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Development of a Rotary Vane Expander for an R-22 Heat Pump System

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Practical Design Procedure of a Rotary Vane Expander for a Conventional Refrigeration System

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ABSTRACT

In recent years, expanders have been proposed to improve the energy efficiency of systems that are based on vapor compression refrigeration principles. In this study, a rotary vane expander is developed to improve the energy efficiency of an existing R22 heat pump system. One of the main advantages of rotary vane expanders is it requires no valve to control the fluid flow. The expander is developed by converting a commercially available automotive air conditioning 4-vane rotary compressor. A step-by-step procedure of this conversion is presented here. A mathematical model of the rotary vane machine is formulated to provide the required design parameters. It was found that the machine has a maximum volume ratio of 8.3. The expander is to be installed in series with an expansion valve into an existing R-22 heat pump system. The arrangement is expected to allow some degree of self-adjustment in the expansion process during operation, which can be a problem for an expander-only system. A simulation model of an existing R-22 heat pump system is developed to compute the expected performance of the concept. It was found that the expander-valve-in-series arrangement can improve the coefficients of performance (COPs) of the system by 4.3% and 3.5% on the cooling and heating sides, respectively. In comparison, with an expander-only arrangement, the improvements were 6.8% and 5.4% on the cooling and heating sides, respectively. From the parametric study, COP improvements of between 4-8% and 3-7% in cooling and heating, respectively, are expected at various evaporating and condensing temperatures studied. COP improvements are higher with increasing evaporating and condensing temperatures.

1. INTRODUCTION

A traditional vapour compression refrigeration system consists of a compressor, a condenser, an expansion device and an evaporator, as illustrated in Figure 1, where the pressure reduction is typically done by an expansion valve. However, there is a significant throttling loss in such process. Replacing it with a device that offers a near isentropic expansion, such as an expander, can significantly offer a better performance (Subiantoro & Ooi, 2009).

1.1 Heat Pumps

The second law of thermodynamics states that heat is transferred from a higher temperature to a lower temperature, but in a mechanical refrigeration and heat pump systems, heat is transferred from a lower temperature reservoir to a higher temperature reservoir using a mechanical system.

Heat pumps deliver heat to a space by absorbing heat from a cold space and releasing heat to a warmer one. The process is reverse of a refrigeration or an air conditioning system in which heat is removed from a cold space. Heat pumps can be differentiated based on sources they use to perform their function, an air source and a ground or geothermal source. Following these, there are many subsets, e.g. hybrid heat pumps, exhaust air heat pump, water source heat pump, solar assisted heat pump and many more (Grassi, 2018).

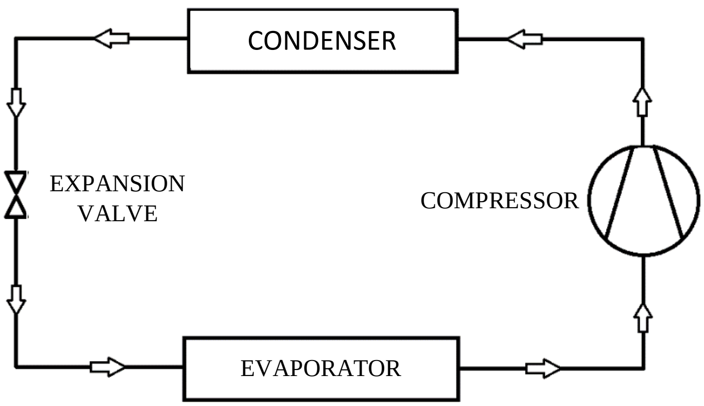

Figure 1: Conventional heat pump system

Air source heat pumps are inexpensive and offer a lower cost of installation but despite its cost-effectiveness, it offers lower CoP values in winter as the ambient air is much cooler than the targeted space and needs a fan system to blow air through its heat exchangers. Due to these reasons, the ground source is preferred to their stable or near constant temperature (5-10°C) which solves the problem of maintaining the required amount of temperature throughout the year despite their higher installation costs (Sarbu & Sebarchievici, 2015). Of course, there are possibilities of water freezing in the system or contribution of contaminants but these can be solved easily by mixing glycol with the water to prevent freezing, and a filter to prevent contaminants (Hundy *et al*., 2016). The thermal conductivity of ground or soil also plays an important factor in determining the efficiency of the heat pump, for example, for a horizontal heat pump type, a helical design proved to be the most efficient in extracting heat from the soil (Ranjeet & Sandeep, 2015). From the literature review, it was found out that many researchers preferred ground source heat pumps to be analyzed and improved (Omer & Mustafa, 2008). Varshney *et al*. (2011) did a field test of air bypass in the vertical stack water source heat pumps in five different buildings and found out that the problem was associated with air bypass, and quantified and solved it. A reversible water-water heat pump was analyzed and modified from an ASHRAE design by Renedo *et al*. (2006) thereby reducing the manufacturing and material cost and presented improvements in performance over previous designs. Removal of fouling or unwanted substances from the heat pump is also an important factor to be considered for optimum performance and efficiency thus Bai *et al*. (2014) did an experimental study on a surface water heat pump and came up with a prediction model which would help in reducing the energy consumption and make the exchanger more efficient.

The current protocols (Velders *et al*., 2007) and Paris agreements (Rogelj, *et al*., 2016) and Global Warming Potential (GWP) (Shine *et al*., 2005) values help in developing various vapour compression systems, in the past, manufacturers did not analyze the damage their systems could do to the environment in the long run thus the old existing systems now need an effective and compatible replacement of refrigerants that do not have high GWP value, one such was performed by Lee *et al*. (2012) where a water source heat pump using R32/R152a mixture was compared with HCFC-22 or R-22. The results showed that the use of the mixture reduced compressor work while the CoP increased. Also, the charge required to in the test rig was reduced to 27% as compared to R-22, thus making it an effective replacement. Other types of heat pump include district heating which is a variation of a ground source heat pump where heat is provided to a location for either space heating or water heating. It employs co-generation of heat by combining heat from the ground and a separate plant that burns fossil fuels or biomass (Lund *et al*., 2010). Hepbasli (2005) performed a thermodynamic analysis of a ground source heat pump which was used for a district heating located in Turkey, the CoP values from the results was found to be in the efficient category.

1.2 Expanders

Due to the negative environmental impacts of fossil fuels and greenhouse gases, steps have been implemented to either switch to alternative sources of energy or to reduce the consumption of fossil fuels or by using less harmful refrigerants that have less global warming potential (GWP) (Shine *et al*., 2005). Recovering the maximum loss from an indispensable device that produces a huge amount of irreversible loss has a potential to enhance the system efficiency (Wang *et al*., 2012). X. Fan (1997) reported that a throttling device induces a loss up to 20 percent of total power input of the compressor. Traditional expansion devices do not have the ability to produce work from the high-pressure drop during expansion thus substituting it with an expander solves the issue. Therefore, the implementation of an expansion device to recuperate the expansion loss could not only be efficient but can also reduce the damage to the environment.

From a financial perspective, Subiantoro & Ooi (2013) calculated the benefits achieved by replacing an expansion valve with an expander with various refrigerants and found reasonable payback periods for certain conditions.
The simplest arrangement is to replace the expansion valve with an expander, resulting in an expander-only system. This, however, may result in issues when the operating conditions change because most expanders do not have a self-adjusting mechanism, unlike a thermostatic expansion valve. One possible solution is to have an expander and an expansion valve arranged in series, as shown in Figure 2.

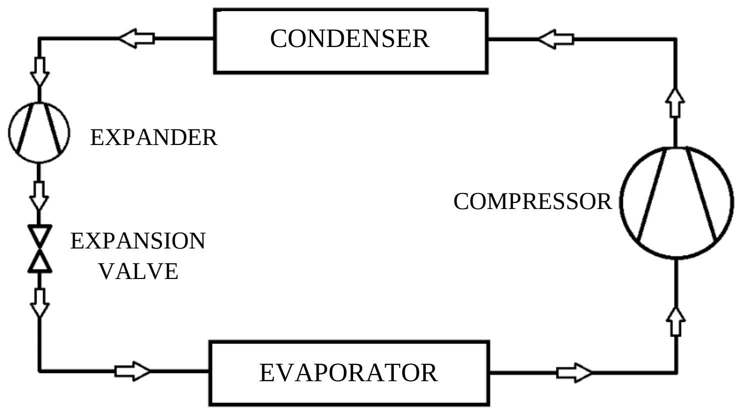

Figure 2: Heat pump system with expander and expansion valve

Many researchers have come up with novel designs to make the expander concept the most efficient. Reciprocating piston design has been considered as it is used widely in internal combustion engines. However, it has certain functional difficulties, significant vibrational and frictional losses. Moreover, because unlike a compressor where the inlet and outlet valves do not function based on the pressure difference, an external mechanism is required with precise timing to open and shut the valve, which is difficult. A rotary mechanism is more compact and has less vibration and noise but tend to have issues with leakages.
As compared to other rotary mechanisms, the rotary vane design has relatively good volumetric efficiencies. Moreover, because of its unique working principles, the machine does not need any valve to control the suction and discharge processes in expander applications. Bassat & Wolgemuth (1972) designed a two stage rotary vane steam expander where the refrigerant expanded in two stages thus increasing the capacity and efficiency, Wang *et al.* (2012) did a simulation study of a novel two stage vane-type expander with R-410A as the refrigerant, it had an oval stator and a concentric rotating cylinder with eight vanes thus creating two chambers to expand the refrigerant. The result showed that a maximum volumetric ratio achieved by the expander was 7.6 with an isentropic efficiency of 55.9% at 2000 rpm. Xia *et al.* (2013) upgraded the expander design and came up with a single stage expander that included five sliding vanes with springs underneath the vanes. The results depicted a volumetric ratio of 7.48 with a maximum efficiency of 32.7% at 1200 rpm.
Other expander machines include that developed by Zhao *et al.* (2014) who did a simulation study of two rolling piston expander which provided large expansion ratios for an R-22 refrigeration system with a theoretical efficiency of 72%. He *et al.* (2009) performed an experimental study using R-410A as a working fluid in a two-phase Pelton-type expander that achieved an isentropic efficiency of 32.8% and an increase in CoP by 5.8%. Wang *et al.* (2011) proposed a single screw expander for compressed air that provided an adiabatic efficiency of 59% at 2850 rpm. Zhang *et al.* (2015) performed an experimental study of a turbo-expander by substituting an expansion valve in a refrigeration system where the maximum isentropic efficiency was 10.4 % at 2000 rpm.
The survey shows that expanders are effective to improve energy efficiencies of refrigeration systems. Each mechanism has its own strengths and weaknesses. A rotary vane expander, in particular, is attractive as compared to other mechanisms because it is compact and does not require a valve or any complex mechanism to control the suction or discharge processes.

2. DEVELOPMENT OF THE EXPANDER PROTOTYPE

The working principles of a 4-vane rotary expander (Figure 3) are as follows. At the beginning of the cycle, high-pressure fluid is suctioned into Volume 1 as its volume increases. When Volume 1 reaches its maximum value and transforms into Volume 2, the suction process stops and the expansion starts. As Volume 2 increases, the fluid pressure decreases. Depending on the location of the discharge ports, this expansion maybe continued in Volume 3. When four vanes are used, the maximum possible chamber volume is reached at Volume 3. When the discharge port is exposed, the expanded fluid will then flow out from the expander. The discharge process continues until the end of the cycle.

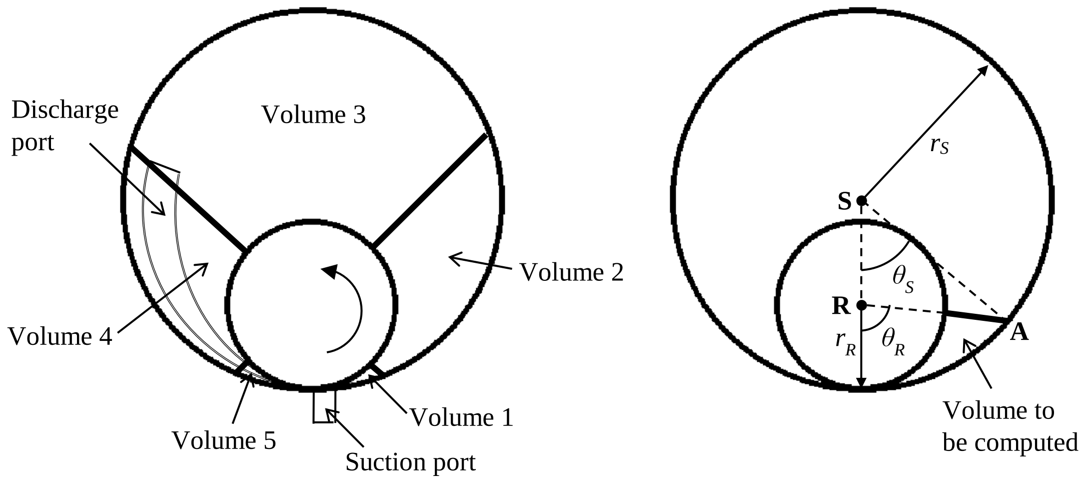

Figure 3: Schematics of Rotary Vane Expander

In this study, a rotary vane expander is developed for an R-22 refrigerant heat pump system by converting a commercial rotary vane compressor, shown in Figure 4. The compressor has a rotor with 4-vanes ($n_v = 4$) and a stator. The main steps of conversion include blocking the existing ports and constructing the new ports. The new suction port has to be located as close as possible to the line contact, as shown in Figure 5, while locating the new discharge ports involves some modelling and calculation, which will be presented in the following discussion.

It is interesting to note that the compressor's stator in Figure 4 does not follow a regular circular design. Therefore, measuring the dimensions were complex. To solve this, the whole stator and the rubber ring's geometries were traced out on a paper, the distances between inner diameters of the rubber ring to the stator at different angles were measured and then, the stator design was replicated on Solidworks, as shown in Figure 5. The chamber length (L) and the rotor radius (r_R) was measured to be 38.5 mm and 28.25 mm respectively, the average stator radius (r_S) was assumed to be 40.5 mm with a ±10% accuracy.

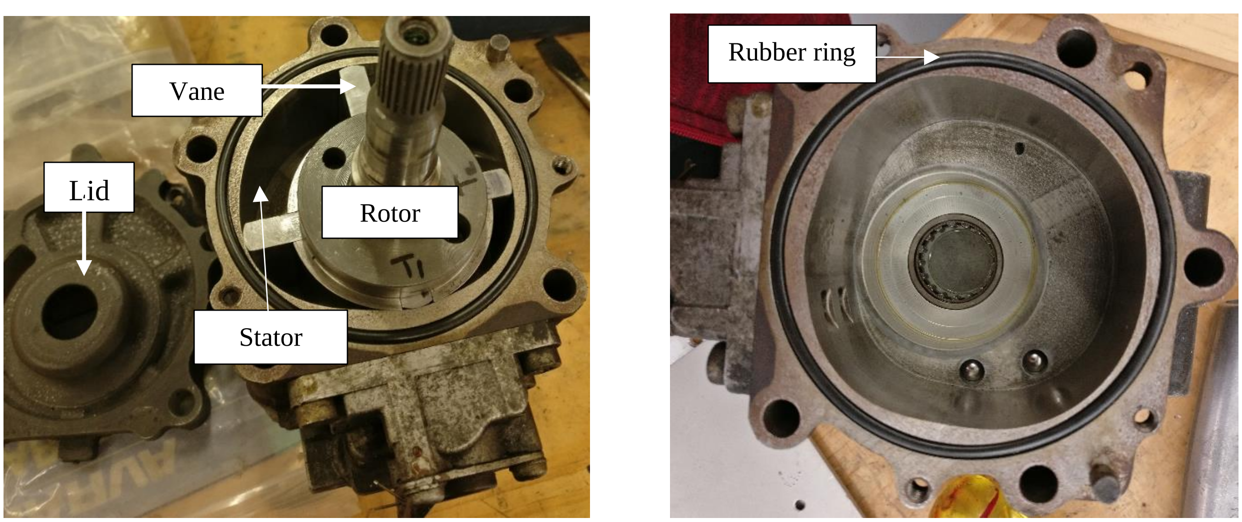

Figure 4: Salvaged compressor top cover removed (Left), rotor removed (Right)

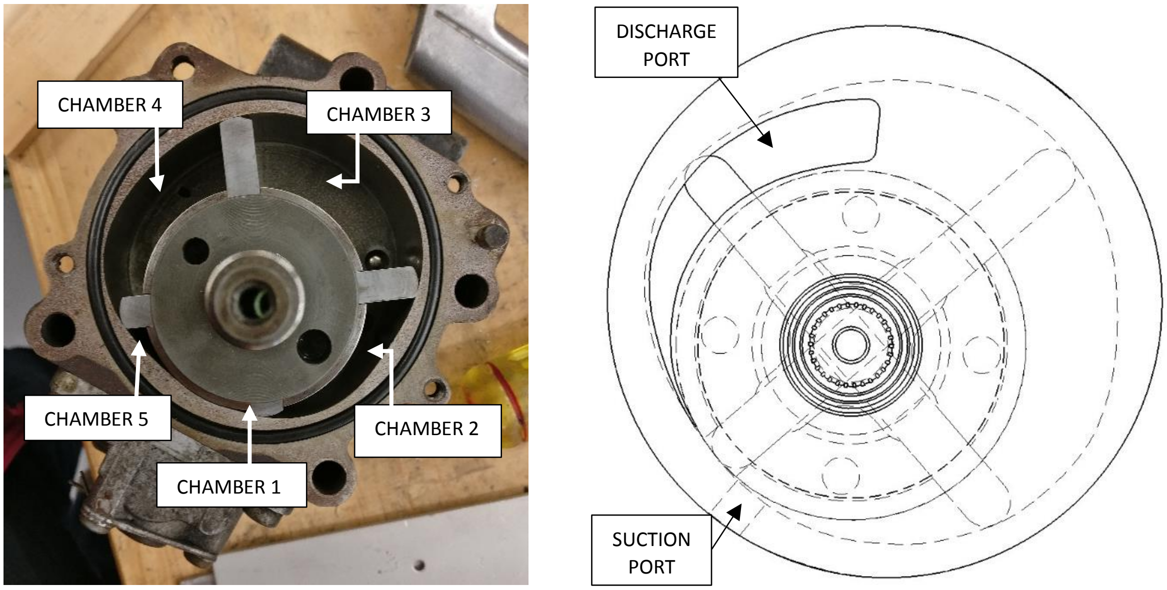

Figure 5: Top view of the expander (left) and Solidworks design (right)

To model the expander, the angle of the rotor (θ_R) was chosen as the independent variable and its value is ranging from $0 \leq \theta_R \leq \frac{2\pi}{n_v}$. From Figure 3, Equations (1) and (2) can be obtained from the triangle ASR.

$$Length\ of\ AR: l_{AR} = -(r_S - r_R)\cos\theta_R + \sqrt{{r_S}^2 - (r_S - r_R)^2 \sin^2\theta_R} \tag{1}$$

$$\cos\theta_S = \frac{(r_S - r_R)^2 + {r_S}^2 - {l_{AR}}^2}{2r_S(r_S - r_R)} \tag{2}$$

The general equation for volume is expressed in Equation (3).

$$V(\theta_R) = \frac{L}{2}\left(\theta_S {r_S}^2 - (r_S - r_R) r_S \sin\theta_S - \theta_R {r_R}^2\right) \tag{3}$$

To compute the individual chamber volumes, Equation (4) can be used.

$$V_i = V\left(\theta_R + (i-1)\frac{2\pi}{n_v}\right) - \sum_0^{i-1} V_j, \text{ where } i = 1,2,\ldots,(n_v + 1) \tag{4}$$

A useful parameter to characterize the maximum expansion that an expander can produce is introduced in Equation (5). It is called 'Maximum Volume Expansion Ratio' (MVER) and is defined as the ratio between the maximum possible chamber volume to the volume of fluid that is drawn in a cycle (i.e. the maximum value of Volume 1).

$$MVER = \frac{V_{i,max}}{V_{1,max}} \tag{5}$$

A Visual Basic Application (VBA) code in Excel was written using a macro which entailed the whole process by the use of Equation (1) to (5) to convert a compressor into an expander by inputting the dimensions of it, an interval of 1 degree was set for a vane till 360 degrees thereby giving the volumes of the refrigerant at every interval. After all the volumes were obtained, the volumes ratios were calculated through which the maximum ratio gave the location of the discharge port. For the machine in the discussion, during suction, up to 6.7 cm^3 of fluid is drawn into Volume 1. It is then expanded up to a maximum volume of 55.1 cm^3 in Volume 3. Therefore, the MVER is 8.3 and can be achieved when the discharge port starts at 225°.

3. SIMULATION OF THE REFRIGERATION SYSTEM

The expander prototype under development is to be installed and tested in a conventional R-22 heat pump system in our laboratory. It has a 63 cm^3 scroll compressor with an average operating speed of 2900 rev/min. It runs in parallel with two water lines for the condenser and the evaporator to exchange heat. Pressure transducers, temperature sensors, and flow meters are installed at various strategic locations to give benchmark experimental data of the refrigeration system. With CoolProp database (Bell, *et al.*, 2014), enthalpies and entropies of the refrigerant at various positions can be obtained. Expansion process in the expansion valve is assumed isenthalpic. The benchmark experimental data of the system is shown in Figure 6. The compressor's isentropic efficiency is around 70%. There are slight pressure drops across the condenser and evaporator. Evaporating temperature is -30°C and condensing temperature is 23°C. Volume ratio across the expansion valve is 37.5. The coefficient of performance (COP) of the systems on the cooling and heating sides are 2.1 and 3.1, respectively. The COP values are calculated using the following formulas.

$$COP_{Benchmark,cooling} = \frac{\dot{Q}_{Evaporator}}{\dot{W}_{Compressor}} \tag{6}$$

$$COP_{Benchmark,heating} = \frac{\dot{Q}_{Condenser}}{\dot{W}_{Compressor}} \tag{7}$$

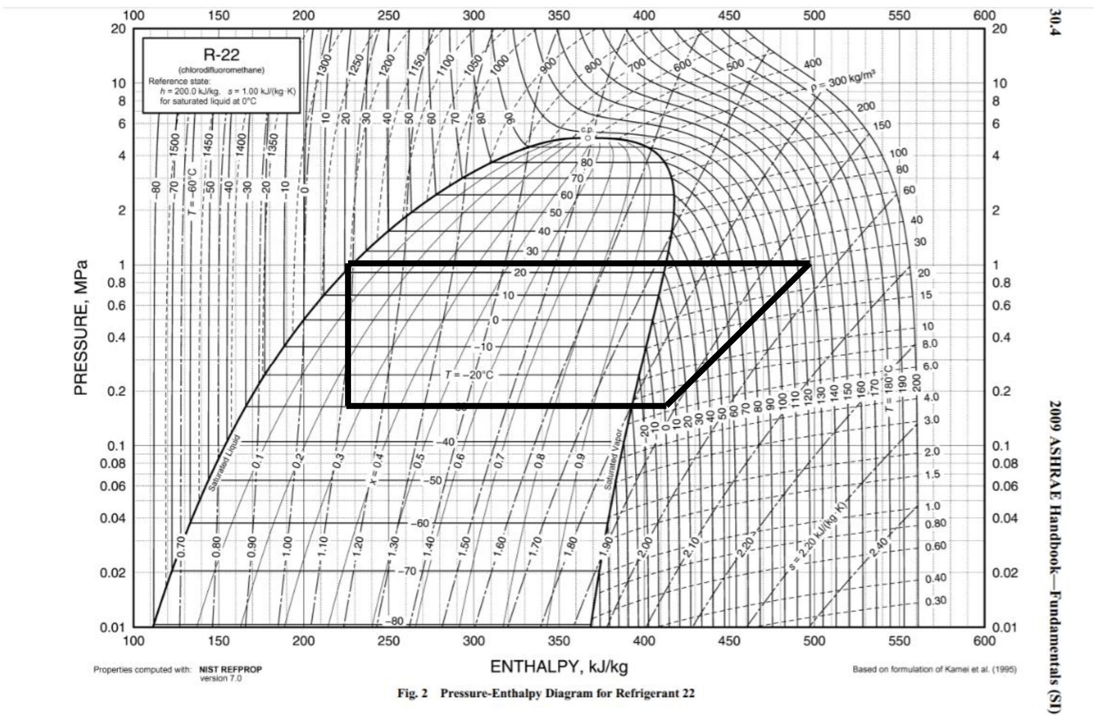

Figure 6: Pressure-enthalpy diagram of the benchmark system (R-22 chart is from ASHRAE)

4. RESULTS AND DISCUSSION

When the expander is installed into the system according to the expander-valve-in-series configuration shown in Figure 2 above, the expansion process of the system is expected to change to that shown in Figure 7. Please note that the pressure-enthalpy diagram in Figure 7 is only for illustration as it is not to scale and is not according to the experimental data in Figure 6. As shown in the diagram, the benchmark (3-5h) is an isenthalpic process, whereas the process 3-4 is of an expander (isentropic) and 4-5 is of an expansion valve (isenthalpic). As mentioned above, the MVER of the expander prototype is only 8.3 while the required volume ratio of the benchmark system is 37.5. Therefore, the expander only partly expands the refrigerant while the rest is done by the expansion valve. For comparison, another case where only an expander is implemented in the system is also considered.

Since the expander is a device that produces work, the COP formula is modified to that in Equations (8) and (10). A small enthalpy reduction during expansion may be resulted by the near-isentropic process in an expander. This is included in Equation (8). With the assumption of near isentropic process taking place in the expander, $\dot{Q}_{Expander}$ was quantified using the Equation (9), whereas the enthalpy values of the expander were obtained by assuming the state of the refrigerant at the saturated liquid line at the expander inlet through which the density was obtained using CoolProp. The refrigerant density at expander outlet was obtained by dividing the inlet density to the expander volume ratio, which changes according to the number of vanes in the expander, and thus the remaining properties of the refrigerant at this point were obtained using CoolProp.

$$COP_{Modified,cooling} = \frac{\dot{Q}_{Evaporator} + \dot{Q}_{Expander}}{\dot{W}_{Compressor} - \dot{W}_{Expander}} \quad (8)$$

where, $\dot{Q}_{Expander} = \dot{m} * (h_{Expander\_isenthalpic} - h_{Expander\_isentropic})$ (9)

$$COP_{Modified,heating} = \frac{\dot{Q}_{Condenser}}{\dot{W}_{Compressor} - \dot{W}_{Expander}} \quad (10)$$

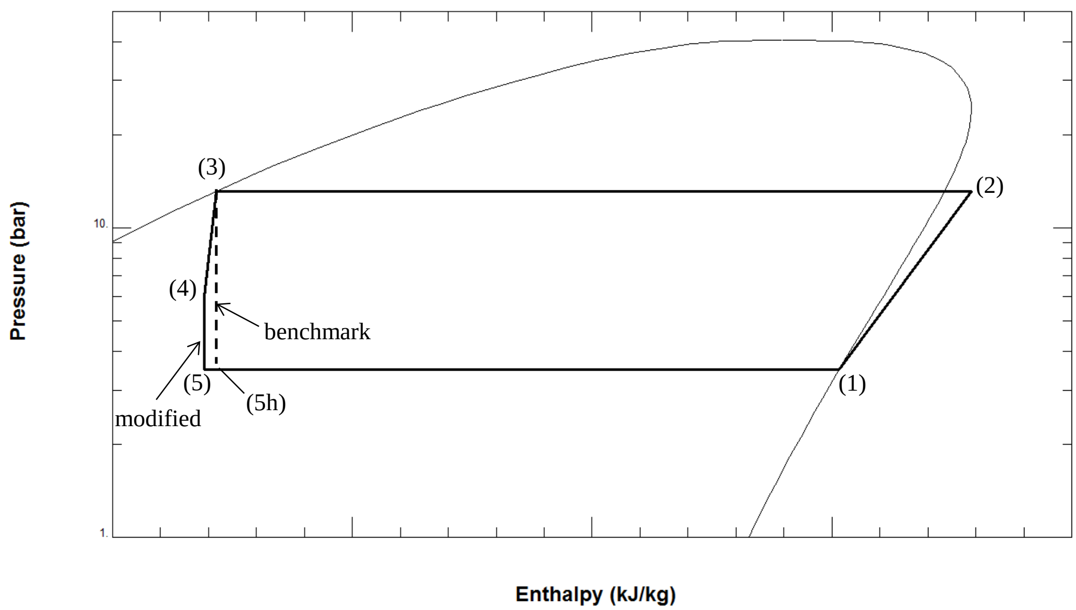

Figure 7: Comparison of the expansion processes in the benchmark and modified systems (not to scale)

From the calculations, it was found that the COPs of the modified system are expected to be up to 4.3% and 3.5% higher on the cooling and heating side respectively, than the benchmark values. For comparison, with only an expander in the system, an increase of 6.8% and 5.4% on the cooling and on the heating side respectively, can be obtained. The intermediate pressure is calculated using the known output pressure and the volume ratio of the expander thus giving the inlet pressure, and the remaining pressure value is reached by employing an expansion valve in series to the expander. In reality, the refrigerant can undergo over, under or perfect expansion, thus assuming a perfect expansion in the expander is not viable, and thus the implementation of the expansion valve in series was considered. In the future of this paper, an experimental study would be carried out where all the parameters would be considered and compared with the theoretical results, and later would be published.

To further investigate the performance of the expander-valve-in-series arrangement, a parametric study is conducted. The parameters in the investigation are evaporating (-30 to -10°C) and condensing (23 to 43°C) temperatures. The benchmark evaporating and condensing temperatures are -30°C and 23°C, respectively. The cycle is assumed to follow that of an ideal vapour compression conditions. COP improvements obtained with the new arrangement are shown in Figure 8. As observed, COP improvements of between 4-8% and 3-7% in cooling and heating sides, respectively, are expected in the range of temperatures simulated. With higher evaporating or condensing temperatures, the COP improvements are higher too.

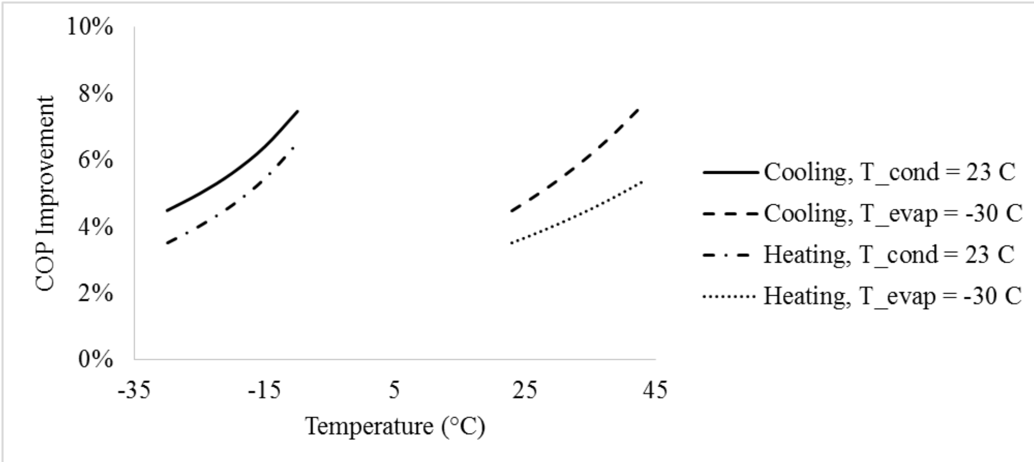

Figure 8: Simulated COP improvements at various evaporating and condensing temperatures

5. CONCLUSION

A four-vane rotary expander was developed from an existing compressor. A major advantage of the rotary vane mechanism is it is relatively simple to convert from a compressor to an expander because no suction valve is needed. The main challenge is in determining the location of the discharge port. A mathematical model of the machine's geometry was formulated for the purpose. The maximum volume expansion ratio was found to be 8.3. The expander is to be installed into an existing heat pump system in series with an expansion valve. This will allow some degree of flow expansion self-adjustment during operation. It is noted that the arrangement will not maximise the available expansion power recuperation potential because a part of the expansion will still be done by the valve.

A simulation model was developed to compute the expected modified system's performance. It was found that the COPs of the modified system are expected to be up to 4.3% and 3.5% higher on the cooling and heating side, respectively, than the benchmark values. For comparison, with only an expander in the system, an increase of 6.8% and 5.4% on the cooling and on the heating side, respectively, can be obtained. From the parametric study, COP improvements of between 4-8% and 3-7% in cooling and heating sides, respectively, are expected in the range of evaporating and condensing temperatures simulated. With higher evaporating or condensing temperatures, the COP improvements are higher too.

In the future, the expander prototype will be installed into the heat pump system and experimented with. Subsequently, since the current system consisted of the banned R-22 refrigerant (Velders *et al.*, 2007), an alternative refrigerant will be used. R-32 is a strong candidate, as shown by Lee *et al.* (2012).

NOMENCLATURE

CoP	Coefficient of Performance	
H	Enthalpy	(J/kg)
L	Length of the chamber	(m)
MVER	Maximum Volume Expansion Ratio	
n_v	Number of vanes	
P	Pressure	(Pa)
$\dot{Q}$	Heat transfer rate	(J)
r_S	Radius of the stator	(m)
r_R	Radius of the rotor	(m)
V	Volume	(m^3)
$\dot{W}$	Power produced or spent	(W)
Greek Symbols		
θ_R	Angle of Rotor	(°)
θ_S	Angle of Stator	(°)

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