THEORETICAL AND EXPERIMENTAL STUDY OF INLET MANIFOLD PRESSURE PROFILES USING COMPENSATED TRANSDUCERS

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Abstract

Computer simulation and experimental findings on single cylinder four-stroke cycle engine were done with a view of getting more realistic inlet manifold pressures. Since there is a direct relationship between the inlet pressure profiles and the volumetric efficiency of the engine, then there is a possibility of utilizing the knowledge of these profiles to enhance better engine air flow characteristics and hence improvement of engine performance.

The simulation program is based on hometropic process with constant specific heats where the results were validated using experimental findings of the engine with varying engine speeds.

Experimental results are obtained by measuring manifold pressures close to the engine inlet valve using a set of pressure transducers whose response characteristics are determined experimentally in advance. The knowledge of the response characteristics is essential as it enables a compensation circuit to be constructed to eliminate the amplitude attenuation and phase lag of the output signal.

The results show that it is possible to use cheap compensated pressure transducer to achieve high accuracy of pressure measurements.

Nomenclature

Symbol	Definition	Units
a	Speed of sound	ms^{-1}
a_A	Speed of sound at reference pressure	ms^{-1}
A	Non dimension speed of sound	-
AVO	Air Valve Open	$(^o)$
AVC	Air Valve Close	$(^o)$
c	Disturbance propagation velocity	ms^{-1}
C	Capacitance	Farad

CR	Compression ratio	-
N	Engine speed	s^{-1}
p	Pressure	Nm^{-2}
PET	Piezo Electric Transducer	-
R	Resistance	Ohm
t	Time	s
T	Temperature	(^{o}K)
TDC	Top Dead Center	-
U	Non-dimensional velocity	-
x	Distance	m
Z	Non dimensional time	-

1.0 Introduction

The proper design of manifolds for four stroke cycle engine has been topical since Nicholas Otto produced the first modern engine in 1876. Gas flow in the adjoining ducts is non steady which results in a propagation of pressure waves in the inlet and outlet manifolds. The inlet pressure profiles (near to the inlet valve) governs the volumetric efficiency of the engine and hence performance.

Some mathematical models have emerged to analyze the formation and propagation of these waves for the prediction of the engine performance. The graphical procedure for the solution of the characteristic equations was employed in the early days(7). With the introduction of high speed computers recently, algorithms have replaced the graphical method(8).

The mathematical model used in this paper assumes a hometropic, unsteady one dimension flow with constant specific heats. The modelling algorithm is written in Fortran 77 for computer application. A single cylinder, four stroke cycle engine model is used to demonstrate the application of the modelling. In order to evaluate the accuracy of the simulation program, the predicted results are compared with experimental findings. The experimental data is obtained by measuring the instantaneous pressures in the inlet manifold during the steady state running of the engine. These pressures are picked up by pressure transducers positioned close to the engine inlet valve.

Most of transducers have a finite response characteristics; the measured values are not necessarily the true instantaneous values. At high frequencies the measured values are attenuated and lag behind the actual ones. In order to correct for such deficiencies the response characteristics of the pressure transducers are also studied. A compensating circuit is included in the instrumentation to cater for correction of such deficiencies.

2.0 Modelling Theory

2.1 Unsteady Flow Modelling

The propagation of pressure waves along the engine pipe system in response to events inside the cylinder as the valves close and open is the basic process which determines the engine breathing behavior. In hometropic flow, this process can be modelled mathematically by the three conservation equations which govern the one dimensional, unsteady compressible fluid flow:

Continuity $$\frac{1}{\rho}\frac{\partial\rho}{\partial t} + \frac{u}{\rho}\frac{\partial\rho}{\partial x} + \frac{\partial u}{\partial x} = 0 \qquad 2.1$$

Momentum $$\frac{1}{\rho}\frac{\partial p}{\partial x} + \frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} = 0 \qquad 2.2$$

Constant entropy $$\frac{p}{p_{ref}} = \left(\frac{a}{a_{ref}}\right)^{2\gamma/(\gamma-1)} \qquad 2.3$$

Since there is no analytical method of solution for the above conservation equations so far, they are then rearranged and transformed into a set of ordinary differential equations and solved by numerical technique based on the method of characteristics. This method relies upon finding certain characteristic directions in the flow field along which the discontinuities in the derivatives of the fluid properties may propagate.

The procedure for the solution in question is that the pipe is segmented into a number of equal lengths and the calculations are advanced in a series of time steps. At each mesh point, if required,the thermodynamic parameters of interest can be evaluated in the time-space grid. Through this way the motion resulting from the superposition of left moving and right moving pressure waves can be calculated over the cylinder and manifold system. Thus, the changes in density, mass flow rate, temperature and pressure in the cylinder can be evaluated.

Thus, along curve of slopes of wave characteristic

$$\frac{dp}{dt} = u \pm a \qquad 2.4$$

$$\frac{dp}{dt} = \pm \rho a\frac{du}{dt} \qquad 2.5$$

where eq (2.5) is the compatibility equation for eq (2.4). Similarly along a curve of slope of the particle

$$\frac{dx}{dt} = u \qquad 2.6$$

$$\frac{dp}{dt} = a^2 \frac{d\rho}{dt} \qquad 2.7$$

where eq (2.7) is the compatibility equation for eq (2.6)

In order to obtain a wave pattern solution with time, boundary conditions are required which relate the conditions outside the pipe with those inside (subroutines are derived in Ref 6). For this method of solution, it is convenient to convert the thermodynamic variables ρ and p to a and a_{ref}. These variables are considered to be a measure of entropy in the fluid flow.

2.2 The Non Dimensional Characteristic and Riemann Variables

In the previous paragraphs, the physical interpretation of results has been in terms of pressure and velocities. A direct representation of pressure can be obtained for hometropic flow if the velocities are divided by reference speed of sound, a_{ref}. Non-dimensionalizing is extended to distance, x and time t by introducing a reference length, L_{ref}.

Thus, $A = \frac{a}{a_{ref}}$; $U = \frac{u}{a_{ref}}$; $X = \frac{u}{L_{ref}}$; $Z = \frac{a_{ref}}{L_{ref}} t$

The original characteristic equations (eqns 2.4 and 2.6 respectively) were

$$\frac{dx}{dt} = u \pm a \qquad 2.8$$

and $$\frac{da}{du} = \mp \frac{\gamma - 1}{2} \qquad 2.9$$

In non dimensional form these become

$$\frac{dX}{dZ} = U \pm A \qquad 2.10$$

$$\frac{dA}{dU} = \mp \frac{\gamma - 1}{2} \qquad 2.11$$

and $$A = \frac{a}{a_{ref}} = \left(\frac{p}{p_{ref}} \right)^{(\gamma - 1)/2\gamma} \qquad 2.12$$

Consider the expression $\frac{dA}{dU} = - \frac{\gamma - 1}{2}$

Integrating with $A = \lambda$ at $U = 0$ we get

$$\lambda = A + \frac{\gamma - 1}{2} U$$

Thus for λ characteristic

$$\frac{dX}{dZ} = U + A \qquad 2.13$$

$$\frac{dA}{dU} = - \frac{\gamma - 1}{2} \qquad 2.14$$

and $$\lambda = A + \frac{\gamma - 1}{2} U \qquad 2.15$$

Similarly for β characteristic

$$\frac{dA}{dU} = \frac{\gamma - 1}{2}$$

Integrating with $A = \beta$ at $U = 0$ we get

$$\frac{dX}{dZ} = U - A \qquad 2.16$$

$$\frac{dU}{dA} = \frac{\gamma - 1}{2} \qquad 2.17$$

and $$\beta = A - \frac{\gamma - 1}{2}U \qquad 2.18$$

Note that λ is constant along λ characteristic; the same for β. From equations (2.15) and (2.18) we get

$$A = \frac{\lambda + \beta}{2} \quad \text{and} \quad U = \frac{\lambda - \beta}{\gamma - 1}$$

For convenience let $$a = \frac{3-\gamma}{2(\gamma-1)} \qquad 2.19$$

and $$b = \frac{\gamma+1}{2(\gamma-1)} \qquad 2.20$$

Thus from equations (2.19) and (2.20) we have for λ characteristic

$$\left(\frac{dX}{dZ}\right)_\lambda = b\lambda - a\beta \qquad 2.21$$

Similarly for β characteristic

$$\left(\frac{dX}{dZ}\right)_\beta = a\lambda - b\beta \qquad 2.22$$

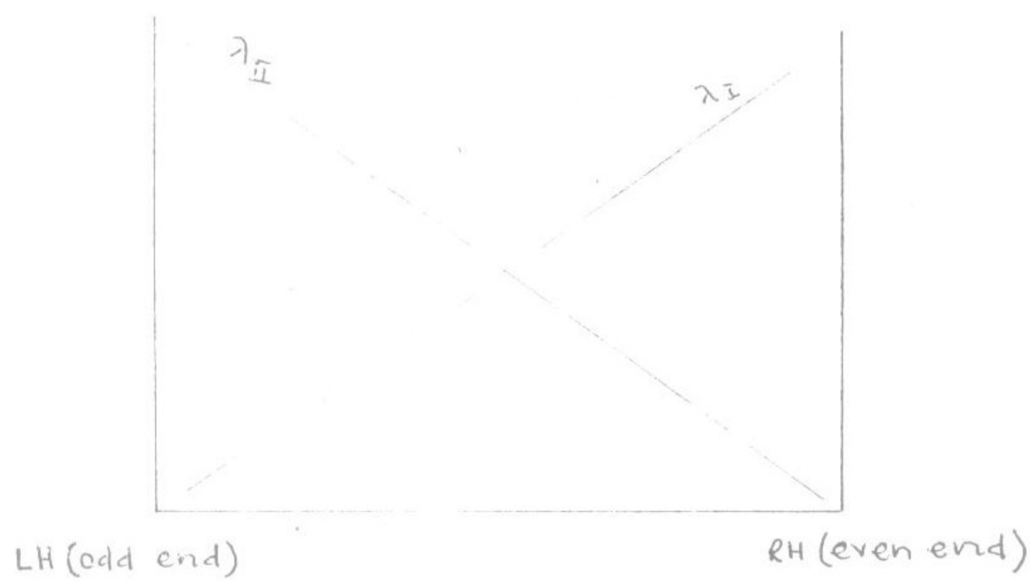

Fig 2.1 Generalized characteristic (λ_I and λ_{II})

Let $\lambda = \lambda_I$ for positive x-axis from left to right

$\lambda = \lambda_{II}$ for positive x-axis from right to left

$$\text{i.e } U = \frac{\lambda_{I} - \lambda_{II}}{\gamma - 1} \qquad 2.23$$

$$A = \frac{\lambda_{I} + \lambda_{II}}{2} \qquad 2.24$$

For the λ slope, $\lambda = \lambda_{I}$ and $\beta = \lambda_{II}$

$$\text{Thus } \left(\frac{dX}{dZ}\right)_{\lambda_{I}} = \frac{\gamma+1}{2(\gamma-1)}\lambda_{I} - \frac{3-\gamma}{2(\gamma-1)}\lambda_{II} \qquad 2.25$$

Similarly for λ_{II} slope we have $\lambda=\lambda_{II}$ and $\beta=\lambda_{I}$

$$\text{Thus } \left(\frac{dX}{dZ}\right)_{\lambda_{II}} = \frac{3-\gamma}{2(\gamma-1)}\lambda_{II} - \frac{\gamma+1}{2(\gamma-1)}\lambda_{I} \qquad 2.26$$

where λ_{I} and λ_{II} are commonly known as Riemann Variables.
The form of equations (2.25) and (2.26) are the basis for the main program for the engine simulation ENGPROG.

2.3 Numerical Solution of the Characteristic Equations

The numerical solution of the characteristic equations is performed wholly in the position diagram, i.e X-Z field. The most convenient form is that which uses rectangular mesh which has the advantage that the conditions at a particular pipe position are determined over a desired period corresponding to measurement of time history of variable at a fixed location. The procedure is that the grid pattern is fixed in the X-direction but the Z-coordinate of the grid lines are adjusted according to the stability criterion (sect 2.3.1) for the numerical solution of quasi-linear hyperbolic partial differential equations. This results in irregular time intervals on spatially fixed lines. The regular pattern of rectangles are fixed by the fact that the grid proportions $\Delta Z:\Delta X$ are fixed at each time interval, ΔZ. At any time Z = Z1, Riemann's variables λ_{I} and λ_{II} are evaluated at each mesh point. The time interval ΔZ for the next time step is evaluated from λ_{I} and λ_{II} at time Z. The next time $Z=Z1+\Delta Z$ is then determined and the procedure repeated.

2.3.1 Stability Criterion

The mesh proportions are selected such that $\frac{\Delta Z}{\Delta X} = \frac{1}{|U|+A}$

This means that the mesh proportions will vary throughout the flow field in the Z-X diagram. In practice the procedure is to fix the length ΔX and allow ΔZ to vary in each time step.Under this condition the values of A and U are determined at each mesh point and ΔZ is selected for the minimum value of $\Delta X/(|U|+A)$. At any time step Z, the values of λ and β at each mesh point are evaluated from the previous values of λ and β. The assumption is that the values of λ and β vary linearly along the mesh length in the X- and Z-directions.

Thermodynamic quantities are then evaluated from λ_{I} and λ_{II} using the relationship:

$$\left(\frac{p}{p_{ref}}\right)^{(\gamma-1)/2\gamma} = \frac{\lambda_I + \lambda_{II}}{2} \qquad 2.27$$

and $$\frac{u}{a_{ref}} = \frac{\lambda_I - \lambda_{II}}{\gamma - 1} \qquad 2.28$$

2.4 Program Organization

The computer solution of the previously discussed equations is organized in such a way that it caters for the subroutines which deal with the boundary conditions. The calculations in this routines determine conditions at all interval mesh points of the duct section. At the duct ends, the boundary equations are used in conjunction with the wave equations to establish the values of λ and β. Depending upon particular engine, the form of the boundary conditions will vary widely. The program structure is therefore arranged so that the calculation of the quantities λ and β is carried out at separate boundary operations. In this way, only boundary conditions of particular interest to the problem at hand need be included in the program.

The master program portion is principally concerned with the organization of the data describing a particular engine under investigation and with printing information about the solution. A flow chart of this control program (ENGPROG) is shown in Fig 2.2. Data read into this segment describe the engine under investigation in terms of engine geometry, valve area, cylinder pressure and temperature when the inlet valve opens. The thermodynamic quantities p and T are converted into λ and β values and stored in mesh arrays. For each boundary calculations the known values of λ and β at the boundary mesh points of each duct section are then used to transfer the collected information to the boundary routines for subsequent calculation of λ and β values.

2.4.1 Program Input Data

The following are the required engine input data for the inlet manifold. The geometrical data listed below are those of the engine under investigation, while the ambient conditions are those which were recorded during the experimentation. The input data are grouped in three categories:

Group I: Input Data

Initial pressure in the inlet pipe	1.013bar
Cylinder pressure at inlet valve open	1.1bar
Cylinder temperature at inlet valve open	473^{o}K
Reference pressure	1.013bar
Number of meshes	12
Ratio of specific heats	1.4
Ambient pressure	1.013bar
Ambient temperature	295^{o}K
Engine speed	(variable)

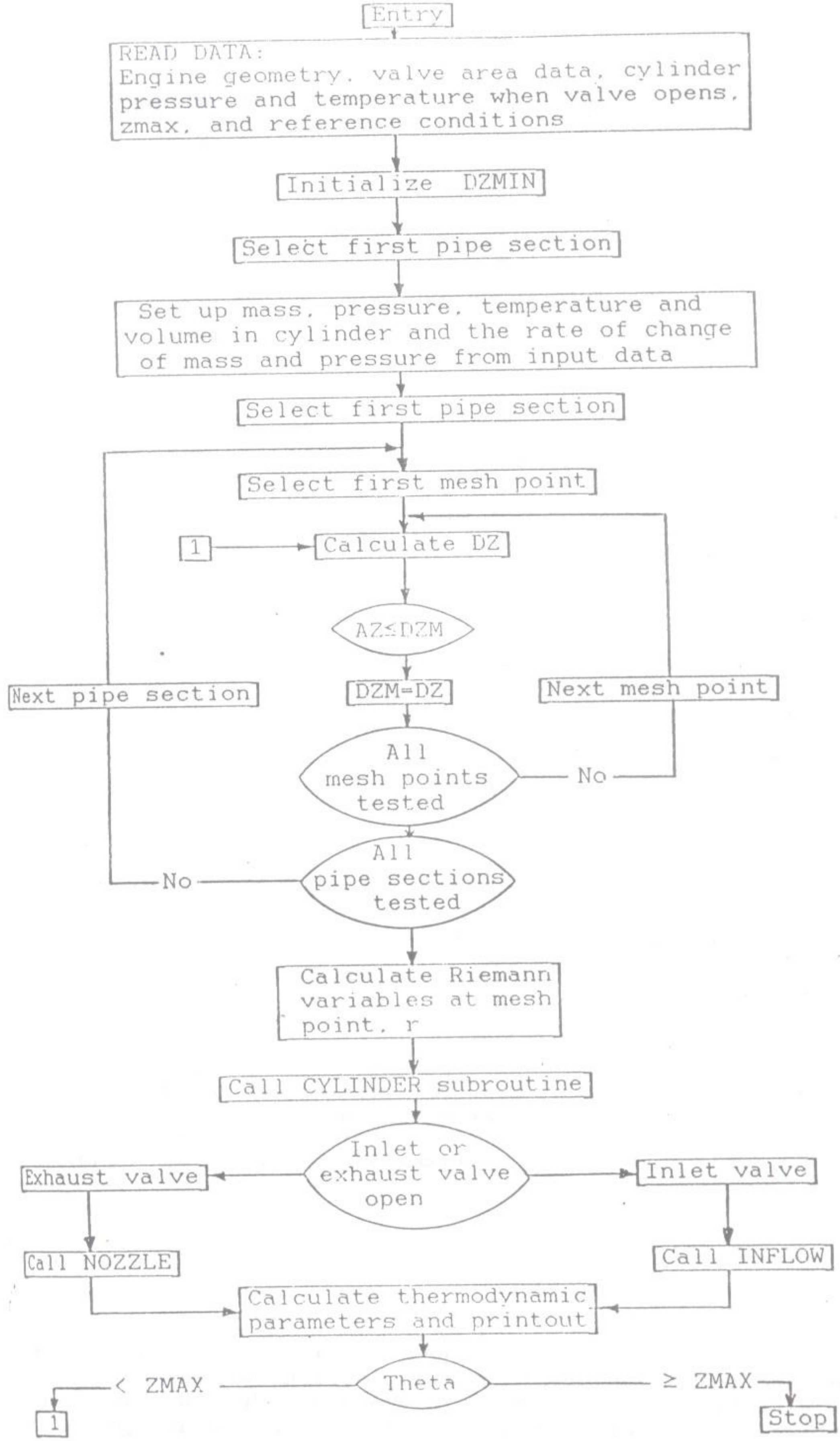

Fig 2.2 Control program (ENGPROG)

Group II: Input Data (Specific to the Engine)

Air valve open after TDC power stroke	355.5^{o}
Cycle of operation	4 (stroke)
Cylinder diameter	0.08m
Stroke length	0.11m
Connecting rod length	0.22m
Inlet pipe diameter	0.03m
Inlet pipe length	0.745m
Nominal compression ratio	16.5
Number of points on the "area curve"	245
Angle from opening including zero degree	220^{o} **
Valve area	$f(\theta)$

Group III: Output Data

Number of output locations	1
Location expressed in mesh points	2
Duration of run in degrees crank angle	720^{o}

** See Fig 3.1

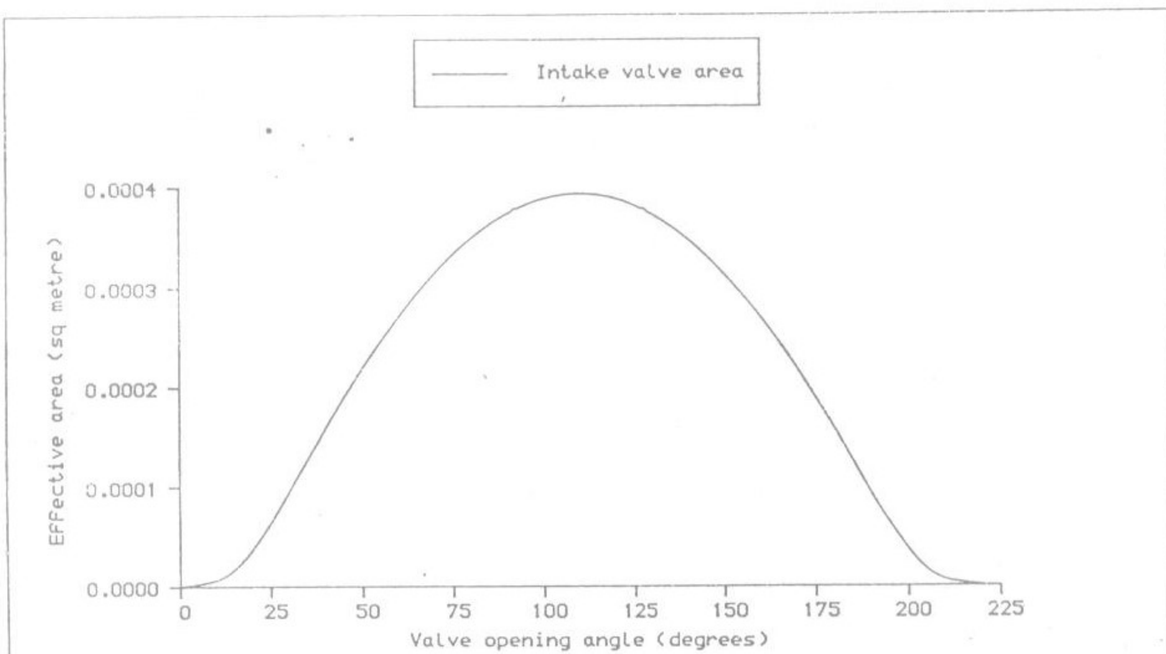

Fig 3.1 | Intake valve opening characteristic (Petter Diesel Engine)

Fig 3.1 Intake valve opening area

3.0 Instrumentation and Experimentation

Instrumentation layout is set as shown in Fig 3.2. The air box eliminates the pulsation due to piston motion while the inclined manometer give the air pressure to the engine. For the reason that the resistive transducer introducers an attenuation and phase lag of the output signal, a compensating circuit (R-C circuit) was incorporated in the instrumentation to eliminate these deficiencies.

The RC circuit was modeled as a first order transfer function, i.e the transducer is of first type instrument. Fig 3.3 shows the compensating configuration while Fig 3.4 depicts the actual compensating (lead) circuit.

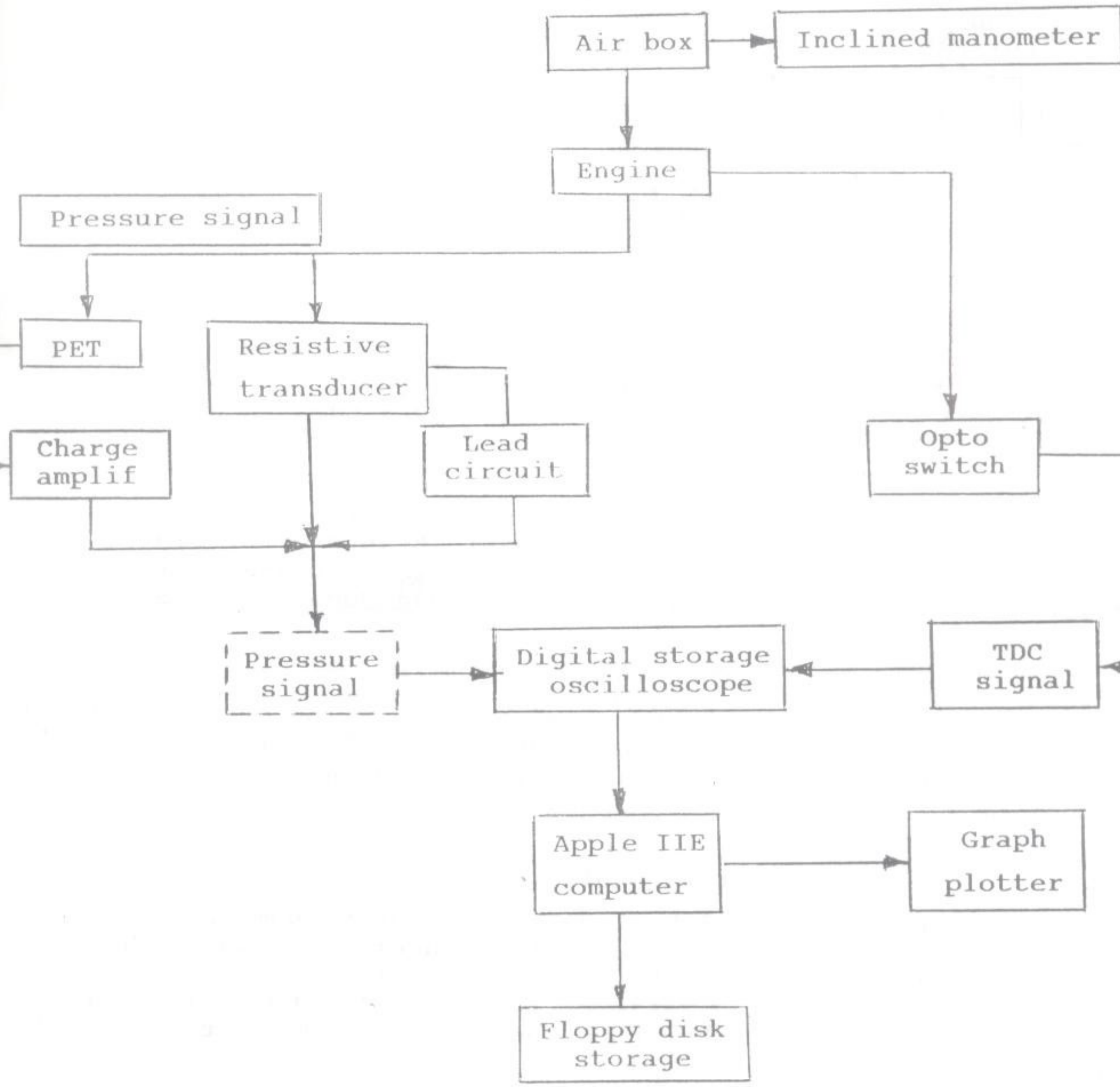

Fig 3.2 Instrumentation and data collection

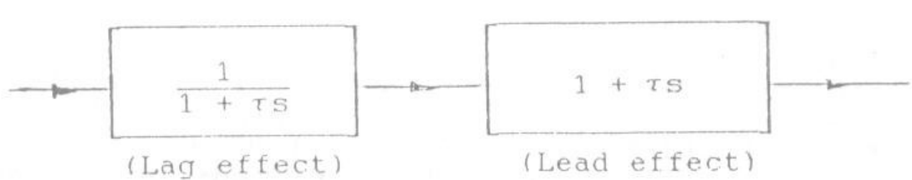

Fig 3.3 Lag-Lead configuration

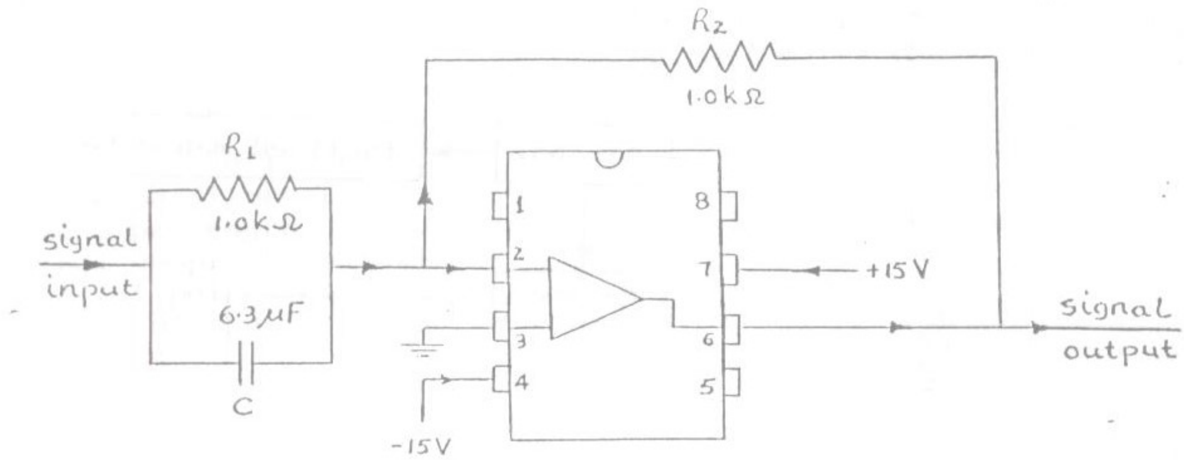

Fig 3.4 Actual Lead-circuit

3.1 Pressure Signal

The pressure variation in the inlet manifold close to the inlet valve is picked up by a set of two pressure transducers located at a common point. These transducers are piezo transducers (Type Kiestler KIAG SWISS 742) and a resistance transducer (Type SA 15).

3.2 TDC Signal

The TDC signal is detected by means of reflective opto-switch which picks the signal as the flywheel rotates. The TDC gives the interval of one complete engine cycle and also enhances the manipulation of accurate engine speed.

3.3 Calibration of Transducers

All the transducers were calibrated by means of vacuum pump. The calibration factor of the resistance transducer alone was found to be 0.204bar/volt while that which passes through the RC-circuit becomes 0.208bar/volt. The ratio of the later to the former gives the gain factor. On the other hand the calibration factor of the PET was found to be 0.219bar/volt.

4.0 Description and Discussion of Results

Experimental and simulation results of the engine are discussed and compared with a view of getting better representation of pressure histories.

4.1 Inlet Manifold Pressures

Experimental and predicted manifold pressures (Figs 4.1, 4.2 and 4.3) reflect the cylinder filling characteristics as a function of engine speed with a fixed length of intake pipe. These profiles have common features which are discussed below:

1) After the valve closure the amplitude dies rapidly for remainder of the cycle

2) The pressure profiles through the RC-circuit and the PET have a lot of improvement in terms of amplitude attenuation and phase lead as compared to those direct from the resistive transducer
3) There are more irregularities in the signal through the RC-circuit than those of PET.This might be due to the property of the RC-circuit as it is basically a differentiator
4) After the valve closure, the experimental amplitudes die faster with time than those of predicted ones. This discrepancy might be due to the omission of friction and heat transfer in the modeling assumptions

5.0 Conclusion and Recommendations

5.1 Measuring Instruments

Through the knowledge of response characteristics of the measuring instruments it has been demonstrated that better results can be obtained without embarking on expensive instruments. The PET was employed in the experimentation to act as a comparator to the RC-circuit.

5.2 Application of the Simulation Program

With the simulation program in the present form, the engine thermodynamic parameters can be studied systematically as it is possible to "run" the engine on the computer and get greater details than experimental measurements. Validation of the program has been demonstrated by comparing experimental and predicted pressure profiles. The reasonably close agreement demonstrates the validation of the modeling accuracy of unsteady gas flow in the engine cylinder with the piping system. Better results could be obtained by modifying the program by incorporating a combustion model and taking into account the friction in the engine system.

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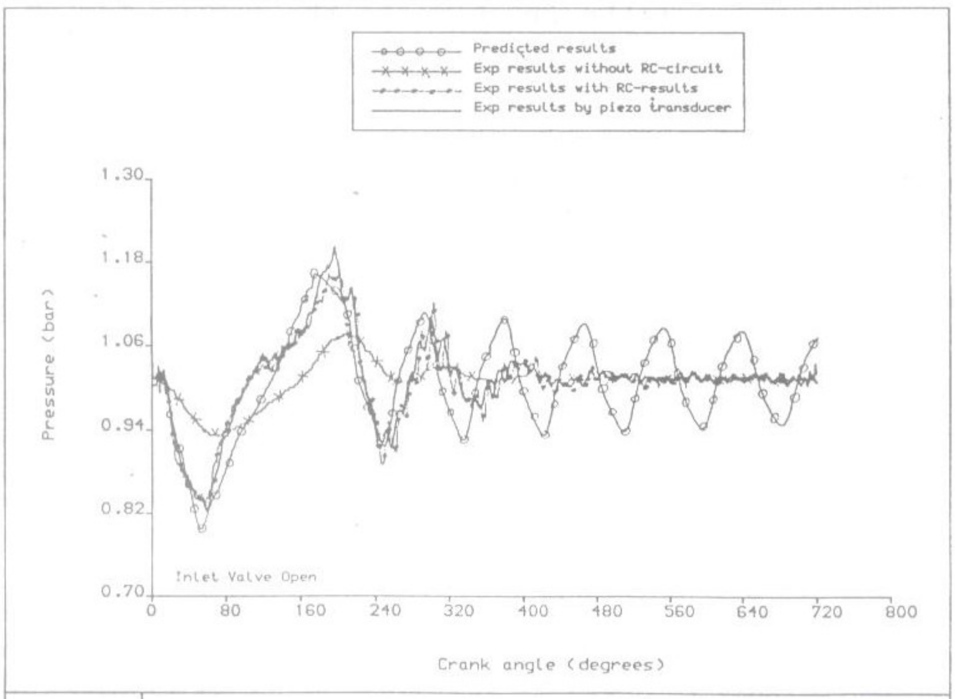

FIG. 4.1 Manifold pressure (N=2090rpm; L=0.745m)

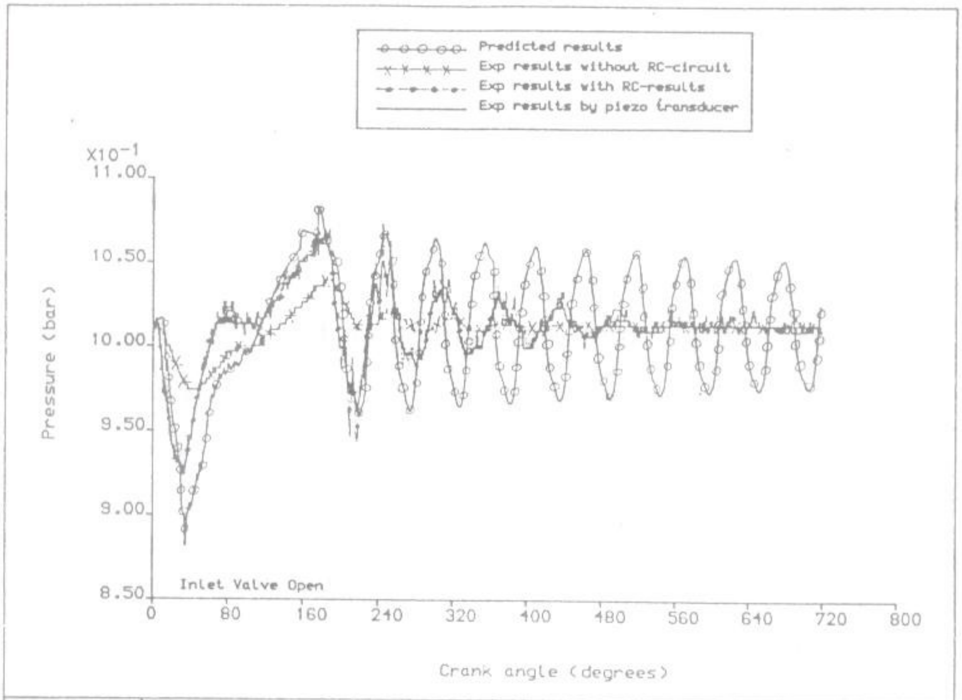

FIG. 4.2 Manifold pressure (N=1300rpm; L=0.745m)

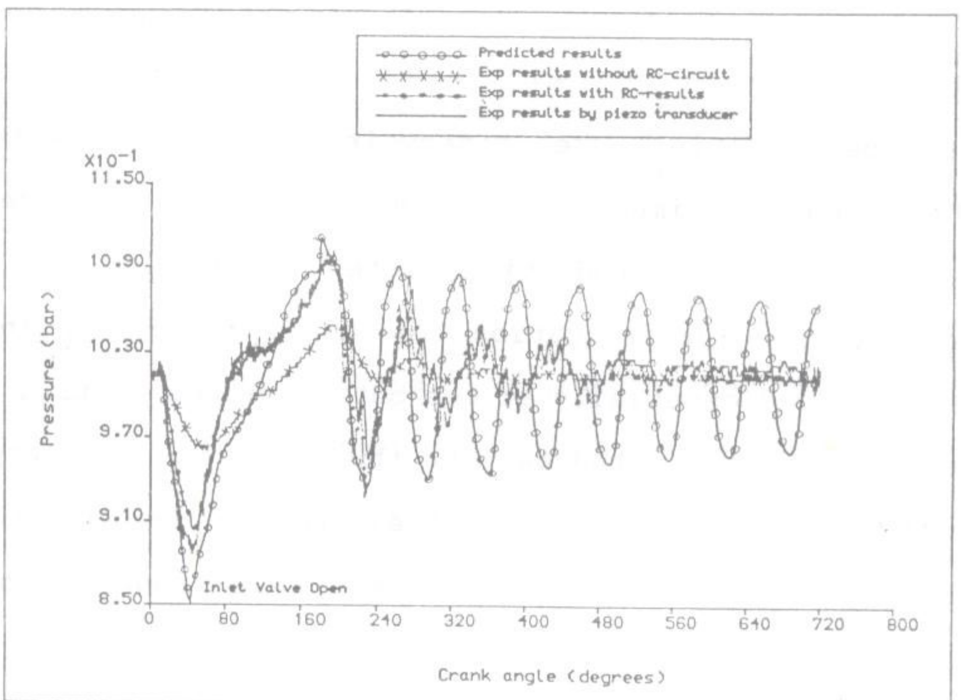

FIG. 4.3 Manifold pressure (N=1580rpm; L=0.745m)