A Simple Approach to Design and Fabricate an Efficient Heat Pump

Abid Karim[1], Shagufta Ishtiyaque[2], Faisal Afridi[3], Rabia Kaneez[4], Shaikh Kamaludin[1], Arif Karim[1], Aqeel Ahmed Khan[1], Ahmed Abouesayed[5], Abdul Rasheed Solungi[6], Faisal Ghanzanfar[1], Mansoor Hai[1], Mohammad Siddique[7]

[1]Applied Physics Computer and Instrumentation Centre, PCSIR Laboratories Complex, Karachi, Pakistan, [2]Department of Chemical Engineering, University of Karachi, Pakistan, [3]Institute of Space and Planetary Astrophysics, University of Karachi, Pakistan, [4]Department of Physics, COMSATS University Islamabad, Pakistan, [5]Physics Division, National Research Centre, Dokki Giza, Egypt, [6]Engineering Services Centre, PCSIR Laboratories Complex, Karachi, Pakistan, [7]Department of Chemical Engineering, Balochistan University of Information Technology, Engineering and Management Sciences, Quetta, Pakistan

Abstract

A low cost and indigenous experimental training, testing and process optimization setup of the heat pumping system is designed, developed and fabricated for the engineering students. The setup can be an effort to provide an indigenous option and import substitution for the local industries and academic institutions. The system uses difluoro-monochloromethane ($CHClF_2$) for the heat transport mechanism which has a heat of vaporization (ΔH_{vap}) of 233.95 kJ/kg at the boiling point of -40^{o}C. The control compression and expansion of the gas is performed by using the electro-mechanical controls and semi-hermetic compressor. The whole experimental setup is installed on the bench top table with vertical installation in open-air inside the laboratory. The change in pressure and temperature values at various stages of the setup is monitored by sensors, gauges, and visual displays. Physical states of the transport medium, i.e. liquid, vapors and super vapors are monitored by installing the transparent glass windows in the passage of transport medium. Hands-on experiments and procedures can be run on this setup in order to acquire deeper knowledge about design and process optimization of thermodynamical parameters, such as coefficient of performance (COP), Carnot cycle, dynamics of pressure-temperature imbalance during operating cycles and phase transformation of the transport medium under several operating conditions and parameters.

Keyword: Heat Pumping; Phases; Refrigerants; Coefficient of Performance (COP); Speed Controllers; Enthalpies; Compressors

Corresponding author's email: *shaguftai@uok.edu.pk*

INTRODUCTION

In normal circumstances, heat flows from hotter regions to a colder while the heat pump is a system which transfers heat from cold to hot body and creates low temperatures. The heat pumps are normally applied to most of the HVAC (Heating, Ventilation and Air-conditioning) system. The refrigerators and air conditioners are the most common examples and extensively used in our daily life and industries. Several air conditioning systems are designed to be operated in reverse and forward cycles and the same unit can be used for cooling and heating applications (Xu et al., 2011; Van Wylen et al., 1985; Wanget et al., 1999). Creating and stabilization of low temperatures is an important field of science and technology which had tremendous industrial applications and used extensively in our daily life. For example, several food chain restaurants and food suppliers are extensively utilizing the low temperature techniques in their cold storage for their food products and recipes. Several food technologies need freezing temperatures for preserving vegetables, meat items and cooked products which can be parboiled prior to the storage. This also helps to reduce the cooking and serving time and preserve the taste and beneficial healthy ingredients of the food (Borgenakke et al., 2013; Bejan, 2016). Furthermore, several unusual phenomena and actions are happening at low temperatures such as superconducting state, metal-to-insulator transitions and frictionless flow of liquid-helium. These exciting new phenomena are also exploited at the industrial scale and used widely for various applications in industries such as conducting infinitely large current from the superconducting wires and use it for creating huge magnetic fields for several scientific and industrial applications. Therefore, the well-trained engineers and technicians are in high demand in the field of heat transport system for industries as well as in scientific laboratories. Experimental setup for training and testing of heat pumping is so far rarely developed indigenously which can be helpful for the engineering students in order to acquire in-depth and hands-on training along with visual illustrations and their tuning control parameters (Karim, 2014; Baldassarre et al., 2012; Okamura et al., 2013; Pashkin et al., 2010; Munkejord et al., 2002).

MATERIALS AND METHODS

An experimental setup for training and testing of the heat pumping system was designed, developed and fabricated for the department of Chemical Engineering, University of Karachi. The simplified block diagram of the setup is shown in Fig. 1 in order to illustrate the working principle, measurement and sensor locations used in order to study their thermodynamical parameters. The snapshot of the experimental setup is shown in Fig. 2 during the process optimization and operational analysis. The setup comprises four basic building blocks (Figure 1), i.e. condenser, thermal expansion valve (TXV), evaporator and compressor. The condenser has coil of pipes which can be made of several materials depending on the type of transport medium and its operating pressure-temperature ranges. The Condenser is connected to the thermal expansion valve (TXV) which is used for the flow control of the transport medium to the evaporator. The evaporator consists of pipes tubing where the transport medium is able to evaporate due to the reduced pressure and absorbed heat from the environment and creates cooling effect (Frazzica et al., 2015; Karim, 2014; Zotter et al., 2015; Pearson et al., 2008; Xu et al., 2011). The compressor is considered a heart of the heat pumping system which is an electromechanical device, having an electric motor coupled to a suction pump and both units are assembled in a single block with the perfectly sealed environment. The electric motor drives the suction pump which sucks the transport medium (refrigerant) from the evaporator and discharges it to the condenser. In this experimental setup, a semi-hermetic compressor was used from the Copeland Corp. (C.C) as shown in Figure 2. The transport medium mostly called the refrigerant which is a gaseous medium that undergoes several phase transitions under various pressure and temperature conditions. In Figure 1 the hot state of the transport medium (refrigerant) is shown with red color and cold state is represented by green color. The gaseous state of the transport medium (refrigerant) is represented as small filled circles. The setup uses R-22 also called difluoro-monochloromethane ($CHClF_2$) for the heat transport mechanism which has a heat of vaporization (ΔH_{vap}) of 233.95 kJ/kg at the boiling point of -40^oC (Xu et al., 2011; Zotter et al., 2015; Chuanyang et al., 2013). The discharge pressure (Figfure. 1) of the compressor is needed to be high enough to transform the state of the transport medium from the gaseous to the liquid phase which appears as the release of heat energy (red color) from the refrigerant to the condenser pipes. The compressed liquefied transport medium allows to expand through the thermal expansion valve (TXV) in the evaporator section where the refrigerant starts to evaporate and absorbed heat energy from the evaporator pipes tubing. The evaporated transport medium from the evaporator is sucked again into the compressor and discharge out to the condenser after compression (Figure 1). Continuous operations of this process in a close loop cycle will gradually decrease the temperature of evaporator while the temperature of the condenser will be increased. Thus, by definition heat is pumped mechanically from the evaporator and discharges to the condenser (Van Wylen et al., 1985; Horn et al., 1976; Bejan, 2016). In this experimental setup, additional measurement and observational gauges / sensors are incorporated at several stages of the setup as indicated in Fig. 1. The Pressure (P), Temperature (T) at the input line of the condenser (C(I)) is measured as P1, T1. The values P2 an T2 are measured at the output line of the condenser represented as (C(O)). The Pressure (P), Temperature (T) at input line of the evaporator is represented as (E(I)) and measured as P3, T3. The values at output line of evaporator (E(O)) is measured as P4, T4. The temperature of the return gas to the compressor is measured as R-T5. The abbreviations of these experimental parameters are also indicated in the snapshot of the setup (Figure 2) where the temperature reader displays are marked as T1, T2, T3, T4, and R-T5. The measured values from these gauges and sensors are recorded in Table 1 and Table 2 which can be used for the calculations and process optimization.

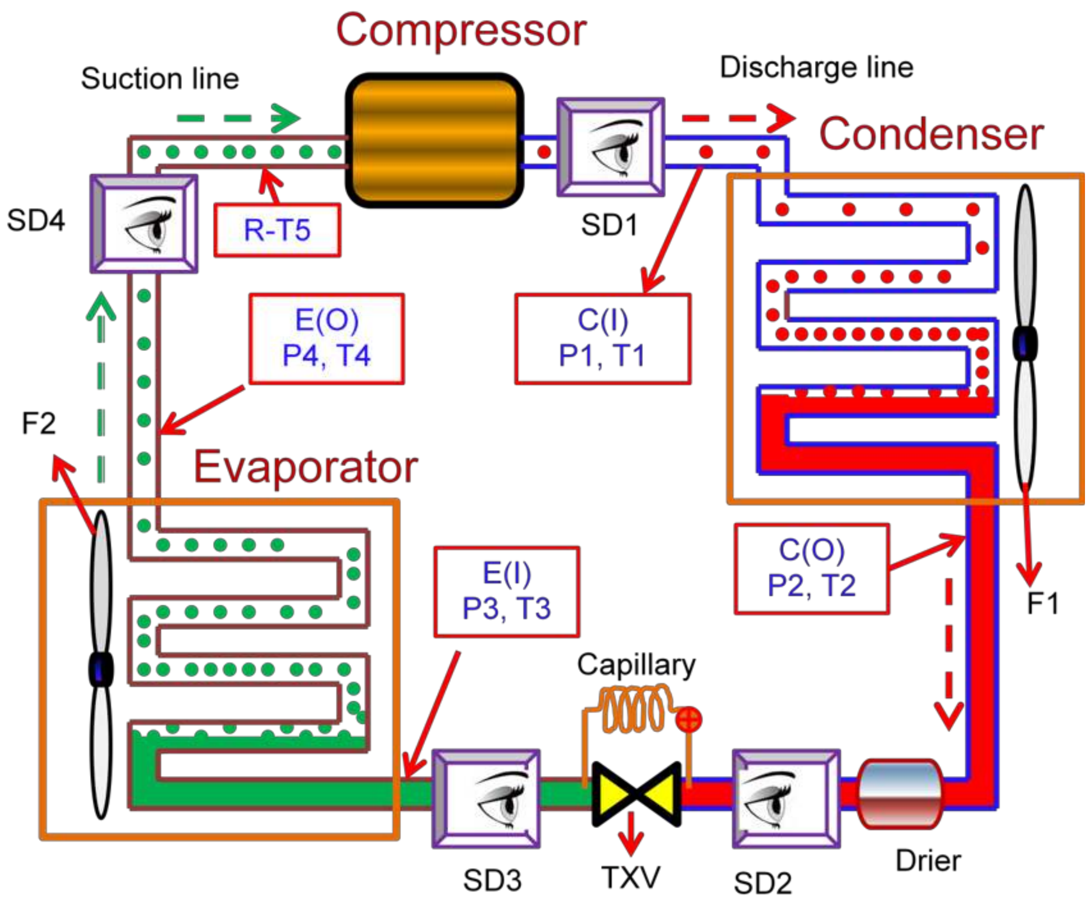

Figure 1: A simplified drawing of the experimental setup with the indicated locations where Pressure (P), Temperature (T) sensors and displays were incorporated. C(I) and C(O) represent Condenser and input and output put lines. E(I) and E(O) represent Evaporator input and output put lines. R – T5 is the temperature of the return gas to the compressor. SD represents the Sight Glass window to observe the physical state of the transport medium. F1 and F2 are the Condenser and Evaporator Fans. TXV represents the thermal expansion valve and the capillary is the coil made from copper tube.

Four sight glass windows (abbreviated as SD in Figure 1) are installed at several stages of the setup to observe the physical state of the transport medium such as vapors, liquid plus vapors, superheated vapors, and gas. The SD1 is installed at the discharge line of the compressor; SD2 is installed between liquid drier and the expansion valve; SD3 is incorporated at the input of the evaporator; SD4 is installed at the return of the compressor. Two fans represented (Figure 1) as F1 and F2 are installed at the back side of condenser and evaporator to control the flow of heat through / over them. For throttling effect two kinds of devices are used, one is an expansion valve (TXV) and the second is the capillary tube coil (Figure 1). The capillary tube coil is made from the copper pipe of diameter 1/8 inch and four coils. The transport medium only passes through either TXV or Capillary tube coil in order to understand their individual functions during the operating cycle. The routing of the transport medium can be changed through the manual valves. This feature helps to understand the design issues relating to the expansion valves and capillary tubes such as sizes, diameter, length and coils in connection with the pressure temperatures values and transport medium.

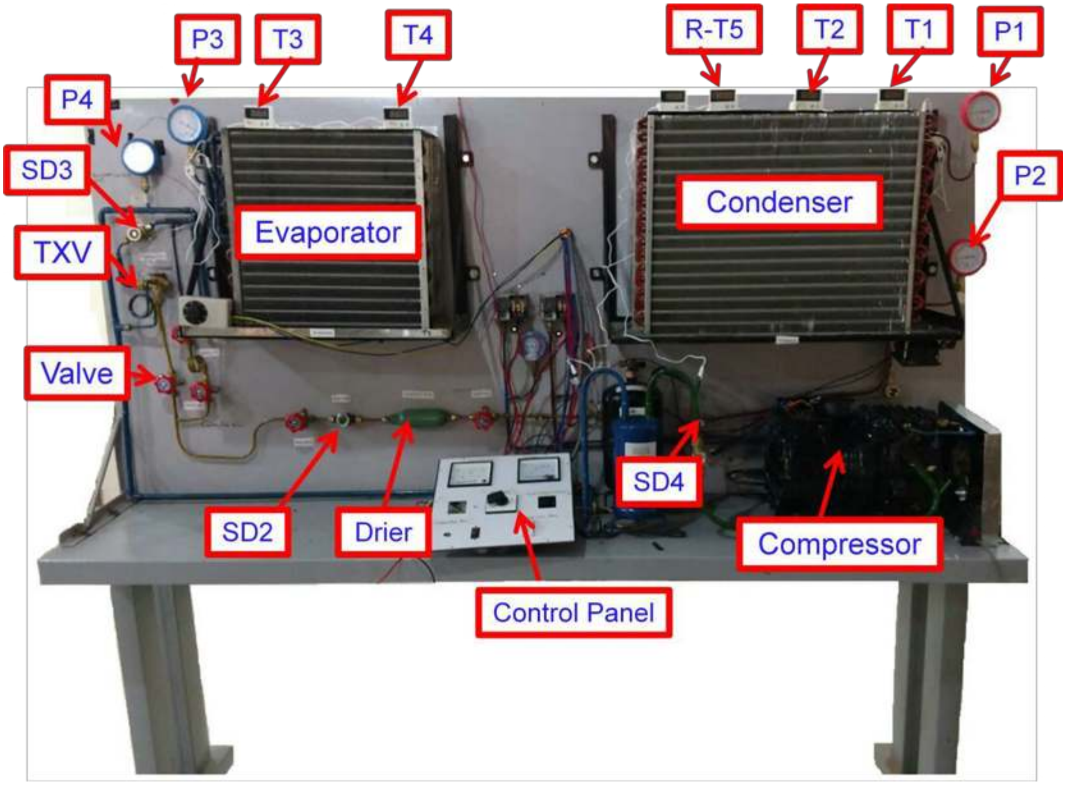

Figure 2: Snapshot of the experimental setup for the training and testing of the heat pumping systems. The measurement parameters and location of the pressure sensor and displays are shown according to Fig. 1. The location of temperature displays is shown only while their sensors are mounted according to the Fig. 1. The Fans, F1 and F2 are mounted behind the evaporator and condenser. The control panel consists of the electrical parameter displays with main ON/OFF switches, fans and temperature controller

RESULTS AND DISCUSSION

The measured values from the gauges and sensors from the setup (Figure 1) are measured and recorded in Table 1 and Table 2 during the operational analysis and design optimization. The pressure and temperature values; P1, P2, T1 and T2 are on the condenser side and P3, P4, T3 and T4 are data measured at the evaporator side. The data in Table 1 is recorded for the condition when the evaporator's fan F2 is powered-off. Table 2 represents the data recorded for the condition when the evaporator's fan F2 is rotating with maximum speed. Since, the setup is made for the training purpose, several important and critical parameters are left open for the student handlings and tuning. This ultimately increases the risk of damaging and failures for the setup and needed to be protected against any over tuned or wrongly sat parameters. For example, the condenser fan F1 should not be powered-off whenever the compressor is powered on otherwise the overheated refrigerant can lead to increase head pressure over the compressor and can cause damages. Therefore, to protect the setup, several electrical / electronic circuits and interlocks were installed and interfaced such as, over current protection, head pressure release interlocks, temperature controlling and compressor restarts delay timers etc. In Table 1 The 'status and Time' tag represent the initial condition of the setup, 'Read at Time' tag represents the time when the measurements were recorded. 'Set point' tag is the set temperature at the thermostat connected to the output line, i.e. T4. The locations of sensors for measuring pressure (P1, P2, P3 and P4) and temperatures (T1, T2, T3, T4, R-T5) is explained in Figure 1. For the condition when the experimental setup was powered-off for more than 24 hours, the input and out pressure and temperature values equalize to the similar values. The pressures P1, P2, P3 and P4 were normalized to 120 Psi and the temperatures T1, T2, T3 and T4 were reached at room temperature (297.0 K). These are the starting pressure and temperature conditions which will be manipulated during the heat pumping cycles. At time 10:38 the setup was powered-on and the measurements were recorded at 10:44, where P1 and P2 were increased to 175 Psi and 160 Psi.

Table 1: Operating temperatures and pressure measurements of the experimental setup at the various stages of the system at the condition when the evaporator fan (F2) is rotating at maximum speed

Evaporator fan (F2) is rotating at maximum speed											
Status and Time	**Read at Time**	**Condenser Side**				**Evaporator Side**				**Set Point** (K)	**Remarks**
		P1 (Psi)	P2 (Psi)	T1 (K)	T2 (K)	P3 (Psi)	P4 (Psi)	T3 (K)	T4 (K)		
OFF	10:30	120	120	297.0	297.0	120	120	297.0	297.0	OFF for > 24 hrs	
ON at 10:38	10:44	175	160	333.7	300.6	28	25	267.2	255.0	283	Running
	10:57	178	162	350.8	320.2	30	25	266.7	255.2	//	**T4** is not decreasing
OFF at 11:00	11:00	150	140	320.8	302.0	30	25	267.6	255.8	//	
	11:08	135	125	298.6	299.1	120	120	293.8	296.6	//	
ON at 11:14	11:20	175	160	351.3	320.2	27	26	268.8	254.9	//	**T4** is not decreasing
	11:28	180	170	355.3	321.6	29	28	269.3	255.2	//	
	11:35	185	175	357.5	323.4	31	29	268.9	255.0	//	

In contrary, at the evaporator side P3 and P4 were reduced to 28 Psi and 25 Psi because of expansion of the transport medium which creates an imbalance in the temperatures of the setup. Meanwhile, the temperature at the condenser side was increased to 333.7 K (T1) and 300.6 K (T2) while at the evaporator side the temperatures were decreased to 267.2 K (T3) and 255.0 K (T4) as shown in Table 1 (at read time 10:44). The temperature, T4 was monitored by a control system (thermostat) which was continuously sensing and comparing the preset, i.e. 283.0 K (10°C) at the thermostat. It was observed that the set point at T4 was not able to achieve the conditions presets for the measurement cycles recorded in Table 1. The compressor was powered-off at 11:00 for releasing the head pressure and restarted after fourteen minutes and the measurements were recorded at 11:20, 11:28 and 11:35, but the temperature at T4 was not going deeper down than 255.0 K. In fact, the input line temperature (T3) of the evaporator is 267.2 K which is much lower than T4 but the force air flow of F2 through the evaporator prevents the temperature from going deeper down. It can also be seen that the pressure values P1 and P2 increases due to the increase in temperature at the compressor head and longer operating time. Table 2 is shown for the experimental data recorded for the condition when the fan F2 was powered-off and condenser fan was rotating at maximum speed. At 11:35, pressure and temperature values show a similar reading as recorded in Table 1 for the off condition for more than 24 hours. As the system was switched ON at 11:39 the P1 and P2 increased to 175 Psi and 170 Psi and T1 and T2 increased to 347.1 K and 299.5 K the set point of 295 K and the achieved in less than thirty seconds and the automatic control system stop the compressor at 11:40.

Table 2: Operating temperatures and pressure measurements of the experimental setup at the input and output lines of the condenser and evaporator. The evaporator fan (F2) is OFF, i.e. 220 VAC is completely disconnected

Evaporator Fan (F2) is OFF											
Status	Read at Time	Condenser Side				Evaporator Side				Setup Point (SP)	Remarks
		P1 (Psi)	P2 (Psi)	T1 (K)	T2 (K)	P3 (Psi)	P4 (Psi)	T3 (K)	T4 (K)		
OFF	11:35	120	120	297.0	297.0	120	120	297.0	297.0	OFF for > 24 hrs	
ON at 11:39	11:39	175	170	347.1	299.5	31	28	268.2	288.5	295	**SP** achieved in less than 30 seconds
OFF at 11:40	11:50	150	112	298.0	300.9	118	116	293.6	295.4	//	**T4** raise slowly
ON at 12:34	12:34	175	160	339.4	320.2	25	21	269.1	280.6	283	**SP** achieved in less than 60 seconds
OFF	12:35	100	75	324.7	296.4	90	90	289.5	287.2	//	**T4** raise slowly
ON at 01:00	01:00	170	150	327.6	319.5	20	20	266.4	271.0	273	**SP** achieved in less than 60 seconds
OFF at 01:01	01:02	75	60	316.2	307.5	85	84	288.1	285.0	//	**T4** raise slowly

According to the phase diagram of the transport medium [difluoro-monochloromethane] ($CHClF_2$)) used in the setup, the superheated state occurs in the vicinity of 283.1 K and 98.6 Psi. The set point (SP) was decreased to 283 K and the setup was powered-on at 12:34 and measurements were recorded in Table 2. The set point temperature 283 K was achieved within sixty seconds, and the superheated vapor state was observed in SD3. The set point (SP) temperature was further decreased to 273 K, which was achieved within sixty seconds due to the stop force air circulation over the evaporator (Table 2 and Figure 1). Also, the temperature difference between T3 and T4 was decreased and the temperature raises quite slowly when the compressor was turned-off by thermostat (Bejan, 2016; Zarrella et al., 2016; Frazzica et al., 2015). The performance efficiency of the heat pumping system is defined by the term ‘Coefficient of Performance (COP)’ is instead of the general term ‘efficiency’. The COP is generally greater than 100 % due to the fact that they are moving heat rather than creating heat. Also, the solid-state cooling devices such as Peltier’s module consumed a huge amount of electrical energy to create the same amount of cooling effect. So, at the moment, heat pumping systems are considered best for heating and cooling applications. Mathematically, the COP can be defined as follows (Pons et al., 1999; Zotter et al., 2015; Fonyo et al., 1996).

$$\mathrm{COP_{Carnot}} \leq \frac{\text{Heat transferd } (Q_H)}{\text{Electrical Energy (W)}} \tag{1}$$

Since, the maximum efficiency is limited by the Carnot Engine, so the equation (1) can be rewrite as follows.

$$\mathrm{COP_{max,\ Carnot}}\ (\text{Heating}) \leq \frac{Q_H}{Q_H - Q_C} \leq \frac{T_H}{T_H - T_C} \tag{2}$$

Where T_C or Q_C is the amount of heat extracted and T_H or Q_H is the amount of heat exhausted while in the case of cooling application the amount of energy extracted is important, which is the temperature of the colder side of the heat pump. The equation (2) can also be rewrite for the cooling application follows.

$$\mathrm{COP}_{\mathrm{max,\ Carnot}}\ (\mathrm{Cooling}) \leq \frac{T_C}{T_H - T_C} \tag{3}$$

In the case of our experimental setup, the situation is different because the system was mounted on a bench top table in the open-air inside the same room. Also, the heat exhausted, and heat absorbed were taken and release in the same environmental space. By referring to Table 1 where the setup was OFF for more than 24 hours, all temperatures (T1, T2, T3, T4) and pressure (P1, P2, P3, P4) are the same which means T_H - T_Cà 0; the equation (3) will give, COP à ∞. This implies that the compressor does not need any electrical power for the heat transport mechanism. On the other hand, if the inside and outside temperatures are too high, i.e. T_H - T_Cà ∞; then equation (3) will give COPà 1. Hence, the two extreme limits for the COP of the heat pumping systems will lie between 1 to ∞. For the condition when setup was powered-on at time 10:44 as shown in Table 1, where the set point is 282 K but the minimum temperature achieved at T4 is 255.2 K due to the force ventilation of air over the evaporator. In this case, the compressor needs to work for an infinite time in order to achieve the desired (set point) temperature. Therefore, in this case, the temperature of the evaporator and condenser can be considered as two separate units where fans F1 and F2 can be used to simulate weather conditions. Table 2 measurements were recorded for the case when the fan F2 was powered-off and the setup was operated for three set point temperatures, i.e. 295 K, 283 K, 273 K which were achieved within a minute. The COP for these set point temperatures can be calculated from Table 2 as follows.

$$\mathrm{COP}_{\mathrm{max}}\ (\text{at } 295\ \mathrm{K}) \leq \frac{T4}{T1 - T4} \leq \frac{288.5}{347.1 - 285.5} \leq \frac{288.5}{61.6} \leq 4.68 \tag{3}$$

$$\mathrm{COP}_{\mathrm{max}}\ (\text{at } 283\ \mathrm{K}) \leq \frac{T4}{T1 - T4} \leq \frac{280.6}{339.4 - 280.6} \leq \frac{280.6}{58.8} \leq 4.77 \tag{4}$$

$$\mathrm{COP}_{\mathrm{max}}\ (\text{at } 273\ \mathrm{K}) \leq \frac{T4}{T1 - T4} \leq \frac{271.0}{327.6 - 271.0} \leq \frac{271.0}{56.6} \leq 4.78 \tag{5}$$

The calculated COP as shown in equations (2), (3) and (4) indicates a gradual increase because of the optimized pressure and temperatures due to the previous cycle of operations. However, the difference between the 'set point temperature' and the 'achieved temperature' (T4) is because of the using ON/OFF type temperature controller is used for temperature controlling. These types of controllers have larger error bars due to the inertial effects during cooling and heating cycles. In fact, PID (Proportional Integral Derivative) controllers can give precise controlling over the temperature ranges, but not well suited for the life span and efficiency of the traditional compressors, which was also used in this experimental setup. In sophisticated scientific instruments where accuracy is more important than efficiency and large volume-size, mostly Peltier modules (solid state device) are used which has no moving parts and required less maintenance. Also, they can easily be controlled with electronics interface circuits along with PID controllers (Xu et al., 2011; Dinc et al., 2011; Granet, 2015).Recently, the market is flooded with inverter type air-conditioning systems which are basically designed to control the speed of the compressor's motor in order to control precisely compression and suction cycles. The use of the speed controllers to the compressor motors decreases the consumption of electrical power substantially, i.e. by 10 to 50 %. In fact, the initial costs for these systems are much higher due to the installation of sophisticated electronics circuits to achieve optimized performance and efficiency. Also, traditionally trained technicians and engineers are not able to handle and perform routine repair/maintenance of these systems and huge well-trained manpower is required in this regard (Xu et al., 2011; Micallef, 2014; Struchtrup, 2014).

CONCLUSION

In this experimental work, an effort was made to design, develop and fabricate a low-cost experimental setup for training, testing and process optimization of the heat pumping and transport systems for the undergraduate chemical engineering students. The dynamics of the pressure, temperature imbalance was observed in relation with the phase transformation of the transport medium. The experimental setup can also be upgraded by using the sophisticated electronics and control circuits in order to extend its training capability according to the latest technology as adopted in the inverter-type air-conditioning systems.

REFERENCES

- Xu X, Hwang Y, Radermacher R. (2011). Refrigerant injection for heat pumping/air conditioning systems: literature review and challenges discussions. International Journal of Refrigeration 34(2): 402-415.
- Pearson MR, Seyed-Yagoobi J. (2008). Experimental study of a two-phase heat transport device driven by electrohydrodynamic conduction pumping. In ASME 2008 International Mechanical Engineering Congress and Exposition. American Society of Mechanical Engineers, pp. 1877-1883.
- Van Wylen GJ, Sonntag RE. (1985). Fundamental of Classical Thermodynamics. 3rd Edition, John Wiley & Sons, New York.
- Zotter G, Rieberer R. (2015). Experimental analysis of a novel concept of a “thermally driven” solution pump operating a small-capacity ammonia/water absorption heat pumping system. International journal of refrigeration 60:190-205.
- Fonyo Z, Benko N. (1996). Enhancement of process integration by heat pumping. Computers & chemical engineering 20:85-90.
- Karim A. (2014). Electronic correlations in $SrFe_{2-x}Co_xAs_2$ pnictides and EuB_6 probed by infrared spectroscopy under high–pressure and low–temperatures. Ph.D thesis, Augsburg University, Germany.
- Baldassarre L, Perucchi A, Postorino P, Lupi S, Marini C, Malavasi L, Jiang J, Weiss JD, Hellstrom EE, Pallecchi I, Dore P. (2012). Electrodynamics of $BaFe_2As_2$ from infrared measurements under pressure oxides. Physical Review B 85(17): 174522.
- Okamura H, Shoji K, Miyata K, Sugawara H, Moriwaki T, Ikemoto Y. (2013). Pressure suppression of spin-density-wave gap in the optical conductivity of SrFe2As2. Journal of the Physical Society of Japan 82(7):074720.
- Pashkin A, Dressel M, Hanfland M, Kuntscher CA. (2010). Deconfinement transition and dimensional crossover in the Bechgaard-Fabre salts: Pressure-and temperature-dependent optical investigations. Physical Review B 81(12): 125109.
- Borgenakke C, Sonntag RE. (2013). The Second Law of Thermodynamics, in Fundamentals of Thermodynamics. 8th Edition, pp 244-245.
- Su CY, Chen G, Zhang Q. (2013). Experimental Study on Pumping and Recharging of Groundwater-Source Heat Pump System in Fuzhou Basin. In Advanced Materials Research 805:574-579.
- Dinc I, Kanoglu M. (2010). Refrigeration Systems and Applications. 2nd Edition. John Wiley & Sons.
- Wang SK. (1999). Air-conditioning and Refrigeration, Mechanical Engineering Handbook. McGraw-Hill Global Education Holdings, LLC.

- Horn JF, Scharf PH. (1976). Design consideration for heat pump compressors. International Compressor Engineering Conference, at Purdue University, Indiana, USA.
- Bejan A. (2016). Advanced Engineering Thermodynamics, 4th Edition. John Wiley & Sons.
- Zarrella A, Emmi G, De Carli M. (2017). A simulation-based analysis of variable flow pumping in ground source heat pump systems with different types of borehole heat exchangers: A case study. Energy Conversion and Management 131: 135-150.
- Micallef D. (2014). Fundamental or refrigeration thermodynamics chrome. Foundation of refrigeration of thermodynamics. 1st Edition, pp. 1-168.
- Frazzica A, Briguglio N, Sapienza A, Freni A, Brunaccini G, Antonucci V, Ferraro M. (2015). Analysis of different heat pumping technologies integrating small scale solid oxide fuel cell system for more efficient building heating systems. Elsevier; International Journal of Hydrogen Energy 40:14746 -14756.
- Granet I. (2015). Thermodynamics and Heat Power. 8th Edition, CRC Press by Taylor & Francis Group, LLC.
- Pons M, Meuniera F, Cacciolab G, Critophc RE, Grolld M, Puigjanere L, Spinnerf B, Ziegler F. (1999). Thermodynamic based comparison of absorption systems for cooling and heat pumping. Elsevier; International Journal of Refrigeration 22: 5-17.
- Struchtrup H. (2014). Thermodynamics and Energy Conversion. Springer-Verlag Berlin Heidelberg, Germany.
- Munkejord ST, Maehlum HS, Zakeri GR, Neksa, P, Pettersen J. (2002). Micro technology in heat pumping systems. Elsevier; International Journal of Refrigeration 25: 471–478.