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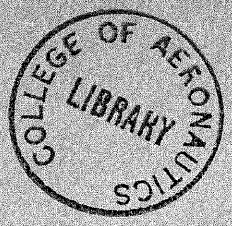


ADVANCED SCHOOL OF AUTOMOBILE ENGINEERING

CRANFIELD

Report to the Worshipful Company of Coachmakers

and Harnessmakers

# 'The digital simulation of a turbo-charged diesel engine'

- by -

H.D. Shroff

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'The digital simulation of a turbo-charged diesel engine'

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This is an interim report. An official report of The College of Aeronautics will be published when the current programme of research is complete.

ACKNOWLEDGEMENTS

The author is grateful to the Worshipful Company of Coachmakers and Harnessmakers for the award of the scholarship (October 1966 to October 1967) which enabled him to complete the work of this report.

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REPORT TO THE WORSHIPFUL COMPANY OF COACHMAKERS

AND HARNESSMAKERS

# "THE DIGITAL SIMULATION OF A TURBO-CHARGED DIESEL ENGINE"

by : H.D.Shroff

SUMMARY

A mathematical simulation of a four stroke six cylinder turbo-charged diesel engine was used to study the changes in performance caused by the variation of engine parameters and operating conditions.

The principle variables were turbocharger size, combustion timing and turbocharger bearing friction. The effect of these and other factors are tabulated and discussed. The results were explained in terms of the mathematical model employed and by comparison with the physical system.

The results suggest that the mathematical model behaves sensibly, in most cases, but that further development is required in order to incorporate the refinements which are necessary for an accurate simulation.

LIST OF SYMBOLS AND ABBREVIATIONS.

TRPM = turbocharger speed R.P.M.

W TURB = turbine work/cycle/cylinder Ft.lb.

W COMP = Compressor work/cycle/cylinder Ft.lb.

X FLOW = Parameter discussed in text.

FLEAK = Parameter discussed in text.

E = exponent to the base ten

e.g. 1.5 E-3 = $1.5 \times 10^{-3}$

TINJ = Combustion timing C.A.

PINJ = Combustion period C.A.

C.A. = Crank angle running from zero to 720°. Zero is set at top dead centre prior to induction stroke.

TFC = Constant for turbocharger bearing friction law.

TFEXP = Exponent for turbocharger bearing friction law

Cp = Specific heat at constant pressure Ft.lb/lb°R.

Cv = Specific heat at constant volume Ft.lb/Lb.°R.

k = Ratio of specific heats Cp/cv.

R = Gas constant per lb. Ft.lb/Lb°R.

T = Temperature °R.

P = Pressure p.s.f.a.

V = Volume $ft^3$

Q = Heat ft. lb.

M = mass lb.

## Introduction

The physical processes that take place in an internal combustion engine are so interelated to each other that the study of an isolated factor is not possible. This necessitates that they be studied collectively, either

i) By experimental work on an engine

ii) By a mathematical simulation of the engine

The first method can be expensive and time consuming and the results may be particular to that engine.

If the various processes taking place in an internal combustion engine can be represented by reliable mathematical models the whole engine can be represented by a group of such models. The simulation would be cheap to run, easy to alter and applicable to various engines. The reliability of the results obtained from such a system, however, depends primarily on the suitability of the models employed. Provided the limitations of the simulation are clearly understood, it can usefully be employed to study many aspects of the engine behaviour which would be difficult or impossible to consider by any other means.

The intention of the work in this report was to produce a mathematical model which would not only incorporate all the refinements necessary for the accurate simulation of a normally aspirated Diesel engine, but would also be suitable for the prediction of the behaviour of the engine when fitted with an exhaust driven turbocharger.

## DESCRIPTION OF THE COMPUTER PROGRAMMES.

Before entering into a detailed description of the latest programme a brief resume of its development is appropriate. This was in three distinct stages:

a) The first programme was written to simulate a single cylinder engine with induction and exhaust at constant pressure levels.

b) The single cylinder was improved by a consideration of the inlet manifold and the exhaust manifold effect on the pressure and temperature etc. of the inducted and exhaust gases.

c) The model was enlarged to incorporate six cylinders, one inlet manifold fed from a compressor, and two exhaust manifolds with each feeding into one half of a twin entry turbine.

The earlier programmes were written in Pegasus Autocode and were tested and run at the College of Aeronautics. The computer was too slow and it was soon apparent that for a comprehensive testing the programmes would have to be re-written for a faster machine. The programmes were written in Hartran and run on the Chilton Atlas at Harwell. With the installation of an I.C.T. 1909 computer at the Cranfield Computing Centre, The College of Aeronautics, in September 1967, the programme was revised and re-written to the A.S.A. Fortran specifications.

The single cylinder programmes are now obsolete and will not be considered henceforth. Further discussion is limited to the latest programme (c). The programme simulates a six cylinder diesel engine fitted with a turbocharger. The engine has two exhaust manifolds and these are connected to the two halves of a twin entry turbine.

The engine turbocharger system is divided into a set of control volumes. These control volumes are isolated from one another except during certain periods of the engine cycle. When the inlet valve of a cylinder is open it is in communication with the inlet manifold and mass transfer may take place between these two. Similarly, during the exhaust period there may be mass transfer between the exhaust manifold and the engine cylinder. The gases within each control volume are assumed to obey the unsteady flow energy equation and the law of conservation of mass. The whole system is then represented by a set of simultaneous differential equations of the type given in appendix I. These equations are numerically integrated to give the gas states in each control volume at each point during the engine cycle.

In order to speed up the running of the programme certain additional facilities are provided but before these can be implemented three assumptions are made:

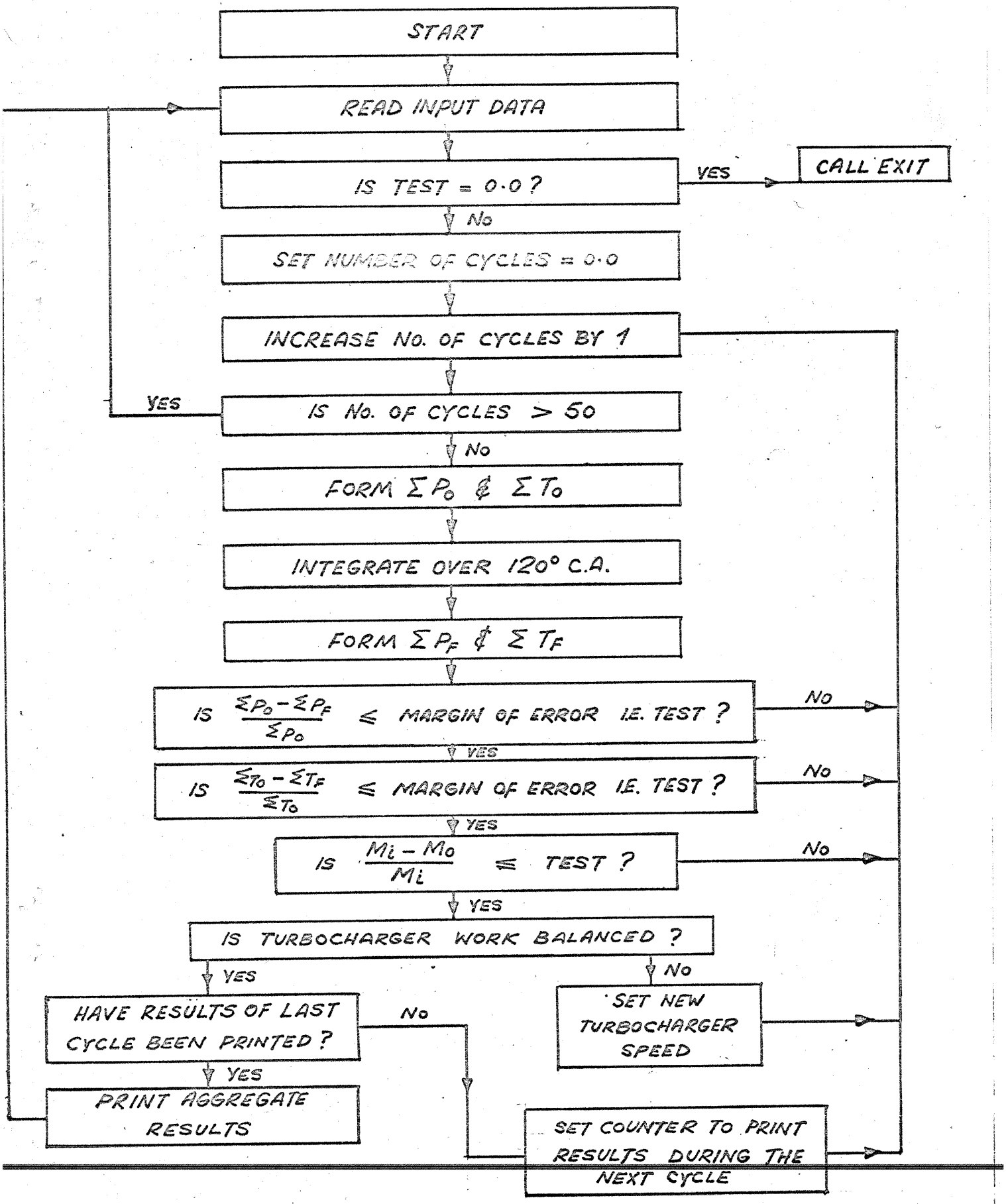
a) The engine cycle is periodic in nature.

b) The engine cylinders are identical.

c) The cylinders are injected with fuel at equal intervals.

Assumption a) relies on the repeatability of the indicator diagram from cycle to cycle. In practice there is always a slight variation but, since in a diesel engine, this variance is small it is not a serious problem. Assumption b) is not strictly valid in that there are differences in the valve layouts, the cooling water flow rates and temperatures, etc. This would introduce a certain degree of error into the solution. However, the implementation of these assumptions allows the computation times to be reduced by a factor of six and it was felt that this consideration outweighs the accuracy by which variations from cylinder to cylinder can be incorporated. Assumption c) is reasonable except for slight discrepancies in the fuel pump system.

With these assumptions the cycle period is reduced from $720^{\circ}$C.A. to $120^{\circ}$C.A. The computation of the temperatures and pressures in each control volume is allowed to run for a period of $120^{\circ}$C.A. and a check is made to see if the cycle has reached periodicity. A certain margin of error (usually 1%) is allowed on the temperatures and pressures within the system and if the tests show that the cycle is periodic to within this error the calculations are terminated. The gas states at each point of the cycle is printed in the output. However, if the test fails the cycle is rerun with the new values of pressures and temperatures as starting conditions for the next cycle and such iterations are repeated until periodicity is reached. There is a master control to stop the programme if periodicity is not achieved in less than fifty iterations.


START
READ INPUT DATA
IS TEST = 0·0 ?
YES
CALL EXIT
No
SET NUMBER OF CYCLES = 0·0
INCREASE No. OF CYCLES BY 1
YES
IS No. OF CYCLES > 50
No
FORM ΣP0 & ΣT0
INTEGRATE OVER 120° C.A.
FORM ΣPF & ΣTF
IS (ΣP0 − ΣPF)/ΣP0 ≤ MARGIN OF ERROR I.E. TEST ?
No
YES
IS (ΣT0 − ΣTF)/ΣT0 ≤ MARGIN OF ERROR I.E. TEST ?
No
YES
IS (Mi − M0)/Mi ≤ TEST ?
No
YES
IS TURBOCHARGER WORK BALANCED ?
YES
No
HAVE RESULTS OF LAST CYCLE BEEN PRINTED ?
No
SET NEW TURBOCHARGER SPEED
YES
PRINT AGGREGATE RESULTS
SET COUNTER TO PRINT RESULTS DURING THE NEXT CYCLE

The flow diagram of the overall control is illustrated by FIG. 0. Details of individual processes are given in reference 1.

USE OF THE PROGRAMME.

To run the programme it is necessary to input the basic data to describe the engine.

There are seven distinct input streams which must be listed in the order given below. Each stream must start with the word DATA on one line and the name of the stream on the next,

e.g DATA

STARTING VALUES

1) STARTING VALUES.

This data stream contains the pressures and temperatures of the control volumes at the start of the computation. If the starting values are close to the end result it results in a considerable saving in computing time before the cycle becomes periodic. In practice this implies a fore knowledge of the result which is not readily available and several iterations are necessary. The actual layout of numbers is given in appendix III.

2) TABLE OF GAS PROPERTIES.

Air is used as the working fluid and its thermodynamic properties ($C_p/C_v$ and $C_v$) are stored as a function of temperature in

a TABLE format as described in appendix IV.

3) TABLE OF VALVE LIFTS.

For a poppet valve the area of gas flow is an annular space formed between the valve edge and the seat. It is effectively reduced by the effect of wall drag and boundary layer build-up on the valve sides. Hence the effective area of flow at any instant is considered as a function of the valve size lift profile, and an instantaneous coefficient of discharge as defined by Kastner et. al. (reference 2). This discharge coefficient is dependent on the valve lift to diameter ratio and on the pressure drop across the valve. Since the effect of valve lift is much more important than the pressure drop, for the purposes of this programme the valve lift only is taken into account. By ignoring the effect of pressure drop on the instantaneous coefficient of discharge, the available area of gas flow at any instant can be computed from a knowledge of the valve geometry and its lift. This area is stored for both the inlet valve and the exhaust valve as a function of crank angle in a TABLE format.

4) TABLE OF HEAT RELEASE.

The programme requires a preset heat release pattern. This is input as a TABLE of Heat Release rate per unit heat release as a function of the fraction of unit period.

5) TABLE OF COMPRESSOR CHARACTERISTICS

6) TABLE OF TURBINE CHARACTERISTICS.

The digital programme does not incorporate the mathematical models to simulate the compressor or the turbine, but during the course of the computation the compresnor and turbine is simulated by reference to their non-dimensional steady state characteristics which are stored as two tables.

a) TABLE OF COMPRESSOR CHARACTERISTICS plotted as compressor non-dimensional mass flow ($M\sqrt{T}/P$) and adiabatic efficiency as a function of pressure ratio and non-dimensional speed ($TRPM/\sqrt{T}$).

b) TABLE OF TURBINE CHARACTERISTICS. The parameters are the same as for the compressor but in this case the non-dimensional mass flow ($M\sqrt{T}/P$) refers to one half of the turbine only.

7) ENGINE AND TEST PARAMETERS.

This input stream describes the engine geometry and the speed and load conditions of a particular test (see appendix II).

RESULTS.

The results of the computation are given on the lineprinter.

At the end of each iteration the programme outputs one line of numbers. These are: i) sum of temperatures, ii) sum of pressures, iii) mass through compressor, iv) mass through forward half of turbine, v) mass through reverse half of turbine. When there is a change in

turbine speed, it prints a line showing the power balance on the turbocharger. The numbers represent i) mechanical work to drive the compressor and to overcome bearing friction, ii) work delivered by the forward half of turbine, iii) work delivered by the reverse half of the turbine, iv) the overall deficiency in work. This is followed by the new turbocharger speed which is printed on a new line.

When the cycle has reached periodicity and the turbine compressor power balance is satisfactory, the programme outputs six lines of output which give the significant parts of the input stream 'ENGINE AND TEST PARAMETERS'.

When this is complete, the programme repeats the cycle and prints out the state of the gases, the work done, the heat transfer, etc. for each control volume at the end of each step length of integration. This layout is described as follows.

There are ten columns. The first nine refer to the six engine cylinders, the inlet manifold and the two exhaust manifolds. The tenth column is a sum of terms in the six engine cylinders. In row order, the layout is:-

a) Crank angle (six columns only).

b) Temperature $^{o}R$ (first nine columns only)

c) Pressures p.s.i.a. (the tenth column represents the pressure in the exhaust pipe.

d) Work done. FT.LB. (column seven represents the compressor adiabatic work. Columns eight and nine refer to the adiabatic work done by the two halves of the turbine).

e) Heat input FT.LB.

f) Heat Losses.

g) Mass inflow.

h) Mass outflow.

When the cycle is complete, the aggregate results such as I.M.E.P., I.H.P., THERMAL EFFICIENCY, JACKET LOSSES and Air/Fuel ratios are also printed. The programme then prints the date and the time.

RESULTS AND DISCUSSION.

The model was used to simulate an A.E.C. AV 410 turbocharged four stroke diesel engine. The turbocharger fitted was a twin entry turbine C.A.V. type 12 unit with a $70^{\circ}$ nozzle ring.

The turbocharger characteristics were predicted characteristics and were made available by M.I.R.A.

The engine details are given below.

| | |
|---|---|
| Bore | 4.13 in. |
| Stroke | 5.12 in. |
| Compression Ratio | 18:1 |
| l/r ratio | 3.57 |

I.V.O. $10^{\circ}$ B.T.D.C. = $710^{\circ}$ C.A.

I.V.C. $38^{\circ}$ A.B.D.C. = $218^{\circ}$ C.A.

E.V.O. $40^{\circ}$ B.B.D.C. = $500^{\circ}$ C.A.

E.V.C. $20^{\circ}$ A.T.D.C. = $20^{\circ}$ C.A.

NOTE: All crank angles are measured from the top dead centre prior to the start of the induction stroke.

| | | |
|---|---|---|
| Valve seat angle | = | $40^{\circ}$ |
| 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 $in^3$

This simulation was used to study the effect of two system parameters and nine operating variables.

The two system parameters were:

1) X FLOW
2) FLEAK

and the nine variables were

3) Turbocharger size
4) Combustion Timing (TINJ)
5) Valve Sizes
6) Turbocharger Bearing Friction
7) Heat Transfer coefficient
8) Exhaust manifold volume.
9) Combustion Period (PINJ)
10) Engine Speed
11) Engine Fuelling Rate (HEAT)

Each of these items will be discussed in detail.

1) <u>X FLOW</u>.

X Flow is a parameter introduced to simulate the pressure rise in the exhaust pipe due to the finite flow capacity of the exhaust system. For small pressure changes the pressure drop due to a restricted passage may be approximated by

$$\Delta p = k V^2$$

$$\Delta p = k \rho \left(\frac{\dot{M}}{\rho A}\right)^2$$

$$= \frac{k}{\rho} \left(\frac{\dot{M}}{A}\right)^2$$

If $\rho$ is assumed constant, then for a given area, A

$$\Delta p = K1 \ (\dot{M})^2$$

In the programme K1 is replaced by X flow as:

$$\Delta p = X \text{ flow } (1000.0 \ \dot{M})^2$$

where,

$\Delta p$ = pressure drop, p.s.f.a.

A = area $ft^2$

$\rho$ = gas density $lb/ft^3$

V = gas velocity f/sec

M = mass flow rate lb/sec

k,K1 = constants

This is only an approximate method for the estimation of the pressure variation in the exhaust pipe and it requires that the pressure drop is small. However, it is thought to be adequate for the purpose of allowing for a small flow resistance in the pipe.

The effect of X flow is tabulated in Table 1 and plotted in Fig. 1, where it can be seen that X flow not only increases the mean pressure level in the exhaust pipe but it also imposes a

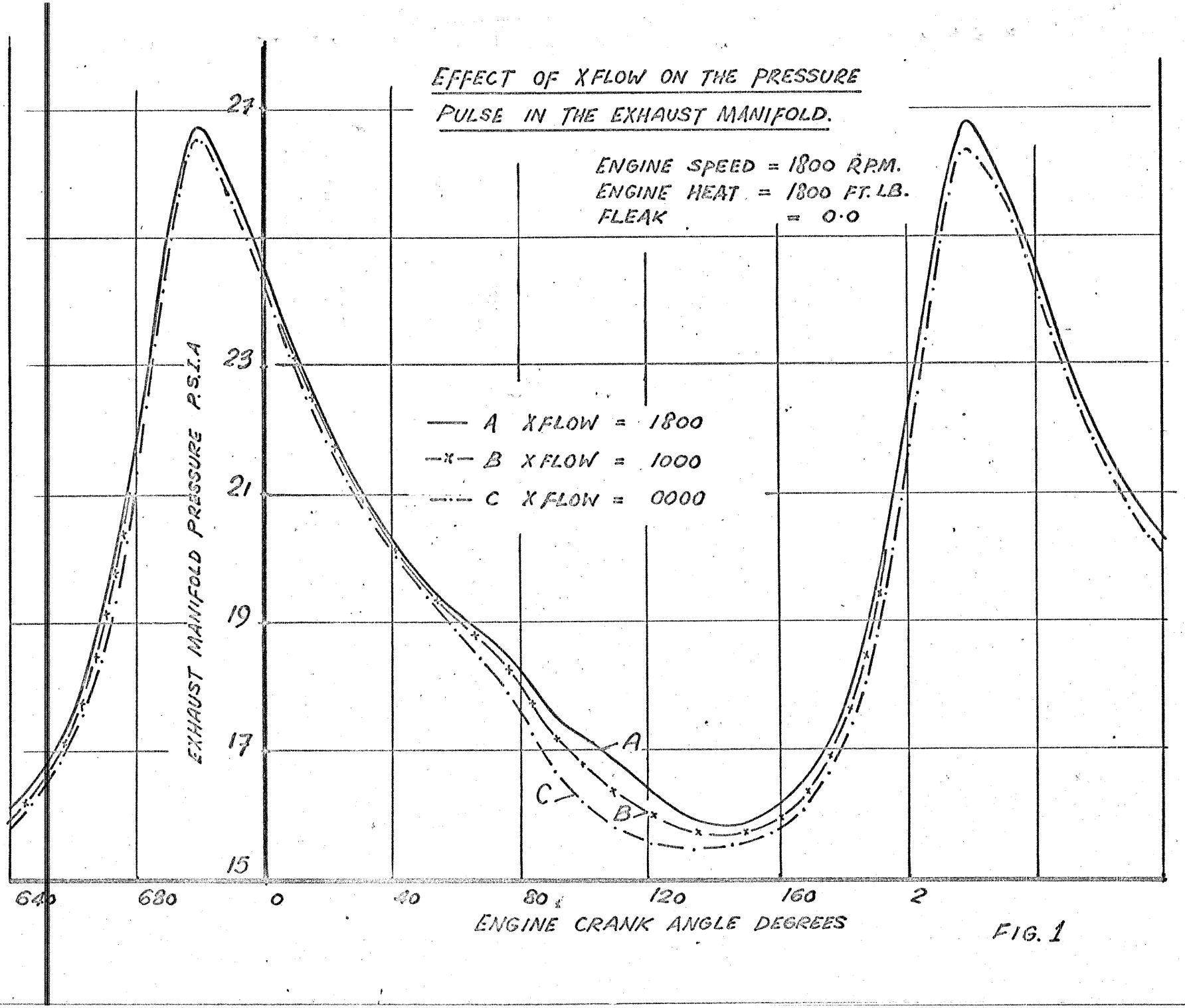

EFFECT OF XFLOW ON THE PRESSURE
PULSE IN THE EXHAUST MANIFOLD.
ENGINE SPEED = 1800 R.P.M.
ENGINE HEAT = 1800 FT. LB.
FLEAK = 0.0
A XFLOW = 1800
B XFLOW = 1000
C XFLOW = 0000
EXHAUST MANIFOLD PRESSURE P.S.I.A
27
23
21
19
17
15
640
680
0
40
80
120
160
2
ENGINE CRANK ANGLE DEGREES

FIG. 1

cyclic variation on this pressure. However the overall effect on the cycle is quite small since the turbine work and the air delivery to the engine are hardly affected.

Since the model employs a constant preset fuelling rate the variation in engine power due to the effect of any parameter or variable cannot be measured directly. It can only be indirectly estimated by the amount of air aspirated to the engine since an increase in air capacity is a potential increase in the power output of the engine. All further discussion will be confined to air capacity.

TABLE 1.

EFFECT OF X FLOW.

Engine Speed = 1800 R.P.M.

Engine Heat = 1800 Ft.lb/Cylinder/Cycle

TINJ = 360° C.A.

PINJ = 60° C.A.

| X FLOW | T RPM R.P.M. | W TURB FT.LB. | W COMP FT. LB. | AIR DELIVERY LB | PEAK EXHAUST PRESSURE P.S.I.A. |
|---|---|---|---|---|---|
| 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 |

2) FLEAK.

The effects of Fleak are tabulated in Table 2 and plotted in Fig. 2. Fig. 2-D shows a typical exhaust manifold pressure history obtained from experimental observations. The pattern exhibits primary peaks at periods of $240^{\circ}$ C.A. and secondary peaks midway between these. It was in order to simulate the occurrance of this secondary peak that the factor Fleak was introduced.

Fleak is a parameter representing the leakage area of flow between the two halves of the twin entry turbine. It allows the two halves of the exhaust manifold to transfer mass between them when they are operating at different pressure levels.

Fleak has quite a marked effect on the shape of the exhaust manifold pressure profile and on the power output from the turbine. An increase of Fleak has a detrimental effect on the air consumption of the engine and therefore on its power potential.

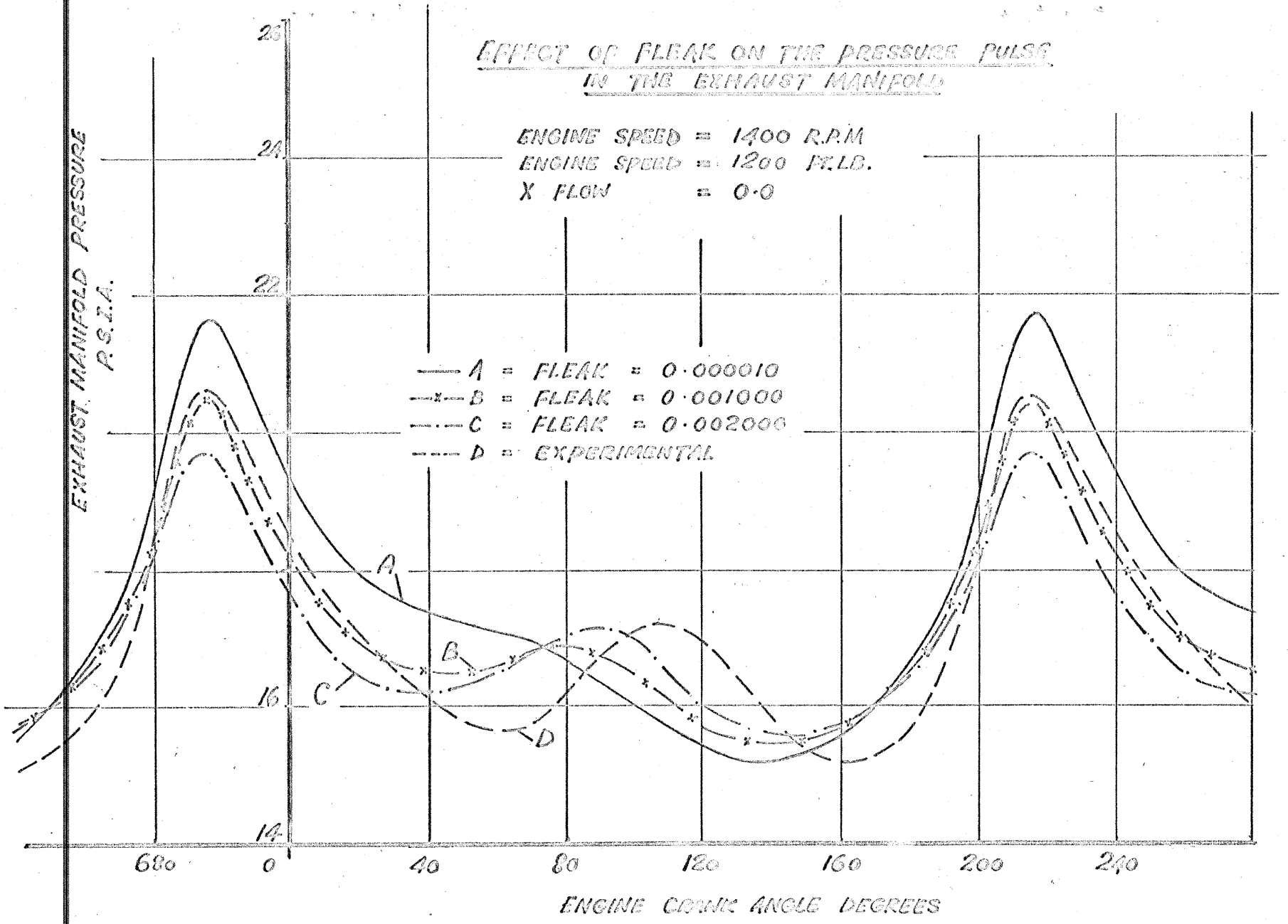

EFFECT OF FLEAK ON THE PRESSURE PULSE IN THE EXHAUST MANIFOLD
ENGINE SPEED = 1400 R.P.M
ENGINE SPEED = 1200 FT.LB.
X FLOW = 0·0
A = FLEAK = 0·000010
B = FLEAK = 0·001000
C = FLEAK = 0·002000
D = EXPERIMENTAL
EXHAUST MANIFOLD PRESSURE P.S.I.A.
26
24
22
16
14
680
0
40
80
120
160
200
240
ENGINE CRANK ANGLE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

FIG. 2

TABLE 2.

EFFECT OF FLEAK.

Engine Speed = 1400 R.P.M.

Engine Heat = 1200 Ft.lb/Cylinder/Cycle

TINJ = 355° C.A.

PINJ = 60° C.A.

| FLEAK | T RPM R.P.M. | W TURB FT.LB | W COMP FT.LB. | AIR DELIVERY LB. | PEAK EXHAUST PRESSURE P.S.I.A. |
|---|---|---|---|---|---|
| 0.0001 | 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 | 24.857 | 17.293 | 2.950E-3 | 19.857 |
| 0.00200 | 28676 | 24.057 | 16.608 | 2.930E-3 | 19.568 |

3) TURBOCHARGER SIZE.

Table 3 lists the performance for three different values of turbocharger size. Fig. 3 illustrates the effect of non-dimensional turbocharger speed on the adiabatic turbine work and the adiabatic compressor requirement for two different turbocharger sizes.

At the high speed end the larger turbocharger delivers a greater turbine power and is capable of delivering a larger air flow to the engine. As the turbocharger speeds decrease this tendency is reversed and towards the low speed end the larger

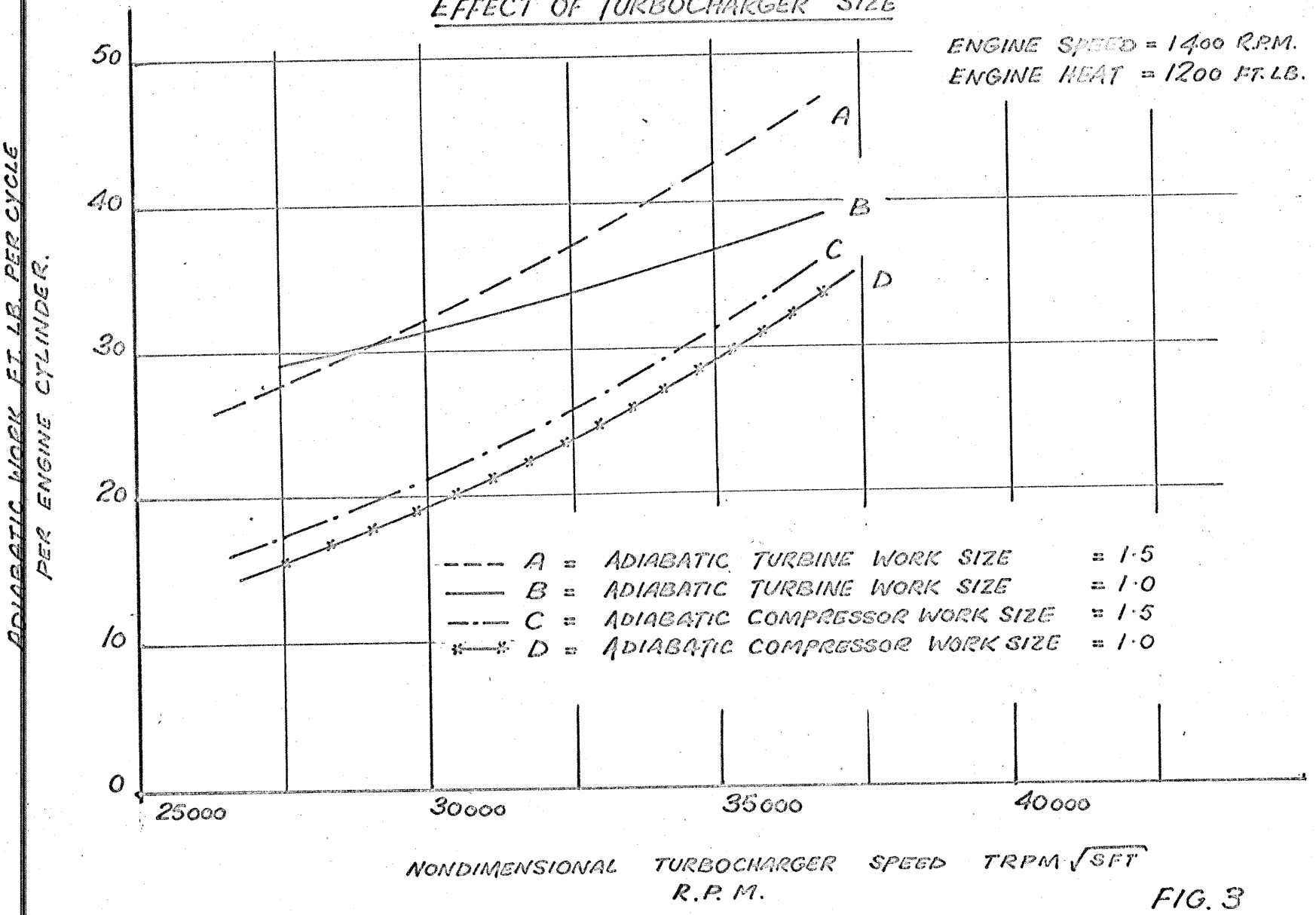

EFFECT OF TURBOCHARGER SIZE
ENGINE SPEED = 1400 R.P.M.
ENGINE HEAT = 1200 FT. LB.
ADIABATIC WORK FT LB. PER CYCLE PER ENGINE CYLINDER.
50
40
30
20
10
0
A
B
C
D
A = ADIABATIC TURBINE WORK SIZE = 1·5
B = ADIABATIC TURBINE WORK SIZE = 1·0
C = ADIABATIC COMPRESSOR WORK SIZE = 1·5
D = ADIABATIC COMPRESSOR WORK SIZE = 1·0
25000
30000
35000
40000
NONDIMENSIONAL TURBOCHARGER SPEED TRPM √SFT
R.P.M.

FIG. 3

turbine delivers less power than its smaller counterpart.

A thorough investigation would require a run through the whole range of engine operating speeds and loads for each size of turbocharger. In this way it would be possible to find the optimum size at each operating point. The results tend to show the advantages of a variable geometry turbine since it is obvious that a smaller turbocharger is needed for low-speed operation whereas at a higher engine load/speed the larger turbocharger becomes beneficial.

TABLE 3.

EFFECT OF TURBOCHARGER SIZE.

Engine Speed = 1400 R.P.M.

Engine Heat = 1200 Ft.lb/Cylinder/Cycle

TINJ = 355° C.A.

PINJ = 60° C.A.

| TURBOCHARGER SIZE | 0.75 NORMAL | NORMAL | 1.25 NORMAL |
|---|---|---|---|
| I.M.E.P. P.S.I.A. | 105.5 | 107.0 | 107.8 |
| T R.P.M. R.P.M. | 36652 | 28983 | 25200 |
| W TURB FT.LB. | 21.761 | 24.530 | 21.900 |
| W COMP FT.LB. | 18.658 | 17.073 | 16.550 |
| AIR DELIVERY LB. | 2.8375E-3 | 2.9380E-3 | 2.9850E-3 |
| EXHAUST GAS LOSSES FT.LB. | 441.0 | 439.0 | 437.0 |
| INDICATED THERMAL EFFICIENCY | 50.1% | 50.8% | 51.2% |

4) COMBUSTION TIMING (TINJ)

The results obtained with three different values of combustion timing are displayed in Table 4.

Earlier combustion results in higher peak pressures and temperatures, an overall increase in the indicated thermal efficiency and a decrease in the exhaust gas losses. This results in a decrease in the turbine work and a lowering of the amount of air aspirated by the engine. However, the change in efficiency is over-riding and results in an increase of power with earlier combustion. As would be expected, there is a decreasing return in power and efficiency compared with the increase of mechanical and thermal loading of the engine.

TABLE 4.

EFFECT OF COMBUSTION TIMING (TINJ)

Engine Speed = 1400 R.P.M.
Engine Heat = 1200 Ft.lb/Cylinder/Cycle
PINJ = 60° C.A.

| TINJ °C.A. | 355° | 360 | 365 |
|---|---|---|---|
| IMEP P.S.I.A. | 107.0 | 103.5 | 98.4 |
| W TURB FT.LB. | 24.53 | 26.49 | 28.61 |
| W COMP FT.LB. | 17.073 | 18.639 | 20.003 |
| AIR DELIVERY LB | 2.938E-3 | 2.967E-3 | 2.998E-3 |
| PEAK TEMP °R | 3094 | 2978 | 2889 |
| PEAK PRESSURE P.S.I.A. | 1126 | 930 | 877 |
| EXHAUST GAS LOSSES FT.LB. | 439 | 463 | 493 |
| INDICATED THERMAL EFFICIENCY | 50.80% | 49.15% | 46.73% |

5) VALVE SIZE.

The valve sizes were altered without altering the cylinder bore. This required that an increase in the size of one valve necessitated a decrease in the other. The results are tabulated in Fig. 5.

These results 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 lowering of the air aspirated by the engine. More computer results are required to determine whether this size is optimum for the whole range of speed/load conditions of the engine. The effect, however, is small - a 12% change in valve size results in a 1.5% change in air aspirated.

TABLE 5.

EFFECT OF VALVE SIZE.

Engine Speed = 1400 R.P.M.
Engine Heat = 1200 Ft.lb/Cylinder/Cycle
TINJ = $355^{\circ}$ C.A.
PINJ = $60^{\circ}$ C.A.

| VALVE SIZES | AIR DELIVERY LB |
|---|---|
| Inlet = 1.25 Normal<br>Exhaust = 0.70 Normal | 2.890E-3 |
| Inlet = 1.0 Normal<br>Exhaust = 1.0 Normal | 2.938E-3 |
| Inlet = 0.70 Normal<br>Exhaust = 1.25 Normal | 2.910E-3 |

6) TURBOCHARGER FRICTION.

The results of varying the turbocharger bearing friction are illustrated in Table 6. The bearing losses are represented by means of a power law of the form

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

Since different bearings have markedly different drag losses the equilibrium turbocharger speed can vary considerably from one type of bearing to another even when the engine speed and load are held constant. Fig. 4 illustrates how different bearing losses result in different matching speeds. The three bearings employed are the first three listed in Table 6.

As the bearing losses are lowered the turbocharger 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, by changing from a plain bush type to a floating bush bearing there is a 12% increase in the air delivered to the engine. Assuming that the engine is operating near its smoke limit, this would mean a power boost of the same order.

# EFFECT OF BEARING FRICTION ON THE MATCHING POINT

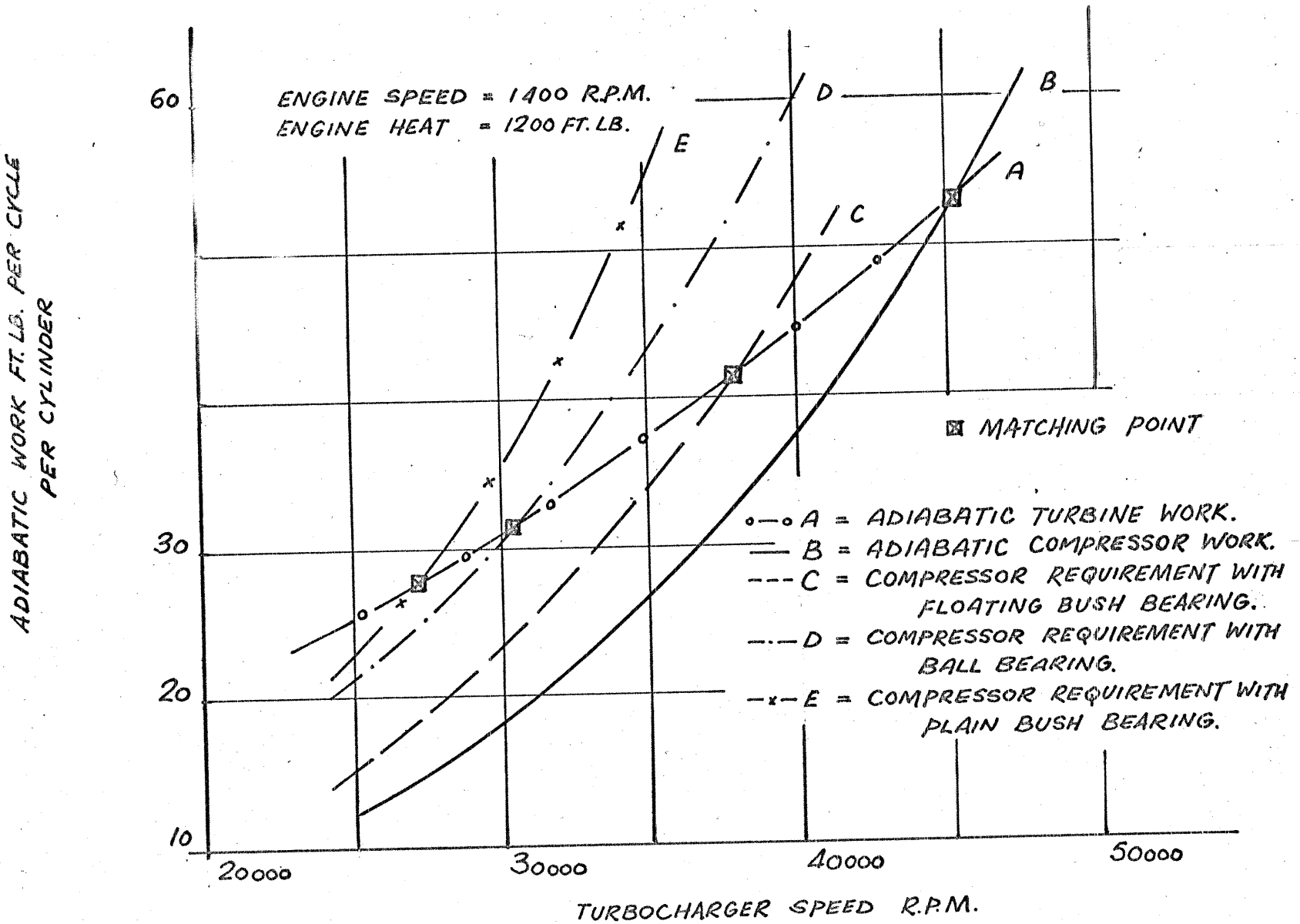



FIG. 4

TABLE 6.

EFFECT OF TURBOCHARGER BEARING FRICTION.

Engine Speed = 1400 R.P.M.

Engine Heat = 1200 Ft.lb/Cylinder/Cycle

TINJ = 355° C.A.

PINJ = 60° C.A.

| Bearing Type and Law B.H.P. = Q(TRPM/1000)$^b$ | W TURB FT.LB. | W COMP FT.LB. | TRPM R.P.M. | AIR DELIVERY LB |
|---|---|---|---|---|
| Plain Bush. a = 2.185 E-4 b = 2.76 | 28.0 | 15.0 | 26800 | 2.849E-3 |
| Ball Bearing. a = 7.05 E-4 b = 2.250 | 32.0 | 19.5 | 31000 | 2.970E-3 |
| Floating Bush a = 2.08 E-4 b = 2.675 | 42.0 | 33.5 | 38750 | 3.184E-3 |
| a = 0.0 b = 0.0 | 52.50 | 52.5 | 45000 | |
| a = 2.47 E-4 b = 2.25 | 27.57 | 22.71 | 32451 | 3.052E-3 |
| a = 4.94 E-4 b = 2.25 | 24.53 | 17.07 | 28983 | 2.938E-3 |
| a = 9.88 E-4 b = 2.25 | 20.05 | 10.23 | 23793 | 2.800E-3 |

7) HEAT TRANSFER.

The heat transfer that takes place between the hot cylinder gases and the relatively cooler 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 that

$$\frac{q}{As} = 8.06E\text{-}5\ (V_p)^{\frac{1}{3}}\ (PT)^{\frac{1}{2}}\ (T - Tw)$$

where

q = Heat transfer rate CHU/sec

As = Surface area exposed to heat transfer $ft^2$

$V_p$ = Mean piston speed ft/sec

P = cylinder gas pressure atm.

T = gas temperature $^oK$

Tw = wall surface temperature $^oK$.

As used in this study the velocity index has been increased from 0.3333 to 0.6 and the constant adjusted so as to give a suitable 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 wall temperature. Tests were conducted for three different values of this constant. The effect that this had on the engine cycle are noted in Table 7.

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 and a reduction of the energy lost to the exhaust gases. This leads to a drop in the turbine work and to a reduction in the air aspirated by the engine. Hence any reduction in the jacket losses would result in a significant improvement in the power output and the thermal efficiency of the engine.

TABLE 7.

EFFECT OF HEAT TRANSFER.

Engine Speed = 1400 R.P.M.

Engine Heat = 1200 Ft.lb/Cylinder/Cycle

TINJ = 355 $^{\circ}$ C.A.

PINJ = 60$^{\circ}$ C.A.

| Eichelberg's Constant | IMEP P.S.I. | W TURB FT.LB. | AIR DELIVERY LB | JACKET LOSSES FT.LB. | EXHAUST GAS LOSSES FT.LB. | 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% |

8) EXHAUST MANIFOLD VOLUME.

The effect of changing the size of the exhaust manifold is illustrated in Table 8. Its effect on the shape of the pressure pulse in the exhaust manifold is illustrated by Fig. 5.

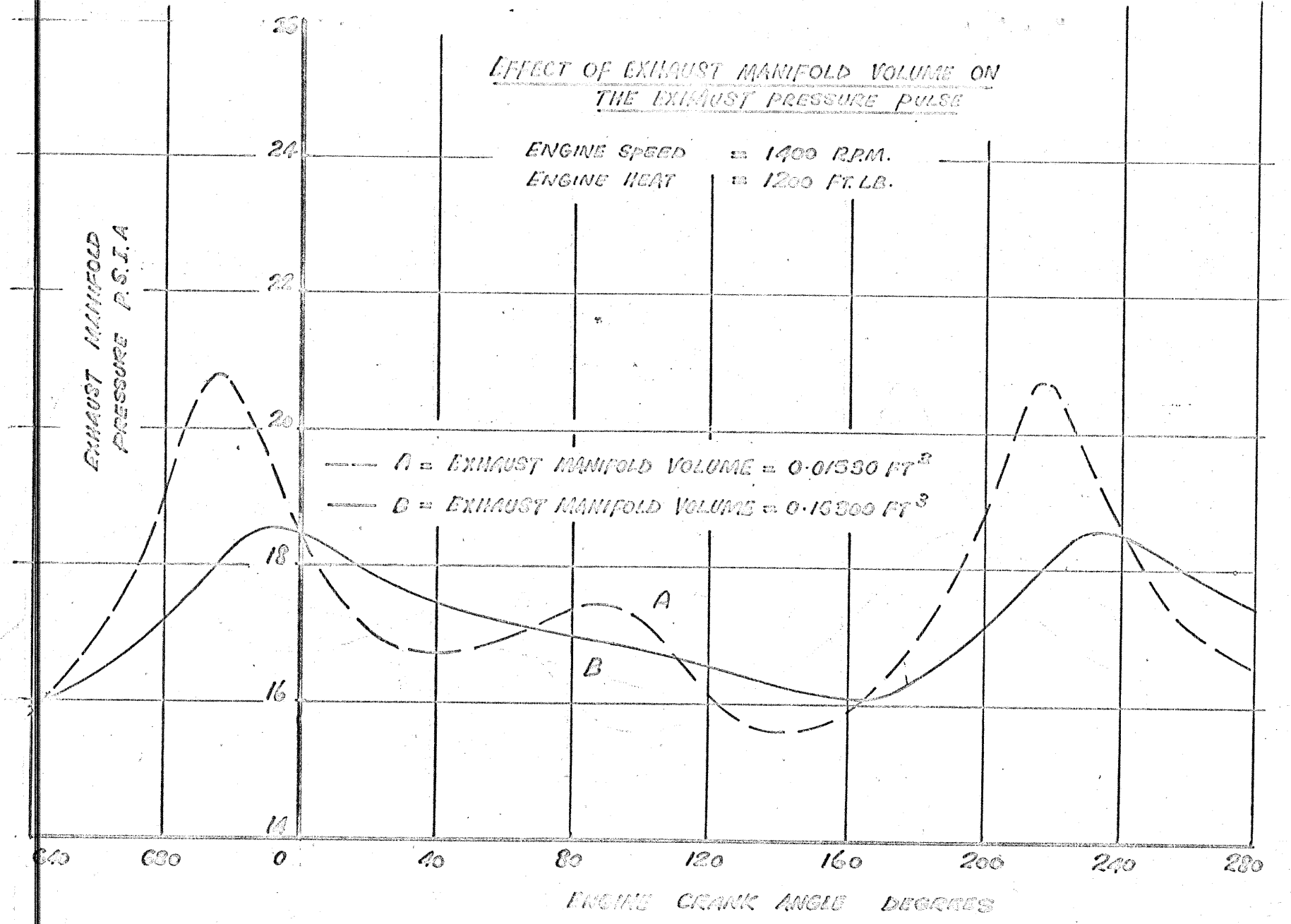

EFFECT OF EXHAUST MANIFOLD VOLUME ON THE EXHAUST PRESSURE PULSE
ENGINE SPEED = 1400 R.P.M.
ENGINE TORQUE = 1200 FT. LB.
EXHAUST MANIFOLD PRESSURE P.S.I.A
26
24
22
20
18
16
14
A = EXHAUST MANIFOLD VOLUME = 0·01530 FT³
B = EXHAUST MANIFOLD VOLUME = 0·16300 FT³
A
B
640
680
0
40
80
120
160
200
240
280
ENGINE CRANK ANGLE DEGREES

FIG. 5

A change in the exhaust manifold volume has a significant effect on the power output of the turbine. As the manifold volume is increased the exhaust pressure pulse is attenuated, the exhaust system tends towards a constant pressure type and the turbine output drops.

These results tend to support previous findings by other investigators who have shown that for low pressure charging a pulsating exhaust system is more efficient than one at constant pressure. They have also suggested that for high levels of supercharging the constant pressure system is superior. As turbocharger data was only available for low pressure ratios, high pressure supercharging tests were not simulated in this series of tests.

TABLE 8.

EFFECT OF EXHAUST MANIFOLD VOLUME.

Engine Speed = 1400 R.P.M.

Engine Heat = 1200 Ft.lb/Cylinder/Cycle

TINJ = 355° C.A.

PINJ = 60° C.A.

| Exhaust Manifold Volume $Ft^3$ | IMEP P.S.I. | W TURB FT.LB. | AIR DELIVERY LB |
|---|---|---|---|
| 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 |

9) COMBUSTION PERIOD (PINJ)

The effect that changes in the combustion period have on the engine are noted in Table 9. As expected late burning results in a lower engine efficiency and an increase in the energy to the exhaust gases. The effect is to bring about a slight increase in the air delivered to the engine but this is insufficient to counteract the power loss due to the drop in the indicated thermal efficiency.

TABLE 9.

EFFECT OF COMBUSTION PERIOD.

Engine Speed = 1400 R.P.M.

Engine Heat = 1200 Ft.lb/Cylinder/Cycle

TINJ = 355° C.A.

| PINJ °C.A. | IMEP P.S.I. | W TURB FT.LB. | AIR DELIVERY LB. | EXHAUST GAS LOSSES FT.LB. | 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% |

10) & 11) ENGINE SPEED AND FUELLING RATE (HEAT).

The behaviour of the engine to changes in the engine speed and fuelling rate are given in Table 10. These results were obtained with a fixed heat release pattern and fixed values of TINJ and PINJ.

It was possible to alter the heat release pattern to suit each speed/load condition but this would in effect have been pre-supposing the results and altering the pattern to fit. It was for the want of a better method that the heat release pattern was unchanged over the range of these tests.

TABLE 10.

EFFECT OF ENGINE SPEED AND FUELLING RATE.

TINJ = 355°C.A.

PINJ = 60° C.A.

| ENGINE SPEED R.P.M. | ENGINE HEAT FT.LB. | IMEP P.S.I. | PEAK TEMPERATURE °R | PEAK PRESSURE P.S.I.A. | JACKET LOSSES | INDICATED THERMAL EFFICIENCY |
|---|---|---|---|---|---|---|
| 1200 | 1200 | 108.2 | 3089 | 1113 | 13.1% | 51.4% |
| 1400 | 1200 | 107.0 | 3094 | 1126 | 12.69% | 50.8% |
| 1400 | 1650 | 146.0 | 3604 | 1334 | 14.46% | 50.6% |
| 1600 | 1200 | 105.8 | 3108 | 1143 | 12.50% | 50.3% |

## CONCLUSIONS.

The tests reported here were a set of preliminary tests that were run to check whether the proposed model was reasonable and what further refinements were necessary. It was appreciated that in some cases the treatment was not entirely satisfactory but the advantage of this method of approach is that parts of the analysis can be changed easily.

The results have indicated that, despite certain flaws, the model is basically sound. It is capable of predicting many aspects of the performance of the engine to within reasonable limits of accuracy. Its qualitative response to a change in many operating variables is representative of the physical system. For example, its behaviour to the variation of TINJ and PINJ is similar to an actual engine. A further example is the effect of valve size on the mass aspirated by the engine.

However, the tests have revealed weaknesses and defects. These are listed and their possible causes are discussed.

a) The predicted indicated thermal efficiency is high. This is due to neglecting the rate of change of specific heat of the working fluid in the basic equations. (see appendix I). By the use of the same semi-perfect gas but with the inclusion of the term for the rate of change of specific heat the same model gives a much more realistic value of indicated thermal efficiency. In a particular case the effect was to lower the thermal efficiency from 51% to 47%. It is also possible that further improvement can

be achieved by allowing the working fluid to change composition from pure air at induction to combustion products after combustion. The thermodynamic properties of the instantaneous gas composition would be used at each point in the cycle.

b) The model utilises a preset heat release pattern. In practice the heat release pattern depends not only on the injection schedule or the gas state at the time of injection, but also on the temperatures and pressures of the gas as the combustion proceeds. Incorporating these effects would require a combustion model fundamentally based on the fuel properties. It is then and only then that it would be possible to make a realistic study of the effect of varying parameters such as TINJ and PINJ.

c) The computation of the heat transfer by Eichelberg's equation is suspect, but the alternatives are also open to objection. However, there is one refinement that is immediately possible. At present a preset value of the gas side cylinder wall temperature is used. A better approach would be to define a mean wall temperature from a heat balance between the gas side and the water side of the heat transfer surfaces. This approach could later be extended to a set of heat transfer surfaces, each with its own gas side temperature and a heat balance over the region covered by that surface. This would be necessary to take into account the effect of hot spots at valves, etc.

It is obvious that the model developed is not completely representative of the physical system but even in its unrefined form

it promises to be a useful tool for engine development.

A more refined model is in the final stages of development. It incorporates the modifications suggested in this report but a comprehensive series of tests is required to show its true value. When these results are available they will be reported in a publication of the Advanced School of Automobile Engineering (College of Aeronautics, Cranfield).

APPENDIX I.

SYSTEM THERMODYNAMICS.

The thermodynamic equations of the system are derived on the following assumptions:

a) The working fluid has properties similar to air and is assumed to be a semi-perfect gas, i.e. it satisfies the two conditions

(i) $PV = MRT$

(ii) $Cv = Cv(T)$, a function of temperature

It is further assumed that $\left(\frac{\partial Cv}{\partial T}\right)_p$ is insignificant and is neglected.

b) The processes taking place within the control volume are quasi-static.

c) The fluid within each control volume is homogenous, isobaric and isothermal.

By the first law of thermodynamics

$$Hi - Ho = U + W - Q$$

Hi = enthalpy added to the system by incoming mass

Ho = enthalpy lost from the system by outgoing mass

U = change in internal energy of the system

W = work output from the system.

Q = heat input to the system.

By the substitution of $H = C_p T = k\,Cv\,T$ and the application of conservation of mass and the state law for the gas, the state of the

gas in the control volume can be represented by the differential equations:-

$$\frac{dT_c}{dt} = \frac{R\,T_c}{P_c Cv_c V_c}\left\{(T_i Cv_i ki - T_c\,Cv_c)\frac{dMi}{dt} - R\,T_o\frac{dMo}{dt} - \frac{P_c dV_c}{dt} + \frac{dQ_c}{dt}\right\}$$

$$\frac{dP_c}{dt} = \frac{P_c}{T_c}\frac{dT_c}{dt} + \frac{R\,T_c}{V_c}\left\{\frac{dMi}{dt} - \frac{dMo}{dt}\right\} - \frac{P_c}{V_c}\frac{dV_c}{dt}$$

where

i = refers to incoming quantities

o = refers to outgoing quantities

c = quantities in the control volume.

N.B. Because of assumption (c) outgoing quantities have the same properties as those within the control volume.

In addition to the two basic equations there are five other auxiliary equations to be solved for mass inflow, mass outflow, heat input, heat losses and $dV_c/dt$. These are explained in detail in reference 1. The complete solution of the state of the gas within each control volume involves the solution of seven simultaneous first order differential equations.

The thermodynamic equations were used in the above form to obtain the results given in this report. It was soon evident that some assumptions were not justified and that if greater accuracy was required these equations would have to be rewritten.

The two assumptions which were invalid are:

i) The rate of change of specific heat is negligible.

ii) $H = C_p T = k C_v T$.

In the more accurate form the two state equations are:

$$\frac{dT_c}{dt} = \frac{R T_c}{P_c V_c \left[C_v + T\left(\frac{\partial C_v}{\partial T}\right)\right]} \left\{ (Hi - Cv_c T_c)\frac{dMi}{dt} - (Ho - Cv_c T_c)\frac{dMo}{dt} - P_c \frac{dV_c}{dt} + \frac{dQ}{dt} \right\}$$

$$\frac{dP_c}{dt} = \frac{P_c}{T_c}\frac{dT_c}{dt} + \frac{R T_c}{V_c}\left\{\frac{dMi}{dt} - \frac{dMo}{dt}\right\} - \frac{P_c}{V_c}\frac{dV_c}{dt}$$

The effect that these changes had on the performance of the engine for a particular test condition are given below.

Engine Speed = 1400 R.P.M.

Engine Heat = 1200 ft.lb/Cylinder/Cycle

TINJ = 355° C.A.

PINJ = 60° C.A.

| | IMEP P.S.I. | W TU/B PER UNIT MASS OF EXHAUST GAS FT.LB/LB | PEAK TEMPERATURE °R | PEAK PRESSURE p.s.i.a. | EXHAUST GAS LOSSES | INDICATED THERMAL EFFICIENCY |
|---|---|---|---|---|---|---|
| Simple Equation | 107.0 | 78.6E3 | 3094 | 1126 | 36.5% | 50.8% |
| More Complex Equation | 98.2 | 105.0E3 | 2602 | 1027 | 43.8% | 46.6% |

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 , | EEKP , | TFC , | TWEAP , |
| SFI , | SFE , | SFC , | SFT , |
| TINJ , | PINJ , | T WALL, | N7, TEST, FLEAK |

where:

STROK = Engine stroke ft.

BORE = Engine cylinder bore ft.

ECC = Eccentricity of connecting rod or l/r ratio

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 p.s.i.a.

HEAT = Fuel heat input per cylinder per cycle FT.LB.

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 turbocharger bearing friction law.

TFEXP = Speed index for turbocharger bearing friction law.

The turbocharger bearing friction is assumed to be given by

$$\text{Bearing Horse Power} = TFC \left(\frac{TRPM}{1000.0}\right)^{TFEXP}.$$

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 turbocharger 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 factor. In addition, the non-dimensional speed is also divided by the quare 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 degrees C.A.

T WALL= Gas side cylinder wall temperature $^{o}R$.

N7 = Sets the number of integration steps before the first output. Normally N7 0 so that there is no output till the iterations have reached a stable solution. Then N7 is automatically set equal to zero and results are output 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.

N.B. All the numbers except N7 are real and should be written with a decimal point. N7 is an integer and should not contain a decimal point.

APPENDIX III.

STARTING VALUES.

This input stream is used to preset 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, | $T_1$, | $T_2$, | $T_3$ |
| $T_4$, | $T_5$, | $T_6$, | $T_7$, |
| $T_8$, | $T_9$, | $P_1$, | $P_2$, |
| $P_3$, | $P_4$, | $P_5$, | $P_6$, |
| $P_7$, | $P_8$, | $P_9$. | |

TRPM = Turbocharger speed R.P.M.

$T_1$ to $T_6$ = Cylinder gas temperatures $^{o}R$.

$P_1$ to $P_6$ = Cylinder gas pressures p.s.i.a.

$T_7$, $P_7$ = Inlet manifold temperature $^{o}R$ and pressure p.s.i.a.

$T_8$, $P_8$, $T_9$, $P_9$ = Temperatures $^{o}R$ and pressures p.s.i.a. in the two halves of the exhaust manifold.

APPENDIX IV.

THE TABLE "FUNCTION".

The table function was originally written by Mr. Palmer of the Department of Propulsion, College of Aeronautics. Its purpose was to stre in an easily accessible tabular form up to three dependent variables as functions of one or two independent parameters. During this study, the "Function" has been written as a Fortram IV subrantive which is capable of storing 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 only limited 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.

| $\min.x_1$, | $\min.x_2$, | $\max.x_1$, | $\max.x_2$, |
|---|---|---|---|
| $x_1$ , | $x_2$ , | $y_1$ , | $y_2$ , |
| 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 subrantive call of the type

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

where appropriate values have been given to I, X1 and, where relevent, X2. The values of $y_1$ (X1,X2) and $y_2$ (X1,X2) are available in Y1 and Y2.

LIST OF REFERENCES.

1) Shroff, H.D.
"A Digital programme for the performance of a Diesel Engine".
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"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.