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Research Article

Design, Optimization and Performance Analysis of Condenser for HVAC Automobile System for R-290.

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Abstract

In automobile air conditioning system condenser plays a vital role. In this paper theoretical and experimental study of acondenser for automobile air conditioning is carried out with refrigerant propane (R290). In automobile air conditioning R134a is widely usedrefrigerant whichis not environmental friendly because of its higher GWP value hence there is need of to replace R134a with the environmentally friendly refrigerant like hydrocarbon (i.e. R290). The optimization of condenser is done by varying different geometric parameters and the performance of the condenser is studied experimentally by carried out the experimentation. In optimization of condenser the optimum value for the condenser is obtained is 5.2 outer diameter, 2 rows and 12 fin per inch and in experimental analysis at 6 m/s -7 m/s air velocity gives optimum result are obtained.

Keywords:*Alternative refrigerants, Condenser, Optimization,R290*

1. Introduction

The heating, ventilation and air conditioning of the automotive system is designed to provide comfort to driver and passengers. One of the important things to be fulfilled is comfort. Human comfortis the state of mind that expresses satisfaction with the surrounding environment according to American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE).The main function of an air conditioning control system is to modulate the A/C system capacity to match the design condition and climate change, in order to maintain the indoor environment within desirable limits at optimum energy use levels during the entire drive duration. One of the important part in air conditioning system is condenser. The condenser is a heat exchanger located at front of the vehicle and receives high-pressure, hot refrigerant from the compressor. In air conditioning system refrigerant flows through the condenser and cools off from the wind when driving at highway speeds or air blowing from electric cooling fan. A condenser in which desuperheating of high temperature vapor changes the phase from vapour to liquid and sub cooling of condensate occurs. The condenser is an important device and the main function of condenser is to remove heat of hot vapor refrigerant discharged from the compressor. The hot vapor refrigerant contains the heat absorbed by the evaporator and the heat of compression due to the mechanical energy of the compressor motor. The heat from the hot vapor refrigerant in a condenser is removed by transmitting it to the walls of the condenser tubes and then from the tubes to the cooling medium. The cooling medium is generally air or water.

J.B. Copettiet. al. have designed and optimized the minichannel parallel flow condenser with refrigerant R-134a for automobile air conditioning system. Optimization was done through variation of geometric parameter like refrigerant side pass arrangement, number of channels, height and width of the tube.

Viveksahuet. al. experimentally studied the wire on tube condenser with different spacing of wire. The optimization was done by varying operating parameter like condenser pressure, condenser temperature.

In present study, optimization of condenser carried out by varying different geometrical parameter and experimental analysis done by varying air velocity through condenser fan.

2. Propane

Propane (R290) is an eco-friendlynatural refrigerant.Which has very low Global Warming Potential and zero Ozone Depletion Potential and has no direct impact on the greenhouse effect. It is a Hydrocarbon refrigerant. Propane has excellent thermal performance, low price, and R290 can be compatible with general machinery lubricants and structural material.

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2.1Thermophysical Properties

Molecular weight of weight of R290 is less as compared to other refrigerant. R290 has the slightly highest critical temperature; the highest is the latent heat of vaporization that normally means higher efficiency. Lower saturation pressure allows ease in manufacturing of refrigeration system.

Table 1 Thermophysical property

Refrigerant	R290
Molecular Weight	44.10
Normal B.P. ^{0}C	-42.2
Critical Temp. Tc^{0}C	96.7
Critical Press MPa	4.25
Latent heat of Evap. kJ/kg	425.4

2.2 Environmental characteristics

Atmospheric life of R-290 is very less as compared to other which means it will sustain in environment for very less time and hence it is environment friendly. HC-290 is a non-ozone depleting refrigerant whereas most of the refrigerant is a non-zero ODP refrigerant. GWP value of HC-290 is 20, which is very low compared to other.

Table 2 Environmental characteristics

Refrigerant	R290
Atm. life (Years)	0.041
ODP(R11=1)	0.000
GWP$_{100yr(CO2=1)}$	20

2.3 Safety characteristics

For the development of HC-290, flammability and toxicity are very important parameters due which it was neglected alternative for so many years. R290 is classified under A3 safety class as per ASHRE34-2010 due to this it has been avoided. But it has good thermo physical properties which is similar to R22, hence can be effectively used with taking proper care of leakage factor during operation. The European standard EN378 gives the safety requirements for the use of flammable refrigerants in various applications. Toxicity safe index for R22 and R290 are similar.

Table 3 Safety characteristics

Refrigerant	R290
LFL by mass kg/m^3	0.075
LFL by volume %	2.1
Burning velocity cm/s	46
Combustion heat MJ/kg	50.3
Toxicity ppm	1000
Safety class	A3

When considering a mixture of a flammable fluid and air, burning or explosion can occur upon contact with a source of ignition in the case that the mixture concentration is between the lower explosive limit (LEL) and the upper explosive limit (UEL) and Stored in a cool, ventilated coffers away from fire and the heat source. Propane sensors have been installed nearby the compressor and inside electrical panels. These sensors detect from 0 - 20% of the LEL and different safety levels have been considered depending on the concentration measured by the sensors.

3. Design of condenser

Design of condenser is done by using various correlations to calculate the heat transfer coefficient from air side and refrigerant side. Heat transfer of finned tube air cooled condenser is given by

$$Q = U_t A_t LMTD \quad (1)$$

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (2)$$

$$U_t = \frac{1}{\frac{1}{h_i}\frac{A_t}{A_i} + \frac{1}{h_f} + \frac{A_t}{A_i}\frac{r_i \ln(d_o/d_i)}{K_w} + \frac{1}{h_o \eta_o}} \quad (3)$$

3.1 Air side heat transfer

Air side heat transfer coefficient calculated using wang and chi correlation (2000).

$$j = \begin{cases} 0.108\,\mathrm{Re}_{dc}^{-0.29}\left(\frac{X_t}{X_l}\right)^{c_1}\left(\frac{p_f}{d_c}\right)^{-1.084}\left(\frac{p_f}{D_h}\right)^{-0.786}\left(\frac{p_f}{X_t}\right)^{c_2} & for\ N_r = 1 \\ 0.086\,\mathrm{Re}_{dc}^{c_3}.N_r^{c_4}\left(\frac{p_f}{d_c}\right)^{c_5}\left(\frac{p_f}{D_h}\right)^{c_6}\left(\frac{p_f}{X_t}\right)^{-0.93} & for\ N_r \geq 2 \end{cases} \quad (4)$$

Where

$$c_1 = 1.9 - 0.23\ln RE_{dc}$$

$$c_2 = -0.236 + 0.126\ln \mathrm{Re}_{dc}$$

$$c_3 = -0.361 - \frac{0.042N_r}{\ln \mathrm{Re}_{dc}} + 0.158\ln\left[N_r\left(\frac{p_f}{d_c}\right)^{0.41}\right]$$

$$c_4 = -1.224 - \frac{0.076(X_l/D_h)^{1.42}}{\ln \mathrm{Re}_{dc}}$$

$$c_5 = -0.083 + \frac{0.058N_r}{\ln \mathrm{Re}_{dc}}$$

$$c_6 = -5.735 + 1.21\ln\frac{\mathrm{Re}_{dc}}{N_r}$$

Outside heat transfer coefficient is given as

$$h = jG_{\max}C_p(P_r)^{-2/3} \quad (5)$$

3.2 Fin efficiency

To determine the overall surface efficiency for a finned tube heat exchanger, it is necessary to determine the efficiency of the fins.Fin efficiency is calculated using Schmidt (1945) correlation the empirical relation for the equivalent radius is given by

$$\frac{R_e}{r} = 1.27\psi(\beta - 0.3)^{1/2} \quad (6)$$

The coefficients ψ and β are defined as

$$\psi = \frac{X_t}{2r} \text{ and } \beta = \frac{1}{X_t}\left(X_l^2 + \frac{X_t^2}{4}\right)^{1/2}$$

The fin efficiency can be expressed as

$$\eta_f = \frac{\tanh(m.l)}{m.l} \quad (7)$$

where, $l = R_e - r$

The total surface efficiency of the fin, η_o is therefore expressed as

$$\eta_o = 1 - \frac{A_f}{A_t}(1 - \eta_f) \quad (8)$$

3.3Refrigerant side heat transfer

The refrigerant heat transfer is calculated using the Dittu-Boelter correlation which is valid for fully developed flow in circular tubes with moderate temperature variations (Incropera& DeWitt, 1996).

$$Nu_D = 0.023\text{Re}_D^{0.8}\text{Pr}^{0.3} \quad (9)$$

The two-phase flow heat transfer model developed by Shah is a simple correlation which is given as

$$\bar{h}_{TP} = \bar{h}_L\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{p_r^{0.38}}\right] \quad (10)$$

For complete condensation, the mean two-phase heat transfer coefficient reduces to the following expression,

$$\bar{h}_{TPM} = \bar{h}_L\left(0.55 + \frac{2.09}{p_r^{0.38}}\right) \quad (11)$$

4. Optimization

Optimization is done by varying the different geometric parameter like outer diameter of the condenser tube, fin per inch and number of rows of condenser and its effect on the coefficient of performance is graphically plotted below by keeping frontal area of condenser constant.

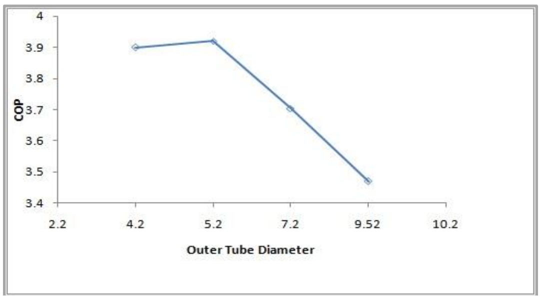

Fig.1 Effect of outer diameter on COP

After optimization it is observed that at 5.2 mm outer diameter, 12 fins per inch and 2 rows condense gives better performance.

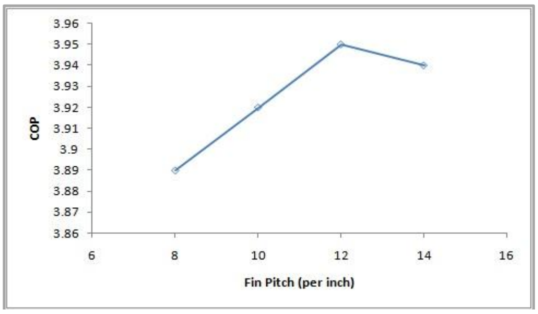

Fig.2 Effect of fin pitch on COP

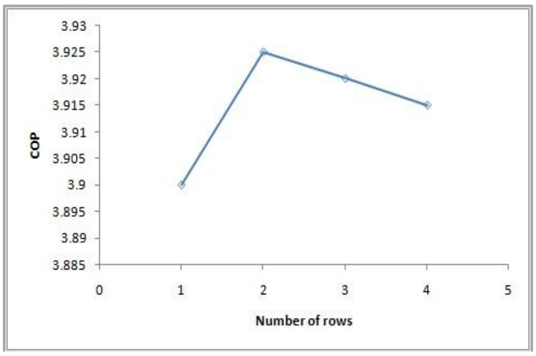

Fig.3 Effect of Number of rows on COP

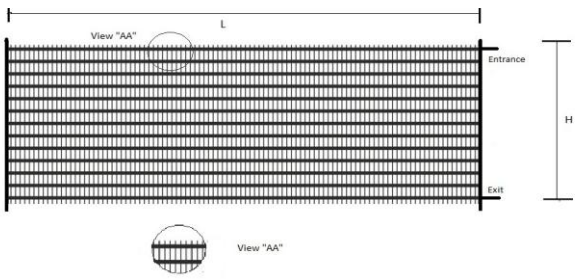

Fig.4 Condenser

Table 4 Specification of Condenser

Tube material	Aluminum
Outer diameter of the tubes D_o	5.2 mm
Number of tubes N_t	13
Number of rows (z)	2
Fin pitch	1 mm
Thickness of the fins δ	0.20mm

5. Experimental Setup

In experiment the performance analysis is done by varying the Air velocity (m/s) by condenser fan. At differentAir velocity different temperature are calculate like temperature at compressor inlet and outlet,temperature after condenser and also temperature after evaporator.

Fig.5 Experimental setup

6. Result and discussion

From above study of design optimization of condenser it concluded that at diameter 5.2 mm 2 rows and 12 fins per inch gives better result as compared to other varying geometric parameter by keeping frontal area of condenser constant. As the diameter, Number of rows and fin per inch increases there will be drop in COP.

In experimentally analysis COP and Refrigerating effect is calculated and it is observed that as the air velocity increases the COP is going to increases at Air velocity 7 m\s gives better coefficient of performance.

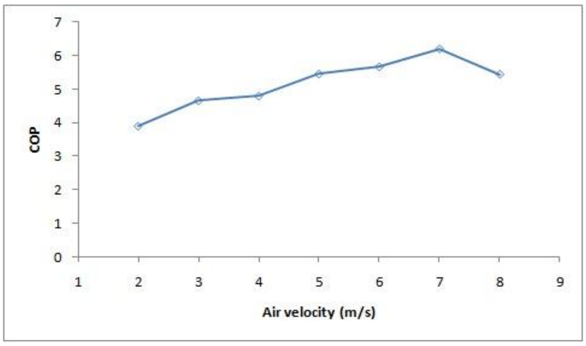

Fig.6 COP vs. Air velocity

Also in Experimental analysis it is observed that as Air velocity increased it directly affect the refrigerant effect. As Air velocity increases Refrigerant effect also increases at Air velocity 6 m\s gives optimum refrigerating effect.

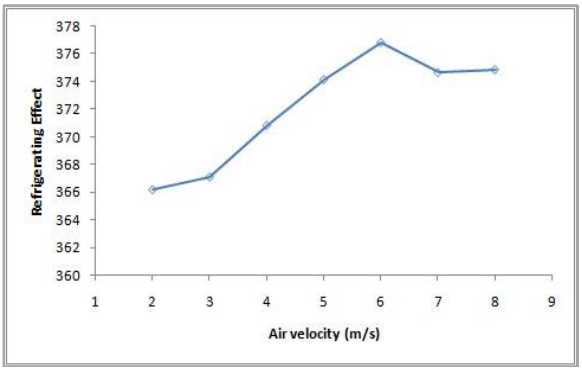

Fig.7 R.E vs. Air velocity

Conclusions

In optimization it is concluded that by keeping the frontal area of condenser constant and varying geometrical parameter it is observed that as the outer diameter increases the COP of the system is decreases. Also from varying the number of rows and fin per inch it is concluded that at 12 fins per inch gives better performance and at 2 number of rows gives better performance.Experimentally analysis it is concluded air flow velocity (condenser fan speed) also contribute effect part in improving the performance of system from experiment it is that observed that COP and R.E of the system increases with Air velocity increased.

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