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T. Morimoto

Matsushita Electric Industrial Co.

S. Yamamoto

Matsushita Electric Industrial Co.

S. Hase

Matsushita Electric Industrial Co.

S. Yamada

Matsushita Electric Industrial Co.

N. Ishii

Osaka Electro-Communication University

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Development of a High SEER Scroll Compressor

Takashi Morimoto¹, Shuichi Yamamoto², Shozo Hase¹
Sadayuki Yamada³, Noriaki Ishii⁴

¹Engineer, Air Conditioning Research Laboratory

²Senior Staff Engineer, Air Conditioning Research Laboratory

³Engineer, Compressor Division

Matsushita Electric Industrial Co., Ltd. (Panasonic),
Noji-cho, Kusatsu, Shiga 525, Japan

⁴Professor, Faculty of Engineering,

Osaka Electro-Communication University,
Neyagawa, Osaka 572, Japan

Abstract

In the present development, to improve the SEER of the compact horizontal-type scroll compressor, we took an overview of the operation frequency of the air conditioner throughout the year and identified existing problems under the commonest operational conditions. As a result, we found that to improve the SEER, the most effective approach was to improve efficiency under the operational conditions of low speed and low compression ratio. Based on this information as our starting point, we performed optimization of the volume ratio, reduction of overcompression loss using a bypass mechanism, and reduction of backflow during undercompression, this resulting in a considerable improvement in efficiency under the conditions of low speed and low compression ratio. At the same time, we improved the motor efficiency and, in the final specifications, realized an improvement in efficiency by 10–20 % under conditions of both low speed and low compression ratio.

1. Introduction

In the field of home-use room air conditioners, apart from conventional expectations of convenience and low noise, there is an increasingly urgent need for better energy efficiency. We therefore have taken an advantage of the low vibration and high efficiency of the scroll compressor and have adopted it in our home-use room air conditioners, resulting in the further improvement of their performance.

At present, from the viewpoint of both energy efficiency and convenience, variable speed operation of the compressor by inverter drive is common, and further improvement in efficiency over a wide operational range, from low speed to high speed, is an important factor in the performance of scroll compressors. It is particularly desired to improve the Seasonal Energy Efficiency Ratio (SEER) in accordance with the actual operational load. To realize this aim, it is important to take an overview of the actual load conditions in the operation of the air conditioner throughout the year and to improve the efficiency under frequent operational conditions. On the basis of this viewpoint, we have been developing the compact horizontal-type scroll compressor due to its high efficiency and low noise^{(1) (2) (3)} and have succeeded in further improvement of the SEER by introducing new technologies. In this paper, we report the newly developed element technologies realizing the higher efficiency and their performance.

2. Configuration and Specifications

The configuration of the newly developed compact horizontal-type scroll compressor is shown in Figure 1 and its specifications are shown in Table 1.

High pressure shell method

The compressor is 110 mm in outside diameter, 280 mm in length, and 10.6 kg in weight. The rated output is 2.5 kW and the operational speed range is 10 – 150 Hz. This compressor adopts a high pressure shell method whereby the refrigerant gas is directly inducted into the compression mechanism section through the induction tube and the compressed refrigerant gas is sent to the airtight vessel to be discharged through the outlet tube.

Compression mechanism

The compression mechanism comprises a fixed scroll and an orbiting scroll using an involute curve. The stroke volume is 13.4 cm^3 . The fixed scroll is made of cast iron and the discharge hole is equipped with a valve to prevent the backflow of the discharged refrigerant. The orbiting scroll is made of aluminum alloy and its tip is equipped with a tip seal. The driven axis is disposed in the center of the end plate and is carried by the variable crank mechanism. One edge of the main shaft on the side of the compression mechanism is carried by a sliding bearing and the other edge is carried by a rolling bearing.

Oil feed path

In this compressor, a positive-displacement pump is disposed on the end of the main shaft to pump up the oil inside the shell and to feed the oil to the compression mechanism through the path inside the main shaft, and the sliding parts of the orbiting bearing, the main bearing, and the orbiting scroll are constantly lubricated. In addition, the oil feed to the compression chamber is optimized in its rate by means of the throttle mechanism.

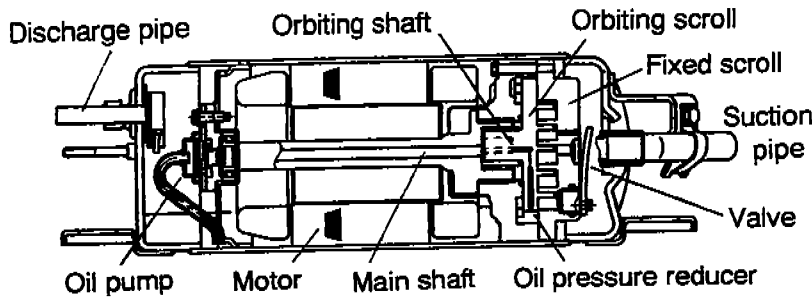


Fig.1 Configuration of horizontal-type scroll compressor

Table 1 Specifications of the compressor

Items	Specifications
Displacement	13.4cc/rev
Operating speed	10 - 150Hz
Volume ratio	2.23
Dimensions (mm)	$\phi 110 \times L280$
Mass	10.6kg
Refrigerant	R22

3. SEER and Operational Conditions for Air Conditioners

The SEER is the sum of the efficiency multiplied by the duration under various load conditions during operation. In the case of an air conditioner, it can be substantially regarded as its annual power consumption. In Japan, the industry has calculation standards for the annual power consumption of the air conditioners. In the case of the inverter-driven air conditioner, the compressor is operated at various frequencies, and the pressure conditions vary accordingly. Therefore, not only the efficiency at the rated point as before, but also the efficiency under longer duration conditions need to be improved.

A case of operation of an air conditioner with a scroll compressor is explained here. Figure 2 shows the operation time and the variation in performance against the outer temperature under typical Japanese meteorological conditions. During both heating operation and cooling operation, the operation time reaches a peak near the half-performance point relative to the rated value. Figure 3 shows the vaporization pressure, the condensation pressure, and the compression ratio against the operation frequency during cooling operation. It represents that the half-performance point with the longer duration has a low operation frequency and a relatively low compression ratio. Therefore, an effective strategy to improve the SEER is to focus on improving the efficiency under low-speed and low-compression-ratio conditions.

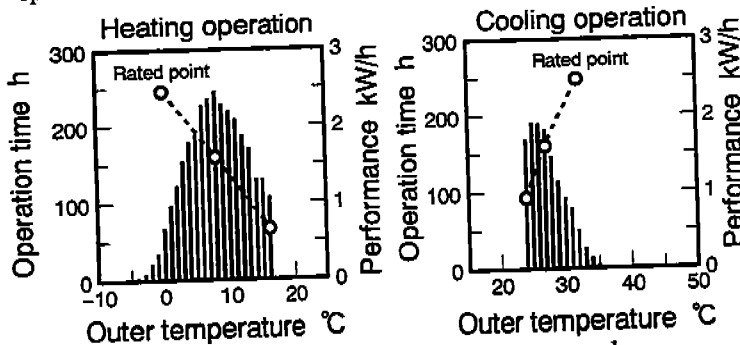


Fig.2 Operation time and performance change against outer temperature

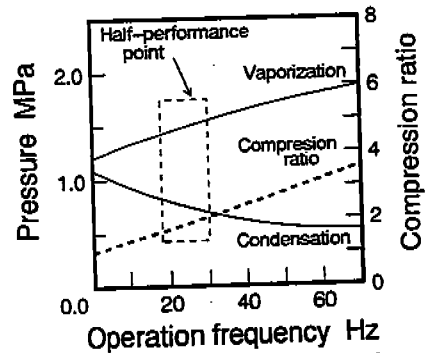


Fig.3 Pressure and compression ratio vs. operation frequency

4. Analysis of the Conventional Scroll Compressor

As an example of a conventional model of the scroll compressor, the P-V diagrams and the results of analysis are shown in Figure 4 for the rated point in heating and the half-performance point in cooling. The built-in volume ratio of the conventional compressor is 2.23. Figure 4 shows that, at the rated point, it is operated under conditions of undercompression and that, regarding the pressure loss, the delay in closing the discharge valve causes backflow, resulting in a large loss. At the half-performance point, it is operated under conditions of overcompression and the overcompression causes a large loss. Therefore, to improve the SEER under low-speed and low-compression-ratio conditions, reduction of the overcompression loss is an important factor.

The following evaluation experiments in this study are made under the two conditions of the rated point and the half-performance point relative to the rated value.

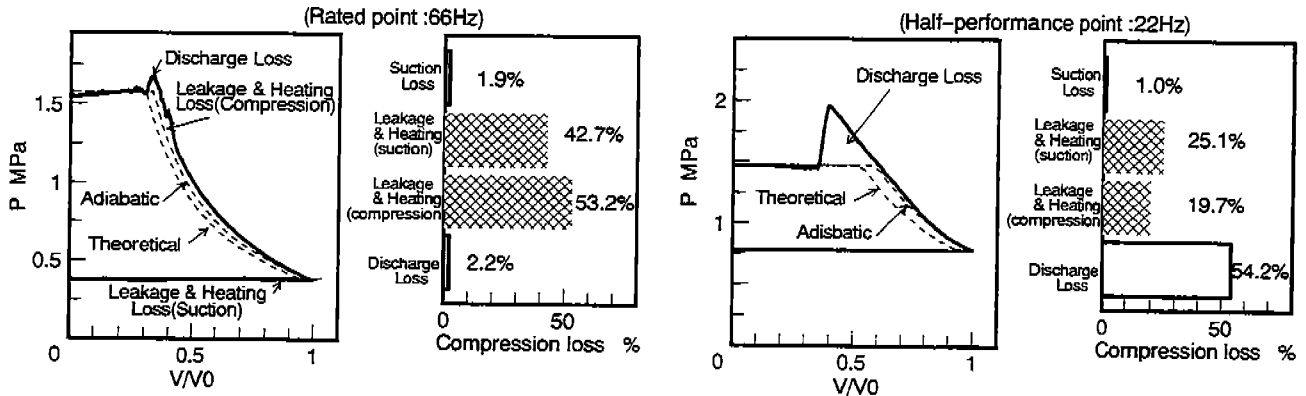


Fig.4 P-V Diagram and results of analysis of the conventional model

5. Major Newly Developed Technologies

5-1 Reduction of the overcompression loss

It was found that reduction of the overcompression loss is useful in improving the SEER. There are two methods for reducing the overcompression loss: optimization of the built-in volume ratio and avoidance of overcompression using the bypass mechanism in the compression chamber. In this development, we studied both methods.

(1) Optimization of the volume ratio

Figure 5 shows the operation compression ratio against the operation frequency in an air conditioner equipped with a scroll compressor. The operation compression ratio near the half-performance point with the longer duration in the air conditioner varies around 1.8 ~ 2.4 between cooling and heating operations. It shows that, to reduce the overcompression loss, selecting the volume ratio is necessary when the compression ratio is in this range. The optimal volume ratio is 1.95, lower than the conventional value of 2.23.

By lowering the volume ratio the overcompression loss is reduced and the efficiency is improved by 12 %. However, at the rated point, where it is under conditions of undercompression, the increase in backflow loss causes a 2 % decrease in efficiency.

Figure 6 shows the (theoretical) rates of the overcompression loss and the backflow loss to the theoretical power against the compression ratio at the volume ratios of 2.23 and 1.95. As obviously shown in Figure 6, if the volume ratio is further reduced, further improvement in efficiency at the half-performance point under conditions of undercompression is expected, but the efficiency at the rated point under conditions of overcompression is inevitably lowered. Since the performance at the rated point is also important for air conditioner compressors, there is a limit to the possible reduction of overcompression loss by optimization of the volume ratio.

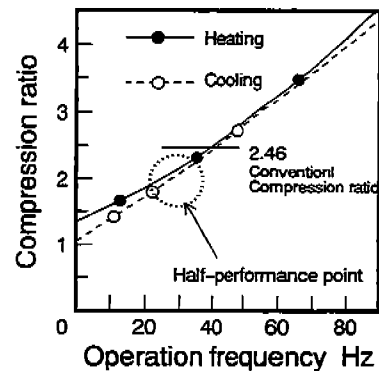


Fig.5 Operation compression ratio

Thus, to improve the efficiency over a wide range, the optimization of the volume ratio alone is not sufficient, and other devices, such as a bypass mechanism, are necessary.

(2) Study of the bypass mechanism

The bypass mechanism comprises a hole connecting to the compression chamber, in the fixed scroll, and a valve preventing backflow from the bypass hole during undercompression. The position of the bypass hole is shown in Figure 7. The bypass hole is designed to connect approximately 160 degrees from the start of compression.

Working range of the bypass

The bypass hole needs to be designed to avoid overcompression over a wide range of conditions taking account of the operation conditions in the air conditioner. Figure 8 shows the working range of the bypass hole. As shown in Figure 8, the bypass hole is located approximately 160 degrees from the start of compression, which allows it to be effective over a wide range, from the minimum-performance point to the half-performance point in the air conditioner. Although the difference in the working range of the bypass hole has an influence on the bypass effect, no difference in the bypass effect was observed at the half-performance point when the bypass was designed to be operable from the minimum-performance point.

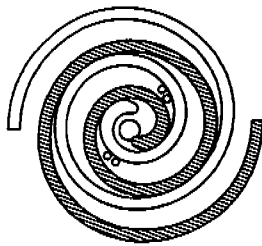


Fig.7 Position of the bypass hole

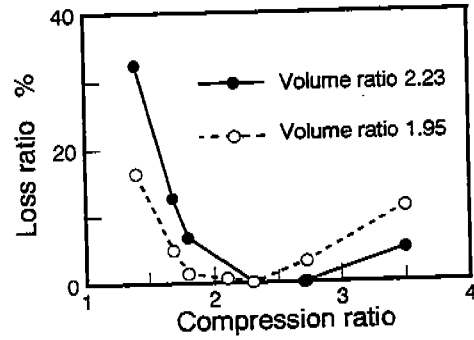


Fig.6 Rates of overcompression loss and backflow loss

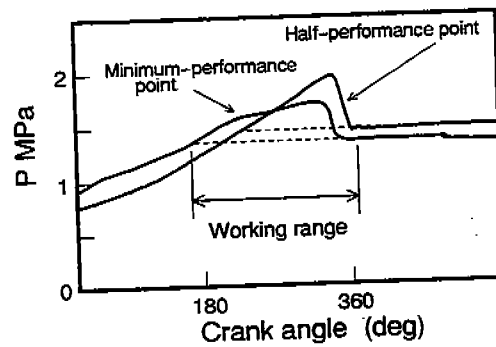


Fig.8 Working range of the bypass hole

Influence of bypass hole diameter

It would initially appear that the bypass hole should be as large as possible to reduce discharge resistance. However, since this has an adverse effect during undercompression because the bypass hole acts as dead volume in the compression stroke, the bypass hole would also seem to need to be as small as possible. The best compromise between these contradictory conditions is that the bypass hole cannot be greater than the thickness of the scroll vane.

Here, the influence of the bypass hole diameter is studied at the volume ratio of 1.95 previously determined during the optimization of the volume ratio. Figure 9 shows the performance simulation and the experimental results with the bypass hole at the half-performance point.

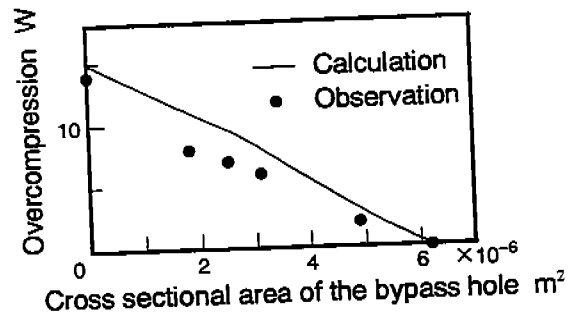


Fig.9 Overcompression loss vs. cross sectional area of the bypass hole

This shows the overcompression loss against the cross sectional area of the bypass hole. In calculations, overcompression is reduced as the cross sectional area of the bypass hole is increased, and overcompression is avoided at a certain cross sectional area. In addition, as for the maximum cross sectional area in the case of a single round bypass hole, overcompression is avoided only by some 50% and thus more than one bypass hole is necessary. Based on this calculated result, experiments were performed by varying the cross sectional area of the bypass hole. Although the observed figures are slightly smaller than the

calculated figures, this disagreement is probably because overcompression is reduced in the actual compressor due to leakage during the compression stroke. As for the bypass hole diameter necessary to completely avoid overcompression, the observed value is in good accordance with the calculated value. Therefore, in the studied model, to assure a path area relative to the width of the tip seal, overcompression was avoided by making 2 adjacent bypass holes of f 2.0.

Influence of the volume ratio when the bypass hole is made

Figure 10 shows the performance coefficient against the volume ratio when two bypass holes of f 2.0 are made. In the case of overcompression, the overcompression loss increases as the volume ratio increases, but, in the case of the bypass holes, no difference was observed in performance. On the other hand, in the case of undercompression, as in the case of no bypass hole with varied volume ratio, the performance is enhanced as the volume ratio increases because the backflow loss decreases. As a result, the volume ratio has little influence on the reduction of the overcompression loss when the bypass mechanism is used and thus the volume ratio can be set relatively arbitrarily.

Performance when the bypass is made

The developed model is configured with two bypass holes of f 2.0 per compression chamber. Figure 11 shows the performance coefficient against the cross section of the bypass hole. The vertical axis represents the performance coefficient ratio and one DIV. is 10 %. At the half-performance point, the performance coefficient is improved by up to 16 % by increasing the cross section of the bypass hole. At the rated point, it is improved by a maximum of 2 %. Thus it is found that the bypass mechanism has a significant effect on efficiency improvement under conditions of low speed and low compression ratio and a minor effect under conditions of high compression ratio. Thanks to this advantage, the bypass mechanism is suitable for the improvement of the SEER.

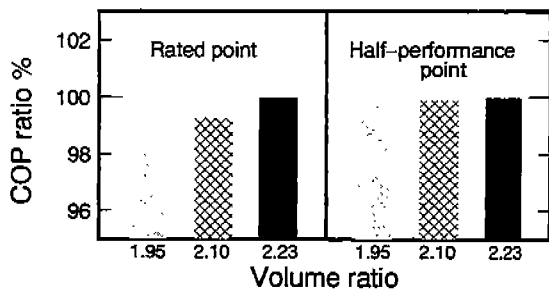


Fig.10 Performance Coefficient vs. volume ratio

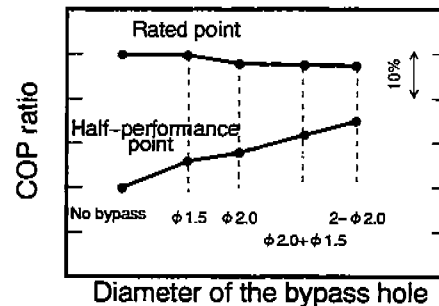


Fig.11 Performance coefficient when the bypass hole is made

5-2 Reduction of backflow loss

Reduction of backflow loss is useful in improving the efficiency during undercompression. Here, by focusing on the detachment velocity of the both scroll vanes after the completion of the contact and the top clearance of the discharge path, the backflow loss can be reduced.

Influence of detachment velocity

Although the backflow of the compressed gas in the top clearance is decreased by reducing the detachment velocity of the vane, the overreduced detachment velocity has a deleterious effect. If the detachment velocity is too low, the backflow loss is reduced but the compression prior to the opening of the discharge valve causes an increase in the discharge loss. Thus, unfavorable phenomena are observed if it is designed based only on the detachment velocity of the vane.

Shape of the vane of the developed model

Figure 12 shows the compared shapes of the vanes near the starting point of the developed model and the conventional model. The developed model, compared with the conventional model, has a thicker shape near the starting point of the vane, resulting in a lower vane detachment velocity and a smaller top clearance volume. The diameter and the position of the discharge hole were optimized based on the detachment velocity. By thickening the starting part

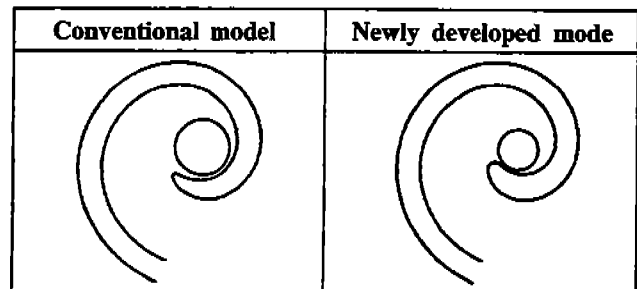


Fig.12 Shape of the vane

more than in conventional models, the volume ratio is reduced and the detachment velocity of the vane is 30 – 50% lower than in conventional model.

Backflow reduction effect

By reducing the top clearance volume and the detachment velocity of the vane, backflow loss is greatly reduced and the efficiency is improved by 3–5 %.

5–3 Study of the backflow valve

Although the backflow valve plays an important role in preventing backflow during undercompression, it increases noise generation owing to the increase of the discharge resistance and the impact of the valve. Figure 13 shows the values of input and noise against valve rigidity. Valve rigidity is closely related to both performance and noise. Lower rigidity results in better performance but higher noise, and higher rigidity causes the opposite.

In this development, by using some devices together with the above-mentioned reduction of the backflow loss, we have succeeded in simultaneously reducing valve rigidity and noise. In the newly developed model, by reducing the valve rigidity to two thirds of that of the conventional model and by optimizing the shape of the discharge hole, we have achieved a reduction of input of 1 % with a lower noise level than the conventional model.

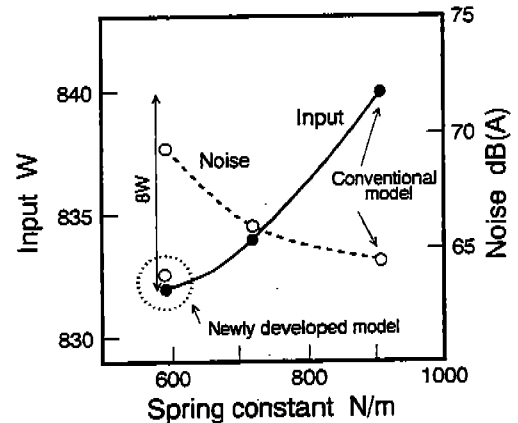


Fig.13 Values of input and noise against valve rigidity

6. Performance Coefficient and SEER of the Newly Developed Model

In addition to the above-mentioned improvement, we have performed the optimization of each gap during operation and the improvement of the motor. In the specifications which include all the improvements, the performance coefficient is reduced by 2 % at the rated point and by 20 % at the half-performance point, and the efficiency is greatly improved under the conditions of low speed and low compression ratio, with a significant contribution to the improvement of the SEER. In an air conditioner featuring the newly developed model, the annual power consumption is reduced by some 10 % compared with the conventional model in addition to an improved SEER.

7. Conclusion

To improve the SEER, the design of a compact horizontal-type scroll compressor was improved. For better SEER, increased efficiency at low speeds and low loads was found to be necessary and, in the scroll compressor, the reduction of the overcompression loss is a useful measure. The bypass mechanism is very useful in reducing the overcompression loss. However, since it causes the reduction of efficiency during undercompression, measures during undercompression, such as reduction of backflow, need to be used together to enhance efficiency. In this study, by combining such major technologies, we have succeeded in developing a high SEER compact horizontal-type scroll compressor.

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