ASAE REPORT NO. 4

THE COLLEGE OF AERONAUTICS

CRANFIELD

THE DIGITAL SIMULATION OF A TURBO-CHARGED DIESEL ENGINE

by

H. D. Shroff, and D. Hodgetts

1403063855

ASAE Report No. 4

ADVANCED SCHOOL OF AUTOMOBILE ENGINEERING

CRANFIELD

The digital simulation of a turbo-charged diesel engine

by

H. D. Shroff, B. Tech. (Mech), D. Au. E.

and

D. Hodgetts, B. Sc., B. Sc. (Eng)., M. I. Mech. E.

SUMMARY

A mathematical simulation of a six cylinder four stroke water cooled diesel engine is described and then used to study the changes in performance caused by variations of engine paramters and operating conditions. The results are discussed with reference to the mathematical model employed and the physical system.

A limited study, to demonstrate the applicability of the simulation, considers the optimisation of a variable geometry system to achieve a performance target.

CONTENTS

ACKNOWLEDGMENTS

The authors are grateful to the Worshipful Company of Coachmakers and Harnessmakers for the award of a scholarship which enabled one of the authors, Mr. H. D. Shroff, to follow part of the research reported here.

Because Mr. Shroff is registered for a higher degree in the subject of this report, the second author, Mr. D. Hodgetts, wishes it to be known that his contribution has been that of supervision of Mr. Shroff.

LIST OF SYMBOLS AND ABBREVIATIONS

A	Area, ft^2
B. D. C.	Bottom dead centre
C	Specific heat, ft. lb_f lbm^{-1} F^{o-1}
C. A.	Crank angle
D	Cylinder bore, ft.
E	Exponent to base ten
FLEAK	Area parameter, ft^2
g_o	Gravitational constant
h	Enthalpy, ft. lb_f lbm^{-1}
k	Ratio of specific heats, $\frac{C_p}{C_v}$
M	Mass, lbm
$\dot{M}$	Rate of mass flow, lbm sec^{-1}
n	Ratio of connecting rod length/crank throw
P	Pressure, lb_f ft^{-2} abs
PINJ	Period for combustion, crank angle degrees
Q	Net energy loss or gain by system due to heat transfer and combustion, ft. lb_f.
q	Rate of heat transfer, ft. lb_f sec^{-1}
R	Gas constant, ft. lb_f. lbm^{-1} F^{o-1}
S	Stroke of piston, ft.
T	Temperature, ^{o} Rankine
t	Time, sec.
T. D. C.	Top dead centre
TRPM	Turbocharger speed, r. p. m.
TINJ	Injection timing, degrees C. A.
TFC	Constant for turbocharger bearing friction law
TFEXP	Exponent for turbocharger bearing friction law
V	Volume, ft^3
W COMP	Compressor work, ft. lb_f $cycle^{-1}$ $cylinder^{-1}$
W TURB	Turbine work, ft. lb_f. $cycle^{-1}$ $cylinder^{-1}$
X FLOW	Constant for pressure drop in exhaust pipe
θ	Crank angle, degrees
ν	Velocity, ft. sec^{-1}

Subscripts

c	control volume
d	downstream
e	effective
i	incoming
o	outgoing
p	constant pressure
u	upstream
v	constant volume
w	wall or surface

Introduction

After many years of almost continuous development of the reciprocating engine it is probable that the rate of further progress will depend upon a fundamental understanding of its combustion and gas flow processes. If the engine is coupled with a gas turbine, as with a turbo-charged or compound engine, the need for an adequate knowledge of the complex interrelation between the components and processes of the system is evident. Should the system have the additional complexities of variable geometry whereby an attempt is made to optimise the effective size of the components or timing of the events according to the operating requirements of the engine, an adequate assessment of the design by simple analyses is impossible and the cost of a full experimental evaluation is prohibitive.

The use of digital simulations as aids to the understanding and solution of complex design problems is well established. In the case of the turbo-charged diesel engine the mathematical models for the processes are rarely exact and depend often upon empirical data. Nevertheless, although the estimates of the system performance may be doubtful in an absolute sense, they can be invaluable for an assessment of the relative effects of the parameters and variables of that system.

It is the purpose of the research at the Advanced School of Automobile Engineering to improve the mathematical models of the processes and to develop digital programmes that will allow full optimisation of the design of turbo-charged or compund engines. It is the intention that the results of the analyses will be checked wherever possible with the performance of real engines. The development of these programmes is a continuous process and a current activity at the School. Presented here is the progress to-date.

Description of digital simulation

Written in the A.S.A. Fortran code, the programme simulates a multi-cylinder four-stroke water-cooled turbo-charged diesel engine with double entry turbine.

The analysis of the flow is based upon a set of control volumes that represent the engine cylinders and manifolds. In the case of a six cylinder engine with a double entry turbine, for example, there are nine control volumes - six to represent the cylinders, one volume for the induction manifold and two volumes for the exhaust manifolds, see FIG. 1. Within each control volume the state of the gas is calculated according to the unsteady flow energy equation and the law of conservation of mass. At the boundaries of the control volumes the heat transfer is estimated by empirical equations and the mass transfer is calculated according to the flow characteristics of the valves, the compressor or the turbine as appropriate.

It follows that for each control volume there are two basic first order differential equations with non-linear coefficients and five auxiliary rate equations to account for the mass inflow, mass outflow, heat transfer, change of volume and energy relaease by combustion of the fuel. Nine control volumes require the simultaneous solution of eighteen (18) basic equations and forty-five(45) auxiliary equations. A fourth order Runge-Kutta integration routine is used to give the change of state of the gases in the control volumes. Trial runs were made to establish the largest integration step length that could be used for consistent results and a stable solution. The step length required cannot exceed the order of one to two crank angle degrees in the critical parts of the cycle.

A full four stroke cycle requires that the integration is taken over seven hundred and twenty degrees of crank angle (720^{o}). With the assumptions that the cylinders are identical and that the processes are truly periodic and similar for each cylinder, the integration period is reduced to one hundred and twenty degrees of crank angle (120^{o}). An integration over the longer period requires the availability of reliable information to quantify the departure from these assumptions.

Wave effects in the manifolds are ignored. Since the research is directed towards power units that are suitable for road vehicles, the manifold dimensions are small and wave effects are insignificant at low or medium speeds of the eingine crankshaft. At high engine speeds where they may become significant, the error due to their omission is likely to be swamped by other errors that arise as a consequence of lack of knowledge of factors such as the flow characteristics of a radial turbine when subject to unsteady flow, unsteady heat transfer in the cylinders and manifolds, and the combustion processes in the cylinder.

A flow diagram, FIG. 2, illustrates the method of computation. Starting values for the turbo-charger speed and the pressures and temperatures in the control volumes are guessed or estimated and supplied with the data at the start of the run. The end values of a cycle are used as starting values for the next cycle until the closing error is less than one percent of the absolute temperatures and pressures in the control volumes. An exit path from the loop is provided should the rate of convergence be unsatisfactory. At the end of the iteration, the balance between the turbine work and the work of the compressor and bearing friction is tested. If the balance is not achieved, a new turbo-charger speed is chosen according to Newton's Rule and the cycle is repeated.

Programme input

Seven input streams of data are required as follows.

1. Starting values

   Turbo-charger speed and pressures and temperatures in the control volumes at the start of the computation.

2. Table of gas properties

   The specific heat at constant volume, C_V, and the ratio of the specific heats, $\frac{C_p}{C_v}$, are stored as a function of temperature in a TABLE format - described in Appendix IV.

3. Table of valve flow areas

   For a poppet valve the area of gas flow, the annular space between the valve edge and the seat, is reduced in effect by wall drag and boundary layer build-up on the valve sides. The effective area of flow at any instant is considered to be a function of the valve size, the valve lift, and an instantaneous coefficient of discharge, as defined by Kastner et. al. (reference 2), which depends upon the valve lift to diameter ratio and, to a lesser extent, on the pressure drop across the valve. By neglecting the effect of pressure drop on the instantaneous coefficient of discharge, the available area for gas flow at any instant can be computed from a knowledge of the valve geometry and its lift. This area is stored in a TABLE format for both the inlet valve and the exhaust valve as a function of crank angle.

4. Table of heat release

The programme requires a present 'heat' or energy release pattern to simulate the energy released by combustion. This is read as a TABLE of non-dimensional heat release rates over a unit period. By this means the total heat release and the period for heat release can be specified according to the engine requirements. The form chosen for the purpose of this analysis is illustrated by FIG. 3.

5. Table of compressor characteristics

The steady state non-dimensional mass flow ($\dot{M}\sqrt{T}/P$) and the adiabatic efficiency are stored as functions of the pressure ratio and the non-dimensional speed ($TRPM/\sqrt{T}$). Scale factors are read with the engine parameters to describe the size of the compressor.

6. Table of turbine characteristics

The steady-state characteristics are stored similarly to those of the compressor.

7. Engine and test parameters

The information describes the geometry of the engine and the speed and load conditions of a particular test. Full details are given in Appendix II.

Programme output

When the work balance is satisfactory and the iteration is complete, the major parameters of the engine, as described in the input, are printed for reference purposes. These are followed by a complete record of the pressures, temperatures, heat release, heat transfer, mass inflow and mass outflow for each control volume at crank angles throughout the cycle. Finally, the aggregated results such as indicated mean effective pressure (I. M. E. P.), indicated horse power (I. H. P.), indicated thermal efficiency, total heat losses to coolent, and the air-fuel ratio are printed.

Effect of system parameters and variables

To test the digital programme and to examine the effect of various parameters and variables, a six cylinder A. E. C. AV 410 turbo-charged four stroke diesel engine was simulated. The C. A. V. type 12 turbo-charger has a twin entry turbine with a 70^{o} nozzle ring. Test results for the engine and characteristics of the turbo-charger were provided by the Motor Industry Research Association (M. I. R. A.)

The dimensions of the engine are as follows:-

Bore	4.13 in.
Stroke	5.12 in.
Compression Ratio	18 : 1
Connecting rod length/crank ratio	3.57
Inlet valve opens at	10^{o} B. T. D. C.
Inlet valve closes at	38^{o} A. B. D. C.
Exhaust valve opens at	40^{o} B. B. D. C.
Exhaust valve closes at	20^{o} A. T. D. C.
Valve seat angle	40^{o}
Inlet valve diameter	1.525 in.
Exhaust valve diameter	1.275 in.
Inlet cam lift	0.3232 in.
Exhaust cam lift	0.3232 in.
Inlet rocker ratio	1.575 : 1
Exhaust rocker ratio	1.575 : 1
Inlet manifold volume	410 in^3
Exhaust manifold volume	56.4 in^3

Two system parameters and six operating variables were studied. These will be considered in turn.

(1) X FLOW

X FLOW is a scaled parameter that allows for a small pressure drop in the turbine exhaust pipe, according to the equation

$$\Delta p = X\ FLOW\ (1000\dot{M})^2$$

where Δp is the pressure drop $lb_f. ft^{-2}$

and $\dot{M}$ is the mass flow rate lbm sec^{-1}

Table 1 shows that the effect of the parameter on the performance of the engine is small. The reason for the interest in this parameter is that the experimental recording of the pressure in the exhaust manifold, FIG. 5, shows a secondary peak that occurs approximately at the time when the other half of the turbine is exhausting. Since this timing is independent of engine speed it is not a wave effect. FIG. 4 shows that the restriction of the turbine exhaust pipe and increase of the back pressure causes a rise at the time of the secondary peak. This effect is small, however, and insufficient to explain the experimental results.

TABLE I

EFFECT OF X FLOW

Engine Speed = 1800 R. P. M.
Engine Heat = 1800 ft. lb_f/Cylinder/Cycle
TINJ = 360° C. A.
PINJ = 60 C. A. degrees.

X Flow	T RPM R. P. M.	W TURB ft. lb_f.	W COMP ft. lb_f.	AIR MASS Flow lbm	PEAK EXHAUST PRESSURE lb_f in^{-2} abs.
0	39111	62	28	2.830E-3	26.558
500	38678	60	27	2.814E-3	26.625
1000	38678	60	27	2.812E-3	26.758
1500	38678	59	27	2.810E-3	26.885
1800	38678	58	27	2.800E-3	26.918

NOTE: For all tabulations, the timing of events is relative to T. D. C. at the beginning of the induction stroke and the work and mass quantities refer to one cylinder taken over one cycle.

(2) FLEAK

The parameter FLEAK is based on a hypothesis that attempts to explain the secondary peak in the exhaust manifold pressure. It represents a leakage area between the two manifolds of a twin entry turbine and allows a mass transfer between them when they are operating at different pressure levels. How this leakage occurs in the physical system is not clear.

FIG. 5 shows that the leakage could have a considerable effect on the variation of pressure in the exhaust manifold. Bearing in mind that the unsteady state characteristics of the turbine are unknown, the leakage indicates a possible reason for the secondary peak.

TABLE 2 suggests that the effect of leakage on the air mass flow through the engine is significant but small. However, care is required in the interpretation of these results because, in this case, the considerable loss of turbine power, is offset by the reduction of bearing friction as the speed of the turbo-charger is reduced. At high boost pressures the loss of power from the turbine would be much more significant and, therefore, the leakage could have a very detrimental effect on the overall performance of the engine.

TABLE 2

EFFECT OF FLEAK

Engine Speed = 1400 R. P. M.
Engine Heat = 1200 ft. lb_f/Cylinder/Cycle
TINJ = 355° C. A.
PINJ = 60 C. A. degrees

TABLE 2 (continued)

FLEAK	T RPM R. P. M.	W TURB ft. lb_f	W COMP ft. lb_f	AIR MASS FLOW lbm	PEAK EXHAUST PRESSURE lb_f in^{-2} abs.
0. 00010	33150	34. 149	23. 895	3. 060E-3	21. 576
0. 00050	30808	28. 991	20. 141	3. 000E-3	20. 797
0. 00100	29709	26. 278	18. 466	2. 970E-3	20. 238
0. 00150	29150	24. 837	17. 293	2. 950E-3	19. 857
0. 00200	28676	24. 057	16. 608	2. 930E-3	19. 568

(3) Combustion timing (TINJ)

The results obtained with three different valves of combustion timing are displayed in TABLE 3.

Early combustion results in higher peak pressures and temperatures, an overall increase in the indicated thermal efficiency and a decrease in the exhaust gas losses. The consequence of the reduction of the exhaust gas energy is a decrease in the turbine work and a reduction of the amount of air aspirated by the engine. The net effect, however, is an increase of power with earlier combustion but, as experience would suggest, there is a small return in power and efficiency compared with the increase of mechanical and thermal loading of the engine.

TABLE 3

EFFECT OF COMBUSTION TIMING (TINJ)

Engine Speed = 1400 R. P. M.
Engine Heat = 1200 ft. lb_f/Cylinder/Cycle
PINJ = 60 C. A. degrees.

TINJ ^{o}C. A.	355^{o}	360	365
IMEP lb_f. in^{-2}	107. 0	103. 5	98. 4
W TURB ft. lb_f	24. 53	26. 49	28. 61
W COMP ft. lb_f	17. 073	18. 639	20. 003
AIR MASS FLOW lbm.	2. 938E-3	2. 967E-3	2. 998E-3
PEAK TEMP ^{o}R	3094	2978	2889
PEAK PRESSURE lb_f. in^{-2} abs	1126	930	877
EXHUAST GAS LOSSES ft. lb_f	439	463	493
INDICATED THERMAL EFFICIENCY	50. 80%	49. 15%	46. 73%

(4) Valve size

The valve sizes were altered without altering the cylinder bore so that an increase in the size of one valve required a decrease in the other. The results, tabulated in TABLE 4, suggest that for a given cylinder bore there is an optimum size for the valves and that any change from this optimum will result in a reduction of the air aspirated by the engine.

TABLE 4

EFFECT OF VALVE SIZE

Engine Speed = 1400 R. P. M.
Engine Heat = 1200 ft. lb_f/Cylinder/Cycle
TINJ = 355° C. A.
PINJ = 60 C. A. degrees

VALVE SIZES	AIR MASS FLOW lb_m
Inlet = 1.25 Normal Exhaust = 0.70 Normal	2.890E-3
Inlet = 1.0 Normal Exhaust = 1.0 Normal	2.938E-3
Inlet = 0.70 Normal Exhaust = 1.25 Normal	2.910E-3

(5) Turbo-charger friction

The results of varying the turbo-charger bearing friction are illustrated in TABLE 5. The bearing losses are represented by means of a power law of the form

$$\text{Bearing horse-power} = \text{TFC} \times \left(\frac{\text{TRPM}}{1000}\right)^{\text{TFEXP}}$$

Different bearing have markedly different drag losses and they affect considerably the equilibrium speed of the turbo-charger. FIG. 6 illustrates the different matching speeds for three types of bearing.

As the bearing losses are reduced the turbo-charger tends to run at higher equilibrium speeds. This results in an increase in the boost pressure and a consequent rise in the air aspirated by the engine. For example, for the case considered, a change from a plain bush to a floating bush bearing gives a 12% increase in the air delivered to the engine and a power boost of the same order.

TABLE 5

EFFECT OF TURBO-CHARGER BEARING FRICTION

Engine Speed = 1400 R.P.M.
Engine Heat = 1200 ft. lb_f/Cylinder/Cycle
TINJ = 355° C.A.
PINJ = 60 C.A. degrees

BEARING TYPE AND LAW B.H.P. = TFC(TRPM/1000)^{TFEXP}	W TURB ft. lb_f	W COMP ft. lb_f	TRPM R.P.M.	AIR MASS FLOW lbm.
Plain Bush TFC = 2.185E-4 TFEXP = 2.76	28.0	15.0	26800	2.849E-3
Ball Bearing TFC = 7.05E-4 TFEXP = 2.250	32.0	19.5	31000	2.970E-3
Floating Bush TFC = 2.08E-4 TFEXP = 2.675	42.0	33.5	38750	3.184E-3
TFC = 0.0 TFEXP = 0.0	52.50	52.5	45000	
TFC = 2.47E-4 TFEXP = 2.25	27.57	22.71	32451	3.032E-3
TFC = 4.94E-4 TFEXP = 2.25	24.53	17.07	28983	2.938E-3
TFX = 9.88E-4 TFEXP = 2.25	20.05	10.23	23793	2.800E-3

(6) Heat transfer

The heat transfer that takes place between the cylinder gases and the surfaces of the combustion chamber and the cylinder wall is difficult to predict and it is usual to fall back on empirical methods. Several workers have proposed empirical and semi-empirical formulae to calculate the instantaneous gas side heat transfer coefficient. The formula used in this simulation is a modified form of Eichelberg's equation.

From a study of heat transfer in large slow speed diesel engines. Eichelberg postulated an equation of the form:-

$$\frac{q}{A_w} = \text{Constant}\,(\mathbf{v})^{\frac{1}{3}}\,(PT)^{\frac{1}{2}}\,(T - Tw)$$

For the purpose of this study, the velocity index has been increased from $\frac{1}{3}$ to 0.6

and the dimensional constant chosen to give a suitable overall heat loss at an engine speed of 1000 r.p.m.

The instantaneous heat transfer rate can be altered either by changing the Eichelberg constant or by altering the value of the mean wall temperature. Tests were conducted for three different values of this constant (TABLE 6).

An increase of heat transfer has a two-fold effect on the engine cycle. It leads to a significant drop in the thermal efficiency of the engine, a reduction of the energy lost to the exhaust gases and, consequently, a reduction of the turbine work and the air aspirated by the engine.

TABLE 6

EFFECT OF HEAT TRANSFER

Engine Speed = 1400 R.P.M.
Engine Heat = 1200 ft. lb_f/Cylinder/Cycle
TINJ = 355° C.A.
PINJ = 60 C.A. degrees

EICHELBERG'S CONSTANT	IMEP $lb_f in^{-2}$	W TURB ft. lb_f	AIR MASS FLOW lbm	JACKET LOSSES ft. lb_f	EXHAUST GAS ft. lb_f	INDICATED THERMAL EFFICIENCY
0.000600	97.8	18.50	2.788E-3	277.44	367	46.46%
0.000300	107.0	24.50	2.938E-3	152.80	439	50.80%
0.000148	112.0	28.44	3.019E-3	79.51	483	53.22%

(7) Exhaust manifold volume

The effect of changing the size of the exhaust manifold is illustrated in TABLE 7. Its effect on the shape of the pressure pulse in the exhaust manifold is illustrated by FIG. 7.

A change in the exhaust manifold volume has a significant effect on the power output of the turbine. As the manifold volume is increased and the exhaust pressure pulse is attenuated, the exhaust system tends towards a constant pressure type and the turbine output falls. These results tend to support previous findings of other investigators that, for low pressure charging, a pulsating exhaust system is more efficient that one at constant pressure.

TABLE 7

EFFECT OF EXHAUST MANIFOLD VOLUME

Engine Speed = 1400 R.P.M.
Engine Heat = 1200 ft. lb_f/Cylinder/Cycle
TINJ = 355° C.A.
PINJ = 60 C.A. degrees

EXHAUST MANIFOLD VOLUME ft^3	IMEP $lb. in^{-2}$	W TURB ft. lb_f	AIR MASS FLOW lbm
0.0100	107.0	24.86	2.940E-3
0.0153	107.0	24.53	2.938E-3
0.1530	106.8	21.55	2.895E-3

(8) Combustion period (PINJ)

TABLE 8 gives the results for various lengths of the combustion period. Late burning results in a lower engine efficiency and an increase in the energy to the exhaust gases. The increase of turbine power results in a small increase in the air delivered to the engine but this is insufficient to counteract the power lost as a consequence of the fall of the indicated thermal efficiency.

TABLE 8

EFFECT OF COMBUSTION PERIOD

Engine Speed = 1400 R.P.M.
Engine Heat = 1200 ft. lb_f/Cylinder/Cycle
TINJ = 355° C.A.

PINJ C.A. DEGREES	IMEP $lb_f.in^{-2}$	W TURB ft. lb_f	AIR MASS FLOW lbm	EXHAUST GAS LOSSES ft. lb_f	INDICATED THERMAL EFFICIENCY
50	109.5	23.58	2.9255E-3	424	51.9%
55	108.4	23.94	2.9260E-3	432	51.4%
60	107.0	24.53	2.9380E-3	439	50.8%
65	106.0	25.15	2.9440E-3	446	50.3%
70	104.8	25.83	2.9540E-3	456	49.7%
75	103.3	26.22	2.9550E-3	464	49.0%

Application of the digital simulation

For the purpose of demonstrating the application of the simulation to a real problem it is assumed that there is an interest in the use of a variable geometry system to achieve a high brake mean effective pressure of about 250 lb_f in^{-2} abs. at a speed of about 1600 crankshaft r.p.m. and a b.m.e.p. of about 180 lb_f in^{-2} at a speed of about 2800 crankshaft r.p.m. These were regarded as desirable characteristics for a particular road vehicle. It was also specified that the peak cylinder pressure should not greatly exceed 1600 $lb_f.in^{-2}$ abs. nor should the air-fuel ratio be less than 26 to 1. After-cooling is assumed to reduce the charge temperature to about 700°R.

Four results based upon the A.E.C. AV 410 engine are tabulated in TABLE 9. Case 1 shows that the lower speed requirement can only be met with a very high boost ratio. Limited to a more reasonable figure of about 3 to 1 boost ratio, the target figures cannot be met even by over-fuelling the engine to 24 to 1 air-fuel ratio. Note that the turbo-charger size is optimised for each case and compared with that required for case 1 - taken as 100%. The higher speed condition is achieved by an increase in the size of the turbo-charger as indicated under case III. Case IV shows the advantage of an increase in the exhaust valve size for that particular load and speed condition.

TABLE 9

	Case I max. torque	Case II max. torque with boost limit of 3.0 to 1	Case III max. power with 'standard' valves	Case IV max. power
Engine Speed, r.p.m.	1600	1600	2800	2800
Engine IMEP (psi)	300	200	200	220
Engine compression ratio	12.0	16.0	14.0	14.0
Inlet valve size	100%	100%	100%	86%
Exhaust valve size	100%	100%	100%	114%
Compressor size	100%	83%	125%	125%
Turbine size	100%	200%	240%	240%
Peak cylinder temp. °Rankine	3150	3530	3200	3232
Peak cylinder pressure $lb_f.in^{-2}$ abs.	1670	1624	1593	1530
Boost pressure ratio	3.7	3.0	2.9	3.1
Air fuel ratio	26.75	24.00	26.80	26.57
Aftercooling change temp. °Rankine	700	680	720	720
Indicated thermal efficiency	38%	38%	34%	37.7%

Conclusions

Tests of the simulation suggest that the model responds sensibly to the changes of combustion period, injection timing and exhaust manifold volume. They stimulate an interest in a possible path for leakage between the two halves of a double entry turbine and the significance of this factor as far as performance is concerned. It is apparent also that the treatment of turbo-charger friction and the heat losses from the engine cylinder have a very significant effect on the validity of the simulation.

Although the model chosen is capable of giving a useful guide to practical design problems such as the optimisation of a variable geometry system, it is yet far from ideal. Combustion, heat transfer, and the mixing processes during induction are particular areas that are receiving close attention. Improved models of these processes are at the final stages of development and will be reported at a later date. Parametric studies are also being followed in an attempt to reduce the time required for optimisation of a complex system.

References

Shroff, H. D.	A digital programme for the performance of a diesel engine. A. S. A. E. Thesis May, 1966.
Kastner, L. J., Williams, T. J. and White, J. B.	Poppet valve characteristics and their influences on the induction processes.
Whitehouse, N. D. Stotter, A., Gondie, G. O. and Prentice, B. W.	Method of predicting some aspects of the performance of a diesel engine using a digital computer. Proc. I. Mech. E. 1962. Vol. 176, No. 9.

APPENDIX I

System equations

The thermodynamic equations for each control volume are based on the following assumptions:

a) The working fluid has properties similar to air and behaves as a semi-perfect gas, that is, it satisfies two conditions

(i) $PV = MRT$

(ii) $CV = C_v\,(T)$, a function of temperature only.

b) The fluid within each control volume is homogeneous isobaric and isothermal, that is, the processes are quasi-steady.

By the first law of thermodynamics, conservation of mass, and the characteristic gas equation, the rate equations for the state of the gas in the control volume are:

$$\frac{d\,T_c}{dt} = \frac{R\,T_c}{P_c V_c\left[C_v + T\left(\frac{\partial C_v}{\partial T}\right)\right]}\left\{(h_i - C_{v_c} T_c)\frac{dM_i}{dt} - (h_o - C_{v_c} T_c)\frac{d\,M_o}{dt} - P_c\frac{d\,V_c}{dt} + \frac{dQ}{dt}\right\} \quad \ldots\ldots\ldots\ldots (1)$$

$$\frac{d\,P_c}{dt} = \frac{P_c}{T_c}\frac{d\,T_c}{dt} + \frac{R\,T_c}{V_c}\left\{\frac{d\,M_i}{dt} - \frac{d\,M_o}{dt}\right\} - \frac{P_c}{V_c}\frac{d\,V_c}{dt} \quad \ldots\ldots (2)$$

Five auxiliary equations were also required. Two equations for the rate of mass flow through the valves were based upon these for flow through a noxxle, that is

$$\frac{dM}{dt} = A_e\,P_u\left\{\frac{2\,g_o\,k}{R\,T_u(k-1)}\left[\left(\frac{P_d}{P_u}\right)^{\frac{2}{k}} - \left(\frac{P_d}{P_u}\right)^{\frac{k+1}{k}}\right]\right\}^{\frac{1}{2}} \quad \ldots\ldots\ldots (3)$$

for subsonic flow at the throat and

$$\frac{dM}{dt} = A_e\,P_u\left\{\frac{g_o\,k}{R\,T_u}\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}\right\}^{\frac{1}{2}} \quad \ldots\ldots\ldots (4)$$

when the flow is choked, that is when $\frac{P_d}{P_u} < \left(\frac{2}{k+1}\right)^{\frac{k}{k+1}}$

A_e, the effective throat area is that defined by Kastner et. al (2), and depends upon the geometry of the valve, the valve lift and a coefficient of discharge.

The rate of charge of cylinder colum is given by

$$\frac{dV}{d\theta} = \frac{S\pi^2 D^2}{1440}\left\{\mathrm{SIN}\theta + \frac{\mathrm{SIN}\theta\ \mathrm{COS}\theta}{\sqrt{n^2 - \mathrm{SIN}^2\theta}}\right\} \quad .... \quad (5)$$

where θ is crank angle in degrees

The rate of heat transfer between the cylinder gases and the wall is calculated according to a form of Eichelberg's equation.

$$\frac{q}{A_W} = 0.0003 \quad (\nu)^{0.6} \quad (PT)^{\frac{1}{2}} \quad (T - T_W) \quad .... \quad (6)$$

As described in the body of the report, the rate of heat release is assumed to follow the form illustrated by FIG. 3. A non-dimensional form of the variation is stored in the computer and called by a TABLE function.

APPENDIX II

The data stream ENGINE AND TEST PARAMETERS contains the necessary information to describe the engine geometry and the speed and load conditions of the test.

The layout of numbers is as follows:

STROK ,	BORE ,	RATIO ,	ECC ,
VINO ,	VEXO ,	VINC ,	VEXC ,
X FLOW,	SHAPE ,	VOL (7) ,	VOL (8) ,
VOL (9) ,	PREX ,	HEAT ,	SPEED ,
EC ,	EEXP ,	TFC ,	TFEXP ,
SFI ,	SFE ,	SFC ,	SFT ,
TINJ ,	PINK ,	T WALL ,	N7, TEST, FLEAK

where:

STROK = Engine stroke, ft.
BORE = Engine cylinder bore, ft.
ECC = Ratio of connecting rod length/crank throw
VINO = Timing of inlet valve opening, degrees C. A.
VEXO = Timing of exhaust valve opening, degrees C. A.
VINC = Timing of inlet valve closing, degrees C. A.
VEXC = Timing of exhaust valve closing, degrees C. A.
X FLOW= A parameter to simulate the finite flow capacity of the exhaust system.
SHAPE = Ratio of surface area of combustion chamber to that of a right circular cylinder of similar bore and clearance volume.
VOL (7) = Volume of inlet manifold, ft^3.
VOL (8) = Half volume of exhaust manifold, ft^3.
VOL (9) = Half volume of exhaust manifold, ft^3.
PREX = Initial value of exhaust pipe pressure lb_f in^{-2} abs.
HEAT = Fuel heat input per cylinder per cycle, ft. lb_f
SPEED = Engine Speed, r. p. m.
EC = Constant for Eichelberg's heat transfer equation.
EEXP = Speed index for Eichelberg's heat transfer equation.
TFC = Constant for turbo-charger bearing friction law.
SFI = Scale factor for inlet valve area.
SFE = Scale factor for exhaust valve area.
SFC = Scale factor for compressor flow.
SFT = Scale factor for turbine flow.

In order to avoid reading in lengthy tables for every change in valve or turbo-charger size, scale factors are provided for the valve flow areas and the flow capacities of the compressor and turbine. The true valve flow area is that read from the table and multiplied by the appropriate scale of factor. In addition, the non-dimensional speed is also divided by the square root of the appropriate scale factor to determine the true speed required when using the table.

TINJ = Combustion timing, degrees C. A.
PINJ = Combustion period, C. A. degrees.
T WALL = Gas side cylinder wall temperature $^{\circ}$Rankine.

N7	=	Sets the number of integration steps before the first output. Normally there is no output till the iterations have reached a stable solution, then N7 is automatically set equal to zero and results are printed at each step of integration.
TEST	=	Accuracy level required before the cycle is assumed to be periodic. If TEST is set equal to zero it causes a normal exit from the computer.
FLEAK	=	A factor to incorporate mass flows between the two halves of the turbine.

APPENDIX III

Starting values

This input stream is used to present the values of the temperatures and pressures in each of the control volumes in order that the computation may proceed.

The layout is as follows:-

TRPM,	T_1,	T_2,	T_3
T_4,	T_5,	T_6,	T_7,
P_3,	P_4,	P_5,	P_6,
P_7,	P_8,	P_9.	

TRPM	=	Turbo-charger speed R. P. M.
T_1 to T_6	=	Cylinder gas temperatures ^oR.
P_1 to P_6	=	Cylinder gas pressures p. s. i. a.
T_7, P_7	=	Inlet manifold temperature ^oR and pressure p. s. i. a.
T_8, P_8, T_9 P_9	=	Temperatures ^oR and pressures p. s. i. a. in the two halves of the exhaust manifold.

APPENDIX IV

The table "Function"

The table function, originated by Mr. Palmer of the Department of Propulsion, College of Aeronautics, stores up to three dependent variables as functions of one or two independent parameters. The "Function" has been written as a Fortran IV routine which stores up to two dependent and two independent variables.

The data is read into a COMMON ARRAY TAB from TAB (I) onwards by means of the master programme. 'I' becomes the address of the table. The number of tables that can be stored is limited only by the size of the array TAB.

The numbers to be read in must be presented to the computer in a particular order.

The first two lines contains the words:

DATA

TABLE OF ....

This is followed by a set of ten numbers whose meaning is given below.

If x_1 and x_2 are the independent variables and y_1 and y_2, the dependent variables, then the number layout is as follows:-

min. x_1,	min. x_2,	max. x_1,	max. x_2,
ηx_1 ,	ηx_2 ,	ηy_1 ,	ηy_2 ,
0.0 ,	0.0		

values of x_1 in ascending order

values of x_2 (if present in ascending order)

values of y_1 for each x_1 for 1st value of x_2

values of y_1 for each x_2 for 2nd value of x_2

values of y_1 for each x_2 for last value of x_2

(if x_2 does not exist, then just the values of y_1 for x_1 in increasing order)

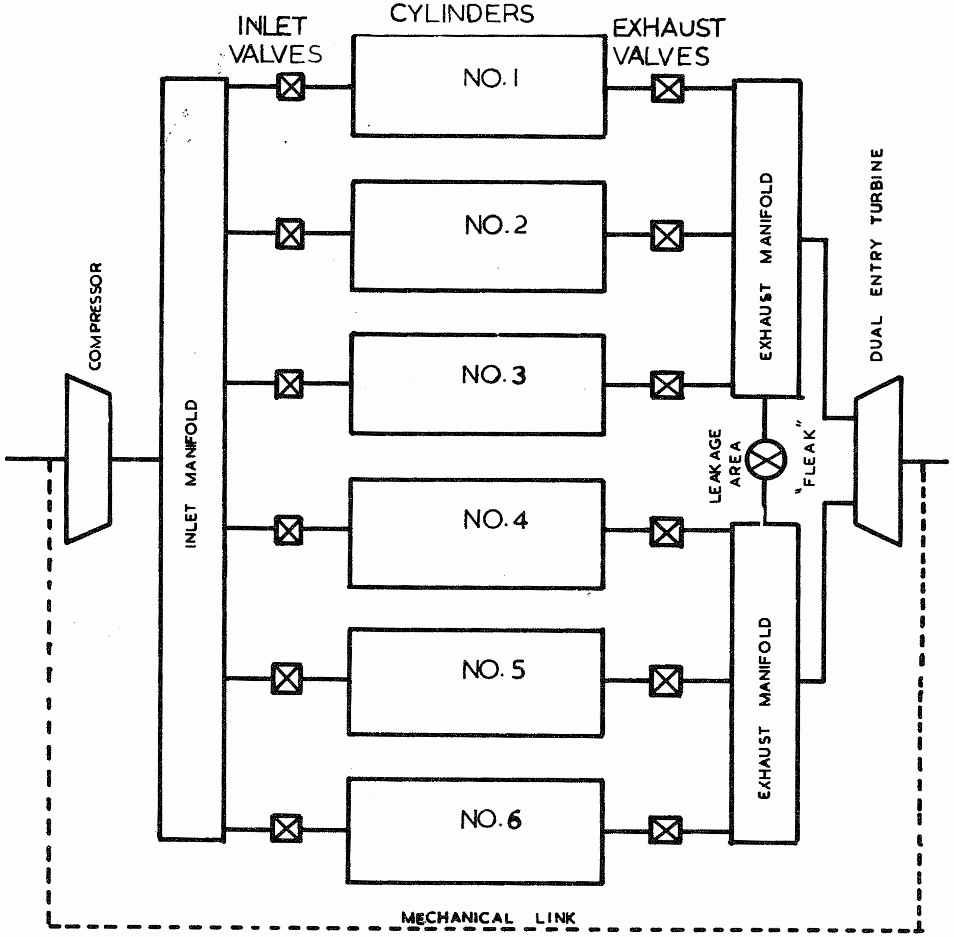
values of y_2 (if present) in a similar order to y_1.

Once the numbers are read, any reference to the table can be made by a sub-routine call of the type

CAL CALL TABLE (I, X1, X2, Y1, Y2)

where appropriate values have been given to I, X1 and, where relevant, X2.

The values of y_1 (X1, X2) and y_2 (X1, X2) are available in Y1 and Y2.



CONTROL VOLUMES FOR SIX CYLINDER TURBOCHARGED ENGINE WITH DUAL ENTRY TURBINE

FIG. 1

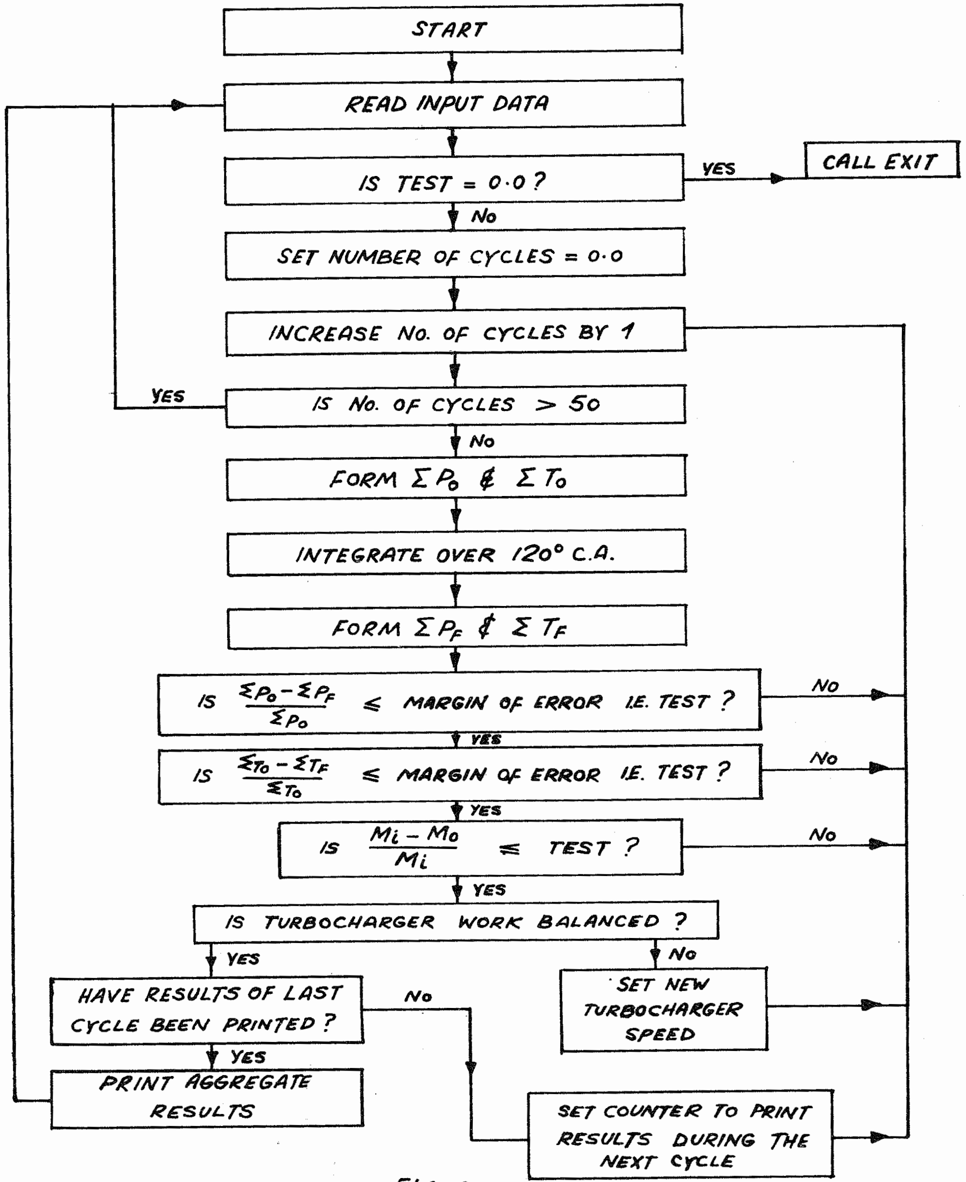


FIG. 2

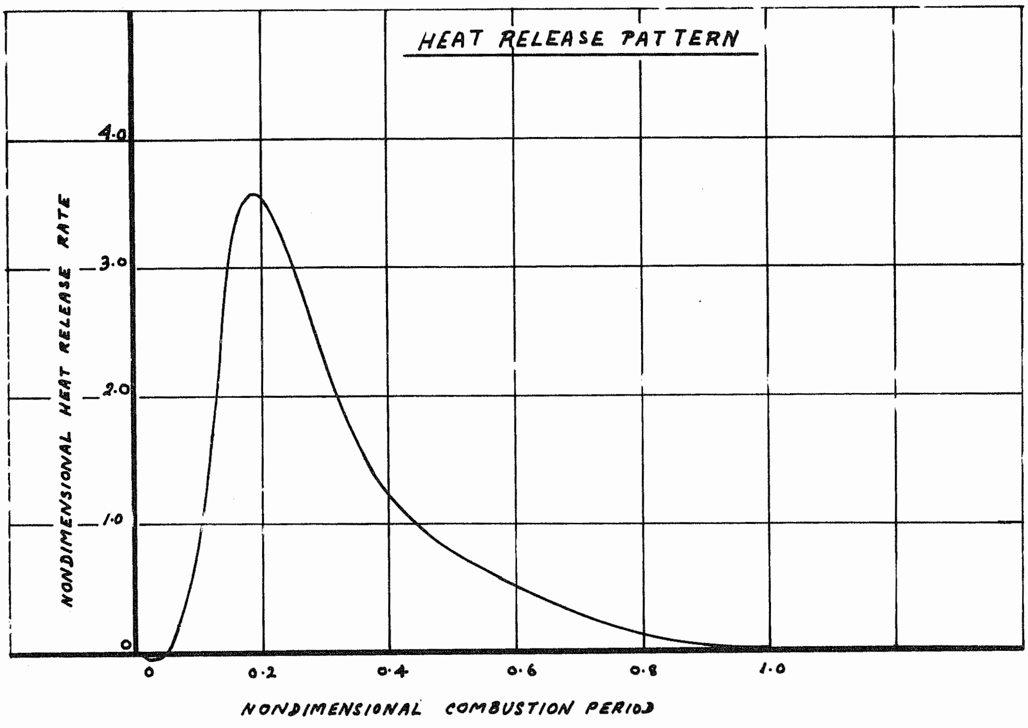

HEAT RELEASE PATTERN
4.0
3.0
2.0
1.0
0
NONDIMENSIONAL HEAT RELEASE RATE
0
0.2
0.4
0.6
0.8
1.0
NONDIMENSIONAL COMBUSTION PERIOD

FIG. 3

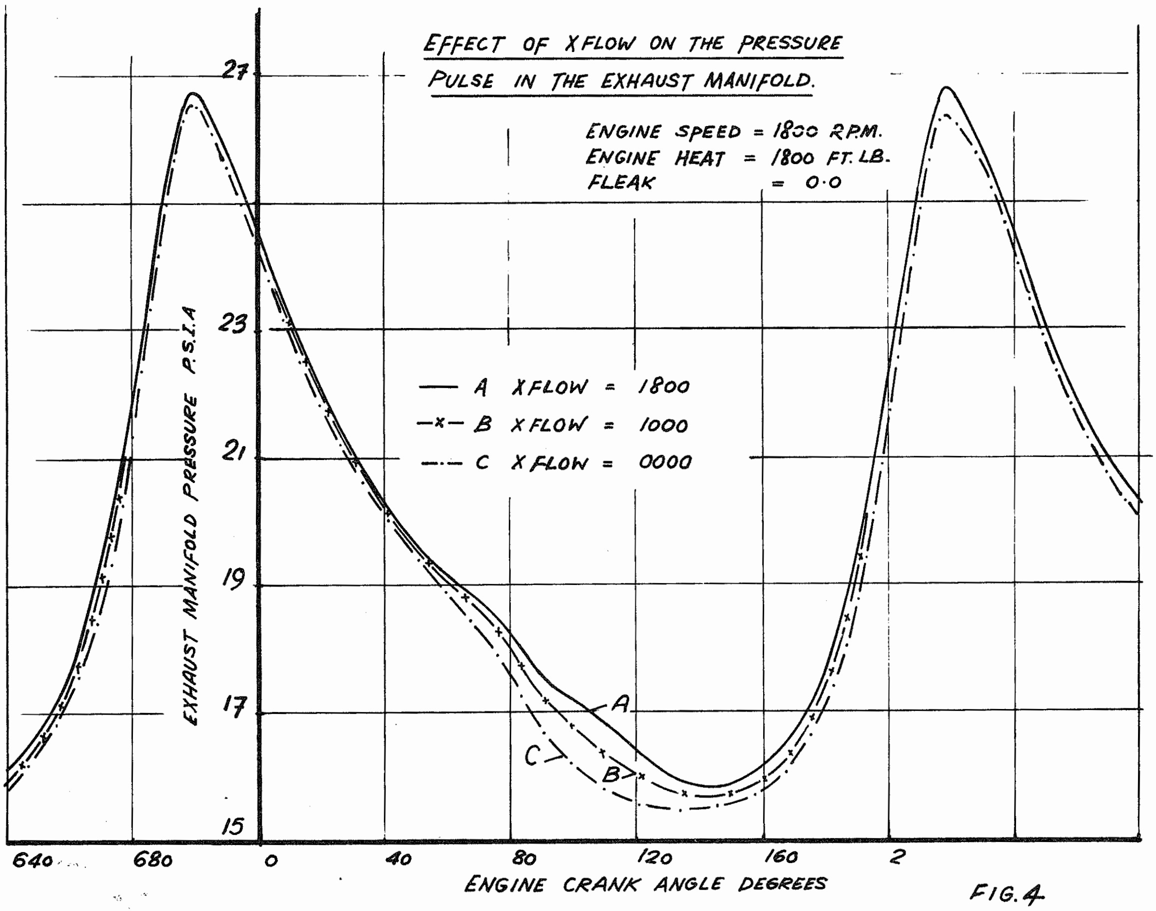

EFFECT OF XFLOW ON THE PRESSURE
PULSE IN THE EXHAUST MANIFOLD.
ENGINE SPEED = 1800 RPM.
ENGINE HEAT = 1800 FT. LB.
FLEAK = 0.0
A XFLOW = 1800
B XFLOW = 1000
C XFLOW = 0000
27
23
21
19
17
15
EXHAUST MANIFOLD PRESSURE P.S.I.A
A
B
C
640
680
0
40
80
120
160
2
ENGINE CRANK ANGLE DEGREES

FIG. 4

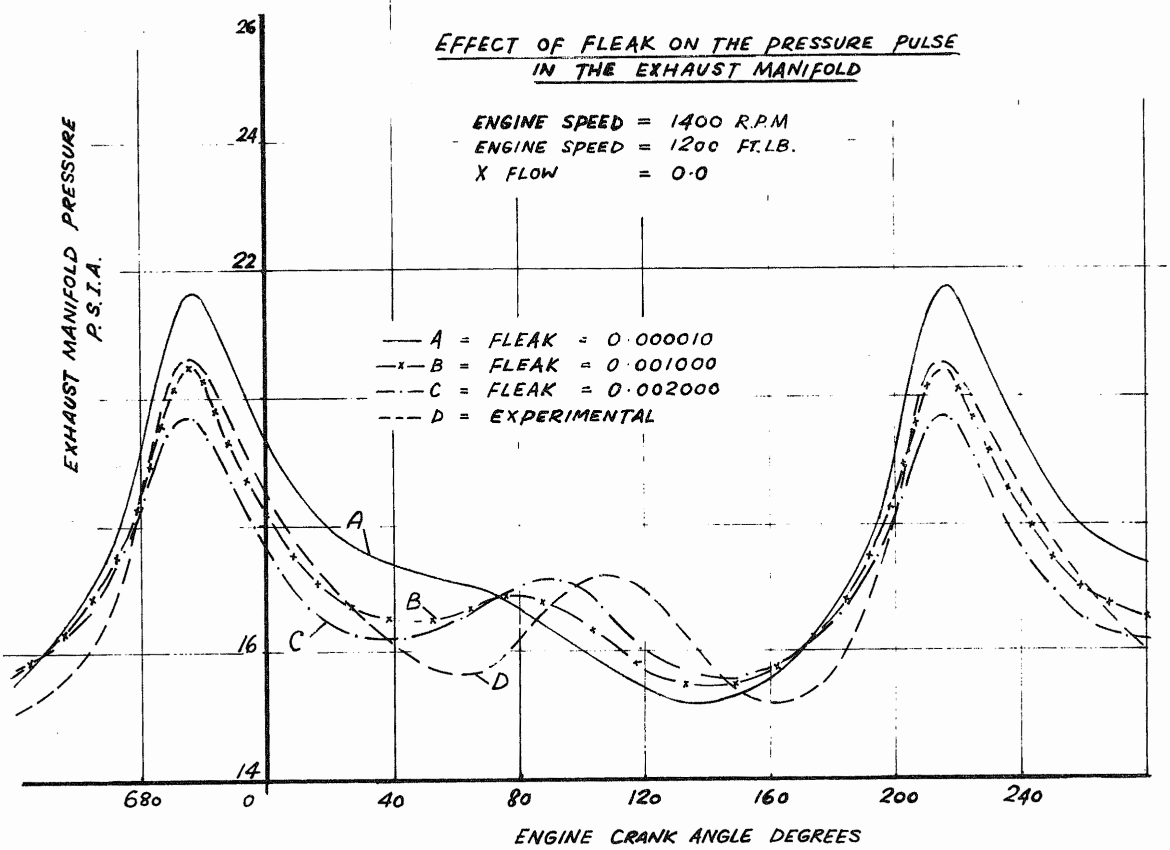

EFFECT OF FLEAK ON THE PRESSURE PULSE IN THE EXHAUST MANIFOLD
ENGINE SPEED = 1400 R.P.M
ENGINE SPEED = 1200 FT.LB.
X FLOW = 0·0
A = FLEAK = 0·000010
B = FLEAK = 0·001000
C = FLEAK = 0·002000
D = EXPERIMENTAL
EXHAUST MANIFOLD PRESSURE P.S.I.A.
26
24
22
16
14
A
B
C
D
680
0
40
80
120
160
200
240
ENGINE CRANK ANGLE DEGREES

FIG. 5

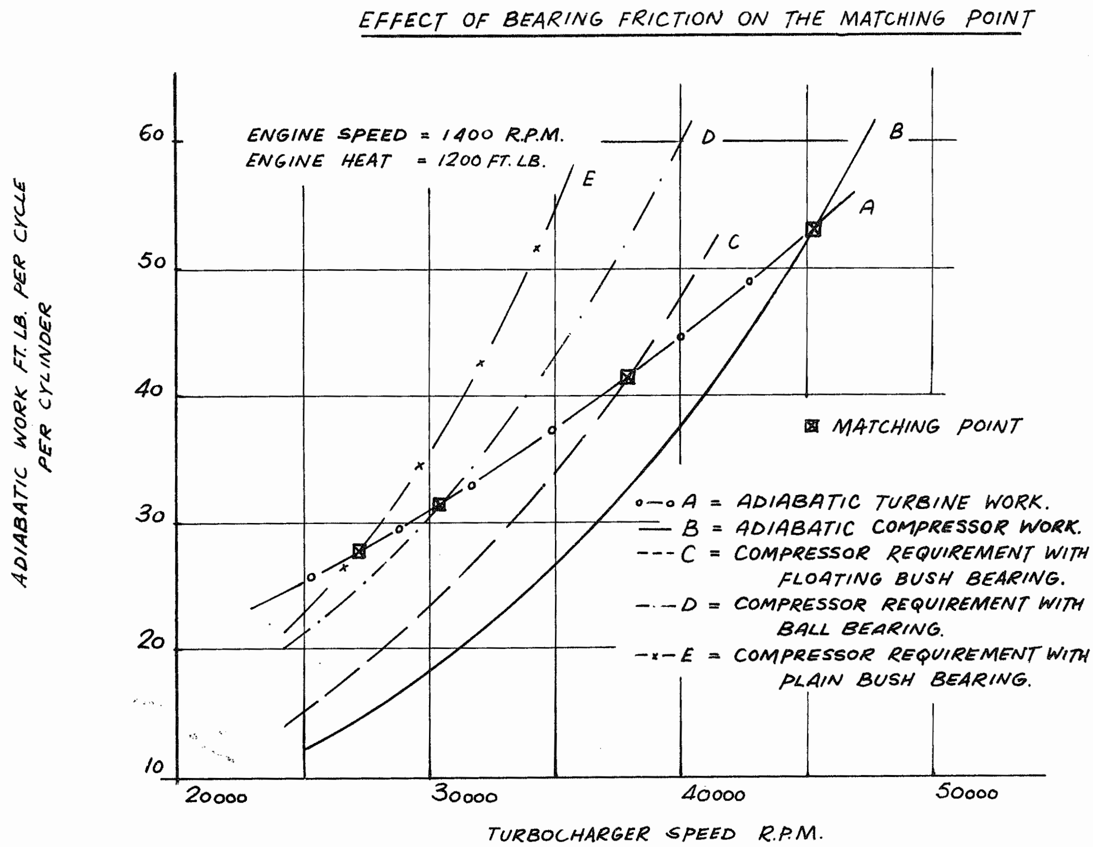

EFFECT OF BEARING FRICTION ON THE MATCHING POINT
ENGINE SPEED = 1400 R.P.M.
ENGINE HEAT = 1200 FT. LB.
D
B
E
A
C
MATCHING POINT
A = ADIABATIC TURBINE WORK.
B = ADIABATIC COMPRESSOR WORK.
C = COMPRESSOR REQUIREMENT WITH FLOATING BUSH BEARING.
D = COMPRESSOR REQUIREMENT WITH BALL BEARING.
E = COMPRESSOR REQUIREMENT WITH PLAIN BUSH BEARING.
ADIABATIC WORK FT. LB. PER CYCLE PER CYLINDER
60
50
40
30
20
10
20000
30000
40000
50000
TURBOCHARGER SPEED R.P.M.

FIG. 6

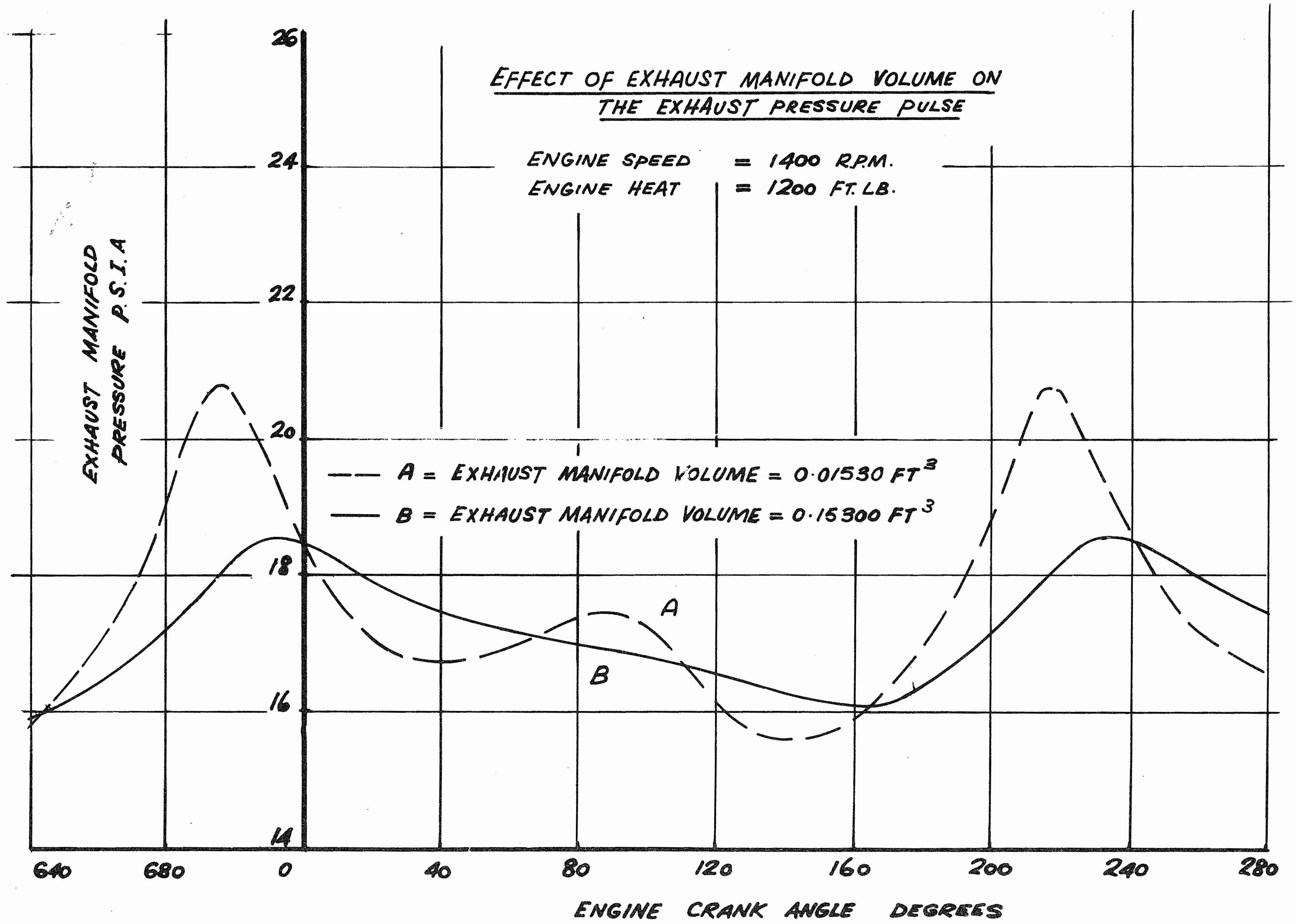

EFFECT OF EXHAUST MANIFOLD VOLUME ON
THE EXHAUST PRESSURE PULSE
ENGINE SPEED = 1400 R.P.M.
ENGINE HEAT = 1200 FT. LB.
A = EXHAUST MANIFOLD VOLUME = 0·01530 FT³
B = EXHAUST MANIFOLD VOLUME = 0·15300 FT³
A
B
EXHAUST MANIFOLD PRESSURE P.S.I.A
26
24
22
20
18
16
14
640
680
0
40
80
120
160
200
240
280
ENGINE CRANK ANGLE DEGREES

FIG. 7