Development of Large Capacity And High Efficiency Rotary Compressor

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ABSTRACT

The authors had studied some elemental technologies to realize the large capacity and high efficiency of rotary compressor in order to achieve energy conservation, resource savings, and weight reduction of heat pump systems. Those elemental technologies are the cylinder dimension optimization, the new discharge structure, the divided vanes, and the compatibility between sliding loss reduction and securing reliability in journal bearings. The new discharge structure is a structure in which two discharge ports are installed per cylinder. The discharge ports at bearing and partition plate were designed with different diameter port to get a high reliability of discharge valve. By applying the new discharge structure, discharge losses drastically reduced during compression process. In order to achieve a high reliability of sliding parts, the divided DLC vanes structure and an optimization of journal bearing within enhanced shaft stiffness were applied. Hence, by adopting high efficiency 6-pole concentrated winding motor driving with PWM converter, an electric harmonic wave can be controlled to get a small loss current.

By above technologies, we succeeded to develop a new compressor that can increase to the maximum refrigeration capacity 20 HP [60kW]. Compared to our conventional large model, the new model has 1.7 times larger maximum capacity, and about 6 % higher COP ratio in most operating condition range. Moreover, by the more compact and lighter weight of 38.2 kg, the new one has contributed to resource saving and weight reduction of the heat pump unit system. 4 developed compressors which are installed in air-cooled modular heat pump unit “Universal Smart X Edge Series” were able to expand up to 70HP in rated cooling capacity and achieved “IPLV 6.0” at 60 HP heat pump unit systems.

1. INTRODUCTION

In recent years, facing on the environmental problem like global warming, natural resource depletion becomes an important issue to build more compact and eco-friendly air conditioner and heat pump. Moreover, considering an easy handling and installation of such systems, it is effective way to reduce system weight by reducing the number of compressor per system by increasing its capacity. According to increasing capacity, additional refrigerant flow loss at cross-sectional area of discharge port will decline compressor’s performance, meanwhile, sliding part reliability get worse during compressor’s load increase. On the other hand, enlarging discharge port diameter will deteriorate discharge valve reliability. In this study, we introduce some technologies to maintain both of compressor’s efficiency and reliability.

2. OPTIMIZATION OF CYLINDER DIMENSIONS

2.1 Cylinder Diameter and Crankshaft Eccentricity

2-cylinder rotary compressor and compression parts cross-section views are shown in fig.1 and fig.2. Suction volume Vs of rotary compressor is defined as correlation of cylinder height H_{cyl}, cylinder inner radius R_{cy} and roller radius R_o. Hence, cross-section suction area A is result of correlation of crankshaft eccentricity E and cylinder inner radius R_{cy} as given in equation (1), (2) below:

$$Vs = \pi(R_{cy}^2 - R_o^2)\, H_{cyl} \quad ; \text{ where } R_o = R_{cy} - E \tag{1}$$

$$A = \pi R_{cy}^2 - \pi(R_{cy} - E)^2 \tag{2}$$

Then, cylinder radius R_{cy} relate to a distance of vane to inner case and inner radius of case as given as follows:

$$R_{cy} = R - B - L_v \tag{3}$$

Where B is distance of backside vane to inner case, Lv is vane length. Refer to the figure 2, in our investigation, the optimum suction volume in equation (1) respectively will be achieved if distance B and vane length Lv with connected to cylinder radius Rcy and eccentricity amount E satisfy a condition as given in (4). The condition will ensure the cylinder allowable stiffness, vane sliding reliability and easy handling in manufacturing side.

$$B \geqq 0.15R \quad \text{and} \quad E \geqq 0.225 L_v \tag{4}$$

In addition, as a pressure vessel, it is desirable not to increase the compressor case diameter and case thickness as much as possible from the viewpoint of pressure resistance, weight reduction and natural resource saving. Therefore, in this development model, we decided to optimize the cylinder inner diameter and crankshaft eccentricity, and also to extend the cylinder height inside of case with inner diameter less than 160 mm.

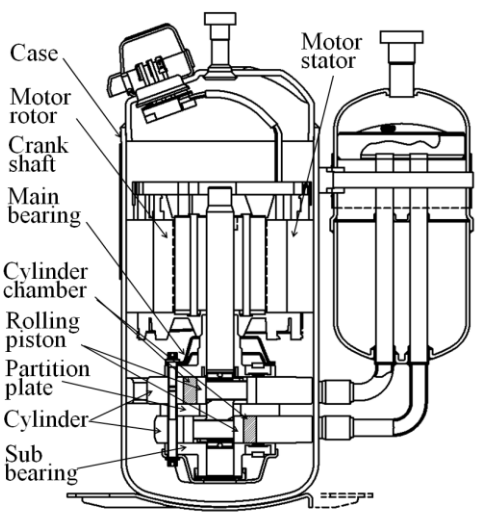


Figure 1: 2-Cylinder rotary compressor

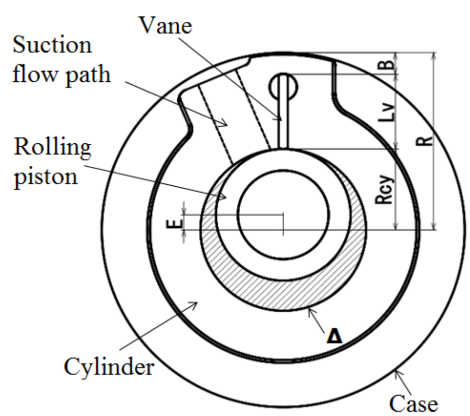
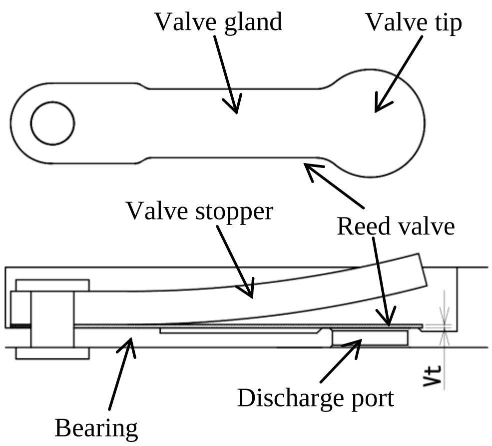


Figure 2: Compression cross-section view

2.2 Cylinder Height with Conventional Discharge Structure

Increasing capacity by extending the cylinder height makes possible to magnify the cross-sectional flow area in order to reduce the suction pressure loss, meanwhile, it becomes a bottleneck in discharge valve. As shown in figure 3, installed reed valve with valve stopper is a typical discharge valve of rotary compressor. Enlarge discharge port to reduce pressure loss makes cracking valve risk higher due to high stress in valve closing on discharge port. On the other hand, thicker reed valve has more damage risk at valve gland because of high bending stress in maximum valve lift.



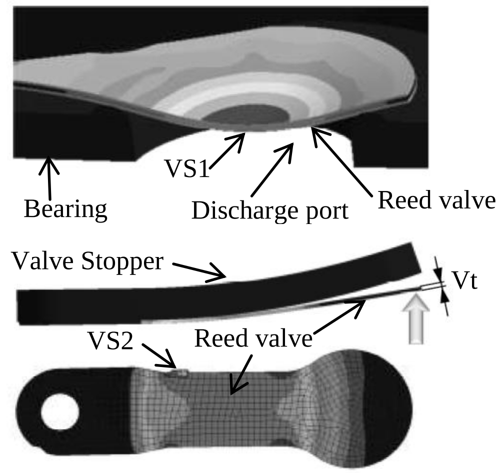


Figure 3: Reed valve and cross-section view

Figure 4: Maximum stress analysis

As shown in figure 4, the structural analysis simulations were used to predict the impact fatigue limit of valve tip VS1 on contact port and the maximum bending stress of valve gland VS2 in maximum lift. VS1 was analyzed in pressure difference 3.7 MPa and VS2 under ASHRAE condition at 90 rps of rotation speed with R410A refrigerant. The results are defined as a relation of discharge port diameter related to the ratio of each stress VS1, VS2 to the allowable stress $VS1_c$, $VS2_c$. As a result, in order to satisfy the reliability standard and to make discharge port diameter larger than 13 mm, it is necessary to design valve thickness V_t thicker than 0.55 mm as seen in figure 5 and to make V_t thinner than about 0.58 as in figure 6. Based on these results, we found that discharge port diameter larger than 13 mm is risky.

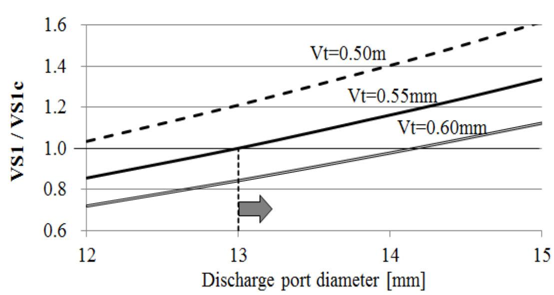


Figure 5: Stress ratio of valve tip versus discharge port diameter

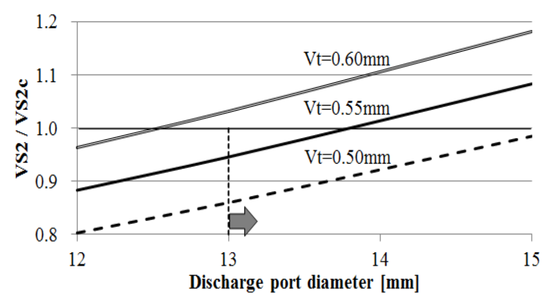


Figure 6: Stress ratio of valve gland versus discharge port diameter

Optimization of cylinder diameter, crankshaft eccentricity, and determined discharge port diameter are applied into a correlation of cylinder height and ratio of reference value Q_c to mean flow velocity Q as given in figure 7. From the correlation above 28 mm or less of cylinder height satisfies the allowable pressure losses throughout discharge process.

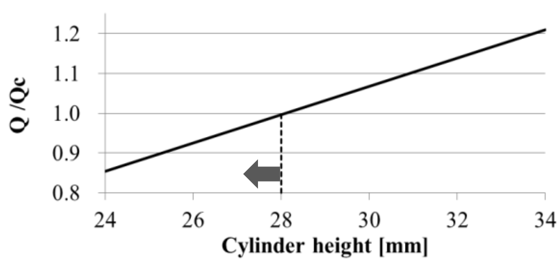


Figure 7: Flow rate ratio versus cylinder height

2.3 Study of New Discharge Structure

Increasing capacity with extending cylinder height possibility increases flow loss throughout discharge process. In this case, for 2-cylinder rotary compressor, multiple discharge valves installment per cylinder is easier rather than enlarge the discharge port diameter due to maintain impact fatigue limit of valve. Since partition plate has divided as 2 parts as shown in figure 8, beside on bearing parts, some additional discharge ports can be installed on each partition plate [1]. By this structure, as shown in figure 9, discharge gas from upper and lower cylinder flow mix in flow-path inside partition plates. Fluid analysis method is used to optimize the flow path. By simulation result, discharge port diameter of partition plate is approximately 10 mm.

Furthermore, as shown in figure 10, from the correlation of the cylinder height and the ratio of flow rate Q/Q_c through discharge port, the maximum cylinder height is less than 44 mm.

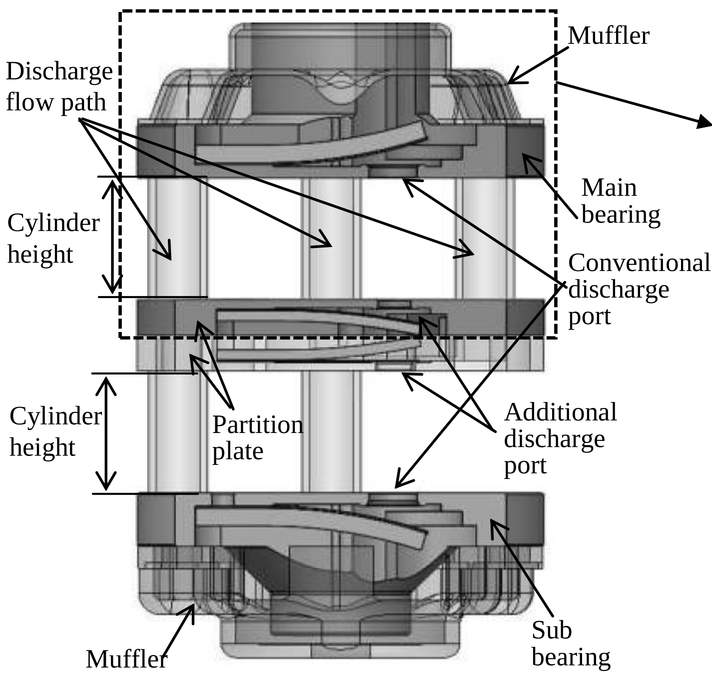


Figure 8: New discharge structure

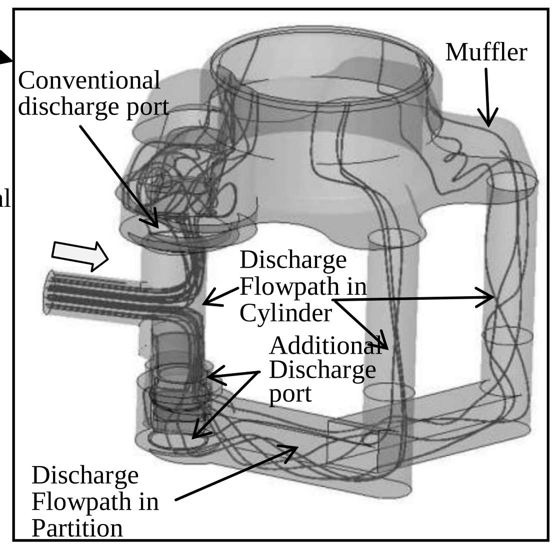


Figure 9: Discharge flow analysis

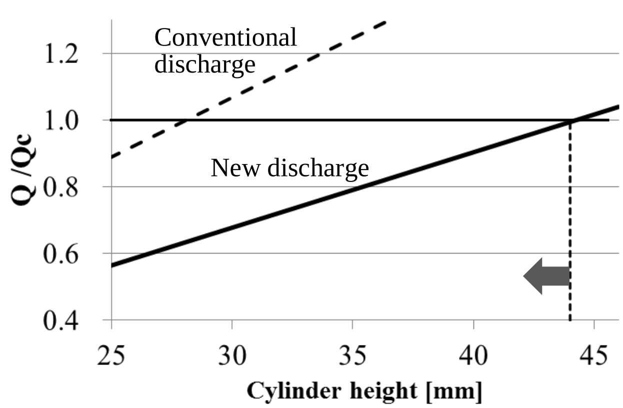


Figure 10: Flow rate ratio versus cylinder height

3. IMPROVEMENT BY NEW DISCHARGE STRUCTURE

3.1 Performance Improvement by New Discharge Structure

Figure 11 shows the Coefficient of Performance (COP) comparison between the new discharge structure and the conventional one in the same cylinder height less than 44 mm respectively, under ASHRAE condition with R410A refrigerant. COP ratio is COP which represents as a ratio to the conventional discharge model at 60 rps. COP of conventional discharge tends to drastically decrease in high-speed region. On other hand, the new discharge valve structure shows improvement about 10% of COP at 90 rps as result of pressure loss decrease in discharge process compared to the conventional one.

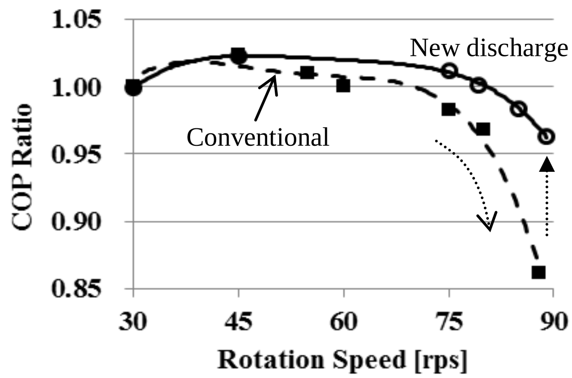


Figure 11: COP comparison of discharge structure

3.2 Compression Pressure and Valve Behavior

Comparison of new discharge valve structure and conventional model for each angle compression process in pressure change and valve behaviors were evaluated under ASHRAE condition at 60 rps of R410A refrigerant. The pressure change comparison which are plotted as figure 12 shows the excessive compression of conventional discharge structure is reduced by applying the new discharge structure.

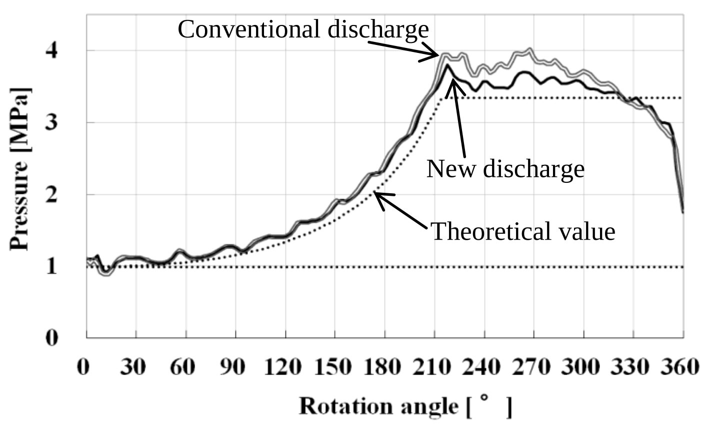


Figure 12: Pressure change inside compressor chamber

Valve behaviors at bearing side and partition plate was measured by pressure transducer sensors which are installed on valve stopper. Measurement result per one cylinder can be seen in figure 13. The reed valve on partition plates confirmed has 2 time open and close before and after discharge process of valve at bearing side. By applying new discharge structure, not only reducing the flow loss throughout discharge port, but also loss by valve fluttering can be decreased. From figure 14 of P-V diagram, the new discharge structure contributes to reduce a half of excessive compression that occurs in conventional model.

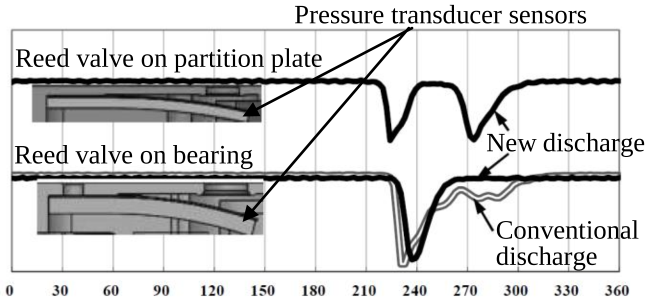


Figure 13: Reed valve behavior on partition and bearing part

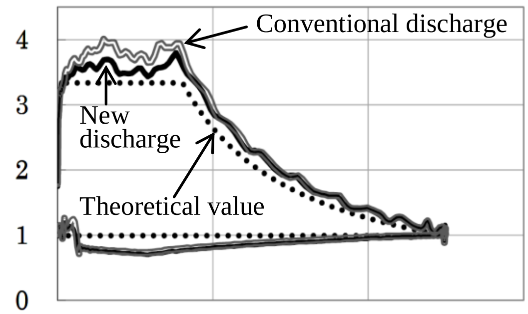


Figure 13: P-V diagram comparison

4. HIGH RELIABILITY PARTS

4.1 Appropriate Geometry of Journal Bearing

Increase capacity using shaft diameter as same as conventional model has some advantages in low sliding loss and low cost in manufacturing, but some excessive surface pressure contact on bearing occurs because of shaft deflection getting worse due to compression load and extended bearing distance. Still, as consequence of increasing crankshaft eccentricity, coupling moment of crank should be encountered by increasing the balancer weight to reduce vibration. On other hand during high-speed rotation, centrifugal force of counter balance leads to the excessive local surface pressure risk on the top surface of bearing.

Regarding described matter above, as shown in figure 14, we optimized the shape of parts such as taper ring grooves at bearing and strengthened the rigidity between the 2 crankshafts. Also, adopting rotor with counter bore to permit an installment of extended main bearing height with tapered shape. Shaft and bearing contact surface pressure was analyzed by structure simulation as show in figure 15. As a result of optimization geometry of sliding part as shown in figure 16, the maximum contact surface of developed model has been reduced about half comparing to the conventional model.

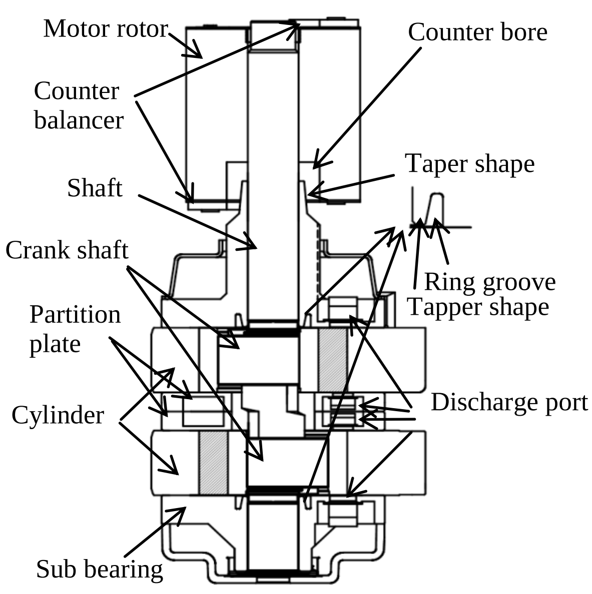


Figure 14: Cross-section view of development model

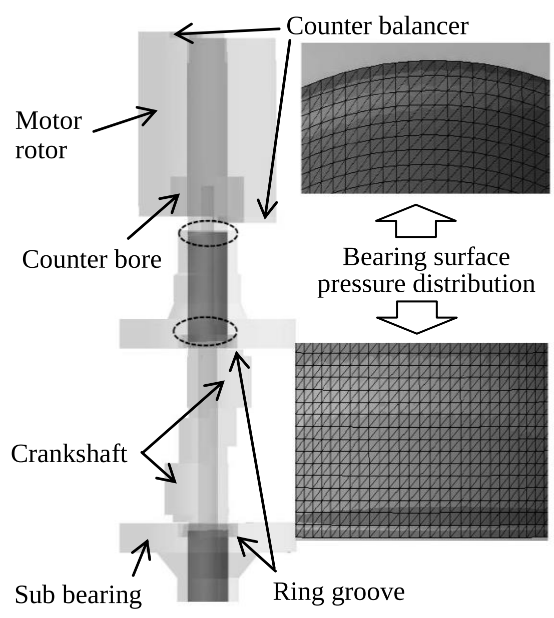


Figure 15: Contact surface pressure by simulation

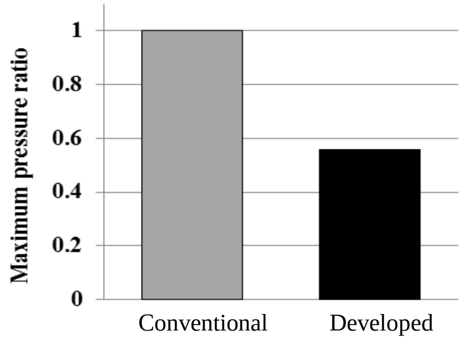


Figure 16: Maximum contact surface pressure of journal bearing

4.2 Divided DLC Vane

As a consequence of increasing capacity by extending dimension of the cylinder height and crankshaft eccentricity, it is necessary to maintain the durability of sliding contact surface between vane tip and rolling piston mechanism due to the severe lubrication environment on its contact surface. Furthermore, since back pressure of vane is ought to be a proportion of cylinder height, excessive surface pressure risk at contact on vane tip increases due to extend the vane height. Beside improve the abrasion proof strength of vane surface layer by adopting our particular Diamond Like Carbon (DLC) coating to increase the adhesiveness surface layer of vane, also vane divided as two parts as shown in figure 17. By divided 2-vane structure, the excessive local surface contact pressure between vane tip and rolling piston as described above can be reduced[3,4].

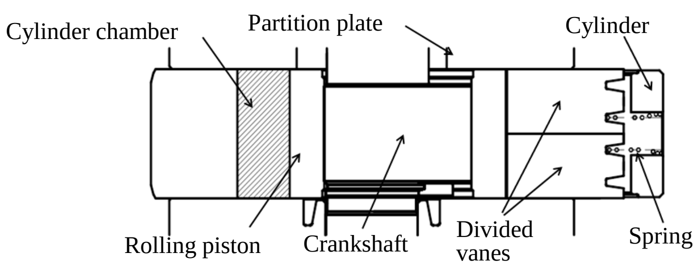


Figure 17: Divided 2-vane structure

5. HIGH EFFICIENCY MOTOR

Referring to high load of large capacity compressor, the concentrated winding 6 pole motor driven by new PWM converter is applied to achieve a high efficiency. A concentrated motor as shown in figure 18 is designed as lighter as possible with small loss of electricity flow. Rotor cross-section area is optimized by magnetic field analysis simulation to get an appropriate rotor's geometry in order to achieve a maximum discharge flow throughout motor. Therefore, high-speed switching of PWM converter has possibility to control electric harmonic wave, to improve power factor of motor's power source and to combine with boosted DC voltage. By those function, new PWM converter contributes to improve the compressor performance because of its ability to control motor electric current in all range of operating condition.

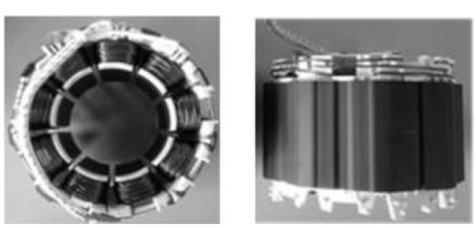


Figure 18: 6 pole concentrated winding motor

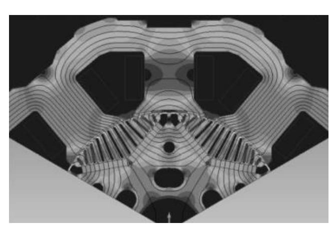


Figure 19: Magnetic field simulation of rotor

6. SPECIFICATIONS & PERFORMANCE

6.1 Performance Improvement and Loss Analysis

Specifications of new developed model comparing to conventional model are described in table.1. Comparison performance of models can be seen in figure 20, shows the cooling capacity tendency related to the performance characteristic which is represented as the ratio to the conventional model at 60 rps. The performance shows that COP improves about 6 % in most operating condition range under ASHRAE condition with R410A refrigerant and also it realizes the capacity equal to 20 HP at 90 rps by optimizing the supercharging effect at accumulator.

	Developed	Conventional
Compressor Type	2-Cylinder Rotary	←
Capacity Ratio	1.7	1
Refrigerant	R410A	←
Case Inner Radius [mm]	< 80	←
Cylinder Height [mm]	< 44	< 28
Discharge Port Diameter [mm]	Dia.13 / Dia.10	Dia.12
Concentrated Motor's Pole	6 pole	4 pole

Tabel 1: Compressor specification comparison

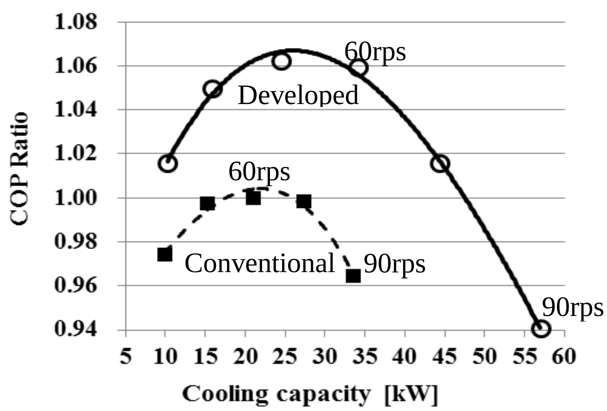


Figure 20: COP comparison

Loss comparison analysis of new model refers to the conventional at 60 rps under ASHRAE rate cooling condition can be seen in figure 21. In this analysis, assuming the conventional model has 100% loss. Even though developed model has loss at TCV (Top Clearance Volume) of discharge port and other mechanical part, totally about 20% loss can be reduced through some improvements such as new discharge valve structure, appropriate journal bearing, concentrated winding motor, PMV converter, and high volumetric efficiency.

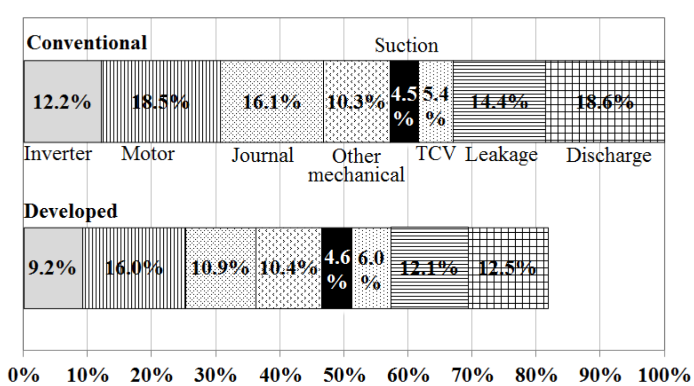


Figure 21: Loss analysis comparison

6.2 Performance Improvement and Supercharging Effect

Suction supercharging effect is defined by optimizing the suction piping dimension inside accumulator as shown in figure 22. Supercharging effect as a result of suction pulsation in a certain frequency was calculated by simple equation as given as follows:

$$\boldsymbol{f} = C/(4\times(L+(V_s/\ A_p\ )) \qquad (5)$$

Where supercharging frequency $\boldsymbol{f}$ is related to refrigerant sound speed C, suction line distance L, suction volume V_s, and cross area of suction pipe A_p.

As shown in figure 23, optimized dimension in suction line shows amount of extraordinary volumetric efficiency occurs at 60 rps or over in rotation speed. Particularly at 90 rps, compression in volumetric efficiency 111% has led to maximum capacity that equal to 20 HP under ASHRAE condition with R410A refrigerant.

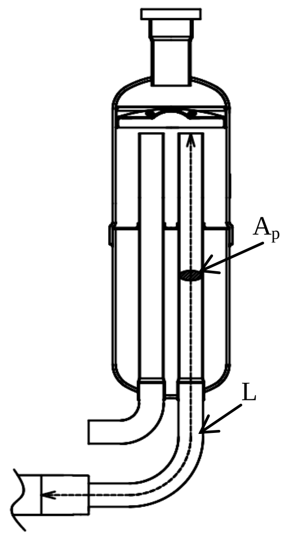


Figure 22: Piping dimension parameter for supercharging effect

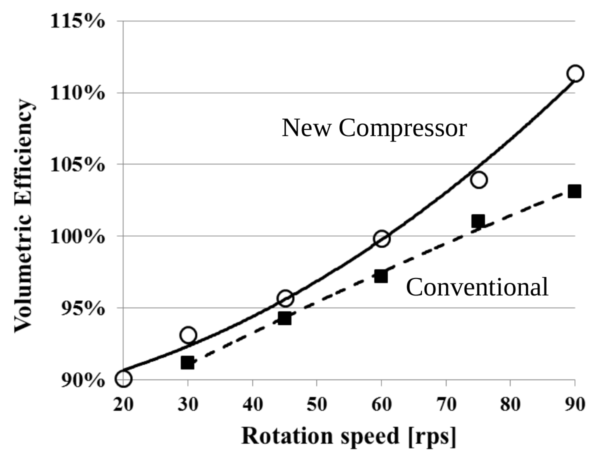


Figure 23: Volumetric efficiency comparison

7. CONCLUSION

By adopting some elemental technologies for large capacity and high efficiency in the rotary compressor, we have developed a new model which can improve the COP ratio by about 6 % in most of all operating condition ranging compare to the conventional model. Optimization dimension of some part such as cylinder inner diameter, cylinder height, crankshaft eccentricity and also adopting the new discharge structure, the divided DLC vanes, 6 poles concentrated winding motor with PWM converter are important keys to increase compressor capacity 1.7 times while maintain a high efficiency.

4 developed compressors models are installed per module in our module type air-cooled modular heat pump "Universal Smart X Series "[7] has increased the rated capacity of the latest unit has increased from 50 HP to 70 HP (200kW). Moreover, in the latest type of 60 HP class, applying developed compressors can achieve high IPLV of 6.0 as top performance in the same class (TCC investigation, 2017th). This developed model is currently the largest capacity rotary type in the world and still possibility to develop for larger scale of air conditioner in future.

NOMENCLATURE

A	Cross-section area of cylinder	(mm^2)
A_p	Cross-section area of piping	(mm^2)
C	Refrigerant sound speed	(m/s)
E	Crankshaft eccentricity	(mm)
f	Supercharging frequency	(Hz)
L	Suction line distance	(mm)
R	Radius of case	(mm)
R_o	Roller outer radius	(mm)
R_{cy}	Cylinder inner radius	(mm)
Q	Average flow velocity	(m/s)
Q_c	Standard of flow velocity	(m/s)
V_s	Suction Volume	(mm^3)
VS1	Head valve maximum stress	(MPa)
VS2	Neck valve maximum stress	(MPa)
$VS1_c$	Standard of head valve maximum stress value	(MPa)
$VS2_c$	Standard of neck valve maximum stress	(MPa)
Vt	Reed valve thickness	(mm)

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