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# Prediction for the Sealing Characteristics of Piston Rings of a Reciprocating Compressor

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# PREDICTION FOR THE SEALING CHARACTERISTICS OF PISTON RINGS OF A RECIPROCATING COMPRESSOR

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## ABSTRACT

Piston rings are essential sealing units of reciprocating compressors. The purpose of this paper is to study a method for predicting the sealing characteristics of piston rings and to evaluate the sealing effects of lubricating oil. Thus both mathematical models of working cycle in a cylinder and the gas leakage through piston rings have been set up. Meanwhile the oil sealing coefficient $\gamma$ , which reflects the sealing effect of lubricating oil, is introduced and $\gamma$ curves are plotted according to the experimental data. The calculated curves of both the pressure in the cylinder and in the volumes between the rings are basically in agreement with the curves measured, and the calculated data of gas leakage through piston rings are approximately equivalent to the data measured. Thus the mathematical models are shown to be applicable.

## SYMBOLS

| | |
|---|---|
| A | flow area of valve |
| b | width of piston ring |
| D | diameter of cylinder |
| $F_{gas}$ | gas force |
| $F_p$ | area of piston |
| $F_{sp}$ | spring force |
| f | flow area of gap of piston ring |
| H | displacement of valve |
| h | enthalpy or height of piston ring |
| k | isentropic index |
| m | gas mass |
| $\dot{m}$ | mass flow rate |

$p$ gas pressure
$R$ gas constant
$s$ piston stroke
$T$ gas temperature
$t$ time
$U$ internal energy
$V$ volume of cylinder
$v$ gas specific volume
$\alpha$ flow coefficient
$\rho$ gas density
$\gamma$ oil sealing coefficient
$\delta$ average clearance of contacting surface
$\theta$ crankshaft angle
$\lambda$ ratio of crank to conducting rod
$\mu$ gas viscosity

## INTRODUCTION

Piston rings are essential units used to seal gas in reciprocating compressors and the sealing characteristics of the rings is an importent factor affecting volumetric efficinecy and power dissipation of compressors. Thus, the study on this respect has been paid attention to widely.

The interests of this paper are mainly in following researches.

The pressure variance in cylinder has great significance to decide gas leakage, but in previous researches when calculating gas leakage people often take the pressure in cylinder as average value within a cycle, or use the pressure variance in a theoritical cycle as original data, which makes it impossible or erroneous to calculate instantaneous pressure distribution between rings and gas leakage through each ring. Although sometimes actual pressure in a cylinder is used, the data frequently come from measured value, which is of no advantage to predict the sealing characteristics of piston rings and to carry out an optimal design. This paper utilizes a mathematical model of working cycle to simulate actual processes in a cylinder, and gets pressure data coinciding with actual data, thus the model sets up a reliable basis to predict the sealing characteristics of piston rings.

Up to now, the experiments and researches on functions of lubricating oil to the sealing of piston ring have been scarcely reported. In this paper the function of oil is preliminarily evaluated by experimental method and a concept of oil sealing coefficient is introduced, which may be certain significant to the studies on the sealing characteristics of piston rings of oil lubricating compressors.

## THEORETICAL ANALYSIS

Theoretical analysis was done in two sections: one is simulation of working cycle of compressors, the other is development of the model of gas leakage through piston rings.

### 1. Simulation of Working Cycle

The mathematical simulation of working cycle in this paper is mainly used to calculate pressure in cylinder, therefore some proper simplications are considered, which involve: thermal parameters in suction and discharge plenums are constant; no heat transfer in the cycle; gas leakage through piston rings is negligible; no gas leakage through valves and the draft coefficient and the flow coefficient of valves are constant; the working medium is ideal gas. The mathematical model involves following basic formulas:

mass conservation

$$dm = dm_{sc} - dm_{dc}$$

energy conservation (without heat transfer)

$$dU = h_s\,dm_{sc} - h\,dm_{dc} - p\,dV$$

gas flow rate

$$dm_i = \alpha_i \rho_i v_i A_i\,dt$$

here the subscript i represents suction or discharge valve as requested.

valves dynamic equation

The movement of valve plate is simplified as a system of single mass and single freedom, and damp forces on the plate by oil viscosity, friction etc. are disregarded.

$$m_v \frac{d^2H}{dt^2} = F_{sp} + F_{gas}$$

equation of working volume

$$\frac{dV}{d\theta} = \frac{1}{2} F_p S \left(\sin\theta + \frac{\lambda \sin\theta\cos\theta}{\sqrt{1-\lambda^2\sin^2\theta}}\right)$$

These equations become simultaneous equations and can be solved with the four orders standard Runge-Kutta method for compression, discharge, expansion and suction processes. Thereby, the mathematical simulation of working cycle is completed.

## 2. The Model of Gas Leakage through Piston Rings

When a piston reciprocates in a cylinder, its lubricating conditions are very complicated which show different lubricating states resulted from oil amount, oil viscosity, geometric configuration and surface roughness of contacting parts, relative velocity, and clearance between two surfaces. The sealing function of oil in lubricating compressors makes gas leakage through piston rings decrease considerably, and finding a theoretical solution for the leakage become very complicated and difficult. However the mechanism analysis of gas leakage in oil free compressors had been well studied, and it was borrowed to analyse the lubricating compressors with introducing a new concept of oil sealing coefficient $\gamma$ to correct the theory on oil free compressors, therefore various sealing characteristics of piston rings in oil lubricating compressor can be obtained approximately.

In oil free compressor there are three possible paths of gas leakage through the rings: rings and surface of cylinder, rings and bottom of slot of piston, gaps of rings. They are denoded by a, b, c in Fig. 1. For the piston with several rings the total leakage mass flow $\dot{m}_i$ through the ith ring becomes

$$\dot{m}_i = \dot{m}_{ai} + \dot{m}_{bi} + \dot{m}_{ci}$$

The flow through the gaps of piston rings is presumed as one dimensional compressible isentropic flow passing a throttle. Therefore the gas leakage from gap of the ith ring would be written as

$$\dot{m}_{ci} = \frac{A_i p_{i+1}}{T_{i+1}} \sqrt{T_i \left[ 1 - \left( \frac{p_{i+1}}{p_i} \right)^{\frac{k-1}{k}} \right]}$$

here $A_i = \alpha f_i \sqrt{2k/[R(k-1)]}$

But if the flow speed in the gap equals the sound speed

$$\dot{m}_{ci} = \frac{B_i p_i}{\sqrt{T_i}}$$

here $B_i = \alpha f_i \sqrt{\frac{2k}{R(k+1)} \left( \frac{2}{k+1} \right)^{\frac{2}{k-1}}}$

The flows between piston rings and cylinder and between piston rings and piston are considered as the flow in a thin clearance between two smooth surfaces. This problem can be solved in theory of two dimensional uncompressible viscous laminar flow.

Navier-Stokes equation is used to conduct the leakage $\dot{m}_{ai}$ between rings and cylinder and the leakage $\dot{m}_{bi}$, rings and piston.

$$\dot{m}_{ai}=\frac{\pi D\delta_a^3(p_i^2-p_{i+1}^2)}{24\mu RT_i h} \quad ; \quad \dot{m}_{bi}=\frac{\pi\delta_b^3(p_i^2-p_{i+1}^2)}{12\mu RT_i\ln[D/(D-2b)]}$$

Let $C=\frac{\pi D}{24h}$ , $E=\frac{\pi}{12\ln(\frac{D}{D-2b})}$ the total leakage through ith ring will be

$$\dot{m}_i=\frac{A_i p_{i+1}}{T_{i+1}}\sqrt{T_i[1-(\frac{p_{i+1}}{p_i})^{\frac{k-1}{k}}]}+\frac{(C\delta_a^3+E\delta_b^3)(p_i^2-p_{i+1}^2)}{\mu RT_i}$$

If flow speed equals the sound speed, the first item in right side of the equation above should be altered correspondingly.

In order to evaluate the leakage through piston rings in oil lubricating compressor in the theory on oil free compressor, the sealing effect of oil should be known, so the oil sealing coefficient $\gamma$ is introduced. Provided that the oil sealing effect in each ring is same, the total leakage through ith ring in lubricating compressor should be

$$\dot{m}_i=\gamma\left\{\frac{A_i p_{i+1}}{T_{i+1}}\sqrt{T_i[1-(\frac{p_{i+1}}{p_i})^{\frac{k-1}{k}}]}+\frac{(C\delta_a^3+E\delta_b^3)(p_i^2-p_{i+1}^2)}{\mu RT_i}\right\}$$

Similarly, if flow speed reaches the sound speed the first item in braces should be altered correspondingly.

The leakage through rings depends upon pressure and temperature in cylinder and pressure distribution in volumes between the rings which are formed with the spaces surrounded by cylinder, piston and piston rings. Fig. 2 shows a simple illustration of piston with several rings, and the portion surrounded with dotted line is control volumes. For each control volume the mass and energy equations are written out from principles of thermodynamics. For solving the simultaneous equations some discussions were carried out on following situations: (1) $p_{i-1}>p_i>p_{i+1}$, (2) $p_{i-1}<p_i>p_{i+1}$, (3) $p_{i-1}<p_i<p_{i+1}$, (4) $p_{i-1}>p_i<p_{i+1}$, and gas temperature before the first ring and between the rings is taken as constant.

Since the experimental device is for measuring gas leakage of a piston with three rings, the numerical calculation is also done for a piston with three rings, therefore there are two volumes between the rings, i.e., two control volumes. Differiential equations are set up for each control volume, and solved consequently.

## EXPERIMENTAL DEVICE AND MEASUREMENT SYSTEM

The purpose of experiments is to examine accuracy of results of the mathematical models. The experiments involved measuring total leakage through piston, pressure between piston rings and temperature at some points on the piston, and controlling and measuring oil amount instilled into cylinder, furthermore, the running speed and discharge pressure were also measured.

Because of the piston reciprocating quickly in cylinder, it is difficult to mount measuring points on the piston, therefore, a special compressor, in which cylinder reciprocates instead of piston and piston maintains static condition, was designed and set up. A part of the compressor is illustrited in Fig. 3.

In the experiments, all measuring points were mounted on the piston which is also designed and made specially. The piston has three sealing rings and a gas-gathering slot that gathers the leakaged gas through piston rings and transmits it to volumeter by passing a passage inside the piston and a rubber-tube. In order to protect the gathered gas from flowing into atmosphere, two extra sealing piston rings are attached to the piston behind the gas-gathering slot.

Three pressure sensors were fixed on the piston to measure pressure in cylinder and in two volumes between the rings respectively, and two thermo-couples were located on the piston before the first ring and behind the third ring to measure average temperature. In order to simulate oil lubrication, two oil instilling holes were positioned before the first ring with 180 degrees symmetrically.

The measurement system is shown in Fig. 4. average leakage was measured with volumeter. Pressure was measured with quartz-pressure sensors and recorded by a oscillograph. The running speed was measured by a digital running speed gauge.

## THE OIL SEALING COEFFICIENT $\gamma$

The oil sealing coefficient $\gamma$ , which is mentioned above, is the percentage of gas leakage through piston rings in oil lubricating compressor to that in oil free compressor. In order to decide $\gamma$ , experiments were carried out, in which HS-19 compressor oil was used (oil density - 0.882 g/ml; oil drop volume - 0.32 ml/drop). Experiment was done on the special compressor, the discharge pressure $p_d$ was 882 KPa and the running speed was

800 rmp and 400 rmp respectively.

Through these experiments, leakage flow under different oil instilling amount and two running speeds were measured. Meanwhile, the leakage flow of oil free machine could be calculated with the mathematical model of gas leakage. When suction pressure was 96 KPa and discharge 882 KPa, the calculated leakage flow was 1.9829 l/min for 800 rpm and 1.9227 l/min for 400 rpm.

The oil sealing coefficient $\gamma$ is defined as ratio of measured leakage Qlr under oil lubrication to the calculated leakage Qli under oil free. The $\gamma$ represents total effect of the oil on the sealing of piston rings and $\gamma$ should be greater than 0 and smaller than 1. For ideal gas, $\gamma$ is also equal to the ratio between leakage mass flows, then

$$\gamma = \frac{Q_{lr}}{Q_{li}} = \frac{\dot{m}_{lr}}{\dot{m}_{li}}$$

The curves of $\gamma$ vs. oil amount are plotted in Fig. 5 according to measured data. Since the oil instilling was not continual, the lubricating condition during a cycle is different slightly, and it made the leakage process unsteady, thus the points measured in Fig. 5 is diffusive a little.

It is known from Fig. 5 that the tendency of leakage change with oil instilling can be devided into two stages. On the first stage, when oil instilling increases from little amount, the sealing effect improves quickly. For example, when running speed is 800 rpm and oil instilling amount increases from 0.04 ml/min to 0.4 ml/min the gas leakage decrease 30%; when running speed is 400 rpm and other conditions are same as above, the gas leakage decrease 24%. On the second stage, when oil instilling amount is more than a certain critical amount the sealing effect of oil is nearly unchanged,. For example, when running speed is 800 rpm and oil instilling amount is more than 0.8 ml/min the gas leakage approximately maintained 65% of the leakage under oil free; when running speed is 400 rpm and oil instilling amount is more than 1.0 ml/min, the gas leakage approximately maintained 70% of the leakage under oil free. Totally, it may be realized that when oil instilling amount is more than 1 ml/min, the lubrication is sufficient in the sence of sealing.

## CALCULATED RESULTS AND EXPERIMENTAL EXAMINATION

### 1. The Simulating Results of Working Cycle

In a working cycle of a compressor, the instantaneous pressure in the cylinder is the basis to calculate pressure distribution between the rings and instantaneous leakage through each ring. The mathematical model of working cycle can simulate all processes in the cylinder exactly and provide accurate instantaneous pressure in the cylinder. The model can also be used to get accurate capacity of compressors. The calculated results and the measured results are plotted in Fig. 6 so as to compare with each other. In the figure, the solid line is the calculated results and the dotted line is the measured results.

## 2. The Prediction of Pressure Distribution between the Rings and Gas Leakage

The pressure distribution between the rings is the important basis for studing leakage, wear and tear, and strength of piston rings. By virtue of the mathematical model of gas leakage, the change of the pressure between the rings with the pressure in the cylinder and instantaneous gas leakage through each ring can be predicted.

The measured results and calculated results of the pressure between the rings are plotted in Fig. 7 and Fig. 8, and differences between two ways are in existance. The main points are: the maximum measured pressure behind the second rings is lower 75 KPa than the calculated and the peak phase is later 10 degrees than the calculated. In the beginning of suction and discharge processes, the measured and calculated are different obviouslly. These differences may be expounded as follow. In front of pressure sensor, there is a small chamber, which is formed by machine processing and for sensor working, and this chamber is connected with the volume between rings by a thin hole. When the pressure between rings is reflected in the chamber through the hole, its amplitude will decrease and the phase will be late, which causes the measured value to differ from the calculated. As the maximum measured pressure behind the first ring is consistent to calculated pressure and the phase is coincident too, it expressed that the mathematical model is applicable to a certain extent.

From Fig. 8 it can be found out that during compression process, the pressure between rings increases progressively with the increase of pressure in the cylinder, the phase difference maintaining constant. In the discharge process the average pressure in the cylinder is steady, but the pressure between rings still tends to rise until the piston moves near to the top dead centre when the pressure between rings arrives at peak position. During expansion process, the pressure both in cylinder and between rings begins to decrease, but the decreasing rates are so different that the pressure in cylinder may be lower than the pressure between rings. In this case, the gas backflow, which flows from the volume between rings to the cylinder, through the first ring appears (Fig.8). Consequently, as the pressure behind the first ring is lower than that behind the second one gradually, the gas backflow through the second ring can also be seen from Fig.8. Actually, when piston gets near the low dead centre, even a little backflow through all the rings comes about. The reason for this is mainly that the pressure in cylinder is lower than atmosphere during suction process and the pressure difference causes air to leak through piston rings into cylinder.

Fig. 9, the curves of instantaneous gas flow, shows that

(1) The gas leakage through each ring strengthens in discharge and expansion processes and weakens in suction and compression proceses.

(2) The total leakage through each ring is equal to one another in a working cycle, that is, algebra sum of each area surrounded by individual curve and the line of $m_l$=0 (Fig.9) should be equal, for instence, here the total equivalent leakage through each ring is

the first ring $Q_{l1}$=1.982901/min;

the second ring $Q_{l2}$=1.984181/min;

the third ring $Q_{l3}$=1.984471/min.

The maxmum relative error between them is smaller than 0.08%.

Fig.10 is the curves of pressure difference between rings. From the figure it can be seen that the pressure difference on the first ring has a change in direction within a cycle, that is, in compression and discharge processes the direction is from cylinder to external and in expansion and suction processes it is from external to cylinder. The pressure difference on other two rings is always from cylinder to external.

With the same running speed, the greater the pressure difference is, the faster the piston rings wear. From the curves of pressure difference, it can be known that for all rings the integral of absolute value of pressure difference on the first ring or the third ring is greater than the second ring, so it can be deduced that the first ring and the third ring are liable to wear.

## CONCLUSIONS

1. The mathematical models of working cycle and gas leakage through piston rings in a reciprocating compressor can be satisfactorily adopted to discribe the working processes in a cylinder, the pressure variance between piston rings and the leakage variance through piston rings. The calculated results of gas leakage through piston rings by the models have certain accuracy. The models provide a way to predict the sealing characteristics of piston rings.

2. The experiments show that under sufficient lubrication the leakage through piston rings in a lubricating compressor is approximately 65-70% of leakage in a oil free compressor.

3. In a compressor with three sealing piston rings the integral of absolate value of pressure difference on the first or the third ring is greater than the second one, so these two rings are liable to wear.

## ACKNOWLEDGEMENTS

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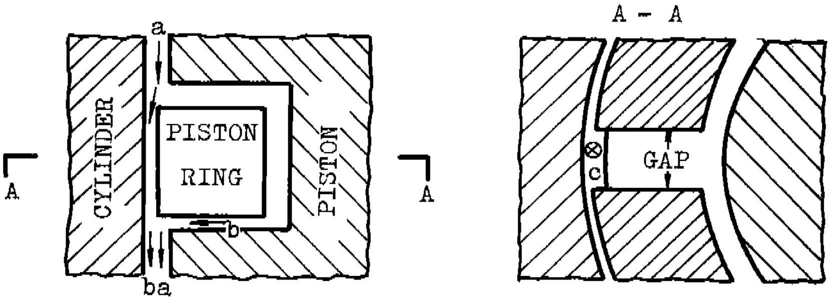



Fig. 1: Paths of Leakage through Piston Rings

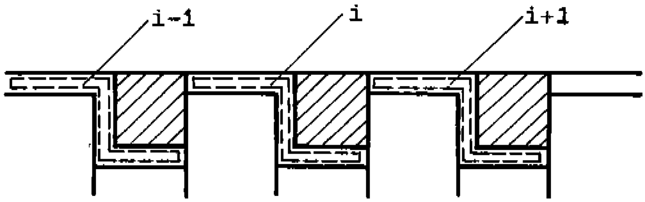



Fig. 2: A Simple Illustration of Piston with Several Rings

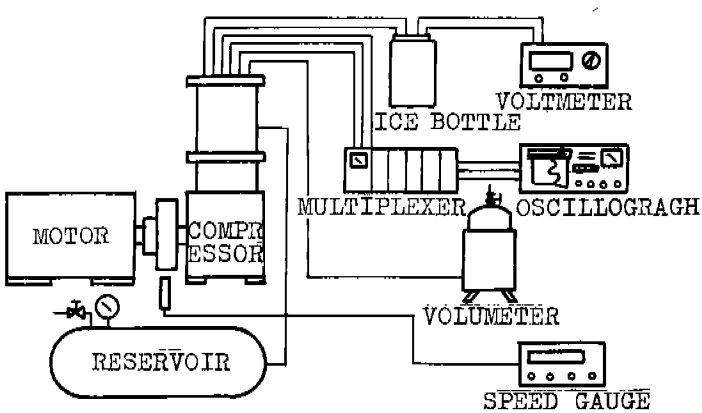



Fig. 4: The Measurement System

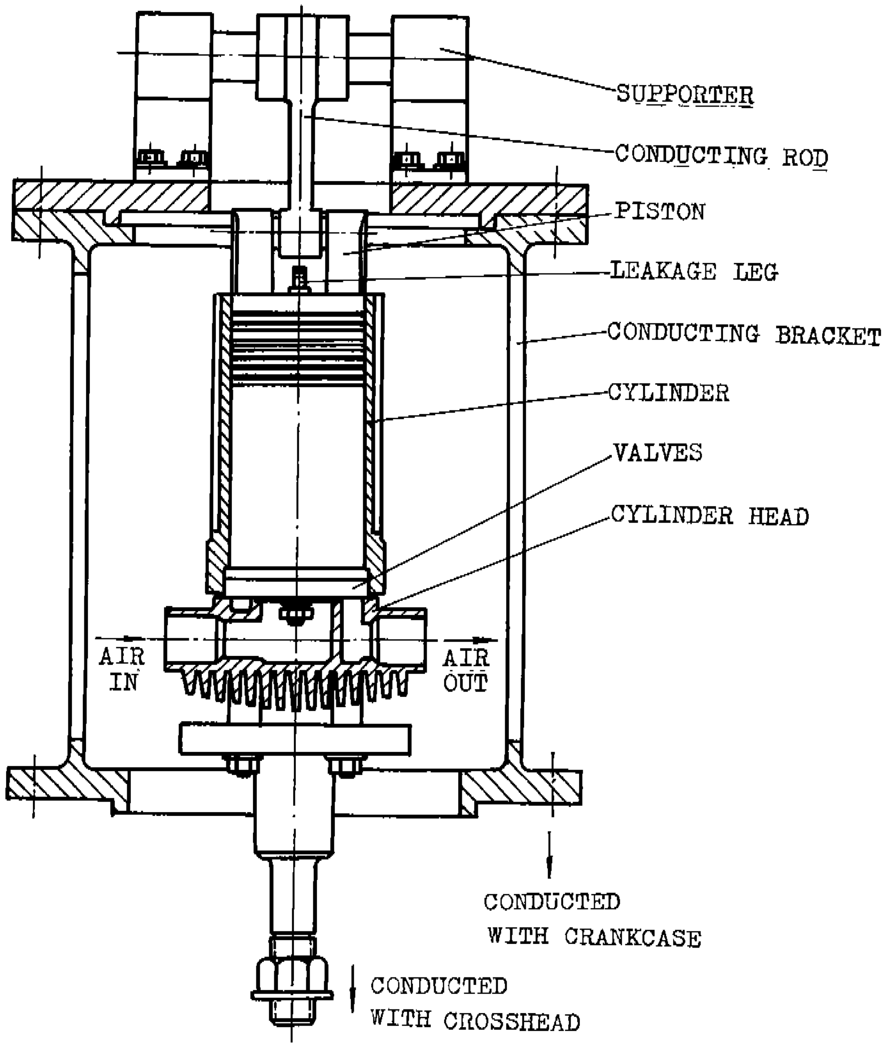
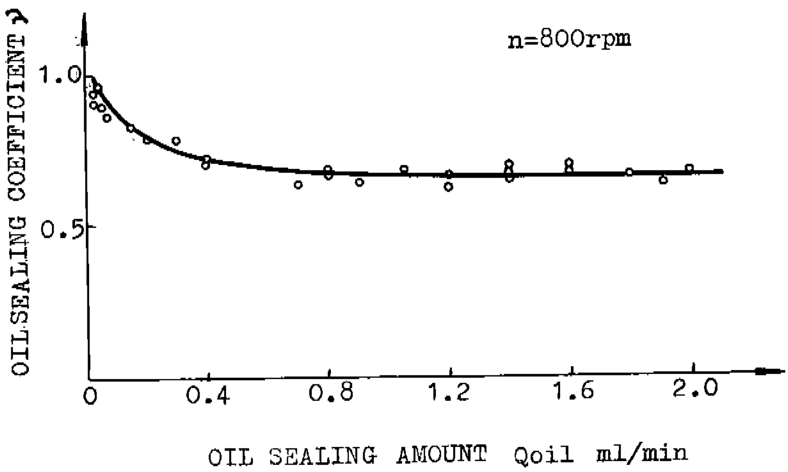



Fig. 3: A Portion of the Compressor for Experiment



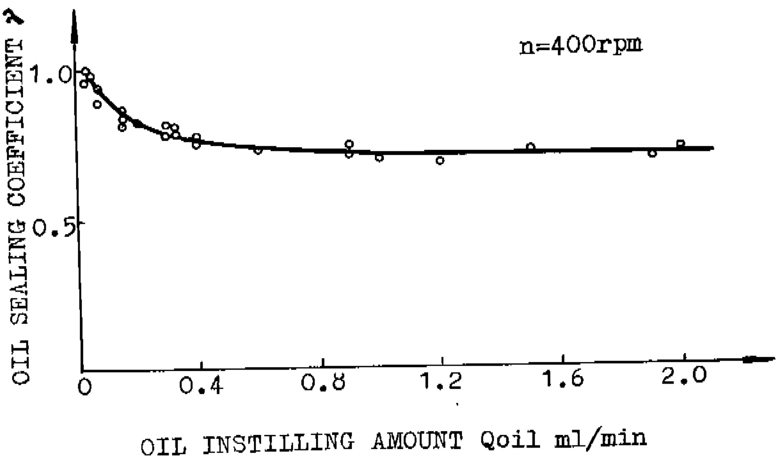




Fig. 5: Oil Sealing Coefficient γ vs. Oil Instilling Amount Qoil

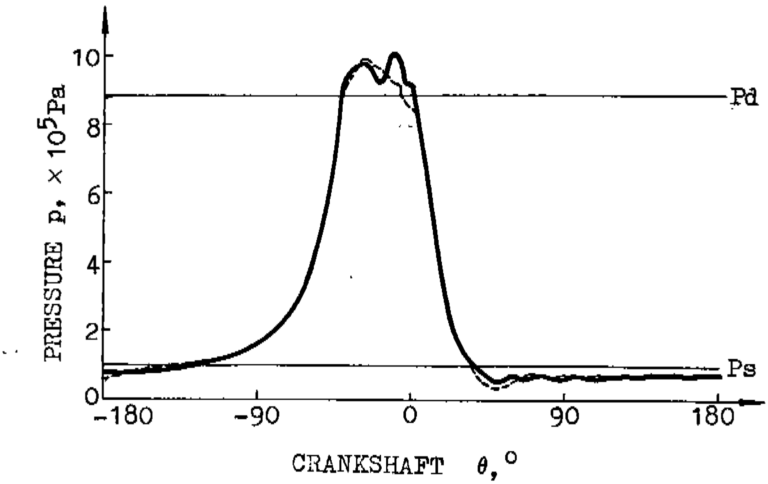



Fig. 6: Pressure Change in the Cylinder

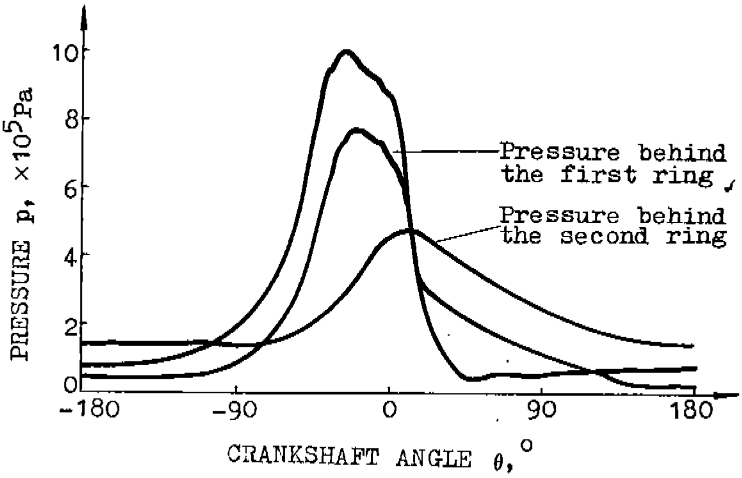



Fig. 7: Measured Pressure Distribution between Piston Rings

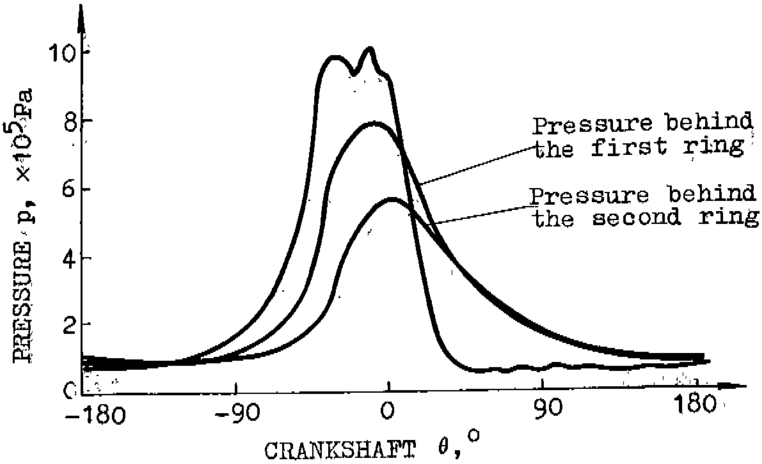



Fig. 8: Calculated Pressure Distribution between Piston Rings

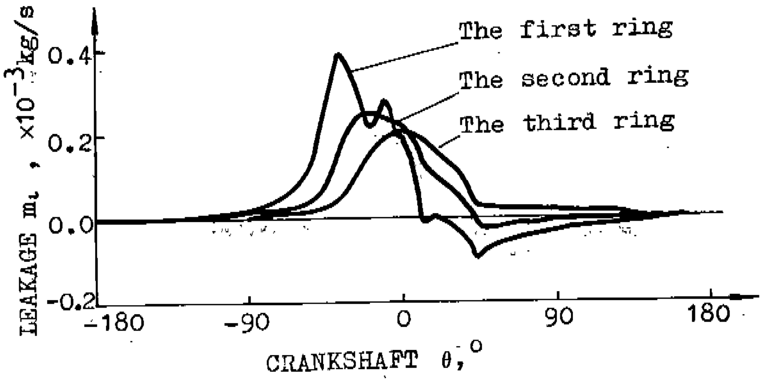



Fig. 9: Instantaneous Gas Leakage through Each Ring

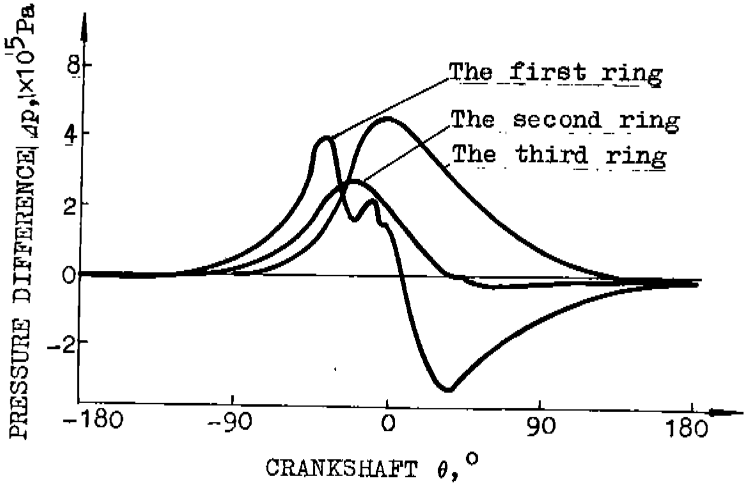



Fig. 10: Pressure Difference between Piston Rings