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AN EXPERIMENTAL METHOD FOR DESIGN AND PERFORMANCE EVALUATION OF A HERMETIC COMPRESSOR

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ABSTRACT

Experimental determination of the effects of parameters which can be used as guidelines in design modifications and the evaluation of their effect on the performance and EER of a hermetic compressor is proposed. The Pressure-crank angle record obtained during standard calorimeter test offers the necessary data. Displacement of the valves is not measured explicitly but the opening and closing of the valves are noted as the points of discontinuity on the continuous polytropic processes. Design changes of the valves were conceived and evaluated using this method. An improvement in performance and energy efficiency ratio was obtained with the modifications adopted.

INTRODUCTION

The pressure crank angle (P-θ) relationship along with the valve opening and closing timings have been some of the main parameters for design improvements in the automobile industry. The accessibility of the cylinder allowed the use of simple mechanical indicator diagram equipment and the cam operated valves gave direct timings of valve action. However, a hermetically sealed compressor poses multiple problems such as::

- High speed running necessiates instruments with good dynamic response
- Limited space and accessibility for fixing the transducers
- Sealing of the shell to be undisturbed by the instrumentation
- Performance to be undisturbed by the instrumentation

- Separate monitoring of valve timings.

Early researchers, Speich[1], Mc Connell & Cohen[2] have measured the pressure-volume (P-V) relation of compressors successfully. The criteria limiting the dynamic range of pressure adaptors were suggested by Elson and Soedel[3]. However, the effect of adaptors on clearance volume is not discussed. Microprocessor based data acquisition helps in recording multipleparameters such as pressure, temperature, displacements and crank angle. Time averages, variation from cycle to cycle and during the cycle can be automatically calculated, Surfuzi & Nakayamma[4], Chang K.Y., Kerr S.V., Maclaren T.F.T.[5], Murati and Fuzitani[6] have acquired data through a microprocessor based system.

Along with the experimental methods, reliable computer oriented numerical models are available to obtain the pressure, volume, valve displacements, and mass flow rates. Soedel[7] has compiled an introductory model with a Fortran listing[8]. The model was enhanced by Hamilton[9]. Many improved models with additional factors taken into consideration are found in literature.

The criteria for analysis of a compressor independent of the refrigeration cycle are not standardized. Pandeya & Soedel[10] have defined the term efficiency of performance which combines all the factors that affect the compressor efficiency. Shaffer[11] has also discussed the mechanical, gas compression, suction gas heating, and overall efficiencies in evaluating compressor performance. He obtained an improvement in performance using larger discharge port areas and reduced suction gas temperature. Jacobs[12] took an overall view of the performance losses due to suction gas heating, manifold pressure drop, and pulsation, valve dynamics, piston ring blowby, and clearance volume.

Since the capacity increases as shown by bare calorimeter data do not always lead to EER increase, the present study is an attempt to make use of the detailed analysis of the pressure-volume diagram for estimation of the effectiveness of the design changes made to improve both the performance and EER of a compressor.

The experimentation is designed following conventional technique in which the transducer output is fed to an oscilloscope and photographed. Displacement of valves were not separately measured but the valve opening and closing timings were read from the pressure-crank angle diagram with reasonable accuracy. The instrumentation required is simplified to avoid the introduction of physical changes which might affect the true performance of the compressor.

EXPERIMENT

The compressor under study is fixed in a bolted shell instead of the normal welded shell, this allows the implementation of successive design changes on the same compressor for a comparative study. Figure-1 shows the pump seated in the bottom demountable shell. An adaptor for the crank angle measurement is fixed on the gas baffle. A cylindrical extension is provided on the bottom shell through which the pressure transducer is fixed on to the cylinder head. The transducer lead comes out of the adaptor through a leakproof seal. These adaptors are quick and easy to affix, do not damage the leads and can be quickly detached. No adhesive is used for either mounting or for sealing the transducers and the leads.

The flow diagram of the instruments used is shown in Figure-2. The reflective sensor measures the crank angle by giving a pulse at every BDC/TDC position. A kistler piezo-electric transducer senses the cylinder pressure which is amplified by the charge amplifier. The crank angle and pressure signals are fed to the two channels of a Philips oscilloscope. A Pentax ME-Super camera is used for taking photographs at a shutter speed of 1/8 a second and an aperture opening of 2.8 mm. The camera is mounted on a tripod stand.

Pressure Measurement

A Piezo-electric transducer of kistler make and type 7001 is used for pressure measurement. The dynamic range of the transducer is about 50 K-Hz. Elson and Soedel[3] have pointed out that the transducer adaptor resonance might cause distortion of the pressure signal. The natural frequency of the path connecting the transducer and the cylinder cavity is calculated to be 235 Hz which limits the dynamic range. The Piezo-electric transducer measures dynamic pressure variation. If pressure at one point is known or assumed, absolute pressures at all other points can be calculated.

Crank Angle Measurement

A magnetic pick up was first tried and the output signal was dominated by stray noise. It was concluded that the motor magnetic field was interfering with the crank angle pulse. An alternate transducer was made with a reflective sensor and its circuit shown in Figure 3. The reflective sensor is placed above the rotor face which is painted black with white radial lines at the BDC or TDC points. Sharp pulses are obtained corresponding to these reflective points.

Details of compressor under test

Crank radius	= 0.5177 in.
Connecting rod length	= 2.1850 in.
Piston area	= 2.40 in^2.

Displacement	=	2.4849 $in.^3$
Clearance volume	=	0.1651 $in.^3$

The clearance volume includes the cylinder clearance at TDC discharge port volume, volume above piston rings and the pressure transducer adaptor clearance less the suction valve volume.

TESTING

The compressor is run at normal load conditions of ASHRAE-T standard in a calorimeter test rig. The pressure-crank angle photographs are taken when the steady state conditions are reached. The following parameters are either read or calculated from the calorimeter test:

Compressor running speed,	ω.
Suction pressure outside the shell,	P_{so}.
Discharge pressure outside the shell,	P_{do}.
Suction temperature outside the shell,	T_{so}.
Discharge temperature outside the shell,	T_{do}.
Mass flow rate through the system,	$\dot{m}$.
Main and auxiliary current,	I_M, I_A.
Suction gas enthalpy,	h_{so}.
Discharge gas enthalpy,	h_{do}.
Main and auxilary resistance,	R_M, R_A.
Thermal work done between suction and discharge gas,	$\dot{m}\ (h_{do} - h_{so})$.
Power input to compressor,	W_I.
Cooling capacity of the compressor,	W_E.
Energy Efficiency Ratio,	EER.
Motor losses,	W_{ML}.

The crank angle and pressure signals are photographed and enlarged. The following parameters are read/calculated from the photograph and with the assumed values of suction pressure and temperature:

PV diagram for the compressor.
Valve opening and closing times.
Duration of cycle processes.
Mean effective over pressure.
Mean effective under pressure.
Compression ratio of cylinder peak pressure.
Compression ratio of cylinder mean pressure.
Mean and peak discharge temperatures.
Polytropic index of compression and reexpansion process.

Mass flow rate.
Blowby during a process.
Indicated work done by the compressor.
Suction under pressure and discharge over pressure losses.
Actual volumetric efficiency.
Heat transfer during the polytropic processes.

Combining the results of calorimeter test and the PV diagram the mechanical efficiency of the compressor, the suction gas heating efficiency and gas compression efficiency defined in (11) can be determined since all other losses are calculated.

Error Evaluation

Accuracy and repeatability of a test determines its reliability. Especially, for a test procedure used for design analysis, error estimation is useful. Error is possible if there is any deviation of the calorimeter test conditions. This error is minimized by using reliable instrumentation and was verified by conducting a repeatability test which is found to be within 0.5%.

The pressure transducer's accuracy is 0.01 psi. The suction inlet pressure is maintained at 76 psig. compensating for the suction path pressure drop (0.5 psi) and the resistance of the suction valve (1.5 psi), the suction pressure at the time of closing was assumed to be 74 psig. This assumption will not induce any error in total work calculation. However, the under pressure and over pressure losses will be incorrectly distributed.

Error in crank angle measurement might arise due to wrong marking of BDC line, misalignment of the reflective transducer and the BDC line. The BDC was marked using a dial gauge and the alignment was assured using an adaptor.

Error from photography can be minimized by proper focussing and adjusting for coincidence of the normals of the camera and the oscilloscope. Accurate reading of the P-θ diagram is important, especially during the compression and expansion processes. An error of ± 0.5 mm on the photograph's time axis leads to a ± 5° error in crank angle and ± 18 psi during the compression process.

The trueness of steady state condition on the calorimeter is tested by taking photographs at different times of steady state condition and the indicated work done compared within 1%.

The reading error during the conversion of P-θ diagram to PV diagram was checked by calculating the indicated work using a planimeter, manual reading, and a computer digitization. A maximum of 3.7% deviation was found.

Error in the measurement of calculations can also be found if any contradiction occurs; discharge mass greater than the suction mass or system mass flow rate greater than the discharge mass flow rate. The valve opening and closing times are reviewed when such occurances takes place.

Pressure-crank angle records

P- photographs of a 1.5 ton capacity compressor with the following successive modification of the valves are obtained:

- Figure 4: for a 0.015" suction valve, a 0.015" valve and a spring stop for discharge.
- Figure 5: for a 0.012" suction valve, a 0.015" valve and a spring stop for discharge.
- Figure 6: for a 0.015" suction valve, a 0.015" valve for discharge and no spring stop.

The results of these three experiments are discussed in this report.

RESULTS AND DISCUSSION

Mass flow

Figure 7 shows the suction, discharge and blowby paths through which the refrigerant enters or leaves the pump. The processes during which leakage occurs is shown in Figure 8. Leakage through valves is due to bad seating and valve flutter.

The discharge mass flow per cycle = $m_s + m_{dl} + m_{bi} - m_{sl} - m_{bo}$.

The blowby mass can be measured with an arrangement as shown in Figure 9. Direct suction is provided for the compressor, the blowby mass is let out into a receiver shell, when the receiver shell pressure equals the suction pressure, the experiment is stopped. The difference between the initial and final mass gives the blowby mass for a given time. Mass can be found by weighing or from the refrigerant charts using pressure, temperature and volume of the shell and receiver. For the compressor under test the blowby mass was found to be less than 1% of the total mass flow.

Cylinder Pressure Frequencies

The cylinder pressure spectrum is shown in Figure 10. The cylinder pressure should consist of running frequency (47.5 Hz) harmonics with monotonically decreasing intensity, if no external influence on the pressure is present. The first three natural frequencies of suction valve are measured to be 116 Hz, 230 Hz, and 340 Hz; and for the discharge valve 318 Hz, 914 Hz, and 940 Hz. - magnification at these frequencies is expected. Also amplification can be expecterd at the resonance frequency of the pressure transducer adaptor (235 Hz). The pressure pulsations become insignificant over 2000 Hz. Cylinder pumping is the basic forcing function for the valves and compressor noise and vibration and the pressure spectra will be useful.

Valve Timing

A large mass flow is achieved with long suction and discharge processes. Heat losses are minimized if the compression and re-expansion processes are made as short as possible. Figure 11 gives the valve opening and closing times are noted as the points of discontinuity on the continuous compression and re-expansion processes. Tuning of the valve timings can be done by changing the valve thickness, valve lift, port area, port entry condition, manifold and clearance volume for a given set of suction and discharge conditions.

Effective Flow Area

The effective flow area is less than the total port area available on a valve system. Port size, shape, entry condition, valve length, valve lift, number of flow restrictions in parallel and series are the factors that determine the effectiveness of the flow area. Soedel[7] gave an experimental method and Hamilton[9] an analytical method for finding the effective flow area, Ae, for a given compressor the following method can be used:

The cylinder mass is a function of crank angle, the time rate of change of cylinder mass, $\dot{m}$, is:

$$\dot{m}(\theta) = \rho(\theta)\,\dot{V}(\theta)$$

where $\dot{v}$ is the rate of change of cylinder volume. For a given compressor the volume is given as a function of crank angle, θ (measured from BDC).

$$V(\theta) = V_C + AR_1\left(1+\cos\theta + \frac{R_2}{R_1}\left(1-\sqrt{\left(1-\left(\frac{R_1}{R_2}\right)^2\sin^2\theta\right)}\right)\right)$$

rate of volume change,

$$\dot{V} = \omega\, dV(\theta)/d\theta$$

where, ω is the angular velocity of the pump.

During suction process, the rate of change of cylinder mass is equal to the rate of mass flow through the suction port. For a pressure difference of ΔP_s across the valve and a gas density of ρ_s, the mass flow rate is given as:

$$\dot{m}(\theta) = Ae(\theta)\sqrt{2\,\rho_s\,\Delta P_s\, g_c}\;.$$

If the cylinder mass is assumed to have a constant density throughout the suction process then $\rho(\theta) = \rho_s$.

The minimum effective flow area, Ae (θ), required to maintain a constant density (pressure and temperature) in the cylinder is:

$$Ae(\theta) = V(\theta)\sqrt{\rho s / 2 \Delta Ps\ gc}\ .$$

Ae (θ) is calculated for the total suction process and the greatest value is chosen as the required effective valve area. Similar calculation can be done for the effective flow area of discharge ports. For the compressor chosen the effective flow area required for suction is 0.174 in^2 and 0.15 $in.^2$ for the discharge.

Polytropic Index

The compression and expansion process are reversible adiatic processes, in an ideal cycle, with the polytropic index, n, equal to the ratio of specific heats, γ . However, in the actual process a heat loss during the compression and heat gain during the re-expansion causes a deviation towards the isothermal process which has a polytropic index of 1, Figure 8. However, for the tests conducted, the index varied from point to point along a process. Also, at certain points of the process the polytropic index went either below 1 or above the value of γ. Irreversibility of the process due to frictional losses and the mass leakage during the process might be the cause for this discrepancy. Average value for the process can be used for calculation. The polytropic index, n, for a process between two points 1 and 2, used as subscripts, is:

$$n = \text{Log}\ (P1/P2) / \text{Log}(V2/V1).$$

Cylinder Pressure and Pressure Ratio

The ratio of discharge to suction pressure of the compressor was maintained at 3.47. However, due to the valve and the flow path losses, Figure 8, the actual pressure ratio is higher. The ratio of maximum discharge pressure to the minimum suction pressure is obtained as 6.97 for a pump with 0.012" suction valve; 6.65 for the pump without spring stop and 6.11 for the pump with 0.015" suction valve. Corresponding mean pressure ratios are 6.04, 5.56 and 5.42. The mean effective pressure ratio is defined to give an average estimate of the cylinder pressure.

The mean effective discharge pressure is the sum of the discharge pressure, P_d, and the mean effective over pressure(m.e o.p.) and similarly the mean effective suction pressure is the difference of the suction pressure, P_s and the mean effective under pressure (m.e.u.p.). The pump might be assumed to operate between these two constant pressures. The difference in the mean effective pressure ratio and the pressure ratio maintained outside the shell gives a quick indication of the suction under pressure and discharge over pressure losses.

Volumetric Efficiency

Volumetric efficiency is defined from Figure 8, as:

$$\eta_V = (V_T - V_R)/(V_T - V_C).$$

For an adiabatic re-expansion, the re-expansion volume V_R is:

$$V_R = V_C\,(Pd/Ps).$$

Thus, for a high volumetric efficiency a small clearance volume and pressure ratio are required.

$$\text{Suction mass} = \rho_s\,(V_T - V_R)$$
$$= \eta_V\,(V_T - V_C).$$

An increase in the suction mass is obtained by an increase in the suction gas density ρs obtained by reducing the superheat; increasing the volumetric efficiency and reducing the clearance volume. The volumetric efficiency was measured to be about 80%, from the PV diagram.

Losses, Performance and EER

The area below the constant suction pressure line P_s, is the loss of work per cycle due to suction under pressure. The pressure in the cylinder drops due to the valve resistance and manifold restriction. A similar loss occurs corresponding to the area above the constant discharge pressure. The suction manifold pressure loss is measured to be 0.5 psi and the discharge manifold pressure loss is 15 psi. A summary of the calorimeter and pressure volume diagram results are shown in Table-1.

Table 1: Effect of valve changes on compressor

Parameter	Mass flow (lbm/hr)	Indicated work (I/W) (HP)	Suction loss (% of IW)	Discharge loss (% of IW)	Cooling Capacity (BTU/hr)	EER BTU/ W.hr.
0.015" suction valve	223.8	2.41	12.86	7.87	16424.7	7.3
No spring support on discharge valve	227.7	2.26	15.26	7.67	16712.8	7.4
0.012" suction valve	235.6	2.99	14.29	7.88	17291.1	7.5

Though the suction and discharge loss % has not changed considerably the increase in cooling capacity and EER results from an increase in the mass flow rate. The increased mass flow rate is associated with a change in the suction and discharge process duration.

CONCLUSION

A complete and systematic study of the compressor is possible by the simple experimental method, suggested. Accurate and adequate results were obtained with the simplified set of instrumentation. Valve modifications were made to achieve an improvement in performance and EER.

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NOMENCLATURE

m_s Suction mass entering the cylinder through suction port

m_{se} Leakage mass leaving the cylinder through suction port

m_d Discharge mass leaving the cylinder through discharge port

m_{dl} Leakage mass entering the cylinder through discharge port

m_{bo} Leakage mass leaving the cylinder through piston rings

m_{bi} Leakage mass entering the cylinder through piston rings

m_c Mass in the cylinder

P_d Discharge plenum pressure

P_s Suction plenum pressure

θ Crank angle

ρ Mass density

γ Coefficient of specific heats

P_c Cylinder pressure

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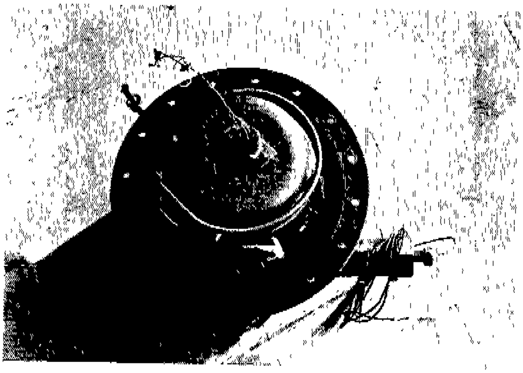

Fig. 1. Test compressor in a demountable shell.

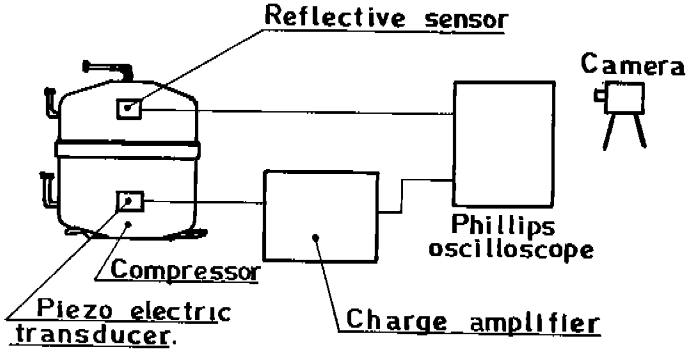

Fig. 2. Flow chart of instrumentation.

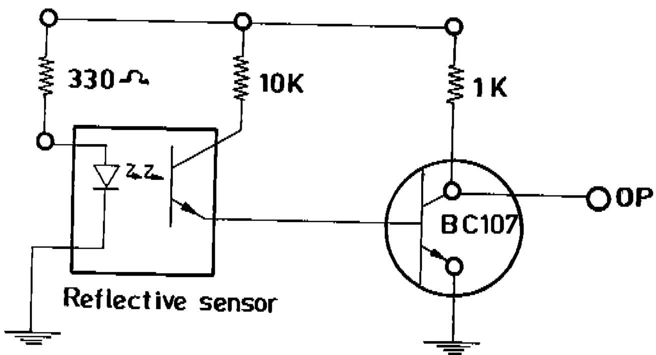

Fig. 3. Crank angle sensor circuit.

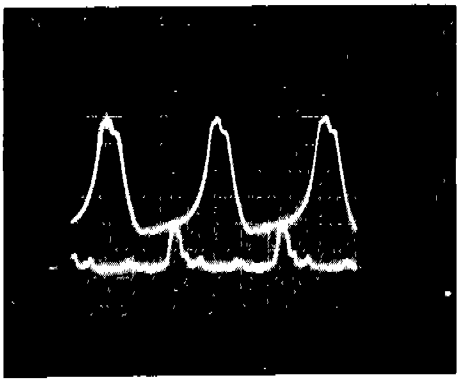

Fig. 4. Compressor with 0.015" suction valve, 0.015" dis. valve with spring stop.

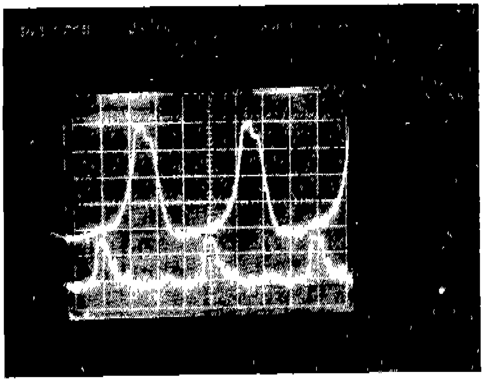

Fig. 5. Comp. with 0.012" suc. valve, 0.015" dis. valve with spring stop.

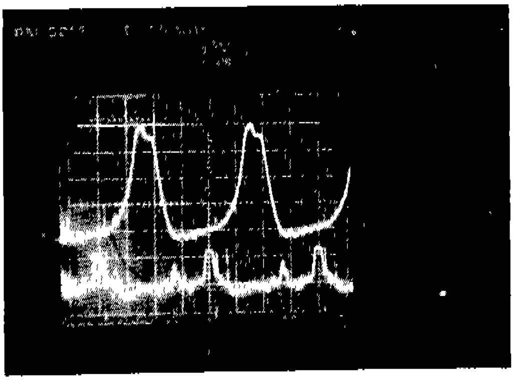

Fig. 6. Comp. with 0.015" suc. valve, 0.015" dis. valve & no spring stop.

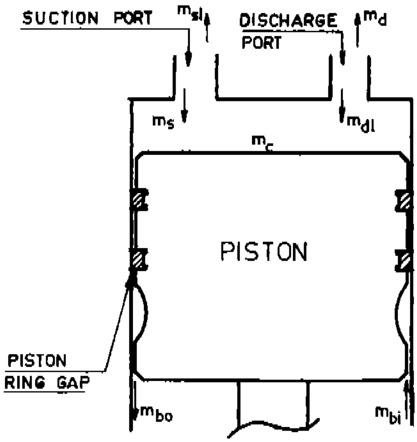

FIG.7 MASS FLOW PATHS FOR REFRIGERANT

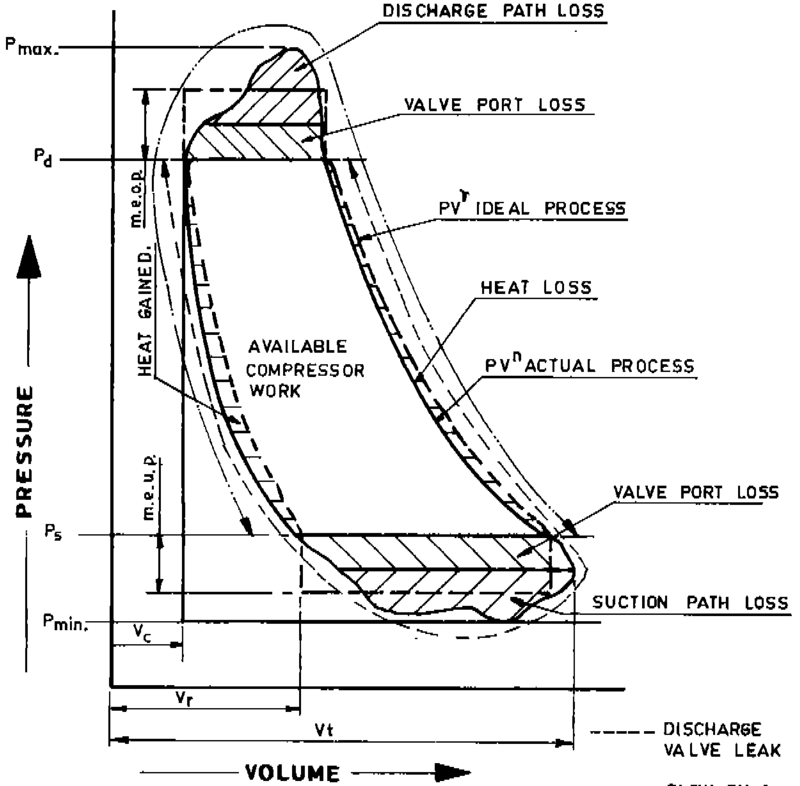

FIG.8. COMPONENTS OF PRESSURE-VOL. DIAGRAM

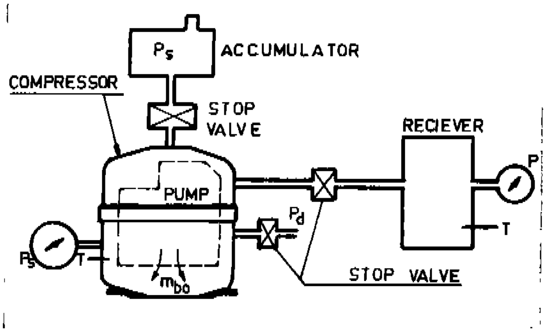

Fig. 9. Blow by measuring equipment.

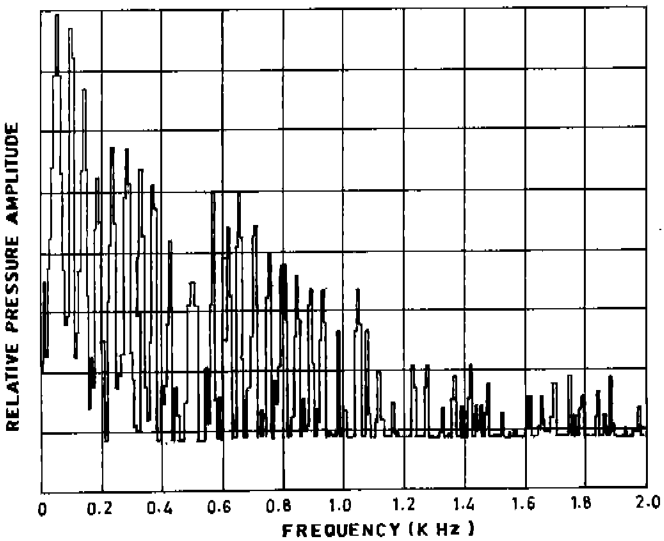

FIG. 10 CYLINDER PRESSURE SPECTRUM

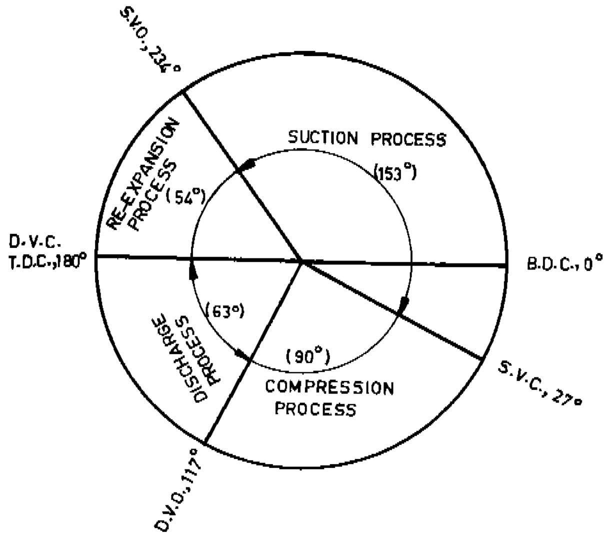

Fig. 11. Valve timing diagram.