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CHARACTERISTICS AND PERFORMANCE OF A CO_2 HEAT PUMP AT 50% OVERCHARGED CONDITIONS

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ABSTRACT

A CO_2 water to water heat pump system was used to investigate the effects of throttle valve opening and water flow rates at higher charge capacity (149% of the full charge). It was observed that the throttle opening affects the gas cooler pressure (and inlet temperature) where the less the opening the higher the pressure. Although the increase in gas cooler pressure was improving the gas cooler heat output and therefore the system efficiencies, there is an optimum pressure which when exceeded, the system performance deteriorates. This optimum pressure also is related to a certain throttle opening which is approximately at 20% in this study. Similarly, as the throttle valve is reduced, the overall system becomes more ideal as indicated by Lorentz efficiency. On the other hand, the higher the flow rates the higher the performance although it's better to keep the chilling water (evaporator) flow rate low and cooling water (gas cooler) flow rates high than vice versa for the best heat pump performance. However, if the main intention is to idealize the system, then it's better to keep the chilling water flow rates high and the cooling water flow rates low as stipulated by the Lorentz efficiency. Many systems tend to be overcharged while the refrigerant is being refilled and therefore will behave as observed in this study, thus its importance.

1. Introduction

Refrigerant selection is a key design decision that influences the mechanical design of heat pump equipment. Factors that must be considered in refrigerant selection include:

- performance
- safety
- reliability
- environmental acceptability
- cost

Though the primary requirements are safety, reliability and nowadays environmentally friendly (in terms of ozone depletion and global warming potential). So far, mainly synthetic, man-made substances have been used as refrigerants, and their negative environmental effects could only be identified on the long term. When safely contained in a proper refrigerator, refrigerants do not cause any harm to the environment, but if the system leaks or during maintenance or at its end of life, these hazardous gases are released to the atmosphere and that's when they become harmful. Furthermore, during their manufacture, toxic products are created and released which are not only harmful to the environment, but also are a health concern. Due to the uncertainty of these artificial refrigerants, their use will be controlled and limited because their full environmental impact isn't known. Therefore, there is a general need for a more permanent solution. As safety and environmental concerns are becoming more important, their impact as a requirement is becoming more crucial to even overshadow reliability, performance and cost. This calls for more natural and freely existing materials.

Natural refrigerants are chemicals which occur as a result of natural process and which can be used to produce a refrigerating effect. They are not synthetic although can also be created synthetically. The most common are air, water, ammonia, hydrocarbons and Carbon dioxide (CO_2 or R744), also known as the gentle five. Natural refrigerants have negligibly small ozone depletion potential and global warming potential unlike most artificial refrigerants [1]. CO_2 can be regarded as the best refrigerant because it is non-toxic, non-flammable and does not contribute to ozone depletion and negligibly to global warming as compared to the artificial refrigerants. CO_2 used as a refrigerant can be recycled from other industrial processes thus reused instead of being released to the atmosphere. CO_2 meets all the basic requirements of a refrigerant in that it is readily available and not expensive. Also, it has excellent thermo-physical properties and transport properties leading to good heat transfer; it is not corrosive and is compatible with various common materials. In addition, it has efficient compression properties and compact system design due to high volumetric capacity and high operating pressures [2-4].

The only concern with the use of CO_2 as a refrigerant is it becomes a super critical fluid at 31.1 °C and 73.7 bar. For low critical temperature refrigerants, a trans-critical heat pump cycle would perform better especially in some applications such as domestic water heating (DHW) because of the good temperature fit between the CO_2 and the water in a counter-flow gas cooler (condenser in trans-critical cycles) [5]. Unfortunately, CO_2 trans-critical cycles operate at high heat rejection pressures. High pressure presents design and cost challenges in terms of component robustness and compressor capability. Typical capital cost of CO_2 heat pumps is approximately double the cost of convectional heat pumps [6]. Therefore, its efficiency of operation must be maximized to justify the capital cost. This study investigates efficiency improvements of such a system by varying the throttle valve and water flow rates while maintaining the water temperature and charge amount constant. Nevertheless, the charge amount was increased from 4.2 kg to 6.3 kg so as to investigate the behavior of the system at excess charge.

2. Methodology

In this study, a CO_2 trans-critical water to water test bed was used to study the output of a typical heat pump. The system contains a compressor, an evaporator, a throttling device and a gas cooler as the basic equipment among other secondary or supporting devices like vapor-liquid separators, oil separators, cooling water system and chilling water system. A schematic diagram of the system is shown in Figure 1. The compressor of the system is Italian designed, special piston, semi-hermetic, Dorin's CO_2 compressor of the second generation with a maximum output of 10 MPa and 110 ºC. The throttle device used adopts a manual throttle valve design so that the amount of refrigerant flowing and thus the maximum operating pressure can be adjusted accordingly.

Both the evaporator and gas cooler in this system have the tube in tube design because of the heat transfer and viscosity characteristics of CO_2. CO_2 flows in the inner tube while water flows in the annulus of the outer tube. In the evaporator, there are three inner tubes while in the gas cooler there are four inner tubes. The inner diameter of the outer tube in both the gas cooler and the evaporator is 26 mm while the inner diameter of the inner tube varied according to the appliance. The evaporator inner tube has a diameter of 6 mm while the gas cooler inner tube has an inner diameter of 4 mm. All tubes are made of copper and have a thickness of 1 mm. To optimize the dimension effect, the gas cooler is made to have 12 passes whereas the evaporator is made to have 10 passes, each pass having a length of 1580 mm. The system also contains temperature, pressure and flow-rate sensors before and after each major component. Signals from these sensors are captured and stored by a data acquisition system. The data acquisition system hardware uses PCAuto industrial control software to collect and store operating and system measurement parameters at each state point and reflect it in the data acquisition interface.

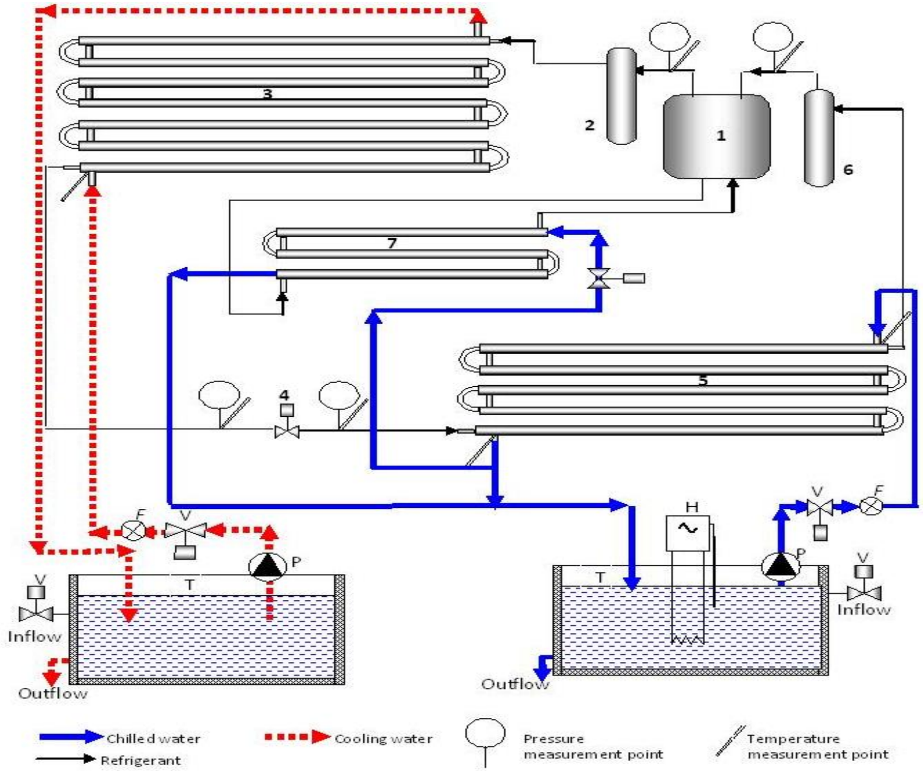

Figure 1: Schematic diagram of the CO_2 heat pump test bed; (1) compressor, (2) oil separator, (3) gas cooler, (4) thermal expansion valve, (5) evaporator, (6) liquid-vapor separator, (7) lubricating oil heat exchanger, (*F*) flow meter, (V) water yield control valve, (P) water pump, (H) heater, (T) water tank.

During experimentation, the test rig was left to run for some time until all the readings stabilized then the output recorded in terms of temperature and pressure at the state points. Each experiment was done at least twice and results taken when their percentage error was less than 5%. These results were used to calculate the heat supplied (or rejected), work done, COP and Lorenz efficiency as explained later. Amount of refrigerant in the system was kept constant at 6.3 kg (approximately 149% of the full value) while both the cooling and chilling water inlet temperature at approximately 25 °C. Effects of the other operational parameters were investigated. These parameters are:

i) Throttle valve opening,
ii) Cooling and Chilling water flow rates.

Throttle valve opening was varied between full open (100%) to 25% open in four steps. An additional experiment at 20% opening was conducted. This was the safe minimum opening that could be achieved considering the maximum pressure and temperature values of the compressor. In the throttle valve experiments, the water flow rates were fixed at maximum value i.e. 0.38 kg/s for cooling water while 0.32 kg/s for chilling water. The difference in these flow rates is due to the fact that the chilling water system is also used in the compressor oil cooler. Finally, to investigate the effect of water flow rate, the throttle valve opening was fixed at 25% then the water flow rates changed from 0.38 kg/s to 0.19 kg/s for cooling water and 0.32 kg/s to 0.16 kg/s for chilling water. Table 1 shows the arrangement of the experiments.

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Table 1: Design of experiments.

Experiment	Throttle valve opening	Cooling – Chilling water temperature	Cooling – Chilling water flow rate
	%	°C	Kg/s
Effect of Throttle valve opening	100 75 50 25 20	≈ 25	0.38 – 0.32
Effect of Water flow rates	25	≈ 25	0.38 – 0.32 0.29 – 0.24 0.19 – 0.16 0.38 – 0.16 0.19 – 0.32

The heat transfer in a segment of the counter flow tube in tube gas cooler or evaporator which exchanges heat with liquid water can be written as [7]:

$$\dot{Q} = (UA)\frac{\left(T_{ref_2} - T_{water_2}\right) - \left(T_{ref_3} - T_{water_3}\right)}{\ln\left(\frac{\left(T_{ref_2} - T_{water_2}\right)}{\left(T_{ref_3} - T_{water_3}\right)}\right)} = \dot{m}_{ref}\left(h_{ref_2} - h_{ref_3}\right) = \dot{m}_{water} c_{water}\left(T_{water_2} - T_{water_3}\right) \qquad \text{(i)}$$

Where the subscript ref means refrigerant and sub-subscript 2 means conditions at the refrigerant entry of the segment while sub-subscript 3 means at the refrigerant exit of the segment of the heat exchanger. In the same way, if the state at entry of the compressor is represented by sub-subscript 1 and assuming there is no energy loss at the exit of the compressor, the work input into the compressor (assuming a perfect compressor with isentropic efficiency of 100%) can be given by:

$$\dot{W} = \dot{m}_{ref}\left(h_{ref_2} - h_{ref_1}\right) \qquad \text{(ii)}$$

Therefore the cycle efficiency of the heat pump represented by the coefficient of performance (COP_{hp}) can be gotten by:

$$COP_{hp} = \frac{\dot{Q}}{\dot{W}} \qquad \text{(iii)}$$

For a trans-critical CO_2 cycle, where heat is given off at a gliding temperature, the Carnot cycle cannot be used as a reference (most efficient) cycle. It is generally accepted that the modified Lorentz cycle is more suitable as the theoretical reference cycle. This cycle is characterized by the following changes of state:

1 – 2s Isentropic compression
2s – 3 Isobaric heat rejection (gliding temperature)
3 – 4s Isentropic expansion
4s – 1 Isothermal heat absorption

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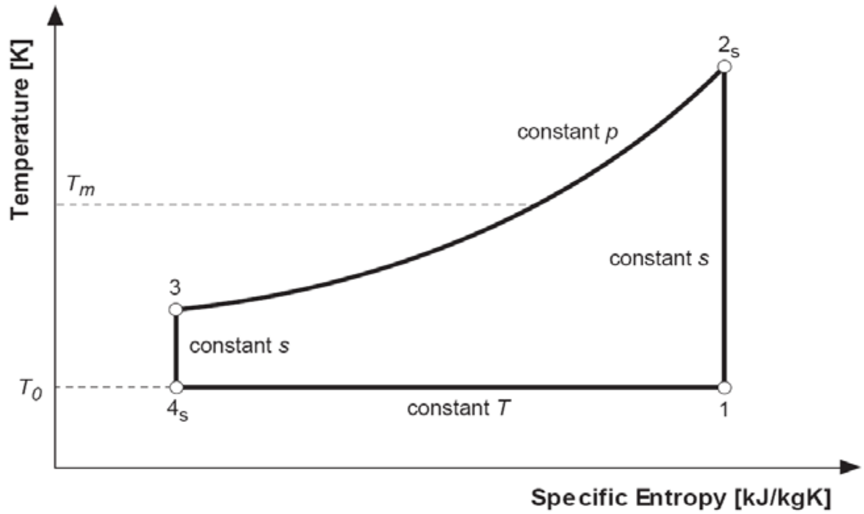

Figure 2: Modified Lorentz cycle [8].

Figure 2 shows the principle of the modified Lorenz cycle in a Temperature-Entropy (T-s) diagram [9]. It should be important to note that in this cycle, the isenthalpic throttle valve is replaced by a perfect expansion turbine which harnesses energy isentropically, thus there is some energy gained. The COP of the modified Lorentz cycle working as a heat pump is defined as:

$$COP_{LZ} = \frac{T_m}{T_m - T_0} \quad \text{and} \quad T_m = \frac{T_{2s} - T_3}{\ln\left(\frac{T_{2s}}{T_3}\right)} \tag{iv}$$

Where T_0 is the temperature of the heat sink, all temperatures being in absolute scale. This gives

$$COP_{LZ} = \frac{T_{2s} - T_3}{(T_{2s} - T_3) - T_0 \ln\left(\frac{T_{2s}}{T_3}\right)} \tag{v}$$

Lorentz efficiency (η_{LZ}) is used in trans-critical CO_2 cycles to evaluate how close the actual cycle is to the reference ideal cycle. This efficiency is used to quantify the gains in reducing irreversibilities. η_{LZ} is defined as:

$$\eta_{LZ} = \frac{COP_{hp}}{COP_{LZ}} \tag{vi}$$

3. Results and Discussion

Table 2 shows the relative results when the throttle valve opening is adjusted. As expected, the smaller the throttle valve, the higher the output pressure and temperature. Due to maximum pressure limitations of the compressor, the lowest valve opening that could be achieved was 20%. Figure 3, 4 and 5 shows the results for the gas cooler heat output and the compressor power input (Figure 3), the heat pump efficiency and the efficiency of the Lorentz cycle both in terms of COP (Figure 4) and the Lorentz efficiency (Figure 5). As noted, there was conspicuous increase of heat output from the gas cooler as the throttle opening reduced from 100% while the compressor power increased marginally (Figure 3). The same happened to the heat pump COP where as the throttle valve closes, the COP_{hp} increased (Figure 4). Unfortunately, due to the limitations of the compressor, the optimum throttle opening amount was not achieved but when the COP_{hp} graph is extrapolated, it can be observed that 20% throttle valve opening is near the optimum point. A possible explanation of this behavior is in the relationship of pressure with throttle valve opening. As the valve opening was reduced, the gas cooler pressure increased due to accumulation of CO_2 in it. As it is widely known, there is always a particular optimum gas cooler pressure for every equipment setting. In this case, the optimum pressure was approximated at slightly below the 20% throttle valve settings with a corresponding pressure of slightly above 9.25 MPa. In these systems, as the pressure increases the COP_{hp} increases initially and then the added capacity no longer compensates for the additional work of compression and hence COP_{hp} decreases. At the optimum pressure, the marginal increase in capacity equals marginal increase of work. High-side pressure regulation can be applied to maintain the maximum COP_{hp} and/or to regulate the heating or cooling capacity [10].

Table 2: Temperature and pressure outputs at different throttle valve openings.

THROTTLE VALVE OPENING	GAS COOLER				EVAPORATOR			
	Inlet		Outlet		Inlet		Outlet	
(%)	T_1 (°C)	P_1 (MPa)	T_2 (°C)	P_2 (MPa)	T_3 (°C)	P_3 (MPa)	T_4 (°C)	P_4 (MPa)
100	37.1	8.05	25.9	6.5	16.2	5.85	22.3	5
75	37.5	8.1	26.6	6.6	15.6	5.75	21.5	4.95
50	38.4	8.1	28.4	6.95	14.5	5.6	20.1	4.85
25	43.7	8.45	26.9	7.7	12.6	5.15	17.1	4.6
20	49.9	9.25	25.4	8.75	12.1	5.05	16.2	4.55

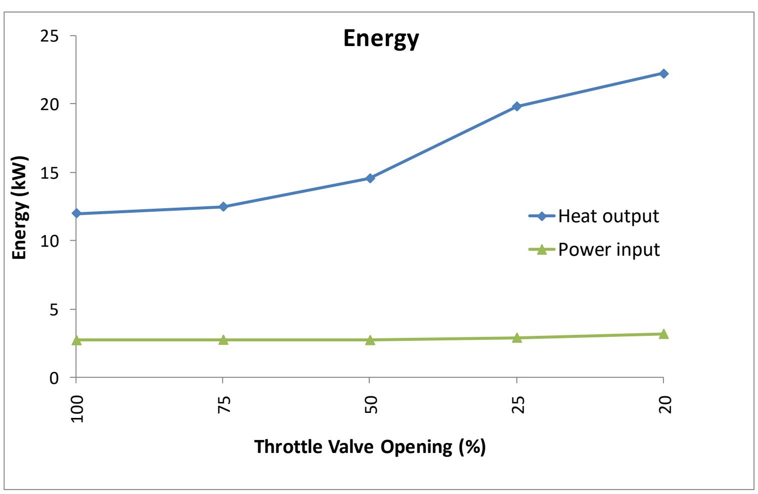

Figure 3: Effect of throttle valve opening on gas cooler heat output and compressor power input

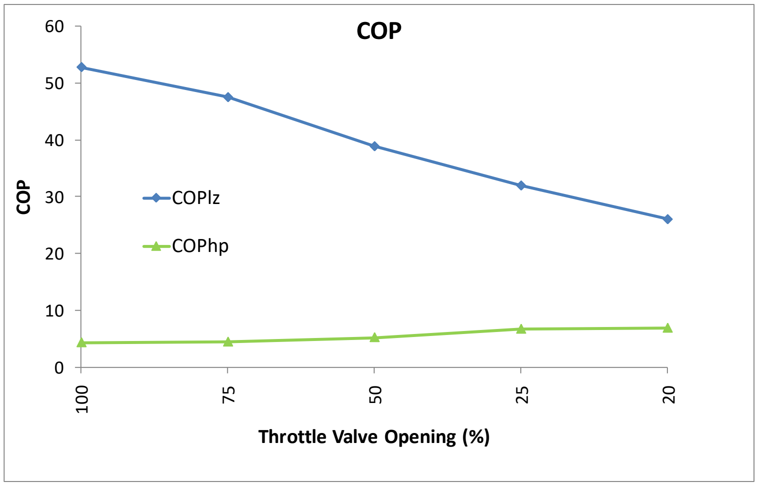

Figure 4: Effect of throttle valve opening on heat pump C.O.P and Lorentz C.O.P

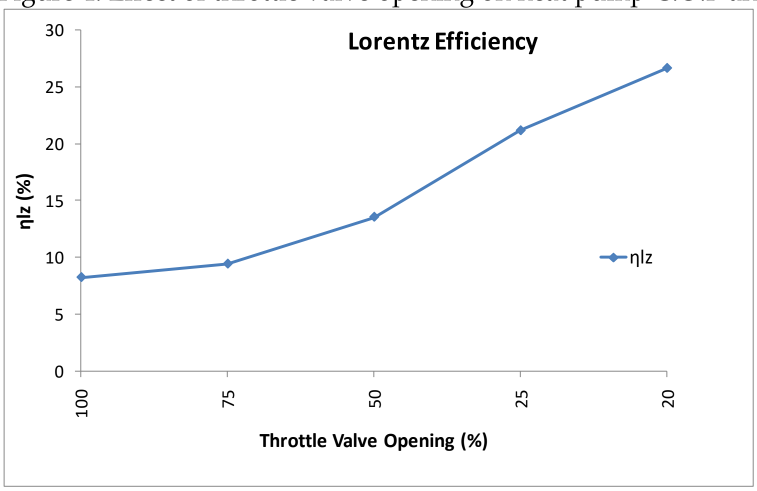

Figure 5: Effect of throttle valve opening on Lorentz efficiency

In all the results, only the Lorentz COP was continually reducing as the throttle valve opening reduced which indicated that ideally, as the throttle valve opening reduces, the best cycle performance also reduces. On the other hand, as the throttle opening reduced, the system became more efficient as evidenced by its increasing Lorentz efficiency (Figure 5). At the lowest valve opening, the Lorentz efficiency was nearly 27% which means that the performance was approaching the ideal value. This behavior is caused by a decrease in pressure loss as the higher pressure increases. Pressure loss in pipes and heat exchanger is one of the highest causes of irreversibility in heat pumps. A reduction in it means an improvement in the efficiency of the heat pump. As observed in Table 2, the difference in the inlet and outlet pressures in the gas cooler reduces as the throttle valve opening reduces. Therefore, with less throttle valve opening (which causes high gas cooler pressures), pressure losses also reduce and the system becomes more efficient.

The effect of high gas cooler pressures in compressors can also be another possibility of the observed results. Compressors operate more efficiently at higher gas cooler pressures. Furthermore, the negative effect of pressure loss becomes less significant as the output pressure increases both in the compressor and gas cooler. This gives a possibility of improving heat transfer through higher flow velocities in high-pressure systems. This is of particular importance for single-phase heat transfer in the gas cooler of CO_2 systems [11]. The optimum gas cooler pressure is a very important parameter, in that, the input power to the compressor is more or less proportional to the gas cooler pressure (p_{gc}). Also the gas cooler pressure has marked influence on the specific enthalpy due to the s-shape of the isotherm in supercritical region. Since the throttling valve inlet condition determines the specific refrigeration effect, it is necessary to control the high side pressure [10].

Table 3 shows the output when the cooling and chilling water flow rates were adjusted. The throttle valve was set at 25% opening while the water inlet temperature set at approximately 25 ^{o}C. The lowest values were achieved when the cooling water flow rate was maximum (0.38 kg/s) and the chilling water flow rate was minimum (0.16 kg/s), while the highest values were achieved at the opposite settings i.e. cooling water flow rate of 0.19 kg/s and chilling water flow rate of 0.32 kg/s. This can be attributed to the effect of the water on the refrigerant. When the cooling water flow rate is maximum, more heat is transferred to the water and thus the gas cooler outlet temperature is reduced and thus low evaporator inlet temperature. Coupled with less chilling water flow rate, very less heat is transferred to the refrigerant which results in a low refrigerant temperature at the evaporator outlet into the compressor. The compressor in turn compressed the refrigerant to an outlet temperature which was less as compared to the other settings of flow rate. Exactly the opposite occurs with low cooling water flow rates and high chilling water flow rates i.e. less heat transfer in gas cooler causes a high temperature refrigerant to enter the evaporator and due to higher heat transfer in the evaporator, a high temperature refrigerant exit the evaporator into the compressor where it is compressed to a higher temperature as compared to the other flow rate settings.

Table 3: Temperature and pressure outputs at different water flow rates.

WATER FLOW RATE	GAS COOLER				EVAPORATOR			
Cooling - Chilling	Inlet		Outlet		Inlet		Outlet	
(kg/s)	T_1 (°C)	P_1 (MPa)	T_2 (°C)	P_2 (MPa)	T_3 (°C)	P_3 (MPa)	T_4 (°C)	P_4 (MPa)
0.38 – 0.32	42.2	8.15	25.3	7.45	11.2	5	15.7	4.45
0.29 – 0.24	43.6	8.45	27.3	7.65	10.5	4.9	15.1	4.35
0.19 – 0.16	46.3	8.75	29.3	8.1	9.5	4.75	14.4	4.25
0.38 – 0.16	39.7	7.85	24.9	7.25	7.5	4.55	12	4.05
0.19 – 0.32	49.5	9.15	28.4	8.45	11.7	5.05	16	4.5

Figure 6 – 8 presents the heat output and compressor input (Figure 6), heat pump and Lorentz COP (Figure 7) and Lorentz efficiency (Figure 8). It can be observed from Figure 6 that the heat output from the gas cooler was immensely affected by the water flow rate change where the higher the water flow rates, the higher the gas cooler heat output. This means that the highest flow rate encountered here was not yet the optimum as its increase might have increased the heat output and COP (both COPs in this case). Less flow rates had less heat output probably because of less heat transfer area available or insufficient covering of the heat transfer surface by the water. On the other hand, the compressor power input was approximately the same and the water flow rate changes seem not to affect it due to negligible change in refrigerant mass flow rate and degree of superheat. Similar results were observed by other researchers [12]. Therefore, the COP_{hp} was changing according to the gas cooler heat output. Figure 8 also supported the notion that the higher the flow rate, the better the system's output. This was indicated by the Lorentz efficiency. As the flow rate increase, the system tends to be more ideal.

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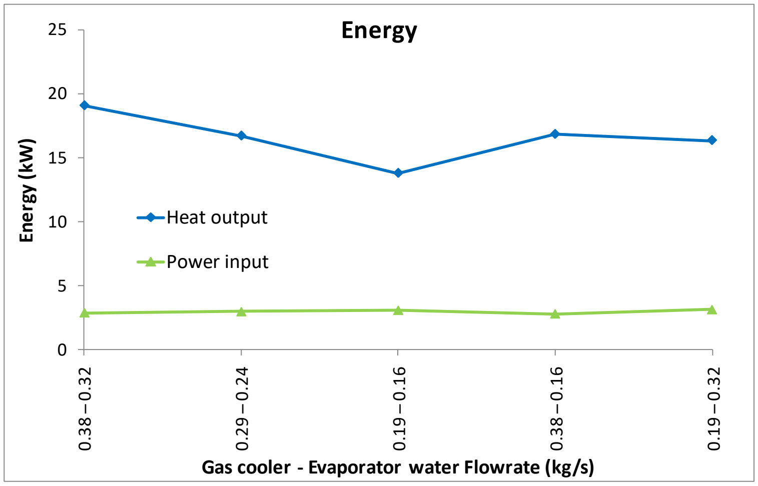

Figure 6: Effect of water flow rate on gas cooler heat output and compressor power input.

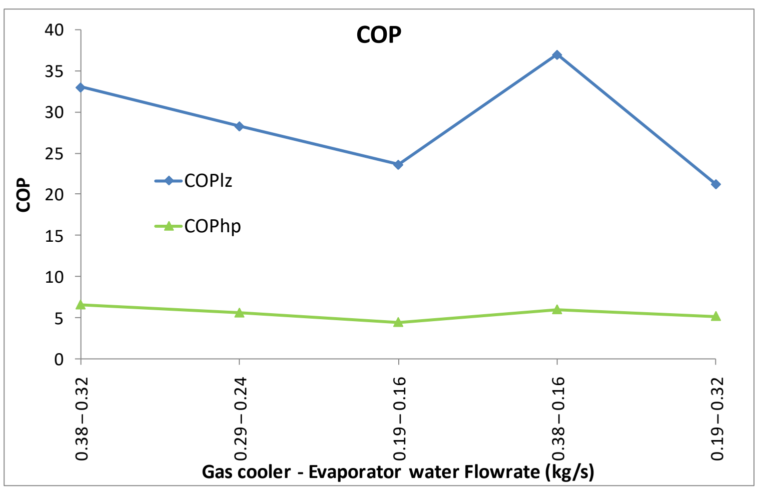

Figure 7: Effect of water flow rates on heat pump and Lorentz C.O.P .

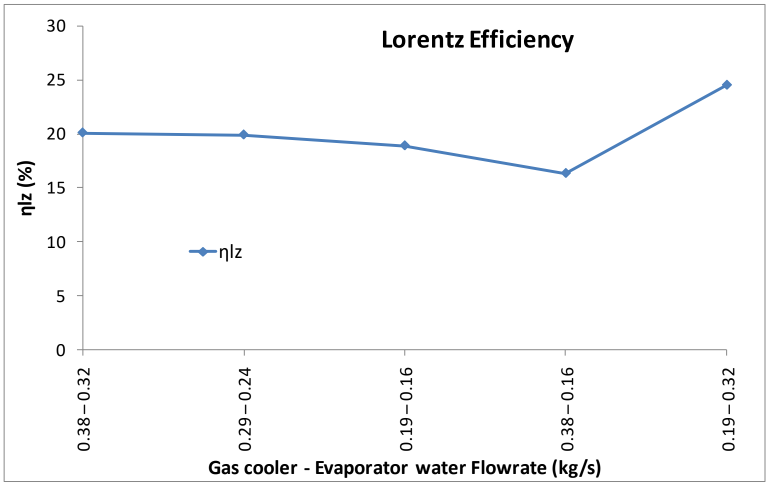

Figure 8: Effect of water flow rates on Lorentz efficiency.

Another critical observation is that the heat pump was more efficient when the chilling water flow rate was low and cooling water flow rate high than vice versa. This can be observed in Figures 7 where the COP_{hp} of cooling – chilling water flow rate 0.38 – 0.16 kg/s had better performance as compared to 0.19 – 0.38 kg/s. These observations lead to the conclusion that the heat transfer in the gas cooler is more important than the heat transfer in the evaporator for a higher overall system efficiency, a result also related to the pressure loss effect especially in the evaporator. Although, with Lorentz efficiency, the opposite was preferred where the chilling water flow rate should be high and the cooling water flow rate low. Thus, for the system to best resemble the ideal cycle, heat transfer in the evaporator becomes more important.

It is a well-known fact that heat transfer depends on the flow rate among other parameters like temperature difference and properties of the fluids. A lower flow rate means a low heat transfer. This can be caused by insufficient fluid for effective heat transfer. Here the fluid temperature increases rapidly thus it reaches a point where the temperature difference is insufficient for effective heat transfer. Also it can be caused by the fact that the fluid was insufficient enough not to cover the heat transfer surface area adequately. On the other hand, low cooling water flow rates causes a high temperature refrigerant to enter the throttle valve and a high chilling water flow rate ensures that a higher quality refrigerant enters the compressor which in turn delivers refrigerant at a higher temperature and pressure and that is why it had a bad performance. The opposite happens with reversed flow rates.

Regulating mass flow rates of refrigerant or cooling medium (cooling water in this case) at a constant chilling water flow rate, will affect approach temperature (difference between the refrigerant temperature and the cooling water temperature at the gas cooler output). For constant refrigerant mass flow rates, increasing mass flow rates of cooling medium will reduce refrigerant temperature out of the gas cooler. If rejected heat is not utilized and just dissipated to the cooling medium, the temperature out of the gas cooler can be reduce as low as possible, hence reducing the approach temperature [13]. The larger cooling water flow rate will in turn reduce the CO_2 outlet temperature from the gas cooler and with that increase the heating capacity of the CO_2 heat pump [14]. For same size of the gas cooler, the approach temperature reduces with increase in the operating pressure. The gas cooler capacity increases with increase in the operating pressure and also with the size of gas cooler. It also increases with increase in cooling medium flow rate as expected but the effect of increase in cooling medium flow rate is insignificant at higher pressure [15].

Conclusions

A CO_2 heat pump was tested at increased refrigerant amount. The effect of throttle valve adjustments and water flow rates were observed. As expected, when the throttle valve opening was reduced, the output of the heat pump in terms of system COP improved. This observation was related to the effect of the throttle valve opening to the gas cooler pressure where at a certain operating condition, the heat pump always have an optimum pressure. Similarly, the Lorentz efficiency continued to rise even when the throttle valve opening was reduced to 20%. On the other hand, the higher the water flow rates, the higher the system's performance. Still, it was found that it's better to keep the chilling water flow rate low and the cooling water flow rate high, for a higher heat pump COP but for a higher Lorentz efficiency, the opposite is true. This study provides an insight into how a CO_2 heat pump system behaves at higher refrigerant charges which is a common occurrence in practical systems.

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