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RESEARCH ARTICLE

Experimental Investigation of the Effects of Temperature and Relative Humidity on Performance of a Heat Pump

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ABSTRACT

This study was aimed at investigating the effects of temperature and relative humidity of atmospheric air on the exergy efficiency and thermodynamic efficiency of a heat pump. For this purpose, an experimental rig was set up and experiments were carried out. The results obtained were put through exergy calculations. Two heat pumps, heaters, and steam engines were employed in the experimental rig. The first heat pump was used for drying and cooling the air. Then, heat and water vapor were imparted to the air in the channel in order to get it to have desired humidity and temperature. The effects of this conditioned air on the second heat pump were determined by temperature, humidity, pressure, and flow rate measurements. Exergy efficiencies and thermodynamic efficiency ratios of the system and its components were determined with the measured values. The COPCM value of the system was observed to decrease with the increase of the relative humidity in the air. The highest COPCM value was at lowest relative humidity values of 30% and 40%. It was graphically shown that the COPCM value remained at low levels when the relative humidity rose to 70% - 80%.

Contents

1. Introduction

Heat pump systems have been in service in heating, cooling, drying, cleaning the ambient air, and condensing water from the air. The humidity and temperature of air are among the factors that affect the efficiency of heat pump systems [1]. Among numerous studies that have been carried out on heat pumps, solar assisted heat pumps have gained popularity in recent years. Kılıç [2] examined the performance of the solar assisted heat pump system of which a 1m2 solar collector consisting of black painted pipes served as the evaporator of the heat pump, and has reported that the system's COP could

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reach 3.79 in the system where R-404a was used as the refrigerant.

Drying/Desiccating with heat pumps has been applied in many fields. For instance, Mirza [3] designed a heat pump system to remove humidity from the ambient air, and thus desiccate leather. Using a computer simulation, Mirza [3] analyzed the efficiency of the heat pump elements and the COP of the system when the desiccation process reached the desired level.

Another drying process by removing the ambient relative humidity was carried out by Fatouh et al. [4] in Egypt, which has a distinct climate. In the study where R134a was used as the refrigerant to dry herbs such as hibiscus, parsley and mint, the thermodynamic performance of the system was presented with graphics.

Drying can be defined as a process of removing water content –mostly- from solids, other liquids or gases. So, a drying process generally refers to the removal of water vapor from solids, which is achieved by removing the relative humidity in the ambient air where the material to be dried is located and/or by blowing hot gases through them. In this way, heat and mass transfer is carried out by radiation and convection from the contacting surface of the material. Air properties in the environment are among the most effective parameters in drying [5].

Ceylan et al. [6] used a heat pump desiccation system in drying timbers and reported that the system was able reduce the moisture content in the poplar and pine timbers from 1.28 kg water / kg dry matter and 0.60 kg water / kg dry matter, respectively, to 0.15 kg water / kg dry matter within a 24-hour period. In a heat pump assisted kiln system, on the other hand, they were able to reduce the moisture in the poplar timbers from 1.28 kg water/kg dry matter to 0.15 kg water/kg dry matter in 70 hours, and for the pine timbers the time required to reduce the moisture content from 0.60 kg water /kg dry matter to 0.15 kg water /kg dry matter was 50 hours.

Kara [7] addressed a dehumidification of an air conditioning system designed to reduce the humidity of fresh air sent to a conditioned environment by passing it over a dehumidifier first and then cooling down by a vapor compression refrigeration cycle. The also coded a computer program to calculate the total heat transfer and energy loss during said conditioning of the air. In another study by Kılıç [8], the heating and cooling performances of a ground source heat pump system were investigated. The effects of cooling, dehumidification and ventilation parameters on the system and its components were revealed by psychrometric analysis. They also investigated the effects of summer and winter periods on the system.

As Turkey is a large country with different climatic characteristics, air-conditioning systems are subject to different operating loads according to the regional conditions. Altın et al. [9]compared different air-conditioning appliances for different regions of Turkey in order to provide the comfort conditions accepted by ASHRAE. They presented on the psychrometric diagram their analyses done based on the meteorological statistics of several provinces with different climatic characteristics such as Ankara, Istanbul, Izmir, Erzurum, Antalya and Urfa.

Using Ansys Fluent, Günay [10] numerically analyzed the heat and mass transfer during the condensation of moist air. Further, the momentum, energy and mass transfer were also analyzed in an experimental rig set up for the purpose by getting the air pass between two plates in laminar flow conditions and the effects of relative humidity, velocity and temperature parameters on heat transfer and temperature variation were investigated by changing the air input conditions. The increase in relative humidity was reported to have an insignificant effect on the sensible heat but that it had a significant effect on the latent heat.

Alkan et al. [11] investigated an air conditioning system of a car and examined the performances of R1234yf and R134a refrigerants under same conditions by using special measurement devices and also carried out exergetic analysis with the data. Zhang et al. [12] tried to optimize a heat pump dehumidification system through exergetic analysis, redesigning the system every once they detect exergy destruction on the heat pump elements. Chen et al. [13] also carried out an exergy analysis of a solar-powered heat pump and reported that the efficiency increased by 29.5% and 25.9% with solar energy assistance. While the exergy efficiency of the system changed in direct proportion to the solar irradiance, the payback time for the solar energy add-on was reported to be 8 years. Al Sayyab et al. [14] made a comparison to reveal the effects of R450A, R513A, R515A, R515B, R516A, R152a, R444A, R1234ze(E), R1234yf, R290 and R1243zf refrigerants on the exergy efficiency in a heat pump system and their respective COP, exergy efficiency, exergy destruction and electricity consumption were reported.

When the studies are examined, the climatic conditions can be seen to have a significant effect on the performances of heating and cooling systems. As a result of the literature survey, it was seen that the effects of climatic conditions on heating and cooling methods were partially analyzed. Climatic conditions, generally considered as the natural environment where the systems are located/installed, involve several parameters such as relative humidity, temperature and air velocity that may change partially but almost constantly. As most of the studies are kind of statistical or analytical/numerical, it would be beneficial to measure and analyze the effects of the changes in climatic conditions on the systems more clearly through experimentation. And, it is regarded as necessary that the analyzes in question be carried out individually for efficiency, energy, exergy and entropy, for the overall efficiency of the whole system.

2. Material and Method

Within the scope of this study, an experimental system employing two heat pumps was set up in the energy laboratory of Engineering Faculty at Ataturk University. The temperature and relative humidity of the air taken from the outside were reduced in the first heat pump and the measurements were made in the second. A schematic representation of the experimental system and its general view can be seen in Figure 1 and Figure 2.

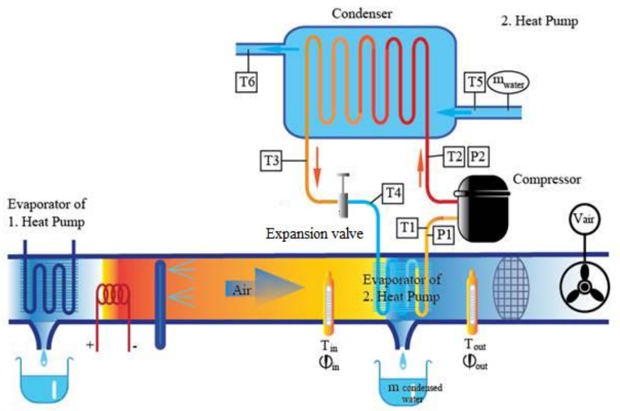

Figure 1 Schematic representation of the experimental setup [15]

Figure 2 General view of the experimental setup [16]

Thermocouples were attached on the second heat pump, so as to be in contact with the fluid, to take temperature measurements. The locations where temperature and pressure measurements for the refrigerant were taken are listed in Table 1.

Table 1 Temperature and pressure measurement locations [16]

Measurement	Definition / Location
T1	Refrigerant temperature at compressor input / evaporator output
T2	Refrigerant temperature at compressor output / condenser input
T3	Refrigerant temperature at condenser output / expansion valve input
T4	Refrigerant temperature at expansion valve output / evaporator input
T5	Water temperature at condenser input
T6	Water temperature at condenser output
P1	Low pressure (bar)
P2	High pressure (bar)

2.1. Coefficient of Performance (COP)

The ratio of the heat energy supplied to the warm environment to the electrical energy used in the heat pumps is called the Coefficient of Performance (COP) [1] as is also valid for cooling machines, and COP is expressed as:

$$COP_{HP} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp}} \tag{1}$$

where $\dot{W}_{comp}$ is the electrical energy consumed by the compressor and $\dot{Q}_{cond}$ the heat dissipated from the refrigerant to the water in the condenser. As for the cooling machines, COP is the ratio of dissipated heat energy to the electrical energy consumption, which is:

$$COP_{CM} = \frac{\dot{Q}_{evaporator}}{\dot{W}_{compressor}}$$
or
$$COP_{CM} = \frac{\dot{Q}_{evaporator}}{\dot{W}_{compressor,elect}} \tag{2}$$

The electrical energy consumption by the compressor can be obtained using:

$$\dot{W}_{comp} = \dot{m}_{refrigerant}(h_o - h_i)$$
or
$$\dot{W}_{comp} = I.V.Cos(\phi).\sqrt{3} \tag{3}$$

where h_i and h_o are the enthalpy values of the refrigerant at input and output, respectively. In the alternative equation, I denotes the electrical current in the compressor, V the voltage, and $Cos(\phi).\sqrt{3}$ the power factor.

The heat dissipated from the refrigerant to the water in the condenser is obtained for the refrigerant and water by using the below equations, respectively [1]:

$$\dot{Q}_{cond} = \dot{m}_{refrigerant}(h_o - h_i)$$
$$\dot{Q}_{cond} = \dot{m}_{water} C_{p,water}(T_{water,out} - T_{water,in}) \tag{4}$$

where $\dot{Q}_{cond}$ denotes the heat transferred from the refrigerant to the water, $\dot{m}_{refrigerant}$ and $\dot{m}_{\mathrm{water}}$ the mass flow rates of refrigerant and water, h_i and h_o the enthalpy values of the refrigerant at input and output of the condenser, $C_{p,water}$ the specific heat of water at constant pressure, and $T_{water,out} - T_{water,in}$ the water temperatures at the input and output of the condenser.

Similarly, the equations for the evaporator are as follows [1]:

$$\dot{Q}_{evaporator} = \dot{m}_{refrigerant}(h_o - h_i)$$
$$\dot{Q}_{evaporator} = \dot{m}_{air} C_{p,air}(T_{air,out} - T_{air,in}) \tag{5}$$

where $\dot{Q}_{evaporator}$ denotes the heat transferred from the air to the refrigerant, $\dot{m}_{refrigerant}$ and $\dot{m}_{air}$ the mass flow rates of the refrigerant and air, h_i and h_o the enthalpy values of the refrigerant at input and output of the evaporator, $C_{p,air}$ the specific heat of air at constant pressure, and $T_{air,out} - T_{air,in}$ the air temperatures at the input and output of the evaporator.

2.2. Air Conditioning Thermodynamics

One of the three basic factors affecting human comfort is the specific humidity of the air, i.e. the amount of water vapor in the air. Air that contains water vapor is defined as atmospheric air whereas the air that does not contain water vapor is defined as dry air. The equations regarding the enthalpy of dry air are as follows [1]:

$$h_{dry\ air} = c_p T = \left(1{,}005\,\frac{kJ}{kg}\,.\,°\text{C}\right)\,T\ (\text{kJ/kg})$$
$$\Delta h_{dry\ air} = c_p \Delta T = \left(1{,}005\,\frac{kJ}{kg}\,.\,°\text{C}\right)\Delta T(\text{kJ/kg}) \tag{6}$$

where T is the current temperature of the air in °C, and ΔT the temperature difference. Air-conditioning studies generally focus on enthalpy changes and the water vapor is assumed to be an ideal gas since this assumption offers convenience and ease at calculations. Thus, the atmospheric air is regarded as a combination of two gases: water vapor and dry air [1].

In this respect, the atmospheric pressure consists the partial pressure of dry air, P_a, and the partial pressure of water vapor, P_v:

$$P = P_a + P_v(\text{kPa}) \tag{7}$$

The enthalpy value for water vapor, on the other hand, is obtained with the following equation [1]:

$$h_g = 2500{,}9 + 1{,}82\,T(\text{kJ/kg}) \tag{8}$$

where h_g is the enthalpy of the water vapor in the air while T denoting the temperature.

The ratio of the mass of water vapor to the mass of dry air, in unit mass of atmospheric air, is defined as specific humidity (ω) is expressed as [1]:

$$\omega = m_v/m_a(\text{kg water vapor/kg dry air}) \tag{9}$$

Air has a humidity capacity, and when the water-vapor content that the air can hold, determined by temperature, is exceeded for a given temperature, the excess vapor becomes a liquid by condensing immediately. In this respect, the ratio of the current mass of water vapor in the air to the maximum mass of water vapor it can hold, is called relative humidity (φ) and is expressed as:

$$\varphi = m_v/m_g \tag{10}$$

Expressing the masses in the terms of ideal gas equation and simplifying identical terms, the relative humidity can also be obtained as a ratio of partial pressures:

$$\varphi = P_v/P_g \tag{11}$$

where P_v denotes the partial pressure of the moisture present in the air, and P_g the partial pressure of saturated air, which can be read from the charts/tables, noting that the P_g is equal to the saturation pressure at given temperature:

$$P_g = P_{sat@T}\ (\text{kPa}) \tag{12}$$

In this respect, relative humidity can be expressed as:

$$\varphi = \omega P/(0{,}622 + \omega)P_g \tag{13}$$

As for the enthalpy calculation equations;

$$H = H_a + H_v = m_a h_a + m_v m_v$$
$$h = \frac{H}{m_a} \tag{14}$$
$$h = h_a + (m_v/m_a)\,h_v = h_a + \omega h_v$$

wherefrom h is obtained as:

$$h = h_a + \omega h_g(\text{kJ/ kg dry air}) \tag{15}$$

were, h_a is the enthalpy of dry air and h_g the enthalpy of water vapor in the air.

In accordance with the first law of thermodynamics, the effect of water vapor in the air on the energy balance:

$$\dot{E}\text{g} = \dot{E}\text{ç}\ (\dot{Q} = 0 \text{ ve } \dot{W}\text{ç} = 0\ )$$
$$\dot{m}_a h_1 + \dot{m}_f h_{f2} = \dot{m}_a h_2 \tag{16}$$
$$\text{or}$$
$$\dot{m}_a h_1 + (\omega_2 - \omega_1)h_{f2} = \dot{m}_a h_2$$

making necessary simplifications, the equations become:

$$h_1 + (\omega_2 - \omega_1)h_{f2} = h_2$$
$$(c_p T_1 + \omega_1\, h_{g1}\,) + (\omega_2 - \omega_1)h_{f2} \tag{17}$$
$$= (c_p T_2 + \omega_2 h_{g2}$$

where, subscripts 1 and 2 represent the first and second state values. From this, the specific humidity can be expressed as:

$$\omega_1 = \frac{(cp\ (T2 - T1)\ +\ \omega 2hfg2)}{hg1 - hf1} \tag{18}$$

assuming φ_2 to be 100%, the equation becomes [1]:

$$\omega_2 = \frac{0{,}622\ Pg2}{P2 - Pg1}(\text{kg water vapor/kg dry air}) \tag{19}$$

2.3. Exergy Efficiency

Exergy efficiency is the ratio of the work potential that a system has when operating under real conditions to the work that the system can do until it becomes dead where no exergy is lost. In other words, the ratio of the work actually done by the system to the maximum work it can do is called exergy efficiency [17] and is expressed as:

$$\psi = \frac{\dot{E}x_{out}}{\dot{E}x_{in}} \tag{20}$$

where, $\dot{E}x_{in}$ denotes the exergy entering the system, and $\dot{E}x_{out}$ the exergy leaving the system. Thereby, the exergy efficiency for the compressor is:

$$\psi_{ex.comp} = \frac{\dot{E}x_{out} - \dot{E}x_{in}}{\dot{W}_{comp,elect}}$$
$$\psi_{ex.comp} = \frac{\dot{E}x_{out,refrigerant} - \dot{E}x_{in,refrigerant}}{\dot{W}_{comp,elect}} \tag{21}$$

Exergy efficiency of heat exchangers (i.e. condenser and evaporator) can be expressed as:

$$\psi_{ex,heat\ exchanger} = \frac{\dot{E}x_{out,cold} - \dot{E}x_{in,cold}}{\dot{E}x_{in,hot} - \dot{E}x_{oou,hot}}$$
$$\psi_{ex,heat\ exchanger} = \frac{\dot{m}_{cold}(\dot{e}x_{out,cold} - \dot{e}x_{in,cold})}{\dot{m}_{hot}(\dot{e}x_{in,hot} - \dot{e}x_{out,hot})} \tag{22}$$

or they can be expressed separately as follows:

$$\psi_{cond} = \frac{\dot{E}x_{out,water} - \dot{E}x_{in,water}}{\dot{E}x_{in,refrigerant} - \dot{E}x_{out,refrigerant}}$$
$$\psi_{cond} = \frac{\dot{m}_{water}(ex_{out} - ex_{in})}{\dot{m}_{refrigerant}(ex_{in} - ex_{out})} \quad (23)$$

$$\psi_{evap} = \frac{\dot{E}x_{in,refrigerant} - \dot{E}x_{out,refrigerant}}{\dot{E}x_{in,air} - \dot{E}x_{out,air}}$$
$$\psi_{evap} = \frac{\dot{m}_{refrigerant}(ex_{in} - ex_{out})}{\dot{m}_{air}(ex_{in} - ex_{out})} \quad (24)$$

Exergy efficiency for the expansion valve is:

$$\psi_{gen} = \frac{\dot{E}x_{out}}{\dot{E}x_{in}} = \frac{ex_{out}}{ex_{in}} \quad (25)$$

2.4. Degree of Thermodynamic Efficiency

The ratio of the total exergy amount leaving a system to the total exergy amount entering the system is defined as the thermodynamic efficiency of that system and is defined as:

$$\varepsilon = \frac{\sum_{out} \dot{E}x}{\sum_{in} \dot{E}x} \quad (26)$$

making necessary simplifications, the equation becomes [18]:

$$\varepsilon = (-\psi)(\dot{E}x_{in} - \dot{E}x_{out}) \quad (27)$$

3. Results and Discussion

Within the scope of this study, a very comprehensive exergetic comparison was made that examines the effects of temperature and relative humidity on air conditioning. For this purpose, an experimental heat pump system was set up, in which the responses of the system components can be measured, making several experimentations on the system.

The atmospheric air entering the duct was first introduced into the heat pump's evaporator at the desired temperature and relative humidity. The heat pump system was subjected to a very comprehensive examination in terms of heating and cooling aspects, related data were recorded and the necessary calculation were made using these data. All other parameters were kept constant at their highest possible levels so that the effect of temperature and relative humidity can be measured accurately. Detailed exergy analyzes done using the data are presented in the next section.

The temperature and relative humidity of the atmospheric air coming into the system were first reduced in the first heat pump and then conditioned to the desired properties with the appliances in the system. Exergy efficiencies of the system and its components, amount of exergy destruction (entropy generation), irreversibility ratios and thermodynamic efficiency degrees were calculated and presented through graphs were calculated and presented graphically in the future. Among the main components of the heat pump system were compressor, condenser, expansion valve and evaporator whereas the secondary components were categorized as other system components. Air, water and refrigerant responses were analyzed separately for evaporator and condenser. In the system, the evaporator which the air duct runs through was used as cooler and the water condenser was used as heater.

3.1. Determination of the Optimum Gas (Refrigerant) Amount

The ideal gas (refrigerant) amount to be given to the second heat pump, where the measurements were made, was determined as a result of meticulous measurements and calculations and the refrigerant, R134a, was filled into the system being weighed on a precision scale starting from 100 grams and gradually increasing by 25 grams in each time. When the amount of gas in the system reached 500 grams, the compressor was observed to have got cold and the data to have become unstable. The gas amount ideal for the system was determined to be 450 grams. In order to obtain an objective result during the evaluations, air was delivered into the duct at 25 °C temperature and 30% relative humidity, and the water flow rate to the secondary condenser was 60 L/h.

As the changes in the data, up to 250 grams of refrigerant, was very small, the evaluations regarding any amount below 250 grams were excluded in the graphs as the corresponding COP value of the system was too low. Figure 3 shows the COP evaluation when the experimental system was considered as a heat pump, which plots the heat given to the condenser versus the amount of gas in the system. It is clearly seen from the figure that the COP increases with gas amount, having 450 grams as the peak value, beyond which the trend is lost and COP values become instable. Due to the relatively small size of the test system, it reaches the highest COP with 450 grams of refrigerant gas in the system. The temperature of the compressor was observed to start reducing when 500 grams or more gas was given to the system. The optimum amount of gas in order for this system to be evaluated as a heat pump system was determined to be 450 grams.

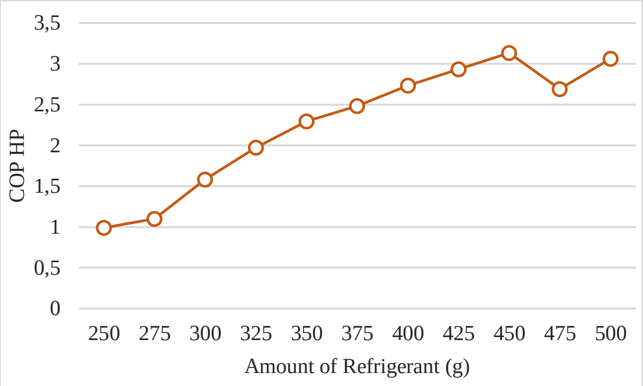

Figure 3 Variation of the COP of the heat pump with the amount of refrigerant in the system

The COP values of the system, when considered as a cooling machine, changed as shown in Figure 4. According to the calculations considering the evaporator of the system as the cooling machine, the COP was observed to increase with the refrigerant amount, starting from 250g until to 450 g, beyond which COP started to decline. Therefore, the optimum amount of R134a refrigerant for this system was determined as 450 grams.

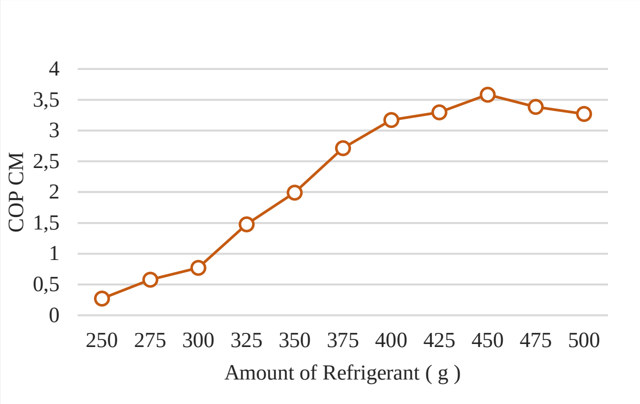

Figure 4 Variation of the COP of the cooling machine with amount of refrigerant in the system

The exergy efficiency when the system is evaluated in terms of refrigerant amount is as presented in Figure 5. As represented by the dotted line in the figure, the uptrend in the exergy efficiency continued up to 475 grams and decreased thereafter.

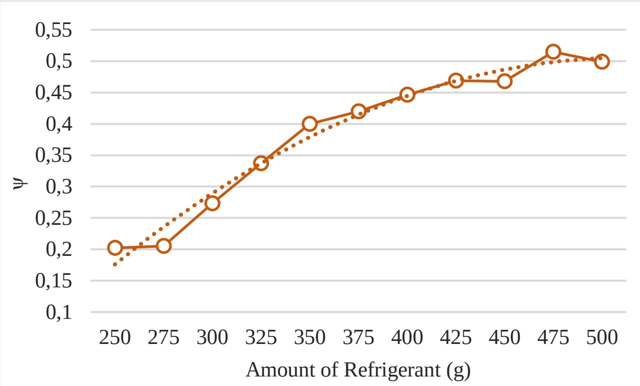

Figure 5 Exergy efficiency for the heat pump versus the amount of refrigerant

3.2. Effects of Relative Humidity and Temperature Change on the System

Water was supplied to the condenser from the mains with 60 L/h flowmeter setting. The air reaching the evaporator was pre-conditioned to desired temperature and humidity levels and fed in a regular manner. The air to the evaporator was delivered at constant velocity by using a single fan and any changes were measured at the exit. Although all parameters were tried to be kept constant at their pre-determined values throughout all experiments, some momentous rises and falls were observed due to laboratory conditions and steam level which are very difficult to keep constant during the experiments even under strict control despite the fact that the amount of steam was tried to be supplied as precise as possible by employing devices that are capable of making precise steam readings. On the other hand, such ups and downs in the graphs are also visible in the similar studies reported in the literature.

Figure 6 shows the effects of temperature and relative humidity on low pressure. Low pressure increases noticeably with the increase in temperature and relative humidity. As seen in Figure 6, the increase occurred at all temperature levels. This pressure increase is attributed to a general stress induced on the system by relative humidity. The fact that the temperature of the refrigerant in the system under load increases every part of the system also brings about the pressure increase. With the increase of the relative humidity, on the other hand, the refrigerant enters the compressor at a higher temperature, forcing the compressor to compress the high temperature gas to higher pressure levels. Although this might not be a problem for powerful large compressors, it may cause more energy loss in systems with small compressors.

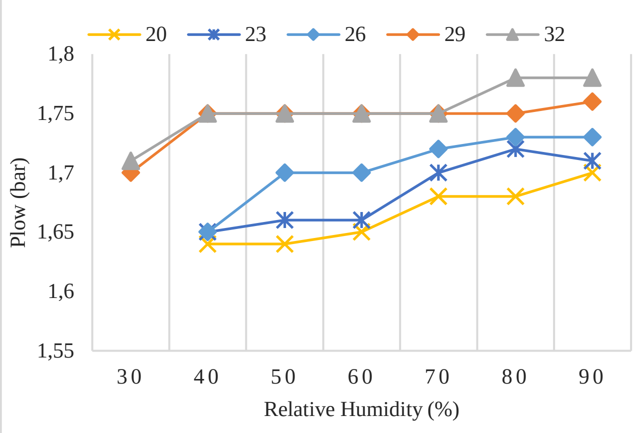

Figure 6 Effects of temperature and relative humidity on low pressure

The pressure values of the refrigerant gas at the exit of the compressor are given in Figure 7. Upon the increase of temperature and relative humidity, the compressor has to increase the pressure more. It is clearly seen in the graphs that high pressure values increase with the increase in temperature and relative humidity. Especially the data for 20 °C clearly shows the effect of relative humidity on high pressure. Due to that the changes at 20 °C require relatively lower performance than the average compressor capacity, the plot for this temperature value is separated from the others.

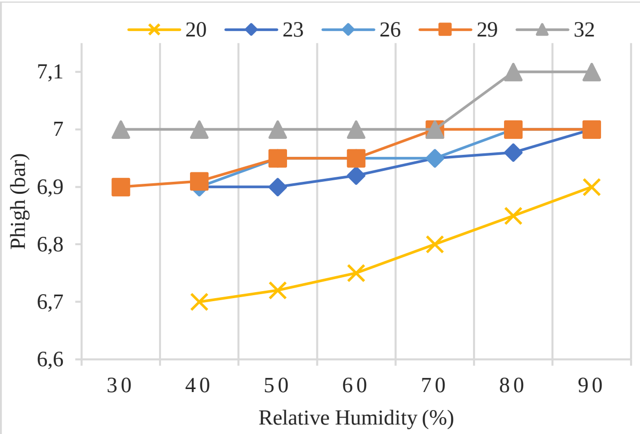

Figure 7 Effects of temperature and relative humidity on high pressure

Figure 8 shows the variation of the temperature of air leaving the evaporator versus relative humidity. It is clearly seen from Figure 8 that the temperature of the air leaving the system increases with the increase in the temperature of the air entering the system. It is seen in the figure that the output temperature increases almost linearly with the increase in relative humidity, and the desired cooling performance decreases. This situation shows that the relative humidity negatively affects the air conditioning performance.

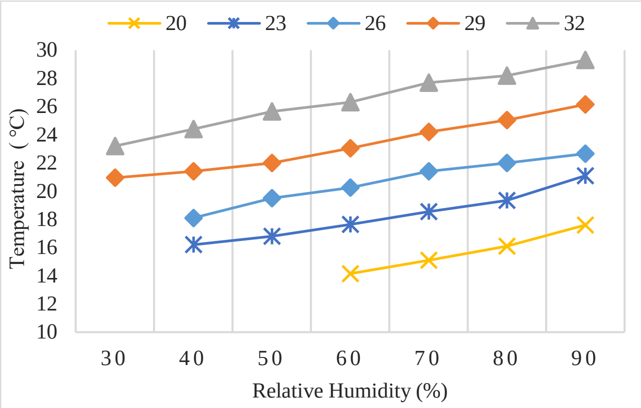

Figure 8 The relationship between temperature and relative humidity of output air

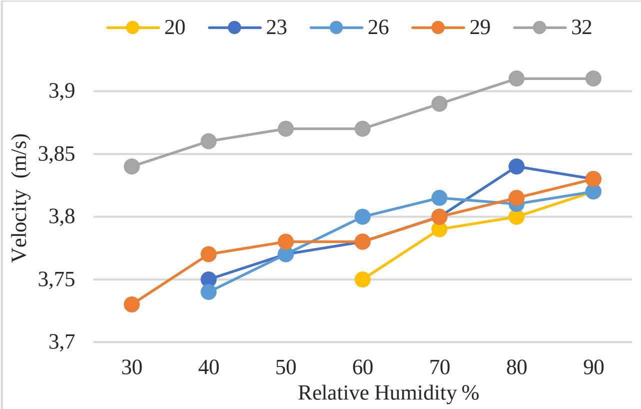

Figure 10 Effect of temperature and relative humidity of input air on the velocity of output air

Figure 9 shows the effect of relative humidity, hence temperature, on the relative humidity of the air leaving the evaporator. The increase in the relative humidity of the air entering the system, as expected, also increases the relative humidity as well as the temperature of the air leaving the system. While the system condenses some of the water vapor held in the input air, the rest water vapor remains in the output air, which clearly shows that air conditioning systems and dryers undertake a serious work in mass transfer while cooling or drying the atmospheric air in humid areas.

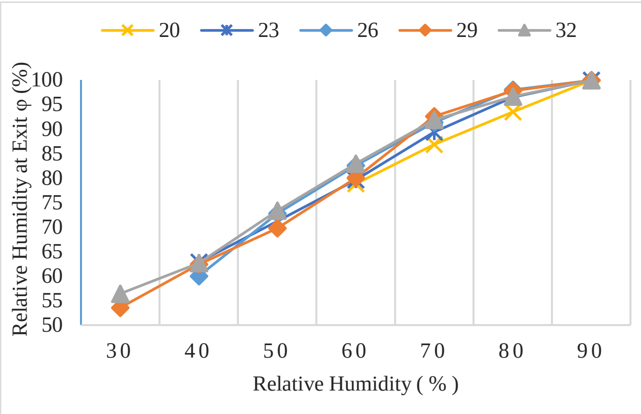

Figure 9 Effect of temperature and relative humidity of input air on the relative humidity of output air

Figure 10 shows the effect of temperature and relative humidity on the velocity of output air. That the general trend is in the upward direction, albeit with small increases, results from the increase in the output air temperature. The increase in the temperature of the output air, on the other hand, is due to the increase in the specific volume of the air. The velocity plot of the output air at 32 °C being distinctly separated from those at other temperature values may be attributed to that the system performance is relatively insufficient with respect to the temperature. This may also mean that refrigeration systems equipped with a compressor same as the one used in the present study, which is widely used, gets loaded more at the same temperature.

In the flow system installed under the evaporator, the amount of condensed water was determined by using scaled beaker and tubes (Figure 11). It can be seen from the graph that mass transfer increases with temperature. And the amount of condensed water increases with the increase of relative humidity. It is seen in this graph that cooling machines consume a significant amount of energy also on mass transfer as well as heat transfer in humid areas.

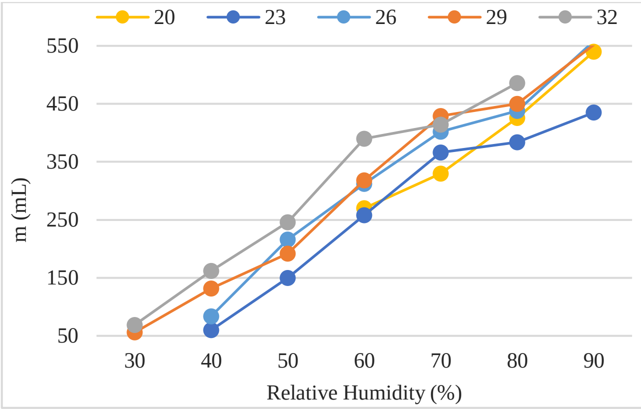

Figure 11 Effect of temperature and relative humidity on the amount of condensed water

Presented in Figure 12 are the effects of temperature and relative humidity increase on compressor performance. Although there seems to be a slight increase in electricity consumption in the experimental system with the increase in temperature and relative humidity, there is no significant change overall, which suggests the compressor operates at the same performance level in all conditions. Compressor not being affected by the changes in temperature and relative humidity allows the changes in other parameters and the effects of such changes to be seen better.

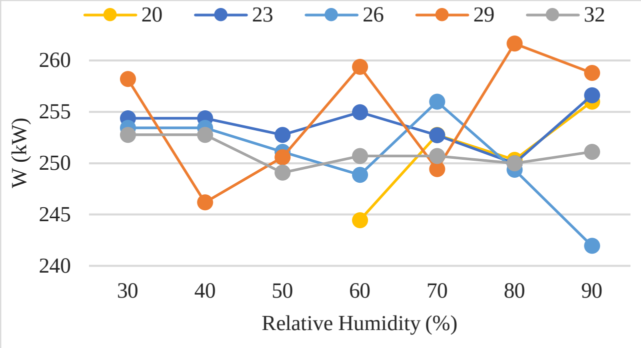

Figure 12 Effects of temperature and relative humidity change on compressor work

Figure 13 shows the effect of relative humidity on energy loss. It can be seen in the figure that the difference between the input and output enthalpies decreases with the increase in the relative humidity, which shows that the relative humidity negatively affects the cooling. It turns out that the energy consumed for cooling is higher in climatic regions with high relative humidity and in other words that in humid areas less cooling is achieved with the same amount of energy consumption.

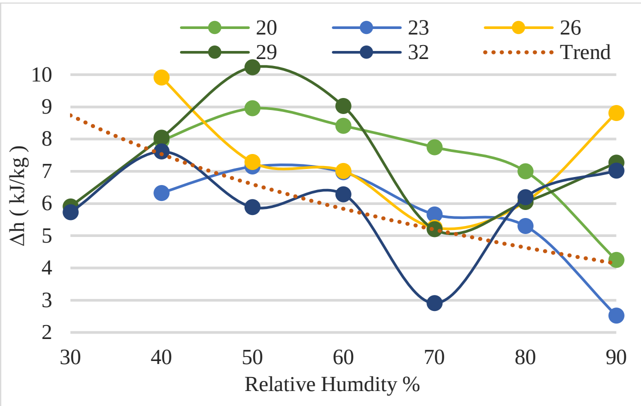

Figure 13 Effect of relative humidity change on enthalpy difference

Figure 14 plots the COP values when the system is evaluated as a cooling machine, which were obtained by proportioning the amount of heat dissipated by the evaporator to the compressor work. The variation of the amount of heat transfer in the evaporator versus 30%, 40%, 50%, 60%, 70%, 80% and 90% relative humidity at 20 °C, 23 °C, 26 °C, 29 °C and 32 °C each can be seen in the graph. While the relative humidity value of 30% could not be conditioned at 20 °C , 23 °C , 26 °C, hence not present in the plot, the increase in the relative humidity of the air causes can be concluded to incur a decrease in the COP values at all temperature values.

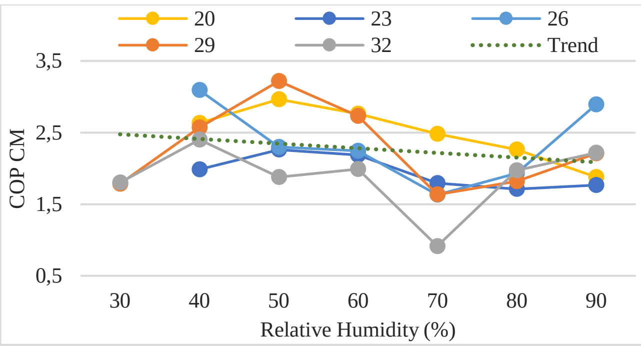

Figure 14 Effects of temperature and relative humidity variation on COP_{CM}

Evaluating the system as a heat pump, the variation of COP values is presented in Figure 15, where the increase in temperature values is seen to induce a decrease in the COP_{HP} whereas on the other hand, the increase in relative humidity increases the COP. Among the examined temperature levels, the most suitable temperature level for the heating need was 20 °C which can be seen to increase in parallel to the increase in relative humidity. This indicates that relative humidity is an advantage to take for heat pump heating systems.

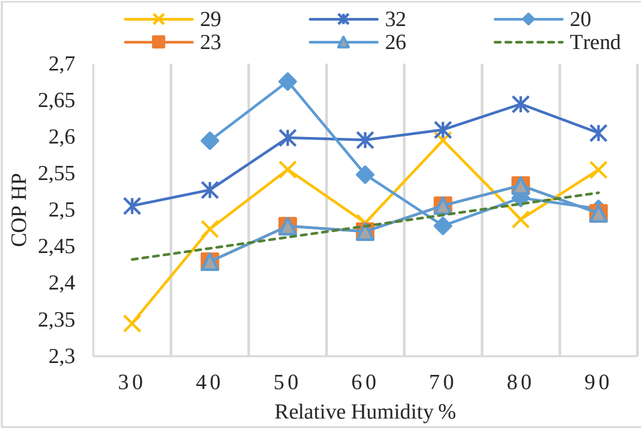

Figure 15 Effects of temperature and relative humidity variation on COP_{HP}

The exergy efficiency of the compressor in the test system is shown in Figure 16, where the exergy efficiency of the compressor is seen to be affected by temperature more than relative humidity. This shows that the compressor capacity is largely suitable for the system and in terms of these temperature values. In the experiments carried out at 32 °C temperature level, the compressor can be seen to have been overloaded after 70% relative humidity level, hence has reduced exergy efficiency.

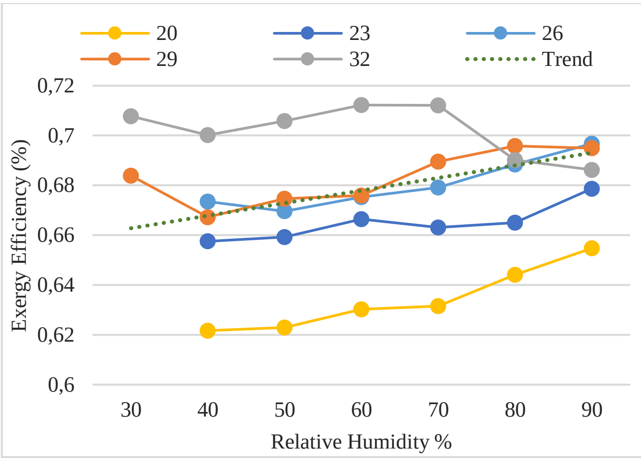

Figure 16 Effects of temperature and relative humidity on exergy efficiency of the compressor

The changes in the thermodynamic efficiency of the compressor of the experimental system are presented in Figure 17, showing it positively related to temperature and relative humidity. The ratio of the exergy at the compressor input to the exergy at the compressor output varies in direct proportion with the increase in temperature and relative humidity. The relative decrease after 60% relative humidity at 20 °C indicates that the increase in relative humidity at low temperatures reduces the thermodynamic efficiency.

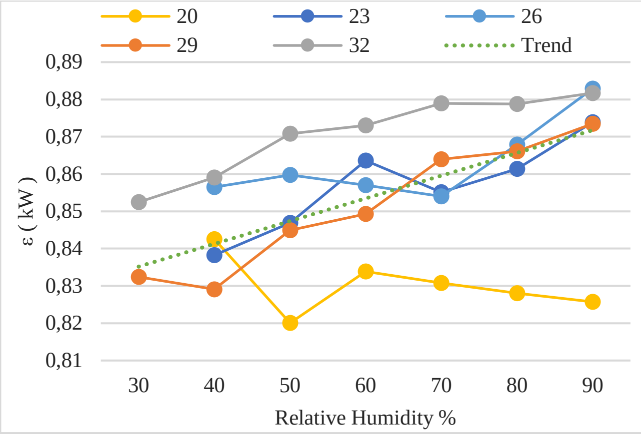

Figure 17 Effects of temperature and relative humidity on thermodynamic efficiency of the compressor

The change in exergy efficiency of the condenser versus temperature and relative humidity is as shown in Figure 18, which indicates that the exergy efficiency decreases with the increase of relative humidity.

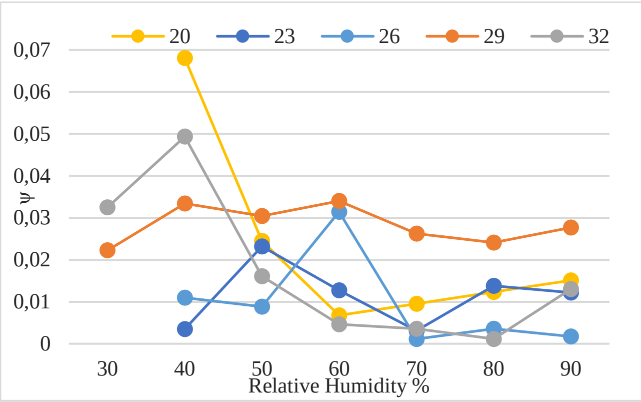

Figure 18 Effects of temperature and relative humidity on exergy efficiency of the condenser

The thermodynamic efficiency of the condenser, shown in Figure 19, tends to decrease with relative humidity at 20 °C, whereas it increased at 23 °C, 26 °C, 29 °C and 32 °C. So, having such a system operate as a heat pump in cold climate regions, the thermodynamic efficiency of the system will be negatively affected by the increase in relative humidity.

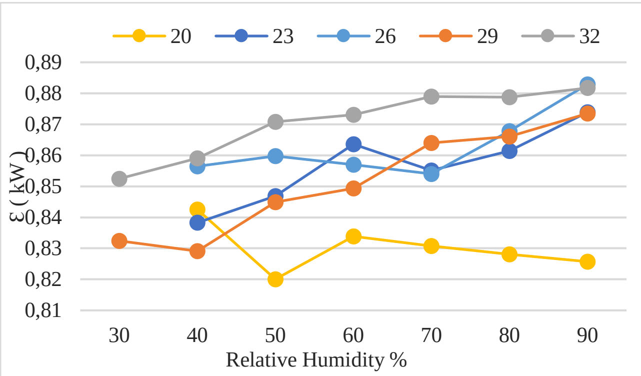

Figure 19 Effects of temperature and relative humidity on thermodynamic efficiency of condenser

The relative humidity of the input air entering the system through the air duct, has a negative impact on the exergy efficiency of the evaporator, as presented in Figure 20. Based on this plot, the variation of the exergy efficiency, which is expressed as the ratio of exergy of the input air to the exergy of the output air, induced by temperature change can be concluded to be insignificant. On the other hand, it is obvious that the relative humidity of 60% or higher negatively affects the exergy efficiency of the evaporator, hence that of the cooling machine, at 32 °C temperature level, which is the highest temperature in the experiments.

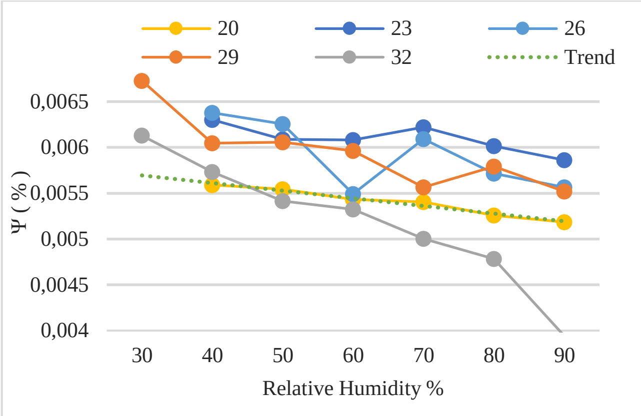

Figure 20 Effects of temperature and relative humidity on exergy efficiency of evaporator

The evaluation of the thermodynamic efficiency of the evaporator is as presented in Figure 21. The thermodynamic efficiency of the evaporator increases with the increase in temperature and relative humidity. It can be seen from the graph that starting from 20 °C, at which the minimum increase was observed, the rate of increase gradually increased as the temperature was risen to 23 °C, 26 °C, 29 °C, and 32 °C. The increase in relative humidity increases the irreversibility ratio of the evaporator, hence the cooling machine, for all cases.

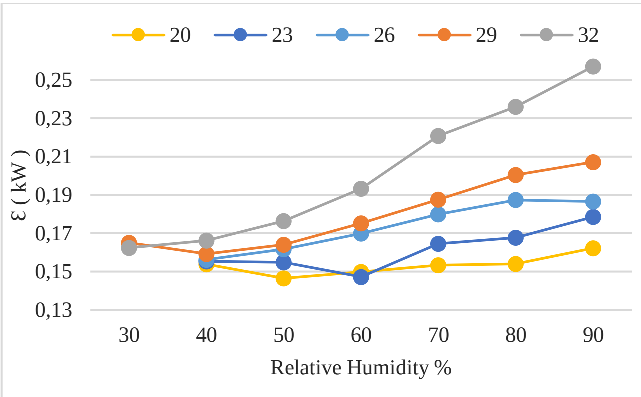

Figure 21 Effects of temperature and relative humidity on thermodynamic efficiency of evaporator

The thermodynamic efficiency analysis of expansion valve (also known as throttle valve), which converts high pressure fluid to low pressure, is as presented in Figure 22. The thermodynamic efficiency analysis of this component revealed a general improvement in inverse proportion with the increase in temperature and relative humidity, with the exception that a slight decrease, as opposed to an expected increase, was observed against the increase in relative humidity in the experiments at 20 °C.

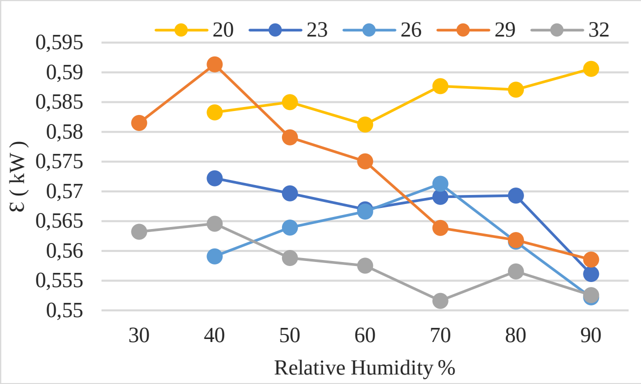

Figure 22 The effects of temperature and relative humidity on the thermodynamic efficiency of expansion valve

Lastly, the variation of the exergy efficiency of the heat pump is presented in Figure 23 with respect to the different values of temperature and relative humidity. The trend of the exergy efficiency change in the experiments at 20 °C is seen to be upwards despite the fact that the exergy efficiency is seen to be in an inverse relationship with temperature and relative humidity, for all other temperature levels. So it can be concluded that the exergy efficiency changes in direct proportion to the relative humidity when the system is used as a heat pump. On the other hand, when the system is used as a cooling machine, the increase in relative humidity causes exergy efficiency to decline.

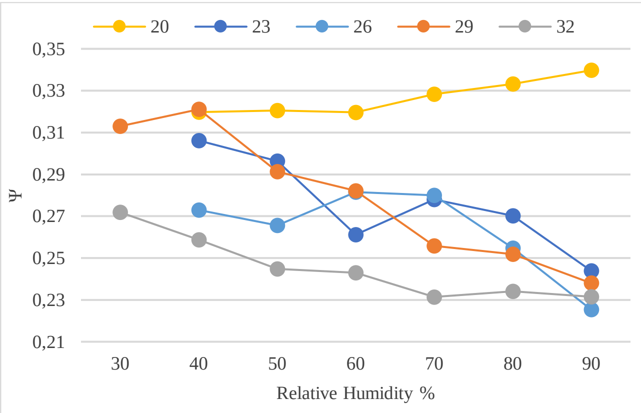

Figure 23 Effects of temperature and relative humidity on exergy efficiency of heat pump (evaluation in terms of refrigerant)

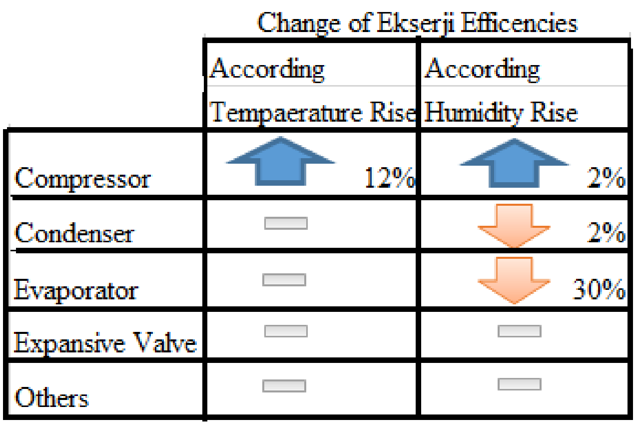

Figure 24 Variation of exergy efficiencies of system components with temperature and relative humidity

The general trends of the exergy efficiencies of system components versus temperature and relative humidity are as shown in Figure 24. As another output of this study, the effect of relative humidity on enthalpy, which can theoretically be deduced using the psychrometric diagram, has been experimentally demonstrated and verified.

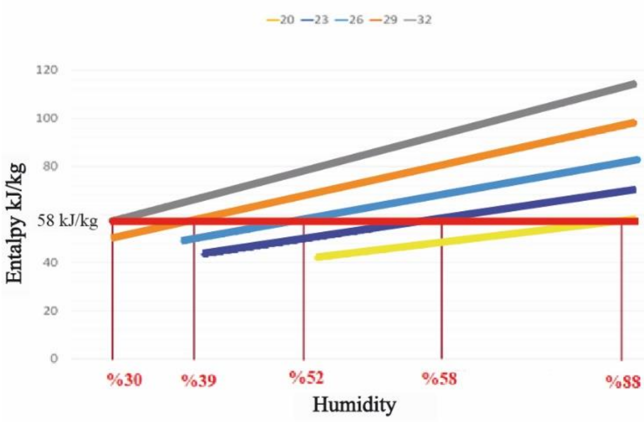

Figure 25 Variation of enthalpy of atmospheric air at different temperatures with relative humidity

In Figure 25, the enthalpy increases caused by relative humidity are presented for different temperatures, which is striking an example of the significance of the ratio of water vapor held in the air. As exemplified by the red line that vertically connect the trend lines of the enthalpy variation of atmospheric air at different temperatures with relative humidity, the same enthalpy value can be obtained at all temperature and relative humidity values. Accordingly, an enthalpy value of 58 kJ/kg is obtainable at certain temperature and relative humidity values. For instance, the enthalpy of ambient air at 20 °C with 88% relative humidity is the same as that at 32 °C with 30% relative humidity. All taken together, this reveals that heating and cooling processes are more difficult in regions with humid climates than in dry or arid climates.

4. Conclusion

An experimental rig has been established to experimentally investigate and reveal the effects of relative humidity and temperature on heating and cooling performances of the system. High precision devices were used, as applicable, in the experimental system and all parameters other than the research variables were tried to be kept constant so that the mere effects of temperature and relative humidity be observable on the system. All changes on the heat pump were recorded and subjected to related calculations.

The evaluations made formed an idea about the importance of component compatibility as well as capacities in creating a system. Here, the significance of exergy analysis in determining the compatibility of components of a heat pump or cooling machine system with each other is recognized. It is important that the system as a whole and all of its components are compatible with the climatic characteristics of the region where they will be operated. When setting up a system, evaluations and analyzes will help select compatible components in terms of capacity and capability.

The highest COP_{CM} value was observed to have occurred at the lowest temperature level, 20 °C and the lowest COP_{CM} at 32 °C, the highest temperature level.

The increase in the relative humidity of the input air was found to cause a decrease in the COP_{SM}. The highest COP_{CM} values were at 30% and 40% relative humidity levels whereas the COP_{CM} values remained low when the relative humidity rose to 70% - 80%.

The experiments showed that the exergy loss in the system occurred mostly in the evaporator. At all temperature levels, exergy destruction occurred the most in the evaporator and the least in the condenser.

The relative humidity of the input air was found to be not that effective on the exergy efficiency of the compressor, due to the fact that the electricity consumption and capacity of the compressor do not vary according to climatic conditions. Another reason is that the capacity of the compressor is much higher than that required by the system.

Thermodynamic efficiency, on the other hand, increases with the increase in temperature and relative humidity. This shows that the compressor works more efficiently, within the range of its capacity, under harsh/difficult conditions.

As presented with graphs that reveals the effects of temperature and relative humidity, the exergy efficiency of the evaporator is high in low relative humidity conditions. Accordingly, cooling devices operating in arid regions undergo less exergy destruction compared to humid regions.

As for the expansion valve, exergy destruction is less at low temperatures and it increases with temperature. The increase in relative humidity, on the other hand, also increases the exergy destruction while it does not cause a large change in the irreversibility ratio. The thermodynamic efficiency shows a decreasing trend against the increase in temperature and relative humidity, which indicates that the expansion valve performs better in high relative humidity and temperature conditions compared to other components of the system.

Evaluated as a cooling machine, with reference to the refrigerant, the efficiency coefficient of the system was observed to decrease with the increase in temperature and humidity, which induce an increase in the temperature and pressure of the refrigerant circulating the system. The result show that temperature and relative humidity, together or separately, have similar effect.

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Conflict of Interest

The authors declared no conflict of interest.

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