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DESIGN AND DEVELOPMENT OF A SINGLE VANE ROTATING SLEEVE ROTARY COMPRESSOR FOR REFRIGERATION SYSTEM

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ABSTRACT

Preliminary test of a single vane rotating sleeve rotary compressor for refrigeration system application is presented in this paper. Generally, single vane rotating sleeve rotary compressor is a type of rotary compressor family that has been developed recent years. The characteristics of the compressor design and compression concept in such that of this compressor is supposed to be better than others rotary compressor in term of leakage and friction. The operation of the compressor mechanism is directly from the shaft to the mechanism compression components. This mean that when the shaft start to rotate, the mechanism compression components will be rotate together in the same direction and this mechanism is able to compress the refrigerant at all speed of rotation. This is because the design of the vane tip is embedded in the swing slot at the rotating sleeve and the other end side of vane is embedded into the rotor slot. The fabrication of prototype of the compressor needs the concentration because it is involving the high precision of machining and mostly of the tolerance was applied in this design is around 5 μm to 20 μm. Computational analysis of the design shows that the design specification is safe to be applied in refrigeration system with the maximum stress distribution of 31 MPa and deflection 0.55 μm.

***Keywords**:* Rotary Compressor, Compressor Design, COP

INTRODUCTION

Generally, compressor is a mechanical device to raise the pressure and the temperature of refrigerant and to provide primary force to circulate the refrigerant through the entire system. The high-pressure vapour is then fed into condenser where the refrigerant is condensed into liquid form, then throttles to a lower pressure through capillary tube and produces the refrigeration effect in the evaporator. According to the compression process, the refrigerating compressor can be divided into two main classifications and each classification can be further sub-divided into several groups, as illustrated in Figure 1. The positive displacement compressor is a type that increases the gas pressure by reducing the internal volume of the compression chamber through the mechanical force that applied to the compressor. Meanwhile, a non-positive displacement compressor is where the compression of the gas depends mainly on the conversion of dynamic pressure into static pressure.

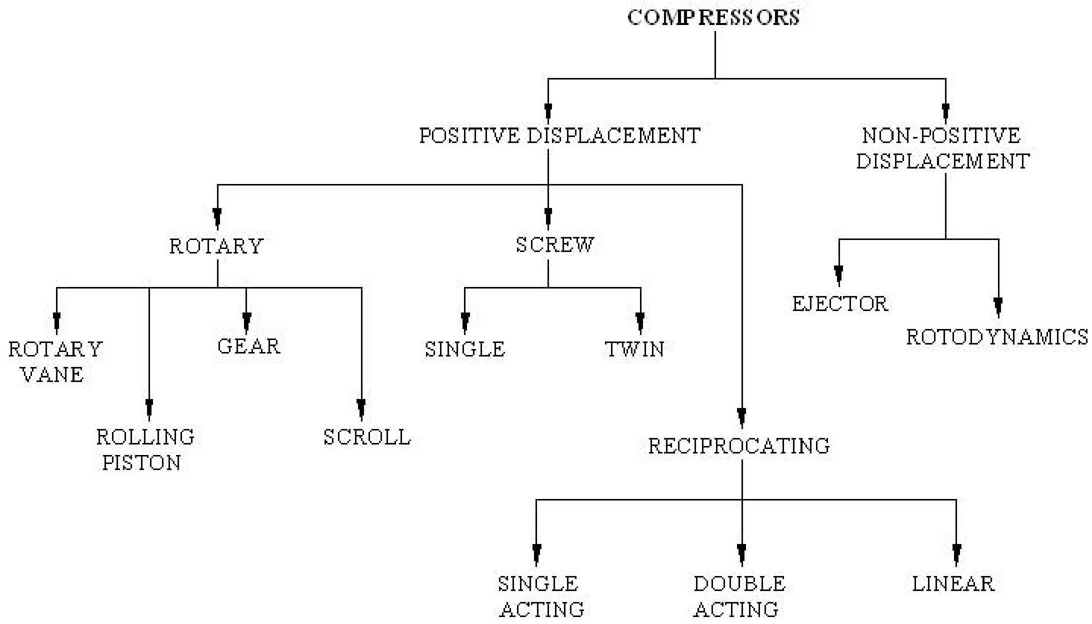

Figure 1: Classification of compressor

The application of reciprocating compressor in a refrigerator and an air-conditioner is already established. Chillers for some big building air-conditioning system are using screw compressors, but the researches are still on-going to improve the performance. Whereas automotive air-conditioning system are using both rotary and reciprocating types and again research in this area is actively pursued. Domestic refrigerator has been known to use reciprocating compressor until lately when rotary compressor has been introduced and appears to be successful. This success is as a result of continuous research carried out by the industry to improve the efficiency and reliability of rotary compressors.

The literature study has been done on the rotary and reciprocating compressors and findings showed that the performance of rotary compressor is better than reciprocating compressor. Single Vane Rotating Sleeve Rotary Compressor (SVRSRC) is a new compression concept that has been developed and it comprises of a rotating vane, a rotating sleeve and a rotor. This is a simple concept compared to the other rotary compressors available in the market today. Details of the new rotary compressor concept are shown in Figure 2. The main components are shaft, rotor, vane, sleeve and cylinder block. These components are assembled to produce the compression mechanism with the crescent-shaped space, created by the eccentrical arrangement of rotor and sleeve. The vane which is driven by the rotor divides the crescent-shaped space into two compartments, comprising of low and high pressure chambers respectively.

Figure 3 shows the working sequence of the compressor, it starts with entering of vapor at 0° angle of rotation and ends with complete vapor delivery at 360° rotation angle. The low-pressure gas enters the suction chamber simultaneously during compression and discharge modes.

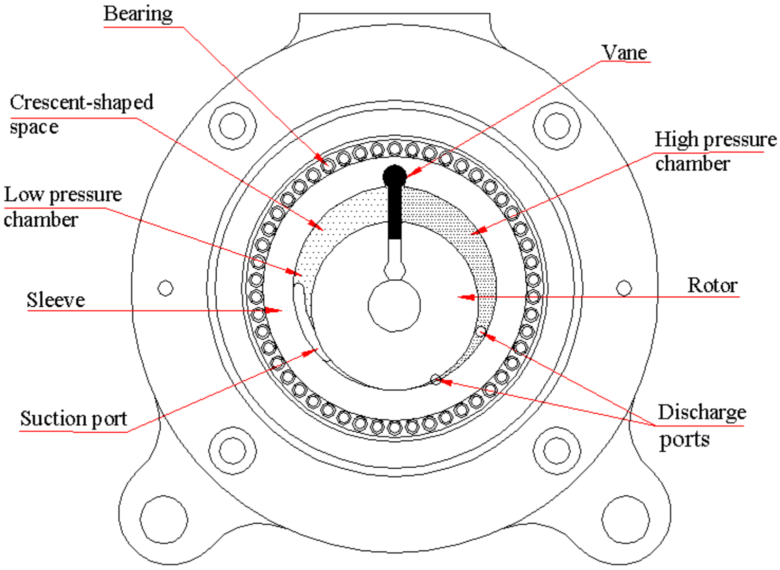

Figure 2: Concept of SVRSRC

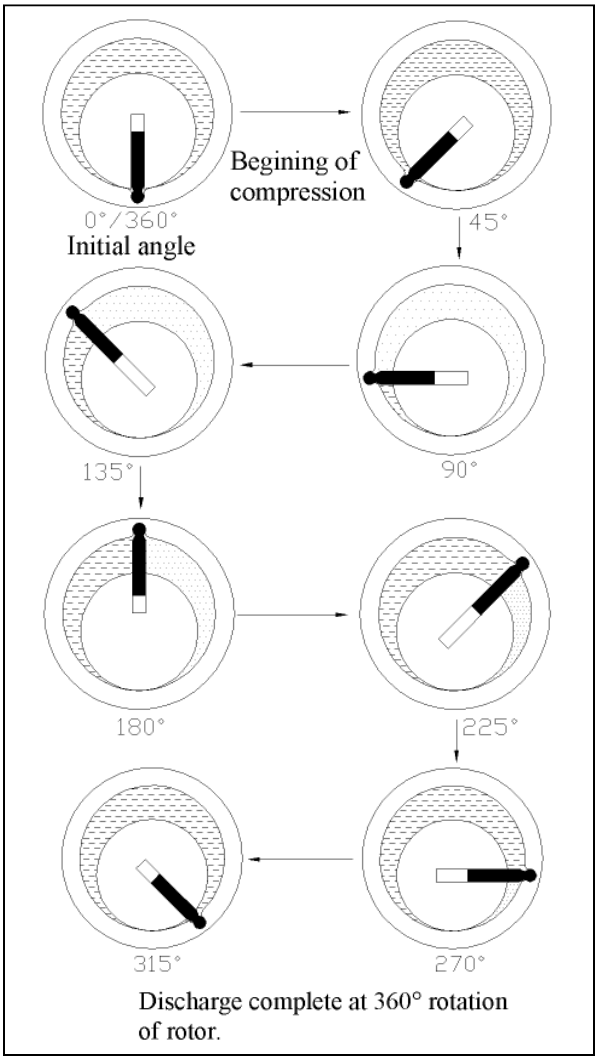

Figure 3: Working Sequence of the Compressor

DESIGN METHODOLOGY

Compressor Geometry Design

Generally, this concept consists of two (2) eccentric circles as shown in Figure 4.

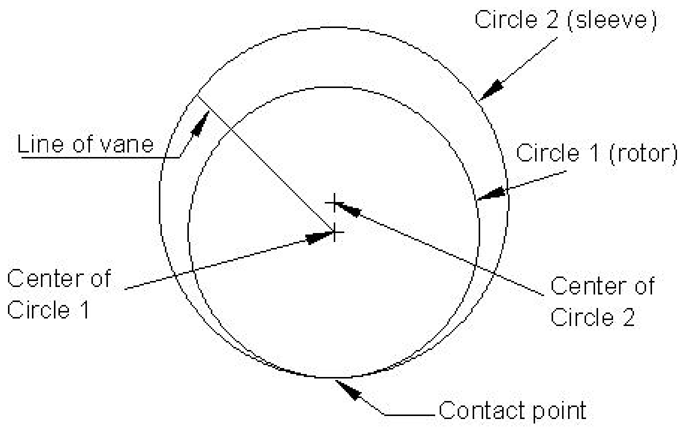

Figure 4: Basic geometry of compression concept

Circle 1 function as a rotor and circle 2 as a sleeve with different center points. These circles touch each other only at one point which is called the 'contact point'. A line from center of circle 1 to any point on circle 2 represents a vane at that particular position of the rotation. Figure 5 shows the detail geometry of compressor concept. The area of cde (Δcde) is the compressed area which is to be derived in term of other geometrical areas. Angle theta (θ) is the rotation angle of the rotor that must be determined. An analysis has been done to get an expression to relate θ with swept area.

The area to be determined is Δcde, which is the compressed area thus;

$$\Delta cde = \Delta ace - \Delta bcd - \Delta abe \tag{1}$$

Where;

$$\Delta ace = \frac{\beta}{360}\left[\frac{\pi D^2}{4}\right]$$

$$= \left\{\theta - \mathrm{Sin}^{-1}\left[\frac{(R-r)\mathrm{Sin}\,\theta}{R}\right]\right\}\frac{\pi R^2}{360} \tag{2}$$

$$\Delta bcd = \frac{\theta \pi r^2}{360} \tag{3}$$

$$\Delta abe = \frac{1}{2}(af)(ef - bf)\text{, where } ef - bf = eb$$

$$= \frac{1}{2}\left[R\,\mathrm{Cos}\left\{\mathrm{Sin}^{-1}\left[\frac{(R-r)\mathrm{Sin}\theta}{R}\right]\right\} - (R-r)\mathrm{Cos}\theta\right]\left[(R-r)\mathrm{Sin}\theta\right] \tag{4}$$

All of the equations involved are expressed in term of *R*, *r* and θ respectively. The value of θ varies from 0° to 360° whereas the values of *R* and *r* are to be specified. The ratio of r to R is called design ratio. The recommended value of the design ratio is taken as 0.83 (Meece, 1974). Thus,

$$\frac{r}{R} = \frac{d}{D} = 0.83 \text{ or } d = 0.83D \tag{5}$$

$V = \frac{\pi}{4} D^2 t$, *t* is the height of rotor and sleeve and is taken equals 20 mm and the volume is 6.6 cm^3 based on volume of existing compressor. Thus,

$$6.6 = \frac{\pi}{4}(D^2 - d^2)(2)$$

$$(D^2 - d^2) = 4.2017 \tag{6}$$

Substituting $d = 0.83D$ into equation 6, thus,

$$D^2 - (0.83D)^2 = 4.2017$$

$$D^2 - (0.6889D^2) = 4.2017$$

$$0.3111D^2 = 4.2017$$

$$D = 36.75 \text{ mm}$$

Substituting $D = 36.75$ mm into equation 5,

$$d = 0.83D$$

$$= 0.83(36.75)$$

$$= 30.5 \text{ mm.}$$

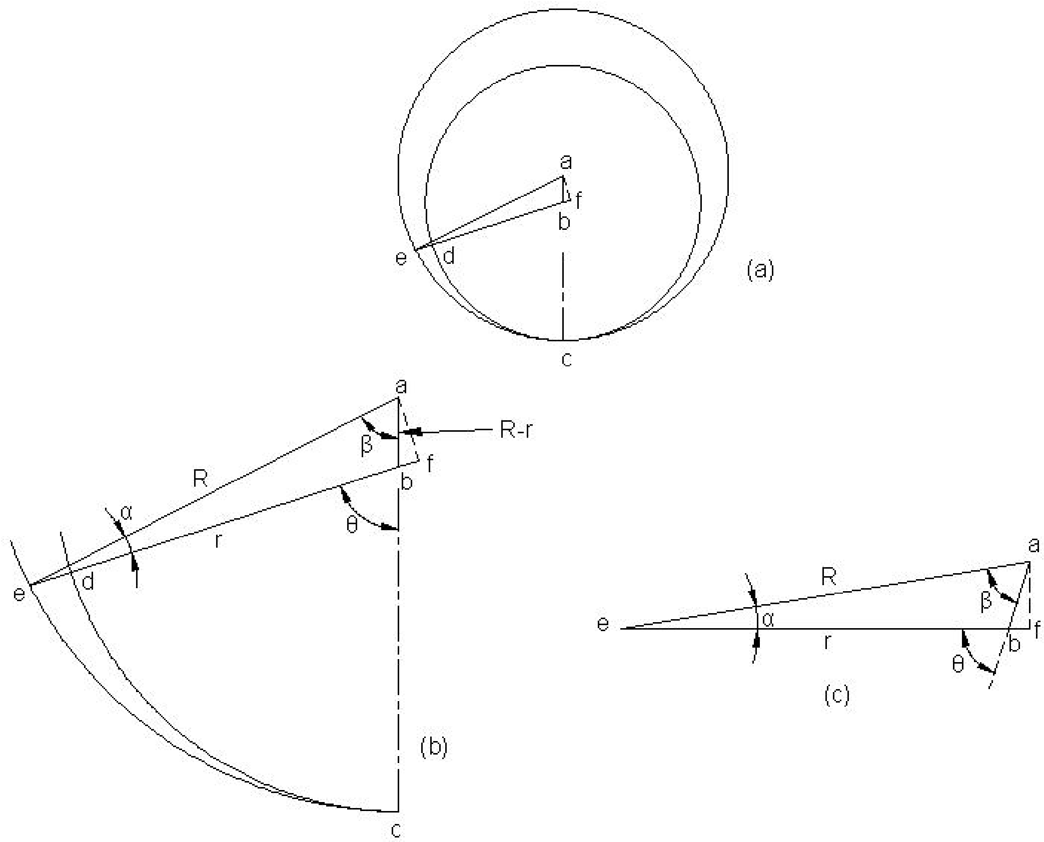

Figure 5: Detail Geometry of Compressor

However, in this design the suction port causes a decrease in the induced volume. Some gas is pushed out as the vane rotates from *c* to *e* with D = 36.75 and d = 30.5 mm, using equation (4.8) the actual swept volume is $6.6-(\Delta cde\times 2)\,\text{cm}^3$. To compensate for this loss of gas and at the same time maintaining the optimum r/*R* ratio of 0.83 both radii have to be increased. By using AutoCAD software and through trial and error, *d* was obtained equals to 32 mm and *D* equals to 38.5 mm. These dimensions are shown in Figure 6.

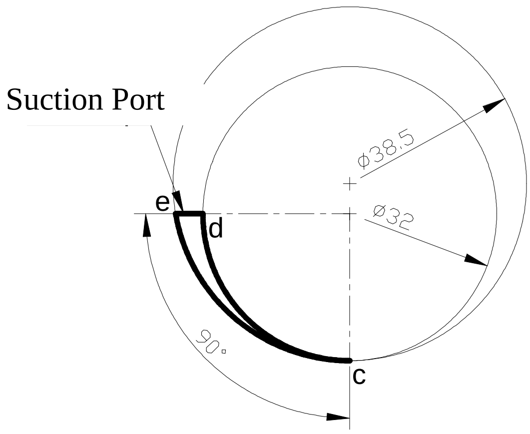

Figure 6: Geometry Design of Suction Port

For further check, with the new values of *d* and *D* respectively, the total swept volume can be calculated as 7.20 cm^3. However, with the compensation of suction port volume (*cde*), the effective swept volume can be shown by solving Equation 1. The value of *r*, *R* and θ used in calculation work are 1.6 cm, 1.965 cm, and 90 degree respectively.

The solution of equation 1 would be 0.27718 cm^2 and by considering volume of the suction port, it would be 0.55436 cm^3.

So, the volume of *cde* is $0.27718(2)=0.55436\ \text{cm}^3$ and this is the amount of gas that is being pushed out. The actual swept volume (V_S) of the compressor is;

$$\begin{aligned} V_s &= 7.20-0.55436 \\ &= 6.646\ \text{cm}^3 \end{aligned}$$

This approximately equals to the swept volume of the existing reciprocating compressor which is 6.6 cm^3.

Determination of Discharge Angle

Compression process starts at pressure p_1 when the vane has just rotated over the suction port. At certain angle of rotation, the pressure reaches p_2. It is at this point that the gas must be discharged. The theoretical discharge angle is calculated using geometrical relationship and using AutoCAD software. Both of these methods are based on the estimation of discharge volume V_2 from the following equation.

$$\frac{V_1}{V_2}=\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} \tag{7}$$

where V_1 is the actual swept volume equals 6.64 cm^3 and p_1 and p_2 are specified values. The polytropic index (n) was calculated using equation 8.

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} \tag{8}$$

where, T_1, p_2 and p_1 are actual values measured from experiment. The value of T_2 was estimated from isentropic process of compression ($s_1 = s_2$). The data of T_1, p_1, and p_2 were taken from a preliminary run of the experiment at which the freezer compartment temperature was constant at -15°C as below:

T_1 = 26.4°C p_2 = 9.8 Bar G p_1 = 0 Bar G

Therefore the values of s_1 and s_2 were estimated equal to 1.036 kJ/kg K. Based on NIST REFPROP software from ASHRAE, the theoretical value of T_2 is 105.2°C. Thus, using Equation 8, the polytropic compression index (n) is obtained equal to 1.11. From Equation 5, the discharge volume (V_2) can be calculated as equal to 0.7866 cm^3.The theoretical discharge angle was determined based on the area of A_2 which is equal to 0.3933 cm^2. By using AutoCAD, it was found that the corresponding discharge angle is about 258°.

Casing, Suction and Discharge Tanks

The compressor is design to have a casing comprises suction and discharge compartments whereby the suction compartment is to accommodate low pressure gas from evaporator. Meanwhile the discharge compartment is to accommodate discharge gas from the compression chamber. The positions of suction and discharge compartments were interchanged with each other. Both compartments were covered with casing as shown in Figure 7. This concept is quite similar to the existing concept of reciprocating compressor, where the compression mechanism is located in a hermetic tank which is filled by a low temperature refrigerant. The advantage of this concept is that the low temperature refrigerant will absorb the heat that is generated during compression process.

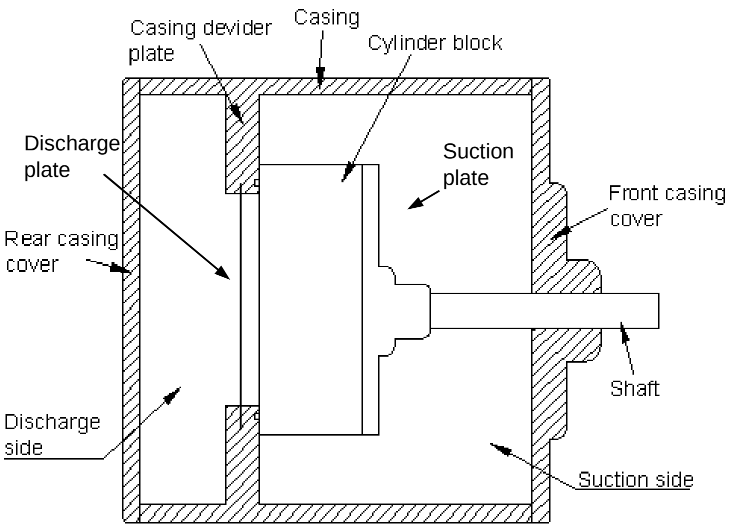

Figure 7: Design concept of casing

Suction and Discharge Ports

The suction port was designed through the front side plate (after this it is called as suction plate) such as depicted in Figure 8 and also can be referred in Figure 10. The profile of the rotor and sleeve form the shape of the suction port which is drilled on the suction plate between 65° to 90° from the sleeve/rotor contact point makes the opening angle of suction port to be 25°.

There is no valve required for suction port. After being compressed, the gas will be discharged through another port on the rear end plate (discharge plate). The discharge port has a reed valve installed. The discharge port concept was referred to that of Heui-Jong, K (Kang, 1998), where the discharge hole on the rotor was drilled partly radially and partly axially so that the axial passage can meet with the groove on the discharge plate. When the passage on the rotor meets the groove on discharge plate, gas will flow to the discharge hole. In the design, a reed valve was used to control pressure during discharge process (Buchanan and hubacker, 1933). Figure 9 describes the above discussion.

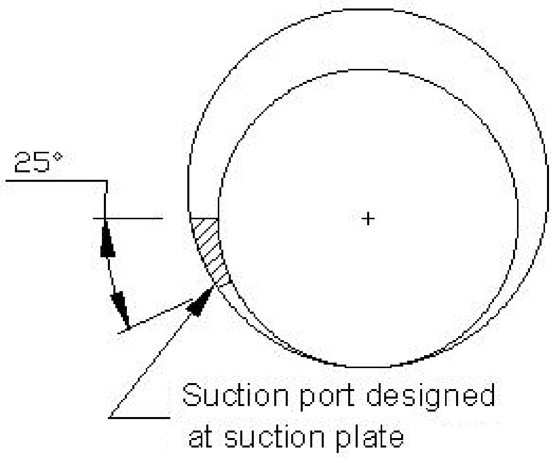

Figure 8: Suction port design of second prototype

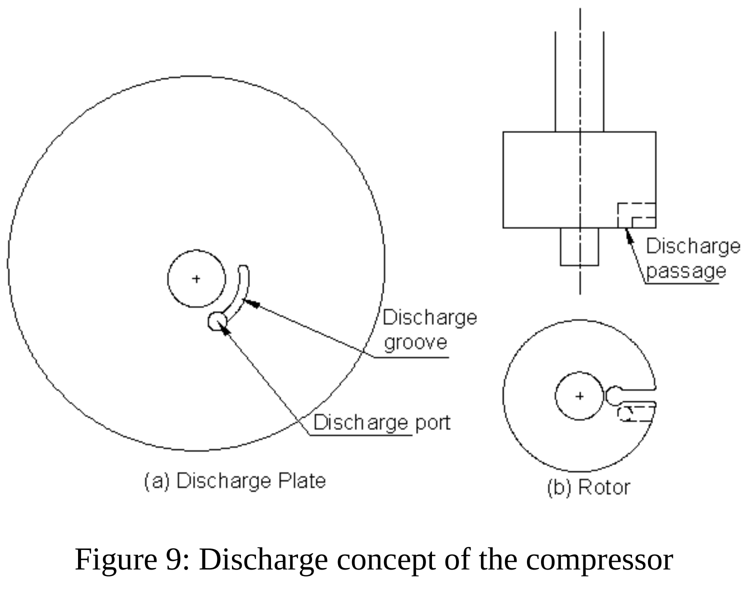

Figure 9: Discharge concept of the compressor

Lubrication Oil System

The lubrication system was designed with a hole 0.5 mm diameter drilled through on the suction plate and small capillary tube was used to connect the hole to an oil sump. Both refrigerant gas and little amount of lubrication oil were sucked into the vacuum chamber created after the vane passed the sleeve/rotor contact point ($\theta = 0°$) as shown in Figure 10.

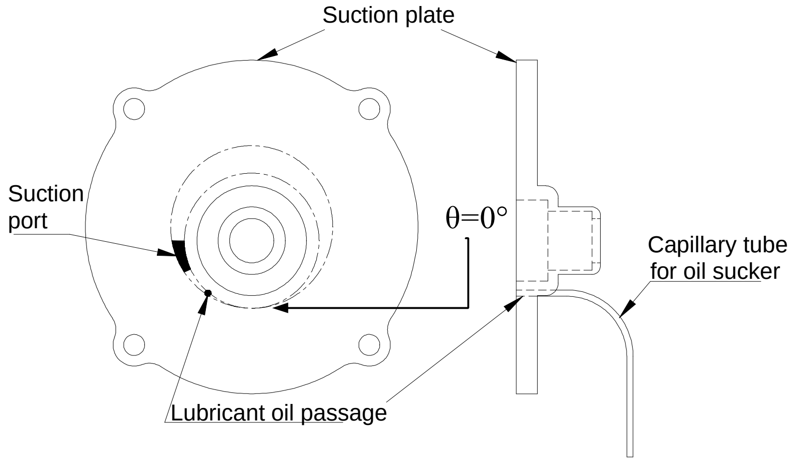

Figure 10: Lubrication Oil System on the Suction Plate

Design of Compression Component

Compression components were defines as components that create the compression chamber such as rotor, vane and sleeve. These components were designed based on the geometrical dimension as discussed before. Heights of these components are equal to 20 mm with tolerance of 0 mm to -10 µm. The rotor was designed as one piece with shaft and dimension of the shaft was matched with the needle roller bearing (SKF, 1991). The tolerance for radius of rotor must be determined by considering the maximum tolerance of contact point between rotor and sleeve. Since the tolerance of contact point was decided as 7.5 µm, the rotor tolerance was obtained as –5 µm to 0 µm in diameter.

The cylinder block was designed with center bore to insert the needle roller bearing HK 5020 SKF and all compression components. Height of cylinder block was designed equal to 20 mm with +5µm tolerance. The sleeve was designed to be putted inside the HK 5020 SKF roller bearing. The tolerance of sleeve inner diameter should be referred to the rotor tolerance and sleeve/rotor contact point clearance. Thus, the tolerance of sleeve inner diameter was obtained as 0 µm to +10 µm. The vane was designed based on the geometry that has been discussed with 20 mm height. The tolerance of vane height is equal to the tolerance of rotor and sleeve height whereas the tolerance of vane width was obtained as –12 µm to 0 µm and vane slot at rotor was obtained as 0µm to +10 µm. Figure 11 and 12 shows the design of the discussed components.

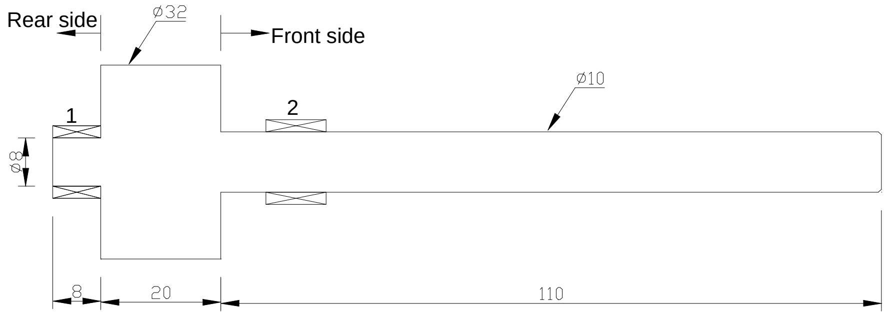

Figure 11: Side View of Rotor Design

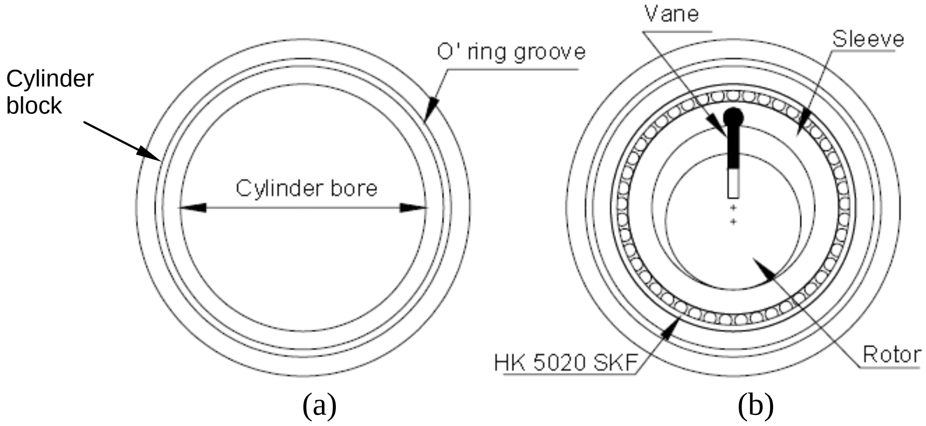

Figure 12: Top view of Compression Components

Material Application

Materials from 718 pre-hardening of ASSAB steel, carbon steel and polyshaft were used as compression component in the third prototype. The materials were treated with full hardening process to increase the material hardness to make it more reliable during operation. Casing was not considered as compression component and was made of mild steel.

RESULT AND DISCUSSION

Overall Design

Figure 13 shows exploded view for the whole design of the compressor. There are 11 parts that have been designed and 10 standard parts. The standard parts are considered as various sizes of screw, various sizes of bearing, oil seal and gasket.

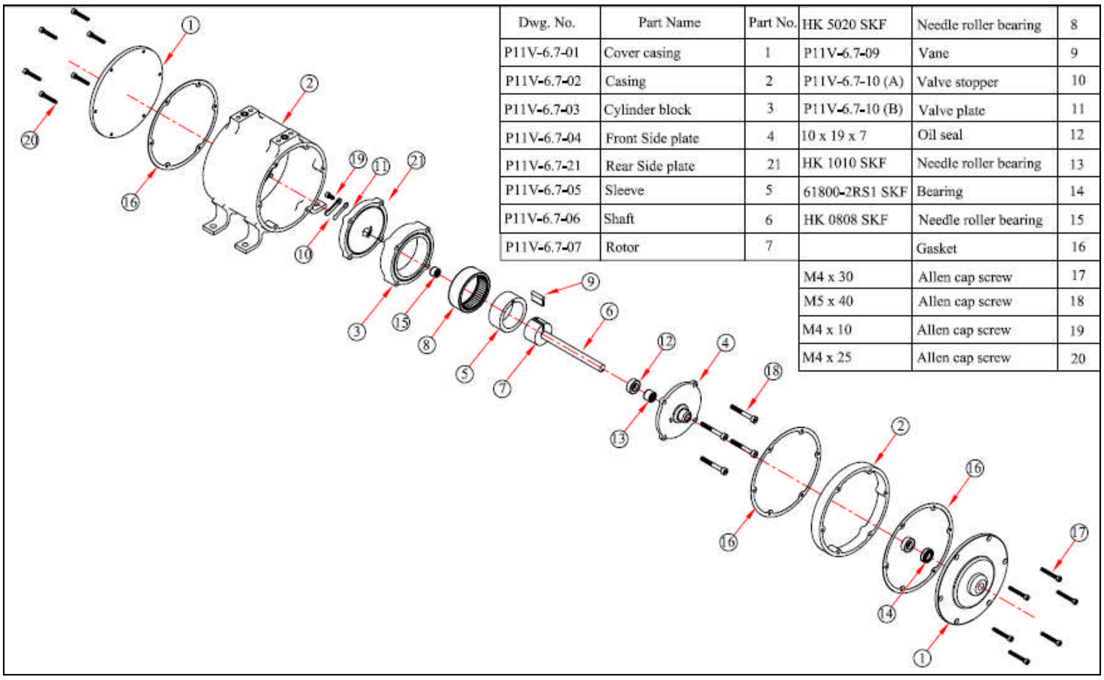

Figure 13: Exploded View of SVRSRC

General Specification

Table 1 shows the general specification of the rotating sleeve rotary compressor. There were five (5) important components need to be focused whereby all of them were designed to form a compression chamber for the compressor.

Table 1: General Specification of the Rotating Sleeve Rotary Compressor

Comp.	Size (mm)	Measured Size (mm)	Allowed Size (mm)	Material	Surface Finishing
Sleeve	D_o=50	50.010	+0.005 to +0.02	718 - full hardening	Mirror finish
	D_i=38.5	38.501	0 to +0.010		
	t=20	20.000	-0.01 to 0		
Rotor	d=32	31.992	-0.005 to 0	Polyshaft	Mirror finish
	t=20	19.990	-0.010 to 0		
Vane	w=2.5	2.480	-0.012 to 0	718 - full hardening	Mirror finish
	l=14	13.986	-		
	t=20	19.999	-0.010 to 0		
Shaft	D=10	10.005	±0.007	Polyshaft	Mirror finish
Cylinder Block	D=79	79	-	Carbon steel	-
	d=58	57.985	-0.033 to –0.014		Not required
	t=20	20.002	0 to +0.005		

The critical parts of this design are the radial clearance between sleeve and rotor at the contact point and the side clearance between sleeve, vane and rotor and the side plates respectively. Radial clearance was decided as 7.5 μm and side clearance was decided as 10 μm. Referring to Gasche, J. L *et. al*. (1998), clearance at 10 μm was reported sufficient to produce good performance and reduce internal leakage. Although the clearance is suitable for this application but the thermal expansion of material in this prototype was not considered. Thermal expansion will definitely increase the component size and if the increase is significant it will create excessive friction between rubbing surfaces causing the compressor to jam. Thermal expansion may to a certain extent be reduced by the refrigerant gas which enters the compressor at extremely low temperature.

Compressor Analysis

An analysis to investigate compressor strength has been done by using COSMOS software. The analysis was done in static condition and the analysis input was decided as the following:

- Theory: Von Mises Stress
- Material: Plain Carbon Steel
- Suction Pressure: 1.013 Bar.
- Discharge Pressure: 11 Bar.
- Motor torque: 2.387 N.m.

The torque value was defined at the shaft, suction pressure at suction chamber and discharge pressure at compression chamber. Figure 14 shows free body diagram subject to the operating condition. Meanwhile Figure 15 describes the element definition for analyzing where it was meshed at 166000 elements with 255000 nodes. Stress distribution can be analyzed as defined by the different of colors in Figure 16 whereby the maximum stress distribution is 31 MPa between vane slot and shaft at the rotor. The red color area indicates the high stress distribution. Meanwhile Figure 17 shows the result of component deformation which may occur with a deflection of 0.55 μm at a deformation scale 1:10519. From the analysis, it has been found that the minimum safety factor is 6.9 and the critical point occurs between vane slot and shaft at rotor.

Generally, the result of the analysis shows that the size of component designed is safe to be used. The safety factor value showed that the sizes that are applied are suitable.

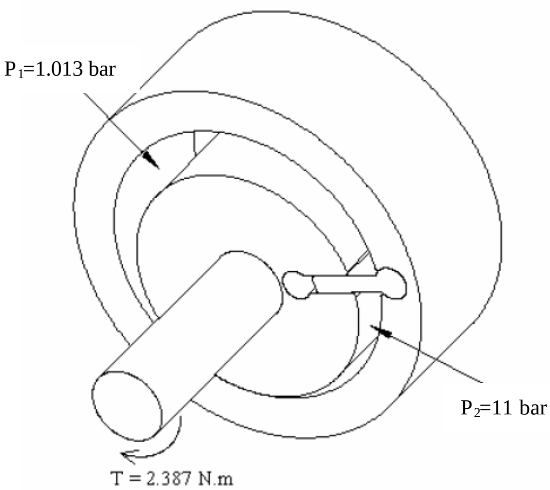

Figure 14: Input Setting

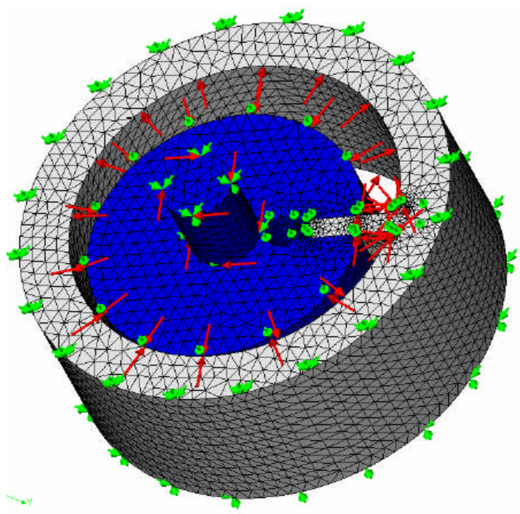

Figure 15: Element definition for Meshing

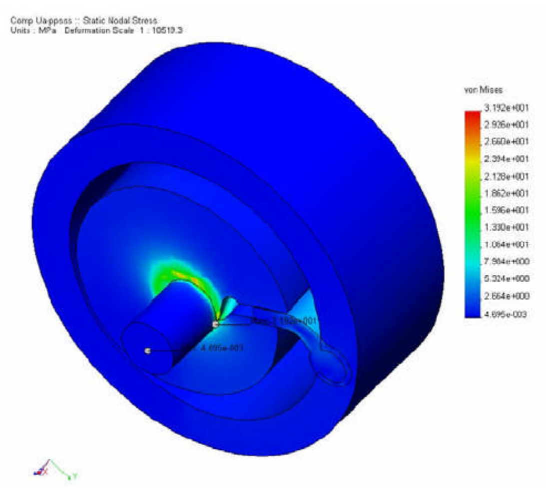

Figure 16: Stress Distribution

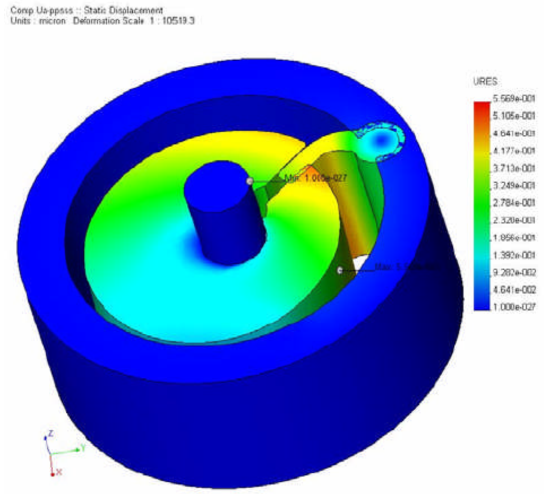

Figure 17: Components Deformation

Figure 18 shows pressure-volume graph that has been calculated by using equation 7. Intake volume (V_1) is taken at 7.2 cm^3 and inlet pressure at 1.0138 bar absolute. Compression process starts at point 2 and will be finished at point 3. Constant line from point 1 to point 2 is representing blade movement across suction port. At this time there is no compression. Constant pressure discharge also occurred from point 3 to point 4 whereby high pressure working fluid will be discharge into discharge tank.

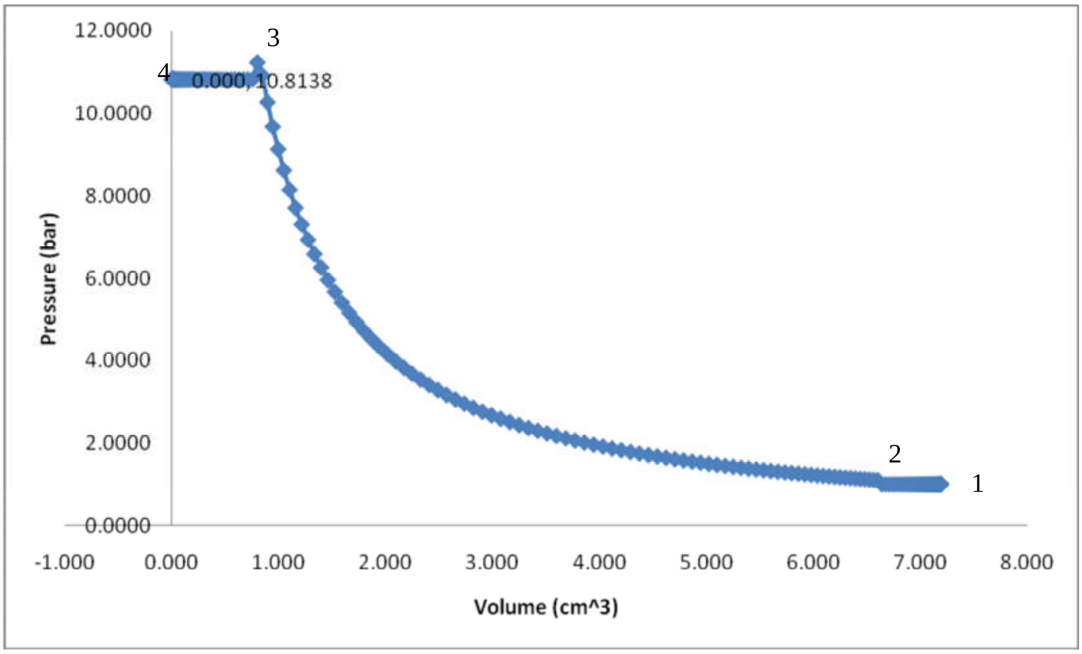

Figure 18: Pressure-volume graph of SVRSRC

CONCLUSSION

SVRSRC is designed comprises of 11 main components and 10 standard parts. The dimension and tolerance of each main components are very close with the maximum tolerance allowed is 0.033 mm and the minimum tolerance is 0.005 mm. The dimension of the design is able to support maximum stress at 31 MPa and 0.55 μm deflection at maximum pressure of 11 bar. Thus, the design is safe to be applied on refrigeration system.

ACKNOWLEDGEMENT

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Nomenclature

A	area
r	radius of rotor
R	radius of sleeve
d	diameter of rotor
D	diameter of sleeve
V_s	swept volume
V_1	volume before compression
V_2	volume after compression
P_1	suction pressure
P_2	discharge pressure
n	polytropic compression index

Greek symbols

β	rotating angle of rotor
θ	rotating angle of sleeve