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THE EFFECT OF SYSTEM PARAMETERS ON THE CONDENSATION PERFORMANCE OF HEAT PUMP SYSTEM USING R290

S. Hocaoglu[1,*], D.B. Ozkan[2,*]

ABSTRACT

Since global warming has reached critical levels, limitations have been placed on the use of certain fluoride-containing refrigerants by F-Gas regulations. The EU F-Gas Regulation has introduced quotas for the use of refrigerants with a global warming potential(GWP) greater than 150. Hydrofluorocarbons(HFCs) from restricted refrigerants are widely used in heat pump systems. Considering the environmental impact of these refrigerants, it is important to look for long-term alternatives to comply with F-gas regulations. Hydrocarbon(HC) refrigerants are shown as suitable alternatives for heat pump applications. R290 as HC refrigerant is a potential refrigerant suitable for existing HFCs systems due to zero ODP and low GWP. In heat pump systems, there are many system components or parameters that are effective in condensing the air passing through the evaporator. It is very important to know how these elements affect the condensation performance in different design situations. In this study, the effect of different parameters such as capillary length, charge amount and evaporator tube volume on the condensation performance of a R290 hydrocarbon refrigerant heat pump was investigated by the experimental design approach. The experimental results obtained was compared with the theoretical model. It has been determined that the most effective parameter on the condensation performance is the capillary tube length with the effect of 35%.

Keywords: *Heat Pump, Hydrocarbons, Propane, R290, F-Gas Regulation, GWP, ODP*

INTRODUCTION

CFCs, HCFCs and HFCs, which types of refrigerants, were widely used in commercial or household appliances in the last couple of decades. Given the increased concern about global warming and environmental pollution and the damage that these refrigerants have on the environment, restrictions and prohibitions have been placed on the use of such refrigerants [1]. As a result of the investigations, it has determined that the halogen-substance contained in the refrigerants of CFCs and HCFCs reacted with ozone and depleted the ozone layer. For this reason, the use of these refrigerants has been abolished by the Montreal Protocol [2]. CFCs refrigerants have high ODP and GWP values. The HFCs group refrigerants have no adverse effect on the depleting of the ozone layer and they have very low ODP. However, because of fluorinated gases, they are greenhouse gases that cause climate change and not an environmentally friendly gas family. Therefore, the emission of six greenhouse gases causing global climate change which has been reduced by the Kyoto Protocol [4]. HFCs are fluorine-containing greenhouse gases with a high GWP value. The new F-gas regulation, released in 2014, aims to reduce F-gas emissions by 2/3 by 2030. With F-Gas regulation, a quota has been applied for the use of refrigerants with more than 150 GWP [1]. A low-carbon road map was identified by F-Gas regulation. It is aimed to phase out and prevent the gases which cause the greenhouse effect according to the low-carbon road map. These developments have affected the industry in search of a new refrigerant and consequently have concentrated on HCs [2]. Hydrocarbons are natural refrigerants with zero ODP and very low GWP. For this reason, the greenhouse effect of hydrocarbon refrigerants is very low. Hydrocarbons and their mixtures, such as R600a, R290 and R1270, are good alternatives to be used instead of HFCs, such as R134a and R32. Even though their high flammability feature prevents their commonly use, it is considered that the hydrocarbons is suitable for household refrigerators and heat pump systems [3].

Bhargav et al. [5] designed a household heat pump refrigerator to work with R134a and investigated its utility with propane, isobutane blend. In their study, they compared to the performance of the refrigerator according to the use

[1]*Department of Central R&D, Arçelik A.Ş, Istanbul, TURKEY*
[2]*Department of Mechanical Engineering, Yildiz Technical University, Istanbul, TURKEY*
**E-mail address: hocaoglusamet@gmail.com*
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of R134a and azeotropic mixture as refrigerant. The theoretical study showed that the COP obtained with the azeotropic mixture (R290/R600A) is 7.6% better than the reference R134a. The result was that it has very low GWP and ODP and it can be an alternative refrigerant for R134a due to its excellent cooling characteristics. Bellomarea et al. [6] experimentally studied on the change of refrigerants in heat pump tumble dryers. The R407C tumble dryer was taken as a reference and experimental studies was done using exactly the same system components with R290 and R441a. Experimental results demonstrated that for the same system, the R290 refrigerant reduces total energy consumption by 6%, drying time by 8%, compressor energy consumption by 9%, and charging by 50% compared to R407C. D. Sánchez et al. [7] experimentally tested five low GWP (R152a, R1234yf, R1234ze (E), R290 and R600a) refrigerants that could be alternative to R134a in the same refrigeration system equipped with a hermetic compressor under the same operating conditions. As a result of the experiment, the best results in terms of cooling capacity and COP were obtained from hydrocarbon R290 (propane), with an increase of 40.5%-67.4% and 22.4-2.8%, respectively. However, as increase in power consumption (up to 44.8%) requires a smaller displacement compressor volume than R134a. This study concluded that propane should create a different system design to be a suitable alternative refrigerant. Bengtsson [8] performed a simulation study of the heat pump dishwasher using R134a, R290 and R600a. When the compressor cylinder volume was optimized, the total electricity usage and TEWI results were examined. The simulation study showed that the electrical usage of the electrical appliance dishwasher is 25% greater than the heat pump system, independent of the refrigerant used (ie R134a, R290 or R600a). The lowest TEWI value was found in refrigerants (R290 and R600A). The TEWI value of R290 and R600a was determined to be 74.5% compared to R134a and 25% less than the electrical element. Longhini [9] studied refrigerators with R32, R152a, R290, R1270, R1234yf and R1234ze (E) which can be used as an alternative to the R-410A used in 10 kW heat pumps in Swedish houses. Engineering Equation Solver(EES) model was designed to compare results such as volumetric heating capacity, COP, seasonal COP(SCOP), TEWI. Theoretical analyzes showed that the lowest TEWI value (54.6% less than R410) belongs to R290. Although the COP of R290 at the lowest temperature was 1% higher than R410A, it was found that the COP value decreased by 8.89% compared to R410A at the highest temperature during the working temperature. Meyer [10] investigated the cooling capacities, COP, and mass flow rates of R-134A, R-290, R-404A, R-407c, R22 and R-410a refrigerants in a vapor compression system between constant condensation and variable evaporation temperatures. As a result of the experiments, the highest and lowest cooling capacities were obtained respectively R410a and R134a (56% less than R410a). Experimental analysis showed that COP values were highest at R134A and lowest at R407C (17% less than R134a). Finally, it was seen that the lowest mass flow rate for the same capacity belongs to the R290 refrigerant (65% less than R410A and R404A). Jwo [11] experimentally compared the mixture of the hydrocarbon refrigerants R290 and R600a (50% R290, 600%), which is supposed to be used instead of R134a in a domestic refrigerator. As a result of the experiment, it was found that the cooling effect of the hydrocarbon mixture was better, the total energy consumed and the amount of charge decreased by 4.4% and 40%, respectively, compared to R134a. Choudhari et al. [12] investigated the performance analysis of refrigerants R290 and R22 in a vapor compression cycle using analytical calculations. Theoretical calculations found out that discharge temperature of R290 is lower than R22. It was also found that the R290 refrigerant flow rate was 50% less than R22 and the COP values were almost equal. The result is that R290 is a very good alternative to R22 because of its excellent environmental and thermophysical properties. Ghoubali [13] studied COP optimization based on the refrigerant charge amount from the three types of condensers (tank-wrapped D-tube, roll-bond and microchannel) in the R290 refrigerant heat pump water heater on the same cooling base. The best performance was obtained by the roll bond HPWH with a charge of 224 g: COP of 3.17 and heating time under 7 h. Nawaz et al. [14] developed a component-based model to evaluate the performance of R290 and R600a, which could be alternative to R134a in a heat pump water heating system. The results of the analysis showed that the charge required for the R290 and R600a system was reduced by 50% compared to the reference R134 system, that the shortest run time was obtained at R290 and that R600a's run time was 33% longer than R134a. The graphs generated for the various performance parameters showed that the R290 which is a very viable alternative to R134a, but the R600a needs a larger compressor size to meet its reference performance values. Zhou et al. [22] compared the system performances of a split type air conditioner with R22 and R290 with various operating conditions. Their results indicated that the refrigerant charge and mass flow rate of R290 were only 44% and 47% of R22, and R290 had 4.7- 6.7% lower cooling capacity and 12.1- 12.3% lower input power than R22. The energy efficiency ratio (EER) of R290 was 8.5% higher than that of R22. Lee et al. [24] discussed condensing and evaporating heat transfer characteristics of R-290 (propane), R-600a (iso-butane) and R-1270 (propylene) as environment-friendly refrigerants and R-22 as

HCFCs refrigerant in heat pump system. The Experimental results demonstrated that the average condensing heat transfer coefficients for R-290, R-600a and R- 1270 were higher than that for R-22 by about 60%, 68% and 70%, respectively. And also its showed that the average evaporation heat transfer coefficients for R-290, R-600a, R-1270 were higher than that for R-22 by about 67.7%, 55.4%, and 72.3%, respectively. Tashtoush et al. [25] carried out an experimentally study on the replacement of R12 in domestic refrigerators by new hydrocarbon/hydrofluorocarbon refrigerant mixtures and investigated the performance of these refrigerants. Experimental results showed the coefficient of performance of R290/R600/R134A (25/25/30) was 5.4% less than R12 at 100 W evaporator duty and 0.8% less than R12 at 350 W evaporator duty and mass flow rate was 40% lower at all conditions. Longo et al. [26] compared HC290, HC1270 and HFC404A during vaporization inside a 4 mm smooth tube. Analysis of experimental result exhibited that frictional pressure drop of HC290 is 20-30% lower than HFC404A and to be as long-term alternative low GWP candidate for HFC404A. Liu et al. [27] tested experimentally heat transfer coefficient and pressure gradient during condensation of R290, R1270 and R22 in circular (dh=1.085 mm) and square (dh=0.952 mm) minichannels. Analyses results demonstrated that the heat transfer coefficients and pressure gradients during condensation increase with increasing mass flux and vapor quality and decrease with increasing saturation temperature for all refrigerants. Its showed that examined parameters for propane were larger than those of R1234ze(E) which are larger than those of R22. Wongwises and Chimres [28] studied as experimentally on the application of hydrocarbon mixtures to replace HFC-134a in a domestic refrigerator. The results showed that R290/R600 (60%/40%) was the most appropriate alternative refrigerant to R134a in terms of energy consumption (5% lower than R134a). Wongwises et al. [29] presented an experimental study on the application of R290, R600 and R600a mixtures to replace HFC-134a in automotive air conditioners. The results showed that blends of R290/R600/R600a (50%/40%/10%) was the most appropriate alternative refrigerant to R134a comparing to others blends. Compared with R134a, the refrigeration capacity, the compressor power consumption and COP value of R290/R600/R600a (50%/40%/10%) was higher by 41%, 21% and 17%, respectively. Ju et al. [30] investigated performances of a heat pump water heater (HPWH) system with the blends of R744/R290 or R22 by simulation and experiment. Results obtained exhibited the R744/R290 with optimal mass fraction of 12%/88%. And results showed that COP and heating capacity of the HPWH with optimal mass fraction blends were 11% and 17.5%, respectively, higher than those of the R22 scheme at the nominal condition. Longo [31] analyzed effect of vapour super-heating on hydrocarbon refrigerant HC-600a, HC-290 and HC-1270 condensation inside a brazed plate heat exchanger. Analyses results indicated that HC-1270 showed super-heated heat transfer coefficient and total pressure drops were 5%,10–15% higher, 20–25%, 50–66% lower than HC-290 and HC-600a, respectively, under the same mass flux. Devotta et al. [32] studied R290 refrigerant as an alternative to R22 in window air conditioners. Their results demonstrated that the cooling capacity and energy consumption of R290 were lower than those of R22 by 6.6 to 9.7% and 12.4 to 13.5%, respectively. The COP ofR290 was higher than that of R22 by 2.8 to 7.9%. Hwang et al. [33] experimentally tested the HFCs (R-404A and R-410A) as compared to R-290 for direct expansion type walk-in refrigeration systems with 4 kW and 11 kW system capacity. Test results showed that, based on same system capacity, the COPs of R-404A and R-410A were 11-12% and 4-9% lower, respectively, than that of R-290 under full load conditions, and based on the same compressor efficiency assumption, the comparisons showed that the COPs of R-404A were 7-10% and 5-6% lower under full load and part load test conditions, respectively, as compared to R290. Oyedepo et al. [34] experimentally investigated the thermodynamic performance of a domestic refrigerator by simultaneously varying the refrigerant charge and the capillary tube length. Experimental studies demonstrated that the cooling capacity of R600a was about 9.18% higher than that of R12, the power consumed by R600a was about 24 % lower than that of R12 and the COP of R600a was about 6.3% higher than that of R12. Sheikholeslami et al. [35-36] examined experimentally the hydrothermal behavior of refrigerant-based nanofluid during condensation inside a horizontal tube. Results indicated that the presence of nanoparticles generally increased the frictional pressure drop. Sheikholeslami et al. [37-39] simulated the magnetohydrodynamic nanofluid via Lattice Boltzmann method in different convective flow types (natural, forced) and different types of porous media (cubic, circular). Sheikholeslami et al. [40] investigated the nanofluid forced convection heat transfer in existence of magnetic field. Results demonstrated that velocity of nanofluid augments with rise of Reynolds number and Al2O3 volume fraction but it reduced with increase of Hartmann number. Sheikholeslami et al. [41] utilized Lattice Boltzmann method to investigate magnetic field impact on nanofluid natural convection inside a porous enclosure with four square heat sources and they modeled influence of Hartmann number on MHD convective flow in a permeable medium. Simulation results indicated that convective heat transfer decreased with increase of Hartmann number but it augmented with increase of Darcy and

Rayleigh. Malvandi et al. [42] studied the transport phenomenon of the nanofluids falling condensate film, taking into account the effects of the nanoparticle migration. They indicated that the intensity and direction of nanoparticle migration are able to manage the thermophysical properties of nanofluids, as well as the control of flow, heat transfer, and mass transfer, in order to improve the cooling performance.

In this study, unlike other studies in the literature, the effect of different parameters on the condensation rate of a heat pump using R290 refrigerant was investigated experimentally and theoretically and the effect of these parameters on the condensation performance was compared in graphical form.

PROPERTIES OF R290

In recent years, the R290 from the hydrocarbon refrigerant group comes into prominence as an alternative to the HFCs used in existing heat pump systems. Refrigerants should have certain chemical and physical properties in order to use economically and reliably perform their tasks in the systems they are using. The high latent heat of vaporization and the low specific vapor volume of R290 allows the amount of refrigerant charge circulating in the system to be less. Because of requires in low quantities refrigerant charge, it allows the use of small volume heat exchangers in heat pump systems.

The critical temperature is high and the freezing temperature is low. Compressors with smaller stroke volumes are needed compared to conventional refrigerants due to the high volumetric capacities and the low amount of refrigerant used. Thus, the volume of the cooling group which used in the system will also decrease. The discharge temperatures are not high, compressor life is longer than the others [9]. Lubrication systems are different as they belong to a completely different group of conventional refrigerants. It is not easy to drop in instead of the refrigerant used in a system, and the whole system needs to be optimized according to R290. Although they appear to be a disadvantage because of being in the high flammability class, they are not dangerous under 150 grams refrigerant charge [15]. Their usage in low refrigerant charge quantities reduces concerns in this regard and facilitates their usage in the appropriate design.

THEORETICAL ANALYSIS

Working Principle of Heat Pump Dehumidifier

The heat pump dehumidifier consists of two separate cycles, the drying air cycle and the refrigerant cycle.

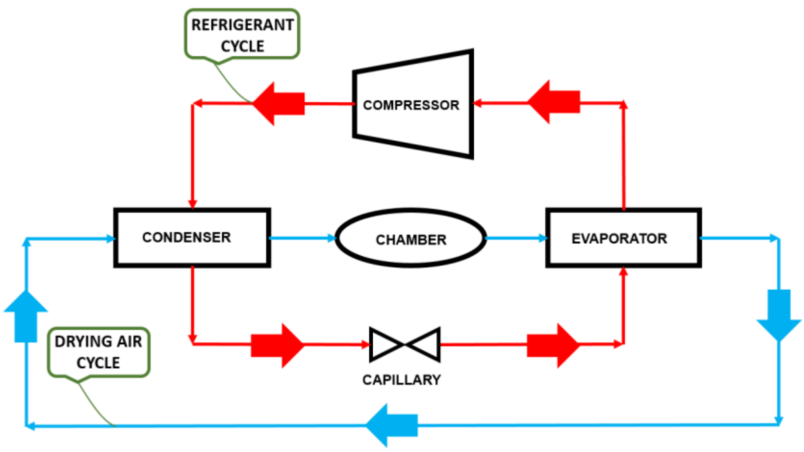

Figure 1. Air cycle and refrigerant cycle in heat pump system

In refrigerant cycle, temperature and pressure of the refrigerant are raised by the compressor and delivered to the condenser inlet via the compressor outlet pipes. The refrigerant in the condenser conveys the thermal energy to the drying air and condenses to leave the condenser in the liquid phase. Temperature and pressure of refrigerant coming from the condenser to the capillary tube are drastically reduced in here. The refrigerant discharged from the capillary tube to the evaporator is circulated through the evaporator tubes and draws heat from the hot drying air and evaporates. The vapor phase refrigerant is absorbed by the compressor, compressed and sent back to the condenser at high pressure and temperature. Thus, the refrigerant cycle is also completed.

Both refrigerant and air cycles in the heat pump dehumidifier occur at the same time.

Theoretical Performance Analysis

This study was conducted on a heat pump system using two heat exchangers (evaporator and condenser), capillary tube, constant speed compressor and R290 refrigerant. The refrigerant properties and vapor compression cycles to be used in the calculations were taken from Refprop 9.0 [23] and CoolPack [19], respectively, and the heat exchanger capacities from CoilDesigner software [21]. Since the effect of different parameters on the condensation performance is examined within a certain period, the capacities vary in every design and the compressor capacities also vary according to the working temperatures.

Heat pump systems are designed according to the refrigeration cycle. Therefore, the ln(P)–h diagram which is used for analyses of an ideal heat pump system and isentropic efficiency of the compressor are shown in Figure 2. When the catalogs of the compressor supplier were analyzed, the isentropic efficiency of the compressors which was used in heat pump systems was determined as 0.7. For this reason, η_s: 0.7 was used for this study. The air position during air cycle is also shown **in Figure 3.**

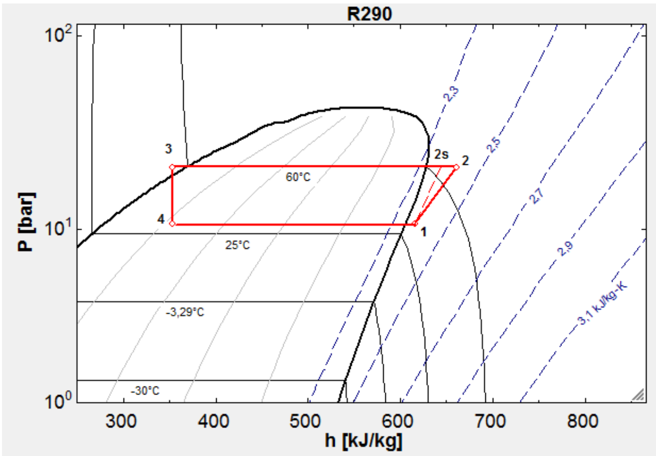

Figure 2. Vapour compression cycle of R290

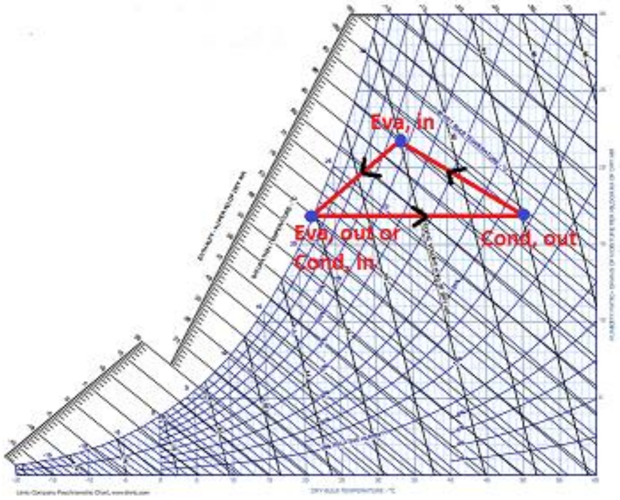

Figure 3. Psychometry process of air

In this study, theoretical calculations were performed according to both air (Figure 3) and the refrigerant side (Figure 2) closed cycles. The results obtained from two sides were compared with each other and verified.

Refrigerant Side Heat Pump System Calculations

All governing equations were developed by Cengel Y. A and Boles M. A. for all sections [17]. The evaporator capacity at which dehumidification has taken place in drying air has been calculated according to equation (1):

$$\dot{Q}_{eva} = \dot{m}_r \, . (h_1 - h_4) \tag{1}$$

Condenser capacities has been calculated according to equation (2):

$$\dot{Q}_{cond} = \dot{m}_r \, . (h_2 - h_3) \tag{2}$$

The compressed refrigerant in the compressor leaves the compressor at 2 in real case, in the ideal case at 2s. From here, the compressor capacity and the volumetric efficiency have been obtained by the equations (3) and (4), respectively:

$$\dot{W}_{comp} = \dot{m}_r \, . (h_2 - h_1) \tag{3}$$

$$\eta_s = (h_{2S} - h_1)/ (h_2 - h_1) \tag{4}$$

Air Side Heat Pump System Calculations

The air passing through the evaporator is cooled as latent and sensible. Condensation rate has been calculated by Equation (5):

$$\dot{m}_v = \dot{m}_a \, . \Delta w \tag{5}$$

If this equation is rewritten:

$$\dot{m}_v = \rho_a \, . \dot{V}_a \, . (\omega_{eva,in} - \omega_{eva,out}) \tag{6}$$

has been obtained.

The cooling capacity of the air passing through the evaporator has been calculated by Equations (7), (8) and (9), respectively:

$$\dot{Q}_{eva,air,latent} = \dot{m}_v \, . \, h_{fg} \tag{7}$$

$$\dot{Q}_{eva,air,sensible} = \dot{m}_a \, . \, c_p \, . (T_{eva,in} - T_{eva,out}) \tag{8}$$

$$\dot{Q}_{eva,air} = \dot{Q}_{eva,air,latent} + \dot{Q}_{eva,air,sensible} \tag{9}$$

The air passing through the condenser is heated as sensible by the condenser. The heating capacity of the air passing through the condenser has been calculated according to equation (10):

$$\dot{Q}_{cond,air} = \dot{m}_a \, . (h_{cond,out} - h_{cond,in}) \tag{10}$$

The amount of heat lost by drying air passing through the chamber is calculated according to equation (11):

$$\dot{Q}_{loss} = \dot{m}_a \, . (h_{cond,out} - h_{eva,in}) - \dot{m}_{vapour} \, . \, h_{fg} \tag{11}$$

The air side energy balance has been shown in equation (12):

$$\dot{Q}_{cond,air} = \dot{Q}_{eva,air} + \dot{Q}_{\,loss} \tag{12}$$

In this study, amount of refrigerant mass flow rate was taken from the compressor calorimeter tests running for specific temperature ranges. Mass flow rates corresponding to the intermediate temperature values were found in the regression model created in the Minitab software [18]. Enthalpy values of the refrigerant at the inlet and outlet of evaporator were obtained from steady state temperature and pressure values of the system by the CoolPack software [19]. The CoolPack software was developed on the EES- Engineering Equation Solver platform. Thermodynamic properties of R290 and condensate water were determined by applying the fundamental equation of state for each phase [20]. After enthalpy and mass flow rates were determined, the refrigerant side evaporator capacity was calculated by Equation (1) shown in the theoretical analysis.

In air side, the air mass flow rate was measured experimentally. The temperature and relative humidity values of moist air were measured through thermocouple and humidity sensors used at inlet and outlet of the evaporator. Specific humidity values of air at evaporator inlet and outlet were obtained per seconds in the Psychrom program, which was added to Excel. Air flow rate and specific humidity values were written to Equation (5) and condensation rate was theoretically calculated by Equation (5). The measured and calculated values were written to Equation (7) and Equation (8) and the cooling capacity for the air side was theoretically calculated by Equation (9).

A deviation of 3% between theoretical calculations made for both air side and refrigerant side was found

EXPERIMENTAL ANALYSIS

Efficiency Expressions in Dehumidification Machines

In order to determine how efficient a system is formed in R290 heat pump system; moisture extraction rate and coefficient of performance must be determined.

Moisture Extraction Rate(MER); is gram of moisture removed per second indicates the dryer capacity. It’s measurement of condensation performance.

$$MER_{theoretical} = \frac{Moisture\ Condensated\ From\ Air}{Second} \tag{13}$$

Theoretical MER was calculated by Psychrom.

$$MER_{experimental} = \frac{Total\ Moisture\ Condensated\ From\ Air\ to\ Water\ Tank}{Operation\ Time} \tag{14}$$

Experimental MER was measured by precision scales.

Coefficient of performance(COP); is a measure of the amount of power input to a system compared to the amount of power output by that system. The COP is also a measurement of efficiency:

$$COP_{HP} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp}} \tag{15}$$

Experimental Setup

In this study, 32 T type thermocouples were used on the air line and the refrigerant line. In addition to thermocouples, pressure transducers were installed at the compressor outlet and inlet points, and system pressures were monitored. Temperature and pressure values were transferred to the computer environment via data collection system. Temperature and relative humidity data were obtained, where the air passes, via SHT-75 type humidity sensors. The obtained data were transferred to the computer environment via the circuit board. The fan, which is connected to the motor shaft on the machine, was disconnected from the motor shaft to allow operation in different air streams. A new fan was fastened in the same position. This newly installed fan was driven at different voltages by an external DC power supply, independent of the rotational speed of the motor shaft. In order to maintain constant ambient conditions in the experiments, thermometers were put in certain places of the test room for temperature control with air conditioning systems. And also all variable parameters were selected taking into account system components used in commercial heat pump systems and their structural limitations.

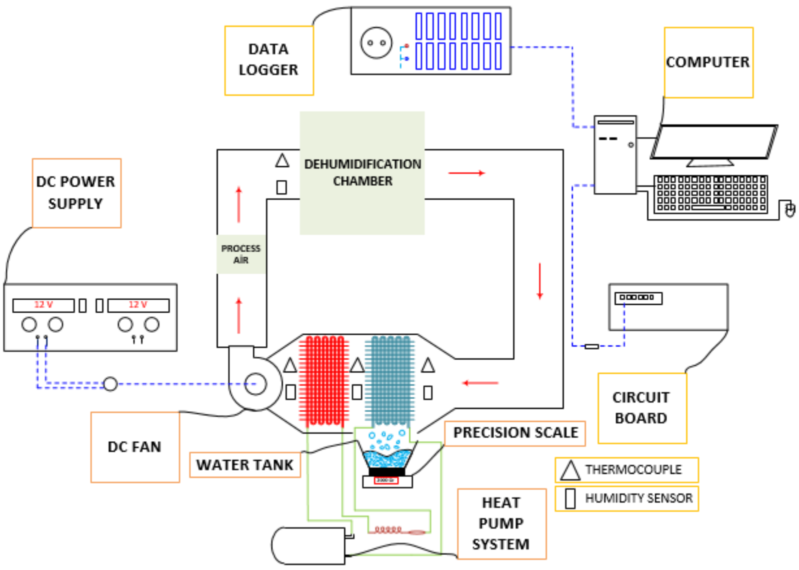

Figure 4. Experimental setup

Experimental Method

Three different parameters were examined in the experiments. These three parameters consist of three levels of capillary length and refrigerant charge, and two levels of evaporator tube volume. At least three retests were performed for each type of experiment. Taking into account parameters, level, and repetitions, the optimum experimental set was planned and implemented in the full factorial design of experiment (DOE) Minitab software [18]. A total of 54 experiments were formed. Due to the fact that R290 refrigerant is a flammable gas, maximum charge according to safety regulations is limited to 150 grams for domestic and commercial refrigerated appliances and similar applications [15]. For this reason, the initial amount of refrigerant was taken at 150 grams and the charge amount was gradually reduced. Experiments were carried out such that ambient temperature and relative humidity were constant at 23 ± 2 ° C and 50 ± 5% relative humidity, respectively. A precision scales and water tank were placed under the evaporator. With these scales, amount of moisture condensed per second was measured. That is, condensation performance was experimentally determined. Since it was difficult to determine the amount of moisture collected per seconds with the precision scales, the amount of moisture collected during the steady-state period was measured and divided into duration time to find condensation performance.

Table 1. The accuracy of measuring instruments and experiment conditions

Parameters	Value
Thermocouple	± 0.2
Humidity Sensor	1.8 %
Air Volume Flow Rate	1.13%
Precision Scales	± 1 g
Environment Temperature	23 ± 2 ◦C
Environment Absolute Humidity	50 ± 5 %
Experiment Duration	100 min.

Experimental Study

In this study, the effect of three different parameters such as capillary length, refrigerant charge amount and evaporator tubes volume on heat pump system were investigated. The lengths of the capillary are 75 cm, 100 cm, and 125 cm, the refrigerant charge amounts are 110 g, 130 g and 150 g, and evaporator have 24 tubes, 30 tubes and a total of 54 tests were carried out.

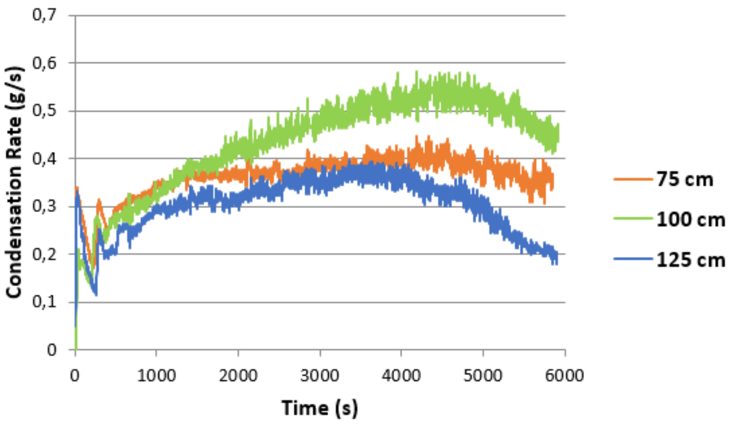

Figure 5. Effect of capillary tube length on condensation performance

Temperature and relative humidity values were measured at specific locations in the air channel. Based on these values, the specific moisture values of the air at the inlet and outlet of the evaporator, the specific moisture change, the density, the mass flow rate and amount of condensate were calculated. The condensation performance analysis was done for the steady-state, where the temperature and pressure of the heat pump system are virtually constant.

Effect of capillary tube length on condensation performance of the heat pump in which all parameters except the capillary tube are kept constant is shown in **Figure 5**. Experimental results showed that evaporation and condensation temperatures decreased and approach at each other when the capillary length increased from 75 cm to 100 cm. In addition, capacities increased, the input power of the compressor decreased and the system COP (5,24 for 75cm and 5,74 for 100 cm) increased. The system has remained in the multi-phase region for a longer time. On the other hand, when length was increased from 75 cm to 125 cm, there was a minor increase in system pressure and temperature, and the power picked up by the compressor increased by 3%. This is an exact opposite situation of that 100cm capillary length system reaction. The cooling capacity of the system has increased slightly but remained in multi-phase region for a shorter time. As a result, condensation performance deteriorated.

When 24-tubes evaporator was used instead of 30-tubes, tube volume was reduced by 20%. When evaporator tube volume was changed from 30-tubes to 24-tubes, no drastic change was observed in condensation and evaporation temperatures. Effect of evaporator tube volume is directly related to air flow. Experimental results showed that the most suitable design for air flow used in this experiment is 24-tubes evaporator. The system has been remained in multi-phase region for a longer time for 24-tubes design and has achieved a more intensive and efficient condensation process.

The refrigerant charge amount was decreased gradually from 150 grams to 130 grams and 110 grams. This severely affected the cooling and heating capacities. When the amount of charge was reduced from 150 grams to 130 grams, the condensation temperature decreased while the evaporation temperature was almost constant. As

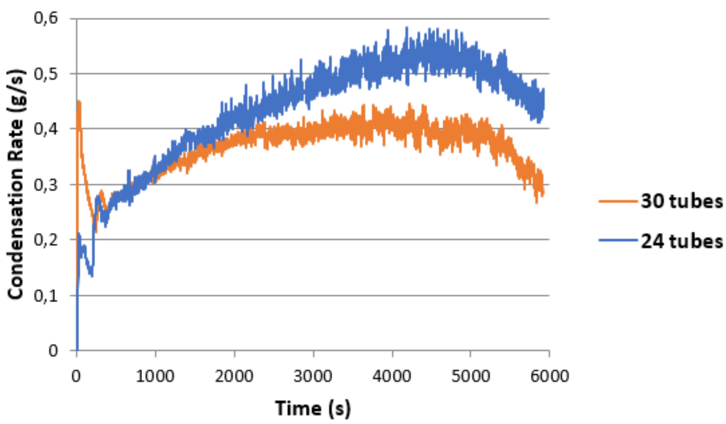

Figure 6. Effect of evaporator tube volume on condensation performance

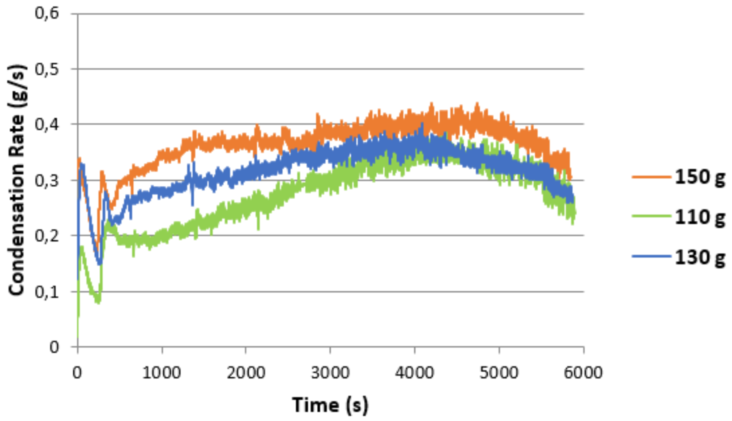

Figure 7. Effect of refrigerant charge amount on condensation performance

system temperatures approached each other, the system COP increased and input power of the compressor decreased. However, the process air has not been sufficiently heated and amount of moisture removed has decreased. Due to decreasing charge amount, refrigerant remained in single-phase region for a longer time. Condensation performance has reduced due to these reasons. Similar effects were observed when the charge decreased from 150 grams to 110 grams. However, temperature of condensation and evaporation have been fallen at same rate as this passing. Reduction of the system temperature and pressures, shrinkage of evaporation distance in multi-phase region, and the decrease of refrigerant flow circulating in the system at these temperatures caused capacities to drop according to 150 grams. For this reason, condensation performance on the evaporator has reduced.

UNCERTAINTY ANALYSIS

The accuracy of experimental results is affected by measurement devices and errors resulting from the experimental setup. A couple of methods have been recommended to determine the error ratios pertaining to parameters that are calculated by using the data obtained from the experiments. The uncertainty analysis method developed by Kline and McClintock [16] for error analysis of experimental findings is one such method. For the error analysis that was conducted for this experimental study, the uncertainty analysis method that was used was a more sensitive one compared to other methods. The values of uncertainties are listed in **Table 1**.

Let us assume that the quantity to be calculated by making measurements in the experiment apparatus is $\dot{m}_v$(condensation rate) and there are n independent variables pertaining to condensation rate (φ1, φ2, T1, T2, $\dot{V}$). In that case, $\dot{m}_v$= f (φ1, φ2, T1, T2, $\dot{V}$). Error ratios pertaining to each independent variable are $w_{\varphi_1}, w_{\varphi_2}$, w_{T_1}, w_{T_2} and $w_{\dot{V}}$, whereas the error ratio of the quantity $\dot{m}_v$ is $w_{\dot{m}_v}$. Uncertainties are expressed as follows:

$$W_R = \pm\left[\left(\frac{\partial R}{\partial x_1}.w_1\right)^2+\left(\frac{\partial R}{\partial x_2}.w_2\right)^2+\left(\frac{\partial R}{\partial x_3}.w_3\right)^2+\left(\frac{\partial R}{\partial x_4}.w_4\right)^2+\left(\frac{\partial R}{\partial x_5}.w_5\right)^2\right]^{1/2} \quad (16)$$

$$W_{\dot{m}_v} = \pm\left[\left(\frac{\partial \dot{m}_v}{\partial \varphi 1}.w_{\varphi_1}\right)^2+\left(\frac{\partial \dot{m}_v}{\partial \varphi 2}.w_{\varphi_2}\right)^2+\left(\frac{\partial \dot{m}_v}{\partial T1}.w_{T_1}\right)^2+\left(\frac{\partial \dot{m}_v}{\partial T2}.w_{T_2}\right)^2+\left(\frac{\partial \dot{m}_v}{\partial \dot{V}}.w_{\dot{V}}\right)^2\right]^{1/2} \quad (17)$$

The result of the calculations is that the maximum uncertainty of condensation rate is 6.4%.

RESULTS

Effect of different parameters such as capillary tube length, charge amount and evaporator tube volume on the condensation performance in R290 heat pump system was examined experimentally and theoretically. Theoretical condensation performance values showed a deviation of average 5%, confirming the experimental results. These measured values showed the correctness of the position of the humidity and temperature sensors, which is similar to the amount of water collected in the water tank.

Experimental and theoretical results;

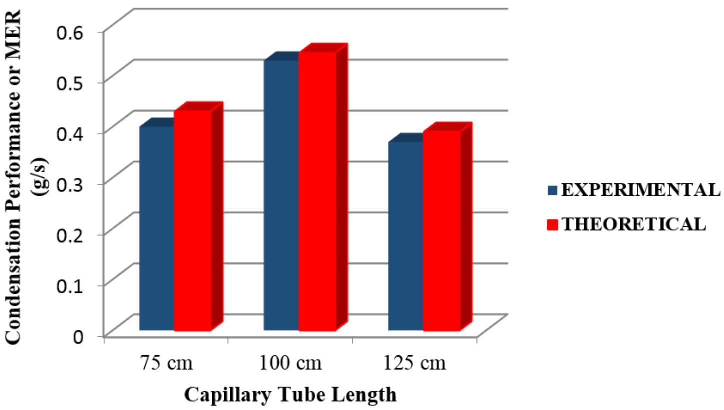

Figure 8. Effect of capillary tube length on the MER

System temperatures of 100cm capillary tube decreased and approach each other compared to 75 cm. Depending on increasing in capacity due to a slight drop in the evaporation pressure, R290 refrigerant remained more in the multi-phase region. As a result, condensation performance of drying air got better by 35%. In the case of 125 cm capillary tube, moisture extraction performance from drying air has deteriorated by 11% due to the fact that system has remained in the multi-phase region for a shorter period of time compared to 75 cm. And also theoretical values showed a deviation maximum %4 for capillary tube length.

Effect of evaporator on the condensation process is directly related to air flow rate. For this reason, it is necessary to realize the optimum evaporator design according to the air flow to meet required capacity at the current system temperatures. In this study, 30% improvement was observed in condensation performance when passing from 30 tubes to 24 tubes. This results in a 30 tubes evaporator that needs to be used in larger air flow rate to perform more effective condensation. In order to support this, the same experiments were repeated at a low air flow rate. Experiments showed that condensation performance of a 24 tubes evaporator was 8% better than that of 30 tubes one. Theoretical condensation performance values showed a deviation of between 1% and 6% for evaporator tube volume.

Condensation performance and the reaction of the system to three different charge quantities was studied experimentally. When 110 grams of charge was increased by 130 grams and 150 grams, condensation performance improved by 8% and 20%, respectively. This is because evaporator remained in the multi-phase region for a longer time and was used more effectively. Theoretical calculations were not different from experimental results for charge amount and confirmed them.

The COP values for each case according to calculations made are shown in **Table 2**. System COP values were calculated as 5.24, 5.74 and 5.55 for 75 cm, 100 cm and 125 cm capillary tube length, respectively. The similar changes were observed for charge amount differences. COP increasing was obtained when passing from 24-tubes to 30-tubes.

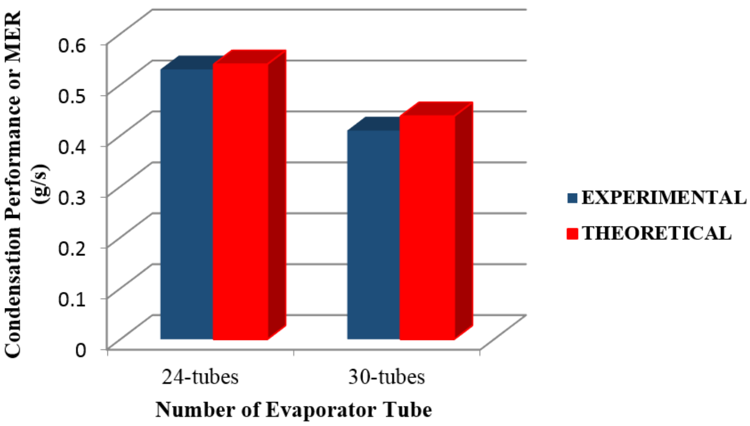

Figure 9. Effect of evaporator tube number on the MER

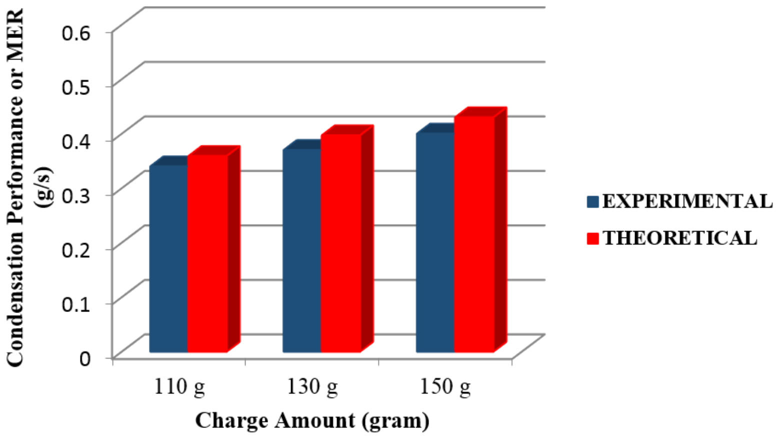

Figure 10. Effect of refrigerant charge amount on the MER

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Table 2. COP values for each case

Parameters	**Capillary Tube Length**			**Charge Amount**			**Number of Evaporator Tube**	
Case	75 cm	100 cm	125 cm	110 g	130 g	150 g	24-tubes	30-tubes
COP	5.24	5.74	5.55	5.24	5.93	5.51	5.4	5.95

Critical system parameters were changed and their effect on the condensation performance was examined experimentally. The system operating temperatures and multi-phase zone length were directly related to condensation performance. Experimental studies have shown that when the humid air passes through evaporator, evaporation temperature and presence of the refrigerant in the two-phase region improve the condensation performance. This also showed how important the optimum system design is when working on dehumidification. In these experimental studies, the most efficient parameter on condensation was found to be the capillary tube with a length of 100cm.

CONCLUSION

It is very important to know thermophysical properties of the refrigerant to be used when designing a system with heat pump and analyze it well. Because of the high evaporation latent heat, use of low refrigerant charge amounts reveals that the required heat exchanger tube volumes must also be small. The high volumetric capacity of the refrigerant shows that a compressor with small stroke volume, which is compatible with the lubrication system R290, needs.

In this study, the effects of capillary tube length, evaporator tube volume and charge amount on the condensation performance investigate experimentally under constant compressor speed. Results indicate that:

- In contrast to compressor used in this study, the effect of variable speed compressors on condensation performance, dehumidification time or energy consumption of R290 heat pump system can be examined.
- Since the evaporator is a very effective parameter on the condensation performance, proper evaporator designs according to R290 can be carried out to improve the condensation performance.
- With the increase of refrigerant charge amount, condensation performance enhances because of system have large components.
- The COP increase does not always positively affect condensation performance. For example, when 100 cm capillary is used instead of 75 cm capillary, COP and condensation performance increased. However, when the 125 cm capillary is used instead of 75 cm capillary, COP increased, but condensation performance was affected adversely. For this reason, research on condensation performance shouldn't be conducted just based on the COP value.

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NOMENCLATURE

GWP	Global warming potential
ODP	Ozone depletion potential
GHGs	Greenhouse gases
CFC	Chlorodifluoromethane
HCFCs	Hydrochlorofluorocarbon
HFCs	Hydrofluorocarbons
HCs	Hydrocarbons
R290	Propane
TEWI	Total equivalent warming impact
COP	Coefficient of performance
SCOP	Seasonal coefficient of performance
EER	energy efficiency ratio
$\dot{m}_r$	refrigerant mass flow rate [kg/s]
$\dot{m}_a$	air flow rate [kg/s]
$\dot{m}_v$	condensation rate [kg/s]
$\dot{m}_{vapour}$	vaporescent water in chamber [kg/s]
$\dot{V}$	air volume flow rate [m^3/s]
c_p	dry air specific heat [kJ/kgK]
$\dot{Q}_{cond}$	condenser capacity [kW]
$\dot{Q}_{eva}$	evaporator capacity [kW]
$\dot{Q}_{loss}$	loss heat [kW]
$\dot{Q}_{latent}$	latent heat capacity [kW]
$\dot{Q}_{sensible}$	sensible heat capacity [kW]
$\dot{W}_{comp}$	compressor power [kW]
h_{fg}	evaporation enthalpy [kJ/kg]
h	air or refrigerant enthalpy [kJ/kg]
T	temperature [K]
η_s	isentropic efficiency
ρ	dry air density [kg/m3]
ω	specific humidity [kg/kg]
MER	moisture extraction rate [g/s]
HP	heat pump
n	number of measurements
R	dimension to measure
x	factors that change the measurement
w	error rates of arguments
W_R	total uncertainty
1, 2,..	refrigerant position number in refrigerant cycle

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