presented with the external load increased from zero to medium (11 bars), taking about 2 seconds.

The comparisons between the measured and predicted cylinder pressures, engine speeds, exhaust temperatures and smoke emissions are given in figures (7.50)-(7.54) respectively. Since the salient features of the comparisons are very similar to the ones discussed in Section 7.3.1, they will not be discussed in detail here.

Compared with the first two types of tests, load acceptance produces a slower engine response, as shown in figure (7.52). It takes about 6 seconds to recover for an external load increased from zero to 11 bars and the engine speed drops from 2000 rpm to 1670 rpm. The percentage overshoot in the exhaust temperature and peak cylinder pressure are smaller. This also applies to the exhaust smoke emission.

Plots of other variables predicted for sudden load acceptance of the engine are presented in figures (7.55)- (7.62). The predicted movement of the governor CS is shown in figure (7.55) and it can be noticed that it takes longer time for the CS to travel to the maximum compared with the ones in the first two types of tests (shown in figures (7.29) and (7.42) respectively). This trend is also applied to the fuelling rate, NO_x emission, friction losses, the start of injection, injection duration, air flow rate and equivalence ratio, given in figures (7.56)-(7.62) respectively.

7.4 RESULTS FROM THE PARALLEL MODEL

In developing the parallel model, care was taken to minimise the communication time. However the simulation model running on three transputers is only twice as fast as the same model running on one transputer. This is due to large data transfer over limited communications and the limited parallelism in the problem.

It takes about 24 minutes to run the model for a 10 seconds transient period of the engine. The same problem takes several hours to execute on a medium speed mainframe, such as Honeywell Multics. This illustrates the benefit of the application of parallel processing.

A comparison of the performance of the parallel model executed with different number of transputers, and on conventional computers is given in Appendix D.

7.5 SUMMARY

At steady operating conditions, agreement between predicted and experimental cylinder pressures is generally good and effects of the changes in the engine operating conditions are accurately represented in the predictions.

At transient operating conditions, the experimental and predicted cylinder pressures, engine speeds, exhaust temperatures and smoke emissions agree favourably. Table 7.1 lists the comparison of the accuracies by the present model and several models reported in the literature ((40), (41) and (42)).

As discussed in the literature survey, one of the problems during transient operations is the sudden increase in smoke emission and the present model has been successful in predicting this trend.

Whilst small discrepancies exist, agreement between the predicted and experimental data is in general good, illustrating the applicability of the model to the investigation of engine performance and emission. Based on the concurrent process computing methods, the investigation can be carried out interactively. Areas of further research will be discussed in Chapter 8.

Table 7.1:

A comparison of the accuracies by the present model and several models reported in the literature ((40), (41) and (42)).

Models	Accuracy	Source
The present	14%	figures (7.25) & (7.35)
Goyal	16%	figure (1.9)
Watson and Marzouk	19%	figure (1.9)
Wallace and Way	25%	figure (1.9)

Fig. 7.1 Cylinder pressure diagram (fuel quantity = 28.1 mm³, engine speed = 1000 rpm, injection timing = 3.5 deg btdc)

Fig. 7.2 Cylinder pressure diagram (advanced injection timing: fuel quantity = 28.6 mm³, engine speed = 1000 rpm, injection timing = 7.3 deg btdc)

Fig. 7.3 Cylinder pressure diagram (retarded injection timing: fuel quantity = 28.2 mm³, engine speed = 1000 rpm, injection timing = 0.9 deg btdc)

Fig. 7.4 Cylinder pressure diagram (fuel quantity = 34.1 mm³, engine speed = 2000 rpm, injection timing = 3.7 deg btdc)

Fig. 7.5 Cylinder pressure diagram (advanced injection timing: fuel quantity = 34.0 mm^3 , engine speed = 2000 rpm, injection timing = 6.1 deg btdc)

Fig. 7.6 Cylinder pressure diagram (retarded injection timing: fuel quantity = 33.5 mm^3 , engine speed = 2000 rpm, injection timing = 0.5 deg btdc)

Fig. 7.7 Cylinder pressure diagram (fuel quantity = 35.5 mm^3 , engine speed = 3000 rpm, injection timing = 7.5 deg btdc)

Fig. 7.8 Cylinder pressure diagram (advanced injection timing: fuel quantity = 35.5 mm^3 , engine speed = 3000 rpm, injection timing = 10.5 deg btdc)

Fig. 7.9 Cylinder pressure diagram (retarded injection timing: fuel quantity = 35.2 mm^3 , engine speed = 3000 rpm, injection timing = 5.0 deg btdc)

Fig. 7.10 Cylinder pressure diagram (fuel quantity = 34.1 mm^3 , engine speed = 4000 rpm, injection timing = 11.3 deg btdc)

Fig. 7.11 Cylinder pressure diagram (advanced injection timing: fuel quantity = 33.9 mm^3 , engine speed = 4000 rpm, injection timing = 13.1 deg btdc)

Fig. 7.12 Cylinder pressure diagram (advanced injection timing: fuel quantity = 33.7 mm^3 , engine speed = 4000 rpm, injection timing = 8.5 deg btdc)

Fig. 7.13 Rate of heat release, accumulated heat release (fuel quantity = 28.1 mm^3 , engine speed = 1000 rpm, injection timing = 3.5 deg btdc)

Fig. 7.14 Mean temperature diagram (fuel quantity = 28.1 mm^3 , engine speed = 1000 rpm, injection timing = 3.5 deg btdc)

Fig. 7.15 Pre-mixed, diffusion and total fuel burning rate (fuel quantity = 28.1 mm^3 , engine speed = 1000 rpm, injection timing = 3.5 deg btdc)

Fig. 7.16 Fuel mass fractions Injected, evaporated and burned (fuel quantity = 28.1 mm^3 , engine speed = 1000 rpm, injection timing = 3.5 deg btdc)

Fig. 7.17 Air entrainment and fuel injection rates (fuel quantity = 28.1 mm^3 , engine speed = 1000 rpm, injection timing = 3.5 deg btdc)

Fig. 7.18 Heat flux diagram (fuel quantity = 28.1 mm^3 , engine speed = 1000 rpm, injection timing = 3.5 deg btdc)

Fig. 7.19 Nitric oxide concentrations, equilibrium and kinetic (fuel quantity = 28.1 mm^3 , engine speed = 1000 rpm, injection timing = 3.5 deg btdc)

Fig. 7.20 Soot formation and oxidation concentrations (fuel quantity = 28.1 mm^3 , engine speed = 1000 rpm, injection timing = 3.5 deg btdc)

Figure 7.21 Effects of Injection Timing

Figure 7.22 Effects of Engine Speed

Figure 7.23 Experimental Cylinder Pressure (Type I)

Figure 7.24 Predicted Cylinder Pressure (Type I)

Figure 7.25 Comparison of Predicted and Experimental Engine Speed Responses (Type I)

Figure 7.26 Comparison of Predicted and Experimental Exhaust Temperatures (Type I)

Figure 7.27 Comparison of Predicted and Experimental Exhaust Smoke Emissions (Unshifted) (Type I)

Figure 7.28 Comparison of Predicted and Experimental Exhaust Smoke Emissions (Shifted) (Type I)

Figure 7.29 Predicted Governor C-S Travel (Type I)

Figure 7.30 Predicted Fuelling Rate (Type I)

Figure 7.31 Predicted NOx Emission (Type I)

Figure 7.32 Predicted Engine Friction Losses (Type I)

Figure 7.33 Predicted Start of Injection (Type I)

Figure 7.34 Predicted Injection Duration (Type I)

Figure 7.35 Predicted Air Flow Rate (Type I)

Figure 7.36 Predicted Equivalence Ratio (Type I)

Figure 7.37 Experimental Cylinder Pressure (Type II)

Figure 7.38 Predicted Cylinder Pressure (Type II)

Figure 7.39 Comparison of Predicted and Experimental Engine Speed Responses (Type II)

Figure 7.40 Comparison of Predicted and Experimental Exhaust Temperatures (Type II)

Figure 7.41 Comparison of Predicted and Experimental Exhaust Smoke Emissions (Type II)

Figure 7.42 Predicted Governor C-S Travel (Type II)

Figure 7.43 Predicted Fuelling Rate (Type II)

Figure 7.44 Predicted NOx Emission (Type II)

Figure 7.45 Predicted Engine Friction Losses (Type II)

Figure 7.46 Predicted Start of Injection (Type II)

Figure 7.47 Predicted Injection Duration (Type II)

Figure 7.48 Predicted Air Flow Rate (Type II)

Figure 7.49 Predicted Equivalence Ratio (Type II)

Figure 7.50 Experimental Cylinder Pressure (Type III)

Figure 7.51 Predicted Cylinder Pressure (Type III)

Figure 7.52 Comparison of Predicted and Experimental Engine Speed Responses (Type III)

Figure 7.53 Comparison of Predicted and Experimental Exhaust Temperatures (Type III)

Figure 7.54 Comparison of Predicted and Experimental Exhaust Smoke Emissions (Type III)

Figure 7.55 Predicted Governor C-S Travel (Type III)

Figure 7.56 Predicted Fuelling Rate (Type III)

Figure 7.57 Predicted NOx Emission (Type III)

Figure 7.58 Predicted Engine Friction Losses (Type III)

Figure 7.59 Predicted Start of Injection (Type III)

Figure 7.60 Predicted Injection Duration (Type III)

Figure 7.61 Predicted Air Flow Rate (Type III)

Figure 7.62 Predicted Equivalence Ratio (Type III)

CHAPTER 8

CONCLUSION AND RECOMMENDATIONS FOR FUTURE WORK

8.1 DISCUSSION

The processing power of the CPU of any processor is limited and the role of parallel processing is more recognised for high speed computation. In fact, the new generation of supercomputers are all being built with multiple CPUs which require researchers to re-examine algorithms and consider problem domain decomposition (the parallelism in the problem).

Application of the parallel processing in the engine technology is still in its early stages and the present work approaches this aim by employing a relatively cheap solution, transputers.

By the application of concurrent process computing methods, the model can be implemented interactively to simulate the engine under transient operating conditions. Figure (8.1) shows the predicted cylinder pressure, governor control-spool movement, fuelling rate and engine speed response at different stages of a five second transient running, demonstrating the uniqueness of the model.

The various transient simulation models available in the literature have been discussed in Chapter 1. Comparisons of the present model with the outstanding work by Watson and Marzouk (40) and Jones and et al (45) are carried out below:

Comparison with the Work by Watson and Marzouk

The model by Watson and Marzouk (40) can predict the transient performance of a diesel engine with reasonable accuracy and has been used in industry for years. The two algebraic equations (2.25) and (2.26) postulated by them to account for the two distinct phases of the burning characteristics of a diesel engine (premixed and diffusion burning) formed the basis of the heat release evaluation in the present model. However, their model differs in two distinct aspects from the present work.

(1) Their model is sequential in nature.

(2) There are no detailed entrainment and turbulent mixing submodels in their work. Therefore, exhaust emissions under transient conditions have not been addressed. Although the later work by Watson (44) has attempted the soot emission during transient period, his correlation is purely empirical.

Comparison with the Work by Jones et al

The pioneering work of Jones et al (45) investigated the use of a multiprocessor simulator to study the transient behaviour of the diesel engine. However, their study differs in two important ways from the work presented here.

(1) Their computational system though having multiple processors (Motorola 68020) was limited by the 'system bus' for all interprocessor communication and access to system memory. This requires significant effort on programming to reduce contention for use of the system bus.

(2) No modelling of exhaust emissions was attempted. Combustion behaviour and pollutant formation during transient excursions of an engine are important and need to be addressed.

From the comparisons above, it is clear that the present model has made contribution towards the two goals emerged from the literature study, i.e. the needs

for detailed but inexpensive dynamic models and the needs for exhaust emission modelling during transient period.

8.2 CONCLUSION

An interactive model to simulate the diesel engine under both steady and transient conditions has been developed, based on a new approach - parallel processing. The engine, governor-pump and load dynamics submodels have been developed to study drive train transient behaviour. The simulation code has been successfully run in parallel on a concurrent computer which is based on four INMOS T800 transputers by using OCCAM as a harness for the simulation code written in FORTRAN. Furthermore graphics routines are also developed to present the results concurrently with the calculation which would not be possible with any sequential computer.

In their conclusion, Jones et al (105) pointed out that the future development on hardware for diesel engine modelling will focus on the application of the latest generation of chips, including the transputer. The present work forms a contribution towards achieving that goal.

Employing a two zone combustion model enables detailed description of the entrainment and turbulent mixing processes, hence prediction of exhaust emissions under transient conditions with economic use of computing time.

The four stroke, direct injection, swirl chamber diesel engine was considered. However, the model can cater for other types if simple modifications are introduced.

A comparison between the measured and the calculated behaviour of a FORD swirl chamber diesel engine shows good agreement for both steady and transient operating conditions.

The Possible Applications of the Model

Because of the advantages of being detailed, inexpensive and interactive, the model can be used in a wide range of applications, such as:

- (1) Investigation of engine transient performance and emissions.
- (2) Advanced diagnostics and parametric studies
- (3) Control system design, new modules can be added in to investigate the dynamics of different type of governors
- (4) Condition Monitoring, used as data bank
- (5) Computer aided teaching, used as data bank and visually demonstrating the operating principles of diesel engines

8.3 RECOMMENDATIONS FOR FUTURE WORK

As illustrated in section 7.2.4, the injection timing has strong effects on the predictions of combustion in diesel engines. The correlations for the start of injection and injection duration in this study (equation (2.82) and (2.83)) must be considered as preliminary in nature. A more thorough investigation for these is needed to improve the accuracy of prediction (e.g. relating the start of injection and injection duration to the cam profile of the engine being used and the cam dynamics).

In the present study, the calculation of ignition delay under transient conditions is based on the steady state data (section 2.2.1.2). The in-cylinder temperatures vary between steady state and transient operations. These variations will cause the ignition delay to be different between the two operating conditions, as illustrated in

figure (8.2) (French (106)). More work should be undertaken to investigate that to what extent the ignition delay under transient conditions can be represented by the steady state data.

The exhaust smoke measurements demonstrated the feasibility of the method, but further development work is needed for more accurate measurements, especially the period three to four seconds after the transient operation is started, by optimisation of the parameters such as the inner walls diameter and the air pressure.

The changes in the patterns of the heat transfer and combustion enable the model to predict the transient performance and emission of the indirect injection engine.

The harnesses developed can be used in some other applications such as producing an engine performance and emission maps like the ones shown in figure (8.3) and (8.4) (5). In fact, this is an ideal case for parallel processing. One processor can be used as the master which determines the number of running conditions, distributes the running conditions to a number of slave processors, waits for the results and presents them in graphics form, while the slave processors calculate the engine performance and emission for a number of running conditions and send the results back to the master. The more processors the system has, the faster the performance and emission maps can be produced. Again, detailed experimental data such as the start of injection, injection duration and ignition delay are needed for a wide range of running conditions (typically, A/F from 15 to 60 and engine speed from 1000 to 4000 RPM).

(a)

(b)

Figure 8.1 Predicted Cylinder Pressure, Governor Control Spool Movement, Fuelling Rate and Engine Speed Response at Different Stages of a Five Second Transient Running. (a-b)

(c)

(d)

Figure 8.1 Predicted Cylinder Pressure, Governor Control Spool Movement, Fuelling Rate and Engine Speed Response at Different Stages of a Five Second Transient Running. (c-d)

(e)

(f)

Figure 8.1 Predicted Cylinder Pressure, Governor Control Spool Movement, Fuelling Rate and Engine Speed Response at Different Stages of a Five Second Transient Running. (e-f)

(g)

(h)

Figure 8.1 Predicted Cylinder Pressure, Governor Control Spool Movement, Fuelling Rate and Engine Speed Response at Different Stages of a Five Second Transient Running. (g-h)

(i)

(j)

Figure 8.1 Predicted Cylinder Pressure, Governor Control Spool Movement, Fuelling Rate and Engine Speed Response at Different Stages of a Five Second Transient Running. (i-j)

(k)

Figure 8.1 Predicted Cylinder Pressure, Governor Control Spool Movement, Fuelling Rate and Engine Speed Response at Different Stages of a Five Second Transient Running. (k)

Figure 8.2 The Steady and Transient Results of the Start of Injection, End of Injection and Ignition Delay (106)

Figure 8.3 CO Emission Map (Vol%) (5)

Figure 8.4 NO Emission Map (ppm) (5)

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APPENDIX A

EXPERIMENTAL ENGINE AND VEHICLE DETAILS

The Vehicle details:

In the present studies, a FORD TRANSIT minibus is taken as the vehicle to be modelled. The details of the minibus is listed as follow:

Load (W)	2.0 tonne
Front area (A)	5.41 m ²

Coefficients in equations (2.94):

a	97.99 N/tonne
b	0.9799 Nh/tonne m
k	0.037 Nh/m ² km

Transmission:

Rolling radius of tyres, r	0.3175 m
Rolling reduction in rear axle, n _b	3.5:1
Gear reduction in gear box, n _g	

1st	3.96
2nd	2.28
3rd	1.41

Moment of inertia of the loaded minibus (neglecting rotational inertia of wheels and tyres), referred to the engine crankshaft:

$$IS=IE+W*r^2*(1/ng)^2*(1/nb)^2 \tag{A.1}$$

and for the second gear:

$$\begin{aligned} IS &= 0.28 + 2.0 * 0.3175^2 * (1/2.28)^2 * (1/3.5)^2 \\ &= 3.446 \text{ (kg-m}^2\text{)} \end{aligned} \tag{A.2}$$

The Details of the Engine:

Engine Type	Diesel, 4 cylinder, in-line OHV	
Bore	mm	93.67
Stroke	mm	90.54
Capacity	cc	2496
Compression ratio	19 : 1	
Connecting rod length	mm	154
Swirl ratio	2.06	
Number of holes of injector	4	
Orifice diameter of injector	mm ²	0.23
Inlet valve closure	ATDC	-141
Exhaust valve opening	ATDC	129
Volumetric efficiency	0.85	

APPENDIX B

OCCAM HARNESSSES AND CONFIGURATION PROGRAM

OCCAM HARNESSSES

OCCAM Harness for the Root Transputer

-----FIRST.OCC-----

PROC print (CHAN OF ANY from.link, to.link, to.root, from.root, from.t21,
to.t21, from.t3, to.t3)

-- Libraries used in this harness:

#USE "\toolset\flibs" -- File server library

#USE "\toolset\boardlib" -- Board support library

-- Import the non-OCCAM program:

#IMPORT "program (INT dummy, VAL INT flag, []INT ws1,
in, out, ws2) = 1"

-- Run time data option to be returned:

VAL option IS 0:

VAL f.ok IS 0:

-- Declaration of channels and variables:

[4]CHAN OF ANY from.prog.chan, to.prog.chan:

[1] INT ws2 :

[4] INT out.program, in.program:

[200000] INT ws1 :

[2] REAL32 number2 :

[2] REAL32 number3 :

[13] REAL32 number1 :

[24] REAL32 number :

INT result, screen.id,d, flag:

INT len, cont3, cont4 :

REAL32 ratio, ratio1, tr4 :

-- Processes start:

SEQ

-- Set flag value:

flag := 1

-- Reserve channels:

SEQ i = 0 FOR 4

SEQ

LOAD.INPUT.CHANNEL(in.program[i] , to.prog.chan[i])

LOAD.OUTPUT.CHANNEL(out.program[i], from.prog.chan[i])

-- Running io.handler and non-OCCAM program in parallel:

PAR

```
io.handler(from.link, to.link,
           [from.prog.chan FROM 0 FOR 2 ] ,
           [to.prog.chan  FROM 0 FOR 2 ] )
program(d, flag, ws1, in.program, out.program, ws2 )
```

-- Thecommunications (hos $\Leftrightarrow$ trout and root $\Leftrightarrow$ targets):

SEQ f = 0 FOR 100

SEQ

SEQ i = 0 FOR 24

SEQ

```
from.prog.chan[2] ? number[i]
from.root ! number[i]
to.t21 ! number[i]
to.t3 ! number[i]
```

SEQ

```
ratio := number[0]/2000.0(REAL32)
ratio1 := number[0]/500.0(REAL32)
cont3 := 1 + (INT TRUNC ((number[17]-number[14])/ratio))
cont4 := 1+(INTTRUNC (((number[16]+720.0(REAL32))
                     -number[17])/ratio1)).
```

SEQ j = 0 FOR cont3

SEQ

SEQ k = 0 FOR 13

SEQ

```
to.root ? number1[k]
to.prog.chan[2] ! number1[k]
```

SEQ m = 0 FOR 2

SEQ

```
from.t3 ? number3[m]
```

```
        to.prog.chan[2] ! number3[m]
SEQ i = 0 FOR cont4
    SEQ j = 0 FOR 2
        SEQ
            to.root ? number2[j]
            to.prog.chan[2] ! number2[j]
        SEQ
            to.root ? tr4
            to.prog.chan[2] ! tr4
    :
```

OCCAM Harness for the First Target Transputer

-----SECOND.OCC-----

```
PROC send1 (CHAN OF ANY from.root, to.root, to.t12, from.t12,
           to.t11, from.t11)
```

-- Libraries used in this harness:

```
#USE "\toolset\flibs"    -- File server library
```

```
#USE "\toolset\boardlib" -- Board support library
```

-- Import the non-OCCAM program:

```
#IMPORT "program (INT dummy, VAL INT flag, []INT ws1, in,
           out, ws2) = 1"
```

-- Run time data option to be returned:

VAL option IS 0:

-- Declaration of channels and variables:

```
[4]CHAN OF ANY from.prog.chan, to.prog.chan :
```

```
[1]    INT ws2 :
```

```
[200000] INT ws1 :
```

```
[4]INT out.program, in.program:
```

```
INT d, flag, result:
```

-- Processes start:

SEQ

-- Set flag value:

flag := 1

-- Reserve channels:

SEQ i = 0 FOR 4

SEQ

LOAD.INPUT.CHANNEL(in.program[i] , to.prog.chan[i])

LOAD.OUTPUT.CHANNEL(out.program[i], from.prog.chan[i])

-- The communication between non-OCCAM programs :

PAR

program(d, flag, ws1, in.program, out.program, ws2)

-- Declaration of channels and variables:

[2] REAL32 number3 :

[3] REAL32 number2 :

[13] REAL32 number1 :

[24] REAL32 number :

INT nwal, cont1, cont2, cont3, cont4 :

REAL32 ratio, ratio1, camig, var1, tr4 :

-- The communications (root $\Leftrightarrow$ target and target $\Leftrightarrow$ target):

SEQ f = 0 FOR 100

SEQ


```

SEQ i = 0 FOR 24
  SEQ
    from.root ? number[i]
    to.prog.chan[2] ! number[i]
SEQ
  ratio := number[0]/2000.00(REAL32)
  ratio1 := number[0]/500.00(REAL32)
  cont1 := 1 + (INT TRUNC (number[15]/ratio))
  cont3 := 1 + (INT TRUNC ((number[17]-number[14])/ratio))
  cont4 := 1 + (INT TRUNC (((number[16]+720.0(REAL32))
    -number[17])/ratio1))
  cont2 := cont3 - cont1
SEQ n = 0 FOR cont1
  SEQ
    SEQ m = 0 FOR 3
      SEQ
        to.t12 ? number2[m]
        to.prog.chan[2] ! number2[m]
      SEQ
        to.t12 ? nwal
        to.prog.chan[2] ! nwal
    SEQ k = 0 FOR 13
      SEQ
        from.prog.chan[2] ? number1[k]
        to.root ! number1[k]
        from.t11 ! number1[k]
  SEQ j = 0 FOR cont2
    SEQ
      SEQ k = 0 FOR 13
        SEQ
          from.prog.chan[2] ? number1[k]
          to.root ! number1[k]
          from.t11 ! number1[k]

```

```
SEQ i = 0 FOR cont4
  SEQ j = 0 FOR 2
    SEQ
      from.prog.chan[2] ? number3[j]
      to.root ! number3[j]
  SEQ
    from.prog.chan[2] ? tr4
    to.root ! tr4
```

:

OCCAM Harness for the Second Target Transputer

-----THIRD.OCC-----

```
PROC send2 (CHAN OF ANY from.t12, to.t12, to.t22, from.t22,
           to.t21, from.t21)
```

```
-- Libraries used in this harness:
```

```
#USE "\toolset\flibs"    -- File server library
```

```
#USE "\toolset\boardlib" -- Board support library
```

```
-- Import the non-OCCAM program:
```

```
#IMPORT "program (INT dummy, VAL INT flag, []INT ws1, in,
           out, ws2) = 1"
```

```
-- Run time data option to be returned:
```

```
VAL option IS 0:
```

```
-- Declaration of channels and variables:
```

```
[4]CHAN OF ANY from.prog.chan, to.prog.chan :
```

```
[1]    INT ws2 :
```

```
[4]    INT out.program, in.program:
```

```
[200000] INT ws1 :
```

```
INT d, flag, result:
```

-- Processes start:

SEQ

-- Set flag value:

flag := 1

-- Reserve channels:

SEQ i = 0 FOR 4

SEQ

LOAD.INPUT.CHANNEL(in.program[i] , to.prog.chan[i])

LOAD.OUTPUT.CHANNEL(out.program[i], from.prog.chan[i])

-- The communication between non-OCCAM programs :

PAR

program(d, flag, ws1, in.program, out.program, ws2)

-- Declaration of channels and variables:

[3] REAL32 number1 :

[24] REAL32 number :

INT nwal, cont1 :

REAL32 ratio :

-- Thecommunications (target $\leftrightarrow$ target):

SEQ i = 0 FOR 100

SEQ

SEQ i = 0 FOR 24

SEQ

to.t21 ? number[i]

to.prog.chan[2] ! number[i]

SEQ

ratio := number[0]/2000.00(REAL32)

cont1 := 1 + (INT TRUNC (number[15]/ratio))

SEQ j = 0 FOR cont1

SEQ

SEQ k = 0 FOR 3

SEQ

from.prog.chan[2] ? number1[k]

to.t12 ! number1[k]

SEQ

from.prog.chan[2] ? nwal

to.t12 ! nwal

:

OCCAM Harness for the Third Target Transputer

-----FOURTH.OCC-----

```
PROC send3 (CHAN OF ANY from.t22, to.t22, to.t3, from.t3,
           from.t11, to.t11)
```

-- Libraries used in this harness:

```
#USE "\toolset\flibs"    -- File server library
```

```
#USE "\toolset\boardlib" -- Board support library
```

-- Import the non-OCCAM program:

```
#IMPORT "program (INT dummy, VAL INT flag, []INT ws1, in,
           out, ws2) = 1"
```

-- Run time data option to be returned:

VAL option IS 0:

-- Declaration of channels and variables:

```
[4]CHAN OF ANY from.prog.chan, to.prog.chan:
```

```
[1]    INT ws2 :
```

```
[4]    INT out.program, in.program:
```

```
[200000] INT ws1 :
```

```
[2] REAL32 number1 :
```

```
[13] REAL32 number :
```

```
[24] REAL32 number2 :
```

INT result, screen.id,d, flag, len, cont3:

REAL32 ratio :

-- Processes start:

SEQ

-- Set flag value:

flag := 1

-- Reserve channels:

SEQ i = 0 FOR 4

SEQ

LOAD.INPUT.CHANNEL(in.program[i] , to.prog.chan[i])

LOAD.OUTPUT.CHANNEL(out.program[i], from.prog.chan[i])

-- The communication between non-OCCAM programs :

PAR

program(d, flag, ws1, in.program, out.program, ws2)

-- The communications (root $\leftrightarrow$ target and target $\leftrightarrow$ target):

SEQ f = 0 FOR 100

SEQ

SEQ i = 0 FOR 24

to.t3 ? number2[i]

SEQ

ratio := number2[0]/2000.0(REAL32)

cont3 := 1 + (INT TRUNC ((number2[17]-number2[14])/ratio))

SEQ i = 0 FOR cont3

SEQ

SEQ k = 0 FOR 13

SEQ

from.t11 ? number[k]

to.prog.chan[2] ! number[k]

SEQ j = 0 FOR 2

SEQ

from.prog.chan[2] ? number1[j]

from.t3 ! number1[j]

:

THE CONFIGURATION PROGRAM

-----MAIN1.OCC-----

-- Define the separate compiled programs:

#SC "first"

#SC "second"

#SC "third"

#SC "fourth"

-- Declaration of channels:

CHAN OF ANY from.link, to.link, from.root, to.root, from.t11,
 to.t11, from.t12, to.t12, from.t21, to.t21, from.t22,
 to.t22, from.t3, to.t3 :

-- Hardware layout of the network and allocation of separate
 complied programs onto it:

PLACED PAR

PROCESSOR 0 T8

PLACE from.t3 AT 7:

PLACE to.t3 AT 3:

PLACE to.root AT 6:

PLACE from.root AT 2:

PLACE from.t21 AT 5:

PLACE to.t21 AT 1:

PLACE from.link AT 4:

PLACE to.link AT 0:

print (from.link,to.link,to.root,from.root,from.t21,to.t21, from.t3,to.t3).

PROCESSOR 1 T8

PLACE from.root AT 7:

PLACE to.root AT 3:

PLACE to.t12 AT 6:

PLACE from.t12 AT 2:

PLACE to.t11 AT 5:

PLACE from.t11 AT 1:

send1 (from.root,to.root,to.t12,from.t12,to.t11,from.t11)

PROCESSOR 2 T8

PLACE from.t12 AT 7:

PLACE to.t12 AT 3:

PLACE to.t22 AT 6:

PLACE from.t22 AT 2:

PLACE to.t21 AT 5:

PLACE from.t21 AT 1:

send2 (from.t12,to.t12,to.t22,from.t22,to.t21,from.t21)

PROCESSOR 3 T8

PLACE from.t22 AT 7:

PLACE to.t22 AT 3:

PLACE to.t3 AT 6:

PLACE from.t3 AT 2:

PLACE from.t11 AT 5:

PLACE to.t11 AT 1:

send3 (from.t22,to.t22,to.t3,from.t3,from.t11,to.t11)

APPENDIX C

COMMANDS TO COMPILE, LINK, CONFIGURE AND EXECUTE THE MODEL

COMMANDS TO COMPILE THE PROGRAMS

To Compile the FORTRAN Programs, Type the Following Commands:

T8F GRAPH/S/T8A
T8F COMBT/S/T8A
T8F ENTR/S/T8A
T8F EMIST/S/T8A

To Compile the OCCAM Programs, Type the Following Commands:

TOFIR
TOSEC
TOTH
TOFO

-----TOFIR.BAT-----
AFSERVER -:E -:B \TOOLSET\OCCAM.B4
AFSERVER FIRST.OCC/M/T8

```
-----TOSEC.BAT-----
AFSERVER -:E -:B \TOOLSET\OCCAM.B4
AFSERVER SECOND.OCC/M/T8
-----
```

```
-----TOTH.BAT-----
AFSERVER -:E -:B \TOOLSET\OCCAM.B4
AFSERVER THIRD.OCC/M/T8
-----
```

```
-----TOFO.BAT-----
AFSERVER -:E -:B \TOOLSET\OCCAM.B4
AFSERVER FOURTH.OCC/M/T8
-----
```

COMMANDS TO LINK THE PROGRAMS

```
-----
LINK1
LINK2
LINK3
LINK4
-----
```

```
-----LINK1.BAT-----
LINKT (PROGRAM:GRAPH.BIN+\TF2V0\FRTL8.BIN+\TF2V0\
      T8HARN.BIN)+(:@FIRST.LNK),FIRST.LSC /C /M /"PRINT"
-----
```

```
-----LINK2.BAT-----
LINKT (PROGRAM:COMBT.BIN+\TF2V0\SAFRTL8.BIN)+
      (:@SECOND.LNK),SECOND.LSC /C /M /"SEND1"
-----
```

-----LINK3.BAT-----
LINKT (PROGRAM:ENTR.BIN+\TF2V0SAFRTL8.BIN)+
(:@THIRD.LNK),THIRD.LSC /C /M /"SEND2"

-----LINK4.BAT-----
LINKT (PROGRAM:EMIST.BIN+\TF2V0SAFRTL8.BIN)+
(:@FOURTH.LNK),FOURTH.LSC /C /M /"SEND3"

COMMAND TO CONFIGURE THE MODEL

CONFIG MAIN1

COMMAND TO EXECUTE THE MODEL

AFSERVER -:B MAIN1.BT -:O 1

APPENDIX D

A COMPARISON OF COMPUTER RUN TIMES OF THE MODEL, ON A SELECTION OF COMPUTERS

The performance of the model run in parallel on differing numbers of transputers and run sequentially on conventional computers are listed below. The comparison is based on the CPU time, and excludes the effects of multi-user operation on the conventional computers.

Table D1: A comparison of the CPU times for the model to run for ten seconds transient on a selection of computers.

Hardware	CPU Time	CPU/Real Time
VICTOR VPC III 286	18799 sec	1880 (IBM-PC AT compatible)
Honeywell Multics	5586 sec	558
Hewlett-packard	398 sec	40
One Transputer	2792 sec	279
Two Transputer Network	1956 sec	195
Three Transputer Network	1424 sec	142