



CFD Analysis of Air-Cooled Heat Exchanger Using Reduced Diameter Inner Grooved Copper Tube

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Abstract: Replacing copper tubes with small tube diameters is the popular trend to lower the cost of heat exchanger. This paper presents heat transfer enhancement using inner grooved copper tubes with small tube diameters as a substitution. Performance comparison as well as cost analysis are conducted for the two tubes of $\phi 12.7\text{mm}$ and $\phi 9.52\text{mm}$ inner grooved copper tube air-cooled condenser. The research results show that: where the testing conditions are same, the heat transfer coefficient with $\phi 9.52\text{mm}$ inner grooved copper tube is about 30% higher than within $\phi 12.7\text{mm}$ inner grooved copper tube. Where the dimension and operating conditions are same, $\phi 9.52\text{mm}$ inner grooved condenser can achieve the same heat exchanger performance as the $\phi 12.7\text{mm}$ inner grooved condenser, the copper material can be saved up to 15.2%.

Keywords: small diameter inner-grooved copper tube; fin tube heat exchanger; heat transfer performance; cost analysis

I. Introduction

Compact heat exchangers are characterized by a large heat transfer surface area per unit volume of the exchanger. Fin Tube Heat Exchanger is a typical compact heat exchanger that is widely used in many industrial power generation plants, chemical, petrochemical and petroleum industries [1]. Inner groove tubes are widely used in refrigerant evaporation and condensation. Heat transfer from a prime surface can be increased by attaching fins or extended surfaces to increase the area available for heat transfer. Fins provide a more compact heat exchanger with lower material costs for a given performance. Inner groove tube can give a better heat transfer coefficient due to increase contact surface between the tube wall and the refrigerant. As the groove surrounded the entire inner wall of the tube, it can produce better flow condition compare to plain tube. The centrifugal force inside the tube causes the cooling fluid to flow in the pattern of the groove, which result to an annular flow inside the tube [2].

Wu Yang *et al.* [3] substituted the copper tubes with small diameters. Performance comparison as well as cost analysis were conducted for two tube types of $\phi 9.52\text{mm}$ and $\phi 5\text{mm}$ inner-grooved copper tube, as well as the heat exchangers using these two types. Kadir Bilen *et al.* [4] experimentally investigated the surface heat transfer and friction characteristics of fully developed turbulent air flow in different groove tubes. Tests were performed for Reynolds number range of 10000 to 38000 and for different geometric grooved shape (circular, trapezoidal and rectangular) tubes. Goto *et al.* [5] investigated the condensation and evaporation heat transfer of R410A and R22 inside internally grooved horizontal tubes. The measured local pressure drop, heat transfer coefficients were compared with the predicted values from previous correlations proposed by the authors. Wei-kun Ding *et al.* [6] developed a discretized computational model and general simulation program in which a special circuit data structure was introduced. The study showed that increasing circuit branches number, decreasing the single branch length and adding more tubes in the circuit design, the refrigerant pressure drop were increased while heat exchange rate remained almost the same. Selvaraj P. *et al.* [7] presented computational fluid dynamics studies on heat transfer, pressure drop, friction factor, Nusselt number of a plain tube and tube equipped with three types of internal grooves (circular, square and trapezoidal). Water was used as the working fluid. Tests were performed for Reynolds number ranges from 5000 to 13500 for plain tube and different geometry inside grooved tubes.

II. Model Description

A. Physical model:

The typical fin-tube heat exchanger section with staggered arrangement is shown in Figure 1. The inner-grooved tube is a kind of high efficiency heat exchange component. Because it possesses 2~3 times the heat exchange capacity of the normal bare tube, it has received extensive attention. The high efficiency of the inner-grooved tube stimulates and promotes the development of energy saving and miniaturization for air-conditioning systems. The typical inner-grooved tube tooth-type parameters are shown in Figure 2. Of them, d is the outer diameter, δ is the tube wall thickness, h is the tooth height, β is the helix angle, γ is the tooth vertex angle and n is the tooth number.

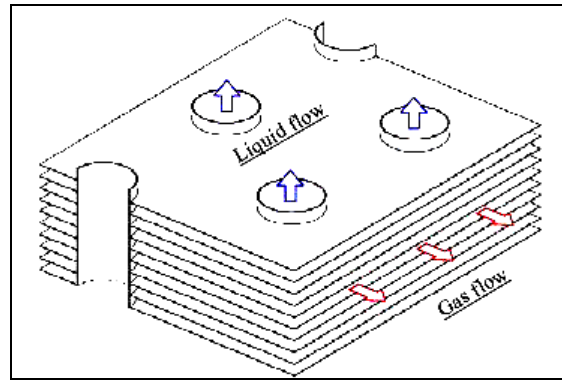


Fig. 1: Typical Fin-Tube Heat Exchanger section with staggered arrangement.

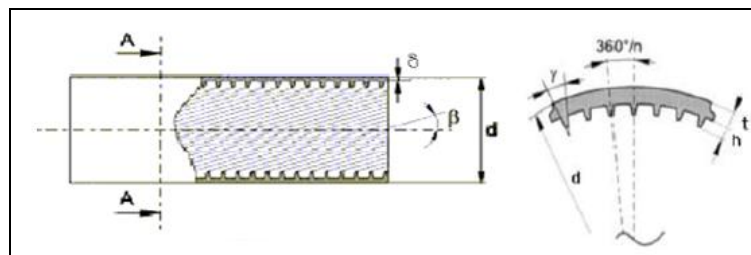


Fig. 2: Inner groove tube structure

B. Computational model and boundary conditions for $\phi 9.52\text{mm}$ heat exchanger:

B.1. Fin

The Gambit Software is used to create and mesh the computational model. A diagram of the studied model is shown in Figure 3, and consists of air flow area between two fins of plain fin geometry and around the surfaces of four rows of tubes.

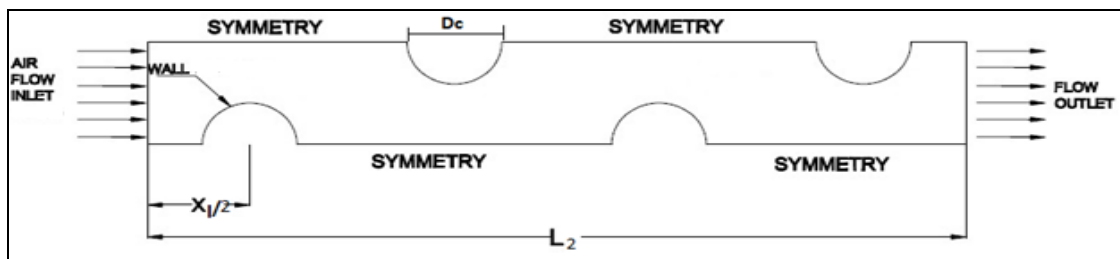


Fig. 3: Computational domain and boundary conditions of model

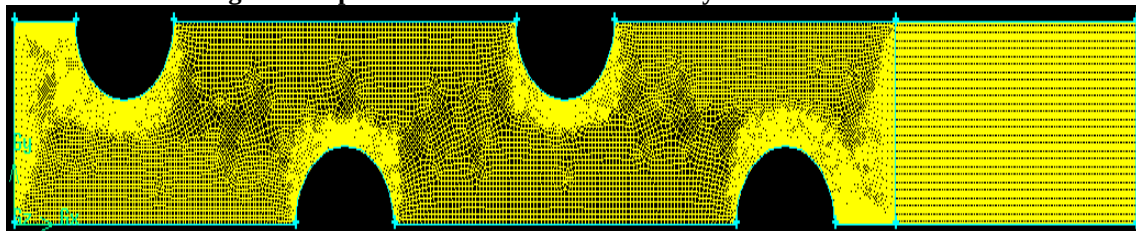


Fig. 4: Computational domain and extended surface of model

The computational domain contains the boundary conditions as shown in Figure 3 with the following conditions:

Tube surfaces as wall, Dirichlet Boundary condition

$$T = T_w = 54.5^\circ\text{C} = 327.5$$

Air velocity $u = v = w = 0$, that is no-slip condition at tube surfaces.

Uniform velocity $u = u_{in} = 3.8 \text{ m/s}$

$$v = w = 0$$

$$T = 40^\circ\text{C} = 313\text{K}$$

Outlet as a pressure outlet, Neumann Boundary condition

Zero gradients, u, v, w , pressure and temperature.
Side planes as a symmetry, symmetry planes
 $(\partial u / \partial y) = 0, v = 0, (\partial w / \partial y) = 0, (\partial t / \partial y) = 0$

The entire computational domain was made up of approximate 34622 finite elements with structure grid throughout most of the domain, while the areas around the tubes are more unstructured.

B.2. Inner groove tube:

The Solidworks software is used to create the fluid part of the inner groove tube as shown in Figure 5. The geometry is then exported in Gambit in ACIS format to mesh the computational domain as shown in Figure 6.

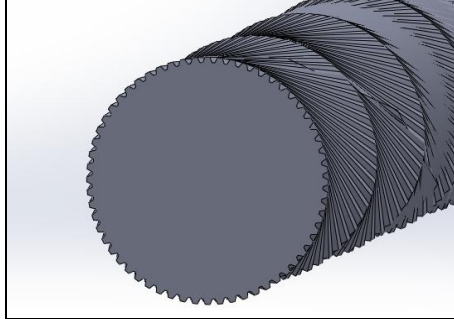


Fig. 5: Computational domain of inner groove tube in Solidworks

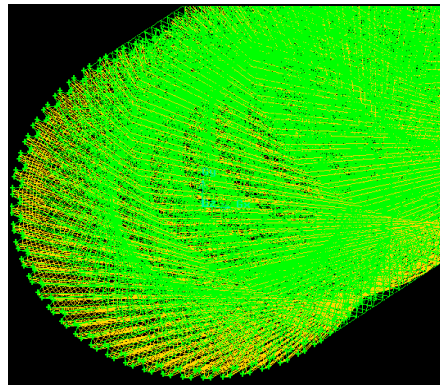


Fig. 6: Computational domain of inner groove tube in gambit

The computational domain contains the following boundary condition:

Inlet as mass flow inlet, Dirichlet Boundary conditions
 $m = m_r = 0.053$ kg/s, along the z -direction

Outlet as pressure outlet, Neumann Boundary condition

Zero gradients, u, v, w , pressure and temperature.

Tube surface as a wall, Dirichlet Boundary condition

Heat transfer coefficient and free stream temperature of air were applied on wall surface.

The entire domain was made up of approximate 161000 finite volumes, with a structured grid throughout most of the domain.

III. Heat Transfer Calculations

Since the condenser is a heat exchanger, one of the heat transfer and pressure drop calculation methods, LMTD (Log Mean Temperature Difference) method is used in this study. Total heat transfer capacity is calculated from Eq. 1,

$$Q = U_o A_o \Delta T_m \quad (1)$$

where, Q is the heat transfer capacity, U_o is the overall heat transfer coefficient, A_o is the total outside heat transfer area, ΔT_m is the Log Mean Temperature Difference (LMTD) for cross flow.

Overall heat transfer coefficient is calculated from Eq. 2 (Kays and London, 1984),

$$U_o A_o = \frac{1}{\frac{1}{h_i A_i} + \frac{r_i \ln \frac{d_o}{d_i}}{k} + \frac{1}{h_o (\eta_f A_f + A_t)}} \quad (2)$$

where, A_i is the tube inside surface area, h_i is the refrigerant side heat transfer coefficient, k is the tube thermal conductivity, d_o is the tube outside diameter, d_i is the tube inside diameter, h_o is air-side heat transfer coefficient, η_f is the fin efficiency, A_f is the finned surface area, A_t is the tube outside surface area.

IV. Results and Discussions

A. Computational Fluid Dynamics Results:

A.1. Fin

A characteristic of flow describes the initial observation found using Contour Display after running the CFD simulation in Fluent. This gives the behavior of the air flow velocity inside the computational domain.

a) Velocity Observations:

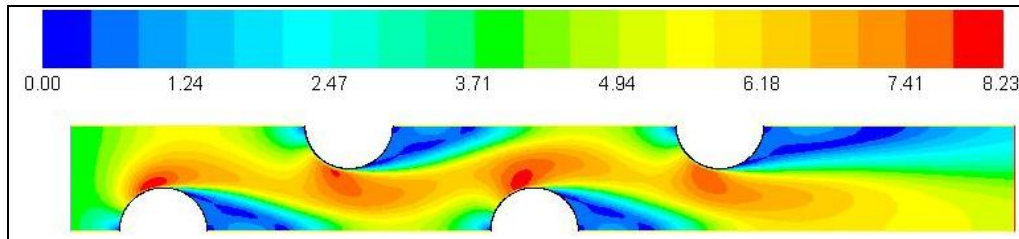


Fig. 7: Contours of velocity field, k-epsilon model, inlet air flow 3.8 m/s of $\phi 12.7\text{mm}$ heat exchanger

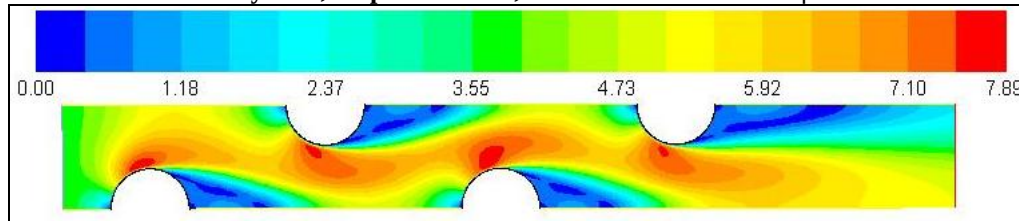


Fig. 8: Contours of velocity field, k-epsilon model, inlet air flow 3.8 m/s of $\phi 9.52\text{mm}$ heat exchanger

The air enters at the inlet and exits at the outlet as shown in the above figures. The outlet is extended twice the height in order to reduce reversed flow. Now the flow characteristic of air in heat exchanger can be easily understood. In the above cases, as the air flow around the first tube, it begins to speed up and then the air velocity increases again as it goes around the second tube. From calculation, it can be seen that, the minimum free-flow area of $\phi 12.7\text{mm}$ heat exchanger is less than that of $\phi 9.52\text{mm}$ heat exchanger, which showed that the velocity going around the second tube is faster than that going around the first tube. The minimum free flow area is the area of the heat exchanger between two transverse tubes, so the area just above tube one or just below the second tube are the minimum free-flow areas. The flow is forced to speed up, as the tubes act as a type of pipe contraction in the air flow channel. As the velocity increases along the flow the Pressure of air flow is decreases accordingly.

It is observed that the size of the tubes impact the Reynolds number of the air flowing around them, since with larger tubes there would be an even smaller minimum free-flow area. In this study, the characteristic length for the Reynolds number is the tube collar diameter, and it can be seen here, that increases in this parameter can induce higher velocities and with it a higher turbulence and Reynolds number.

b) Temperature Observation:

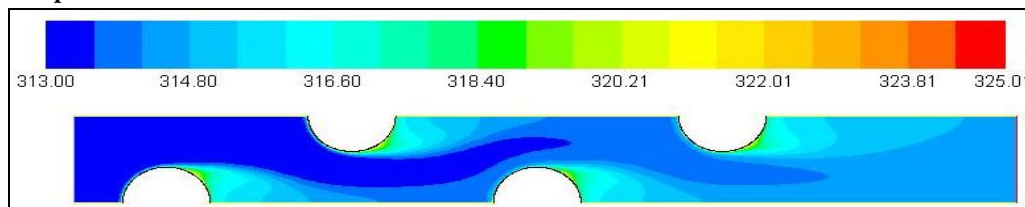


Fig. 9: Contours of temperature field, k-epsilon model, inlet air flow 3.8 m/s of $\phi 12.7\text{mm}$ heat exchanger.

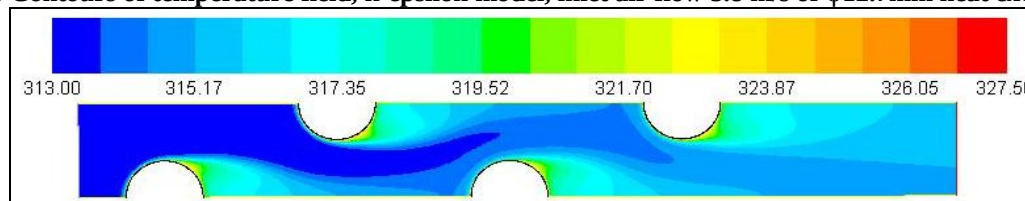


Fig. 10: Contours of temperature field, k-epsilon model, inlet air flow 3.8 m/s of $\phi 9.52\text{mm}$ heat exchanger.

The temperature contours for the two kinds of heat exchangers is shown in the figure 9 and 10. For high Reynolds number as in case of air flow for $\phi 12.7\text{mm}$ heat exchanger, shown in figure 9 the air has not sufficient

time to absorb the heat from the tube and fin. So the air temperature at the outlet is lower than as in low Reynolds number flow. The largest temperature changes for the cases are occurring in the recirculation and slow velocity zones just after each of the tubes. As in the slow-moving flow in the case with $\phi 9.52\text{mm}$ heat exchanger, the slow-moving areas of the heat exchanger are also better able to absorb heat.

A.2. Inner Groove Tube

a) Velocity Observation:

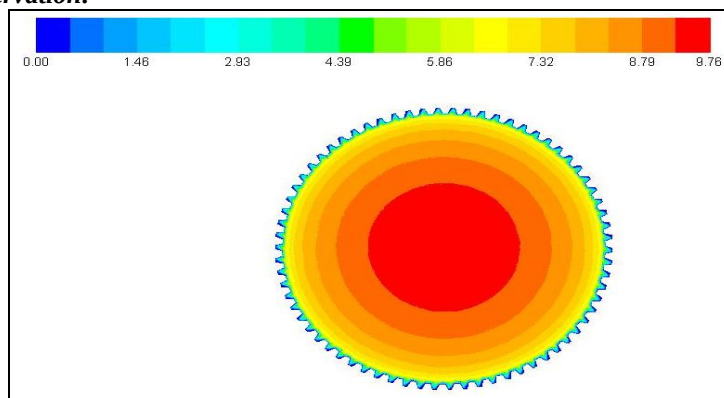


Fig. 11: Contours of velocity field, k-epsilon model and mass flow rate 0.086kg/s of $\phi 12.7\text{mm}$ heat exchanger

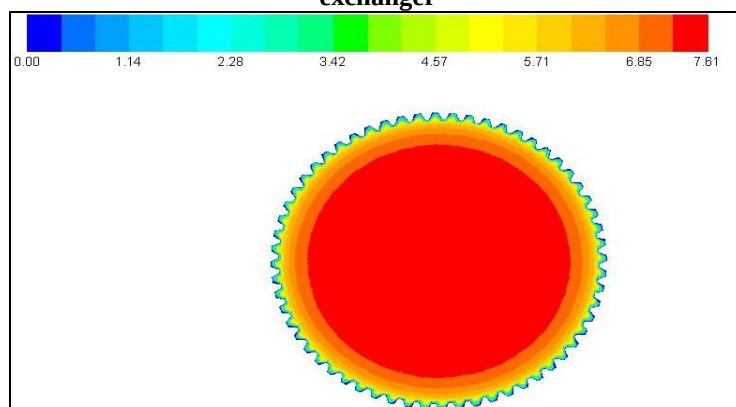


Fig. 12: Contours of velocity field, k-epsilon model and mass flow rate 0.053kg/s of $\phi 9.52\text{mm}$ heat exchanger.

The velocity contours of inner groove tube of both kinds of heat exchangers are shown in Figures 11 and 12. It can be seen that the refrigerant closest to the wall tends to move slower than the fluid at the center. The internal enhancement in Inner-Groove tube reduces boundary layers and increase heat transfer through the tube wall. The additional mixing of the refrigerant that occurs inside the tube because of these grooves increases the amount of refrigerant that comes in contact with the tube wall. From the above figure, it can be seen that, the high velocity increases in $\phi 9.52\text{mm}$ tube due to the homogeneity of the refrigerant.

b) Temperature Observations:

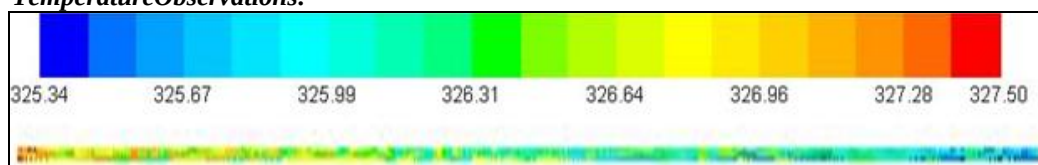


Fig. 13: Contours of temperature field, k-epsilon model and mass flow rate 0.053kg/s of $\phi 12.7\text{mm}$ heat exchanger.

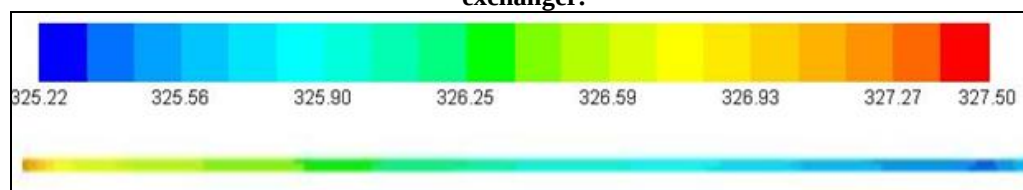


Fig. 14: Contours of temperature field, k-epsilon model and mass flow rate 0.053kg/s of $\phi 9.52\text{mm}$ heat exchanger.

It is speculated that the grooved tubes provide more increase in heat transfer augmentation, due to not only producing more turbulence and good fluid mixing than the smooth tube but also increase in heat transfer area. They also provide periodic redevelopment of the boundary layers and cause a more effective heat transfer. The temperature contours of the two kind of inner groove tube are shown in figures 13 and 14. Higher pressures are typically required to condense the environmentally friendly refrigerants. The temperature drop of the refrigerant is uniform in $\phi 9.52\text{mm}$ tube as compared to $\phi 12.7\text{mm}$ tube. This is because the working pressure is more in $\phi 9.52\text{mm}$ tube.

B. Performance and cost analysis of heat exchangers:

Geometric Parameter	Symbol	Value	Unit
Heat Duty	Q	65	KW
Air Volume	-	5.62	m^3/s
Air Inlet Velocity	V	3.8	m/s
Air inlet Temperature	$T_{a,i}$	40	$^{\circ}\text{C}$
Condensing Temperature	T_c	54.5	$^{\circ}\text{C}$
Refrigerant	-	R22	-

Table 1: The operating conditions of Air-Cooled Condenser

Geometric Parameter	Symbol	Value	Unit
Finned Length	L_1	1220	mm
Finned width	L_2	110	mm
Finned height	L_3	1206.5	mm
Fin Spacing	N_f	2.1167	mm
Vertical Tube Pitch	X_t	31.75	mm
Horizontal Tube Pitch	X_l	27.5	mm
Fin Thickness	δ'	0.15	mm
Tube Outer Diameter	d_o	12.7	mm
Tube Inner Diameter	d_i	11.48	mm
Collar Diameter	D_c	13	mm
Bottom wall thickness	δ	0.36	mm
Helix angle	β	18	$^{\circ}$
Tube Fin Height	h	0.25	mm
Tube Fin angle	γ	40	$^{\circ}$
Number of grooves	n	70	-
Number of tubes	N_t	152	-
Tube weight/meter	-	140	g/m

Table 2: Structure parameters for $\phi 12.7\text{mm}$ heat exchanger

Geometric Parameter	Symbol	Value	Unit
Finned Length	L_1	1220	mm
Finned width	L_2	88	mm
Finned height	L_3	1219.2	mm
Fin Spacing	N_f	2.1167	mm
Vertical Tube Pitch	X_t	25.4	mm
Horizontal Tube Pitch	X_l	22	mm
Fin Thickness	δ'	0.15	mm
Tube Outer Diameter	d_o	9.52	mm
Tube Inner Diameter	d_i	8.72	mm
Collar Diameter	D_c	9.82	mm
Bottom wall thickness	δ	0.3	mm
Helix angle	β	18	$^{\circ}$
Tube Fin Height	h	0.2	mm
Tube Fin angle	γ	53	$^{\circ}$
Number of grooves	n	60	-
Number of tubes	N_t	192	-
Tube weight/meter	-	94	g/m

Table 3: Structure parameters for $\phi 9.52\text{mm}$ heat exchanger

Output variables	Unit	Baseline model	3/8" heat exchanger	Difference (%)
Refrigerant-side Pressure drop	KPa	9.945	15.774	58.6
Refrigerant-side Nusselt number	-	1445.7	1514.8	4.78
Refrigerant heat transfer coefficient	$\text{W/m}^2\text{K}$	1869	2588.36	38.49
Total mass flow rate	kg/s	6.536	5.088	-22.15
Copper tube weight	kg	27.55	23.365	-15.2

Table 4: The comparison of modeling output and their corresponding results from baseline model

To verify the modeling results, the simulation output $\phi 9.52\text{mm}$ heat exchanger were compared with the corresponding $\phi 12.7\text{mm}$ heat exchanger. The two kinds of heat exchangers are simulated using the same entry parameters as shown in Table 1. Table 2 and Table 3 show the structure parameters of the $\phi 12.7\text{mm}$ heat

exchanger and $\phi 9.52\text{mm}$ heat exchanger. Also it can be seen that weight of tube is reducing in $\phi 9.52\text{mm}$ heat exchanger. The Numerical results shows that after adjusting the tube number and tube spacing, the $\phi 9.52\text{mm}$ inner-grooved heat exchanger can reach the almost the same heat exchanger performance as that of the $\phi 12.7\text{mm}$ inner-grooved heat exchanger. The comparison of modeling output and their corresponding results are shown in Table 4. It can be seen that the copper tube usage amount is reduced by 15.2%. The length and height are the same for both kinds of heat exchangers. Therefore, the aluminum foil usage for the $\phi 9.52\text{mm}$ heat exchanger is reduced by 20%.

C. Model Validation

The $\phi 9.52\text{mm}$ heat exchanger is also compared with the following correlations:

The Nusselt number for plain fin, staggered arrangement heat exchanger is defined by Zhukauskas' (1972) [8],

$$Nu = 0.3115 Re_{D,\max}^{0.6} Pr^{0.36} \quad (3)$$

For inner-grooved tube is defined by Kadir Bilen *et al* [4],

$$Nu = 0.014 Re^{0.893} Pr^{\frac{1}{3}} \quad (4)$$

The Nusselt number of $\phi 9.52\text{mm}$ heat exchanger for the plain fin and inner-grooved tube, when compared with the correlation, is reduced by 10% and 21% respectively.

V. Conclusions

The conclusion obtained is as below:

- (1) Under the same testing conditions, the refrigerant-side heat transfer coefficient within the $\Phi 9.52\text{mm}$ inner-grooved copper tube is around 38.49% higher than that within the $\Phi 12.7\text{mm}$ inner-grooved copper tube. The material cost is thus greatly reduced.
- (2) For the same dimensions and under the same operating conditions, the $\Phi 9.52\text{mm}$ inner-grooved copper tube heat exchanger can achieve almost the same heat exchange performance as the $\Phi 12.7\text{mm}$ inner-grooved copper tube heat exchanger. The copper tube usage amount can also be reduced by 15.2%. The aluminum foil cost is reduced by around 20%.
- (3) While keeping the windward cross-sectional dimensions unchanged, through adjusting the refrigerant manifold and tube spacing, as well as structural parameters like the fin spacing, the heat exchanger with small tube diameter can obtain the same performance as the copper tube heat exchanger with big tube diameter.

Nomenclature

Q	Heat Transfer Capacity (W)
U_o	Overall heat transfer coefficient ($\text{W}/\text{m}^2\text{K}$)
A_o	Total outside heat transfer area (m^2)
ΔT_m	Log Mean Temperature Difference ($^{\circ}\text{C}$)
A_i	Tube inside surface area (m^2)
h_i	Refrigerant side heat transfer coefficient ($\text{W}/\text{m}^2\text{K}$)
k	Tube thermal conductivity (W/mK)
d_o	Tube outside diameter (m)
d_i	Tube inside diameter (m)
h_o	Air-side heat transfer coefficient ($\text{W}/\text{m}^2\text{K}$)
η_f	Fin efficiency
A_f	Finned surface area (m^2)
A_t	Tube outside surface area (m^2)

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Acknowledgments

This work was supported by members of VJTI at Mumbai and Star Coolers and Condensers Pvt Ltd., Jalgaon.