Çukurova Üniversitesi Mühendislik Mimarlık Fakültesi Dergisi, 32(3), ss. 91-102, Eylül 2017
Çukurova University Journal of the Faculty of Engineering and Architecture, 32(3), pp. 91-102, September 2017

Performance and Economic Analysis of a Variable Refrigerant Flow (VRF) System

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Geliş tarihi: 04.05.2017   Kabul tarihi: 25.09.2017

Abstract

This study deals with the exergetic modeling and performance/cost evaluation of a variable refrigerant flow (VRF) air conditioning system. An experimental setup was established to investigate the system performance under cooling conditions. System mainly consists of one outdoor unit and two indoor units. Outdoor unit equipped with two compressors (one variable speed and one constant speed), condenser, and four way valve is connected to two indoor units. Exergy, cost, energy and mass (EXCEM) analysis was applied to this system for the first time to the best of the authors` knowledge. The relations between thermodynamic losses and capital costs were also parametrically investigated. Experimental results show that the greatest irreversibility (exergy destruction) occurs in the condenser, followed by the evaporators. Exergy efficiency of the whole system on the exergetic product/fuel basis was calculated to be 85.84% at a reference state temperature of 25 ^{o}C. Exergy efficiency and exergy loss rate were in the range of 85.27-86.55% and 0.919-0.916 MW/USD, respectively, based upon the conditions and parameters considered in the present study.

Keywords: Variable refrigerant flow, Energy analysis, Exergy analysis, Exergoeconomic

Değişken Soğutucu Akışkan Debili Bir Sistemin Performans ve Ekonomik Analizi

Öz

Bu çalışma kapsamında, değişken soğutucu akışkan debili (VRF) bir klima sisteminin ekserjetik modellemesi ve performans değerlendirmesi ele alınmıştır. Soğutma koşulları için sistem performansını araştırılabilmek için bir deney düzeneği kurulmuştur. Sistem esas olarak bir dış üniteden ve iki iç üniteden oluşmaktadır. İki adet kompresör (bir değişken hız ve bir sabit hız), konderser ve dört yollu valf ile donatılmış dış ünite, iki içi üniteye bağlanmıştır. Bu çalışmada sisteme ekserji, maliyet, enerji ve kütle (EXCEM) analizi uygulanmış ve termodinamik kayıplar ile maliyetler arasındaki ilişkiler parametrik

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olarak incelenmiştir. Deneysel sonuçlar, en büyük tersinmezliğin (ekserji tahribatının) kondenserde meydana geldiğini ve bunu evaporatörlerin izlediğini göstermektedir. Ekserjetik ürün/yakıt bazında sistemin ekserji verimi, 25 °C referans sıcaklığında % 85,84 olarak hesaplanmıştır. Bu çalışmada ele alınan koşullarda, sistemin ekserji verimi ve ekserji kayıp oranları sırasıyla % 85,27-86,55 ve 0,919-0,916 MW/USD aralığındadır.

Anahtar Kelimeler: Değişken soğutucu akışkan debi, Enerji analizi, Ekserji analizi, Ekserjekonomi

1. INTRODUCTION

One of the main purposes of buildings is to provide a comfortable environment for its occupants. Comfort conditions are provided with heating, cooling, ventilating and air conditioning (HVAC) systems. These systems are major energy users in residential and commercial buildings. Since the standard of living and utilization of HVAC systems are rising dramatically in the world, the amount of energy consumed for heating/cooling is also increasing and is estimated to be more than half of the total energy consumption in buildings [1-3].

There are a wide range of air conditioning systems such as basic window-fitted units, small split systems, medium scale package units, large chilled water systems, and currently the variable refrigerant flow (VRF) systems. VRF is an air-condition system configuration where there is one outdoor unit and multiple indoor units. The term variable refrigerant flow refers to the ability of the system to control the amount of refrigerant flowing to the multiple indoor units, enabling the use of many indoor units of differing capacities and configurations connected to a single outdoor unit. VRF systems include sophisticated controls integrated with the units that may not require a separate building automation system, when such a system is part of the project requirements. VRF systems include self-diagnostics and monitoring points, as well as the ability to communicate with a wide variety of other building systems with non-proprietary building automation communication protocols. VRF systems combine many of the features of other HVAC systems, which offer energy efficiency with a limited number of components relative to systems with central plants. VRF systems have limited space requirements, particularly for the distribution system inside the building [4-6].

A number of investigations are reported in the literature regarding of the VRF systems. Aynur [7] presented the overview of the configurations of the outdoor and indoor units of a multi-split variable refrigerant flow system, and its operations, applications, marketing and cost. This review study revealed that even though the main drawback of the VRF system was the high initial cost compared to the common air conditioning systems, due to the energy saving potential of the VRF system, the estimated payback period of the VRF system compared to an air cooled chiller system in a generic commercial building could be about 1.5 year. Jain et al. [8] considered the problem of designing a scalable control architecture for large-scale variable-refrigerant-flow systems. The results showed that the ability of the proposed control architecture and design to provide both high performance and reduced energy consumption was demonstrated through a simulated case study. Kwon et al. [9] installed multifunctional variable refrigerant flow (MFVRF) in an office building and fully instrumented to measure the performance of the system under a wide range of outdoor weather conditions. The effects of a part-load ratio, a hot water demand and a heat recovery operation mode on the performance of the MFVRF system were investigated in a field test for the heating and shoulder seasons. They showed that the daily performance factor was 2.14 and 3.54 when the ratio of daily total cooling energy to daily total energy was 13.0% and 28.4%, respectively, at the similar outdoor weather conditions. Aynur et al. [10-11] investigated energy saving and indoor air condition enhancing potentials by integrating the variable refrigerant flow and heat pump desiccant (HPD) systems in a field performance test during heating and cooling season. Three different operating modes: non-ventilated, HPD ventilation assisted and HPD ventilation–dehumidification assisted VRF systems were investigated. It was concluded that the HPD ventilation–dehumidification assisted VRF outdoor units consume less energy than the HPD ventilation assisted ones, but more than the non-ventilated ones, while providing the best indoor thermal comfort and indoor air quality conditions. For the total system, the HPD ventilation–dehumidification assisted VRF systems consume less energy than the HPD ventilation assisted ones. Zhu et al. [12] presented an optimal control strategy for minimizing the energy consumption of variable refrigerant flow (VRF) and variable air volume (VAV) combined air conditioning systems. The combined system was proposed to take advantages of VAV systems to solve the ventilation problem of VRF systems. Results indicated that the optimal control strategy reduces energy consumption of the combined system by 32.17% in summer and 2.47% in winter. The overall energy efficiency was enlarged by 12.18% in summer and 3.37% in winter, compared with the benchmark operation strategy. Aynur et al. [13] compared variable air volume (VAV) and variable refrigerant flow (VRF) systems in an existing office building environment under the same outdoor conditions and internal load profiles for an entire cooling season. It was found that the VRF system promised 27.1–57.9% energy-saving potentials depending on the system configuration, indoor and outdoor conditions, when compared to the VAV system. Liu and Hong [14] conducted a preliminary comparison of energy efficiency between the air-source variable refrigerant flow and ground source heat pump (GSHP) systems using available building energy analysis software and the performance data/curves from VRF and GSHP equipment manufacturers. It was shown that, for conditioning the same small office building, GSHP system is more energy efficient than VRF system. Kwon et al. [15] investigated the effects of the subcooling heat exchanger (SCHX) on the performance of the multi-split variable refrigerant flow system with long pipe in a field test during the cooling season. It was found that VRF system with SCHX improved the cooling performance factor (CPF) about 8.5% under similar outdoor temperature profiles, as compared to the baseline without SCHX. However, when the fraction of total refrigerant that passes through the SCHX was higher than 5.27%, the CPF starts to decrease due to the decreased refrigerant mass flow rate through the evaporators.

During the past decade, there has been an increasing interest in using exergy as a potential analysis tool for design, analysis and performance evaluation of energy systems [16-19]. The thermodynamic quantity exergy, which can be used to assess and improve energy systems, can help better understand the benefits of utilizing green energy by providing more useful and meaningful information than energy provides. Exergy analysis is employed to detect and to evaluate quantitatively the causes of the thermodynamic imperfection of the process under consideration. Exergy analysis has been applied to different types of air conditioning systems by various researchers [20-24].

Although VRF systems are introduced in the world more than 25 years ago and currently very popular in many countries, their exergetic performance is yet unknown and works related to exergetic and exergoeconomic analysis of a VRF system using EXCEM analysis are not available in current literature. This provided the prima motivation behind doing the present study. In this study, we have conducted a comprehensive exergy and exergoeconomic assessment of a VRF system.

2. EXPERIMENTAL SET-UP

Figure 1 shows schematic view of VRF system studied. System mainly consists of one outdoor unit and two indoor units. Outdoor unit equipped with two compressors (one variable speed and one constant speed), condenser, and four way valve is connected to two indoor units and each indoor unit is installed into an office rooms. A variable speed compressor provides the variable refrigerant mass flow rate to the system depending on the heating or cooling load of the thermal zones by changing the compressor operation frequency. Instead, the constant speed compressor runs in order to cover the higher cooling or heating loads than what the variable speed compressor can cover. In the system, ceiling/floor and cassette type indoor units are used for air conditioning of different zones. Each indoor unit equipped with fan to force the air through the heat exchanger and electronic expansion valve to control the refrigerant mass flow rate. Figure 2 shows the photographic view of outdoor and indoor units used in this study.

In this study, the system was operated in cooling mode so that refrigerant flow paths in the cooling mode are shown in Figure 1. The refrigerant (R410A) enters the compressor (I) and is compressed to the condenser pressure. The temperature of the refrigerant increases during this compression process to well above the temperature of the surrounding medium. The refrigerant then enters the condenser (II) at state 1 and leaves as saturated liquid as a result of heat rejection (process 10-11) to the surroundings. The refrigerant at state 2 is throttled to the evaporator pressure by passing it through an expansion valves (III, V). The temperature of the refrigerant drops below the temperature of the rooms during this process. The refrigerant enters the evaporators (IV, VI) at state 4 and 7 as a low-quality saturated mixture, and it completely evaporates by absorbing heat (process 12-13 and process 14-15) from the rooms. The cycle is completing as the refrigerant leaves the evaporators (state 5 and 8) and reenters the compressor (state 9).

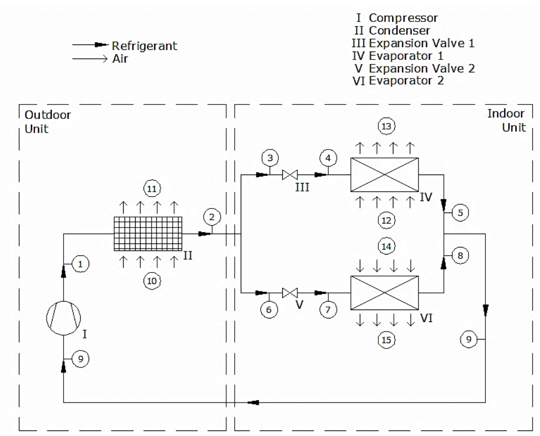

Figure 1. A schematic view of the VRF system studied

(a) (b) (c)

Figure 2. The photographic view of outdoor (a) and indoor units (b,c)

Duracomm digital thermometers and Pakkens manometers were installed on the refrigerant circuit to measure the temperature and pressure of the refrigerant at the inlet and outlet of each component, as shown in Figure 1. A turbine type flow meter was used to measure the total refrigerant flow rate of indoor units. Watt meter was used to measure the power consumption of the outdoor and indoor units. The real time system information, such as the expansion valves opening, thermostat "on/off", compressor frequency, and fan speed, were recorded via the local multi-split VRF system network program. Uncertainty analysis is needed to prove the accuracy of the experiments. An uncertainty analysis is performed using the method described by Holman [25]. Accuracies of the measuring devices and uncertainty of the calculated parameters are presented in Table 1.

Table 1. Accuracy of the measuring devices and the uncertainty of the calculated parameters

Measurements	*Accuracy*
Temperature (air)	±0.2 ^{o}C
Temperature (refrigerant)	±0.4 ^{o}C
Relative humidity	±3%
Pressure	±4.7 kPa
Flow meter	±0.5% of flow rate
Watt meter	±0.5% of measur.
Calculated parameters	*Uncertainty (%)*
Power consumption	±1.8
COP	±2.5
Exergy efficiency	±3.2

3. MODELING AND ANALYSIS

Mass, energy and exergy balances are employed to find the heat input, the rate of exergy destruction, and energy and energy efficiencies [23].

The mass and energy balance for a steady state open system can be written as:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

$$\dot{Q}-\dot{W}=\sum \dot{m}_{out}\, h_{out}-\sum \dot{m}_{in}\, h_{in} \tag{2}$$

Here, subscripts in and out shows inlet and outlet states, $\dot{Q}$ is the heat rate, $\dot{W}$ is the work rate, h is the specific enthalpy and $\dot{m}$ is the mass flow rate.

The general exergy balance can be expressed in the rate form as:

$$\dot{E}x_{in} - \dot{E}x_{out} = \dot{E}x_{dest} \tag{3a}$$

where $\dot{E}x_{in} - \dot{E}x_{out}$ stands for the rate of net exergy transfer by heat, work and mass and $\dot{E}x_{dest}$ stands for the rate of net exergy destruction. The general exergy balance can also written as:

$$\dot{E}x_{heat} - \dot{E}x_{work} + \dot{E}x_{mass,in} - \dot{E}x_{mass,out} = \dot{E}x_{dest} \tag{3b}$$

Using Eq. (3b), the rate of formation of the general exergy balance can also be written as:

$$\Sigma\left(1-\frac{T_0}{T_k}\right)\dot{Q}_k - \dot{W} + \Sigma\dot{m}_{in}\psi_{in} - \Sigma\dot{m}_{out}\psi_{out} = \dot{E}x_{dest} \tag{4}$$

where $\dot{Q}_k$ is the heat transfer rate through the boundary at temperature T_k at location k, $\dot{W}$ is the work rate, ψ is the flow (specific) exergy, h is the enthalpy and the subscript zero indicates properties at the reference (dead) state of P_o and T_o.

The specific flow exergy of refrigerant or water is evaluated as:

$$\psi_{ref,\,water} = (h - h_0) - T_0(s - s_0) \tag{5}$$

The total flow exergy of air is determined as [16]:

$$\begin{aligned}\psi_a &= (C_{p,a} + \omega C_{p,v})T_0[(T/T_0) - 1 - \ln(T/T_0)] \\ &\quad + (1+1.6078\omega.6\ {}_aT_0\ \ln(P/P_0) \\ &\quad + R_aT_0\left\{\begin{matrix}(1+1.6078\omega.60[(1+1.6078\omega_0)/ \\ (1+1.6078\omega.6+1.6078\omega.60\omega/\omega_0)\end{matrix}\right\}\end{aligned} \tag{6}$$

where s is the entropy and the specific humidity ratio is:

$$\omega = \dot{m}_v/\dot{m}_a \tag{7}$$

The exergy rate is determined as:

$$\dot{E}x = \dot{m}\psi \tag{8}$$

Coefficient of the performance (COP) of the system is defined as the ratio between the total cooling capacity of the indoor units ($\dot{Q}_{TCC}$) and total energy input ($\dot{W}_T$) to the system:

$$COP = \frac{\dot{Q}_{TCC}}{\dot{W}_T} \tag{9}$$

where,

$$\dot{Q}_{TCC} = \dot{Q}_{evap,z1} + \dot{Q}_{evap,z2} \tag{10}$$

$$\dot{W}_T = \dot{W}_{comp} + \dot{W}_{condfan} + \dot{W}_{evapfan} \tag{11}$$

Exergy efficiency can be expressed as the ratio of the exergetic product ($\dot{P}$) to the exergetic fuel ($\dot{F}$):

$$\varepsilon = \frac{\text{Exergetic Product}}{\text{Exergetic Fuel}} = \frac{\dot{P}}{\dot{F}} \tag{12}$$

Van Gool's improvement potential on a rate basis, denoted $\dot{IP}$, is expressible as:

$$\dot{IP} = (1 - \varepsilon)(\dot{Ex}_{in} - \dot{Ex}_{out}) \tag{13}$$

The relative irreversibility (RI) is evaluated as:

$$RI = \frac{\dot{Ex}_{dest,i}}{\dot{Ex}_{dest,tot}} = \frac{\dot{I}_i}{\dot{I}_{Tot}} \tag{14}$$

where the subscript "i" denotes the it device.

Cost is an increasing, nonconserved quantity. The general balance equation can be written for cost as:

$$K_{in} + K_{gen} - K_{out} = K_a \tag{15}$$

where K_{in}, K_{out}, and K_a represent, respectively, the cost associated with all inputs, outputs and accumulations for the system. K_{gen} corresponds to the appropriate capital and other costs associated with the creation and maintenance of a system.

$$K_{eq} + K_{C,M} = K_{gen} \tag{16}$$

Exergy losses can be identified from the exergy rate balance in Eq. (3). There are two types of exergy losses: the "waste exergy output" which represents the loss associated with exergy that is emitted from the system, and the "exergy consumption" which represents the internal exergy loss due to process irreversibilities. These two exergy losses sum to the total exergy loss. Hence, the loss rate based on exergy, $\dot{L}_{ex}$, is defined as [26],

$$\dot{L}_{ex} = \dot{Ex}_{con} + \dot{Ex}_{out,W} \tag{17}$$

For a thermal system operating normally in a continuous steady-state steady-flow process mode, the accumulation terms in balance equations are zero. Hence all losses are associated with $\dot{L}_{ex}$. The exergy loss rate can be obtained through the following equations [26]:

$$\dot{L}_{ex} = \sum_{in} Ex.\text{flux rates} - \sum_{p} Ex.\text{flux rates} \tag{18}$$

where the summations are over all input streams and all product output streams.

A parameter, $\dot{R}$ is defined as the ratio of thermodynamic loss rate $\dot{L}$ to capital cost K as follows [26]:

$$\dot{R} = \frac{\dot{L}}{K} \tag{19}$$

The value of $\dot{R}$ generally depends on whether it is based on energy loss rate (in which case it is denoted $\dot{R}_{en}$), or exergy loss rate ($\dot{R}_{ex}$), while in this analysis $\dot{R}_{ex}$ values were used:

$$\dot{R}_{ex} = \frac{\dot{L}_{ex}}{K} \tag{20}$$

The following assumptions were made during the analyses:

- All processes are steady-state and steady-flow with negligible potential and kinetic energy effects and no chemical or nuclear reactions.
- Heat transfer to the system and work transfer from the system are positive.
- Heat transfer and refrigerant pressure drops in the tubing connecting the components are neglected.

Mass and energy balances as well as exergy destructions obtained from exergy balances for each of the components illustrated in Figure 1 can be expressed as follows:

Compressor (I):

$$\dot{m}_9 = \dot{m}_1 = \dot{m}_r \tag{21a}$$

$$\dot{W}_{comp} = \dot{m}_r (h_1 - h_9) \tag{21b}$$

$\dot{E}x_{dest,I} = \dot{E}x_9 + \dot{W}_{comp} - \dot{E}x_1$ (21c)

Condenser (II):

$\dot{m}_1 = \dot{m}_2 = \dot{m}_r$; $\dot{m}_{10} = \dot{m}_{11} = \dot{m}_{a,cond}$ (22a)

$\dot{Q}_{cond} = \dot{m}_r (h_1 - h_2)$; $\dot{Q}_{cond} = \dot{m}_{a,cond} (h_{11} - h_{10})$ (22b)

$\dot{E}x_{dest,II} = \dot{E}x_1 + \dot{E}x_{10} + \dot{W}_{cond\ fan} - (\dot{E}x_2 + \dot{E}x_{11})$ (22c)

Expansion valve 1 (III):

$\dot{m}_3 = \dot{m}_4 = \dot{m}_{r,z1}$ (23a)

$h_3 = h_4$ (23b)

$\dot{E}x_{dest,III} = \dot{E}x_3 - \dot{E}x_4$ (23c)

Evaporator 1 (IV):

$\dot{m}_4 = \dot{m}_5 = \dot{m}_{r,z1}$; $\dot{m}_{12} = \dot{m}_{13} = \dot{m}_{a,evap,z1}$ (24a)

$\dot{Q}_{evap,z1} = \dot{m}_{r,z1} (h_5 - h_4)$;
$\dot{Q}_{evap,z1} = \dot{m}_{a,evap,z1} (h_{12} - h_{13})$ (24b)

$\dot{E}x_{dest,IV} = \dot{E}x_4 + \dot{E}x_{12} + \dot{W}_{evap.fan} - (\dot{E}x_5 + \dot{E}x_{13})$ (24c)

Expansion valve 2 (V):

$\dot{m}_6 = \dot{m}_7 = \dot{m}_{r,z2}$ (25a)

$h_6 = h_7$ (25b)

$\dot{E}x_{dest,V} = \dot{E}x_6 - \dot{E}x_7$ (25c)

Evaporator 2 (VI):

$\dot{m}_7 = \dot{m}_8 = \dot{m}_{r,z2}$; $\dot{m}_{14} = \dot{m}_{15} = \dot{m}_{a,evap,z2}$ (26a)

$\dot{Q}_{evap,z2} = \dot{m}_{r,z2} (h_8 - h_7)$ $\dot{Q}_{evap,z2} = \dot{m}_{a,evap,z2} (h_{14} - h_{15})$ (26b)

$\dot{E}x_{dest,VI} = \dot{E}x_7 + \dot{E}x_{14} + \dot{W}_{evap.fan} - (\dot{E}x_8 + \dot{E}x_{15})$ (26c)

Exergy efficiencies of the variable refrigerant flow (VRF) system studied and its components are evaluated as follows:

Overall VRF system (I-VI):

$$\varepsilon_{sys} = \frac{\Sigma \dot{P}_{sys}}{\Sigma \dot{F}_{sys}} \quad (27)$$

Compressor (I):

$$\varepsilon_I = \frac{\dot{E}x_1 - \dot{E}x_9}{\dot{W}_{comp}} \quad (28)$$

Condenser (II):

$$\varepsilon_{II} = \frac{\dot{E}x_{11} - \dot{E}x_{10}}{\dot{E}x_1 - \dot{E}x_2 + \dot{W}_{condfan}} \quad (29)$$

Expansion valve 1 (III):

$$\varepsilon_{III} = \frac{\dot{E}x_4}{\dot{E}x_3} \quad (30)$$

Evaporator 1 (IV):

$$\varepsilon_{IV} = \frac{\dot{E}x_{12} - \dot{E}x_{13}}{\dot{E}x_5 - \dot{E}x_4 + \dot{W}_{evapfan}} \quad (31)$$

Expansion valve 2 (V):

$$\varepsilon_V = \frac{\dot{E}x_7}{\dot{E}x_6} \quad (32)$$

Evaporator 2 (VI):

$$\varepsilon_{IV} = \frac{\dot{E}x_{14} - \dot{E}x_{15}}{\dot{E}x_8 - \dot{E}x_7 + \dot{W}_{evapfan}} \quad (33)$$

4. RESULTS AND DISCUSSION

A series of experiments were performed during the cooling season of 2014 to determine the performance characteristics of the system investigated. All the experiments were carried out during continuous eight hours, from 9.00 a.m. to 5.00 p.m. In the present study, the results obtained from the experiments on 15 July 2014 at 14:00, which were typical, are given and discussed. In the calculations, the dead (reference) state values were considered to be 25 ^{o}C and 101.325 kPa. The value for the dead state humidity ratio was taken to be daily mean value of ambient air humidity ratio. The thermodynamic properties of air and R410A were found by using Engineering Equation Solver (EES) software package program.

Temperature, pressure and mass flow rate data for refrigerant R410A and air are given in Table 2 according to their state numbers specified in Figure 1. The exergy rates were also calculated for each state as presented in Table 2, while exergy destruction, exergy efficiency, improvement potential rate ($\dot{IP}$) and relative irreversibility (RI) data for representative components of the whole system are given in Table 3.

Table 2. Exergy analysis results of the VRF system

State no.	Description	Fluid	Phase	Temperature (°C)	Pressure (kPa)	Specific humidity ratio (kg water/ kg dry air)	Specific enthalpy (kJ/kg)	Specific entropy (kJ/kgK)	Mass flow rate (kg/s)	Specific exergy (kJ/kg)	Exergy rate (kW)
0	-	Moist air	Dead state	25	101.325	0.0166	-	-	-	-	-
0'	-	Refrigerant (R410A)	Dead state	25	101.325	-	321.1	1.411	-	-	-
1	Compressor outlet/Condenser inlet	Refrigerant	Super heated vapor	84.7	2840	-	341.5	1.121	0.149	106.820	15.92
2	Condenser outlet	Refrigerant	Liquid	45.1	2740	-	135.6	0.487	0.149	89.852	13.39
3	Expansion valve 1 inlet	Refrigerant	Liquid	38.0	2688	-	122.6	0.446	0.084	89.070	7.48
4	Expansion valve 1 outlet/Evaporator 1 inlet	Refrigerant	Mixture	26.0	1694	-	122.6	0.451	0.084	87.580	7.36
5	Evaporator 1 outlet	Refrigerant	Super heated vapor	17.3	994	-	295.0	1.075	0.084	74.028	6.22
6	Expansion valve 2 inlet	Refrigerant	Liquid	39.2	2650	-	124.3	0.452	0.065	88.982	5.78
7	Expansion valve 2 outlet/Evaporator 2 inlet	Refrigerant	Mixture	29.0	1833	-	124.3	0.455	0.065	88.088	5.73
8	Evaporator 2 outlet	Refrigerant	Super heated vapor	19.2	1103	-	294.3	1.063	0.065	76.904	5.00
9	Compressor inlet	Refrigerant	Super heated vapor	12.1	710	-	296.6	1.114	0.149	64.006	9.54
10	Condenser inlet	Air	Gas	36.1	101.325	0.017	81.9	-	3.920	0.230	0.90
11	Condenser outlet	Air	Gas	40.2	101.325	0.017	89.6	-	3.920	0.400	1.57
12	Evaporator 1 inlet	Air	Gas	26.1	101.325	0.012	55.5	-	0.660	0.019	0.01
13	Evaporator 1 outlet	Air	Gas	13.3	101.325	0.008	34.3	-	0.660	0.596	0.39
14	Evaporator 2 inlet	Air	Gas	25.9	101.325	0.012	58.4	-	0.600	0.072	0.04
15	Evaporator 2 outlet	Air	Gas	15.8	101.325	0.009	39.4	-	0.580	0.390	0.23

Table 3. Exergy, improvement potential rate ($\dot{IP}$) and relative irreversibility (RI) data for representative components of the whole system

Item number	Component	Exergetic product (exergy output) rate $\dot{P}$ (kW)	Exergetic fuel (exergy input) rate $\dot{F}$ (kW)	Exergy destruction rate $\dot{Ex}_{dest}$ (kW)	Exergy efficiency ε (%)	Exergetic improvement potential rate $\dot{IP}$ (kW)	Relative irreversibility RI (%)
I	Compressor	6.38	6.45	0.07	98.97	0.0007	1.26
II	Condenser	0.67	2.53	2.71	26.36	1.9981	51.54
III	Expansion valve 1	7.33	7.45	0.12	98.33	0.0021	2.37
IV	Evaporator 1	0.38	1.13	1.25	33.62	0.8316	23.80
V	Expansion valve 2	5.76	5.82	0.06	99.00	0.0006	1.11
VI	Evaporator 2	0.18	0.73	1.05	24.99	0.7867	19.92
I-VI	Overall system	20.71	24.12	5.26	85.84	0.7452	100.00
COP=3.096							

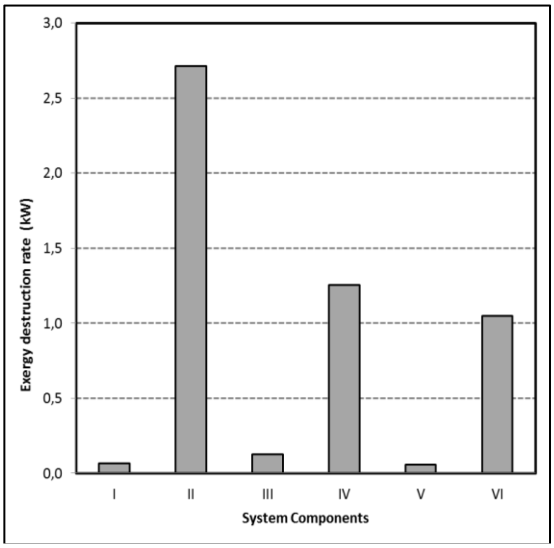

Figure 3. Exergy destruction rate values of the components used in the system

As can be seen from Table 3 and Figure 3, the greatest exergy destruction on the system occurs in the condenser, followed by the evaporator 1, evaporator 2 and the other components. It is clear from Table 3 that the highest irreversibility occurs in condenser, evaporator 1 and evaporator 2 with the relative irreversibility of 51.54%, 23.80% and 19.92% for the whole system, respectively. As can be observed the influence of the irreversibility in the outdoor unit represents more than 50% of the whole system. This is mainly due to heat transfer process in the condenser and also in the heat and friction losses in the mobile parts of the compressors and condenser fans. The exergy efficiency and the coefficient of performance of the system were calculated to be 85.84% and 3.09, respectively.

Van Gool's improvement potential on the rate basis ($\dot{IP}$) given in Eq. (13) is calculated for the each component of the system using the values listed in Table 3. It is found that the condenser has the highest $\dot{IP}$ value with 1.998 kW, followed by the evaporator 1 and evaporator 2 with 0.832 and 0.787 kW, respectively.

The main parameters for performing exergoeconomic analysis that were calculated from the experimental data are listed in Table 4. The costs shown in this table are in 2014 US dollars. The exergoeconomic analysis for the system components showed that condenser, evaporator 1 and evaporator 2 were inefficient due to the overall system (OS) results. Particularly, evaporator 1 is important as its exergy loss rate ($\dot{R}_{ex}$) value was 3.55 times greater than OS.

Table 4. Performance parameters for the VRF system investigated

Item No	Component	K^a (USD)	ε (%)	$\dot{IP}$ (kW)	$\dot{L}_{ex}$ (kW)	$\dot{R}_{ex}$ (MW/USD)
I	Compressor	1350	98.97	0.0007	0.0700	0.0519
II	Condenser	2785	26.36	1.9981	2.7100	0.9731
III	Expansion Valve 1	250	98.33	0.0021	0.1200	0.4800
IV	Evaporator 1	385	33.62	0.8316	1.2500	3.2468
V	Expansion Valve 2	250	99.00	0.0006	0.0600	0.2400
VI	Evaporator 2	742	24,99	0.7867	1.0500	1.4151
I-VI	Overall system	5762	85,84	3.619	5.2600	0.9129

The analyses were performed at different dead state temperatures ranged from 20 to 35 ^{o}C. Figure 4 and 5 illustrate variation of exergy efficiency (ε) and $\dot{R}_{ex}$ with different dead state temperatures for VRF system. First of all, Figure 5 indicated that the variation of $\dot{R}_{ex}$ was obtained to be linear and decreases with the increase of dead state temperature. While the exergy efficiencies were obtained to vary between 85.27– 86.55%, $\dot{R}_{ex}$ were in the range of 0.916–0.919 MW/USD, respectively (Figure 4 and 5). It is also obvious from Figure 4-5 that exergy efficiency values increased as the temperature increased to the contrary of the $\dot{R}_{ex}$.

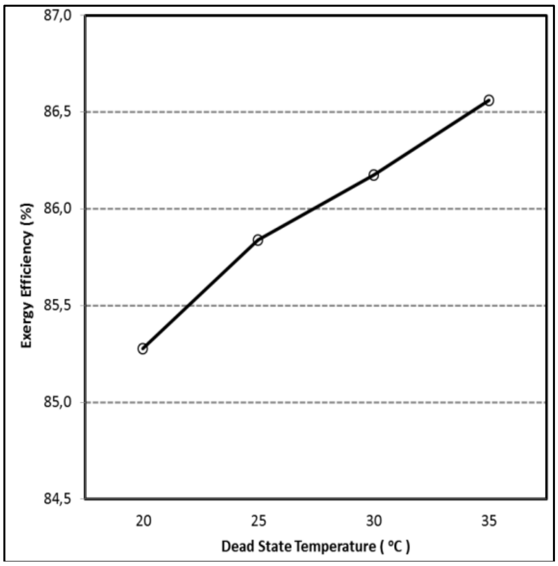

Figure 4. Variation of exergy efficiency values with dead state temperatures for the VRF system

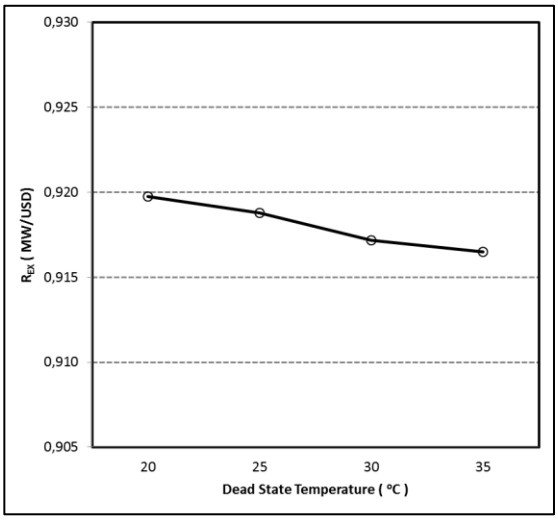

Figure 5. Variation of $\dot{R}_{ex}$ values with dead state temperatures for the VRF system

5. CONCLUSIONS

Variable refrigerant flow system for residences were exergetically modeled in this study, while the performance of a VRF system along with their essential system components was assessed through a comprehensive exergy analysis in the cooling mode. The main conclusions drawn from the results of the present study may be listed as follows:

- Exergy efficiency of the whole system on the exergetic product/fuel basis was calculated to be 85.84% at a reference state temperature of 25 ^{o}C.
- The COP value was found to be 3.09 for the whole system.
- The condenser has the maximum exergy destruction rate, followed by the evaporators.
- According to Van Gool's improvement potential rate ($\dot{IP}$), the condenser had the highest $\dot{IP}$ value, followed by the evaporators.
- The exergy efficiency of the system increased from 85.27-86.55% with increasing the reference state temperatures from 20 to 35 ^{o}C. These values can be increased by eliminating the factors like heat and friction losses that cause irreversibilities in the system.
- Exergy loss rate value of the system decreased from 0.919-0.916 MW/USD with increasing the reference state temperatures from 20 to 35 ^{o}C.
- There are various ways to describe exergy efficiency in the literature. In this regard, the use of the efficiency definition on the benefit/fuel basis is more convenient than that on the output/input basis.
- It may be concluded that exergy analysis is a useful tool for determining the locations, types and true magnitudes of energy losses, and therefore help in the design of more efficient energy systems. It is also a way to a sustainable development and reveals whether or not (and by how much) it is possible to

improve variable VRF systems by reducing inefficiencies.

- For a future work, the performance of the system could be evaluated in the heating mode.

6. REFERENCES

1. Huang, W.Z., Zaheeruddin, M., Cho, S.H., 2006. Dynamic Simulation of Energy Management Control Functions for HVAC Systems in Buildings, Energy Conversion and Management 47 (7–8), 926-943.
2. International Energy Agency (IEA), World Energy Outlook, OECD/IEA, France, 2012. www.worldenergyoutlook.org/publications/weo-2012/.
3. Ürge-Vorsatz, D., Cabeza, L., Serrano, S., Barreneche C., Petrichenko, K., 2015. Heating and Cooling Energy Trends and Drivers in Buildings, Renewable and Sustainable Energy Reviews 41, 85-98.
4. Goetzler, W., 2007. Variable Refrigerant Flow Systems, ASHRAE Journal 49 (4), 24-31.
5. Aynur, T.N., Hwang, Y., Radermacher, R., 2008a. Experimental Evaluation of the Ventilation Effect on the Performance of a VRV System in Cooling Mode-Part I: Experimental evaluation, HVAC&R Research, Vol.14, No.4, 615-630.
6. Aynur, T.N., Hwang, Y., Radermacher, R., 2008b. Simulation Evaluation of the Ventilation Effect on the Performance of a VRV System in Cooling Mode—Part II: Simulation Evaluation. HVAC&R Research, Vol. 14, No. 5, 783-795.
7. Aynur, T.N., 2010. Variable Refrigerant Flow Systems: A review, Energy and Buildings 42: 1106-1112.
8. Jain, N., Koelna, J., Sundaramb, S., Alleynea, A., 2014. Partially Decentralized Control of Large-scale Variable-Refrigerant-Flowsystems in Buildings, Journal of Process Control 24, 798-819.
9. Kwon, L., Lee, H., Hwang, Y., Radermacher, R., Kim, B., 2014. Experimental Investigation of Multifunctional VRF System in Heating and Shoulder Seasons, Applied Thermal Engineering 66, 355-364.
10. Aynur, T.N., Hwang, Y., Radermacher, R., 2010a. Integration of Variable Refrigerant Flow and Heat Pump Desiccant Systems for the Cooling Season, Applied Thermal Engineering 30, 917-927.
11. Aynur, T.N., Hwang, Y., Radermacher, R., (2010b). Integration of Variable Refrigerant Flow and Heat Pump Desiccant Systems for the Heating Season, Energy and Buildings 42, 468-476.
12. Zhu, Y., Jin, X., Fang, X., Du, Z., 2014. Optimal Control of Combined Air Conditioning System with Variable Refrigerant Flow and Variable Air Volume for Energy Saving, International Journal of Refrigeration 42, 14-25.
13. Aynur, T.N., Hwang, Y., Radermacher, R., 2009. Simulation Comparison of VAV and VRF Air Conditioning Systems in an Existing Building for the Cooling Season, Energy and Buildings 41: 1143-1150.
14. Liu, X., Hong, T., 2010. Comparison of Energy Efficiency between Variable Refrigerant Flow Systems and Ground Source Heat Pump Systems, Energy and Buildings 42, 584-589.
15. Kwon, L., Hwanga, Y., Radermachera, R., Kimb, B., 2012. Field Performance Measurements of a VRF System with Sub-cooler in Educational Offices for the Cooling, Season. Energy and Buildings 49, 300-305.
16. Hepbasli, A., 2008. A Key Review on Exergetic Analysis and Assessment of Renewable Energy Resources for a Sustainable Future, Renewable and Sustainable Energy Reviews 12, 593-661.
17. Hürdoğan, E., 2016. Thermodynamic Analysis of a Diesel Engine Fuelled with Diesel and Peanut Biodiesel, Environmental Progress & Sustainable Energy, 35 (3), 891-897.
18. Peng, S., Hong, H., 2015. Exergy Analysis Of Solar Gas Turbine System Coupled With Kalina Cycle, Int. J. of Exergy, 18(2), 192–213.
19. Ceylan, İ., Gürel, A.E., 2015. Exergetic Analysis of a New Design Photovoltaic and Thermal (PV/T) System, Environmental Progress & Sustainable Energy, 34 (4), 1249-1253.
20. Taufiq, B.N., Masjuki, H.H., Mahlia, T.M.I., Amalina, M.A., Faizul, M.S., Saidur, R., 2007. Exergy Analysis of Evaporative Cooling for

Reducing Energy use in a Malaysian Building, Desalination 209, 238-243.

21. Sakulpipatsin, P., Van Der Kooi, H.J., Itard, L.C.M., Boelman, E.C., 2008. The Influence of Possible Definitions of a Reference Environment to Determine the Exergy of Air in Buildings, Int. J. of Exergy, 5(3), 275-295.

22. Rafique, M.M., Gandhidasan, P., Rehman, S., Alhems, L.M., 2016. Performance Analysis of a Desiccant Evaporative Cooling System under Hot and Humid Conditions, Environmental Progress & Sustainable Energy, 35 (5), 1476-1484.

23. Hürdoğan, E., Büyükalaca, O., Hepbasli, A., Yılmaz, T., 2011. Exergetic Modeling and Experimental Performance Assessment of a Novel Desiccant Cooling System for Buildings, Energy and Buildings 43, 1489-98.

24. Özbek, A., 2016. Exergy Characteristics of a Ceiling-type Residential Air Conditioning System Operating under Different Climatic Conditions, Journal of Mechanical Science and Technology 30 (11), 5247-5255.

25. Holman, J.P., 1989. Experimental Methods for Engineers, Fifth edn, McGraw Hill, NY.

26. Rosen, M.A., Dincer, I., 2003. Exergoeconomic Analysis of Power Plants Operating on Various Fuels, Appl Therm Eng, 23:643–58.