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A Fresh Look at Vortex Tubes used as Expansion Device in Vapor Compression Systems

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ABSTRACT

The vortex tube is an intriguing device that separates an incoming high-pressure fluid stream into a two low-pressure streams. Work interaction during the expansion process causes a temperature decrease in one of the two exit streams, while the other one experiences a temperature increase. The overall expansion process in a vortex tube therefore approaches isentropic rather than isenthalpic expansion, and the internal flow separation is achieved without any moving parts, resulting in robust and inexpensive designs. Commercially available vortex tubes are almost exclusively used for spot cooling in industrial applications and use compressed air as the working fluid. In addition, vortex tubes have been gaining lots of attention in air-conditioning and refrigeration research, because of the possibility to replace the expansion valve of vapor compression systems with this low-cost device that can recover expansion work that would otherwise be lost in the isenthalpic throttling process. Most of the work on vortex tubes used for refrigeration has been on numerical studies, and many of them predict very optimistic energy efficiency improvements. However, the few papers available that describe experimental validation of vortex tubes in HVAC&R systems are far less optimistic, which is often caused by the selection of cycle architectures that seem inappropriate for vortex tubes. This paper takes a fresh look at vortex tubes used as the expansion device in refrigeration systems. Vortex tube performance is assessed on a fundamental level for different working fluids, including air and R134a. Suitable vortex tube geometries and operating conditions have been identified and actual work recovery effects have been demonstrated experimentally for both air and R134a. Based on these new findings it is possible to devise novel vortex tube cycles that are able to utilize the demonstrated improvement potentials when applied to vapor compression systems.

1. INTRODUCTION

In vapor compression refrigeration cycle, throttling devices such as capillary tubes, short tube orifices and expansion valves are used as robust and cost-effective solutions for expanding the refrigerant from higher condenser pressure to lower evaporator pressure. However, the physical process during throttling is irreversible, isenthalpic which inflicts a dual penalty on the system in the form of reduction in cooling capacity as well as increase in required compression work. This results in a decrease in COP of the actual vapor compression refrigeration cycle compared to an ideal Carnot refrigeration cycle. In an ideal Carnot refrigeration cycle, the expansion process is isentropic as shown in Figure 1. The expansion process in throttling devices is isenthalpic rather than isentropic as shown in Figure 2, therefore, process 3-4C in Figure 2 would be 3-4R along the constant enthalpy line in an actual system. This reduces the area of cooling capacity by an area of a-d-4R-4C and also increases the work required by an equal amount denoted as Q_{exp} and W_{exp} respectively. Therefore, isenthalpic expansion inflicts a two-fold penalty on COP of actual cycle which is expressed as Equation (1).

$$COP_{act} = \frac{Q_{act} - Q_{exp}}{W_{act} + W_{exp} + W_{sup}} \quad (1)$$

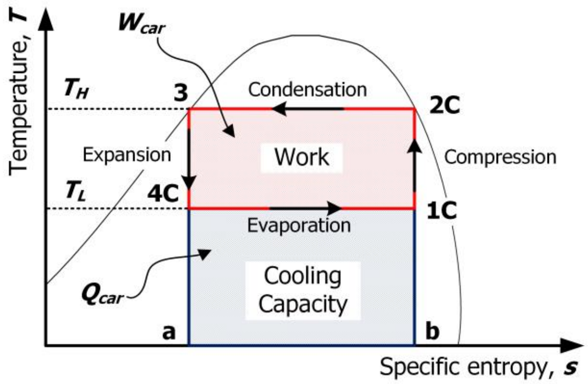

Figure 1: Carnot cycle *T-s* diagram

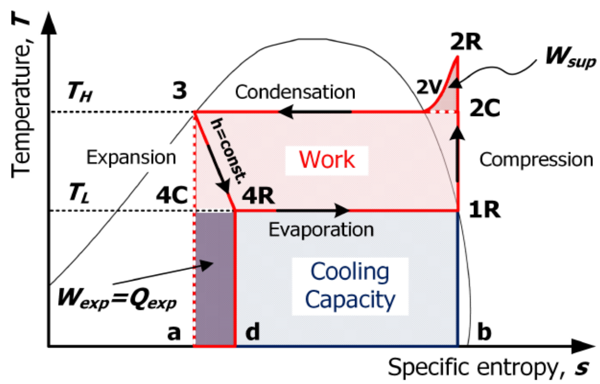

Figure 2: Evans-Perkins cycle *T-s* diagram

COP reduction due to the isenthalpic process in the expansion valve can be reduced by many different methods. One of the simplest is internal heat exchange that reduces the generation of flash gas during expansion by introducing more subcooling at the inlet of the expansion device. However, methods that involve expansion work recovery are known to be more beneficial in terms of cycle efficiency and cooling capacity. A common feature of methods that involve work recovery is that they attempt to utilize the kinetic energy released during the pressure reduction of the fluid as it passes from the high to the low-pressure side instead of dissipating it in a throttling process. Thus, an isentropic expansion process is approached rather than isenthalpic throttling. This increases the cooling capacity, because the specific enthalpy at the evaporator inlet is reduced. Therefore, devices that can approach expansion closer to isentropic process are worth exploring.

The Ranque-Hilsch vortex tube is an unconventional expansion device which was discovered by French physics student George Ranque in 1930. The physical phenomenon was further explained by Hilsch (1947) who named the device as 'vortex tube'. As per the available literatures, it is found that vortex tube has the potential to perform better than isenthalpic expansion. The device also possesses the advantage of being robust and cost-effective as it does not have any moving parts. However, a thorough investigation to measure the performance of existing vortex tubes operating with refrigerants is still due. Therefore, it was the aim of this work to experimentally and numerically investigate the capability of commercially available vortex tubes to improve the COP of the actual refrigeration cycle through incorporation of the device in the system. Special emphasis was given to identify the areas of limitation of operation with two phase refrigerant flow. Once the operating parameters were investigated and clearer understanding of the vortex tube operation was obtained, focus was shifted to devise novel vortex tube cycles suitable for heating and cooling applications.

2. LITERATURE REVIEW

2.1 Expansion work recovery with different devices

Devices such as the refrigerant expander utilize the kinetic energy released during pressure reduction of the fluid as it passes from the high to the low-pressure side instead of dissipating it in a throttling process. Thus, an isentropic expansion process is approached rather than isenthalpic throttling. This increases the cooling capacity as the specific enthalpy at the evaporator inlet is reduced. Simultaneously, the extracted work rate can be used to reduce the power required by the compressor. Therefore, the COP of the system increases for two reasons.

Refrigerant expanders are centrifugal as well as positive displacement type including scroll, rotary vane, rolling piston and free piston devices that aim to achieve isentropic expansion. Many of these designs are built as turbo-expanders in which the expander unit shares the same drive shaft as the compressor as shown in Figure 3. Robinson and Groll (1998) numerically investigated the contribution of different components in producing irreversibility in vapor compression refrigeration cycle using carbon dioxide as working fluid. They found that, replacing the expansion valve with an expansion work recovery turbine with an isentropic efficiency of 60% reduces the process contribution to total cycle irreversibility by 35%. Despite having some potential to improve cycle performance, these devices, being highly integrated, can be subjected to several operational difficulties. If there is only one compressor in the system, and both the compressor and the expander are of the positive displacement type, the volumetric flow rate through the expander is fixed by its volume displacement rate since the compressor and the expander are operated on the same shaft. Other difficulties can be the heat conduction through the shared housing and shaft which can severely reduce the desired work recovery effect. One more adversity commonly encountered in centrifugal expanders is the possibility of two-phase flow damaging the equipment surfaces by erosion.

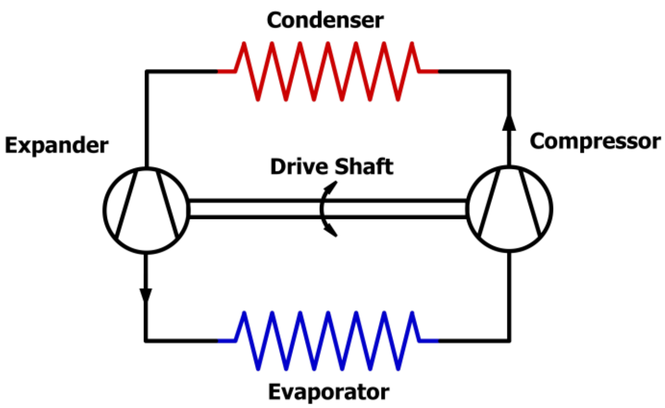

Figure 3: Refrigerant expander used in vapor compression refrigeration cycle

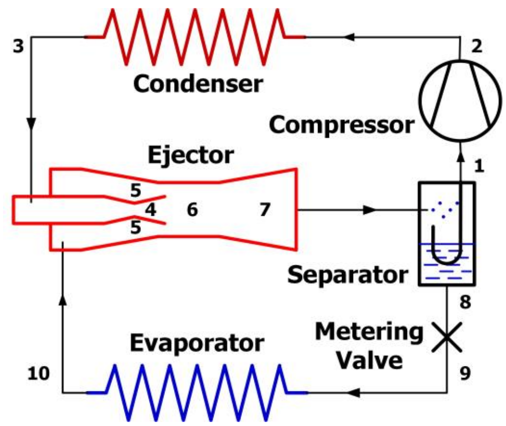

Figure 4: Standard two-phase ejector system layout

An alternative to expanders are two-phase ejectors that also aim to achieve isentropic instead of isenthalpic expansion, thereby recovering throttling losses. A schematic of a typical two-phase ejector cycle is shown in Figure 4. In reality however, the expansion process might occur too rapidly for the two-phase mixture to maintain hydrodynamic and thermodynamic equilibrium. Consequently, these metastability effects might cause a delayed flashing of the flow which could potentially influence the performance of the ejector as investigated by Lawrence and Elbel (2012).

2.2 Vortex tube as an expansion device

The vortex tube, shown in Figure 5 can act to separate and incoming flow of single-phase high pressure fluid flow into two low pressure streams of different temperatures. Temperature of the fluid coming from one of the low pressure exits can rise above that of incoming stream. While on the other low pressure exit, the fluid temperature becomes lower than that of the incoming stream. This pressure reduction and flow separation is achieved without any moving parts, leading to a significant advantage over other devices for being robust and inexpensive.

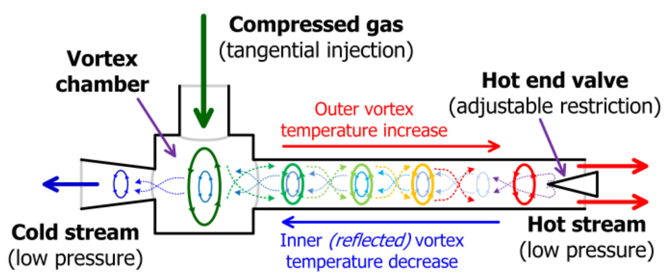

Figure 5: Vortex tube schematic diagram

Hilsch (1947) provided an explanation to the vortex tube phenomenon which is widely accepted until today. As per that explanation, fluid is flown tangentially into the vortex chamber by using tangential inlets. The swirl generator that is used in newer designs provides tangential inlet vanes that also serve the same purpose. The fluid entering tangentially through the swirl generator moves with a screw-like motion along the wall of the vortex tube. The centrifugal force and the internal friction of the gas produce a lower pressure in the axial region. Hence, the fluid can get entrained through the cold end opening as it is closer to the vortex chamber as shown in Figure 5. This is avoided by throttling the flow at the hot end by attaching an adjustable valve. The valve is placed sufficiently away from the vortex chamber so that the gas reaching it loses most of its screw-like motion because of internal friction. By partially closing of the valve, it is possible to force a fraction of the fluid stream to escape through the cold end. This fraction increases with increasing internal pressure and it originates in the region near the axis of the vortex tube. The fluid escaping through the cold end is expanded in the centrifugal field from a region of high pressure near the wall of the vortex tube to a pressure in the region near the axis. The inner fluid flow transfers a considerable part of its kinetic energy by means of internal friction to the peripheral layers. It is worth mentioning here that, in the absence of internal friction, and with a sufficient pressure gradient, the velocity of the fluid would increase during expansion between the circumference and the axis of the vortex tube, acting like a free vortex. The internal friction, however, is particularly effective in this region as the two concentric vortices flowing in opposite directions are in contact with each other. The internal friction causes a flow of energy from the axis to the circumference by trying to establish a constant angular velocity throughout the cross section of the tube resembling a solid body rotation. The inner vortex loses its momentum which is imparted on the outer vortex. Therefore, a decrease in the heat content of inner fluid stream and an increase in the heat content of the outer fluid stream occur if the heat exchange with the surrounding though the wall of the tube is prevented. Hence, two exit streams of different temperatures are achieved.

2.3 Vortex tube studies with single and two phase working fluids

A good amount research work had been done using compressed air with the vortex tube in the early stages of development. Stephan *et al.* (1983) experimented with vortex tube using compressed air with inlet pressure ranging from 150 kPa up to 500 kPa. The valve at the hot exit was adjusted to provide different mass flow distribution between the two exits. The ratio of mass flow rate going out through the cold exit and the incoming mass flow rate was denoted as the cold mass fraction as shown in Equation (2).

$$\dot{y}_c = \frac{\dot{m}_c}{\dot{m}_o} \tag{2}$$

The maximum hot end temperature observed by them was around 75°C at 500 kPa inlet pressure with hot end valve set to a condition such that, cold mass fraction was around 0.9. However, the cold side temperature at that valve setting was 5°C. As the hot end valve was gradually closed, the temperature of the hot exit decreased. However, the temperature at the cold end also dropped with the gradual closure of the hot end valve. The minimum cold exit temperature was observed when cold mass fraction was around 0.3. In this case, the cold and the hot exit fluid temperatures were observed to be -35°C and 15°C respectively. This indicates that, it is not possible to achieve the maximum hot side and minimum cold side temperatures at single setting of the hot end valve.

Aydin *et al*. (2010) stated that, although the vortex tube can be considered to possess certain advantages than other refrigerating or heating devices because of being simple, having no moving parts, using no electricity or chemicals and having long operation time, yet their critical disadvantage is their low thermal efficiency. As per their investigation, the maximum isentropic efficiency of the cooling side was found to be around 45% with cold mass fraction of around 0.5 for air with inlet pressure of 300 kPa and outlets open to atmosphere.

Wu *et al.* (2012) investigated the vortex tube operation with single phase refrigerant. Using R-22 with inlet pressure of 383 kPa (T_{sat} = 8°C), they observed a temperature drop of around 10 °C at the cold outlet from the inlet refrigerant temperature. The temperature drop caused by isenthalpic throttling and isentropic expansion for the same inlet and outlet conditions obtained in the experiment are calculated as around 7 °C and 53 °C, respectively. The observation suggested the fact that the vortex tube could achieve lower temperature at the cold outlet than what is achievable in isenthalpic throttling for similar inlet and outlet conditions.

Collins and Lovelace (1979) investigated the behavior of vortex tube with two-phase propane. They observed that, the temperature separation between the hot and cold outlet was significant when the inlet quality remained above 80% extending in to the superheated region. At 80% quality, for vortex tube inlet pressure of 791 kPa (T_{sat} = 31 ^{o}C) and outlets being open to atmosphere, the hot and cold exit temperatures were found to be 35 °C and 5 °C, respectively. The cold mass fraction in this case was 0.8. They hypothesized that, the possible appearance of the liquid droplet on the inner wall during vortex tube operation at lower qualities diminish the temperature separation, as cooler liquid droplets are evaporated on outer hot walls.

Liu and Jin (2012) analyzed a transcritical CO_2 two-stage compression refrigeration cycle using vortex tube expansion by thermodynamics method. In their thermodynamic model, the gas expanding from gas cooler pressure to evaporation pressure in the vortex tube is assumed to be divided into three fractions: saturated liquid, saturated vapor and superheated gas. The saturated liquid and vapor are mixed again and sent through the evaporator to provide useful cooling effect. The superheated gas is cooled in the heat exchanger and mixed with the gas coming from the evaporator before entering the compressor. In the calculation condition of the study, COP of the vortex tube cycle improved from 2.4% to 16.8% than that of the conventional vapor compression cycle. However, in practical applications, the existence of saturated liquid is expected to diminish the temperature separation. Also, extraction of liquid directly from the vortex tube can be highly challenging from a design perspective. These limitations make this cycle unrealistic despite its theoretical potential.

Another application of vortex tube being used refrigeration cycle is numerically analyzed by Li *et al.* (2000) where the device is used to replace the expansion valve of the transcritical carbon dioxide system. According to the authors, due to Ranque-Hilsch effect, saturated CO_2 liquid at evaporation pressure leaves the cold end of the vortex tube, while superheated CO_2 vapor at evaporation pressure exits at the hot end of the vortex tube. However, again in case of practical application, the existence of saturated CO_2 liquid will diminish the possibility of achieving superheated vapor at the hot end of the vortex tube. Therefore, this takes this cycle to a distant possibility of becoming realistic.

The literature review done so far indicate that, the vortex tube has the potential to obtain better expansion performance than isenthalpic expansion device for refrigerants if it is operated with above 80% quality liquid-vapor mixture, saturated and superheated vapor. However, this capability is yet to be explored by designing proper refrigeration cycle. Therefore, exploring the performance of commercially available vortex tubes with refrigerants is necessary. Based on the performance of vortex tube with refrigerants, realistic refrigeration cycles can be proposed and numerically analyzed to estimate overall gain in system COP and capacity.

3. EXPERIMENTAL STUDY OF VORTEX TUBE

3.1 The vortex tube

The vortex tube used in this research has a measured length of 0.105 m. The vortex tube is rated at 786 kPa. The components of the vortex tube are shown schematically in Figure 6. The vortex tube can produce a number of flow rates as determined by the internal plastic part called the swirl generator. For the vortex tube used in this study, there are three swirl generators available with the vortex tube namely the 2R, 4R and 8R. The number indicates the maximum capacity (expressed in standard cubic feet per minute of air consumption) with that particular generator installed inside the vortex tube.

Figure 6: Vortex tube with its components

3.2 Experimental setup and its components

In order to ensure a constant supply of refrigerant at desired thermodynamic condition at the inlet of the vortex tube, the system needs to be closed. In the closed system, the refrigerant is pressurized and heated to attain desired pressure, temperature and quality before being supplied to the vortex tube inlet. The cold and hot discharge from the vortex tube are collected and cooled properly to be pressurized and heated again to be supplied back into the vortex tube. To pressurize the liquid refrigerant, a reciprocating pump is used. The pressurized liquid refrigerant is then passed through a heat exchanger acting as an evaporator to raise the refrigerant enthalpy. Hot water is supplied to the heat exchanger from a tank with an immersion heater and a submersible pump. Refrigerant at desired temperature and pressure is supplied to the vortex tube. Temperature separation between the hot and the cold sides are measured. Discharged refrigerant from the outlets are collected and supplied in to a condenser and then passed through a receiver and then through a subcooler. The receiver separates any refrigerant vapor present after the condenser and allows only liquid to pass in to the subcooler. Both the condenser and the subcooler are heat exchangers running chilled water supply to remove heat from the refrigerant and liquefy it. The liquid refrigerant is again supplied to the pump inlet. Chilled water supply enters the subcooler first and then passes onto the condenser which makes both of those heat exchangers at a counter flow arrangement with the refrigerant. A bypass line is fitted at the pump outlet connecting it to the pump inlet through copper tubing and a valve. The bypass is used to adjust the flow of refrigerant to the vortex tube. Coriolis-type mass flow meters are used; one at the pump outlet and another at the hot outlet of vortex tube as in these two sections, the refrigerant flow will be single phase. Similar to the previous experimental setup built to testing with air, the temperature readings are obtained from ungrounded T-type immersion thermocouple and piezo-electric transducers are used to read absolute pressures. Sight glasses are installed at various locations of the system to monitor refrigerant flow. A schematic diagram of the experimental setup is provided in Figure 7.

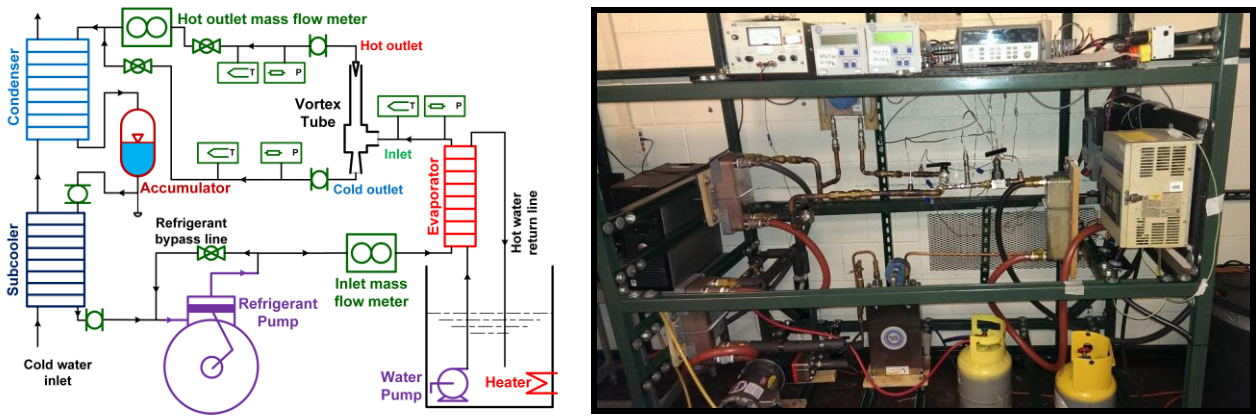

Figure 7: Experimental setup schematic diagram (left) and laboratory facility (right)

3.3 Experimental results

The temperature separation for R134a and air is provided in Figure 8. The results show that, temperature separation exists for refrigerant using the vortex tube which is primarily built for operation with compressed air. The maximum hot side and minimum cold side temperature occurs at around 0.8 and 0.4 cold mass fractions, respectively for both the cases. However, the temperature separation in case of R134a is lower than that of the air. The reason may be attributed to presence of liquid refrigerant droplet during experimentation with R134a as the inlet condition is very close to saturation condition. Temperature separation is also evident with 2R and 8R generators. However, the difference in temperature is smaller compared to results from the 4R generator. The results are shown in Figure 9. The lower temperature separation in 2R generators with refrigerant R134a can be attributed to lower mass flow rates that are insufficient to develop proper flow field for creating a vortex. For the 8R generator, presence of more liquid droplet can diminish the temperature separation.

From the pressure and temperature data collected at the inlet and the outlets of the vortex tube, thermodynamic properties are calculated using Engineering Equation Solver (2013) software. For comparing the expansion process in the vortex tube with an isenthalpic and an isentropic expansion device, actual cold outlet temperature is compared to the calculated isenthalpic and isentropic expansion temperature for the inlet and outlet pressures. The result is shown in Figure 10. The vortex tube is seen to provide lower expansion temperature on the cold side than what is achievable through isenthalpic expansion. This proves its merit to work with refrigerants at high operating pressure to provide better than isenthalpic expansion. For the 4R generator, temperature at the cold end is lower than the isenthalpic expansion temperature by 4 to 8 °C. The isentropic efficiency is calculated in this case by using Equation (3). The maximum isentropic efficiency observed is around 33% observed during experimentation with the 4R generator with the inlet operating close to saturated condition. The efficiency peaks at a range of cold mass fraction between 0.4 and 0.6. Similar trend is also observed with experiments with air. The results are shown in Figure 10 on the right.

$$\eta_{c,isen} = \frac{h_o - h_c}{h_o - h_{c,isen}} \tag{3}$$

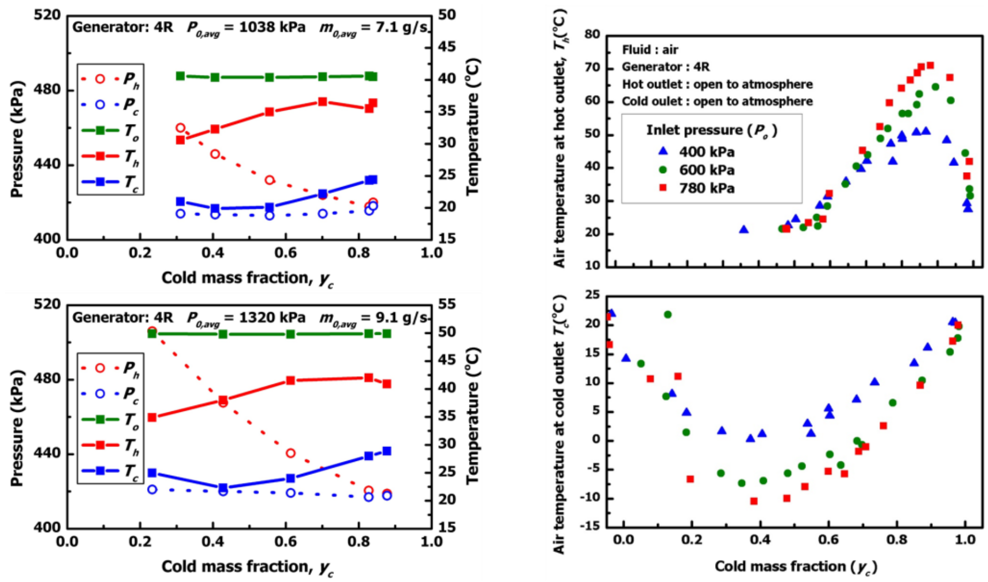

Figure 8: Temperature separation in vortex tube using R134a (top and bottom left) at different cold mass fractions and inlet pressures. Cold and hot side temperature distribution when using air (top and bottom right)

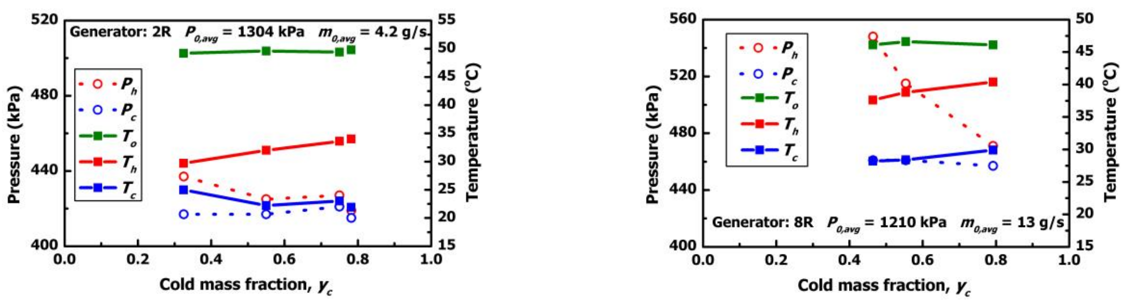

Figure 9: Temperature separation in vortex tube using R134a with 2R generator (left) and 8R generator (right) at different cold mass fractions.

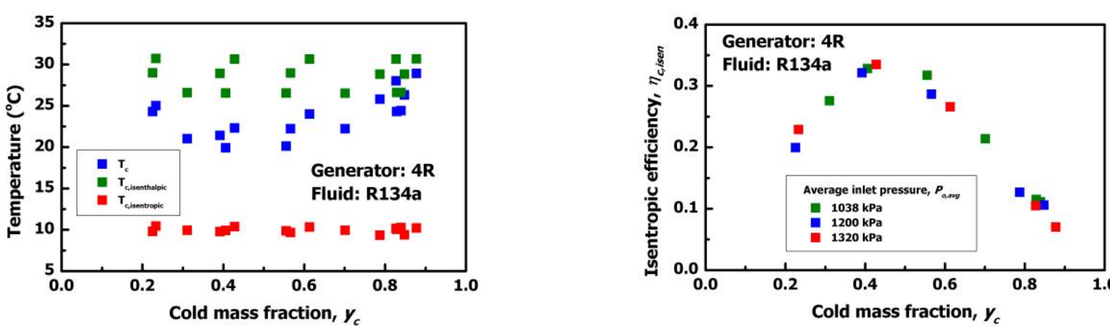

Figure 10: Vortex tube cold side temperature distribution at different cold mass fraction compared to isenthalpic and isentropic temperature using 4R generator (left) and isentropic efficiency of the vortex tube cold side using 4R generator (right)

4. NOVEL VORTEX TUBE CYCLES

The research conducted indicates that vortex tube operation is much more efficient when the fluid streams inside the vortex tube remain single phase. This could be the reason why some of the earlier investigations showed very good potential when investigated numerically, but showed much less (in fact little to no) improvement when evaluated experimentally as shown by Christensen *et al* (2001). The potential problem is illustrated in Figure 11. When the vortex tube is used a replacement for the expansion device in a conventional vapor compression system, it is likely that the expansion process to lower pressures will result in saturated conditions at one or both outlets of the vortex tube. That means that even if temperature separation was achieved inside the vortex tube, the temperature of the warmer fluid that is traveling towards the hot end will be reduced due to vapor being cooled by the two-phase refrigerant that is present. In case both exits are located inside the two-phase region no temperature separation can be achieved, and the potential of utilizing the effect of temperature separation to increase cycle COP diminishes.

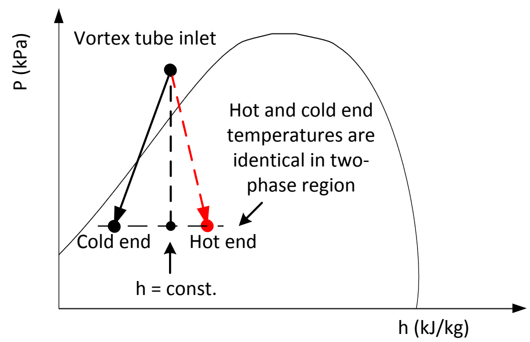

Figure 11: Reduced temperature separation potential of vortex tube for operation in two-phase region

What is needed are novel vortex tube cycles that take into account that vortex tube operation is most efficient when all fluid streams remain single phase. Figure 12 shows a novel vortex tube cooling cycle that was developed from an earlier idea in which partial condensation with subsequent phase separation was used to ensure single-phase operation of the vortex tube. However, it was found that the additional cooling capacity obtained by additional subcooling created by the cold end of the vortex tube was insufficient to overcome the performance loss by incomplete condensation. Therefore, the cycle in Figure 12 achieves complete condensation before the flash gas separated after a first expansion is used to drive the vortex tube. From further expansion of the flash gas in the vortex tube additional subcooling is created which increases the performance of the cycle. Depending on the refrigerant and the operating conditions, COP improvements on the order of 5 to 10% have been obtained with a simplified thermodynamic state point model. Further improvement can be obtained by adding an internal heat exchanger, which in vortex tube cycles gives extra opportunity for the expanding flows inside the vortex tube to remain single-phase.

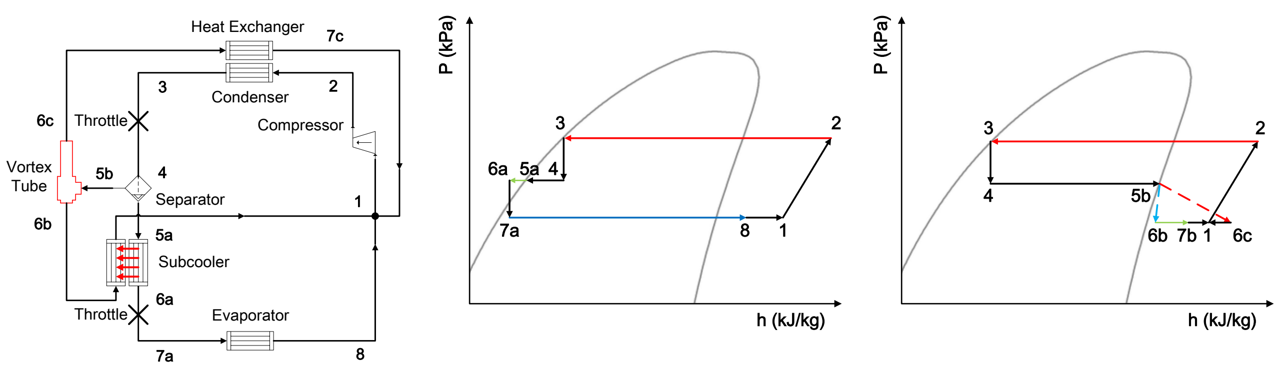

Figure 12: Novel vortex tube cooling cycle that ensures single phase operation inside vortex tube (left) and corresponding pressure-specific enthalpy diagrams (middle and right)

Two additional vortex tube cycles are proposed in Figure 13. The cycle on the left is intended for cooling; COP improvements are achieved by using the hot end of the vortex tube to increase heat rejection in comparison to a baseline system with expansion valve. It should be noted that the improvement mechanism is very different from the cycle shown in Figure 11, where the kinetic energy recovered by the vortex tube is used to increase cycle COP by vortex tube generated subcooling. In order for the vortex tube streams to remain single phase, it is proposed to realize the cycle shown in Figure 13 as a transcritical CO_2 cycle. Due to the increased heat rejection created by the vortex tube, the simplified vortex tube cycle model shows COP improvements on the order of 40% for a transcritical CO_2 cycle operated at ambient temperatures of 45°C.

Figure 13: Novel vortex tube cooling (left) and heating (right) cycles that ensure single phase operation inside vortex tube

The cycle on the right side of Figure 13 is an example of how the vortex tube can be used to improve efficiencies of heating applications. The proposed cycle can be utilized to increase the temperature of a low-grade waste heat source, and might be beneficial in applications such as industrial steam generation or water heating. A liquid pump is used to circulate refrigerant in a round-around heating loop. Low-temperature waste heat is used to vaporize the working fluid in Heat Exchanger 1. The energized stream is fed to drive the vortex tube where the hot side temperature reaches higher levels than the original waste heat source. High temperature heat can then be extracted through Heat Exchanger 2 and be used in applications that benefit from increased temperatures. Efficiencies of the proposed vortex tube cycle are very high, because in today's applications, the desired temperature increase can often only be achieved with electric resistance heat; therefore these new vortex tube cycles offer very attractive options for both heating and cooling applications.

5. CONCLUSIONS

The vortex tube designed and used primarily for air is utilized in this study to investigate the potential to use as a work recovery expansion device for refrigerants. The results show that the vortex tube is capable of producing temperature separation for R134a at higher inlet pressure. The cold side temperature is found to be lower than the isenthalpic expansion temperature which shows the potential to recover work. Although the hot side temperature does not climb above the inlet temperature, the reason might be operating conditions being close to saturation. Currently ongoing modification of the experimental setup will enable to take accurate measurement of the inlet refrigerant vapor quality and even operate at superheated inlet condition that will further enable further insight into the work recovery capabilities of the vortex tube. The very promising results obtained in this study warrant further investigation on the cycle level. For that purpose, a variety of novel cooling and heating cycles have been proposed, with the aim of implementing single-phase vortex tube operation in realistic HVAC&R applications.

NOMENCLATURE

COP	coefficient of performance	
h	specific enthalpy	(kJ/kg)
HVAC&R	heating ventilation air conditioning and refrigeration	
P	pressure	(kPa)
T	temperature	(°C)
y_c	vortex tube cold outlet mass fraction	(-)

Subscript

act	actual
c	vortex tube cold side
exp	related to expansion process
h	vortex tube hot side
o	vortex tube inlet
sat	saturated fluid condition
sup	superheated fluid condition

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