SIMULATION ANALYSIS OF FINNED TUBE COIL HEAT EXCHANGER FOR SPLIT-AIR CONDITIONER

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Abstract- Decreasing the diameter of copper tube is advantageous to Decrease the quantity of copper material Consumption in heat exchanger. The reduction of the diameter of the copper tube can decrease the volume of the heat exchanger. The aim of this study is to find-out the finned tube coil heat exchange performance with design in split air-conditioners. Finned tube coil is cad-model using simulation & heat transfer method. Results will be comparing with the 7mm Fin-Tube model. So, it is possible to find-out the performance without experimental test which are costly & takes lengthy duration. Main reason of this study is to simulate split air conditioner finned tube coil heat exchanger performance with heat transfer & design modification together for higher energy efficiencies demands & affordable in residential air conditioning systems with higher capacities for heat transfer. By using the simulation software EVAP-COND to simulate the heat exchange performance and compare the refrigerant pressure drop, refrigerant mass flow rate, copper mass consumption with the condenser and Optimize using ISHED Simulation.

***Keywords*- Split air conditioners, finned tube coil, Simulation, Heat transfer performance, circuit optimization, R32**

I. INTRODUCTION

Copper metal is the most essential material in HVAC&R industry, and the magnification rate of the copper consumption is approx 9%~13% per year. Cost increase of copper producing the traditional heat exchanger cost rise, decreasing the consumption amount of the copper is the mode of evolution of HVAC industry. reduce diameter heat exchanger has the notable merit and it is highly changing the larger tube diameter heat exchanger, such as 7.0mm, 7.94mm and etc. Simulation is mostly for the product design development in the application of heat exchanger. The maximize design of the heat exchanger is achieved by simulation. In the past era, the simulation application was underdeveloped, so it was essential to construct a machine and verify the advantages and problems of the exchanger, and then enhance it. This procedure is not only expensive but also takes a great of time. Simulation application has been broadly used in present years, which can save the resources and reduce duration of the growth cycle. Decreasing the diameter of the copper tube can decrease the heat exchanger volume, with the R32 which environmentally friendly refrigerants widely used in this era. At present, the development of heat exchanger is mainly focused on the heat exchanger with 7.0 mm copper tube, and the development of the reduce diameter heat exchanger.

II. THEORETICAL STUDY

Air-side heat transfer coefficient (h) (Incropera & DeWitt, 2001),

$$h = \frac{Nu\,.\,k}{D_h}, \quad Nu = 0.38.Re^{0.6}.Pr^{\frac{1}{3}}.\left(\frac{A}{A_p}\right)^{-0.15}, \quad Re = \frac{G.D_h}{\mu}, \quad D_h = 4.L.\frac{A_o}{A} \qquad (2.1)$$

h: Heat exchange coefficient (W/m^2K), Nu: Nusselt number, k: Conduction heat exchange coefficient of the air (W/mK), Dh: Hydraulic diameter (m), Re: Reynolds number, G: Mass flux rate (kg/m^2s), μ: Air dynamic viscosity at mean temperature (Ns/m^2), umax: Maximum velocity (m/s).

Air side pressure drop condenser (Kakaç & Liu, 1998)

$$\Delta P_{air} = \frac{G^2_{air,c}}{2\rho_{air,in}}\left[\frac{A_{air}\rho_{air,in}}{A_c\rho_{air,m}}f_{air} + \left(1+\sigma^2\right)\left(\frac{\rho_{air,in}}{\rho_{air,out}} - 1\right)\right], \quad \sigma = \frac{A_o}{A_{fr}}, \quad \frac{1}{\rho} = \frac{1}{2}.\left(\frac{1}{\rho_i} + \frac{1}{\rho_0}\right) \qquad (2.2)$$

ρ: mean air density (kg/m^3), ρi: air density at inlet temperature (kg/m^3), ρo: air density at surface temperature (kg/m^3), Ao: minimum flow area, Afr: frontal area, Aair: total area of coil

Refrigerant Side Correlation

Shah Correlation heat transfer coefficient of the refrigerant side (Shah & Sekulic, 2003)

$$h_r = h_l\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{x^{0.38}}\right]$$

$$h_l = 0.023\,Re_l^{\,0.8}\,Pr_l^{\,0.4}\,k_l / D \qquad (2.3)$$

Or

$$h_r = \frac{Heat\ flux_{ref}\left(\frac{W}{m^2}\right)}{(T_{w,i} - T_b)K} \qquad (2.4)$$

Refrigerant Side Pressure Drop (Shah & Sekulic, 2003)

$$\Delta P_{ref} = \Delta P_f + \Delta P_a + \Delta P_g \qquad (2.5)$$

Where Pf, Pa, Pg, are the friction term, the acceleration term & the gravitational term

$$\Delta P_f = 4f_{\text{ref}}\left(\frac{L_{\text{ele}}}{D_i}\right)\left(\frac{G^2}{2\rho_{\text{ref},m}}\right)$$

$$\Delta P_a = G^2\left(\frac{1}{\rho_{\text{ref,out}}} - \frac{1}{\rho_{\text{ref,in}}}\right)$$

$$\Delta P_a = \rho_{\text{ref},m} g L_{\text{ele}} \sin\theta \quad (2.6)$$

Kl: liquid thermal conductivity (W/(mK)), X: thermodynamic vapour quality, Re: Reynolds Number, Pr: Prandalt Number, D: hydrodynamic diameter, F: friction factor, p: refrigerant density, g: gravity, G: mass flux, L: length of Element, P: density of refrigerant, Twi: temperature of the inner tube surface , Tb: average refrigerant tube inlet and outlet temperature.

Fin Efficiency (Kakaç & Liu, 1998)
Nf = tanh (m^2)/mL (2.7)
m^2 = (2*ha)/(k*tf) (2.8)

Overall Efficiency (Kakaç & Liu, 1998)
No = 1-Af/A(1-Nf) (2.9)

Ha: airside HTC, K: thermal conductivity of fin, Tf: fin thickness.

III. DESIGN DATA

In this research, the 7 mm Fin-tube of condenser of 2-1 inlet-outlet with the refrigerant R32. The new condenser is enhanced to the heat exchanger with 5.0 mm coil diameter and R32. 5mm Fin-tube flow circuit layout 5-5, 6-6 inlet-outlet. The heat transfer between the air and the refrigerant is the opposite cross heat transfer, which exit position of refrigerant is the entrance of air. The effect of different flow circuit on the heat exchange performance of the 5 mm condenser is simulated by simulation software EVAP-COND & ISHED, and the results are compare with the 7.0 mm condenser. The original condenser, modified condenser specification are shown in Table 1

Table 1 Specification

Sr. No.	Data	Dimension	7mm	5mm
1	Copper Tube O.D.	mm	7	5
2	Copper Tube I.D. Without Groove	mm	6.14	4.18
3	Copper Wall Thickness	mm	0.25	0.23
4	Aluminium Thickness	mm	0.1	0.095
5	Collar Height	mm	1.90	1.90
6	R.B. Pitch (Transverse \ Longitudinal)	mm	20\16	20\16
7	Collar O.D	mm	8.5	6.5
8	Coil Size (Tube Length L1* Fin Width L2 * Fin Length L3)	mm	791*25*520	809*25*520
9	FPI (Fin Per Inch)	inch	16 (fin pitch 1.5875 mm)	16 (fin pitch 1.5875 mm)
10	Groove Height Hf * bottom thickness Tw * groove Width W3	mm	0.18*0.25*0.29	0.18*0.23*0.29
11	Groove thickness = Hf + Tw	mm	0.180+0.250	0.180+0.230
12	Super Slit 5 cut thickness (Geometry)	mm	1 (Lanced)	1 (Louver)
13	Pitch Ratio (Transverse \ Longitudinal)		20/12.49=1.60	20/12.49=1.60
15	Hairpin * No. of Holes		26HP * 52Holes	26HP * 52Holes

Table 2 Calculated Parameter

	Condenser Coil Calculated Parameters			
Sr. No.	**Data**	**Dimension**	**7mm**	**5mm**
1	Nusselt No.	-	94.996	81.506
2	Reynolds No.	-	23213.21	19840.32
3	Prandlt No.	-	0.71	0.71
4	Air Side Heat Transfer Coefficient	W/m^2K	38.8825	28.4756
5	J-Factor	-	5.76E-03	4.60E-03
6	Air Side Pressure Drop	Pa	24.82	20.239
7	F-Factor	-	0.02648	0.04488
8	A/F-Factor	-	0.2176	0.1025
9	Fin Efficiency	-	69.60%	76.70%
10	Overall Efficiency	-	71.40%	77.80%
11	**Area & Volume By Equations**			
12	Ap copper Area	m^2	0.847124	0.6209386
13	Af fins area	m^2	11.014906	12.02615389
14	A total Area	m^2	11.862031	12.8824775
15	Ao minimum flow Area	m^2	0.250514	0.2966267
16	Afr Frontal Area	m^2	0.41132	0.42068
17	Copper Weight	kg	3.4	2.3
18	Aluminium Weight	kg	1.5	1.6
19	Copper Volume	m^3	3.79E-04	2.58E-04
20	Aluminium Volume	m^3	5.49E-04	5.78E-04
21	Hydraulic Diameter	m	0.0668205	0.0782841

IV. SIMULATION CONDITIONS

Simulation Operating Condition for 7mm fin-tube and 5mm fin-tube heat exchanger given below in Fig. 1, 2.

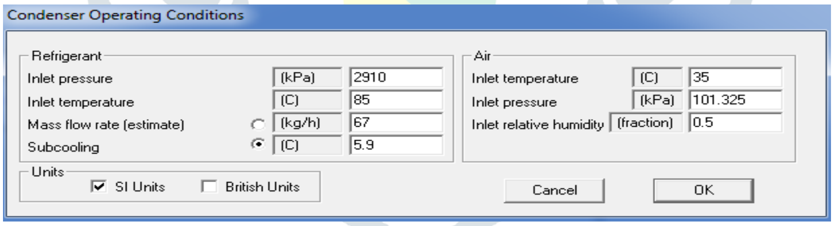


Fig. 1 7mm Operating Condition

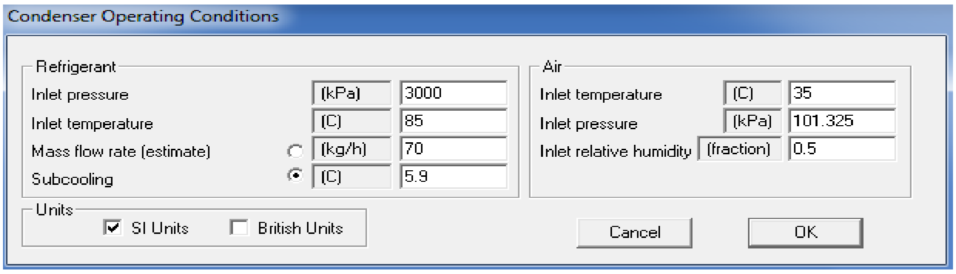
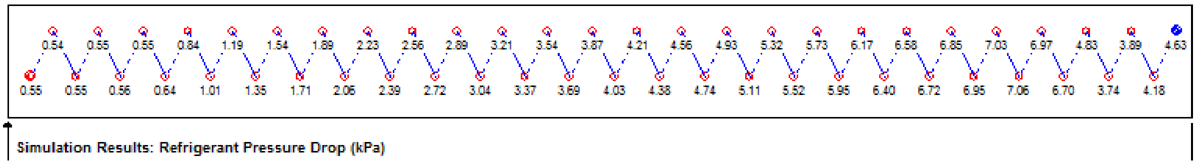


Fig. 2 5mm Operating Condition

V. SIMULATION

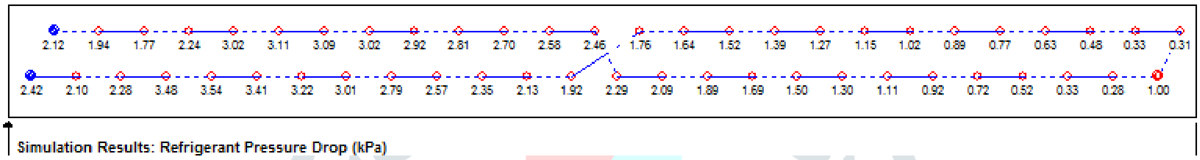
7mm Simulation

Air		Refrigerant		Results	
Inlet temperature (C)	35.0	Inlet pressure (kPa)	2810.0	Total capacity (kW)	4.17
Inlet pressure (kPa)	101.32	Inlet temperature (C)	85.00		
Inlet relative humidity (fraction)	0.50	Outlet quality (-)	0.00		
Vol. flow rate (m³/min)	35.00	Outlet subcooling (C)	6.06		
		Mass flow rate (kg/h)	48.00		



(a) 7mm Layout 1 Simulation

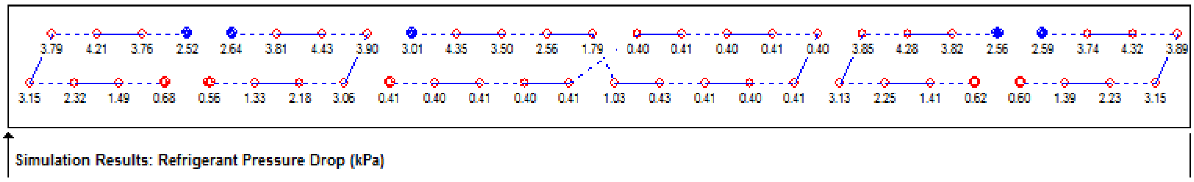
Air		Refrigerant		Results	
Inlet temperature (C)	35.0	Inlet pressure (kPa)	2910.0	Total capacity (kW)	5.61
Inlet pressure (kPa)	101.32	Inlet temperature (C)	85.00		
Inlet relative humidity (fraction)	0.50	Outlet quality (-)	0.00		
Vol. flow rate (m³/min)	35.00	Outlet subcooling (C)	6.07		
		Mass flow rate (kg/h)	67.00		



(b) 7mm Layout 2 Simulation

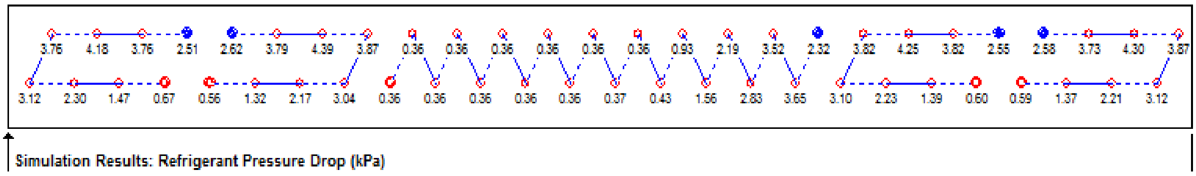
5mm Simulation

Air		Refrigerant		Results	
Inlet temperature (C)	35.0	Inlet pressure (kPa)	3000.0	Total capacity (kW)	5.75
Inlet pressure (kPa)	101.32	Inlet temperature (C)	85.00		
Inlet relative humidity (fraction)	0.50	Outlet quality (-)	0.00		
Vol. flow rate (m³/min)	35.00	Outlet subcooling (C)	5.86		
		Mass flow rate (kg/h)	70.00		



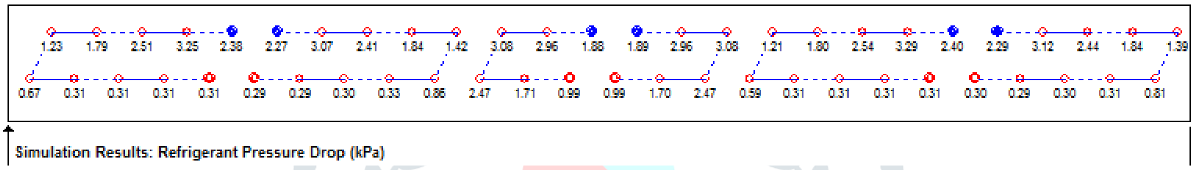
(c) 5mm New Layout 1 Simulation

Air		Refrigerant		Results	
Inlet temperature (C)	35.0	Inlet pressure (kPa)	3000.0	Total capacity (kW)	5.67
Inlet pressure (kPa)	101.32	Inlet temperature (C)	85.00		
Inlet relative humidity (fraction)	0.50	Outlet quality (-)	0.00		
Vol. flow rate (m³/min)	35.00	Outlet subcooling (C)	5.79		
		Mass flow rate (kg/h)	69.00		



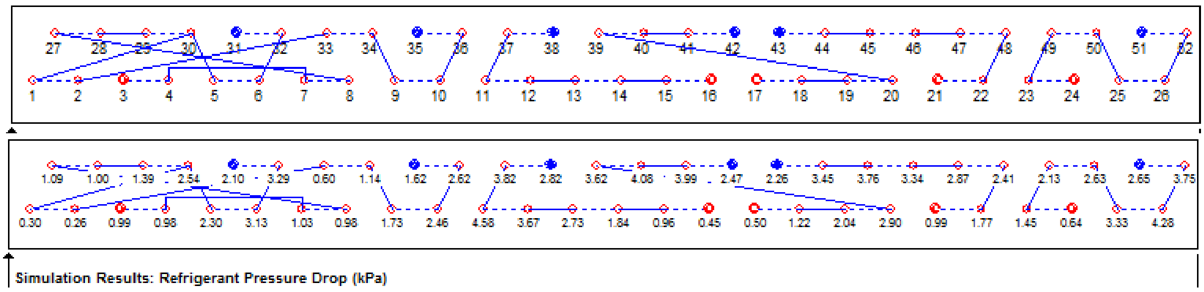
(d) 5mm New Layout 2 Simulation

Air		Refrigerant		Results	
Inlet temperature (C)	35.0	Inlet pressure (kPa)	3000.0	Total capacity (kW)	5.81
Inlet pressure (kPa)	101.32	Inlet temperature (C)	85.00		
Inlet relative humidity (fraction)	0.50	Outlet quality (-)	0.00		
Vol. flow rate (m³/min)	35.00	Outlet subcooling (C)	5.45		
		Mass flow rate (kg/h)	71.00		



(e) 5mm New Layout 3 Simulation

Air		Refrigerant		Results	
Inlet temperature (C)	35.0	Inlet pressure (kPa)	3000.0	Total capacity (kW)	6.48
Inlet pressure (kPa)	101.32	Inlet temperature (C)	85.00		
Inlet relative humidity (fraction)	0.50	Outlet quality (-)	0.00		
Vol. flow rate (m³/min)	35.00	Outlet subcooling (C)	5.98		
		Mass flow rate (kg/h)	79.00		



(f) ISHED Simulation

Fig. 3 Simulation

VI. RESULTS

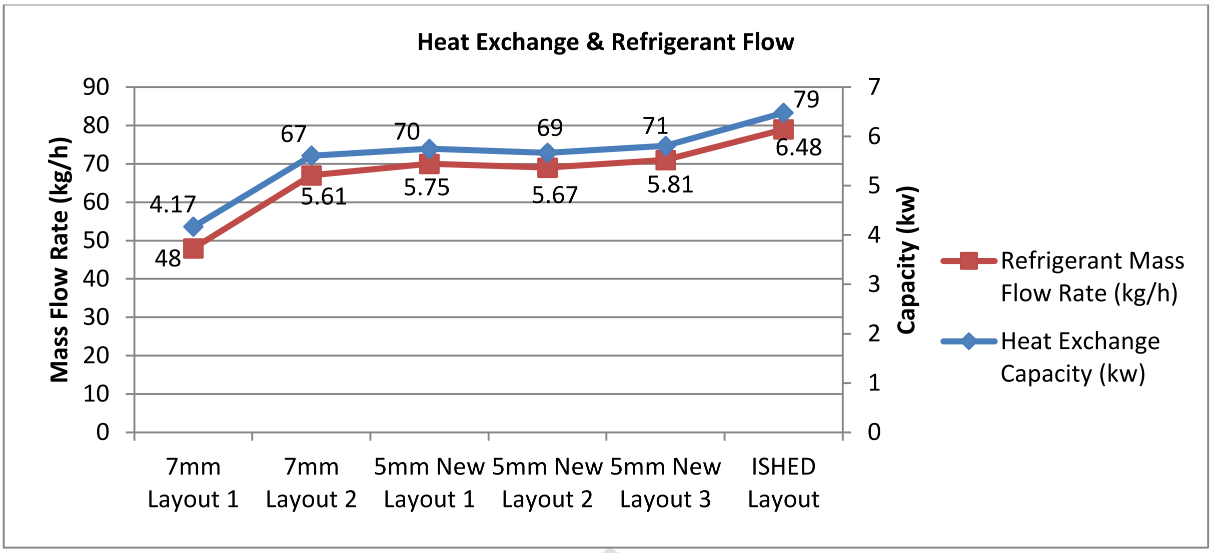


Fig. 4 Heat Exchange & Refrigerant Flow Chart

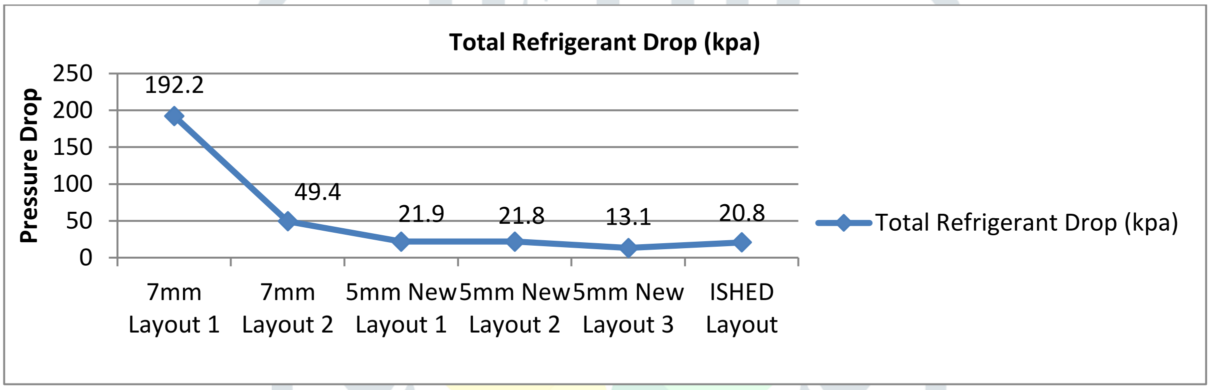


Fig. 5 Total Refrigerant Pressure Drop Chart

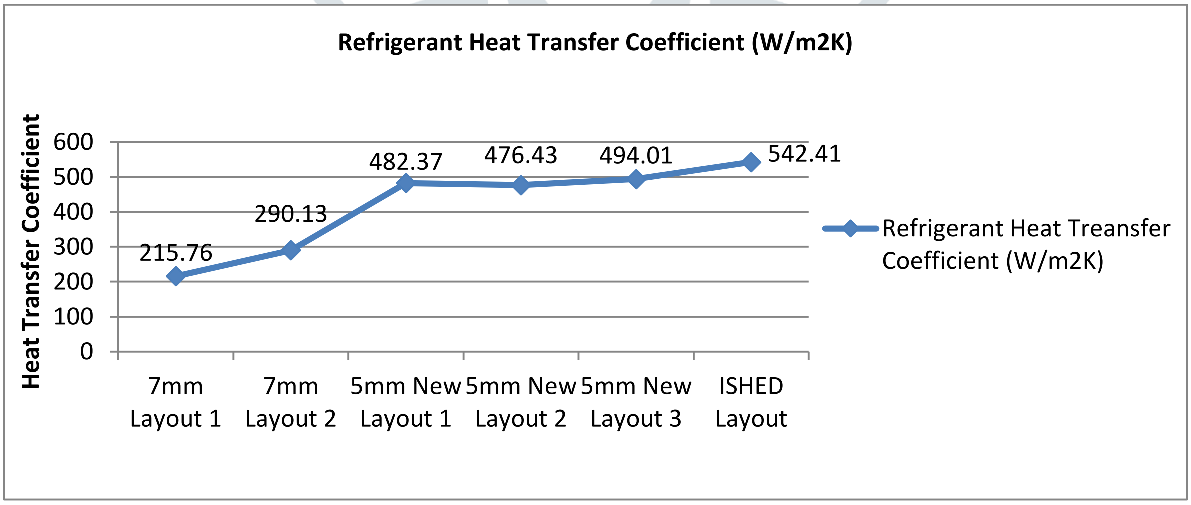


Fig. 6 Refrigerant Heat Transfer Coefficient Chart

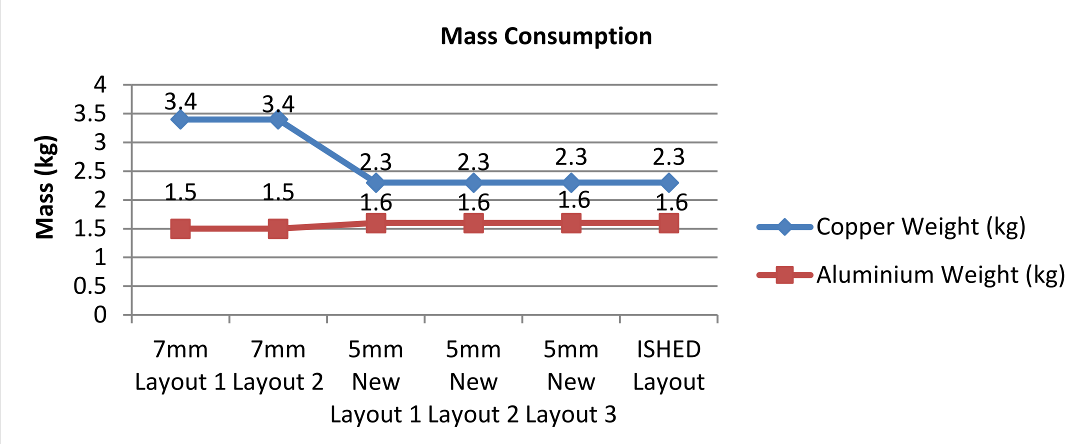


Fig. 7 Mass Consumption Chart

Table 3 Comparison Parameter

Sr. No.	Parameters	7mm Layout 2	5mm New Layout 1	5mm New Layout 2	5mm New Layout 3	ISHED Layout
1	Heat Exchange Capacity(kw)	5.61	5.75	5.67	5.81	6.48
		+	2.49%	1.06%	3.56%	15.50%
2	Refrigerant Mass Flow Rate(kg/h)	67	70	69	71	79
		+	4.47%	2.98%	5.97%	17%
3	Refrigerant Pressure Drop(kpa)	49.4	21.9	21.8	13.1	20.8
		-	55.60%	55.87%	73.48%	57.42%
4	Refrigerant Heat Transfer Coefficient(w/m^{2}k)	290.13	482.37	476.43	494.01	542.41
		+	66.25%	64.21%	70.27%	86.95%
5	Copper Mass Consumption(kg)	3.4	2.3	2.3	2.3	2.3
		-	32.35%	32.35%	32.35%	32.35%
6	Aluminum Mass Consumption(kg)	1.5	1.6	1.6	1.6	1.6
		+	6.66%	6.66%	6.66%	6.66%

VII. CONCLUSION

In this project, the 7mm condenser with 2-1 inlet-outlet circuit of refrigerant R32 is improved the new condenser which had 5-5,6-6 inlet-outlet circuit and decreased the cooper tube diameter from 7.0 mm to 5.0 mm. The condenser is simulated by software EVAP-COND & ISHED. Final results are as follows:

- Compare to 7mm Layout 2 condenser, the heat exchange, refrigerant flow and heat exchange co-efficient of the ISHED layout of R32 finned-tube condenser are raised by 15.50%, 17% & 86.95% and pressure drop is reduced by 57.42%.
- Compare with the original heat exchanger, the condenser has a notable reduction in the consumption of copper, the consumption of copper condenser is reduced by 32.35% and aluminium consumption minor increased 6.6%.

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