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PERFORMANCE ANALYSIS OF OIF SINGLE SCREW COMPRESSOR

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ABSTRACT

This paper presents the performance analysis and the internal pressure measurement of the oil injection-free single screw compressor. The geometric shape of the single screw compressor and its theoretical performance with the slide valve have been analyzed. The internal pressure has been measured with piezo type pressure sensors, and the pressure-volume diagram has been obtained. The experimental results agree fairly well with the theoretical predictions, and the volumetric and adiabatic efficiency can be estimated by this analysis.

SYMBOLS

A	cross sectional area
D_{sg}	distance between screw axis and gate rotor axis
F	coefficient of flow rate
G_c	mass of gas in the groove
G_i	mass of gas which enters the groove
G_o	mass of gas which comes out of the groove

L_1	leakage line length (screw rotor and casing high pressure side)
L_2	leakage line length (screw rotor and casing at discharge side)
L_3	leakage line length (screw rotor and casing at suction side)
L_4	leakage line length (screw rotor and casing lip)
L_5	leakage line length (gate rotor teeth side)
L_6	leakage line length (gate rotor teeth tip)
n	polytropic index
P_1	upstream pressure
P_2	downstream pressure
P_c	pressure of the groove
p	cross point of gate rotor tip
q	cross point of screw rotor surface
R_g	gate rotor radius
R_s	screw rotor radius
T_c	temperature of the groove
T_i	temperature of gas which enters the groove
v_1	upstream specific volume
v_c	specific volume of the groove
W	width of gate rotor
α	engaging angle during compression and discharge
β	engaging angle
Θ	gate rotor rotation angle
θ	screw rotor rotation angle

Subscript

d	discharge
s	suction

INTRODUCTION

Rotary positive displacement compressors are widely used today in coping with market needs, as they provide for improved efficiency, reliability, and light weight. The single screw compressor invented in 1960 by B.Zimmern is a type of rotary positive displacement compressor. It has the following inherent features:

(1) Minimal axial load on the screw rotor shaft.
(2) No requirement for the suction and discharge valve.
(3) Small number of moving parts.
(4) No risk of seizure and wear.

The semi-hermetic type single screw compressor for air conditioning application is being developed under Omphal's license. It is the oil injection-free type. It injects liquid refrigerant instead of oil.

To design the single screw compressor it is necessary to calculate the geometric shape and its theoretical performance. It is also necessary to measure the internal pressure of the groove by using pressure sensors.

This paper explains the method of the performance analysis of the oil injection-free type single screw compressor. This model predicts the swept volume, the port area, the efficiency, the torque and the pressure, etc. The measurement technique and the result of the screw rotor internal groove pressure measurements are explained.

ANALYSIS OF GEOMETRIC SHAPE

The single screw compressor consists of a screw rotor and two gate rotors. The screw rotor has six grooves, and the gate rotors have eleven teeth. Fig. 1 shows the geometric variables of this analysis. The cylindrical coordinate is used in this analysis. The origin is the cross point of the screw rotor axis and the perpendicular line from the gate rotor axis. R-direction is toward the screw rotor surface. Z-direction is toward the discharge side. When Rs and $\alpha(=160°)$ is given, the points of p_s, p_d, q_s, q_b can be calculated by the following formulas.

$$R_g=R_s \tag{1}$$

$$W=0.3R_s \tag{2}$$

$$D_{sg}=1.6R_s \tag{3}$$

$$\Theta=6\theta/11 \tag{4}$$

$$\beta_d=\cos^{-1}((D_{sg}-R_s)/R_g) \tag{5}$$

$$\beta^*=\sin^{-1}(W/(2R_g)) \tag{6}$$

$$\beta_s=-(\alpha-\beta_d-\beta^*) \tag{7}$$

$$Z_s=R_g\cdot\sin\beta_s \tag{8}$$

$$Z^*=(D_{sg}-R_s)\cdot\tan\beta_s-W/(2\cos\beta_s) \tag{9}$$

$$Rp_d=D_{sg}-R_g\cdot\cos(\Theta+\beta^*) \tag{10}$$

$$Rp_s=D_{sg}-R_g\cdot\cos(\Theta-\beta^*) \tag{11}$$

$$Zp_d=R_g\cdot\sin(\Theta+\beta^*) \tag{12}$$

$$Zp_s=R_g\cdot\sin(\Theta-\beta^*) \tag{13}$$

$$Zq_d=(D_{sg}-R_s)\cdot\tan\Theta+W/(2\cos\Theta) \tag{14}$$

$$Zq_s=(D_{sg}-R_s)\cdot\tan\Theta-W/(2\cos\Theta) \tag{15}$$

To obtain the swept volume the area which engages with the screw rotor and the gate rotor is calculated. The swept volume is obtained by integrating the area from β_s to $\beta_d+\beta^*$. The torque is obtained by calculating the force which operates the gate rotor multiplied by the distance from the screw rotor axis.

The leakage line lengths are defined in Fig. 1. Each leakage line length can be calculated by the following formulas.

$$L_1=R_s\sum_{L1}\Delta\theta \tag{16}$$

$$L_2=\sum_{L2}\sqrt{(R_s\Delta\theta)^2+(\Delta Z)^2} \tag{17}$$

$$L_3=\sum_{L3}\sqrt{(R_s\Delta\theta)^2+(\Delta Z)^2} \tag{18}$$

$$L_4=Zq_d-Zq_s \tag{19}$$

$$L_5=\overline{p_dq_d}+\overline{p_sq_s} \tag{20}$$

$$L_6=2R_g\cdot\beta^* \tag{21}$$

The single screw compressor we are developing has two slide valves to control the cooling capacity (Fig. 2). When the slide valve is moved to control the cooling capacity, the area of the bypass port and the discharge port are changed. Therefore it is necessary to calculate the area at each slide displacement. In this analysis it is assumed that the area consists of two or three triangles or rectangles. It becomes possible to calculate the area automatically at each slide displacement.

ANALYSIS OF PERFORMANCE

To calculate the leakage of the single screw compressor it is assumed that the leakage flow is isentropic. Mass flow rate $\dot{G}$ is given by the following formula (R-22).

$$\dot{G}=F\cdot A\sqrt{\frac{2n}{n-1}\;\frac{P_1}{v_1}\;\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}\qquad P_1/P_2<1.8 \quad (22)$$

$$\dot{G}=F\cdot A\sqrt{n\,\frac{P_1}{v_1}\;\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}}\qquad P_1/P_2>1.8 \quad (23)$$

The mass flow rate from the bypass port and the discharge port is calculated by using the same formula. As the area of the bypass port and the discharge port changes depending on the screw rotor rotational angle, it is necessary to calculate the area at each screw rotational angle.

The groove pressure of the screw rotor can be calculated by using those conditions. At each angle the leakage mass flow rate is calculated, and the pressure and temperature in the groove are obtained by solving the equation of state.

$$\frac{dP_c}{P_c}=n\left(-\frac{dv_c}{v_c}+\frac{T_i}{T_c}\frac{dG_i}{G_c}+\frac{dG_o}{G_c}\right) \qquad (24)$$

The pressure and temperature of the forward neighbor groove are not known, therefore the calculation is repeated until the differance between the predicted and corrected pressure is below the tolerance limit.

The volumetric efficiency, the adiabatic efficiency and the compression torque are calculated from the pressure, temperature and the mass flow rate.

RESULT OF ANALYSIS

Fig. 3 shows the relation between the screw rotor angle and the volume. The volume change starts at 0 degrees, and ends at 160 degrees. It is linear from 0 degrees to 120 degrees.

Fig. 4 shows the bypass and the discharge area ratio against the screw rotor angle. This area ratio is divided by the screw rotor outer surface area $[\pi R_g(Z_s+Z_d)]$. 100% load means full loading; 30% load means unloading by the slide valve moving. The area changes like a sine curve.

Fig. 5 and Fig. 6 show the relation between the leakage line length (normalized by $2\pi R_s$) and the screw rotor angle. Fig. 5 shows the leakage line of the screw rotor. Fig. 6 shows the leakage line between the gate rotor and the casing. The leakage line length between the screw rotor and the casing is the longest. Therefore the clearance between the screw rotor and the casing is critical for the performance.

Fig. 7 shows the calculated volumetric and the adiabatic efficiency. The volumetric efficiency changes linearly with the pressure ratio. The adiabatic efficiency has a peak at pressure ratio 4, because the built-in volume ratio is fixed.

Fig. 8 shows the fluctuation of the compression torque. The compression of the single screw compressor occurs every 60 degrees of the screw rotor angle. Therefore the torque fluctuation is small(±10%). Then the noise and the vibration is lower than our reciprocating compressor.

PRESSURE MEASUREMENTS

It is necessary to measure internal groove pressure of the single screw compressor. As it is difficult to plug the sensor into the screw rotor, the pressure sensors are plugged into the casing. The pressure is measured and stored in the digital memory and is processed by a personal computer.

The pressure sensor made of crystal is a piezo type. It has high reliability and durability. Five pressure sensors are plugged. Fig. 9 shows the pressure measurement system. Each sensor covers about 40 degrees of the screw rotor angle. The pressure change is transduced by the sensor and the charge amplifier. The pressure change is fast (8.3msec/cycle), and it is necessary to measure in the same time. The digital memory stores the pressure data.

The data is sent to the personal computer for each channel. The rotational speed is measured by the frequency counter. The personal computer transformes the pressure time history to the pressure-rotor angle relationship. Fig. 10 shows the pressure change of each sensor. NO.1 is the suction side pressure, and NO.5 is the discharge side pressure.

As the sensor is the piezo type, it only measures the alternative part of the pressure. It is necessary to calibrate by the other pressure sensor. The strain gauge type is used as a reference.

RESULT OF PRESSURE MEASUREMENT

The experiment is conducted by using the oil injection-free single screw compressor. The motor nominal output is 37 kw.. Fig. 11 shows the relation between the pressure and the screw rotor rotational angle in the optimum pressure condition. The experimental result agrees fairly well with the calculated result.

Fig. 12 shows the pressure and the volume ratio. The volume ratio is the groove volume divided by the swept volume. The over-shooting pressure is a energy loss, and it must be reduced.

CONCLUSIONS

The performance analysis of the single screw compressor is shown. It is possible to predict its performance and to calculate the force and the torque change, etc., by using a computer.

The internal groove pressure of the single screw compressor is measured using digital memory, a personal computer and piezo type pressure sensors and compaired with the analysis. It is proved that the analysis is satisfactory.

Furthermore, for the single screw compressor with slide valves it is easy to predict the performance of the unloading operation.

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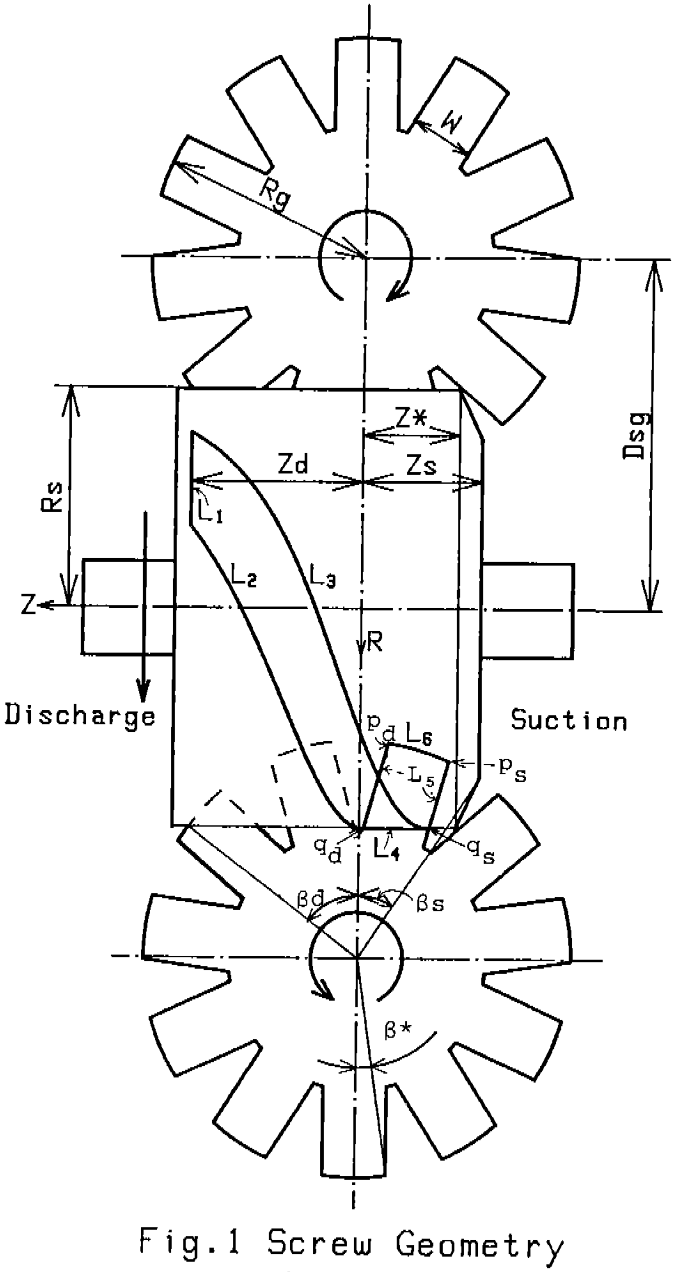

Fig.1 Screw Geometry

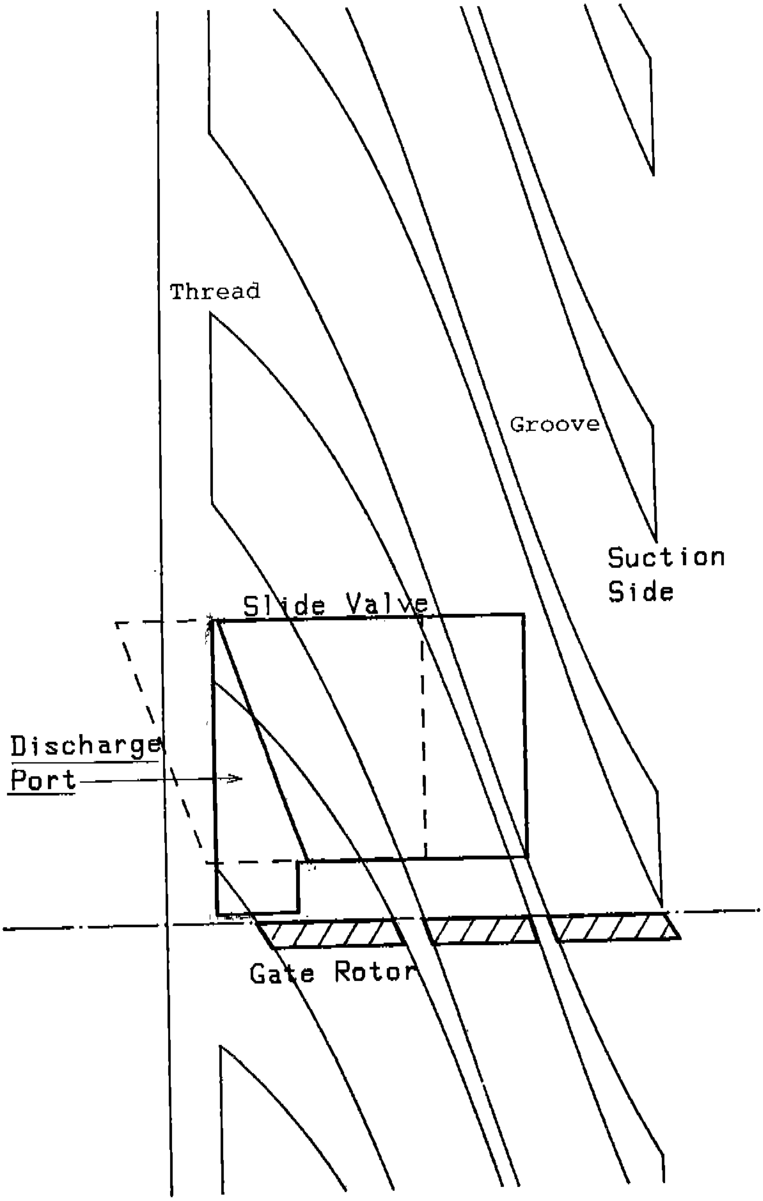

Fig.2 Slide Valve

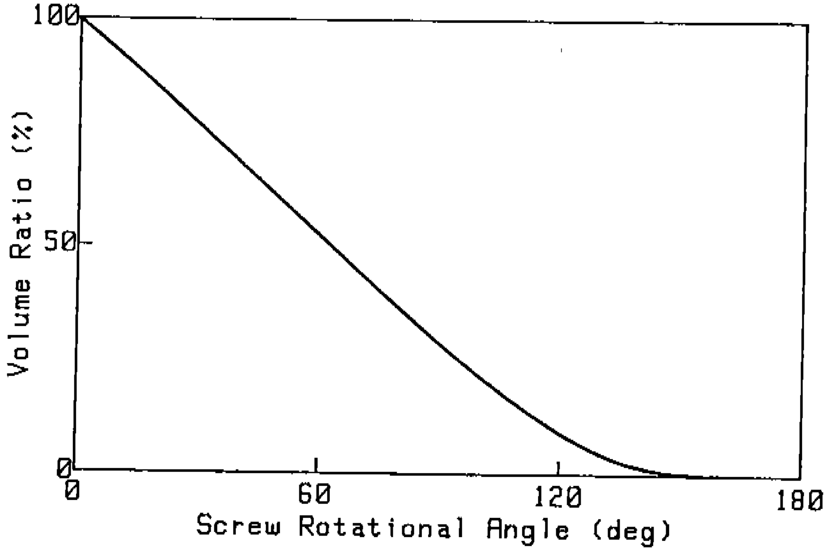

Fig.3 Volume Change

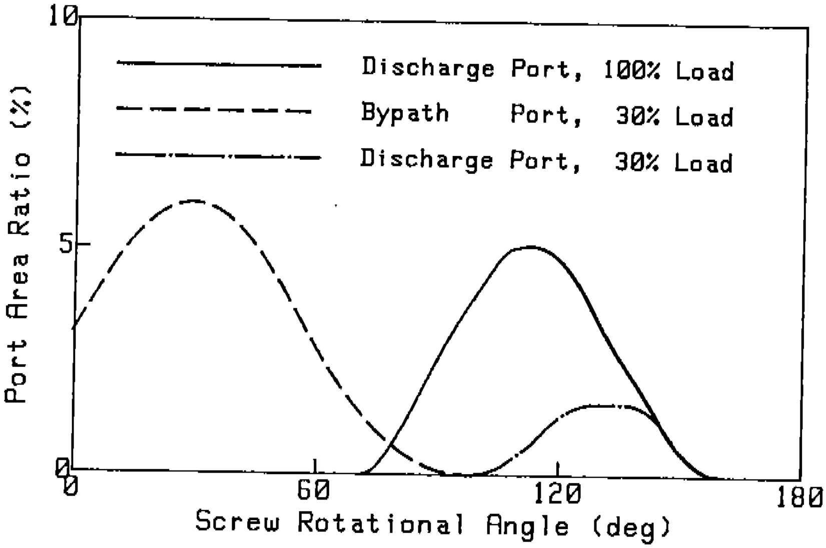

Fig.4 Port Area

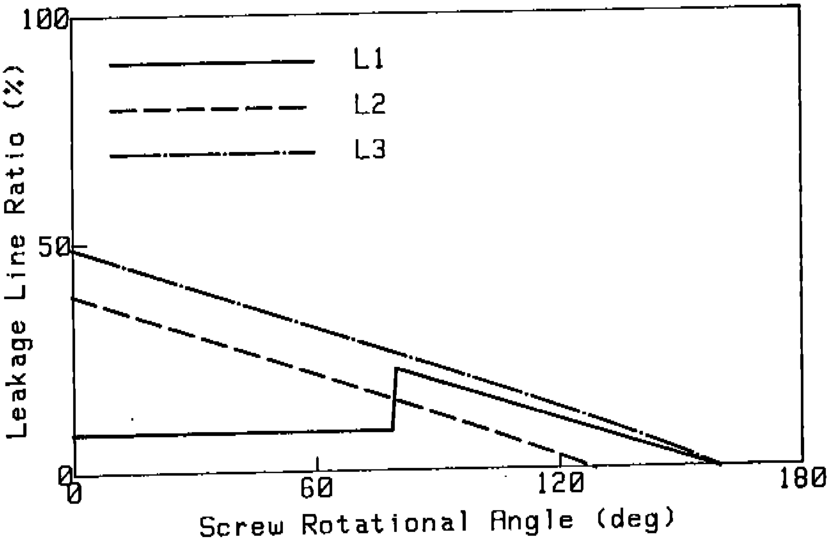

Fig.5 Leakage Line (Screw Rotor)

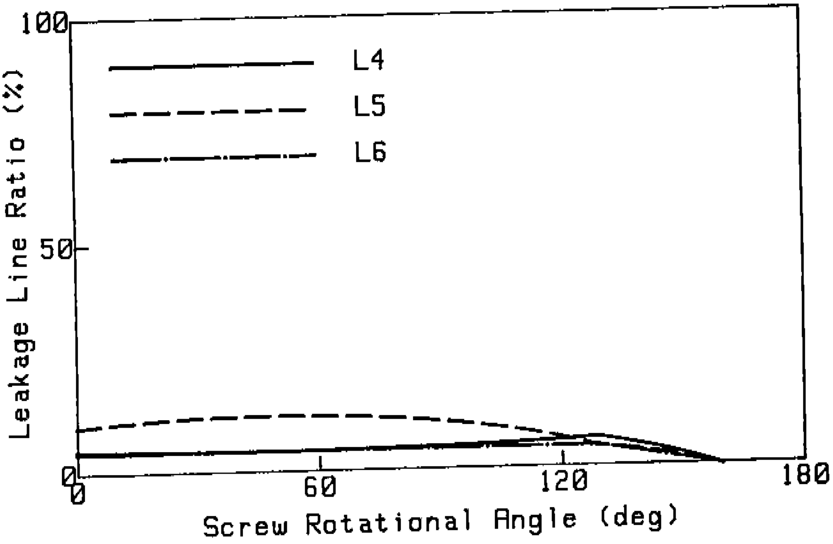

Fig.6 Leakage Line(Gate Rotor)

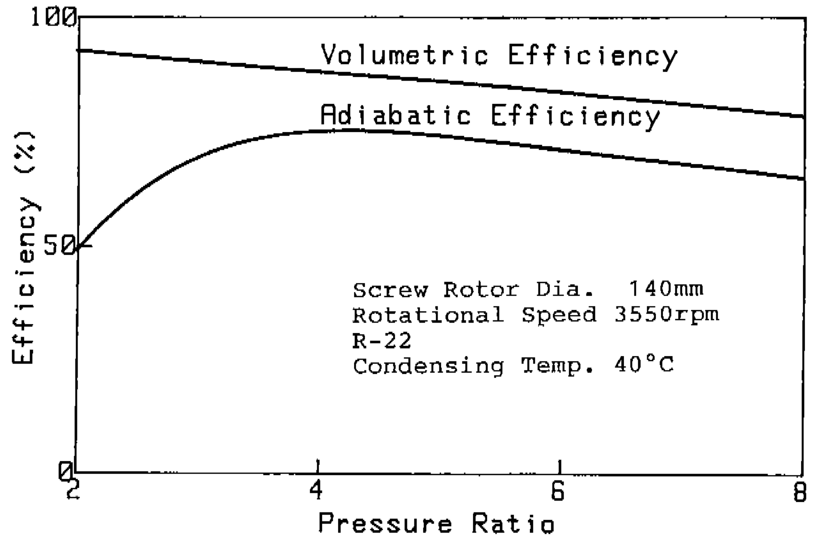

Fig.7 Efficiency

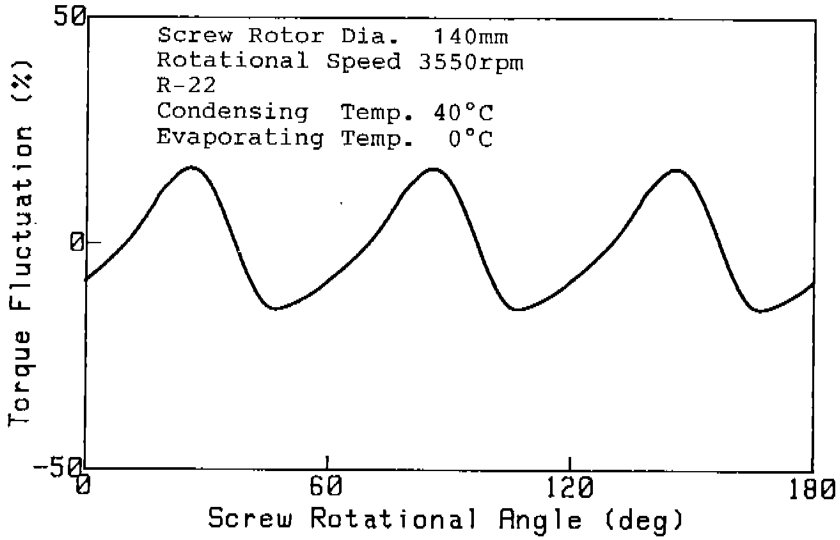

Fig. 8 Torque Fluctuation

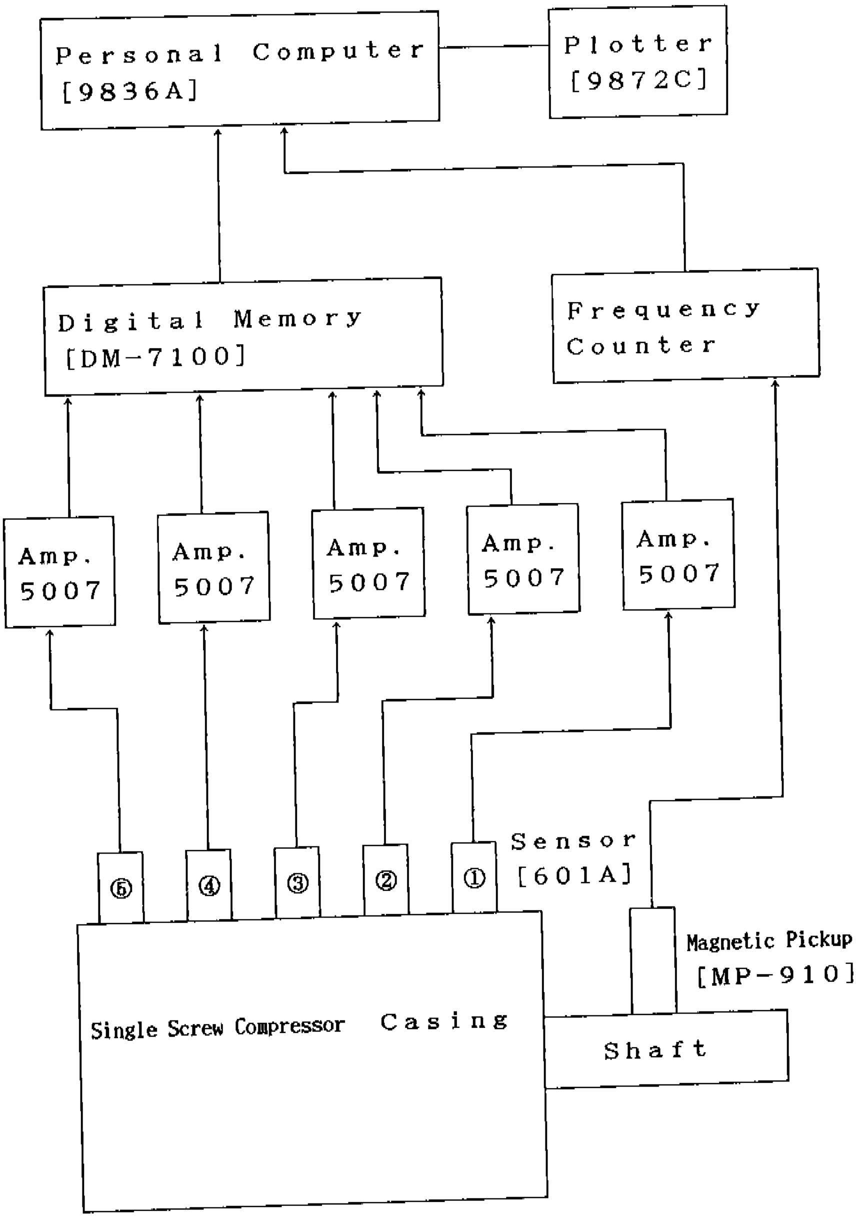

Fig. 9 Measurement System

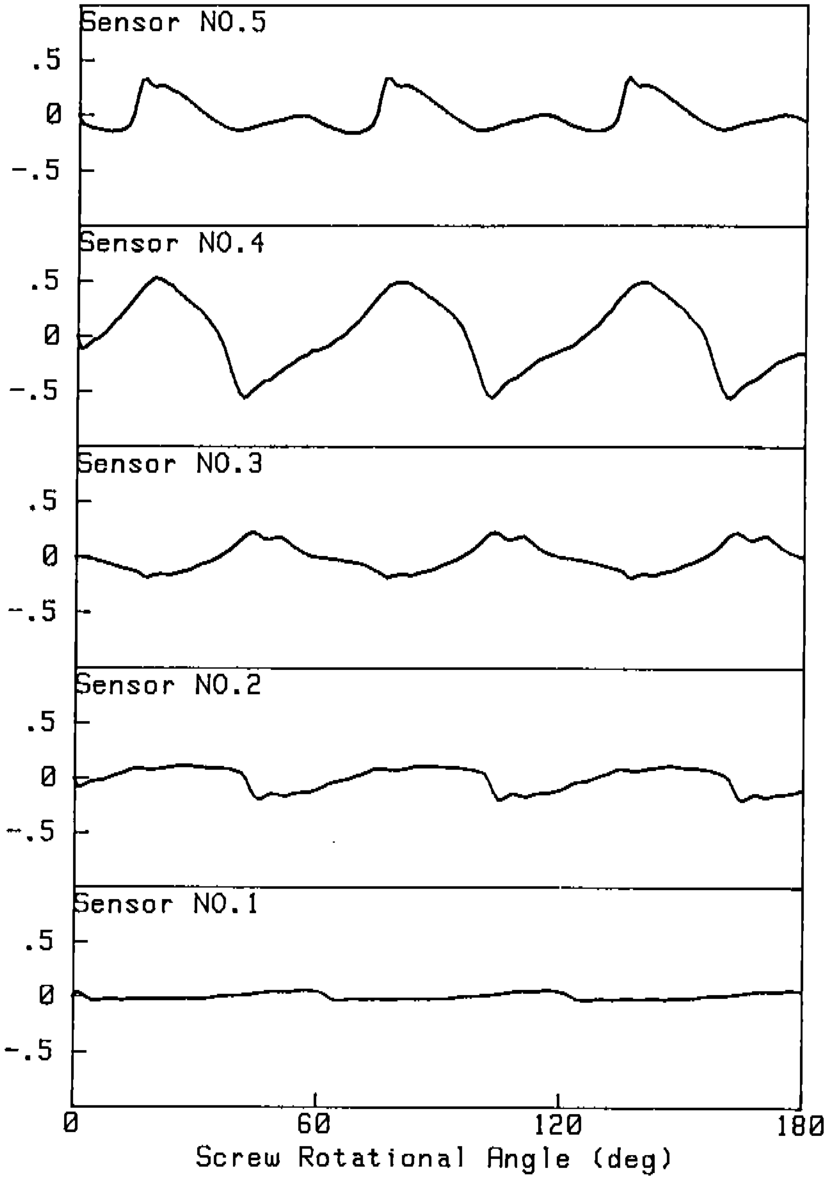

Fig.10 Pressure Data

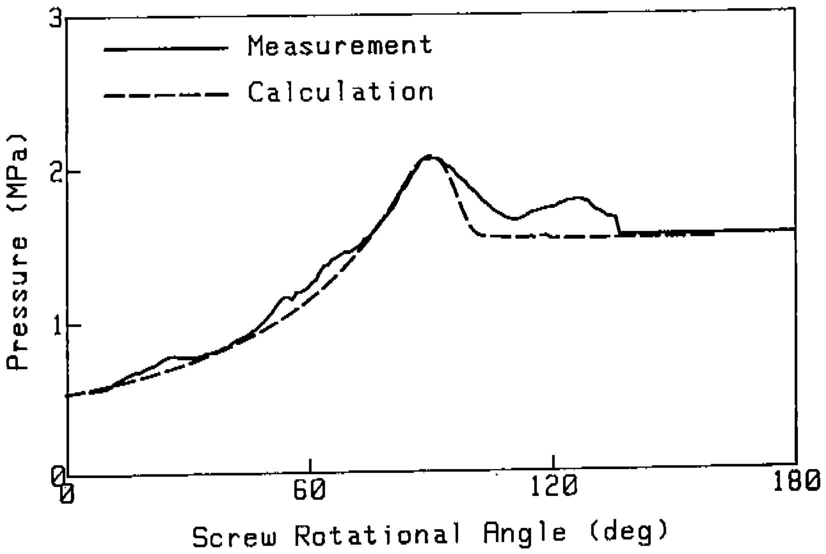

Fig. 11 Pressure Change

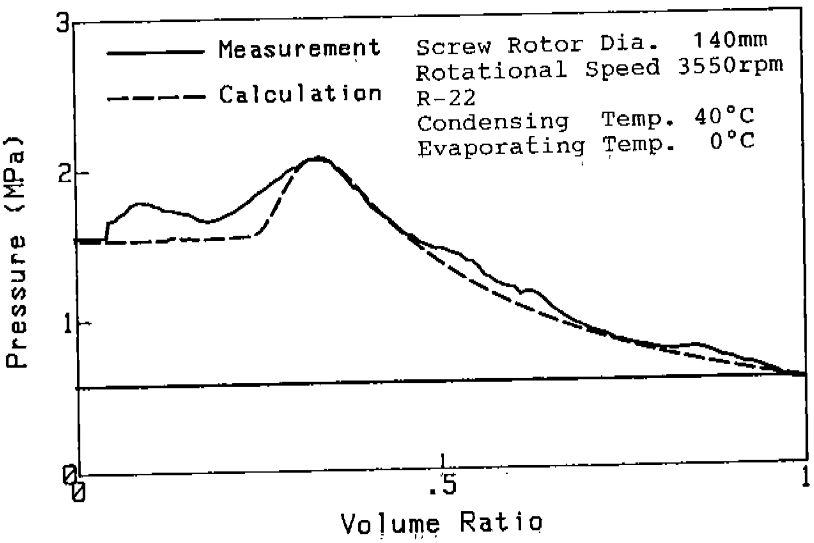

Fig.12 P-V Curve

HEAT TRANSFER IN OIL-FLOODED SCREW COMPRESSORS

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ABSTRACT

Thermodynamic efficiency of the compression process in oil-flooded screw compressors depends greatly on the oil-gas heat transfer process. The amount of heat transfer is a function of many parameters such as mode of oil injection, oil inlet temperature, etc. This paper describes a mathematical model to calculate this heat transfer, assuming that the oil is injected in the form of non-interacting spherical droplets. The droplet trajectories are calculated from the point of injection to the point where the droplets hit the moving boundaries of the compressor rotor. The overall heat transfer is calculated by summing the heat exchange over all the droplets during their free-flight time. This model is then used to calculate the effect of such heat transfer on compressor performance. Some guidelines on ways to enhance heat transfer are also provided.