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NUMERICAL AND EXPERIMENTAL INVESTIGATION OF SWING COMpressor CHARACTERISTICS

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ABSTRACT

This paper discusses the results of the numerical analysis and experiment on swing compressors, scroll compressors and rotary compressors. The performance of these compressors were numerically analyzed between the cooling capacity of 2.3kW and 31kW with R22 and R410A. In addition, we tested newly designed 2-cylinder swing compressors of 15.5kW with R22 and R410A, and evaluated from the viewpoint of efficiency, noise and vibration.

The investigation results show that from the viewpoint of efficiency, the swing type with R22 should be used in the capacity range of 7.7kW or less and that with R410A in the capacity range of 10.7kW or less. The scroll type with axial-radial compliance mechanism should be used above these ranges. In addition, the analysis results show that the swing type has an advantage over the rotary type from the viewpoint of efficiency in the capacity range of 15.5kW or less.

NOMENCLATURE

H: cylinder height  D: cylinder diameter
Tc: condensing temperature  Te: evaporating temperature
Sc: sub cool  Sh: super heat
CP: Radial clearance between the piston and cylinder wall
CR: Axial clearance between the piston and front-head, rear head and middle-plate

INTRODUCTION

Daikin industries Ltd. has developed swing compressors for residential air conditioners and has been manufacturing this type since 1994. Swing compressors are mechanically similar to rotary compressors. However, swing compressors are different from rotary compressors in one point. That is, the vane and the roller are made of one body. Therefore, there is no leakage and no friction loss between the vane tip and the roller. Consequently, its efficiency is higher than that of the rotary compressors.\(^{(1)}\)

In addition, since the HFC refrigerants do not contain chlorine in their molecular structure, if they are used for rotary compressors, the lubricating ability between the vane tip and the roller deteriorates. On the other hand, since the vane and the roller are made of one body, it is not necessary to consider the deterioration of lubricating ability. Therefore, a capillary tube does not clog with deteriorated oil contamination. As a result, the reliability of swing compressors is higher than that of rotary compressors.\(^{(2)}\)

In spite of those advantages of swing compressors mentioned above, until now those of which capacities are
larger than 6.2kW have not been applied to air conditioners. The purpose of this investigation is to clarify the upper capacity limit of swing compressors and broaden their application range.

**NUMERICAL AND EXPERIMENTAL RESULT**

**EFFICIENCY**

This paper deals with the numerically analyzed efficiencies of swing type, rotary type and scroll type compressors of cooling capacity between 2.3kW and 31kW with R22 and R410A. In addition, it deals with the test results of newly designed swing compressors of 15.5kW with R22 and R410A.

**ANALYSIS MODEL**

Fig. 1 shows the scheme of compression mechanism of each compressor.

**SWING COMPRESSOR:**

The method for numerical analysis of mechanical bearing loss is based on the equation of journal bearing of finite width. The diameter of discharge port and the dimension of the cylinder were optimized according to each capacity with R22 and R410A.

**SCROLL COMPRESSOR:**

The scroll compressors used for analysis have a axial-radial compliance mechanism. A pressure chamber which is pressurized by discharge gas is constructed on the backside of orbiting scroll with seal ring. Optimization of the force on the scroll backside is the key of compliance mechanism. Therefore, we optimized the chamber size for optimization of the force applied to the backside, so that the thrust bearing loss may be minimized without tipping the orbiting scroll under the low compression ratio.

**STRUCTURE OF 2-CYLINDER SWING TYPE COMPRESSOR USED FOR TEST**

**CROSS SECTION OF NEWLY DESIGNED COMPRESSOR:**

Fig. 2 shows the cross section of the compressor used for test. The casing size is designed to be equivalent to that of the scroll compressor currently in production. A 3-phase induction motor is mounted. Two compressors were prepared, one for R410A and the other for R22. The lubricants for R410A is ether VG68 and for R22 is mineral VG56.

**IMPROVEMENT OF PERFORMANCE:**

1. **OPTIMIZATION OF H/D RATIO**

   Parameter of numerical analysis is the ratio of H/D. Since the compressor size is equivalent to that of the scroll, the minimum obtainable ratio of H/D is 0.24. According to the analysis results shown in Fig.3, the volumetric and indicated efficiency increase as the ratio of H/D decreases. This is due to decrease of leakage from CP clearance and heat loss. Therefore, the total efficiency of swing compressor can be improved by decreasing the ratio of H/D.

2. **MINIMIZATION OF CLEARANCE**

   Parameter of numerical analysis is the clearances CP and CR. The clearances CP and CR can be decreased by improving precision of machining parts. According to the analysis results shown in Fig.4, the volumetric efficiency increases as the clearances CP and CR decrease. This is due to the decrease of leakage from CP and CR. Therefore, the total efficiency of swing compressors can be improved by upgrading the machining precision.
NUMERICAL AND EXPERIMENTAL RESULT

1. COMPARISON OF PERFORMANCE BETWEEN SWING TYPE AND ROTARY TYPE

Fig.5 shows the numerical results of 15.5kW compressor performances of the swing type and rotary type. Fig.5 shows that the capacity of the swing type is slightly higher than that of the rotary type, because the swing type has no leakage between the vane tip and the roller like the rotary type. However, the input of the swing type is smaller than that of the rotary type. As a result, the COP of the swing type is higher than that of the rotary type.

Fig.6 shows the comparison of mechanical loss between the swing type and the rotary type. The losses of the crank pin bearing and the bush of the swing type is larger than that of the rotary type. However, the loss between the vane and the roller disappears. As a result, the mechanical loss of the swing type is approximately 0.9 times that of the rotary type.

2. COMPARISON OF PERFORMANCE BETWEEN SWING TYPE AND SCROLL TYPE

Fig.7 shows the relation of efficiency and capacity of the swing type. As the capacity increases, though the volumetric efficiency increases, the indicated and mechanical efficiency decrease. The decrease rate of the indicated and mechanical efficiency is larger than the increasing rate of the volumetric efficiency. Fig.8 explains the reason. In addition, the decreasing rate of the indicated efficiency using R22 is larger than that using R410A. Fig.8 shows the relation of indicated loss and the capacity of the swing type. As the capacity increases, the loss in compression process and the over compression loss increase. Here "loss in compression process" means leakage, heat and re-expansion loss. The former is due to the decrease of the leakage from CP per suction volume decreases and the latter is due to the delay of the reed valve opening, which is caused by excessive reed valve rigidity. However, this rigidity is required to properly close the valve at the compression start angle and the reed valve is also required more rigid as the capacity increases.

Fig.9 shows the relation of efficiency and capacity of the scroll type. On the other hand, in case of the scroll type, the volumetric, indicated and mechanical efficiency increase as the capacity increases. In addition, the drop of the volumetric efficiency using R410A is larger than that using R22 in the small capacity range.

Fig.10 shows the numerical and experimental results of the relation of total efficiency and capacity of each compressor using R22. As the capacity increases, the total efficiency of the swing type increase in the capacity range of 4.5kW or less. However, in the capacity range of 4.5kW or over, the total efficiency of the swing type decreases as the capacity increases. On the other hand, the total efficiency of the scroll type increases as the capacity increases. As a result, the swing type using R22 has higher efficiency than the scroll type with axial-radial compliance mechanism in the capacity range of 7.7kW or less.

Fig.11 shows the numerical and experimental results of the relation of total efficiency and capacity of each compressor using R410A. As the capacity increases, the total efficiency of the swing type increases in the capacity range of 6kW or less. However, in the capacity range of 6kW or over, the total efficiency of the swing type decreases as the capacity increases. The drop of the total efficiency of swing type using R410A is smaller than that using R22. On the other hand, the drop of the total efficiency of the scroll type using R410A is larger than that using R22 in the small capacity range. As a result, the swing type using R410A has higher efficiency than the scroll type with axial-radial compliance mechanism in the capacity range of 10.7kW or less.
Fig. 12 shows the experimental results of vibration value of the scroll, rotary and swing compressors. The amplitude of vibration of the swing type is considered to be equivalent to that of the rotary type due to their similarity in compression mechanism. The refrigerant used for the scroll and rotary types is R22 and that used for the newly designed swing type is R410A. According to the test results, the amplitude of vibration of the 2-cylinder swing type is slightly smaller than those of other types though the refrigerant is R410A. This is due to the large inertia moment of the compressor casing. The wall thickness of the swing type casing is greater, because the design pressure for R410A is higher than that of R22. In conclusion, the difference of the amplitude of vibration between the 2-cylinder swing type and the scroll type is small.

[NOISE]

Fig. 13 shows the experimental results of overall sound pressure level of the rotary type, the swing type and the scroll type with and without compliance mechanism. It is said that the sound pressure level of scroll compressors is lower than that of other type of compressors. However, the test results show that the sound pressure level of scroll type with compliance mechanism is larger than that of the scroll type without it. In conclusion, the sound pressure level of scroll type with compliance mechanism does not show any advantage over those of 2-cylinder swing and rotary types.

CONCLUSION

The conclusions based on the evaluation of numerical analysis and experiment results carried out from the viewpoint of efficiency, noise and vibration are follows:

1) The total efficiency of swing type is higher than that of rotary type in the capacity range of 15.5kW or less.
2) The total efficiency of swing type with R22 drops in the capacity range of 4.5kW or over. This is due to the remarkable drop of indicated and mechanical efficiency when the capacity increases.
3) The total efficiency of swing type with R410A drops in the capacity range of 6 kW or over.
4) The swing type with R22 should be used in the capacity range of 7.7kW or less and that with R410A in the capacity range of 10.7kW or less. The scroll type with axial-radial compliance mechanism should be used above these ranges.

REFERENCES

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rotary compressor

swing compressor

scroll compressor

Fig. 1 Scheme of compression mechanism of each compressor

Fig. 2 Cross section of newly designed compressor

Fig. 3 Influence of H/D on efficiency

Fig. 4 Influence of the clearance CP and CR on efficiency

Reference value of clearance CP and CR of rotary type currently in production is regarded as 1.0 (calculated)

Fig. 5 Comparison of the performance between rotary compressor and swing compressor

Reference values of input, capacity and COP of rotary type are regarded as 1.0 (calculated)

Fig. 6 Comparison of the mechanical loss between rotary compressor and swing compressor

Reference value of mechanical loss of rotary is regarded as 100 (calculated)

Fig. 7 Comparison of volumetric, indicated and mechanical efficiency of swing compressor

Reference value of volumetric efficiency of 2.3kW swing type with R22 is regarded as 1.0 (calculated)
Fig. 8 Ratio of indicated loss against input of swing compressor

Fig. 9 Comparison of volumetric, indicated and mechanical efficiency of scroll compressor
Reference value of volumetric efficiency of 2.3kW scroll type with R22 is regarded as 1.0 (calculated)

Fig. 10 Comparison of total efficiency with R22
Reference value of total efficiency of 15.5kW scroll type without compliance mechanism currently in production with R22 is regarded as 1.0 (measured and calculated)

Fig. 11 Comparison of total efficiency with R410A
Reference value of total efficiency of 15.5kW scroll type without compliance mechanism currently in production with R22 is regarded as 1.0 (measured and calculated)

Fig. 13 Comparison of vibration value of compressor (R22 is used except for the newly designed compressor) (measured)

Fig. 14 Comparison of the sound pressure level of compressor (R22 is used except for the newly designed compressor) (measured)