Study on leakage loss via the radial clearance in a double-swing vane compressor for electric vehicle air conditioning systems

To cite this article: Hua Zhong *et al* 2019 *IOP Conf. Ser.: Mater. Sci. Eng.* **604** 012071

Study on leakage loss via the radial clearance in a double-swing vane compressor for electric vehicle air conditioning systems

Hua Zhong, Xiaoran Li, Xu Yang

School of Energy and Power Engineering, Xi'an Jiaotong University, 28 Xianning West Road, Xi'an, Shanxi 710049, China

yangzx@mail.xjtu.edu.cn

Abstract. The leakage loss in a novel double-swing vane compressor (DSVC) for use in electric vehicle air conditioning systems is theoretically analysed. The leakage flow through the radial clearance between the rotor and cylinder in the DSVC is modelled by considering both the section change of the leakage channel and the viscous friction of the working fluid. Under the same operating conditions and dimensions, the radial leakage loss of DSVC is more than that of the swing vane compressor (SVC) with a single swing vane. However, the theoretical suction volume of the DSVC is larger than that of the SVC. The relative volumetric leakage loss is almost equal to that of the SVC, which indicates that compared to the SVC, the effects of the structural changes of the DSVC on the volumetric efficiency is neglectable. Considering the larger work capacity and higher mechanical performance of the DSVC, it is suggested that the DSVC is a better choice for the electric vehicle air conditioning system.

Keywords: Leakage loss, Double-swing vane compressor, Simulation, Comparison.

Nomenclature

e	Distance between cylinder centre and rotor centre [m]	l_f	Length of straight channel[m]
w	Rotor axial length [m]	M	Mach number
P	Pressure [Pa]	ξ	Pressure ratio

q_m	Leakage mass flow rate [kg·s^{-1}]	μ	dynamic viscosity [Pa·s]
R	Radius [m]	δ	clearance value [m]
T	Temperature of the fluid[K]	φ	rotation angle [rad]
V	velocity[m·s^{-1}]	κ	Adiabatic exponent
R_g	Gas constant[J/(kg·K)]	λ	Friction factor
		Subscripts	
r	Rotor	b,c	Suction and compression chamber
cy	Cylinder	e,t	e and t sections
s	Suction	ξ	Pressure ratio
d	Discharge	*	Critical

1. Introduction

As the key component in air conditioning systems in modern vehicles, the compressor is independently driven by an electric rotor instead of belt driven by a combustion engine. So the rotational speed of the compressor and work capacity of the air conditioning system should be easily adjusted by a variable speed motor, which required to work at variable speeds in a wide operational range[1]. For the limited battery capacity of electric vehicles, the compressor unit is preferred to be more efficient, smaller in volume and lighter in weight to promote the performance of the vehicles[2].

So far, several types of compressors which include scroll compressors, wobble plate compressors, swash plate compressors and rotary vane compressors have been applied in the traditional vehicle air conditioning systems. Each type of these mechanisms carries with its inherent operational characteristic. For the compressor is driven by an electric rotor, the rotational speed of compressor is high. So the issues of noise and friction arising from the high operating speed limit the reciprocating type compressors' application in electric vehicles which means that the rotary type compressors exhibit better performance in electric vehicles compared to reciprocating type compressors. For example, the scroll compressor is considered to be suitable for application in electric vehicles due to its high efficiency, small volume and light weight[3]. However, this type of compressor has a higher manufacture cost because its main components are structurally complex.

To develop a low cost rotary compressor for electric vehicle air conditioning systems, a novel double-swing vane compressor (DSVC) is proposed by Yang[4]. This type of compressor is designed on the basis of the swing vane compressor (SVC) presented by Hu[5], which is structurally simple and easy to manufacture and also has a higher mechanical efficiency. Compared with the swing vane compressor (SVC), the performance of the DSVC is significantly improved under the same operating conditions and dimensions by adding an additional vane in the mechanism. The design of the DSVC brings a larger work capacity compared to SVC, which suggests that the DSVC can be constructed with a smaller volume and lighter weight when used in electric vehicles. It also found that the mechanical efficiency of the DSVC is higher than that of the SVC under the same operating conditions and dimensions. The previous investigation shows that the DSVC is a type of promising compressor for the application of electric vehicle air conditioning systems.

As is known to all, leakage loss in the compressor is one of the most important factors that affect its performance by reducing the volumetric efficiency. In this way, in order to develop a novel type of compressor such as the DSVC, it is necessary to figure out the leakage characteristics of the DSVC, which is helpful to implement the parametric optimization work of the DSVC for a higher performance. Referring to previous investigations of leakage losses in rotary type compressors, the leakage loss via the radial clearance between the rotary and the cylinder occupies more than two thirds of the total leakage losses in the compressor, which indicates that the radial clearance leakage is the key factor that affect the performance of the DSVC. In this paper, we theoretically analyzed leakage losses via the radial clearance in the DSVC. Then a comparison study between the leakage losses via radial clearances was also conducted, which provide a demonstration on the feasibility of the DSVC used for the electric vehicle air conditioning systems.

2. Working principle

Figure 1 shows the structural characteristics and working principle and of the DSVC. It can be observed that the DSVC essentially consists of a rotor, cylinder and two swing vanes. Unlike to the SVC with a single swing vane, the DSVC has two swing vanes named upper vane and lower vane. One end of the upper vane is inserted into the vane slot on the rotor while the other end is jointed with the cylinder. One end of the lower vane is jointed with the rotor while the other end is inserted into the guideway assembled with the cylinder. This brings a significant modification of the DSVC of that the trapped volume within the rotor and cylinder is divided into half by the two vanes which results in significant changes in the work process and the suction volume during each shaft revolution.

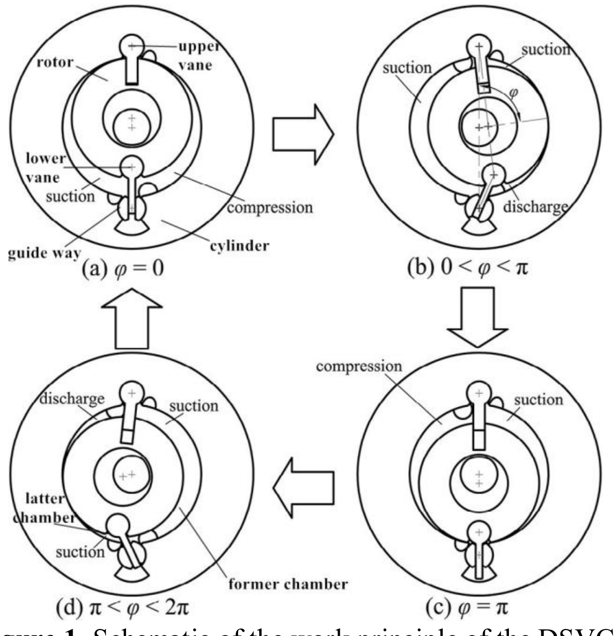

Figure 1. Schematic of the work principle of the DSVC.

As shown in figure 1, in the DSVC, the crescent chamber trapped within the cylinder and rotor is divided by the two vanes into two work chambers, namely the former work chamber and the latter work chamber. Each work chamber is connected with a suction valve assembly and a discharge valve assembly. During the operation of the DSVC, the eccentric shaft drives the rotor such that the tangent point moves along the inner wall of the cylinder. As the rotor rotates, the upper vane swings around the center of the vane joint on the cylinder. At the same time, the lower vane swings around the center of the vane joint on the rotor and slides in the guide way. Due to the motion of the components, the volume of each chamber varies periodically, which results in the suction, compression and discharge of the work fluid.

As shown in figure 1, where the rotation of the shaft is in the clockwise direction and φ is defined as the shaft angle, for an element volume of each work chamber, one and half shaft revolutions ($\varphi = 0 \sim 3\pi$) are needed to complete the whole work cycle including the suction, compression and discharge processes. For the former work chamber, the suction process starts from a rotor angle of $\varphi = 0$ and the element volume gradually increases until the rotor angle nears $\varphi = 3\pi/2$ where it reaches its maximum value. Then, the element volume gradually decreases until the rotor angle of $\varphi = 3\pi$, which results in the compression and discharge processes. For the latter work chamber, there is an angle difference of π for the work cycle of the element volume, which begins at $\varphi = \pi$ and ends at $\varphi = 4\pi$. Figure 3 shows the variations in the element volume and pressure for the former and latter work chambers during a work cycle.

As shown in figure 1, in the shaft angle range of $0 \sim 2\pi$, the volume of work fluids that enters into the former work chamber is equal to the maximum chamber volume V_{max}. Similarly, in the shaft angle range of $\pi \sim 3\pi$, the volume of work fluids that enters into the latter work chamber is also equal to V_{max}. Because the chamber volumes vary periodically, the effective suction volume of the compressor during a shaft revolution can be considered to be equal to $2V_{max}$, which is greater than the volume of the crescent chamber trapped by the rotor and cylinder walls. This means that the effective suction volume of the DSVC is greater than that of an SVC with the same dimension of rotor and cylinder. According to the calculated results of Yang et al[4], for given dimensions of rotor and cylinder in the literature, the suction volume of the SVC is 17.59 cm^3 while the suction volume of the DSVC is 28.77 cm^3, or about 1.6 times that of the SVC.

3. Leakage model

Which is similar to other types of rotary compressors, the radial clearance exits the virtual tangent line between the surfaces of rotor and cylinder. The working fluid therefore leaks from the high pressure chamber to the lower pressure chamber due to the pressure difference. As shown in figure 1, during the shaft angle of $0 \sim \pi$ in a shaft revolution, the former chamber is divided into a suction chamber and a compression chamber and the latter chamber is undivided. Due to the pressure difference, leakage occurs at the radial clearance of the former chamber and no radial clearance leakage exists in the latter chamber. Similarly, during the shaft angle of $\pi \sim 2\pi$ in a shaft revolution, leakage occurs at the radial clearance of the latter chamber and no radial clearance leakage exists in the former chamber. In order to simplify the calculation of leakage loss, during the operation of DSVC, the radial clearance is assumed to be constant which is defined as an assembly value.

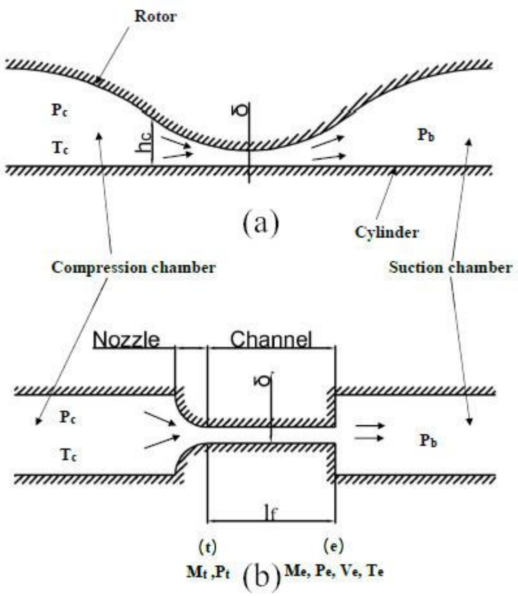

Figure 2. Modeling of the radial leakage channel (a) Leakage channel in the cylinder, (b) Modelling of the leakage channel.

Since the geometric characteristics of the radial channel in the DSVC is quietly similar to that in the rolling piston compressor, the radial leakage flow model proposed by Yanagisawa and Shimizu[6] can be applied in this work. Figure 2(a) shows the schematic of the leakage channel between the inner surface of cylinder and the outer surface. Considering the section change of the leakage channel and the viscous friction of the leakage fluid, the radial leakage channel can be modeled as the flow channel as shown in figure 2(b) which consists of a compression chamber, a convergent nozzle, a straight channel with viscous resistance and an suction chamber. The rectangular section of the straight channel remains unchanged and the height of straight channel is equal to the radial clearance, δ. Since the angular velocity of the rotor is negligible relative to the sound velocity, it can be assumed that the two walls of the channel are stationary. For the friction loss in the model channel is equivalent to that in the practical channel, the length l_f satisfies equation[6] (1):

$$l_f = \frac{2\pi\delta R_c}{[e(1-\chi^2)^{(\frac{1}{2})}]} \tag{1}$$

where $\chi = 1 - \frac{\delta}{e}$.

The mass flow in the channel is calculated as an adiabatic flow with fluid friction. If the fluid velocity at the exit of the channel equals the sound velocity at the entrance of the channel, the Mach number M_t, satisfies the equation[6] (2):

$$\lambda\frac{l_f}{2\delta} = \frac{1-M_t^2}{\kappa M_t^2} + \frac{\kappa+1}{2\kappa} + In\frac{(\kappa+1)M_t^2}{2+(\kappa+1)M_t^2} \tag{2}$$

In the formula, the friction coefficient of the two-dimensional channel, λ, is given by the equation as a function of Reynolds number, and Re is expressed by the equation (3):

$$\lambda = \begin{cases} \dfrac{96}{\mathrm{Re}} & (\mathrm{Re} \le 3560) \\ \dfrac{0.3164}{\mathrm{Re}^{0.25}} & (\mathrm{Re} > 3560) \end{cases} \tag{3}$$

$$\mathrm{Re} = \frac{2q_m}{\mu w} \tag{4}$$

Since the viscosity of refrigerant 134a used as working fluid does not change much in the range of pressure and temperature used in air conditioning, we can use the average viscosity of refrigerant 134a in the formula.

From the equation[6] (2), the pressure ratios of P_t/P_e at the channel and P_c/P_t at the convergent nozzle are given as follows:

$$\frac{P_t}{P_e} = \frac{1}{M_t}\left(\frac{\kappa+1}{2+(\kappa-1)M_t^2}\right)^{\frac{1}{2}} \tag{5}$$

$$\frac{P_c}{P_t} = \left(1+\frac{\kappa-1}{2}M_t^2\right)^{\frac{\kappa}{\kappa-1}} \tag{6}$$

$$\xi = \frac{P_c}{P_t}\frac{P_t}{P_e} \tag{7}$$

If the total pressure ratio, ξ, is less than the given pressure ratio, P_c/P_b , the flow between the upper and lower chambers will be blocked.

When the flow chokes, the Mach number, M_e, at the channel exit is equal to unity and the exit pressure is equal to P_c/ξ . Then the temperature, the velocity, and the mass flow rate at the exit of the channel can be calculated as:

$$T_e = T_c \Big/ \left[1+(\kappa-1)\frac{M_e^2}{2}\right] \tag{8}$$

$$V_e = M_e\left(\kappa R_g T_e\right)^{\frac{1}{2}} \tag{9}$$

$$q_m = \delta w P_e V_e / \left(R_g T_e\right) \tag{10}$$

When the flow is not chocked, we start by assuming the Mach number at the inlet of the channel. Using the assumed value, the virtual critical channel length is calculated to correspond to l_f^* in the equation[6] (10). The critical pressure ratio and the nozzle pressure ratio can be calculated by equations respectively. The Mach number at the exit of the channel is obtained by the inverse equation.

$$\frac{l_f^* - l_f}{2\delta} = \frac{1-M_e^2}{\kappa M_e^2} + \frac{\kappa+1}{2\kappa} + In\frac{(\kappa+1)M_e^2}{2+(\kappa+1)M_e^2} \tag{11}$$

Substituting M_e for M_t in Equation, the critical pressure ratio can be calculated. The final pressure ratio is obtained from the following equation.

$$\frac{P_c}{P_e} = \frac{P_c}{P_t}\frac{P_t}{P_*}\frac{P_e}{P_*} \tag{12}$$

If is equal to the given pressure ratio, the values of and are valid and T_e, V_e and q_m can be calculated from equations (8) ~ (10), respectively.

Table 1. Operating specifications and main dimensions of the DSVC.

Operational speed	3000 rpm
Suction pressure, P_s	0.3 MPa
Discharge pressure, P_d	1.57 MPa
Working fluid	R134a
Cylinder radius, R_{cy}	30 mm
Rotor radius, R_r	25 mm
Rotor axial length, w	26 mm

4. Results and discussion

In this work, the calculations of the leakage flow rate through the radial clearance in the DSVC caused by pressure difference have been carried out by using the established model in section 3. The operating specifications and the main dimensions of the DSVC prototype are given in Table 1. During the simulation, the pressure variation in each of chambers in the DSVC is calculated by using the polytropic process equation as follows,

$$P_s V_{\max}^n = P_c V_c^n \tag{13}$$

where V_c is the volume of the compression chamber, n is polytropic exponent, which is assumed to be a constant value of 1.2.

On the basis of pressure calculation of each chamber, the variation of pressure difference at the radial clearance can be obtained for a shaft revolution. As a comparison, the pressure calculation was also implemented on the SVC which has the same geometric parameters and operating specifications as shown in Table 1. Figure 3 shows the pressure difference, ΔP, of suction chamber and discharge chamber in the DSVC and SVC during a shaft revolution. The solid line represents the theoretical differential pressure of the DSVC while the broken line shows the theoretical differential pressure of the SVC. For the DSVC, during the shaft angle range of the 0~ 2π, the curve of pressure difference is divided into two segments which corresponds respectively to the former chamber and the latter chamber. It can be observed that the pressure difference for the former chamber is similar to that of the latter chamber due to similar work processes of suction and compression for the two chambers. Unlike the DSVC, during the shaft angle range of the 0~ 2π, the curve of pressure difference of the SVC is continuous for the SVC just has one work chamber. We can find that the pressure difference of the DSVC is much higher than the differential pressure of SVC in the shaft angle range of 0~ π. And by integral of the differential pressure of the SVC and DSVC in the shaft angle range of 0~2 π, the average differential pressure of the SVC is 0.65MPa, while the average differential pressure of the DSVC is 1.06MPa, or about 1.63 times that of the SVC.

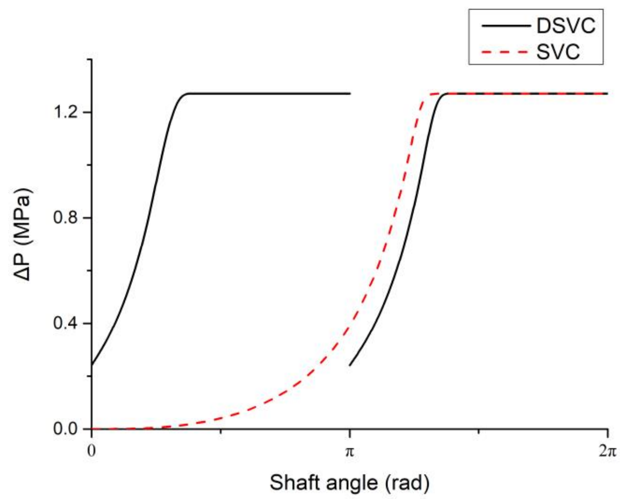

Figure 3. Variations of pressure difference between the suction chamber and discharge chamber in both the SVC and DSVC

Figure 4 shows the simulated resulted of mass flow rate of radial leakage under the radial clearance values of 20μm, 40μm and 60μm respectively. It can be found that for a given clearance value, the mass leakage flow rate through the radial clearance is in direct proportion to the pressure difference and its varying trend is quite similar to that of differential pressure. It is observed that the leakage caused by pressure difference is very sensitive to the change of the radial clearance, and increases rapidly with the raise of the clearance. So it is necessary to reduce the radial clearance value to promote the efficiency of the compressor.

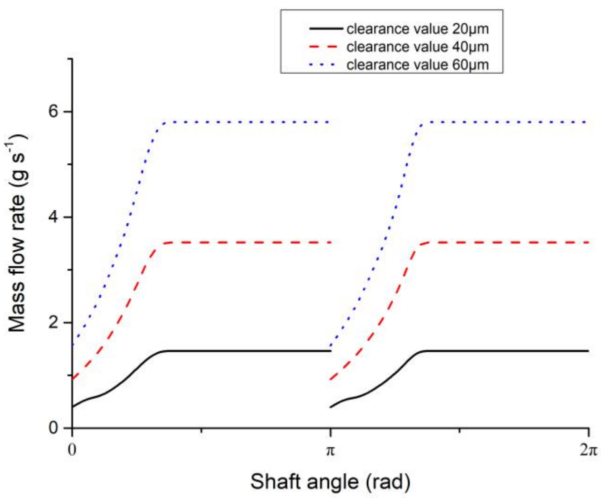

Figure 4. Instantaneous leakage mass flow rates via different radial clearance values in the DSVC

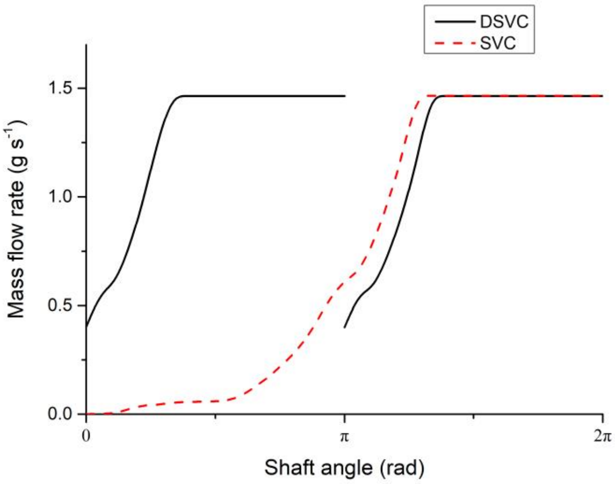

Figure 5. Instantaneous leakage mass flow rates of the SVC and DSVC

Referring to the above discussion, during a shaft revolution, the variation of pressure difference at the radial clearance in the DSVC is quite different with that in the SVC. It can be deduced that the mass flow rate of radial leakage flows for the two types of compressors are different from each other. Figure 5 shows theoretical results of the instantaneous leakage via radial clearance mass flow rate of SVC and DSVC when the radial clearance value is 20μm. The solid line expresses the theoretical leakage mass flow rate of the DSVC while the broken line shows the theoretical leakage mass flow rate of the SVC. For the SVC and DSVC have the same largest differential pressure, the largest mass flow rate of SVC is equal to DSVC. The instantaneous leakage through the radial clearance changes similarly with the change of the differential pressure of compression chamber and suction chamber, *ΔP*, in the both SVC and DSVC. By integration of the instantaneous leakage via radial clearance mass flow rate of the SVC and DSVC in the shaft angle of 0~2 *π*, we can find that the average mass flow rate of DSVC is 1.253$g{\cdot}s^{-1}$, while the average mass flow rate of the SVC is 0.731 $g{\cdot}s^{-1}$. So the DSVC has larger leakage loss via radial clearance than SVC. By the way, the axial leakage loss has also been calculated. And the calculation is based on the model which was proposed by Yanagisawa T and Shimizu T[7]. The average mass flow rate of axial leakage loss of SVC is 0.089 $g{\cdot}s^{-1}$ and that of DSVC is 0.082 $g{\cdot}s^{-1}$. Compared to the leakage loss via the radial clearance, the axial leakage loss can be neglectable.

According to the discussion in section 2, under the same dimensions of rotor and cylinder, the work capacity of the DSVC is much larger than that of the SVC. In such case, it is necessary to compare the volumetric leakage flow rates between the DSVC and the SVC to discuss the volumetric efficiencies of the two compressors. Under the same operating conditions and dimensions as defined in Table 1, when the speed is 3000rpm and the radial clearance value is 20μm, the average volumetric leakage flow rate through radial clearance of the SVC is 0.201m^3/h, while the average leakage flow rate through the radial clearance of the DSVC is 0.344m^3/h, or about 1.71 times of that of the SVC. When the suction volume of the SVC is 17.59 cm^3, while the suction volume of the DSVC is 28.77cm^3, or about 1.6 times that of the SVC. In such case, the relative volumetric leakage loss (the ratio of volumetric leakage loss through radial clearance to the theoretical volumetric flow rate of the compressor) is 6.35% for the SVC, while the relative volumetric leakage loss for the DSVC is 6.63%, which is almost equal to that of the SVC. This indicates that compared to the SVC, the effects of the structural changes of the DSVC on the volumetric efficiency can be neglected.

IOP Conf. Series: Materials Science and Engineering **604** (2019) 012071 doi:10.1088/1757-899X/604/1/012071

Meanwhile, considering the larger work capacity and higher mechanical performance of the DSVC, the DSVC can be considered a better choice for modern electric vehicle air conditioning system.

5. Conclusion

In this paper, the leakage through the radial clearance in the DSVC with different radial clearances has been investigated. On the basis of established model, leakage flow through the radial clearance the DSVC caused by pressure difference were calculated respectively for different values of radial clearances. A comparison was made on the radial leakage losses for the DSVC and SVC. The following conclusions have been obtained from the results:

The influence of the refrigerant leakage caused by pressure difference on the total leakage increases with the increase of the value of the radial clearance. Therefore, the influence of clearance value and pressure difference on the leakage through the radial clearance in the DSVC must be both considered in order to calculate the leakage accurately.

Under the same operating conditions and dimensions, due to the changes of structural design, during a shaft revolution, the average pressure difference at the radial clearance in the DSVC is higher than that of the SVC. As a result, the average mass leakage loss of the DSVC is 1.71 times of that of the SVC. However, the theoretical suction volume of the DSVC is 1.6 times that of the SVC. In this way, the relative volumetric leakage loss is almost equal to that of the SVC, which indicates that compared to the SVC, the effects of the structural changes of the DSVC on the volumetric efficiency is neglectable. Therefore, considering the larger work capacity and higher mechanical performance of the DSVC, it can be suggested that the DSVC is a better choice for modern electric vehicle air conditioning system.

References

[1] Wang Z, Wei M, Peng F, Liu H, Guo C, Tian G. Experimental evaluation of an integrated electric vehicle AC/HP system operating with R134a and R407C. *Applied Thermal Engineering*. 2016;100:1179-88.

[2] Qi Z. Advances on air conditioning and heat pump system in electric vehicles – A review. *Renewable & Sustainable Energy Reviews*. 2014;38(5):754-64.

[3] Akazawa, T., Kawahara, S., Abe, Y., Makino, M., Itou, D., 1996. Development of a high performance scroll compressor for automotive air conditioners. In: Proceedings of Purdue Compressor Technology Conference, pp. 311e316.

[4] Yang X, Dong C, Qu Z. Design and dynamic analysis of a novel double-swing vane compressor for electric vehicle air conditioning systems. *International Journal of Refrigeration*. 2017;76:52-62.

[5] Xu H, Qu Z, Xu Y, Sun J. Theoretical study on frictional losses of a novel automotive swing vane compressor. *International Journal of Refrigeration*. 2013;36(3):758-67.

[6] Yanagisawa T, Shimizu T. Leakage losses with a rolling piston type rotary compressor. I. Radial clearance on the rolling piston. *International Journal of Refrigeration.* 1985;8(2):75-84.

[7] Yanagisawa T, Shimizu T. Leakage losses with a rolling piston type rotary compressor. II. Leakage losses through clearances on rolling piston faces. *International Journal of Refrigeration.* 1985;8(2):75-84.