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# SINGLE STAGE, OIL-FREE SCREW COMPRESSOR WITH A COMPRESSION RATIO OF EIGHT

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## ABSTRACT

The development of a series of single-stage, oil-free screw air compressors with a compression ratio of 8, previously obtainable only with two-stage compressors, is described. A new rotor profile, which reduces leakage loss to achieve higher efficiency, and designed for ease of manufacture,is detailed. Additionally, a new design method for the clearance between rotors, to compensate for rotor deformation due to thermal expansion, is also introduced. This oil-free, screw air compressor with a rating of 37-55 kW has been marketed since 1982 and 15-22 kW since 1984.

## INTRODUCTION

Oil-free air compressors having a discharge pressure of approximately 0.8 MPa (8 ata), and widely used in electrical, food and chemical industries were traditionally reciprocating type compressors. Recently, however, the advantages of oil-free rotary screw compressors have been recognized. These include mechanical simplicity, high reliability,low noise and low vibration. Two-stage, oil-free screw air compressor use has predominated in the capacity range above 550 $m^3/h$ (motor power above 65 kW).
Nevertheless, for compressor capacity below this range, where oil-free air is needed, this type of compressor was not available.

Due to the complicated construction of two-stage, oil-free screw compressors, it was thought that they would not be economical, compared to reciprocating oil-free compressors, even if development in the range below 550 $m^3/h$ was possible. Therefore, development of single-stage compressors was considered preferable because of their simple construction and anticipated low cost. Several problems remained, relating to severe operating conditions, such as the higher compression ratio and resulting increase in temperature of the discharge air. Even given solutions to these difficulties, performance and reliability were considered to be too poor to warrant use.

Addressing these problems, a new rotor profile, yielding higher efficiency at a high compression ratio was initially developed, followed by a new design method of the rotor clearance. This method takes into account of rotor deformation due to thermal expansion during operation and optimizes clearance. This optimization greatly affects compressor efficiency. These fundamental improvements enabled development of a single-stage, oil-free screw compressor with a compression ratio of 8. Such a ratio was previously considered attainable only with two-stage compression. Applying these developments, a packaged,oil-free screw compressor with a rating of 37 - 55 kW was made, representing the first and smallest oil-free screw air compressor. This paper describes the developmental process of the new compressor.

## TECHNICAL OBJECTIVES

A screw compressor is a rotary positive displacement compressor having one male and female rotor meshing with each other and housed in a casing. Air drawn into the chamber between the rotors and the casing walls is compressed due to rotor motion. Air leakage, occuring as a result of clearance between the rotors as well as between the rotors and the casing walls, reduces compression efficiency. Oil-injection screw compressors seal these clearances with an oil film in the compression chamber. Oil-free compressors lack sealing oil in the chamber to prevent this leakage.

This is especially true with single-stage compressors having higher compression ratios, and greatly reduced efficiency as a result.

Fig.1 shows how the clearance between rotors (interlobe clearance) of an oil-free, screw air compressor influences the compression efficiency by computer simulation. In this figure, volumetric and overall adiabatic efficiency at a male rotor tip speed of 100 m/s and a clearance ratio of 0.001 are taken as 1.0. Increasing rotor rotation speed is one solution to this problem, but not a satisfactory one. The fundamental solution is to minimize clearance so as to lessen air leakage. Thus, the first technical problem was development of a new rotor profile with minimum clearance for minimum air leakage, along with machining technology improvements enabling a high-precision rotor finish.

The second difficulty concerned is the air temperature increase during compression and the resulting thermal expansion of the rotor. Air temperature rises with increase in the compression ratio. The lubricant in oil-injection screw compressors also functions as a coolant, permitting the temperature rise to be controlled.
In the case of oil-free rotary screw compressors:
(a) Hot air leaked from the high-pressure side can heat the low-pressure, low-temperature air.
(b) The temperature of this heated air increases further after compression.
(c) This air again leaks and heats up the low-temperature air.
Thus, the air temperature progressively rises. Heated by this air, the rotors expand. This reduces the rotor clearance to a critical degree. Contact between them causes compressor damage or failure. Hence, conventional oil-free screw air compressors operating at a compression ratio above 4 had a two-stage design with an intercooler.

Fig.2 shows a comparison of operating conditions between a two-stage and a single-stage screw air compressor at a discharge pressure of 0.8 MPa (8 ata). The compression ratio of each stage of the two-stage compressor is 3 and the discharge air temperature is less than 150°C. The compression ratio of the single-stage compressor is 8 and the discharge air temperature may be higher than 300°C.
The temperature difference implies very severe operating conditions for the single-stage compressor.

Thus, the technical challenge involves development of a new single-stage, oil-free rotary screw compressor with reduced internal air leakage and optimum clearance compensation for the thermal expansion of the rotors.

Furthermore, Fig.1 indicates that one performance characteristics of oil-free screw compressors involves lowering of efficiency with slower rotor tip speed.

Therefore, small compressor rotors need to be driven at a higher rotation speed to obtain higher efficiency. The resulting rotor vibration increase must be taken into consideration. The technical problems to be solved in the development of a sophisticated compressor are sumarized in Fig.3.

## TECHNICAL SOLUTIONS

In the light of these difficulties, intensive R&D efforts to develop new technology were made by fundamental analysis and experiments. Some of these are outlined below.

### New Rotor Profile

The rotor profile of an oil-free screw compressor is one of the most influential factors in achieving higher efficiency. The reouirements for such a profile are as follows.

(1) Seal lines to prevent air leakage between rotors and between rotors and casing walls should be made. These lines should be short and the clearance should be of a shape that functionally prevents leakage. As an example, surface to surface sealing is more efficient than sharp edge sealing.

(2) The clearance between rotors and between rotors and casing walls should be smaller than one thousandth of the rotor diameter, resulting in a need for super precise rotor manufacturing. Basically, the rotor profile should be a shape suitable for manufacturing.

Given the time and expense involved in developing a screw rotor profile, a computer aided design (CAD) system can be very useful. The CAD system used to develop the new rotor profile is shown in Fig.4. This system is composed of two sub-systems, one estimates screw compressor performance, and the other calculates rotor and cutting hob profiles.

Applying this CAD system, many rotor profiles acceptable for high compression ratio, oil-free screw compressors were investigated. The most promising new rotor profile is shown in Fig.5. This profile has the following advantages over a conventional profile.

(a) The respective lobe combination of five for male and six for female rotors is suitable for higher compression ratio applications, creating one more working chamber than the conventional profile. The additional chamber reduces the pressure difference and compressed air leakage between chambers.

(b) The blow hole area is only 27 % of the conventional profile, and the seal line between rotors is 20 % shorter than the conventional one. Additionally, the new profile is created by continuous, smooth curves. The smaller blow hole, shorter seal line and surface to surface sealing results in less air leakage.

(c) The continuous, smooth curves of the profile made it possible to machine rotors with higher precision by hobbing, resulting in smaller clearance between the rotors.

As a result of computer simulation, the new rotor profile showed 7 % higher volumetric efficiency and 17 % higher overall adiabatic efficiency compared with the conventional rotor profile (Fig.6).

Compensation for Rotor Thermal Expansion

It is neccessary to know rotor temperatures for optimization of rotor clearances. Fig.7 shows rotor temperatures of a prototype single-stage, oil-free air compressor measured at a discharge pressure of 0.8 MPa (8 ata), compared with a calculated rotor temperature. The discharge side rotor temperature was approximately 300 °C. This temperature was much higher than predicted beforehand. A new design method was neccessary to minimize the rotor clearance along the whole rotor profile at such a high temperature.

The fundamental design concept of the new method is shown in Fig.8.

Ⓐ is an initial profile of a male rotor at room temperature, from which the heat-expanded profile Ⓑ is obtained. With allowances for timing gear backlash and rotation reliability, profile Ⓒ is determined. The heat-expanded female rotor profile Ⓓ is generated by Ⓒ. The female rotor profile at room temperature Ⓔ is obtained by contracting Ⓓ. Cutting hob profiles are transferred from rotor profiles Ⓐ and Ⓔ.

Applying this method, the rotor clearance is optimized at a higher operating temperature. The new design produced a 40-50 % smaller clearance compared with a conventional design. This is approximately equal to a 13 % higher overall adiabatic efficiency at a rotor tip speed of 100 m/s according to Fig.1.

## PROTOTYPE COMPRESSOR

Following this fundamental research, several prototype single-stage compressors were made, all being applicable to the 37-55 kW ratings. Some of them had the new rotor profile while others had the conventional one.

### Compressor Structure

Figure 9 shows a cutaway view of a new single-stage compressor. Female and male rotors are radially suported by cylindrical roller bearings and axially by combined angular ball bearings. A pair of timing gears at the end of rotor shafts adjusts the clearances between the lobes of the rotors. Therefore, these lobes do not contact one another. A pinion gear at the drive end of the male rotor is driven by a large gear contained in a gear casing on which the compressor is flang-mounted horizontally. The male rotor was tested at up to 22,000 rpm.

Visco-type seals prevent the lubricant oil from entering the compression chamber while carbon rings seal the air. An oil hole is present in the rotor shaft for cooling. A special heat-proof coating is applied to the rotor surfaces. The casing of the compressor is cooled by a water jacket.

### Performance of the Prototype Compressor

Performance test results of the prototype compressors are shown in Fig.10. The overall adiabatic efficiency, the volumetric efficiency and the increase in discharge air temperature of the conventional profile compressor at a compression ratio of 8 (design compression ratio) are taken as 1.0. Volumetric efficiency and overall adiabatic efficiency obtained with the new rotor profile are greatly improved, and the increase in discharge air temperature is reduced. The higher the compression ratio, the higher the improvement. This indicates significant reduction in air leakage loss by means of the new rotor profile. At the design compression ratio (8), the relative efficiency improvements from the conventional to the new rotor profiles are 27 % in volumetric efficiency and 38 % in overall adiabatic efficiency.

The performance characteristics of the new profile compressor at various rotation speeds is shown in Fig.11. A very flat overall adiabatic efficiency curve is seen over a wide range of rotor rotational speed.

These test results allowed development of a new machine able to deliver 0.8 MPa (8 ata) air by a single-stage compression. Table 1 lists standard specifications of a series of single-stage, oil-free screw air compressors.

## CONCLUSION

The development of a single-stage, oil-free screw air compressor was described. This is the world's first and smallest single-stage compressor operating at a compression ratio of 8, previously obtainable only with two-stage compression.

As a result of intensive R&D effort, reliable and low cost oil-free screw compressors became available in the capacity range from 120 to 455 $m^3/h$. Formerly, oil-free reciprocating compressors had predominated in this range.

## ACKNOWLEDGMENTS

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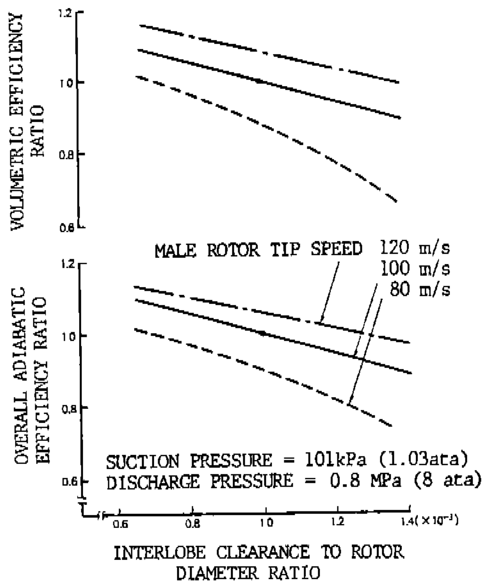



Fig.1: Relation between Interlobe Clearance and Compressor Efficiency by Computer Simulation (Volumetric efficiency and overall adiabatic efficiency at a male rotor tip speed of 100 m/s and an interlobe clearance ratio of 0.001 are taken as 1.0.)

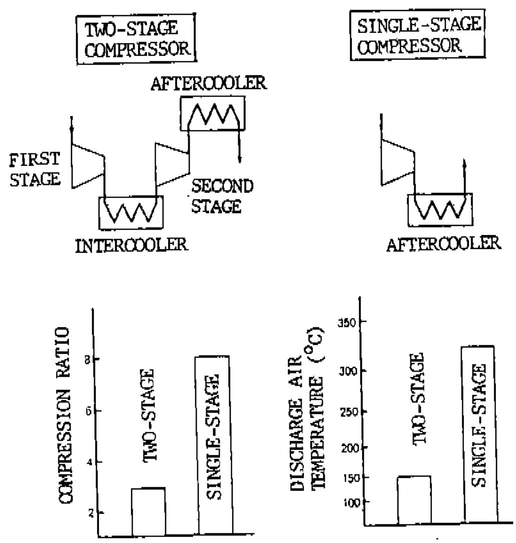



Fig.2: Comparison of Operating Conditions between Two-stage and Single-stage, Oil-free, Screw Air Compressors (Discharge air temperature is calculated assuming an adiabatic efficiency of 0.8.)

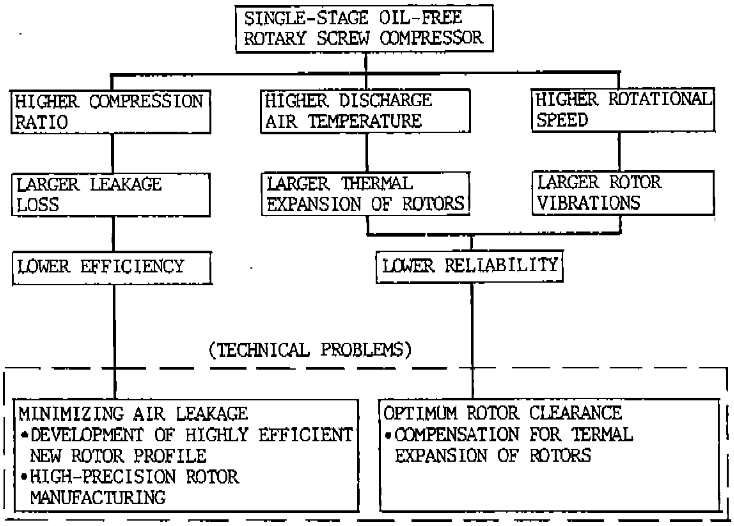



Fig.3: Technical Problems in the Development of a Single-stage Compressor

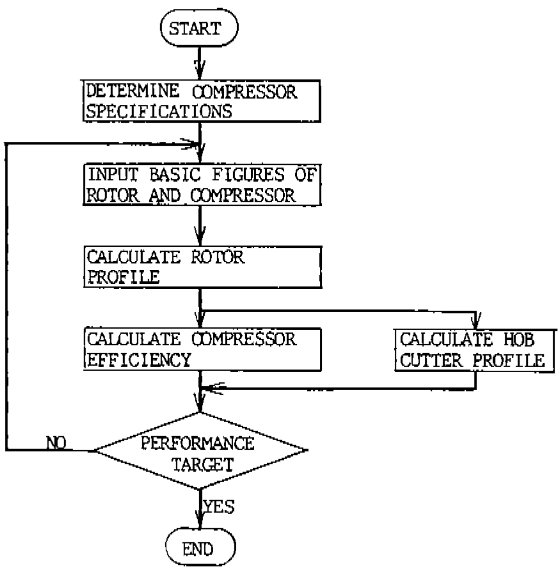



Fig.4: CAD System to Develop a New Rotor Profile

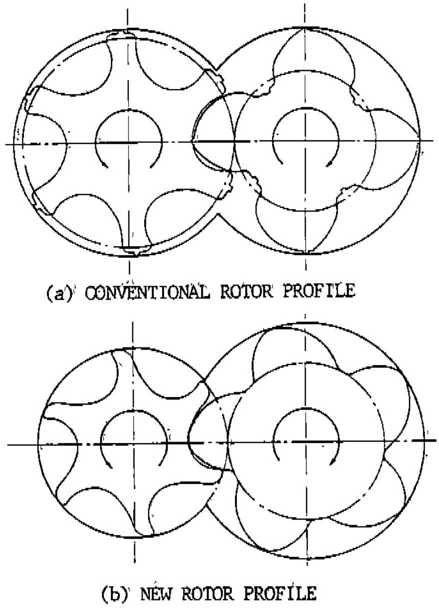



Fig.5: New and Conventional Rotor Profiles

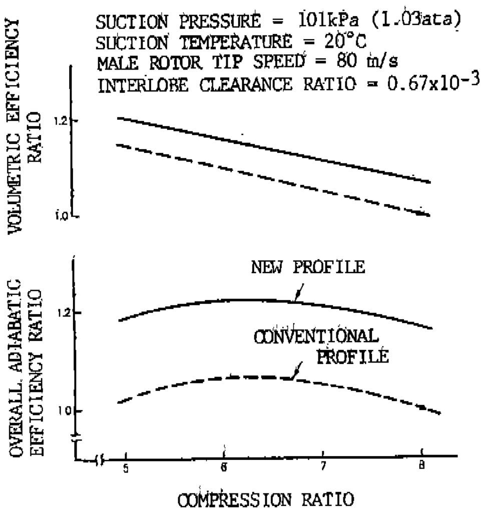



Fig.6: Comparison of Compressor Efficiency between New and Conventional Rotor Profiles by Computer Simulation (Compressor efficiencies with the conventional profile at the compression ratio of 0.8 are taken as 1.0.)

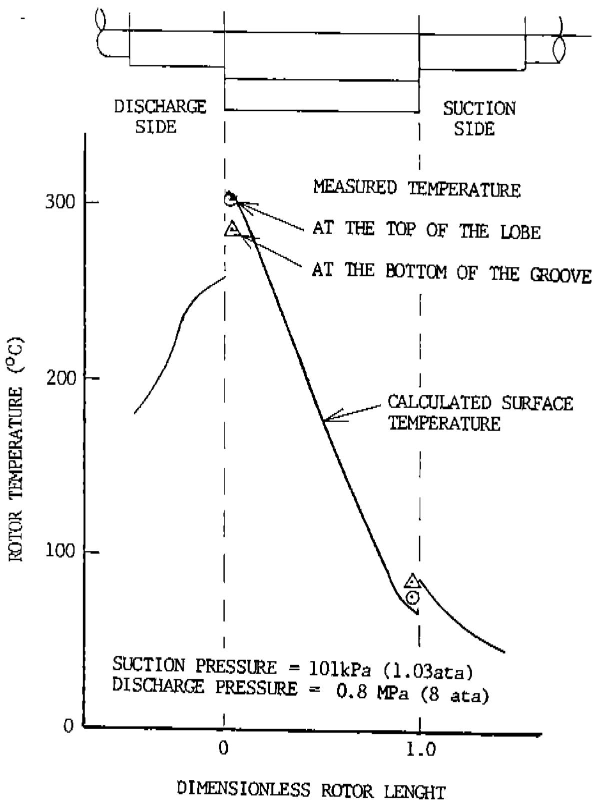
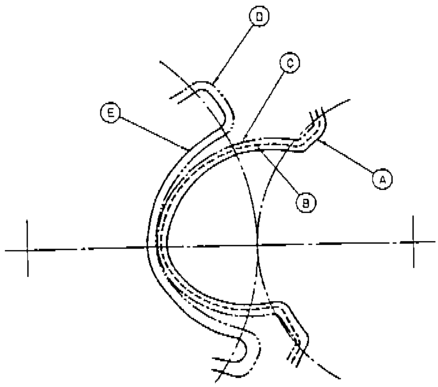



Fig.7: Measured and Calculated Rotor Temperatures of a Single-stage, Oil-free Screw Compressor (Rotors are non-cooled.)




A MALE ROTOR PROFILE AT ROOM TEMPERATURE
B HEAT-EXPANDED MALE ROTOR PROFILE
C PROFILE WITH AN ALLOWANCE FOR GEAR BACKLASH

D HEAT-EXPANDED FEMALE ROTOR PROFILE GENERATED BY C
E FEMALE ROTOR PROFILE AT ROOM TEMPERATURE

Fig.8: Compensation for Rotor Thermal Expansion

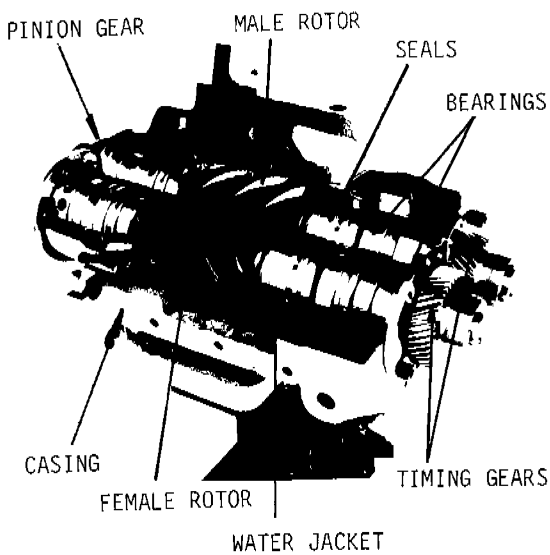
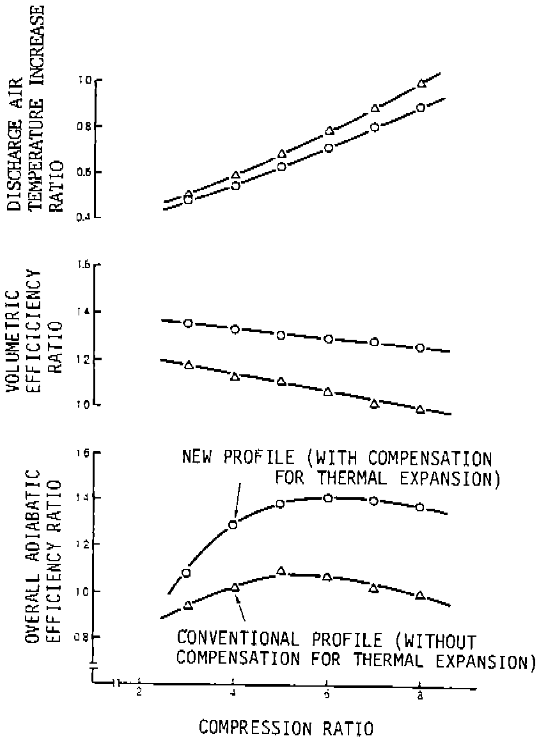
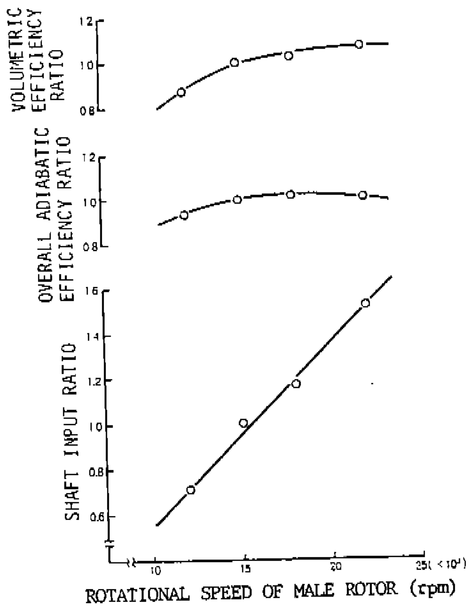



Fig.9: Prototype Compressor




SUCTION PRESSURE = 101kPa (1.03ata)
SUCTION AIR TEMPERATURE = 20 °C

Fig.10: Performance Test Results (Efficiencies and discharge air temperature increase of the compressor with the conventional profile at the compression ratio of 8 are taken as 1.0.)




SUCTION PRESSURE = 101kPa (1.03ata)
SUCTION AIR TEMPERATURE = 20 °C
DISCHARGE PRESSURE = 0.8 MPa (8 ata)

Fig.11: Performance Characteristics of the New Profile Compressor versus Rotational Speed (The characteristics at 15,000 rpm are taken as 1.0.)

Table 1: Standard Specifications

| MODEL | DSP-20CA | DSP-30CA | DSP-20WA | DSP-30WA | DSP-50WA | DSP-60WA | DSP-75WA |
|---|---|---|---|---|---|---|---|
| MOTOR POWER(kW) | 15 | 22 | 15 | 22 | 37 | 45 | 55 |
| CAPACITY($m^3$/h) | 120 | 180 | 120 | 180 | 280 | 365 | 455 |
| DISCHARGE PRESSURE(ata) | 8 | | | | 8.7 | | |
| COOLING METHOD | AIR COOLED | | WATER COOLED | | | | |
| DIMENSION(LxWxH) (mm) | 1,100x1,100x1,100 | | 1,100x700x1,100 | | 1,550x770x1,200 | | |
| NET WEIGHT(kg) | 575 | 625 | 500 | 550 | 750 | 780 | 800 |
| NOISE LEVEL dB(A) | 70 | 72 | 68 | 70 | 72 | 72 | 72 |

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