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Performance Analysis of Oil-Injected Screw Compressors and its Applications

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PERFORMANCE ANALYSIS OF OIL-INJECTED SCREW COMPRESSORS AND ITS APPLICATIONS

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ABSTRACT

A computer model for performance analysis of rotary screw compressors was introduced in a previous paper by the authors (1). In this paper, experimentally obtained flow and heat transfer characteristics are used in the performance simulation. Heat transfer coefficient is determined from experimental relation between the volumetric efficiency and the inlet temperature. Flow coefficients are obtained from the efficiency-clearance curves. Applying those coefficients to the performance simulation, good agreements are obtained between tested and calculated performance for three different prototype compressors. A new rotor profile aimed at higher performance is designed based on the simulation results. The tested performance of the new profile compressor is much higher than that of a conventional compressor, as predicted by the simulation. The new profile compressor has been applied to a commercial series of packaged screw compressors.

INTRODUCTION

Computer simulation appears to be a suitable tool for analysis of screw compressor processes and useful in determining the optimum rotor shape which is one of the requirements of high performance. In recent years, many studies have been pursued in simulating compressor performance for both oil-free type and oil-injected type. Fujiwara et al. (1) previously presented a computer modeling for an oil-injected screw compressor, in which the effects of oil on the gas leakage and cooling were considered.

Some flow and heat transfer characteristics are required for the computer simulation, which were empirically assumed in the previous paper. However, these characteristics directly affect the accuracy of the performance prediction. Therefore, correct evaluations of these characteristics are essential in the accurate prediction performance.

Heat transfer between gas and oil is especially important. In a suction process, gas is warmed by high temperature oil and consequently, the compressor performance goes down. On the other hand, gas temperature rises in the compression and discharge processes and the gas is generally cooled by the injected oil, thus reducing power consumption. Singh and Bowman (2) analyzed the movement of oil droplets in a working space and calculated the heat transfer between gas and oil. Stošić et al. (3) also used an oil droplets model and studied the influence of the droplet size on the working process.

From a more practical point of view, the authors tried, in this paper, to determine flow and heat tranfer coefficients using experimental performance data.

As an application of the computer simulation, a new rotor profile aimed at higher performance is designed, and comparisons between experimental and calculated efficiencies are presented.

In the final part of this paper, a commercial series of oil-injected screw compressors applying the new profile rotor, is introduced.

DETERMINATION OF COEFFICIENTS

Heat Transfer Coefficient

The heat transfer coefficient between gas and oil is determined from an experimentally obtained volumetric efficiencies as follows.

The volumetric efficiency η_v is defined by

$$\eta_v = \frac{V_s}{V_0} \qquad (1)$$

where,

V_s = discharged air volume per unit time at the inlet conditions

V_0 = displacement volume per unit time.

Figure 1 shows typical volumetric efficiency curves of an oil injected screw compressor, plotted against inlet air temperature, in which the supplied oil is fixed at 50℃.

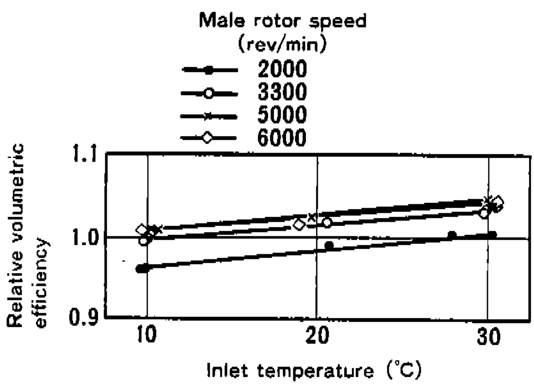

Fig. 1 An example of experimental volumetric efficiency curve presented against inlet temperature (Efficiencies are relative to inlet temperature of 10℃ and rotational speed of 3,300rev/min)

The fall in volumetric efficiency at a lower inlet temperature may be attributed mainly to a higher temperature rise in the inducted air due to heat exchange with the oil. As the oil temperature is fixed in these data, the amount of heat exchange increases as the inlet temperature becomes lower and the charging efficiency goes down.

A schematic model of an air screw compressor in a working space at the end of the suction process is shown in Fig. 2. The working space is filled with inducted air through the inlet port and leakage air from higher pressure working spaces. For simplicity, both airs are treated separately in this model. Heat exchange between oil and leakage air is assumed to be negligible because the temperature difference between them is assumed to be small. The difference in pressure on both sides of the inlet port is also negligible.

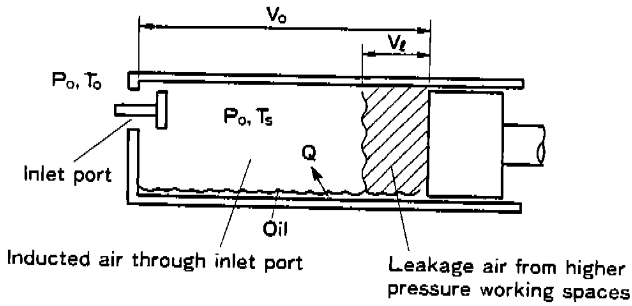

Fig. 2 Air screw compressor model in working space at the end of the suction process

The transferred heat from the oil to the inducted air in a suction process is

$$Q= c_p M_s (T_s - T_0) \quad (2)$$

where,

c_p =specific heat capacity of air at constant pressure
M_s =mass of inducted air
T_s =inducted air temperature at the end of the suction process
T_0 =inlet air temperature.

When the air is assumed to be an ideal gas, the state equations of the inducted air at the inlet conditions (3) and at the end of the suction process (4) are represented by

$$P_0 V_s = M_s R T_0 \quad (3)$$

$$P_0 (V_0 - V\ell) = M_s R T_s \quad (4)$$

where,

P_0 =inlet pressure
V_s =inducted air volume at the inlet conditions
$V\ell$=volume of leakage air at the pressure of P_0
R= gas constant of air

Substituting the value for T_0 from Eq. (3) and T_s from Eq. (4) into Eq. (2) results in

$$Q= \frac{c_p}{R} P_0 V_0 \left(1 - \frac{V\ell + V_s}{V_0}\right) \quad (5)$$

When the temperature rise of inducted air is small compared with the temperature difference between T_{oil} and T_0, Q is also given by

$$Q= Ah (T_{oil} - T_0) t_s \ , \quad (6)$$

where,

A= heat transfer area
h= heat transfer coefficient
t_s =time required for the suction process for the working space
T_{oil} =temperature of leakage oil.

Eliminating Q from Eqs. (5) and (6) results in

$$Ah(T_{oil}-T_0)t_s = \frac{\kappa}{\kappa-1}P_0 V_0 (1 - \frac{V_l+V_s}{V_0}) \tag{7}$$

where,
κ= ratio of specific heats.

Since the oil heat capacity in the working space is so large compared with air, the temperature in the compression and discharge working spaces is little affected by the inlet air temperature. Therefore, T_{oil}, V_l and h are assumed to be independent on T_0. Thus, differentiating both sides of Eq. (7) with respect to T_0, heat transfer coeeficient h is obtained as follows;

$$h= \frac{\kappa P_0 V_0}{(\kappa-1)At_s}\cdot\frac{d\eta_v}{dT_0} \tag{8}$$

Eq. (8) relates h to the tangent of η_v-T curve. Applying test data to this equation, h can be determined. However, no information exists concerning the heat transfer area A. Therefore, in this paper, A is defined as a representative area by

$$A= V_{th}^{2/3} \tag{9}$$

where,
V_{th}= displacement volume per one pair of male and female rotor grooves.

The results are presented in Fig. 3 showing the relationship between Nusselt number Nu and rotational Reynolds number Reω. Where,

$$Nu= \frac{h\cdot D_m}{\lambda} \tag{10}$$

$$Re\omega= \frac{\omega\cdot D_m^2}{\nu} \tag{11}$$

in which,
D_m= male rotor diameter
λ= thermal conductivity of air
ω= rotational velocity of male rotor
ν= kinematic viscosity of air.

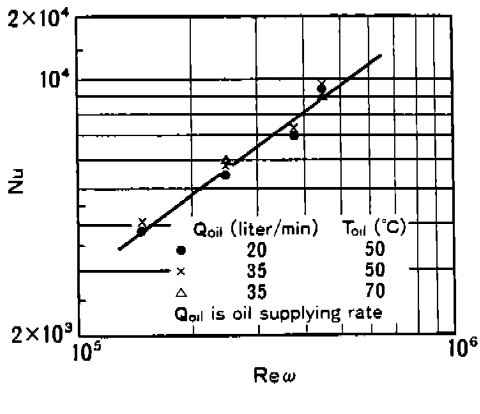

Fig. 3 Experimental relation of Nusselt number versus rotational Reynolds number

In the figure, the logarithmic Nu is represented by a common straight line against the logarithmic Re for three different oil supplying conditions. Though the Nusselt number is determined based on the suction process, it is also applied to the compression and discharge processes in the computer simulation.

Flow Coefficient

In general, screw compressor efficiencies fall with increasing internal clearances. This tendency is also obtained from the computer simulation, but the tangent of a calculated efficiency curve depends on the assumed flow coefficient. Hence, the correct flow coefficient can be obtained if the value is chosen so as to get the best agreement of tangents between the calculated and experimentally obtained efficiency curves.

The authors assumed in the former paper (1) that the lobe tip clearance was filled with oil due to the action of centrifugal force and the oil leakage flow was in a single phase. That has been confirmed by visualizing the working space, as shown in Fig. 4. Therefore, the leakage flow rate through lobe tip clearance is calculated using the equation for incompressible viscous flow, as mentioned in the previous paper.

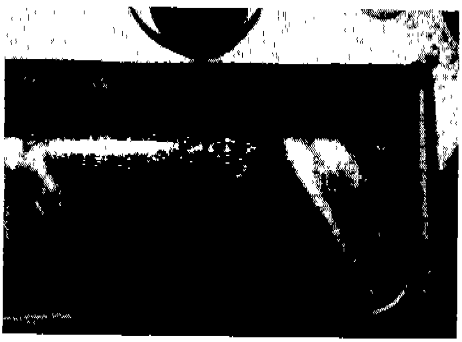

Fig. 4 Oil distribution around lobe tips
(Rotor diameter=212mm, Speed=1900rev/min)

DESIGN OF THE NEW PROFILE

A new rotor profile aimed at higher performance has been developed applying computer simulation. By changing the design parameters such as rotor profile, combination number of teeth, and wrap angle of the lobe, the effects of such parameter changes on compressor performance were studied. The rotor profiles used in the parameter study, in which the number of teeth are variously combined, are shown in Fig. 5.

In addition, machinability of the rotor surface, transmission torque between the rotors, and rotor stiffness must also be considered in the profile design, to improve surface precision, commercial productivity, and compressor reliability.

It is felt that the most promising new rotor profile is that shown in Fig. 6, where the conventional rotor profile is also shown for comparison.

The advantages of the new profile are:

(1) According to the computer simulation, a five and six combination number of teeth offers the highest performance among variously changed ones.

(2) The blow hole area is only 31% of the conventional profile, and the length of the sealing line between rotors is 22% shorter than a conventional one. This results in less air leakage.

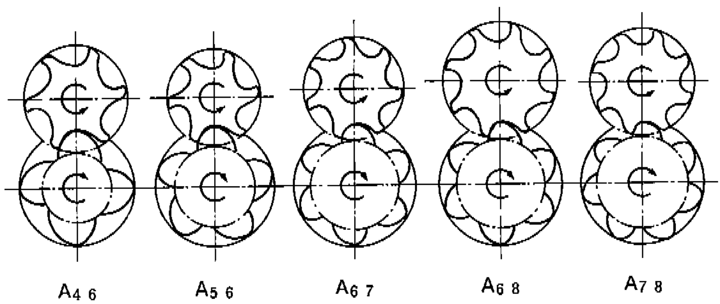

Fig. 5 Profiles of sample rotors for simulation

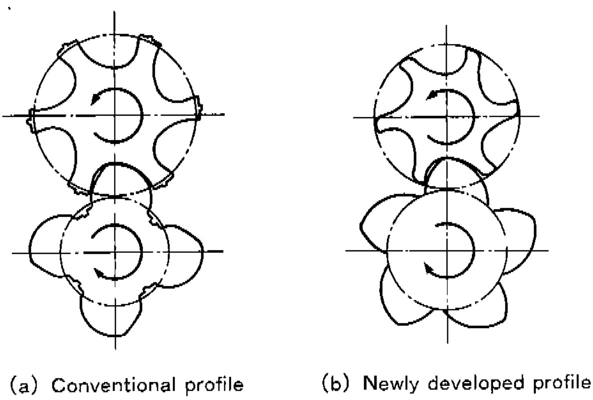

Fig. 6 New and conventional rotor profiles

EXPERIMENTAL PERFORMANCE AND DISCUSSION

Three prototype compressors were made to verify the simulation results. The specifications of these compressors are presented in Table 1. The rotor profile of compressor A is the conventional one shown in Fig. 6(a), while the rotor profile of compressors B and C is the newly developed one shown in Fig. 6(b). V_{th} of compressor C is about twice as large as V_{th} of compressors A and B. A cutaway view of compressor B is shown in Fig. 7.

Performance tests were conducted using the test apparatus shown

Table 1 Rotor specifications of prototype compressors

Compressor	A	B	C
Profile	Fig. 6 (a)	Fig. 6 (b)	Fig. 6 (b)
Combination number of teeth	4+6	5+6	5+6
Outer diameter of male rotor (mm)	102	105	125
Outer diameter of female rotor (mm)	102	84	100
Wrap angle of male rotor	300°	300°	300°
Rotor length (mm)	107	124	175
V_{th} (cc/rev)	545	544	1082
Interlobe clearance (mm)	0.025	0.022	0.022

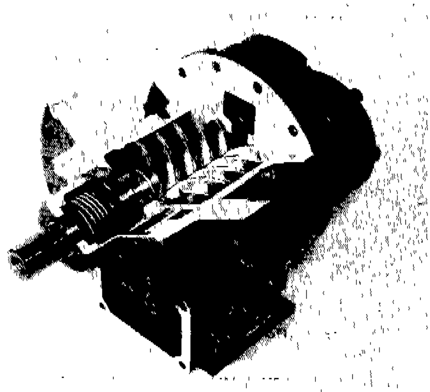

Fig. 7 Prototype of new profile compressor

Table 2 Operating conditions

Rotation speed of male rotor (rev/min)	2000~6000
Discharge pressure (MPa)	0.93
Inlet Pressure (MPa)	0.10
Supplied oil temperature (℃)	50
Supplied oil rate (liter/min)	35

in Fig. 8. Operating conditions of the tests are listed in Table 2

Test results for compressors A and B are shown in Fig. 9, compared with the calculated results of the simulation. The experimental coefficients employed in the simulation are summarized in Table 3. Mechanical losses are not included in these adiabatic efficiencies. It can be seen that the performance of the new profile compressor is much higher than that of a conventional profile one. It is also obvious that the calculated performance accurately

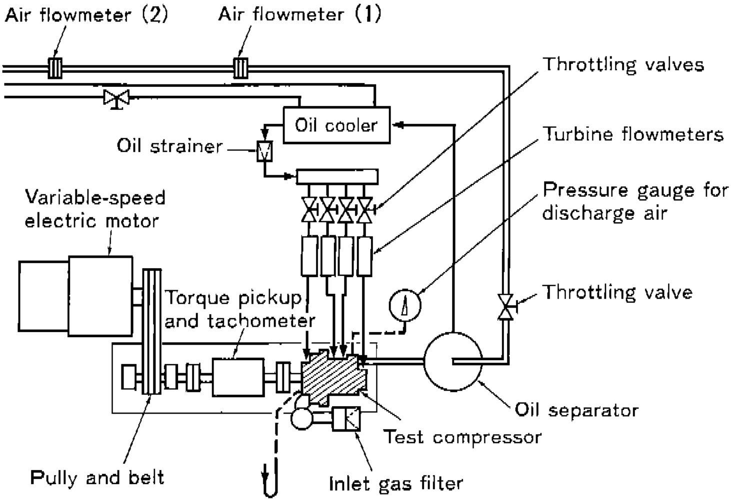

Fig. 8 Test apparatus

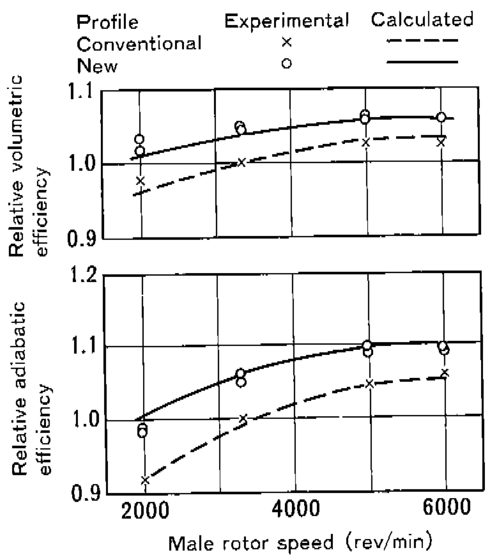

Fig. 9 Peformance test and calculated results – comparison between conventional and newly developed profiles.
(Experimental efficiencies with the conventional profile at the male rotor speed of 3,300rev/min are taken as 1.0)

Table 3 Flow coefficient values

Path name	Value
Interlobe clearance	0.7
Clearance between lobe tip and casing bore	0.7
Blow hole on compression side	0.7
Blow hole on expansion side	0.6
Clearance between rotor end and casing wall	0.4
Discharge port	0.6
Inlet port	1.0

correlates with the experimental results, despite the fact that the heat transfer coefficient determined for the suction process was also applied to the compression and discharge process using the principle of similitude.

The tested results for compressor C are shown in Fig. 10, compared with the calculated results. Both results are in reasonable agreement with each other.

It is concluded that the performance is well predictable, using previously described heat transfer and flow coefficients, even for compressors with different profile and size.

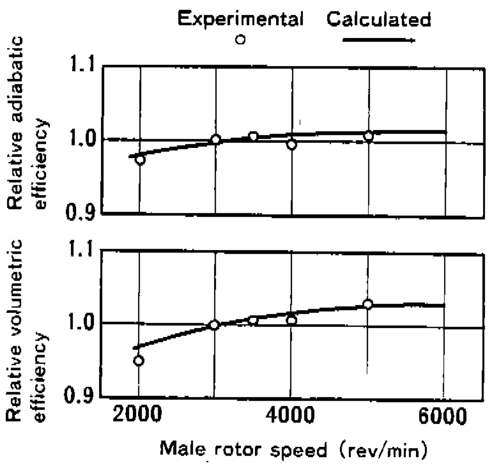

Fig. 10 Comparison between experimental and calculated efficiencies for the prototype compressor C
(Experimental efficiencies at the male rotor speed of 3,000rev/min are taken as 1.0)

OIL-INJECTED SCREW COMPRESSORS APPLYING THE NEW PROFILE

The previously mentioned new profile rotors have been used in commercial compressor units with capacities of 2.2 kW to 110 kW.

The design specifications for the air ends are determined using the present method. Although the five teeth male rotor requires a special tool for measuring its outer and inner diameters, high priority is given to performance merits and such a tool has been developed . Every surface of the rotor lobes is finished precisely by a computer-controlled grinding machine to reduce the clearances.

All compressors are set at a wrap angle as much as 300°. A large wrap angle is efficient in reducing internal leakage and discharge flow resistance, which was predicted by computer simulation. Thus, compressor performance is highly improved, compared with conventional compressors, and meets energy-saving requirements very well.

Advantages of this series in addition to higher performance include:

(1) Low noise operation: 22 kW models of this series reaches sound power level of only 65 dB(A).

(2) High reliability:the maintenance cycle is 24,000 hours.

(3) The oil content of the air is so small as to be 0.02 cc/m³.

(4) Efficient capacity control.

(5) Operation and viewing at the remote location are available, by using high electronics technology.

SUMMARY AND CONCLUSIONS

Heat transfer coefficient between gas and oil in an oil-injected screw compressor has been determined from experimental observations that volumetric efficiency decreases with decreasing inlet air temperature. The relation of the Nusselt number to the Reynolds number has been represented by an exponential function of a single term. Flow coefficients have also been determined from experimentally obtained compressor efficiencies.

A new rotor profile aimed at higher performance has been designed as an application of performance simulation. Close agreements have been obtained between experimental and calculated efficiencies for both the new rotor profile and a conventional profile. The performance of the new profile compressor is much higher than a conventional profile one, as was predicted by the simulation.

The new rotor profile has been applied to the commercial series of oil-injected screw compressors.

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