Analysis of Air-Water Cooled Condenser in Vapour Compression System

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***Abstract*—The development and performance enhancement of heat exchanger is major concern of emerging nascent technology. Performance enhancement contributes to minimize the operating cost and energy consumption. In this paper theoretical analysis of air-water cooled condenser is presented and a 3D (three dimensional) model of air-water cooled condenser has been made in ANSYS. The tubes in the model are half dipped in water and the other half exposedtoair. Copper tubes were used as they are the most thermally conductive common metal and aluminium fins were used because they are lightweight and flexible. The validation of theoretical values is carried out by the Computational Fluid Dynamics (CFD) software ANSYS.The use of such condensers might prove to be more beneficial for residential purposes.**

Keywords:- Heat transfer, CFD,Viscous Laminar models, Simulation, ANSYS

I. INTRODUCTION

With the rapid growth of industrialisation and clearing of land, the average temperature of the environment has been increased within the last two decades. The reason for this gradual/exponential increase in temperature is the greenhouse effect and global warming. As the earth's surface temperature rises and to achieve thermal comfort the use of air conditioning has also been increased which in turn has increased power consumption. The researchers of this domain have continuously tried to enhance the performance of air conditioners so that they become more efficient and consume less power.PrachaYeunyongkul, Passawat Watcharadumrongsak, Sampan Rittidech [1] tried to increase the efficiency of ACC by cooling ACC coils with the use of a hybrid condenser. K. Sumeru, C Sunardi, M.F.Sukri [2] in their work used condensate water to provide subcooling to the compressor discharge for split air conditioning system and reported about the augmentation in COP. Hua Chen, W.L.Lee, F.W.H.Yik [3] experimentally show about 17.4% increment in COP of split air conditioning system while using water cooled condenser relative to the air cooled condenser. A. Sirichaoroenpanich, S.Wiriyasart, R.Prurapark, P.Naphon [4] boosted the thermal performance of an air conditioning system using a cooling water loop with the concentric helical coiled tube heat exchanger between the compressor unit and condenser unit for cooling the refrigerant whereas S.S.Hu, B.J.Huang [5] employed residential water-cooled air conditioners with a cellulose pad as filling material for cooling tower and observed that the COP of this kind of air conditioner is higher than the conventional air conditioner. W.L.Lee, Hua Chen, F.W.H.Yik [6] developed an empirical model for predicting the operational performance and consumption of energy for the use of water-cooled air conditioners. Jingyuan Xu, Jianying Hu, Limin Zhang, Ercang Luo [7] simulated novel shell tube and conventional shell tube heat exchanger to achieve better heat transfer power. E. Hajidavalloo [8] reported about the decrement in the power consumption by installing cooling pads on both sides of the air conditioner.Xiaojing Zhu et al [9] in their work found that the wet-bulb temperature has a huge effect on the heat transfer performance than the relative humidity for an evaporative condenser.Huanwei Liu, Qiushu Zhou, Yuling Liu, Peifeng Wang, Defa Wang [10] in their study of evaporative condenser reported about the increment in COP with the air velocity and spray rate whereas the decrement in COP with the increaseininlet air dry bulb temperature and frequency of compressor. M.R.Islam, K.A.Jahangeer, K.J.Chua [11]experimentally reported about the 28% hike in the COP of an evaporatively cooled air conditioning unit than the traditional air conditioning unit. M.Fiorentino, G.Starace [12] worked upon increasing heat transfer of evaporative condensers by applying a hybrid method on a countercurrent evaporative condenser. Zhiwei Huang, Yunho Hwang, Reinhard Radermacher [13] have proposed a new design to apply biomimicry to heat exchangers which increases heat transfer coefficients per unit area as well as reduce water consumption and pumping power. Manpreet Singh, Babool Rai, Vikul Vasudev [14] installed evaporative pads in vapour absorption and vapour compression refrigeration system and compared the COP of air-cooled and water-cooled condensers and concluded that WCC is much more efficient. M. Hosoz, A.Kilicarslan [15] have performed a comparison of all three condensers including evaporative, air-cooled and water-cooled and concluded that water-cooled condenser has a better coefficient of performance than the evaporative condenser and evaporative condenser have a better coefficient of performance than the air-cooled condenser. G.P. Maheshwari, A.A Mulla Ali [16] have investigated both air-cooled and water-cooled based on peak power and energy consumption. K. Harby, Doaa.R.Gebaly, Nader.S.Koura, Mohamed.S.Hassan [17] reported that the evaporative condensers are much more effectivethan traditional condenser for hot region. Zongwei Han et al [18] tried to reduce the power consumption by maximizing the utilization of natural cold energy using an evaporative condenser with water cooling condensation. Chnag Yong Park, Pega Hrnjak [19], numerically and experimentally investigated the performance of microchannel condenser and conclude that microchannel

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condenser are much more efficient than round tube condenser.P Dalai, P.Nanda, C.Mund, D.Mishra, Abhijeet Gupta [20] using the concept of humidification-dehumidification focused on the utilization of condensed water for drinking and other purposes. H.N.Patel, Y.R.Patel, D.Patel, J.Patel, S.K.Kulkarni [21] optimized performance coefficient of air-cooled condenser by spraying water on the fins of the condensing unit at regular intervals. H.Yang et al [22] reduced the energy consumption around 2.37%-13.53% with the combination of a spray evaporative cooling system and an air cooled chiller for a refrigeration unit. J.Wei, J.Liu, X.Xu, J.Ruan, and G.Li [23] using vertical tube evaporative condenser instead of conventional finned tube type air cooled condenser increased the coefficient of performance by 30%. Mulla Irfan Ahmad, Jameel Basha S.M, Viswanatha Chari V, Srinivas Kumar G [24] have used a combination of R290 and R600a as a refrigerant which in turn is less toxic as well as ozone friendly and provides as an alternative. T Hussain, AK Singh, A.Mittal, A Verma, Z Alam [25] has increased the coefficient of performance by employing evaporative cooler using cellulose and steel wire pads.

From the literature reviewed above, it has been observed that a lot of work has been performed to increase the COP of an air conditioning unit by proposing the different design of water cooled, air-cooled and evaporative condensers or by installing cooling pads and by the usage of microchannel condenser. In contrast to the above, the present literature proposed a novel concept of water cum air cooled condenser for the split air conditioning system.

II. THEORETICAL ANALYSIS:

A. *Air Side*

Assumptions

The assumptions which are to be considered in thepresent analysis are as follows:

- Hydrodynamically and thermally fully developed flow
- Tube wall remains at constant temperature
- Steady State with no heat generation
- High fins with In-line array

Geometrical Parameters

External diameter of condenser tube at fin root (D_r) = 22.7 mm

External diameter of fin(D_t) = 32.7 mm

Height of fin (L) = 5.0 mm

Thickness of fin (w) = 0.5 mm

Fins spacing(s) = 5.0 mm

Total length of air side condenser tube(L_t) = 1.5 m

Number of air side tubes in a row (n_a) = 3

Total number of air side tubes (N_a) = 3

Pitch of tubes in plane perpendicular to flow (P_1) = 50.0 mm

Boundary Conditions

Mass flow rate of air ($M^{\cdot}$) = 0.1687 kg/s

Density of air (ρ) = 1.2170 kg/m^3

$kinematic\ viscosity\ of\ air(v)$ = 15.9*10^{-6} m^2/s

Prandtl number(P_r) = 0.705

Thermal conductivity of air (K_{air})= 26.9*10^{-3} W/mK

Specific heatcapacity of air (C_{pa}) = 1007 J/kg.K

Refrigerant inlet temperature (T_{rinlet}) = 335 K

Air free stream temperature ($T_{airinlet}$) = 313 K

Governing Equations

Surface area of the fins (A_f) is given by,

$$A_f = \frac{N*L_t*\pi}{(s+w)}\left[\frac{1}{2}\left(D_t^{\ 2} - D_r^{\ 2}\right) + D_t w\right] \quad (1)$$

Surface area of tubes between fins(A_w) is given by,

$$A_w = \frac{N*L_t*\pi}{(s+w)}[D_r s] \quad (2)$$

Total surface area of air side condenser (A_a),

$$A_a = A_f + A_w \quad (3)$$

Min cross sectional flow area (S_{min}) is given by,

$$S_{min} = n_a * L_t\left[P_1 - D_r - \frac{2wL}{w+s}\right] \quad (4)$$

Total tube surface area (without fins) A_t is given by,

$$A_t = N*L_t*\pi*D_r \quad (5)$$

Reynolds Number(R_e) is gven by,

$$V_{max} = \frac{M^{\cdot}}{\rho*S_{min}} \quad (6)$$

$$R_e = \frac{V_{max} \times D_r}{v} \quad (7)$$

where,

v is the kinematic viscosity of air .

$V_{max}$is maximum velocity

The Nusselt number ($\bar{N}_u$) is given by

$$\bar{N}_u = 0.3(P_r)^{0.333}[\![(R]\!]_e)^{0.625}\left(\frac{A_a}{A_t}\right)^{-0.375} \quad (8)$$

Average heat transfer coefficient (h_{air}) for air is calculated using the formula below,

$$h_{air} = \frac{\bar{N}_u K_{air}}{D_r} \quad (9)$$

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The efficiency (η) of the radial fin can be determined by modifying the efficiency of the longitudinal fin otherwise it can be calculated by graph.

$$\eta_{fins} = \frac{\tanh m\psi}{m\psi} \quad (10)$$

Where,

$$\psi = \frac{D_r}{2}\left(\frac{D_t}{D_r} - \frac{1}{1}\right)\left(1 + 0.35 \ln \frac{D_t}{D_r}\right) \quad (11)$$

Hence the effective average heat transfer coefficient ($\hbar$) is given by,

$$\hbar = \left[\frac{\eta_{fins * Af + Aw}}{A_a}\right] * h_{air} \quad (12)$$

Log mean temperature difference experienced by air (ΔT) is given by,

$$\Delta T = (T_{rinlet} - T_{airinlet})\left[\frac{1-e^{-A_a*\hbar/Cpa_{*M\cdot}}}{-A_a*\hbar/Cpa*M\cdot}\right] \quad (13)$$

Rate of heat rejected to air side ($\dot{Q}_{air\ side}$) is calculated by,

$$\dot{Q}_{air\ side} = \hbar * A_a * \Delta T \quad (14)$$

B. WATER SIDE :

Assumptions

The assumptions which are to be considered in thepresent analysis are as follows:

- Counter flow between hot fluid and cold fluid.
- Turbulance is present in side the shell.
- Ignore sensible heat removal by the condenser.

Geometrical Parameters

Length of water box = 588 mm

Total length of water side condenser tube(L_w) = 1.5 m

Total number of water side tubes (N_w) = 3

Boundary Conditions

Mass flow rate of refrirerant ($\dot{m}_r$) = 0.047 kg/s

Mass flow rate of water($\dot{m}_w$) = 0.1 kg/s

Saturation temperature of the refrigerant(T_{sat}) = 335 K

Inlet temperature of the water (T_{win}) = 285 K

Specific heatcapacity of water (C_{pw}) = 4184 J/kg.K

Specific heatcapacity of refrigerant (C_{pr}) =1560 J/kg.K

Latent heat of condensation of refrigerant(h_{fg}) = 161 KJ/kg.K

Condensation pressure= 25 bar

Total Surface area of water side tube (A) = 0.106917 m^2

Governing Equations

Heat transfer effectiveness of heat exchanger is:

$$\mathcal{E} = \frac{\dot{Q}_{actual}}{\dot{Q}_{max}} \quad (15)$$

Where, $\dot{Q}_{actual}$ = actual heat transfer rate between refrigerant and cooling water

$\dot{Q}_{max}$ = Thermodynamically limited maximum possible heat transfer rate.

$$\dot{Q}_{max} = (\dot{m}C_p)_w\,(T_{sat} - T_{win}) \quad (16)$$

$$\dot{Q}_{actual} = \dot{m}_r * h_{fg} - \dot{Q}_{air\ side} \quad (17)$$

NTU is number of transfer unit for the heat exchanger which phase change occurs

$$\text{NTU} = -\ln(1 - \mathcal{E}) \quad (18)$$

Over all heat transfer coefficient (U) is given by,

$$U = \frac{(\dot{m}C_p)_w}{A} NTU \quad (19)$$

Log mean temperature difference experienced by water (θ_m) is given by,

$$\theta_m = \frac{\dot{Q}_{actual}}{U*A} \quad (20)$$

The formula below is used to obtain water outlet temperature (T_{wout})

$$\dot{Q}_{actual} = \dot{m}_w \times C_p \times (T_{wout} - T_{win}) \quad (21)$$

Here $\dot{Q}_{actual} = \dot{Q}_{water\ side}$ represents the heat given out by condenser for the three tubes dipped in water.

$$\text{Total heat rejected} = \dot{Q}_{water\ side} + \dot{Q}_{air\ side} \quad (22)$$

$$COP = \frac{h_1 - h_{3'}}{h_2 - h_1} \quad (23)$$

$h_1 h_2$ are found out using p-h chart and $h_{3'}$ is calculated using total heat rejected

III. CFD ANALYSIS:

A. Modelling

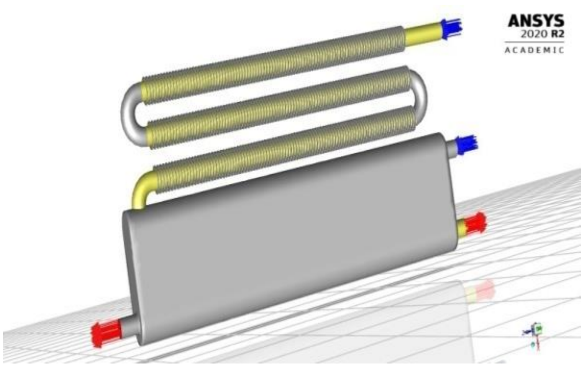

Fig 1. Prototype of AWCC Condenser

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In this section 3D (three dimensional) model of a condenser which is both air cooled and water cooled has been made in **ANSYS** by Computational Fluid Dynamic Modelling (CFD modelling) .

The condenser modelled for present study is show in Fig.1. The model consists of air cooled part and a water cooled part. The refrigerant used in condenser pipes in the study is R-22. To accommodate water for the water cooled part a box has been designed as shown and condenser tubes dipped in the box in order to be water cooled. Inlet and outlet is provided in the box for the entry and exit of the water. Fins made of aluminium have been provided on the air cooled part to disperse the heat more quickly.

Table I. Condenser specification dimensions

Total length of condenser tubes	3 m
Length of water box	588 mm
Internal diameter of fins	22.7 mm
External diameter of fins	32.7 mm
Fins spacing	5 mm
Thickness of fins	0.5mm
Diameter of condenser tubes	22.7 mm

B. Numerical Scheme

Ansys fluent software is used to find out the solution of governing equations. Viscous-Laminar model is used for simulation. Coupled method is selected for solution. To obtain the convergence in the solution 1e-06 is the absolute criteria for residuals.

Table II. Properties of R-22

Density(kg/m^3)	Specific Heat(J/Kg-K)	Thermal Conductivity(W/m-k)	Viscosity (Kg/m-s)
1069	1560	0.0658	0.001

Table III. Boundary Conditions at prototype

S.No	Location	Boundary Conditions	Thermal Conditions
1	Refrigerant inlet	mass-flow-inlet	Temperature
2	Water inlet	mass-flow-inlet	Temperature
3	Refrigerant outlet	Mass-flow-outlet	-
4	Water outlet	Mass-flow-outlet	-
5	Adiabatic wall	wall	Heat flux
6	Air side tube	wall	Convection
7	fins	wall	Convection
8	Refrigerant side tube	Interface	Temperature
9	Air side tube	Interface	Convection
10	Water side tube	Interface	Convection
11	fins	Interface	Convection

Table IV. Residual Table

Residual	Absolute Criteria
Continuity	1e-06
x-velocity	1e-06
y-velocity	1e-06
z-velocity	1e-06
Energy	1e-06

C. Grid Test

Grid independence study is performed to eliminate/reduce the influence of the number of grids/grid size on the computational results.If the results tend towards identical, the grid is considered as grid independent

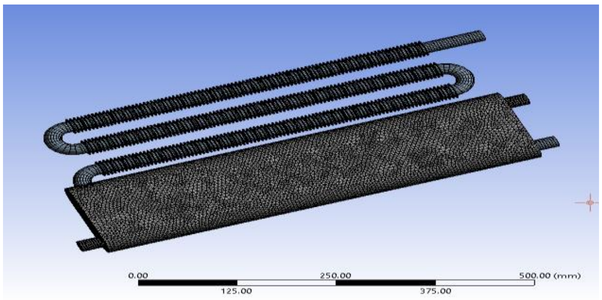

Table V. Comparison for Grid Independence

S.No	Element Size	Nodes	Elements	Water Outlet Temp (K)	Mesh Behaviour
1	4	191281	496170	299.74	Fine
2	5	178279	407367	299.99	Fine
3	6	164320	371827	302.34	Fine
4	7	162471	363090	310.23	Fine
5	15	162024	360856	315.47	Medium
6	25	161981	360571	319.76	Coarse

It has been observed from the table above that there is a sudden change in water outlet temp after 5 mm element size. Variation of water outlet temp is 0.42% and change in pressure at water outlet is approching to zerotherefore mesh with Element size 5 mm (Fig.2) which is fine in nature and has 178279 nodes and 407367 elements is selected for analysis.

D. Result

The temperature of the refrigerant at the inlet and outlet is 62°C and 43°C which clearly validates our assumption, the outlet temperature of refrigerant turns out to be slightly less than assumed because of sub cooling and the temperature of the water at the inlet and outlet is found to be 12°C and 31°C which also validate the temperature obtained through calculation (equation 21). Temperature around the fins turns out to be 42°C.

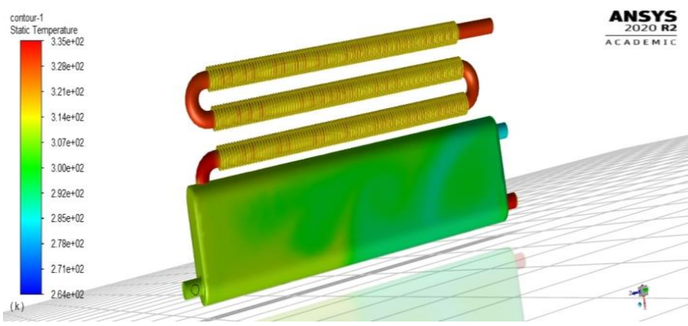

Fig 3. Contour of static temperature

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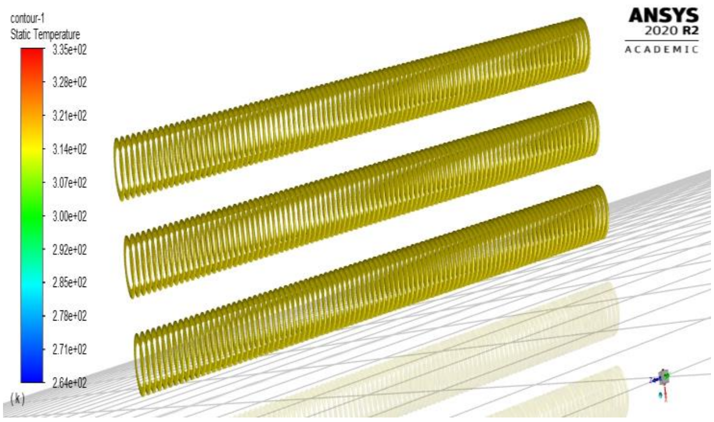

Fig.4 Contour of Static Temperature of Fins

Fins have been taken on air cooled condenser side and the air flowing over the fins is around 313 K. The material chosen for fins is aluminium and the convective heat transfer rate over the fins is 44.84 W/ m^2 K .Temperature of the fins using CFD analysis comes around 42 °C.

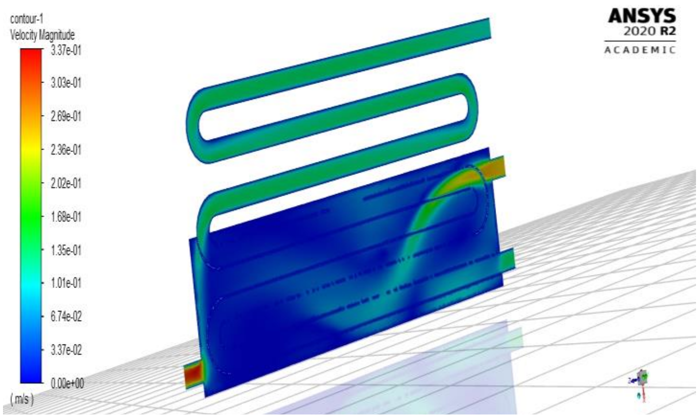

Fig.5 Contour of Velocity Magnitude

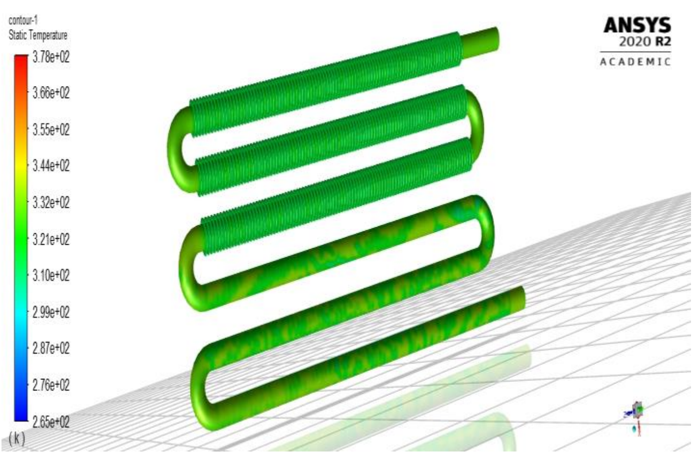

Fig.6 Contour of Static temperature of condenser tubes

IV. CONCLUSION

This paper gives you a 3D model of air water cooled condenser designed in ANSYS by CFD modelling.thecomparision of parameter obtained using theoretical calculations is validated using CFD analysis.The COP obtained for AWCC is 6.22 which is much higher than traditional air conditioner.Therefore,itis concluded that the coefficient of performance of residential air conditioner can be enhanced by using air water cooled condenser in place of air-cooled or water-cooled condenser.

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