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Assessing the Effect of Thermodynamic, Hydrodynamic and Geometric of an Air Cooled Condenser on COP of Vapor Compression Cycle

Hosein Shokohmand, Mahmood Hosein Zare, and Abdorreza Qolibeik

Abstract**—**In this paper, the effects of thermodynamic, hydrodynamic and geometric of an air cooled condenser on COP of vapor compression cycle are investigated for a fixed condenser facing surface area. The system is utilized with a scroll compressor, modeled based on thermodynamic and heat transfer equations employing Matlab software. The working refrigerant is R134a whose thermodynamic properties are called from Engineering Equation Software. This simulation shows that vapor compression cycle can be designed by different configurations and COPs, economical and optimum working condition can be obtained via considering these parameters.

Keywords**—**Vapor compression cycle, air cooled condenser, COP, heat exchanger, thermal modeling.

I. Introduction

NOWADAYS, due to the energy crisis and global warming caused by pollution from fossil fuels, HVAC engineers are working on redesigning cooling and heating equipment in order to energy consumption minimization, size reduction and make them work much more efficient. Therefore performance prediction under condenser geometrical, hydrodynamic and thermodynamic parameters and effects of selecting refrigerants on HVAC cooling systems is that important. As matter of fact, the main purpose of these analyses on residential HVAC systems is to provide thermal comfort in a better way and avoid the energy waste to decrease the HVAC equipment operating cost. One of the most common HVAC systems is split air conditioner. This cycle is supposed to be an air-cooled type. It has two main parts, which are outdoor unit and indoor unit. Outdoor unit consists of 3 major parts: compressor, condenser and expansion valve. Indoor unit consists of just evaporator.

The most influence of environmental conditions on system performance is on the condenser, because it's located outside the room. Therefore by simulating the condenser performance under its condition, thermal efficiency of outdoor unit can be found very well.

Cherem-pereira et al [1] used empirical correlation for four room air conditioner in order to predict EER for each system in different climates. They tested them by experimental apparatus and showed that derived mathematical correlation are accurate enough, allowing to predict building energy consumption, electric power demand and equipment performance for a wide range of climates. Chen et al [2] modeled the compression process of a scroll compressor under different operating conditions in detail. They considered geometrical parameters of compressor, compression process, and heat transfer between refrigerant and compressor parts, internal refrigerant leakage and overall energy balance of the compressor. Thus with their modeling, influence of heat transfer, leakage, and geometrical parameters on scroll compressor performance can be investigated. Corberan et al [3] modeled thermodynamically fin plate tube condenser, working with R134a. They used condensing heat transfer and pressure drop correlations to predict performance of the condenser. They validated theoretical results by experimental test results. Their simulation shows just five percent deviations from experimental result. . Cabello et al [4] studied vapor compression cycle performance theoretically and experimentally. They used three different refrigerants: R134a, R407c, and R22. They evaluated mass flow rate, cooling capacity, compressor power consumption and COP with respect to superheating degree, evaporating and condensing temperature. They also defined their simulation accuracy by a test bench. Overall speaking, all the above researchers studied thermodynamic or geometrical parameters separately, and evaluate COP. Most of them found a correlation for designing the cycle but no one consider condenser geometrical, thermodynamic and hydrodynamic parameters together, to investigate COP with respect to it. Also they didn't evaluate overall COP in a cooling season. It is important to evaluate overall COP in a cooling season, because this parameter shows how much energy the device required in a cooling period of year. The higher overall COP leads to the less energy consumption in a cooling season.

In this paper, the effects of degrees of subcooled, number of rows and fins per inch on the COP and overall COP of outdoor unit are investigated for a fixed condenser facing surface area. Fixed facing surface area condenser is due to the space

Hosein Shokohmand is with the department of Mechanical Engineering, Faculty of Engineering, University of Tehran, North Karegar (Amirabad) St., Tehran 11365-4562, Iran (hshokoh@ut.ac.ir).
Mahmood Hosein Zare is with the department of Mechanical Engineering, Faculty of Engineering, University of Tehran, North Karegar (Amirabad) St., Tehran 11365-4562, Iran (mhzare@ut.ac.ir).
Abdorreza Qolibeik is with the department of Energy engineering Science and Research branch, Islamic Azad University, Tehran, Iran. (a.qolibeik@srbiau.ac.ir).

constraint occupied by the residential systems. The condenser is modeled with thermodynamic and heat transfer equations, employing Matlab software to simulate. R134a is the working refrigerant whose thermodynamic properties are called from Engineering Equation Software. To simulate the cycle and compare the COPs of different configuration, a base condenser is considered. Table.1 shows condenser geometrical parameters of the base model. This simulation shows that split air conditioner can be designed with different configuration and varied COPs.

II. System and Model Descriptions

A. System Description

Split residential air conditioning system is based on the vapor compression cycle. Fig 1 shows the schematic of this cycle. The vapor compression cycle uses a circulating liquid refrigerant as the medium which absorbs and removes heat from the space to be cooled and subsequently rejects that heat

TABLE I
CONDENSER BASE MODEL GEOMETRICAL PARAMETERS

Tube spacing	1.25in.×1in.
Tube thickness	0.4 mm
Tube outer diameter	9.52 mm
Height	2.5 ft.
Finned width	4.5 ft.
Fin pitch	12
Number of rows	3
Number of circuits	12

to the ambient. as the analysis of this cycle is complicated, it is suggested the cycle to be divided in different sections and modeled as following . Outlet superheat gas of evaporator enters compressor and compressed up to discharge pressure. After heat exchange with the air passing over the condenser tubes, the gas condensed and become subcooled liquid refrigerant. Then it goes through the expansion valve and expands to a lower pressure [5].

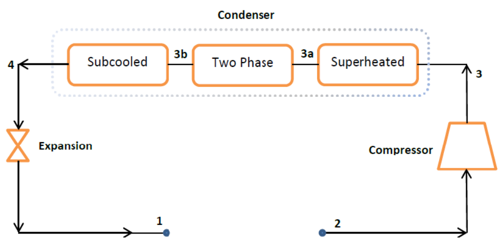

Fig. 1 Outdoor components

B. Model Descriptions

The cycle is divided to the following components.

1. Compressor

This system utilizes a positive displacement compressor (scroll compressor). The refrigerant is compressed when one spiral orbits around a second stationary spiral which creates smaller and smaller volume and higher pressures. By the time the refrigerant is discharged, it is fully pressurized to discharge pressure. Power consumed by scroll compressor is

$$w_{comp} = \dot{m}_r \frac{h_3 - h_2}{\eta_c} \tag{1}$$

Klein and Rindle [6] had correlated thermal efficiency for scroll compressors with respect to critical temperature and pressure:

$$\eta_c = -60.25 - 3.814P_r - 0.281P_r^2 + 111.3T_r - 50.31T_r^2 + 3.061P_rT_r \tag{2}$$

2. Condenser

The condenser is a heat exchanger which exchange heat from the refrigerant to the ambient air. There are a lot of air cooled heat exchangers configurations to employ in residential air conditioning system but plain fin plate tube type heat exchanger considered here.

It is important to have subcooled at the outlet. This is due to allowing the expansion valve to work properly. Specific heat rejected from the condenser is expressed as

$$q_{condenser} = h_3 - h_4 \tag{3}$$

$\varepsilon -$ NTU Method is employed to calculate the performance of the condenser.

$$Q = \varepsilon C_{min}(T_{h,i} - T_{c,i}) \tag{4}$$

For a cross flow heat exchanger with both fluids unmixed, the effectiveness depends on NTU and heat capacity ratio and expresses as [7]:

$$\varepsilon = 1 - \exp\left\{\left(\frac{1}{C_r}\right) NTU^{0.22}\left[\exp\left(-C_r\left(NTU^{0.78}\right)\right) - 1\right]\right\} \tag{5}$$

By neglecting wall thermal resistance and Sedimentation on both sides, overall heat transfer coefficient can be calculated by equation (6) where η_f is air side fin efficiency calculated from the procedure introduced by Schmidt [8].

$$\frac{1}{UA} = \frac{1}{\eta_a \bar{h}_a A_a} + \frac{1}{\bar{h}_r A_r} \tag{6}$$

Heat transfer coefficient on refrigerant side is dependent on each section. In single phase sections of condenser which are subcooled and superheated, it can be calculated by Kays&London correlation as

$$StPr^{2/3} = a_{st}Re^{b_{st}} \tag{7}$$

where a_{st} and b_{st} are coefficients, can be found at [9]. Heat transfer coefficient in two phase section depends on two phase flow pattern, but Shah [10] developed a correlation which could be used for all flow patterns. By assuming that quality varies linearly by length and integrating Shah Correlation over the length, the mean two phase heat transfer coefficient will be determined.

$$h_{tpm} = h_L \left(0.55 + \frac{2.09}{P_r^{0.38}}\right) \quad (8)$$

Pressure drop depends on both type of tubes and flow regimes. For single phase flow and straight tubes it expresses as

$$\Delta P_{s,sp} = \frac{fG^2L}{\rho} \quad (9)$$

For straight tube and two phase flow, Hiller and Glickman method [11] is employed to determine pressure drop. In this study, horizontal heat exchanger is used. Thus pressure drop only divided into frictional and momentum component as follows

$$\frac{dP}{dz} = \left(\frac{dP}{dz}\right)_f + \left(\frac{dP}{dz}\right)_m \quad (10)$$

$$\left(\frac{dP}{dz}\right)_m = -\frac{G^2}{g_c\rho_c}\frac{dx}{dz}\left(2x + (1-2x)\left(\frac{\rho_v}{\rho_l}\right)^{\frac{1}{3}} + (1-2x)\left(\frac{\rho_v}{\rho_l}\right)^{\frac{2}{3}} - 2(1-2x)\left(\frac{\rho_v}{\rho_l}\right)\right) \quad (11)$$

$$\left(\frac{dP}{dz}\right)_f = -0.09\frac{\frac{G_V^2}{\rho_V}}{g_cD}\left(\frac{\mu_V}{G_VD}\right)^{0.2}(1 + 2.85\chi_{tt}^{0.523}) \quad (12)$$

Quality is assumed to have linear relation with length and varies from 1 to 0. Therefore by integrating equations (10), (11) and (12), two phase flow pressure drop of two phases in straight tube is calculated. In two phase flow at U bend tube, pressure drop can be found by Chisholm method [12]. Due to this method, pressure drop in a bend is the product of the bend pressure drop for liquid only and the two phase multiplier.

$$\Delta P_{b,tp} = \Delta P_{b,l}\varphi_{b,l}^2 \quad (13)$$

$$\varphi_{b,l}^2 = 1 + (\Gamma_b^2 - 1)\left(B_{180^\circ}x^{\frac{2-n}{2}}(1-x)^{\frac{2-n}{2}} + x^{\frac{2-n}{2}}\right) \quad (14)$$

In equation (14), Γ_b *and* B_{180° are the coefficients, can be found in [12].To evaluate convective heat transfer coefficient, McQuiston method [13] for dry coils is employed. By using the following correlation, j factor is calculated for heat exchangers having 4 rows or less.

$$\frac{j_z}{j_4} = \frac{1-1280zRe_{rs}^{-1.2}}{1-1280\times4\times Re_{rs}^{-1.2}} \quad (15)$$

$$Re_{rs} = \frac{GX_{rs}}{\mu} \quad (16)$$

In equation (17), X_{rs} is row spacing and z is the number of rows. Air side pressure drop consists of two components: pressure drop due to tubes and pressure drop due to fins. Pressure drop due fins has been calculated with Rich equation which can be expressed as below:

$$\Delta P_{fin} = f_{fin}v_m\frac{G^2}{2}\frac{A_{fin}}{A_c} \quad (17)$$

For evaluating pressure drop due to tubes, Zukauskas method [14] is used. It can be expressed as

$$\Delta P_{tubes} = Eu\frac{G^2}{2\rho}z \quad (18)$$

C. Fan Power

Natural convection is not sufficient to reject enough heat from refrigerant to the air, thus a fan is employed to flow the air over condenser. Only the work which is done by fan is important in this study. It is calculated by equation (20).

$$W_{fan} = \frac{V_a\Delta P_{air\ side}A_{face}}{\eta_{fan}} \quad (19)$$

D. Expansion valve

Expansion valve is used to expand the refrigerant to a lower pressure and temperature. Due to adiabatic process occurred in expansion valve and no work, its governing equation is

$$h_4 = h_1 \quad (20)$$

E. Overall COP

The overall COP is calculated as

$$COP_{overall} = \frac{Q_{overall}}{W_{overall}} = \frac{\sum_{i=1}^{8}(T_i - 100)fr_i}{\sum_{i=1}^{8}\frac{(T_i-100)fr_i}{COP_i}} \quad (21)$$

fr_i is fraction of Total Temperature Bin Hours, introduced by ARI [15].

III. Results and Discussion

Here effects of the velocity of air over condenser, subcooled temperature, and fins per inch on COP and overall COP are investigated.

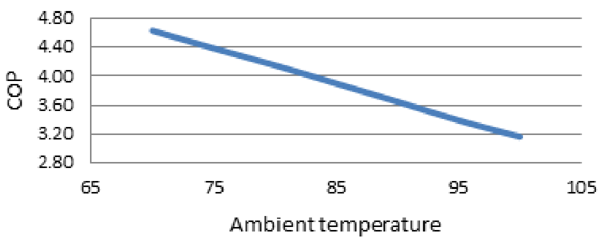

Fig. 2 COP versus ambient temperature for 15 F degrees of subcooled

Fig. 2 shows how COP varies with ambient temperature for 15 °F degrees of subcooled and 8.5 fps air velocity over condenser. As the ambient temperature goes up, the saturation pressure in the condenser increases. Thus compressor consumes much more energy to provide required discharge pressure. Therefore COP decreases with respect to increasing compressor power consumption.

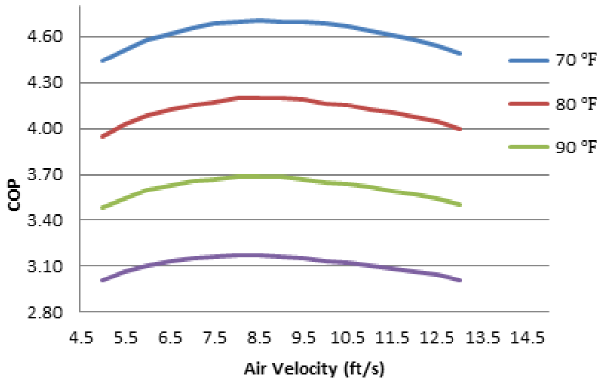

Fig. 3 Effect of Air Velocity on the COP for varying degrees of Subcooled

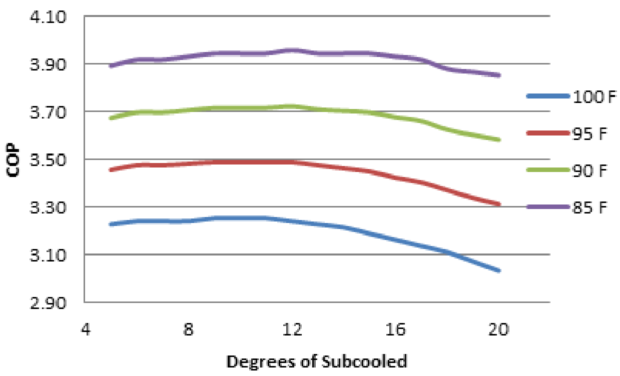

Fig. 4 effect of degrees of subcooled on COP for varying condenser fins per inch

Fig. 3 shows the effect of air velocity over base system on the COP for different ambient temperatures. As it shows, COP increases with increasing air velocity magnitude but it reaches a peak value and then decreases with increasing air velocity magnitude. This trend is because of invers effect of heat transfer rate and fan consumption on COP. This figure also shows that as the ambient temperature decreases, the peak COP increases. Fig. 4 shows how the COP varies with the ambient temperature for various degrees of subcooled. The less the ambient temperature leads to the less the saturation pressure in the condenser. Thus compressor needs to work less and COP increases. By increasing degrees of subcooled, cooling capacity of evaporator increases and as a result COP increases.

IV. Conclusion

In this study, the effects of degrees of subcooled, ambient temperature and air velocity on the COP of vapor compression cycle are investigated for a fixed condenser facing surface area. The system is utilized with the scroll compressor and R134a is considered as the working refrigerant. The simulation is employed for evaluating the performance of that cycle in Tehran climate. For the specified temperature between 65 ° F and 100 ° F, it is found that the performance of the cycle is highly dependent to the air velocity used for cooling the condenser. It is shown that the COP is the function of the velocity and temperature of the air over the condenser as well as degrees of subcooled. This simulation is proved to be a promising tool for design purposes by means of arranging practical figures.

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