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Performance Testing of a Unitary Split-System Heat Pump with an Energy Recovery Expansion Device

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ABSTRACT

Due to the rising demand of using energy resources more efficiently, the HVAC&R industry is constantly facing the challenge of meeting stricter energy consumption requirements. The goal of this research is to present a study that focuses on developing a device that can improve the efficiency of a residential split-system vapor compression heat pump using R410A as the refrigerant. Currently, the industry standard for the expansion process of a heat pump is to use either an electronic expansion valve (EXV) or thermostatic expansion valve (TXV). Both of these are passive expansion devices that do not recover the potential energy from the high pressure refrigerant. As a result, there is a meaningful potential for efficiency improvements by replacing the passive expansion valve with a work-generating device – the Viper Energy Recovery Expander. The Viper Expander has been designed by Regal Beloit Corporation in conjunction with Purdue University's Ray W. Herrick Laboratories. The expander functions by using a nozzle to convert the pressure energy of the refrigerant at the condenser outlet into a high speed flow. The nozzle is designed to ensure refrigerant phase change and accelerate the flow into the impeller of a micro-turbine. The rotating turbine impeller is connected to an internal generator which generates electrical energy. This electrical energy is used in the system to augment the power into the air conditioner's indoor fan motor. The expander enables the realization of decreased power consumption and increased evaporator cooling capacity. The expander has been implemented into a 5-ton split system heat pump and tested in both heating and cooling modes. Experimental testing has shown that the expander can generate up to 30% of the required fan motor power. Results have led to suggestions for optimizing the design of the device to achieve higher efficiencies.

1. INTRODUCTION

Improving the energy efficiency of thermal systems like heat pumps or refrigeration systems is one of the main ways to compensate for the increasing global energy demand. The U.S. Energy Information Administration reported in 2014 that over 40% of the total energy consumption in the United States was consumed in both residential and commercial buildings. Vapor compression cycles are a main application for providing building heating and cooling. Therefore improving the efficiency of vapor compression cycles provides a means to reduce energy consumption in the space heating and cooling sector. Additionally, by 2017 the required SEER and HSPF for air-cooled split-system heat pumps with a capacity of less than 65,000 Btu/h (19 kW) will be increased from 13.0 to 14.0 and from 7.7 to 8.0, respectively (U.S. Department of Energy, 2015).

Many different sources of losses and inefficiencies reduce the performance of vapor compression cycles. This paper investigates using an energy recovery expansion device to harness the otherwise lost work potential in standard systems with passive expansion devices like thermostatic expansion valves (TXVs) or electronic expansion valves (EXVs). Such an expansion process is called a free or passive expansion process because it does not harvest the potential energy of the high pressure refrigerant at the inlet as the energy is dissipated in the form of heat due to friction.

In the past, trying to harvest the wasted energy potential from the expansion process in terms of useful power has typically been neglected due to the comparatively small quantity available. However several attempts at developing expander devices for various working fluids have been pursued. In 1992, Zhang and Yu stated that two-phase expanders should become commercial products to replace throttling devices. A theoretical investigation was performed to compare the potential performance of an expander with R502, R12, R22 and R717 at condensing temperatures of 303.16K and 323.16K. The best performing refrigerant was R502 and showed an increase in cycle efficiency ranging for 23.7% to 38.1% for an expander compared to a standard throttling device.

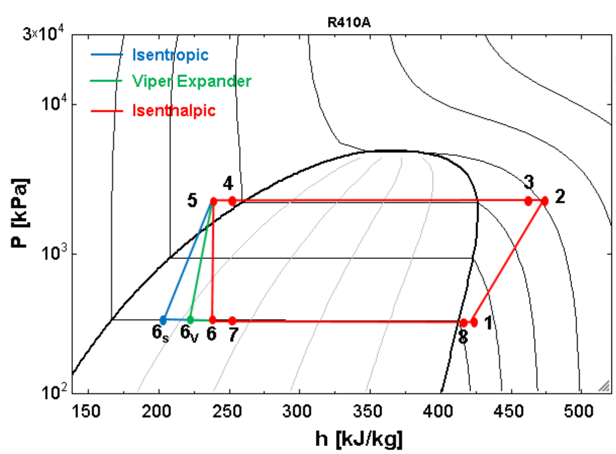
Yang et al., (2009) performed an experimental investigation of installing a double acting rotary vane expander into a transcritical CO_2 cycle. An electrical generator was connected to the shaft in order to consume the recovered power. Tests were conducted at expansion inlet pressures ranging from 7.85 – 8.35 MPa and the power output from the expander ranged from 348 – 379 W. This corresponded to an increase in COP of 11.1 – 14.2%. For this expander the isentropic efficiency was recorded to be between 19.1 – 22.63%.

Another work output expansion device for transcritical CO_2 systems was developed by Baek and Groll (2005). The device was chosen to be a piston cylinder expander and is known as the Expansion Device-With Output Work (ED-WOW). Three different conditions were tested with indoor and outdoor room temperatures of 20°C and 35°C, respectively The increase in COP ranged from 7.07% to 10.46% which includes the increased cooling capacity resulting from the non-isenthalpic expansion process.

The majority of literature on this subject suggests that the CO_2 has been the main area of interest for installing an energy recovery expansion device. This paper attempts to understand the feasibility of using a similar device in an R410A system.

2. THE VIPER EXPANDER

The Viper Expander is a power generating device that has been designed by Regal Beloit Corporation in corporation with Purdue University's Herrick Laboratories. The Viper Expander consists of a nozzle and a micro-turbine that is connected to a generator. The purpose of the Viper Expander is to harvest the normally wasted work potential during the expansion process of a residential split-system heat pump with R410A as the working refrigerant. The nozzle is used to convert the high pressure subcooled R410A into a jet of low pressure, high velocity fluid that will impinge on the impeller. The shaft of the Viper impeller is connected to a generator that will produce electrical energy as the impeller rotates, thus converting the kinetic energy of the refrigerant into electrical energy. The generator outputs a 3-phase alternating current (AC) that is then rectified to direct current (DC). The direct current is then fed into a specially designed fan with an electronically commutated motor (ECM) to supplement the power consumption. ECMs operate with direct current and have many advantages over standard AC motors like higher efficiency and lower cost. This added power will reduced the overall system power consumption. As the expander turns the isenthalpic expansion process into partially isentropic, the cooling capacity will also be increased when the unit is operating in air conditioning mode. Both the reduced power consumption and added cooling capacity could increase the system COP. The log p-h diagram for the standard isenthalpic, Viper Expander and pure isentropic expansion process is shown in Figure 1.



St. Pt	Description
1	Compressor Suction
2	Compressor Discharge
3	Condenser Inlet
4	Condenser Outlet
5	Expansion Device Inlet
6	Passive Expansion Device Outlet
6_V	Viper Expander Outlet
6s	Isentropic Expansion Outlet
7	Evaporator Inlet
8	Evaporator Outlet

Figure 1: Pressure – enthalpy diagram for standard, Viper Expander and isentropic expansion devices.

2.1 Housing and Impeller Designs

The first Viper design presented in this paper is a radially in – axially out housing design. The impeller has 20 blades shaped as sickles with both the leading and trailing edges being pointed tips. The flow enters the blade with an angle of attack of roughly 50° and then exits towards the center of the impeller. The exiting fluid flows into a cone shape around the rotor which turns the refrigerant by 90° and directs it axially outwards. Additionally, there is a top plate that covers the impeller to minimize leakage from the top of the blade. The nozzle is a separate component that screws into the housing in the radial direction. The impeller and housing of the Viper I are shown in Figure 2.

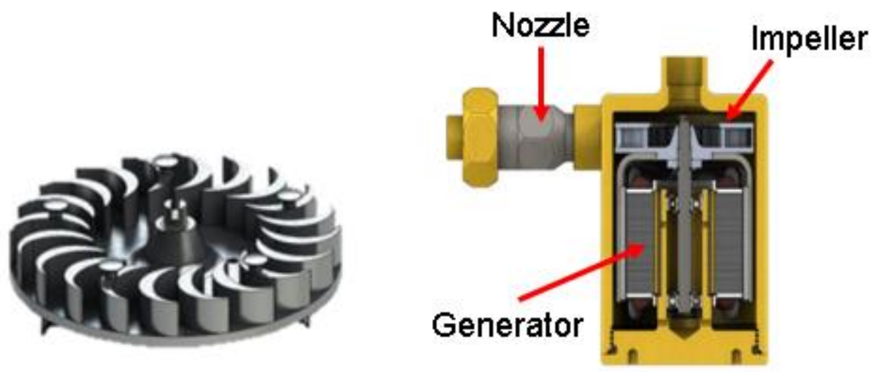


Figure 2: Viper I impeller (left) and housing (right)

The Viper II is a radially in – radially out design that was designed with the purpose of reducing the pressure drop across the housing and impeller to force the pressure drop across the nozzle. A larger pressure drop across the nozzle should result in a higher refrigerant exit velocity and larger power output. The Viper II impeller was designed based on a cross-flow impeller, which allows the flow to pass through the impeller and exit radially. The nozzle inlet port is still radial, but the housing has changed for the outlet port which is now in the radial direction as well. Figure 3 shows the cross sectional view of the impeller (left) and housing (right) of the Viper II.

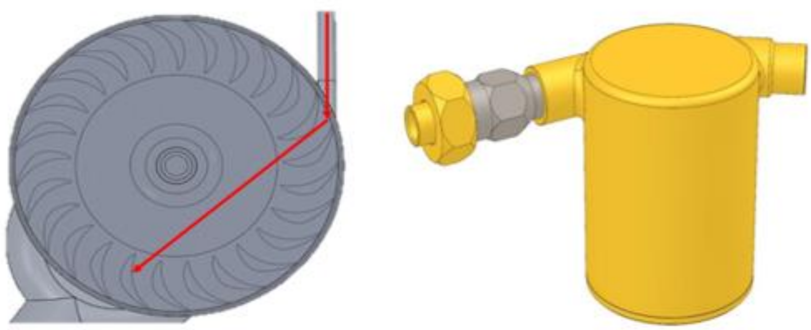


Figure 3: Viper II impeller (left) and housing (right).

2.2 Nozzle Designs

Different nozzle iterations were also designed due to experimental results indicating that the power output is very sensitive to nozzle geometry. The exit diameter was seen to impact the degree of subcooling at the nozzle inlet as well as the power output. The nozzle geometry is a key factor in the Viper design because the larger the pressure drop across the nozzle, the higher the velocity of the refrigerant impinging on the Viper impeller. The power output from the Viper is related to the velocity of the incoming refrigerant by:

$$\dot{W}_{Viper} = \eta_{Viper}\left(\frac{1}{2}\dot{m}V_{exit}^{2}\right) \tag{1}$$

Unfortunately, the exit speed of the nozzle was not able to be experimentally measured due to how the nozzle connects to the housing. There is not enough space to install sensors and other non-intrusive methods yielded to be too costly. In total 17 different nozzles were tested with the expander, however not all nozzles will be presented as they can be classified in 3 main categories shown in Figure 4.

Figure 4: Nozzle types used with the Viper Expander.

The nozzle on the left of Figure 4 has a short converging section at the inlet followed by a long constant diameter diverging section. Since this expansion process is two-phase, the long constant diameter was added to allow the refrigerant to expand into the saturation dome and drop in density. According to the conservation of mass , the velocity of the refrigerant will increase if either the area or density decrease for a constant mass flow rate.

$$V = \frac{\dot{m}}{\rho A} \tag{2}$$

The second type of nozzle is an elliptical contoured nozzle shown in Figure 4 (middle). The purpose of the elliptical nozzle is two-fold. This refrigerant will accelerate through the nozzles due to both the decreasing density and continual decreasing area. The reason this type of nozzle is called an elliptical nozzle is because the contour takes the shape of two different ellipses.

The last type of nozzle tested was a converging-diverging nozzle shown in Figure 4 (right). It is believed the refrigerant might be choked in the straight and elliptical nozzles. For choked flow, the refrigerant reaches a Mach number equal to 1 and will only continue to accelerate with a diverging contour. Therefore a converging-diverging (C-D) nozzle has been designed and implemented. The C-D nozzle was pursued as a theoretical analysis for the two-phase expansion processes predicts a supersonic velocity in order to achieve a target power output of 200 W or 33% of the fan power consumption.

3. EXPERIMENTAL SET-UP

The unit used for testing is an off-the-shelf split-system heat pump manufactured by Carrier Corporation. The system is rated at 5 refrigeration tons (17.6 kW) and has a SEER of 15.3. The unit is designed to operate in both heating and cooling modes and uses R410A as the refrigerant. As this is a split-system unit, the system components are separated into both an outdoor unit and an indoor unit. The units have been set up in psychrometric chambers to simulate respective indoor and outdoor conditions. A schematic of the test unit is shown in Figure 5 (left) and the indoor unit with the Viper Expander installed is shown in Figure 5 (right).

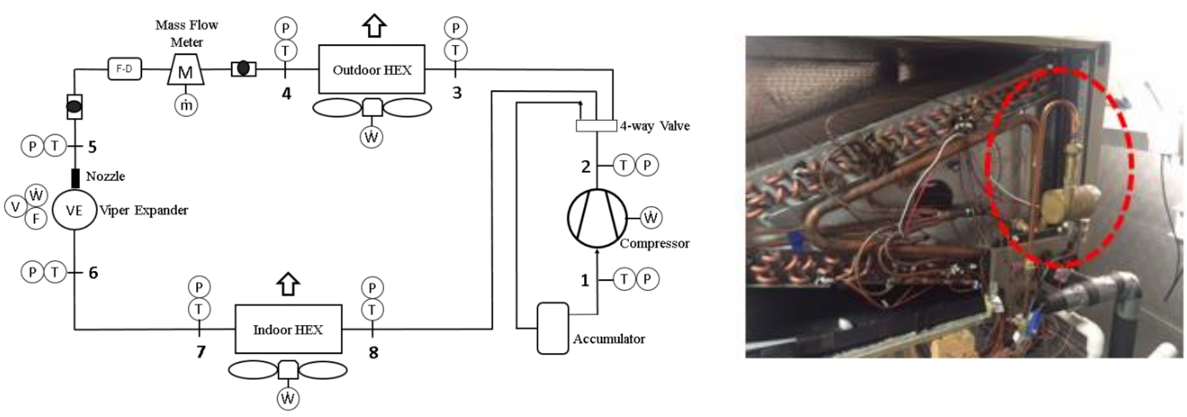


Figure 5: Schematic of test stand (left) and indoor room with Viper Expander installed (right).

The state points of Figure 5 are the same as in Figure 1. Along with the Viper, various measurement instrumentation like pressure transducers and thermocouples are installed for both the refrigerant and air sides. The power, voltage and frequency of the expander are also read recorded. The expander generator output is connected to either the ECM or a variable resistor ranging from 25 - 1000Ω. Varying the resistance of the load showed to impact the rotational speed of the Viper and thus the power output.

Additionally a mass flow meter has been added on the refrigerant lines as well as sight glasses at both the condenser outlet and nozzle inlet. The sight glasses allow for visual confirmation of the refrigerant phase entering the nozzle. Also, a further modification was made with the purpose of trying to maximize the pressure drop across the nozzle. Factory units are typically not designed to maximize the pressure drop across the expansion device and significant pressure drop can occur in the distributor and tubes before the evaporator inlet. It was seen that 170 kPa or 12% of the total pressure drop from condensing pressure to evaporation pressure occurred across the distributor tubes with 3/16 in (4.76 mm) outer diameter. Therefore the distributor and tubes were replaced by tubes with larger diameter of 1/4 in (6.35 mm) outer diameter.

4. EXPERIMENTAL TESTING AND RESULTS

The tests for cooling mode and heating mode were based on the ANSI/AHRI Standard 210/240 specifications. The test matrix is shown in Table 1 and consists of one cooling test (Test A) and three heating mode tests (Test H1, H2 and H3). The Viper Expander was tested at other tests points, but these were the four main operating points tested and thus only the data for tests will be shown. All of these four tests are steady-state tests that in which the unit was run for 30 minutes at steady-state. The average of each parameter was then computed and recorded. The temperature and relative humidity values listed in Table 1 are of the ambient air in each of the rooms.

Table 1: Test Matrix for performance testing

Test Type	Test Name	Indoor Unit Air			Outdoor Unit Air		
		Dry-Bulb		Relative Humidity	Dry-Bulb		Relative Humidity
		°F	°C	%	°F	°C	%
Cooling	**Test A**	80.0	26.7	50.68	95.0	35.0	39.87
Heating	**Test H1**	70.0	21.1	56.46	47.0	8.33	72.70
Heating	**Test H2**	70.0	21.1	56.46	35.0	1.67	81.85
Heating	**Test H3**	70.0	21.1	56.46	17.0	-8.33	69.51

For the cooling more testing, the indoor room had a temperature of 80°F (26.7°C) and the outdoor room had a temperature of 95°F (35°C). For the heating mode tests the indoor temperature is now higher than the outdoor temperature. The indoor air temperature stayed constant at 70°F (21.1°C) and the outdoor temperature ranged from 47 – 17°F for the three operating conditions.

4.1 Cooling Mode Results

In total 5 expander and nozzle combinations will be presented for cooling mode. The results of the Viper I with the following four nozzles at the Test A operating conditions will be presented:

1. 0.090” (2.29 mm) exit diameter by 2.25” (57.15 mm) length – Straight
2. 0.125” (3.18 mm) exit diameter by 3.5” (88.9 mm) length – Straight

3. 0.065” (1.65 mm) exit diameter by 2.25” (57.15 mm) length – Elliptical
4. 0.085” (2.16 mm) exit diameter by 2.25” (57.15 mm) length – Elliptical

The results for the Viper II with only one nozzle will be presented and the nozzle is as follows:

5. 0.110” (2.79 mm) exit diameter by 3.5” (88.9 mm) length – Converging - Diverging

The naming convention used for the nozzles will be the nozzle exit diameter followed by the nozzle type (Straight, Elliptical or C-D). The cooling mode results have been listed in Table 2 and the power output versus the nozzle exit diameter is plotted in Figure 6.

Table 2: Cooling mode Test A Viper power

Viper	Nozzle	Power [W]
I	0.065” – Elliptical	17.0
I	0.085” – Elliptical	27.0
I	0.090” – Strght	27.8
II	0.110” – C-D	40.0
I	0.125” – Straight	45.0

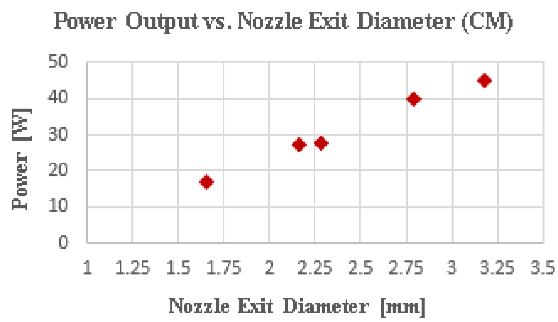


Figure 6: Power output vs. nozzle exit diameter

Figure 6 clearly shows that as the diameter of the nozzle increases for this set of nozzles, the power output also increases. This suggests that the nozzle exit diameter is a key factor in the Viper Expander power output and should be investigated in more detail. Based on experimental testing, the difference in power output between the Viper I and Viper II is very minimal. Also the data suggests that the nozzle contour may not have as strong of an impact on the power output as the nozzle exit diameter.

4.2 Heating Mode Results

For heating mode only the Viper II was tested since previous experimental testing showed that the performance was very similar to the Viper I. Three nozzles were tested with the Viper II at each of the three operating conditions (H1, H2 and H3). The nozzle used in the testing are as follows:

1. 0.080” (2.03 mm) exit diameter by 2.25” (57.15 mm) length – Straight
2. 0.090” (2.29 mm) throat by 0.110” (2.79 mm) outlet by 3.5” (88.9 mm) length – Converging-Diverging
3. 0.132” (3.35 mm) exit diameter by 3.5” (88.9 mm) length – Straight

The nozzles were chosen to with more or less evenly spaced exit diameters to better understand the how the diameter affects the power output. . The power output for the Viper II coupled with each nozzle at all heating mode operating conditions is shown in Table 3.

Table 3: Heating mode tests Viper power output

Nozzle	H1 [W]	H2 [W]	H3 [W]
0.080” – Straight	20.7	18.2	16.4
0.110” – C-D	37.2	35.0	37.0
0.132” – Straight	31.1	30.5	39.6

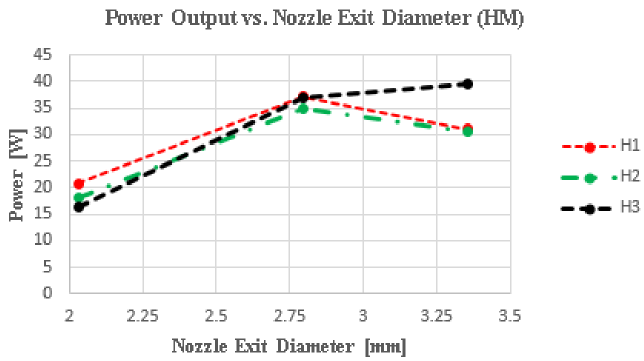


Figure 7: Power output vs. Nozzle Exit Diameter

Figure 7 shows that the trend in power output does not always increase with increasing nozzle exit diameter. The power decreases for both H1 and H2 as the nozzle diameter is increased from the 0.110 in (2.79 mm) to 0.132 in (3.35 mm). However, at the H3 test condition the power output does increase from 16.4 to 39.6W as the nozzle exit diameter increases. For both heating and cooling mode the target power output of 200W was not achieved any of the standard operating conditions. The results indicate that they may be other factors influencing the power output aside from the nozzle diameter. The following section shows a more detailed investigation of how the expander power output varies with several parameters.

4.3 Investigation of Parameters Affecting Power Output

Understanding how various parameters impact the power output will be very crucial in designing a more efficient device to achieve at least 200W. For the following investigation, the system was operated in heating mode. The outdoor air temperature will be set at either H1, H2 or H3 and held constant while the indoor temperature is raised from 70 to 110°F (21.1 to 43.3°C). Therefore, the evaporating pressure will remain constant while the condensing pressure increases and thus the pressure ratio across the Viper Expander will increase. This will change the properties of the refrigerant entering the nozzle and the impact of the change in these parameters on the power can be recorded. Figure 8 shows the power output versus increasing pressure ratio with the outdoor temperature set at the H1 temperature of 47°F (8.33°C).

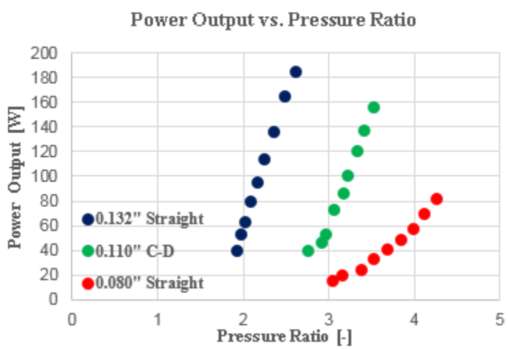


Figure 8: Power vs. pressure ratio

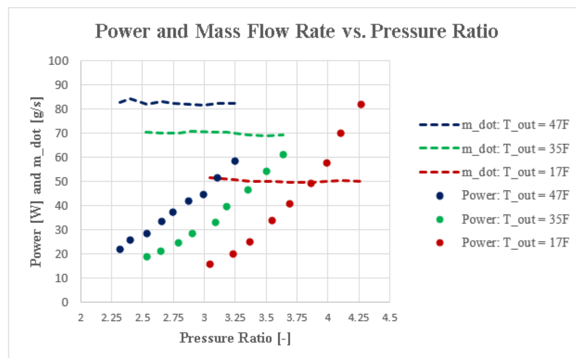


Figure 9: Power and $\dot{m}$for 0.080" Straight nozzle

Three different nozzles tested are the 0.080" Straight, 0.110" C-D and the 0.132" Straight. For each of the three nozzles the power output increases with an almost linear trend. The 0.132" Straight nozzle resulted in 40 – 186 W, the 0.110" C-D in 40 – 160 W and the 0.080" Straight in 18 – 82 W. It can be concluded that the Viper does have the potential to generate a significant amount of power, but with the current device the system may need to be run at extreme conditions. The maximum power generated for each of the nozzles in Figure 8 correlates to an ambient indoor air temperature of 110°F.

Furthermore, the comparison in Figure 8 indicates that a single pressure ratio does not directly correspond to a single power output. For example, a pressure ratio of 3.5 results in nearly 160 W from the 0.110" C-D nozzle and only 38 W from the 0.080" Straight nozzle. It can be stated that the Viper is more likely to generate more power at higher pressure rations, but the power output is truly a function of many parameters.

Another parameter influencing the Viper power output is the mass flow rate. This can be seen in Figure 9, which shows the power output and mass flow rate versus the pressure ratio for the 0.080" Straight nozzle with outdoor ambient air conditions at 47°F (8.33°C), 35°F (1.67°C) and 17°F (-8.33°C). For a given nozzle the mass flow rate does not change with increasing pressure ratio, however the mass flow rate is different at each of the three system operating conditions. As the evaporating pressure decreases due to the ambient air temperature decreasing from 47 - 17°F, the mass flow rate consequently decreases from 80 - 50 g/s. Observing the pressure ratio of 3.25 clearly shows that the mass flow rate has an influence on the power output. At the 17°F outdoor air condition, the mass flow rate is 50 g/s and the power output is 20W. Moving vertically upward to shows that both the mass flow rate and power output increase. This indicates that the power is also a function of the mass flow rate which corresponds to Equation 1.

The inlet condition to the nozzle is also important to investigate. Figure 10 shows a comparison between the density and the power output for the 0.080”- Straight nozzle and the 0.110”- C-D nozzle. The 0.080” Straight nozzle is plotted in red and the 0.110” C-D is blue. The density has the square markers and the power has the triangle markers with the dashed line. The inlet state point to the 0.080”- Straight nozzle was subcooled over the entire range of increasing pressure ratio and has a larger density compared to the 0.110” C-D which was always two-phase. The plot shows that as the density decreases for both nozzles, the power increases. Comparing the power output between the two nozzles at the 3.25 pressure ratio also indicates that the power output is higher for the nozzle with the lower density. For both nozzles, there is an inverse relationship between the power output and the density. As the refrigerant becomes less dense, the power output increases. This suggests that the Viper does in fact generate more power with a high mass flow rate of refrigerant exerting less inertia enters the Viper impeller. Comparing the power generated for subcooled liquid at the nozzle inlet state to that of the two-phase refrigerant, significantly more power is generated for the two-phase refrigerant.

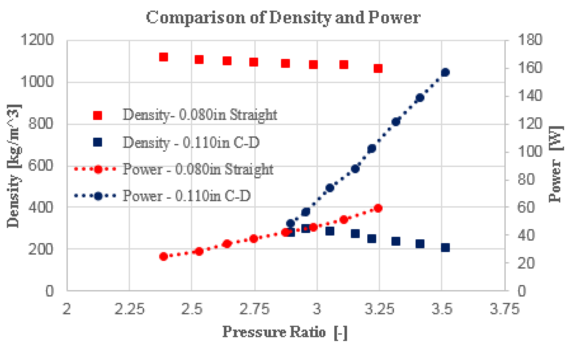


Figure 10: Power and density

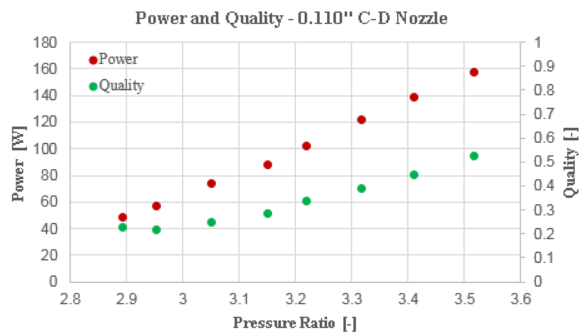


Figure 11: Power and quality

The density for the inlet state point for the 0.080” Straight nozzle is in the subcooled region and ranges from 1066 to 1163 kg/m^3 with a power output ranging from 22 - 59W. The inlet state point of 0.110” C-D nozzle is in the two-phase region and the density ranges from 200 - 430 kg/m^3 with a power output of 40 - 160W. The two-phase inlet state corresponds to 82 - 183% more power output compared to the subcooled inlet state point. This alludes to the fact that the expander performs better with higher vapor content and is shown in Figure 11. The power output is in red and corresponds to the left ordinate, whereas the quality is green and corresponds to the right ordinate.

As expected, the quality increases with increasing pressure ratio, increasing power and decreasing density. This confirms that the power output does in fact increase with a higher vapor content. The viscosity of R410A at single-phase vapor is an order of magnitude less than the viscosity of R410A at single-phase liquid. For a fluid accelerating through a nozzle, the velocity will experience less losses due to viscous forces for a vapor compared to for a pure liquid. Since the losses due to viscosity are less, it is probable that the exit velocity will be greater for a vapor than a liquid for the same inlet and outlet pressure. A higher velocity jet of R410A may be entering the turbine impeller for a higher vapor content at the nozzle inlet.

Testing has shown that the rotational speed of the Viper Expander varies with the resistance the load that is receiving the power. Figure 12 shows that the rotational speed increases with resistance. Figure 13 shows that the power directly correlates to the rotational speed. At higher rotational speed results in a larger power output, therefore achieving a higher velocity refrigerant is likely to spin the impeller faster and thus generate higher power output.

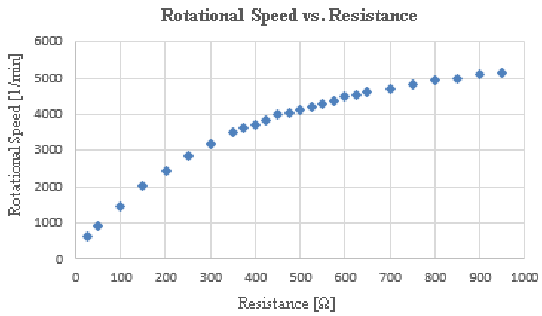


Figure 12: Rotational speed vs. resistance

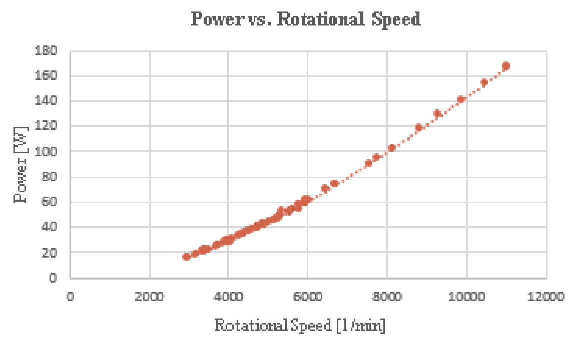


Figure 13: Power vs. rotational speed

The data in Figure 13 shows the power output ranging from 20 to 167W and the rotational speed ranging from 3000 to 11,000 RPMs. In order to reach the target power output of 200W, it is estimated that a rotational speed of 12,300 RPMs must be achieved.

5. CONCLUSIONS AND FUTURE WORK

5. 1 Conclusions

This paper has investigated the performance testing a device to replace the passive expansion valve in an R410A split system unitary heat pump. The device is referred to as the Viper Expander and consists of a nozzle, impeller and a generator. The Viper Expander outputs a 3-phase alternating current that is rectified to direct current and feeds its power to an ECM. The main conclusions of this paper are the following:

- The power output was 17 to 45W (3 to 7.50% of fan power consumption) at the cooling mode Test A conditions
- The power output was 16.4 to 39.4W (3 to 6% of fan power consumption) at the heating mode Test H1, H2 and H3 conditions
- The power output is a function of several parameters including, pressure ratio, refrigerant mass flow rate, exit diameter of nozzle, viscosity of refrigerant at nozzle inlet and the resistance of the load
- The Viper Expander can generate the target 200W, but so far not at standard operating conditions. A higher efficiency device is required.

5.2 Future Work

As this device is to completely replace the self-regulating function of the TXV or EXV, the Viper Expander must also be self-regulating. Future work is to develop a variable nozzle and control algorithm for the device to find the optimum combination of all parameters influencing the performance. By changing the nozzle diameter to best match the system, testing predicts the possibility to improve the COP of the heat pump.

By redesigning the expander impeller and housing to be more like a pure impulse turbine, higher power output could be achieved. It is suggested that a Pelton wheel turbine design be used and tested with the R410A heat pump system. Also, since the experimental results show higher power output with a less viscous fluid entering the nozzle, it is suggested to test the expander in a transcritical carbondioxide (CO_2) refrigeration unit. The lower viscosity and high pressure of a CO_2 system could result in higher power output.

NOMENCLATURE

A	area	(m^3)	**Acronyms**	
$\dot{m}$	mass flow rate	(kg/s)	AC	Alternating Current
$\dot{W}$	power	(W)	C-D	Converging-Diverging
V	velocity	(m/s)	CM	Cooling Mode
			CO_2	Carbon Dioxide
Greek symbols			COP	Coefficient of Performance
η	efficiency		DC	Direct Current

ρ	density	ECM	Electronically Commutated Motor
		ED-WOW	Expansion Device - With Output Work
Subscript		EXV	Electronic Expansion Valve
exit	exit of nozzle	HEX	Heat Exchanger
s	isentropic	HM	Heating Mode
V	Viper Expander	HSPF	Heating Seasonal Performance Factor
Viper	Viper Expander	SEER	Seasonal Energy Efficiency Ratio
		TXV	Thermostatic Expansion Valve

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