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A SCROLL COMPRESSOR WITH SEALING MEANS AND LOW PRESSURE SIDE SHELL

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ABSTRACT

A scroll machine has not been attempted manufacturing for long years because of the difficulties in productivity, in spite of the advantageous characteristics. But in recent years, productive techniques have remarkably developed, and several manufacturers have been paying their attentions to a scroll machine. A few of them have already commercialized.

Authors also remarked a scroll machine for refrigerant equipment, and have accomplished manufacturing engineering of a scroll machine as a hermetic compressor for medium capacity range air conditioners. The scroll compressor was designed to achieve axial and radial sealing by simple means, and cool down the motor by suction gas. Therefore, it is possible to attain high efficiency and high reliability under wide range of compression ratio. And the scroll compressor is suitable for the application to heat pump air conditioners for these reasons.

INTRODUCTION

After Energy Crisis, energy conservation has been the most emphasized topics for air conditioners. In Japan, the regulation of the efficiency started in 1979, in the range of over 3Kw. Besides the legal demand, air conditioners with higher efficiency have been required from the distributors to meet the customer's needs concerning energy saving. These requirements remains even in the latest few years, although there seems to be sufficient oil supply.

Under such a situation, there came a strong demand for higher efficiency compressors, because the improvement of compressor efficiency is the most economical method in spite of technological difficulties. Thus, most compressor manufacturers have been eagerly improving conventional compressors and developing new compressors. Generally, rolling piston type compressors have been replacing reciprocating compressors in the range of smaller size equipment, for the reason of high efficiency and low manufacturing cost. And in application to larger size equipment, screw type compressors have been growing for the same reasons. However, for the medium size equipment an adequate type of compressor replacing the reciprocating compressor has not been appeared.

Authors remarked a scroll compressor for this capacity range. The reason for the choice of a scroll compressor is the possibilities of a superior compressor owing to the inherent characteristics as follows;

(INHERENT CHARACTERISTICS)(ADVANTAGES AS A COMPRESSOR)	
Less variation of driving torque	
Less leakage of compression gas	
Less pressure drop at suction process	
No influence of reexpansion to suction process	
Rotative motion	
No valves	<p>High efficiency</p> <p>High capacity at high compression ratio</p> <p>Low noise, Low vibration</p> <p>High reliability</p> <p>Simple structure, Compact</p>

On the other hand, the rate of heat pump air conditioners has been increasing year by year as well known. By the latest investigation, the rate of heat pump reached 70% in Japan. Heat pump systems require high reliability of compressors because of wide range operation and large amount of refrigerant, etc. Therefore adaptable compressors to heat pump systems have been desired.

Under these requirements, authors have developed a scroll compressor as a high efficiency heat pump compressor.

In this paper, the features and performance, especially the features of sealing mechanisms of the scroll compressor are described.

STRUCTURE OF THE SCROLL COMPRESSOR

An overall structure of the scroll compressor is shown in Fig. 1. Compression elements, the fixed scroll and the orbitng scroll, are located upper side of the

welded shell, and the motor is located lower side. The fixed scroll is attached to the frame, and the orbiting scroll is driven as keeping the same angle by the Oldham coupling. These internal elements are tightly inserted in the shell without internal suspension. Gas enters the shell through the suction pipe, cools down the motor, goes up in the shell, then sucked into the compression chambers through the suction ports. Compressed gas is exhausted through the discharge port. Namely, the compressor is a low side shell compressor.

While, a high side shell was also investigated, but this study showed high side shell is not adaptable for our purpose. Because in high side shell, the motor is easily overheated under the condition of high compressi-

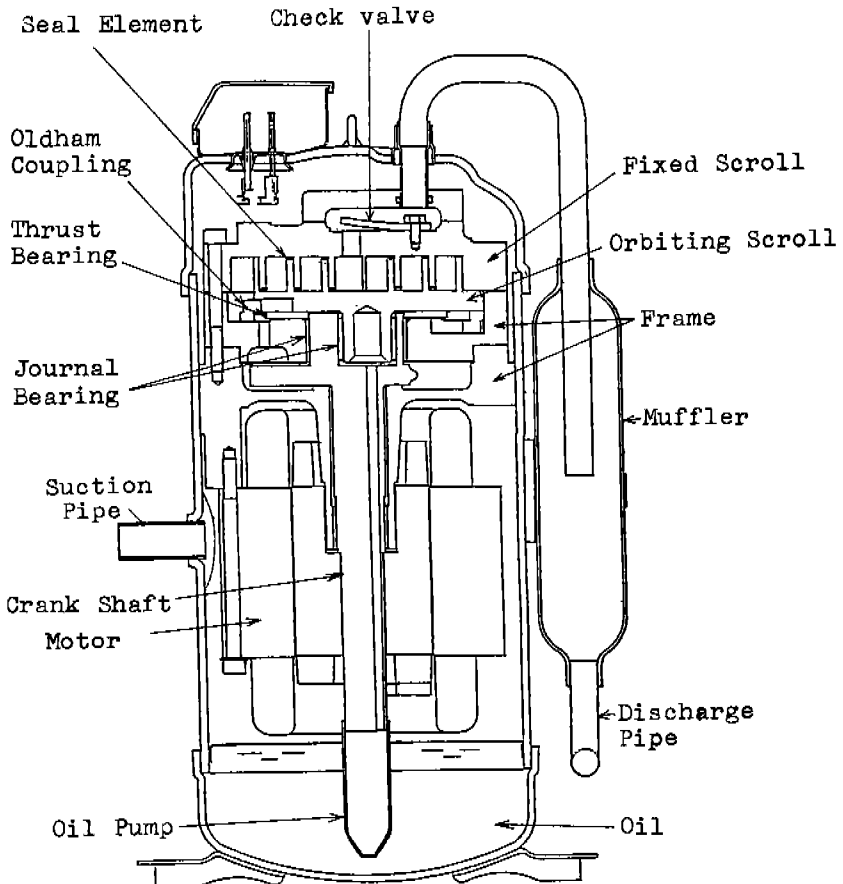


Fig.1: Structure of Low Side Shell Scroll Compressor

on ratio. So it requires another cooling method. Therefore authors adopted the low side shell.

For the purpose of attaining high efficiency, two kinds of sealing means are applied. The details of the sealing means will be described later.

A thrust force due to internal gas pressure is supported by the thrust bearing located below the orbiting scroll. A radial force is supported by the journal bearings, being located at the near longitudinal portion to eliminate the moment. Lubricating oil is supplied to each bearing through the crank shaft by centrifugal pumping action, and the used oil is guided to return to the bottom of the shell. The Oldham coupling has also a duty for preventing oil from being carried over into the system.

A check valve is attached to the fixed scroll to eliminate the noise in reversing.

To minimize vibration and noise of the system, a discharge muffler is attached to the discharge line.

Structural improvements compared with reciprocating compressors are as follows;

Number of parts	50%
Weight	80%
Volume	60%

Leakage of the scroll compressor

Minimizing leakage from the compression chamber is a significant subject in any type of compressors.

Compression chambers of the scroll compressor are the spaces configurationally formed between the fixed scroll wrap and the orbiting scroll wrap, so there occurs leakage through the fine clearances from the higher pressure chamber to the lower pressure chamber, as shown in Fig.2. There are two kinds of leakage, one is the radial leakage flows through the axial clearance, and another one is the tangential leakage flows through the radial clearance.

The mass flow rate of leakage from the compression chamber is calculated by the following equation, assuming the leak flow is one-dimensional flow of a compressible fluid.

$$\frac{dG}{dt} = C \cdot A \cdot \sqrt{2 \cdot g \cdot \frac{k}{k-1} \cdot P_2 \cdot \gamma_2 \cdot \left(F \frac{2}{k} - F \frac{k+1}{k} \right)}$$

$$P_1/P_2 \geq F_c \quad F = P_1/P_2$$

$$P_1/P_2 < F_c \quad F = F_c = (2/(k+1)) \frac{k}{k-1}$$

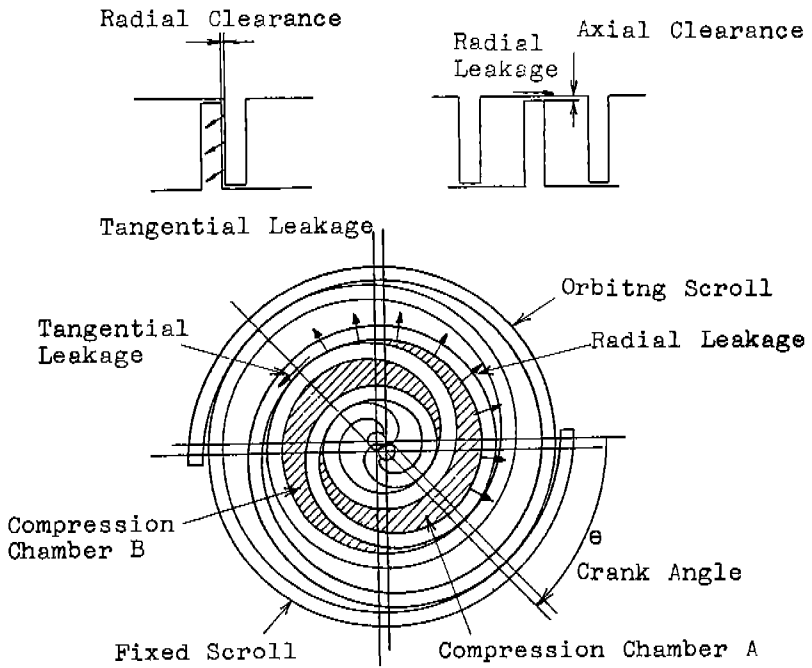


Fig.2: Leak Flow from Scroll Compression Chamber

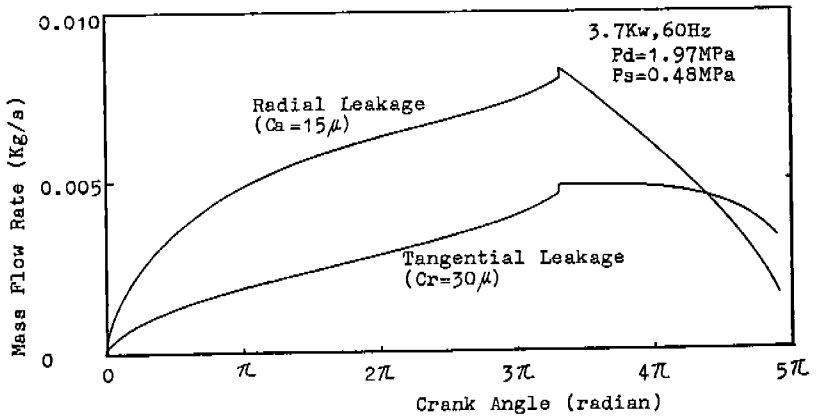


Fig.3: Mass Flow Rate of Leakage during a Stroke

Where, dG/dt is the mass flow rate of leakage, C is the coefficient of flow, A is the cross sectional area of the clearance, g is the acceleration of gravity, k is the adiabatic exponent, P_1 is the lower pressure, P_2 is the higher pressure, γ_c is the specific weight, and F_c is the critical compression ratio.

There are plural compression chambers at the same time, as shown in Fig.2. The instantaneous volumes of compression chambers are calculated. The calculation is introduced in the reference⁽¹⁾. And the instantaneous pressure is calculated by assuming polytropic process.

An example of calculation is shown in Fig.3, assuming that the axial clearance is 15%, and the radial clearance is 30%. The coefficient of flow C was obtained experimentally by measuring the flow of refrigerant-oil mixture through the clearance. As shown in Fig.3, the leakage is maximized at the crank angle when the compression chamber transferred to the innermost chamber. And the radial leakage is larger than the tangential leakage at the most of crank angle, in spite of smaller clearance. This is due to the longer sealing line of axial clearance.

AXIAL SEALING MECHANISM

A unique axial sealing mechanism is adopted, as shown in Fig.4. A seal element inserted tightly in the groove machined on the top of the wrap prevents radial leakage. Remaining the axial clearance between the top of the seal element and the opposite surface is important. Authors have developed the assembling techniques so as to insert the seal element sufficiently close to the bottom surface of the opposite scroll.

While, a seal element inserted loosely in the groove was also investigated. However in this type of axial sealing, the leak loss was relatively larger because of the leakage flowing through the side clearance of the seal element. The comparison of leak losses around the seal element is show in Fig.5. Leakage A is the radial leakage, leakage B is the tangential leakage flows through the side and back clearance, and leakage C is also the tangential leakage flows through the top of the wrap. In case of tightly inserted seal element, leakage A is larger, but leakage B is reduced, so total loss is less.

An example of an actual view of seal element is shown in Fig.

There formed small steps in the groove to prevent the seal element sinking downward by an axial force like during slugging.

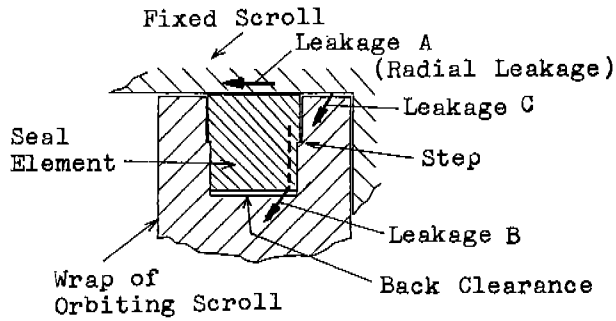
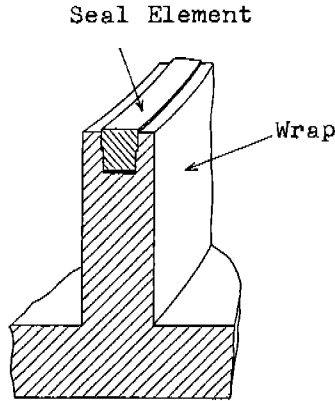


Fig.4: Axial Sealing Mechanism

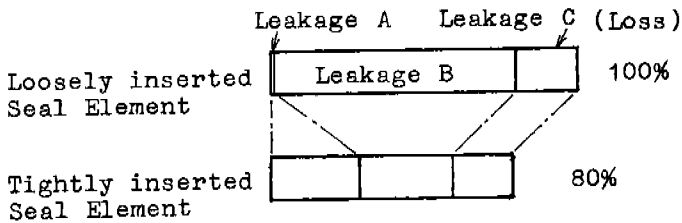


Fig.5: Comparison of Leak Loss
(Difference between Two Types of Seal Element)

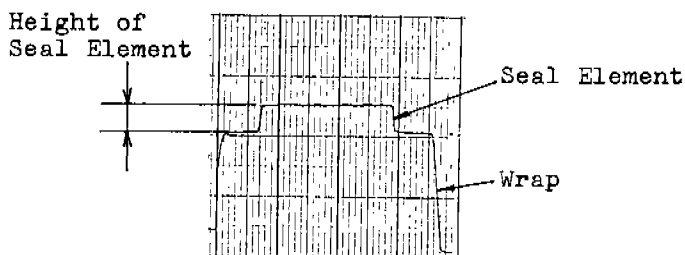


Fig.6: An Actual View of Seal Element
(Height of Seal Element is magnified)

RADIAL SEALING MECHANISM

As well known, radial sealing is achieved by radial contacting of both wraps, by means of swing link like as shown in U.S.P. (2)

Authors utilized a simple constant-radius crank shaft for a radial sealing mechanism. A simple constant-radius crank shaft is the most simplified method for driving, but it requires machining with high accuracy for the leveled radial sealing. Authors have accomplished high accuracy machining of scroll wraps, and furthermore the optimum crank radius was investigated by estimation of leakage loss and friction loss between the wraps.

A concept of radial sealing mechanism is shown in Fig.7. The force F is a resultant force from the force F_g and the force F_c . Where, the force F_g is a gas pressure load obtained by combination of internal gas pressure, and the force F_c is a centrifugal force of the orbiting scroll. The force F_b , a component of the force F , acts on the bearing. And the force F_s , another component, acts on the wrap of the fixed scroll as a radial sealing force. These forces are variable due to the value of crank radius, so the determination of crank radius is important.

A concept of the determination of the optimum crank radius is shown in Fig.8. In the range of relatively smaller crank radius, the tangential leakage increases and in the range of relatively larger crank radius, the leakage decreases but friction loss increases. Therefore the total losses owing to radial sealing gives the optimum crank radius. If the scroll wraps are machined in completely high accuracy, the loss will be zero as shown with a chain line in Fig.8. Namely, the crank radius that gives the loss zero is desirable. However in actual case, as machining deviation is not zero, radial

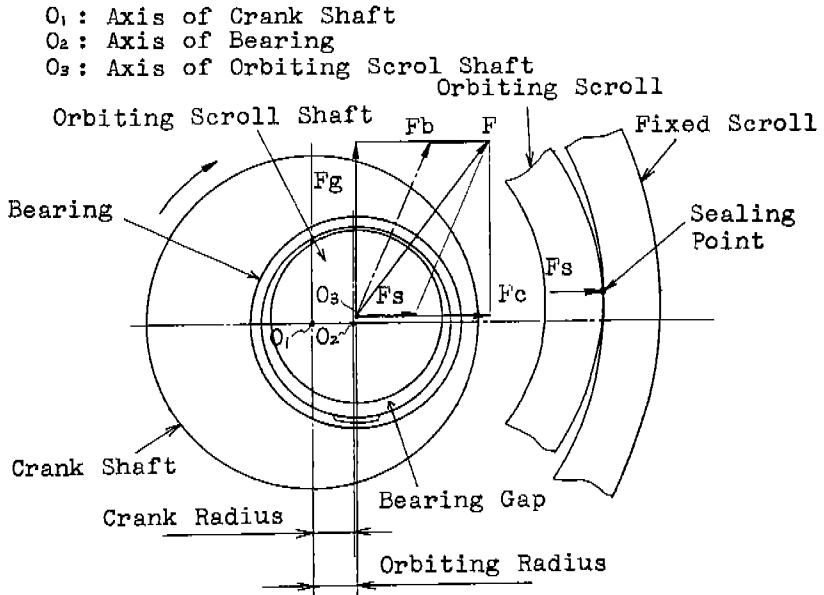


Fig.7: Radial Sealing Mechanism

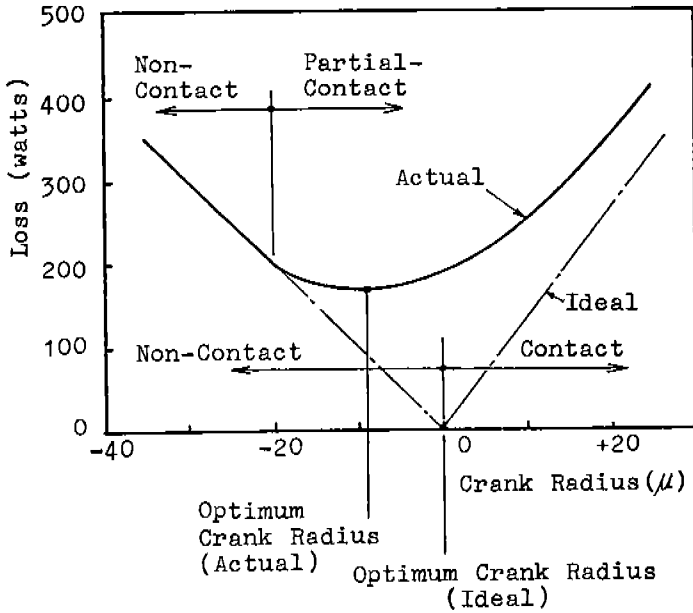


Fig.8: Optimum Crank Radius

clearance presents even under the optimum crank radius, as shown in Fig.9. The actual clearance is obtained by combination of the actual accuracy of both scroll wraps. Under this condition, the wraps are partially contacting with each other, resulting there occurs a little leakage and a little mechanical loss at the same time. These losses can be decreased by further improved machining and assembling.

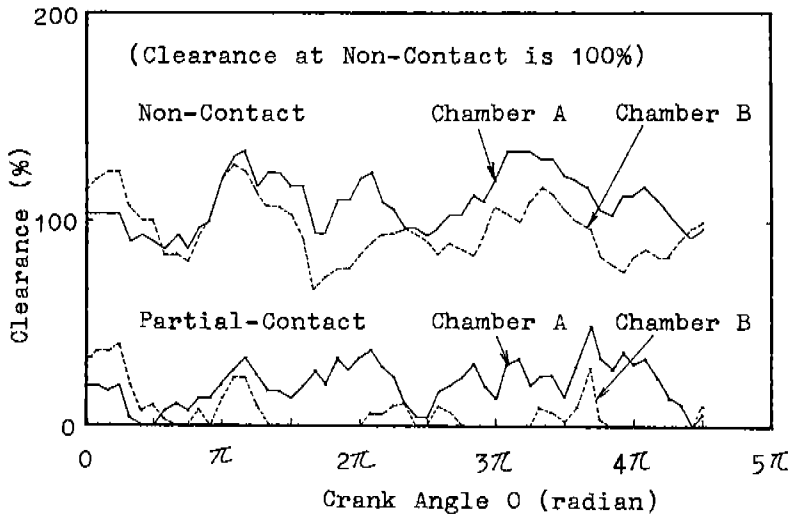


Fig.9: An Example of Actual Radial Clearance during one Stroke.

PERFORMANCE

At present, the scroll compressor has higher efficiency than the most advanced reciprocating compressor, and much higher capacity at high compression ratio.

Coefficient of performance

The comparison of power consumption between the scroll compressor and the reciprocating compressor is shown in Fig.10. The capacities at normal cooling condition are the same.

If no sealing means, power consumption of the scroll is larger than that of the reciprocating. Namely, COP of the scroll compressor is inferior to that of the reciprocating compressor. This is mainly due to the remarkable power consumption of leakage. If adopted sealing means, the performance of the scroll compressor reaches higher than the most advanced reciprocating compressor.

We have noticed there are some losses left in present scroll compressor, for example, tangential leakage and others. These losses will be reduced as described. We can estimate COP of the scroll compressor will be improved about 10% in future.

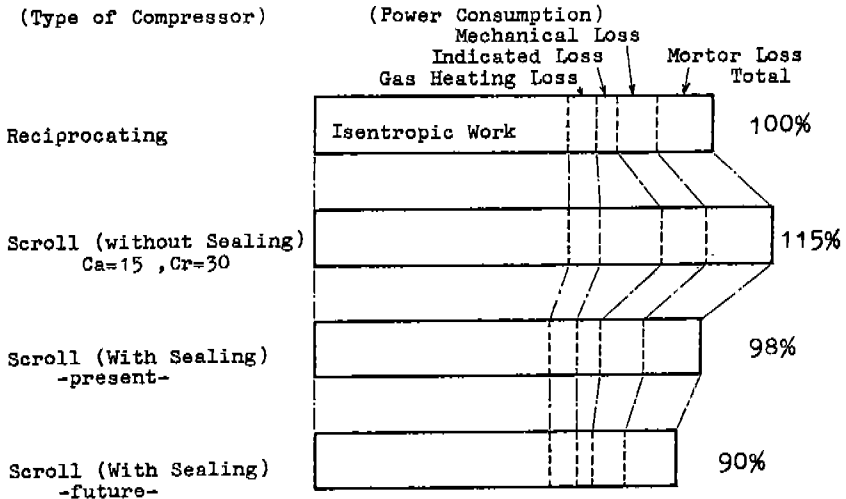


Fig.10: Comparison of Power Consumption

Capacity

The remarkable feature of capacity is the less reduction at high compression ratio, as shown in Fig.11.

Comparing the capacity of the scroll compressor and the reciprocating compressor which has the same capacity at compression ratio 3.5, the scroll compressor can provide about 30% higher capacity at the condition of compression ratio 8, in cooling and heating both. This is mainly due to the fact that there is no influence of re-expansion of discharge gas to suction process, in the scroll compressor. Therefore, the scroll compressor is more applicable to heat pump systems which are faced with wide compression range.

NOISE AND VIBRATION

Noise level of the scroll compressor depends upon the amount of radial contact between the wraps, and the accuracy of wrap machining.

The relation between the noise level and the crank radius is shown in Fig.12. Generally, noise increases

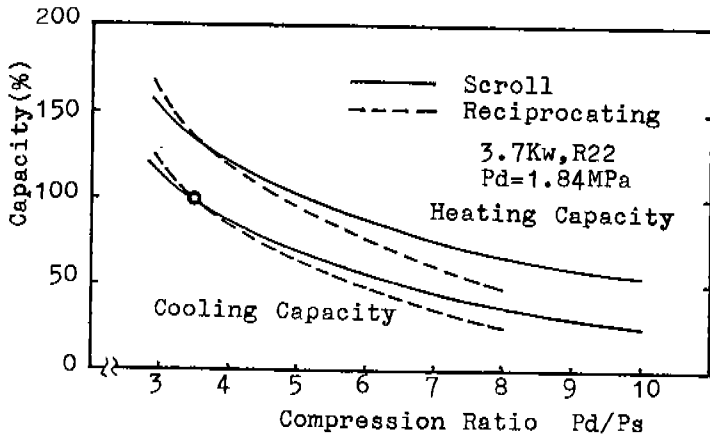


Fig.11: Comparison of Capacity

as the contact between the wraps occurs. In case of 3.7 Kw with the nominal crank radius, sound pressure level is about 67 dB-A.

The source of vibration of the scroll compressor is mainly the reciprocating motion of the Oldham coupling, in principle. In case of 3.7Kw scroll compressor, the maximum radial vibration is about 50 μ .

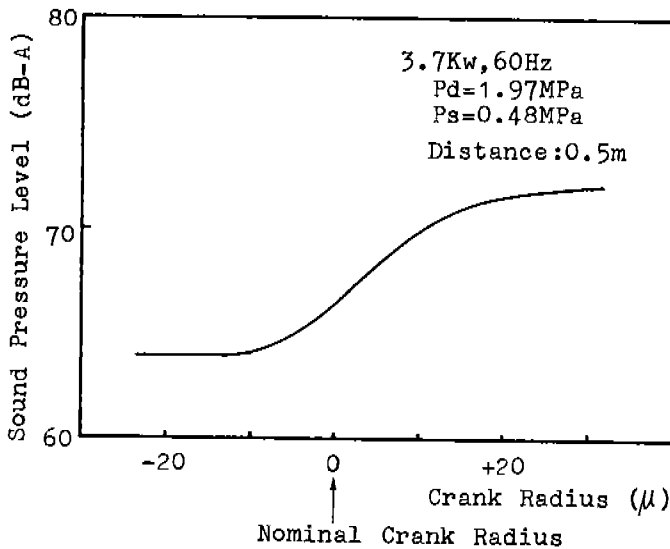


Fig.12: Noise vs. Crank Radius

RELIABILITY

Compressors installed in the heat pump equipment, happen to be faced with many difficult situations, as well known.

Authors also investigated the phenomena inherently occurring in the heat pump equipment, and came to know the compressors are needed to be tolerable to the various conditions, as follows. For example, the internal elements of the compressor are washed away by the condensed refrigerant, which occurs after cycling transferring of refrigerant caused by the variation of ambient temperature, especially in air to air heat pump equipment. In such a condition, the compressor is forced to start up without oil film on the bearings. And there happens a stagnancy of circulating oil in the equipment, especially in the accumulator. And a lot of refrigerant migrates into the compressor, diluting oil and raising the density of oil in a moment.

Authors have designed the scroll compressor to be durable in the above conditions, by choosing the adequate material of bearings, by designing the suitable lubricating path, and by keeping the oil carry-over low.

CONCLUSION

The features of the low side shell scroll compressor with sealing means we developed are as listed below;

- 1) The performance of the scroll compressor is higher than the most advanced reciprocating compressor, at present.
- 2) The cooling and heating capacities at high compression ratio are remarkably higher than those of the reciprocating compressor. (30% higher at compression ratio 8)
- 3) The scroll compressor is compact. (50% in number of parts, 80% in weight, 60% in volume, comparing with the reciprocating compressor)
- 4) The scroll compressor is durable in the various operating conditions of heat pump equipment.

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