TYPICAL INITIAL OUTPUT OF A CO_2 HEAT PUMP

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Abstract
Many of the refrigerants currently being used in heating, ventilation, air conditioning and refrigeration systems have high global warming potential. One potential, environmentally friendly replacement refrigerant is carbon dioxide (CO_2). In this study, a CO_2 trans-critical water to water test bed was used to study the output of a typical heat pump. Initial experimental results and thermo-physical properties were analyzed by NIST REFROP and plotted in a temperature-entropy (T-S). The energy output in the gas cooler were compared to the energy input in the compressor and the efficiency of the system in terms of coefficient of performance (COP) was observed to vary from 3.7 to 3.9. It was observed that there was a wide difference between the theoretical and experimental results when the same state conditions are considered. This was related to change on theoretical processes due to efficiency consideration of the equipment involved and flow process variables like friction which affects the heat transfer process.

1 Introduction

1.1 Refrigerants

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

- performance,
- safety,
- reliability,
- environmental acceptability, and
- cost.

Though the primary requirements is safety, reliable and nowadays environmentally friendly (in terms of ozone depletion and global warming potential). Table 1 summarizes the properties of some refrigerants. This table shows that no progress has been made in terms of global warming potential when switching from HCFCs to the HFC family. When securely contained in a properly operating system refrigerants do not impact climate change; however, system leaks and improper recovery of refrigerants during repairs or at end of life result in these harmful gases entering the atmosphere. Furthermore, during production of refrigerants, toxic and harmful wastes are also released to the environment which even cause air, water and land pollution in addition to releasing green gasses. An alternative to HFCs is to apply naturally occurring and ecologically safe substances, the so-called natural working fluids. The most important substances in this category are hydrocarbons, ammonia, carbon dioxide (CO_2), water and air.

Table 1: Properties of some refrigerants [1, 2].

Refrigerant	Critical temperature (°C)	Critical Pressure (bar)	ODP	GWP (100 years)	Flammable or Explosive	Toxicity
CFCs and HCFCs						
R12	100.9	40.6	0.9	8100	No	No
R22	96.2	49.8	0.055	1500	No	No
Pure HFCs						
R32	78.4	58.3	0	650	Yes	No
R134a	101.1	40.7	0	1200	No	No
R152a	113.5	45.2	0	140	Yes	No
HFC mixtures						
R404A	72.1	37.4	0	3300	No	No
R407C	86.8	46.0	0	1600	No	No
R410A	72.5	49.6	0	1900	No	No
Natural refrigerants						
Propane (R290)	96.8	42.5	0	3	Yes	No
Isobutane (R600a)	135.0	36.5	0	3	Yes	No
Ammonia (R717)	132.2	113.5	0	0	Yes	Yes
Carbon dioxide (R744)	31.0	73.8	0	1	No	No

As seen in table 1, CO_2 can be regarded as the best refrigerant because it is non-toxic, non-flammable and does not contribute to ozone depletion or global warming. CO_2 meet 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 [3-5]. In this study, a heat pump that uses CO_2 as the refrigerant was constructed and utilized to give out typical output of these types of equipments.

1.2 Trans-critical cycle

Unlike other convectional refrigerants, CO_2 has a very low critical temperature (see table 1). This makes it difficult for use in a sub-critical (convectional) cycle where typical condensing temperatures range between 80-110 °C. In a sub-critical cycle, a low critical temperature is a disadvantage because it limits the operating temperature range; heat cannot be delivered at temperatures greater than the critical temperature. Further, at temperatures less than but near critical temperature, the enthalpy of vaporization is reduced. This leads to a reduction in heating capacity and poor performance of the system. Therefore, for low critical temperature refrigerants, a trans-critical heat pump cycle would perform better than a sub-critical cycle. In a trans-critical heat pump, heat rejection pressures are greater than the critical pressure and heat delivery temperatures are no longer limited by critical temperature. Heat rejection occurs by single-phase sensible cooling (gas cooling) and thus takes place via a gas cooler rather than a condenser. A typical trans-critical cycle is depicted in a temperature-entropy (T-S) diagram as shown in figure 1. The process is characterized by isentropic compression (process 1-2), isobaric heat output (2-3), isenthalpic expansion (3-4) and isobaric heat intake (4-1).

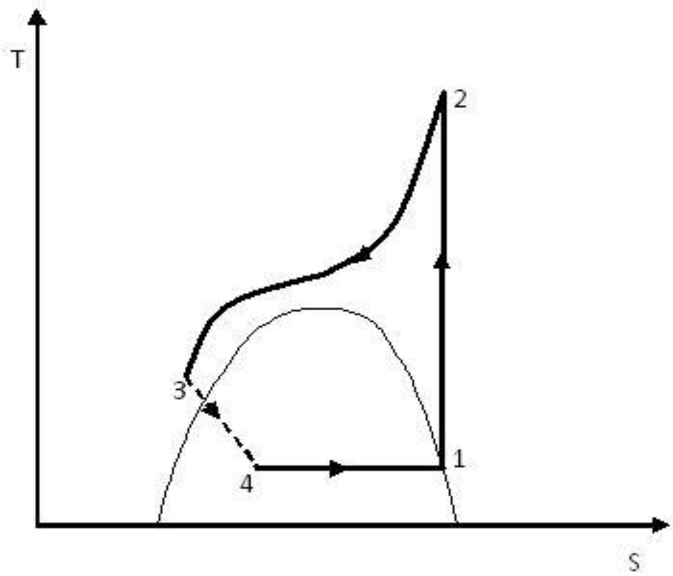

Figure 1: Theoretical trans-critical cycle.

This cycle is advantageous in some application such as domestic water heating (DHW) where it is possible to maintain a low temperature difference between the refrigerant and the heated fluid in the high-pressure side heat exchanger, in comparison with refrigerants that function within the sub-critical regime [6]. This is possible because in the supercritical state, the vapor–liquid state does not exist and the refrigerant temperature constantly changes in the high-pressure side heat exchanger. Since the gas cooler rejects heat by sensible cooling, the difference between the inlet and outlet temperatures (temperature glide) is much greater than in a condensing process. Thus the trans-critical cycle is more beneficial for heating applications requiring large temperature increase. The CO_2 heat pump water heater has the capability of supplying high-temperature DHW, which eliminates the requirement for supplementary heating. This is made possible because of the high energy efficiency of the

CO_2 heat pump water heater due to the good temperature fit between the CO_2 and the water in a counter-flow gas cooler [5, 7].

Unfortunately, CO_2 trans-critical cycles operate at high heat rejection pressures. As seen in table 1, CO_2 critical pressure is 73.8 bar, therefore, above the critical point, pressures are high. Pressures from 80 to 110 bar or more are common in these cycles. High pressure presents design challenges in terms of component robustness and compressor capability. Components for R744 systems have to withstand much higher pressures than their HFC and HCFC counterparts. Due to higher operating pressures, the heat exchangers for R744 require either smaller tube diameters or thicker walls. Other issues are introduced by the particularity of R744, e.g. high discharge temperatures, compatibility with lubricating oils, potential degradation of seals after decompression, etc. These issues hinder the fast development of a "component chain" and make CO_2 heat pumps to be a very expensive venture [1]. Typical capital cost of CO_2 heat pumps is approximately double the cost of convectional heat pumps [8]. However, today's manufacturing capabilities allow production of components which can meet these demands. In addition, high pressure presents some benefits e.g. CO_2 has a relatively high vapor density and correspondingly a high volumetric heating capacity (3 – 10 times larger than CFC, HCFC, HFC and HC refrigerants [9]), this allows a smaller volume of CO_2 to be cycled to achieve the same heating demand which allows for smaller components and a more compact system [10].

1.3 Irreversibilities

The cycle in figure 1 is a theoretical cycle where there are no losses. Practically, this cycle is hard to achieve because of effects of friction, leakage and other sources of losses. For instance, the effect of leakage and pressure losses inside a compressor has influence on the energetic (or isentropic) and volumetric efficiency of the compression process. Friction and heat dissipation in the compressor also contribute to irreversibility. Still when compared to compressors for convectional units, trans-critical compressors are better in terms of irreversibility [9]. This is related to less pressure drop due to higher pressure operation. For a fixed mass velocity, the pressure drop decreases with the operating pressure because the variation in the physical properties becomes smaller as the operating pressure increases [11]. The same effect occurs in the other parts of the equipment i.e. throttle valve and the heat exchangers. These losses modify the theoretical cycle to the one shown in figure 2 where the dotted line represent the actual (practical) cycle. In order to identify locations of energy loss within the carbon dioxide cycles, an analysis with respect to the Second Law of Thermodynamics is made on each component in the cycle. The irreversibility of the individual components is identified throughout the range of operating conditions. This indicates which components to focus on improving with respect to availability utilization.

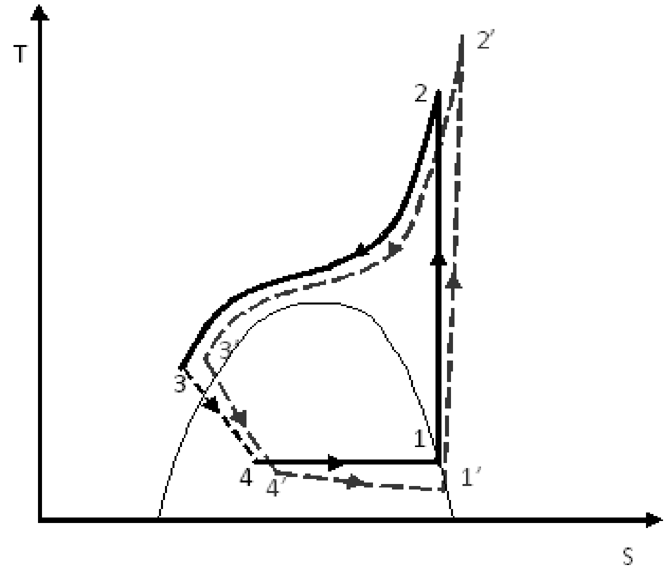

Figure 2: Actual trans-critical cycle.

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

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

Where $\dot{Q}$ is the heat output from the gas cooler while $\dot{W}$ is the power supplied to the compressor. Actual values of heat output and power supply will be affected by irreversibilities, thus making the actual COP_{hp} less than the theoretical one.

2 CO_2 Heat Pump Test Bed

In this study, a CO_2 trans-critical water to water test bed was designed and 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 equipments among other secondary or supporting part devices like vapor-liquid and oil separators; and cooling and chilling water systems. A schematic diagram of the system is shown in figure 5. Due to the very high-pressure existing in the system and other special characteristics of the working fluid and operating conditions, all system components were designed carefully for smooth operation. 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 power consumption of the compressor when operating optimally is 3 kW. The throttle device used adopts a manual throttle valve design so that the amount of refrigerant flowing 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 8 mm while the gas cooler inner tube had an inner diameter of 6 mm. The diameters were chosen because of the state and flow characteristics of CO_2 in these devices brought by conditions experienced in them and also to minimize pressure drop as much as possible. All tubes had 1 mm thickness which is sufficient to sustain the expected refrigerant pressure. 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.

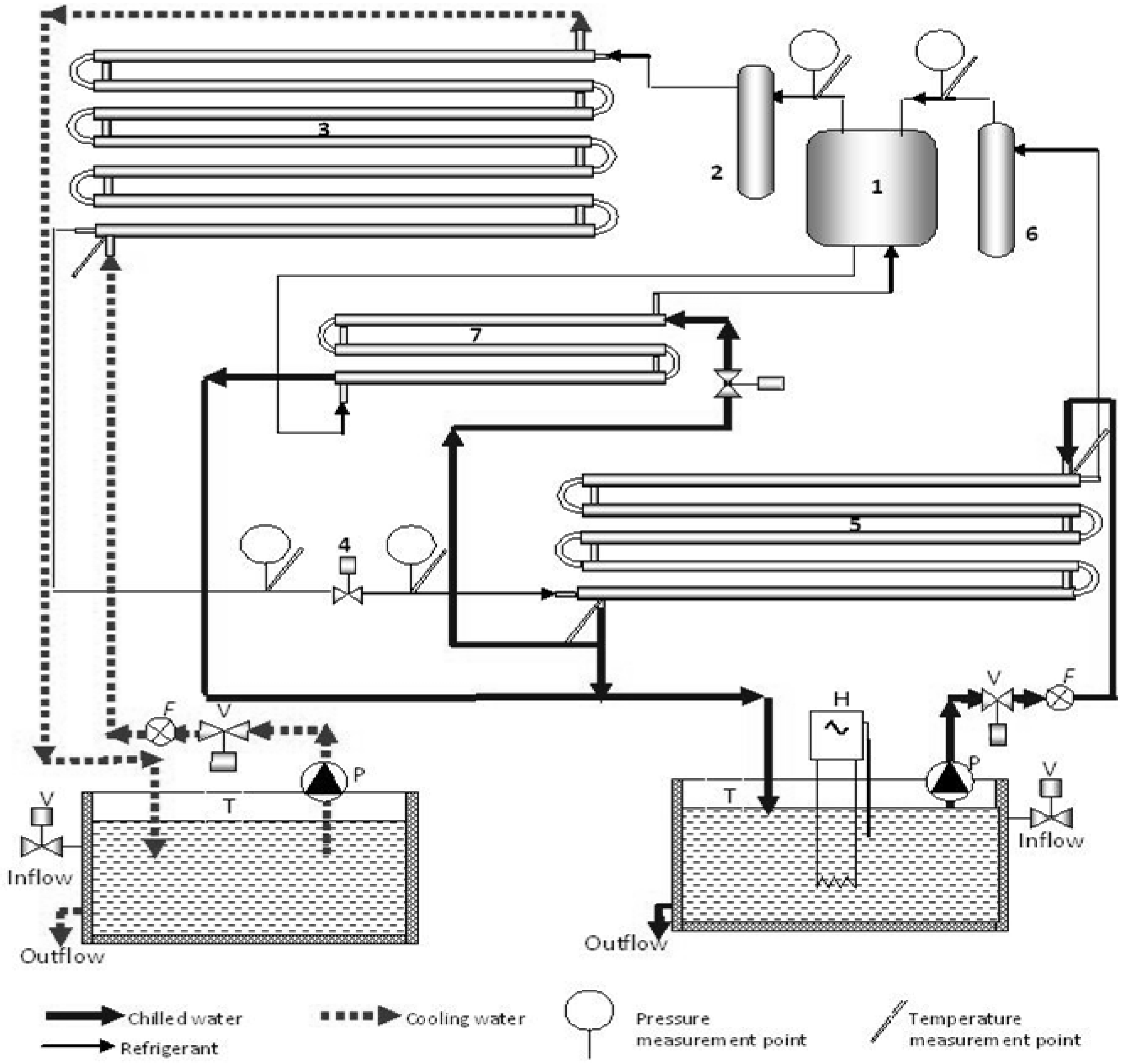

Figure 3: Schematic diagram of the CO_2 heat pump test bed; (1) compressor, (2) oil separator, (3) gas cooler, (4) thermal expansion valve, (5) evaporator, (6) gas-liquid separator, (7) lubricating oil heat exchanger, (*F*) flow meter, (V) water yield control valve, (P) water pump, (H) heater, (T) water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

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 (Force control supervisory control configuration software) to collect and store operating and system measurement parameters at each state point and reflect it in the data acquisition interface. 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. Initial experiments were conducted with normal conditions when the test rig is fully charged.

Several tests were done to examine consistency and stability of the system. The dependence of thermo-physical properties on temperature and pressure at a certain point or state was calculated using NIST REFPROP software. A temperature-entropy (T-S) diagram was used to present the output of the system. To see the output of the CO_2 system when run under sub critical conditions, the charge was reduced to quarter of the full amount and the test rig operated. Several experiments were conducted in these conditions and the average taken. The standard deviation of the output was less than 5%.

3 Results and Discussion

Table 2: Preliminary tests output.

Run	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)
1	75.37	7.61	31.00	7.30	4.94	3.76	5.95	3.48
2	75.37	7.61	31.00	7.30	4.94	3.77	6.65	3.49
3	75.57	7.59	31.00	7.27	5.15	3.80	8.57	3.52
4	75.57	7.59	31.00	7.27	5.15	3.80	8.57	3.52
5	79.60	7.72	29.79	7.45	4.14	3.71	10.08	3.45
6	79.80	7.72	29.79	7.45	4.14	3.71	10.08	3.45
7	79.80	7.01	29.99	7.44	4.14	3.70	8.87	3.44
8	79.80	7.01	29.99	7.44	4.14	3.70	8.87	3.44
9	80.00	7.70	29.99	7.44	4.14	3.70	8.37	3.43
10	80.00	7.70	29.99	7.44	4.14	3.69	8.37	3.43
11	79.80	7.69	29.99	7.42	3.94	3.67	6.65	3.41
12	81.71	7.77	29.09	7.54	2.93	3.57	5.44	3.32
13	81.91	7.77	29.09	7.54	2.93	3.57	5.44	3.32
14	64.2	1.8	20.3	1.6	17.2	1.15	18.9	1

After the initial experiments, the output was put in tabular form as shown in table 2. Run 1 to 13 represent the initial output when the system is fully charged while run 14 is the output when the system is charged by less amount of the refrigerant. NIST REFROP was used to calculate the relevant thermo-physical properties using the values gotten. Figure 3 shows the practical cycle of the system when run 1 was considered. As it is seen, the cycle has some inefficiency thus the use of dotted line. Other researchers in the same field got approximately the same values of COP in their initial experiments [12]. The same researchers improved their system's efficiency by optimizing the working conditions and improving the system apparatus. In the same way, the system in this study is under investigation to improve its performance. Variation of operation parameters like the evaporation temperature, the gas cooler temperature and pressure, the cooling water temperature and flow rates and amount of refrigerant is being undertaken. These parameters were found to affect the operation of similar systems by a wide margin, thus deemed very important [13]. Optimizing these parameters will reduce irreversibility causes especially friction and pressure drop because these causes depend on operating conditions and their interactivity. The effect of refrigerant amount of them is discussed briefly in this study.

Table 3: Heat output and efficiency of the cycle.

Run	Heat Output (kW)	Compressor Work Input (kW)	COP_{hp}
1	10.975	2.958	3.71
2	10.968	2.952	3.72
3	11.136	2.964	3.76
4	11.171	2.964	3.77
5	11.443	3.018	3.79
6	11.549	3.026	3.82
7	11.699	2.992	3.91
8	11.740	2.998	3.92
9	11.780	3.004	3.92
10	11.795	2.998	3.93
11	11.688	2.99	3.91
12	11.789	3.022	3.90
13	11.686	3.022	3.87
14	1.115	1.666	0.669

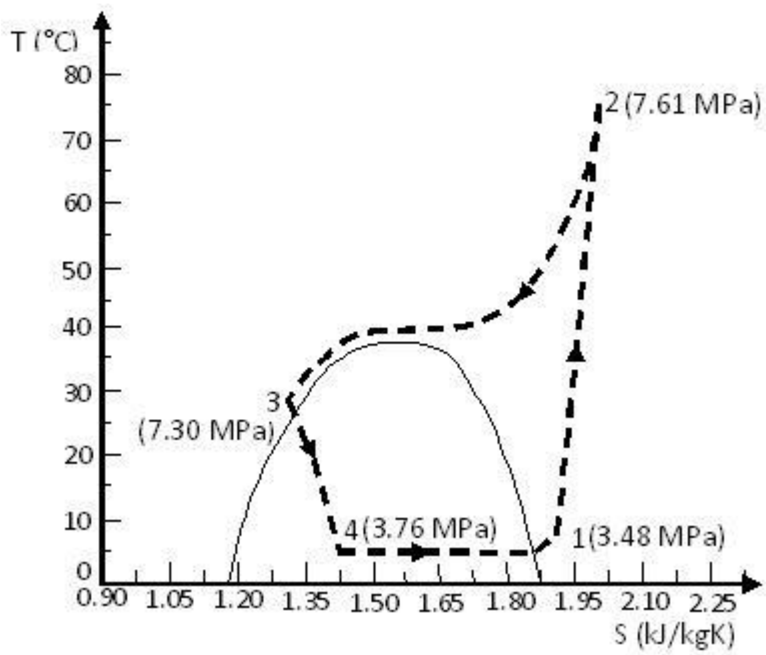

Figure 4: T-S diagram of the average output values at full charge.

After reducing the amount of charge, the output of the system as seen in table 2 and 3 (run 14) was inferior to the other outputs. In fact its COP was less than 1 meaning that the heat gained is less than the amount of power used. This is a very undesirable and unrealistic output which defies the basic advantage of using a heat pump in the first place. A further analysis of the system shows that the system underwent a complete cycle in the superheated state. Figure 5 presents the cycle in a T- S diagram as drawn by the simulating software. The fact that the refrigerant in this cycle didn't undergo any phase change also contributes to its inefficiency. The evaporator in the system under study was designed considering the properties of a wet vapor. When a superheated fluid is passed through it, definitely its efficiency will greatly reduce and thus the output. The same applies to the throttle valve and other system components. Furthermore, an operation in these conditions will end up in some components like the compressor underperforming and parts deteriorating.

Refrigerant charge is an important parameter that affects the performance of a heat pump system. There is an optimum charge at which the system yields the best COP. Performance deterioration is more severe at undercharged condition than at overcharged condition mostly because of the effects of less mass flow rates and reduced compressor efficiency. The CO_2 system performance is reported to be more sensitive to system refrigerant charge compared to that in conventional systems because of its low liquid to vapor density ratio [14]. Therefore, the refrigerant charge must be controlled more precisely in a CO_2 system than in other conventional systems to achieve high performance under various operating conditions. Charge optimization of the trans-critical CO_2 cycle is directly related with optimizing the gas cooling pressure to maximize the COP (or heating capacity) since the second derivative of the pressure with respect to the enthalpy at constant temperature for CO_2 becomes zero in supercritical region near the critical point where the gas cooler outlet condition is approaching. This characteristic would result in the gas cooler capacity increase in superior degree to that of the compressor power increase with the gas cooling pressure increase until the gas cooling pressure reaches to the pressure corresponding to this reflection point. Since the higher refrigerant charge will result in higher system operating pressures and the higher ambient temperature will result in higher optimum gas cooling pressure [15, 16].

Entropy generation can be used to analyze the second law efficiency for the trans critical CO_2 system under various charging conditions. Expansion loss is the dominant factor affecting system performance at undercharged condition while the gas cooler loss became the major parameter at overcharged conditions. The losses in the gas cooler increases as the amount of refrigerant charge increase due to the increase of the heat transfer rate. The enthalpy at the inlet of the evaporator significantly varies with a change of the gas cooler pressure; this change greatly varies the cooling capacity. Generally, the enthalpy at the exit of the gas cooler decreases with an increase of gas cooler pressure, resulting to lower quality at the inlet of the evaporator. Although the refrigerant temperature at the exit of the gas cooler gradually increase with the addition of refrigerant charge at undercharged conditions, the quality at the inlet of the evaporator decrease due to a reduction in the enthalpy at the exit of the gas cooler, thus increasing the cooling capacity. However, for overcharging conditions, the performance of the CO_2 system decreases with refrigerant charge because the enthalpy difference across the compressor increase more than that across the gas cooler. Furthermore, compression ratio decrease with the addition of the refrigerant charge; this decrease increase the refrigerant mass flow rate. Therefore the compressor power consumption slowly but continuously increases with normalized charge, while the cooling capacity rapidly increases at lower normalized charges and then the slope gradually decreases with an increase of the normalized charge[14].

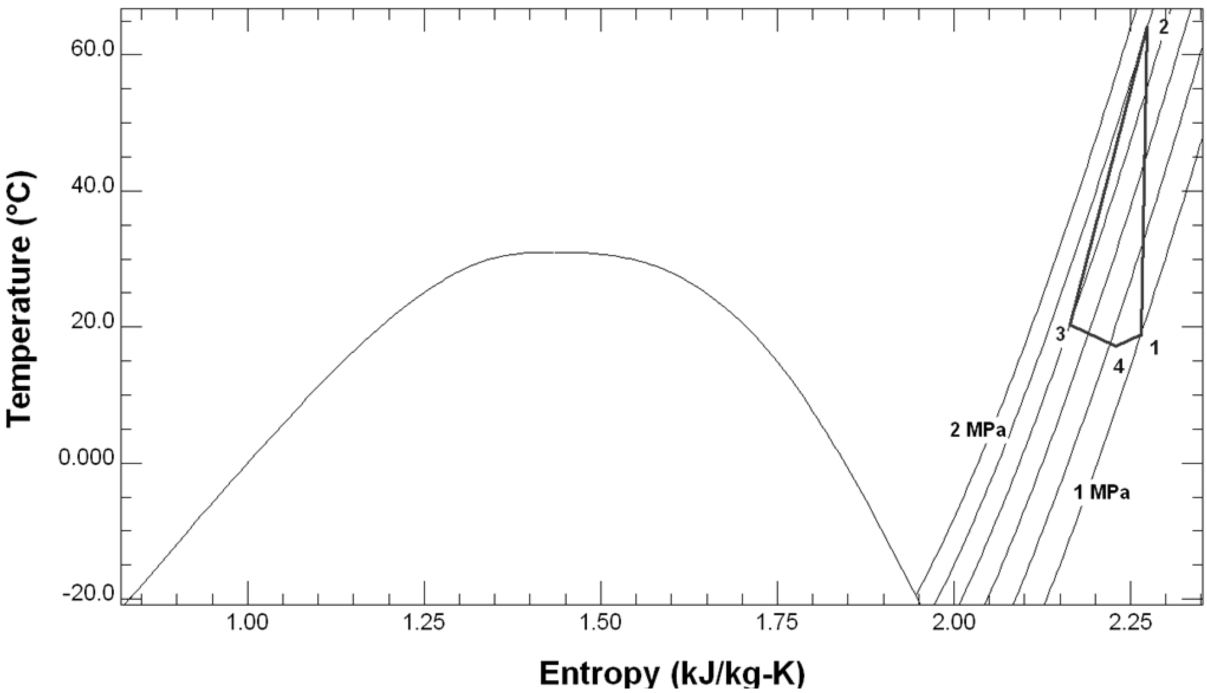

Figure 5: T-S diagram of the output values at less charge values.

4 Conclusions

A CO_2 trans critical water to water heat pump system was designed and utilized to produce typical output of such systems. The system had reasonable heating capacities and system efficiencies though there is room for improvement. At under-charged conditions however, the output and efficiency drastically reduced and the performance was highly undesirable. This result was also observed by other related studies and was related to effects of reduced mass flow rates and system components efficiency. Currently, optimization studies of the system are on progress where several system operation parameters are varied to yield the best heat output and efficiency of such a system.

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