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Improving the Heat Transfer Rate for Ac Condenser by Material and Parametric Design Optimization

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Abstract

Air conditioning systems have condenser that removes unwanted heat from the refrigerant and transfers that heat outdoors. The primary component of a condenser is typically the condenser coil, through which the refrigerant flows. Since, the AC condenser coil contains refrigerant that absorbs heat from the surrounding air, the refrigerant temperature must be higher than the air.

In my paper I have designed an air-cooled Condenser for an air conditioner. Presently the material used for coils is Copper and the material used for Fins is Copper or aluminum G Al Cu 4IMG 204 whose thermal conductivity is 110-150W/m K. A 3D model of the condenser is done in parametric software Pro/Engineer. To reduce the cost of condenser, we are optimizing the design parameters by changing the thickness of the fin for the same length without failing the load conditions. To validate the temperatures and other thermal quantities like flux and gradient, thermal analysis is done on the condenser by applying copper for coil and Fin materials G Al Cu 4IMG 204, Aluminum Alloy Al99 and Magnesium alloy. Thermal analysis is done in Cosmos works. And also we are varying inside cooling fluid Hydrocarbon (HC) and Hydrochloroflourocarbon (HCFC).The best material and best fluid for the condenser of our design can be checked by comparing the results.

1. Introduction

An air conditioner Fig.1 (often referred to as AC) is a home appliance, system, or mechanism designed to dehumidify and extract heat from an area. The cooling is done using a simple refrigeration cycle. In construction, a complete system of heating, ventilation and air conditioning is referred to as "HVAC". Its purpose, in a building or an automobile, is to provide comfort during either hot or cold weather.

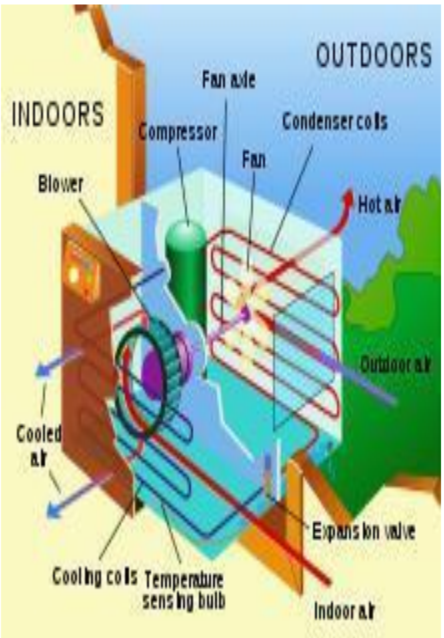

Fig.1. A typical home air conditioning unit.

1.1. Air Conditioning System

Fig.2 shows A simple stylized diagram of the refrigeration cycle: 1) condensing coil, 2) expansion valve, 3) evaporator coil, 4) compressor.

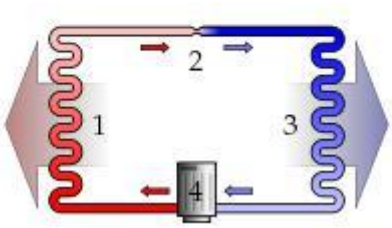

Fig.2. Refrigeration cycle

In the refrigeration cycle, a heat pump transfers heat from a lower-temperature heat source into a higher-temperature heat sink. Heat would naturally flow in the opposite direction. This is the most common type of air conditioning. A refrigerator works in much the same way, as it pumps the heat out of the interior and into the room in which it stands. his cycle takes advantage of the way phase changes work, where latent heat is released at a constant temperature during a liquid/gas phase change, and where varying the pressure of a pure substance also varies its condensation/boiling point.

By placing the condenser (where the heat is rejected) inside a compartment, and the evaporator (which absorbs heat) in the ambient environment (such as outside), or merely running a normal air conditioners refrigerant in the opposite direction, the overall effect is the opposite, and the compartment is heated. This is usually called a heat pump, and is capable of heating a home to comfortable temperatures (25 °C; 70 °F), even when the outside air is below the freezing point of water (0 °C; 32 °F).

1.2. Refrigerants

"Freon" is a trade name for a family of haloalkane refrigerants manufactured by DuPont and other companies. These refrigerants were commonly used due to their superior stability and safety properties. However, these chlorine-bearing refrigerants reach the upper atmosphere when they escape.[6] Once the refrigerant reaches the stratosphere, UV radiation from the Sun cleaves the chlorine-carbon bond, yielding a chlorine radical. These chlorine atoms catalyze the breakdown of ozone into diatomic oxygen, depleting the ozone layer that shields the Earth's surface from strong

UV radiation. Each chlorine radical remains active as a catalyst unless it binds with another chlorine radical, forming a stable molecule and breaking the chain reaction. The use of CFC as a refrigerant was once common, being used in the refrigerants R-11 and R-12. In most countries the manufacture and use of CFCs has been banned or severely restricted due to concerns about ozone depletion. In light of these environmental concerns, beginning on November 14, 1994, the Environmental Protection Agency has restricted the sale, possession and use of refrigerant to only licensed technicians, per Rules 608 and 609 of the EPA rules and regulations; failure to comply may result in criminal and civil sanctions. Newer and more environmentally-safe refrigerants such as HCFCs (R-22, used in most homes today) and HFCs (R-134a, used in most cars) have replaced most CFC use. HCFCs in turn are being phased out under the Montreal Protocol and replaced by hydro fluorocarbons (HFCs) such as R-410A, which lack chlorine. Carbon dioxide (R-744) is being rapidly adopted as a refrigerant in Europe and Japan. R-744 is an effective refrigerant with a global warming potential of 1.

1.3. Equipment capacity

Air conditioner equipment power in the U.S. is often described in terms of "tons of refrigeration". A "ton of refrigeration" is approximately equal to the cooling power of one short ton (2000 pounds or 907 kilograms) of ice melting in a 24-hour period. The value is defined as 12,000 BTU per hour, or 3517 watts. Residential central air systems are usually from 1 to 5 tons (3 to 20 kilowatts (kW)) in capacity.

Unit size, BTU/h × hours per year, h × power cost, \$/kW·h ÷ (SEER, BTU/W·h × 1000 W/kW)

(72,000 BTU/h) × (1000 h) × (\$0.08/kW·h) ÷ [(10 BTU/W·h) × (1000 W/kW)] = \$576.00 annual cost

A common misconception is that the SEER rating system also applies to heating systems. However, SEER ratings only apply to air conditioning.

Air conditioners (for cooling) and heat pumps (for heating) both work similarly in that heat is transferred or "pumped" from a cooler heat source to a warmer "heat sink". Air conditioners and heat pumps usually operate most effectively at temperatures around 10 to 13 degrees Celsius (°C) (50 to 55 degrees Fahrenheit (°F)). A balance point is reached when the heat source temperature falls below about 4 °C (40 °F), and the system is not able to pull any more heat from the heat source (this point varies from heat pump to heat pump).

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Similarly, when the heat sink temperature rises to about 49 °C (120 °F), the system will operate less effectively, and will not be able to "push" out any more heat. Geothermal heat pumps do not have this problem of reaching a balance point because they use the ground as a heat source/heat sink and the ground's thermal inertia prevents it from becoming too cold or too warm when moving heat from or to it. The ground's temperature does not vary nearly as much over a year as that of the air above it.

1.4. Introduction to Condenser

A condenser or evaporator is a heat exchanger, allowing condensation, by means of giving off, or taking in heat respectively.

The construction principle:

Refrigerant and air will be physically separated, at air conditioner condenser, and evaporator. Therefore, heat transfer occurs by means of conduction.

We would like the heat exchanger that enables these processes, to have,

- High conductivity– this property will ensure that the low temperature difference between the outside wall, and inside wall
- High contact factor– this property ensures the passing air mass, will come in contact with the tubes, as much as possible

1.5. Specifications of condenser

The length and size of air conditioner condensers and evaporators have to be sized such that,

- the refrigerant is completely condensed before the condenser's exit, and
- the refrigerant is completely boiled before the evaporator's exit

2. State of the Art

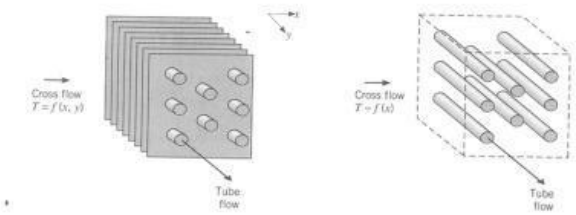

Fig.3.Typical Plate Fin-and-Tube Cross Flow Heat Exchange

Wang et al. (1999) conducted an experimental study on the air-side performance for two specific louver fin patterns and their plain plate fin counterparts. This study investigated the effects of fin pitch, longitudinal tube spacing and tube diameter on the air-side heat transfer performance and friction characteristics. This study found that for plain plate fin configurations ranging from 8 to 14 fins per inch, the effect of longitudinal tube pitch on the air-side was negligible for both the air-side heat transfer and pressure drop. However, the heat transfer performance increased with reduced fin pitch [1].

Chi et al. (1998) conducted an experimental investigation of the heat transfer and friction characteristics of plate fin-and tube heat exchangers having 7 mm diameter tubes. In this study, 8 samples of commercially available plate-fin-and-tube heat exchangers were tested. It was found that the effect of varying fin pitch on the air-side heat transfer performance and friction characteristics was negligible for 4-row coils. However for 2-row coils, the heat transfer performance increased with a decrease in fin pitch. This study used a plate-fin-and tube heat exchanger configuration with louver fin surfaces, which are widely used in both automotive and residential air-conditioning systems. The transverse fin spacing ranged from 21 mm to 25.4 mm and longitudinal fin spacing ranged from12.7 mm to 19.05 mm [2].

Wang et al. (1998) also collected experimental data on a plate-fin-and tube heat exchanger configuration. They examined the effect of the number of tube rows, fin pitch, tube spacing, and tube diameter on heat transfer and friction characteristics. This study found that the effect of fin pitch on the air-side friction pressure drop was negligibly small for air-side Reynolds numbers greater than 1000. It was also found that the heat transfer performance was independent of fin pitch for 4-row configurations. Furthermore, the results indicated that reducing the tube spacing and the tube diameter produced an increase in the air-side heat transfer coefficient. The fin surfaces utilized in this study were of the louver type, with transverse fin spacing ranging from 21 mm to 25.4 mm, and longitudinal fin spacing ranging from 12.7 mm to 19.05 mm. The longitudinal tube spacing investigated for this studied ranged from 15 mm to 19 mm and the tube diameters ranged from 7.94 mm to 9.52 mm [3].

One of the earliest and most complete investigations of heat exchanger heat transfer and pressure drop characteristics was performed by Kays and London (1984). An extensive amount of experimental heat transfer and friction pressure drop data were compiled for several different plate-fin-and-tube heat exchanger configurations as part of this study. However, no optimization of the heat transfer surfaces and geometry was performing [4].

Shepherd (1956) experimentally tested the effect of various geometric variations on1-row plate fin-and-tube coils. He investigated the effects of varying the fin spacing, fin depth, tube spacing, and tube location on the heat transfer performance of the coil. The results of Shepherd's study showed that as the fin pitch increased, the air-side heat transfer coefficient, for a given face velocity, increased only slightly. He also found that as the fin depth and tube spacing increased, with all other variables constant, the air-side heat transfer coefficient decreased [5].

Rich (1973) studied the effect of varying the finspacing on the heat transfer and friction performance of multi-row heat exchanger coils 8 Rich found that over the range from 3 to 14 fins per inch, the air-side heat transfer coefficient was independent of fin pitch[6].

Neither Rich's nor Shepherd's investigations involved the optimization of the heat exchanger operating conditions and geometric parameters. All of the above studies provide valuable insight into the effects of varying different geometric parameters on the heat transfer and friction performance of plate-fin-tube heat exchangers.

In my paper, I have designed an air-cooled Condenser for an air conditioner by changing the fin material and also optimisation by changing thickness of fin.

3. CONTACT FACTOR

It is the amount of media that needs to be heated up or cooled down, that comes directly in contact with the tube walls. Contact factor will be very low, if the air inside a duct is passed through a straight tube with refrigerant. This happens as the amount of air that contacts the tube will be very low. Therefore, we will increase the contact factor, by constructing the condenser and evaporator to have many passes within a given duct area. Thus, the passing air will "see" a lot of tubes on its passage. Hence the contact factor will be improved the maximum theoretical contact factor is 100%. We will have contact factors around 80% for commercially produced air conditioner evaporators and air conditioner condensers. The real figures really depend on each manufacturer. The reciprocal of the contact factor, is the bypass factor, where it is equal to *1 – contact factor*.

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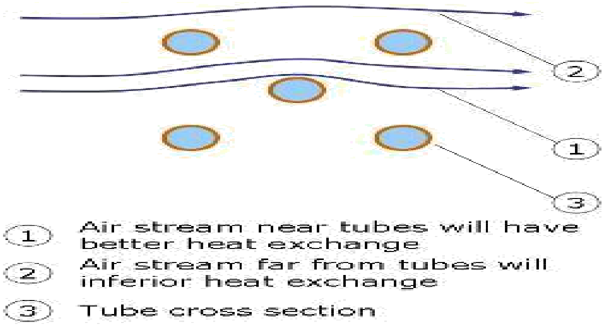

3.1. Cooling Load Calculations

Cooling load calculations for air conditioning system design are mainly used to determine the volume flow rate of the air system as well as the coil and refrigeration load of the equipment to size the HVAC&R equipment and to provide the inputs to the system for energy use calculations in order to select optimal design alternatives. Cooling load usually can be classified into two categories: external and internal

3.1.1 External Cooling Loads.

These loads are formed because of heat gains in the conditioned space from external sources through the building envelope or building shell and the partition walls.

Sources of external loads include the following cooling loads:
1. Heat gain entering from the exterior walls and roofs
2. Solar heat gain transmitted through the fenestrations
3. Conductive heat gain coming through the fenestrations
4. Heat gain entering from the partition walls and interior doors
5. Infiltration of outdoor air into the conditioned space

3.1.2. Internal Cooling Loads.

These loads are formed by the release of sensible and latent heat from the heat sources inside the conditioned space. These sources contribute internal cooling loads:

1. People
2. Electric lights
3. Equipment and appliances

If moisture transfers from the building structures and the furnishings are excluded, only infiltrated air, occupants, equipment, and appliances have both sensible and latent cooling loads. The remaining components have only sensible cooling loads. All sensible heat gains entering the conditioned space represent radiative heat and convective heat except the infiltrated air, radiative heat causes heat storage in the building structures, converts part of the heat gain into cooling load, and makes the cooling load calculation more complicated. Latent heat gains are heat gains from moisture transfer from the occupants, equipment, appliances, or infiltrated air. If the storage effect of the moisture is ignored, all release heat to the space air instantaneously and, therefore, they are instantaneous cooling loads.

3.1.3. External cooling load calculations

Roof&wall:

$$Q = U * A * (CLTD)$$

Solar load through glass:

Conductive:

$$Q_{Glass\ Conductive} = U * A * CLTD_{Glass\ Corrected}$$

Solar Transmission $Q_{Glass\ Solar} = A * SC * SCL$

Partitions ceilings and floors:

$$Q = U\ A\ (Ta - Trc)$$

3.1.4. Internal Cooling Loads

People

$$Q_{sensible} = N\ (Q_S)\ (CLF)$$

$$Q_{latent} = N\ (Q_L)$$

Lights

$$Q = 3.41\ x\ W\ x\ F_{UT}\ x\ F_{SA}\ x\ (CLF)$$

Power Loads

If the motor and the machine are in the room the heat transferred can be calculated as

$$Q = 2545 * (P / Eff) * F_{UM} * F_{LM}$$

Appliances

$$Q_{Sensible} = Q_{in}\ x\ F_u\ x\ F_r\ x\ (CLF)$$

$$Q_{Latent} = Q_{in}\ x\ F_u$$

Infiltration Air

$$Q_{sensible} = 1.08\ x\ CFM\ x\ (To - Ti)$$

$$Q_{latent} = 4840\ x\ CFM\ x\ (Wo - Wi)$$

$$Q_{total} = 4.5\ x\ CFM\ x\ (ho - hi)$$

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Heat Gain from Miscellaneous Sources Supply Fan Heat Load

$Q = 2545 \text{ x } [P / (Eff_1 \text{ x } Eff_2)]$

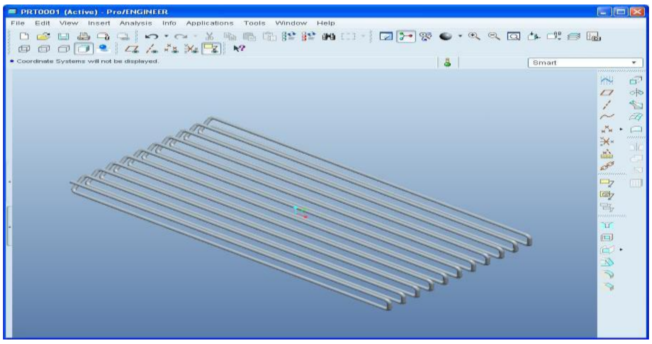
Ventilation Air

$Q_{sensible} = 1.08 \text{ x } CFM \text{ x } (T_o - T_c)$

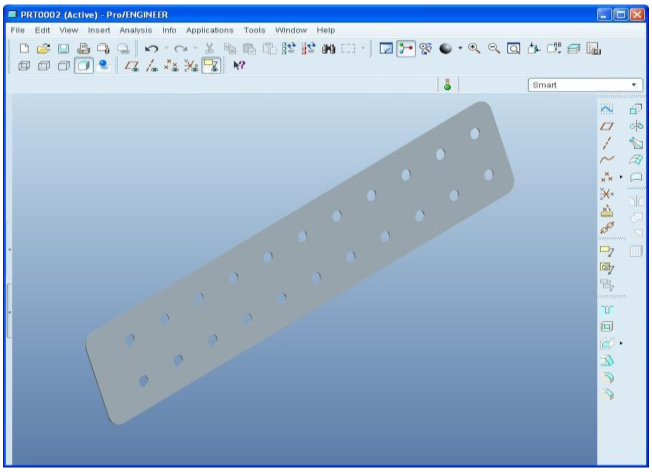
$Q_{latent} = 4840 \text{ x } CFM \text{ x } (W_o - W_c)$

$Q_{total} = 4.5 \text{ x } CFM \text{ x } (h_o - h_c)$

3.2. MODEL OF CONDENSER TUBE



PLATE



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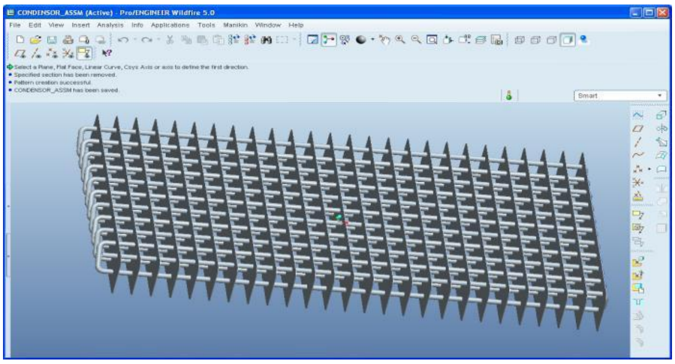
ASSEMBLY



3.2.1. Thermal analysis of condenser

COSMOS Works is a design analysis automation application fully integrated with Solid Works.

This software uses the Finite Element Method (FEM) to simulate the working conditions of your designs and predict their behaviour. FEM requires the solution of large systems of equations. Powered by fast solvers, COSMOS Works makes it possible for designers to quickly check the integrity of their designs and search for the optimum solution.

A product development cycle typically includes the following steps:

1 Build your model in the Solid Works CAD system.

2 Prototype the design.

3 Test the prototype in the field.

4 Evaluate the results of the field tests.

5 Modify the design based on the field test results

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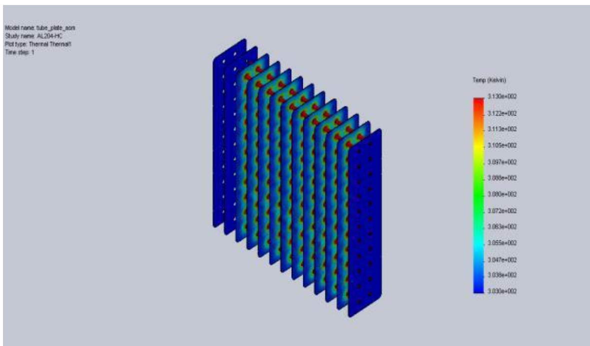

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Fig.4. Temperature contours of Al204 with HC working fluid

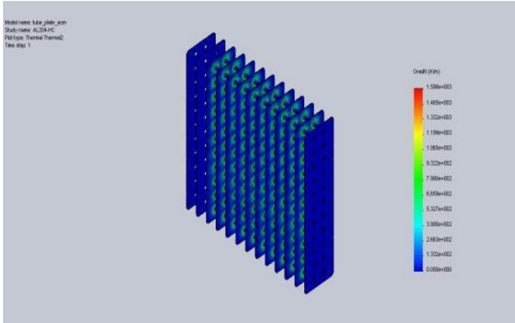

Fig.5. Temperature gradient contours of Al204 with HC working fluid

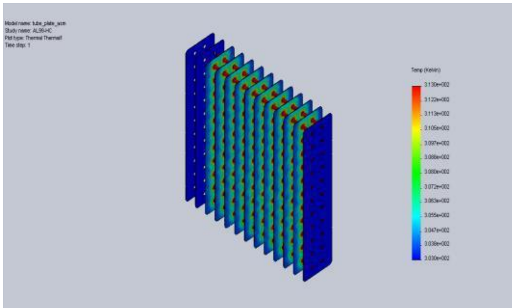

Fig.6. Thermal flux contours of Al204 with HC working
fluid

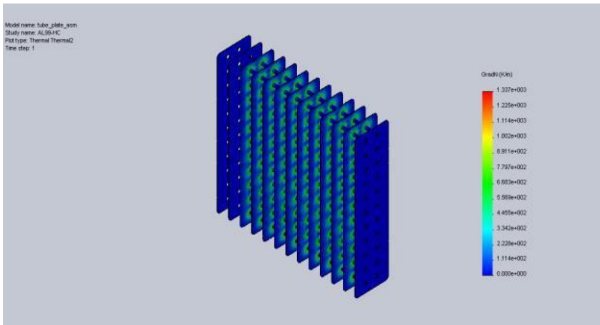

Fig.7. Temperature contours of Al99 with HC working fluid

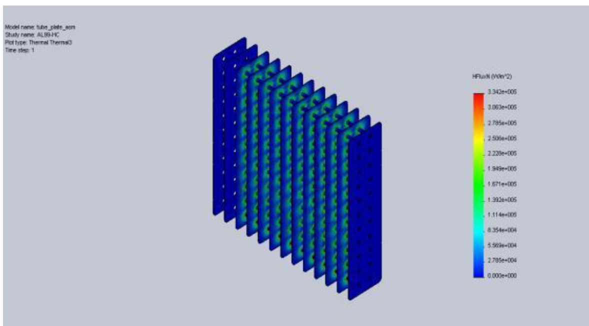

Fig.8. Temperature gradient contours of Al99 with HC working fluid

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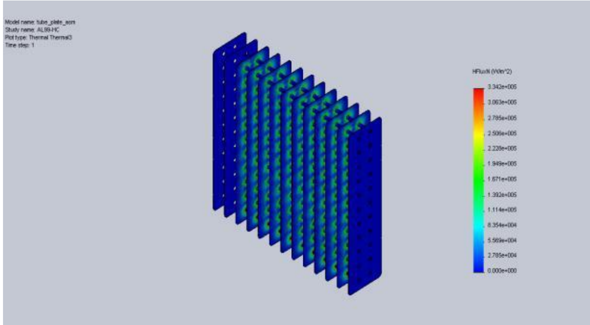

Fig.9. Thermal flux contours of Al99 with HC working fluid

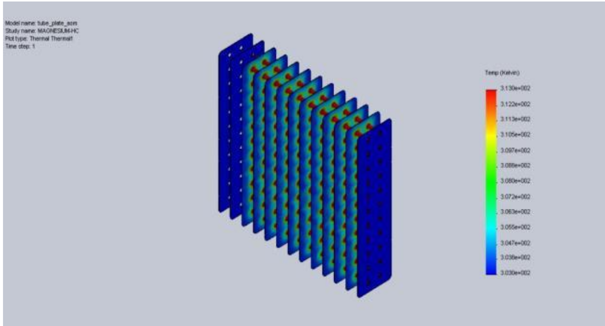

Fig.10.Temperature contours of Mg with HC working fluid

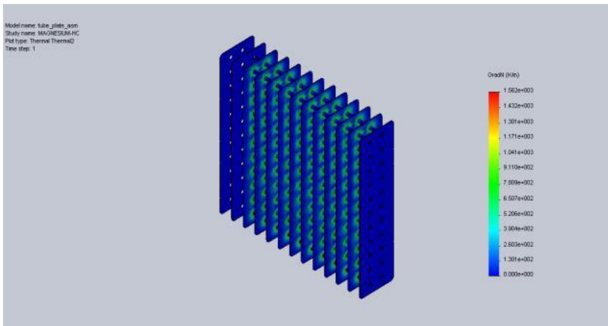

Fig.11. Temperature gradient contours of Mg with HC working fluid

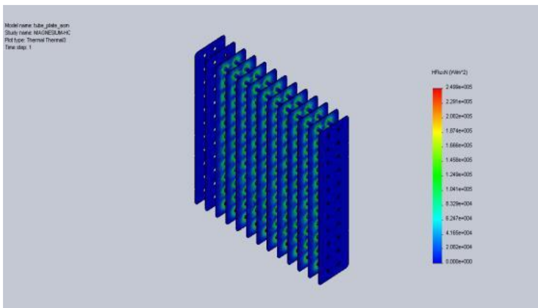

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Fig.12. Thermal flux contours of Mg with HC working fluid

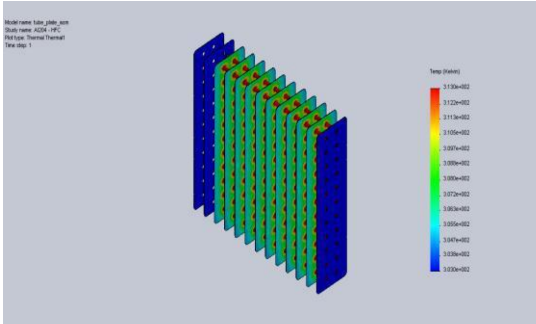

Fig.13. Temperature contours of Al204 with HFC working fluid

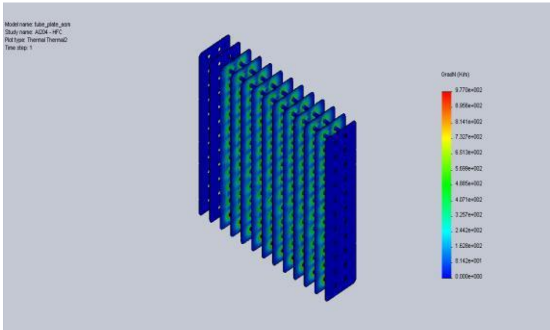

Fig.14. Temperature gradient contours of Al204 with HFC working fluid

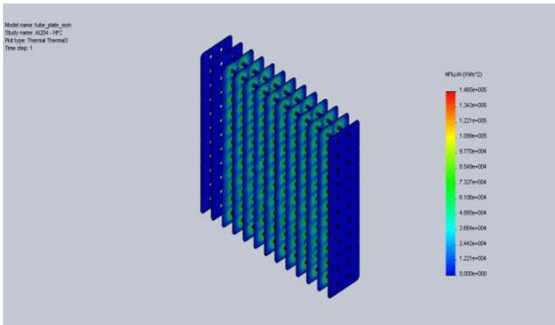

Fig.15. Thermal flux contours of Al204 with HFC working fluid

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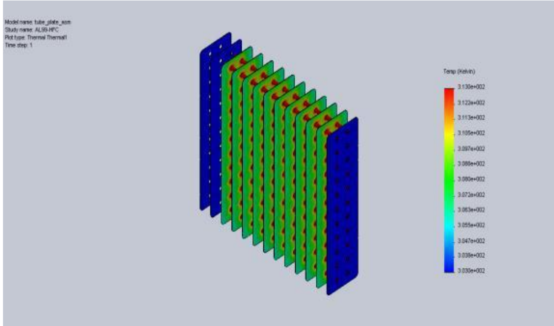

Fig.16. Temperature contours of Al99 with HFC working fluid

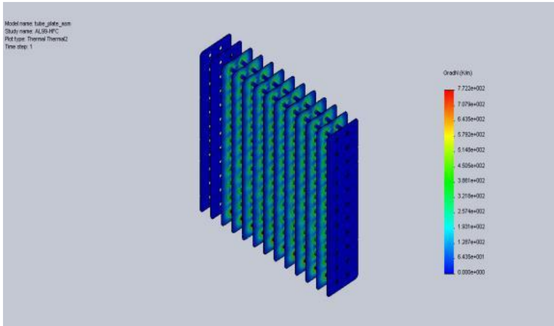

Fig.17. emperature gradient contours of Al99 with HFC working fluid

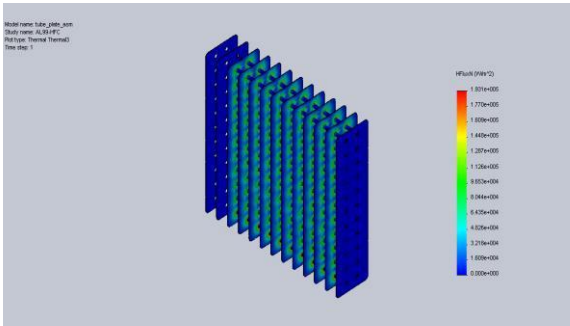

Fig.18. Thermal flux contours of Al99 with HFC working fluid

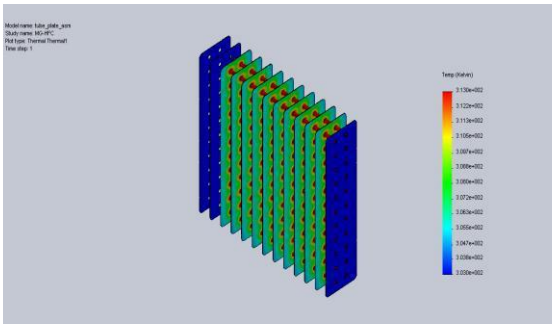

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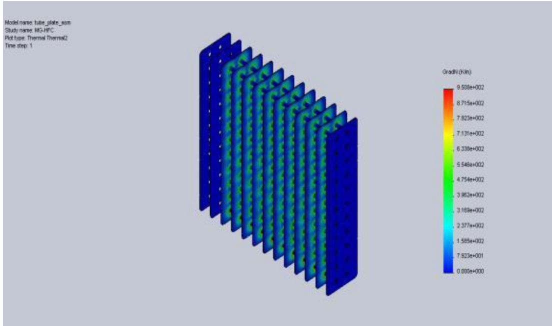
Fig.19. Temperature contours of Mg with HFC working fluid



	AL2 04	AL9 9	Mg	AL2 04	AL9 9	Mg
TEM P(K)	313	313	313	313	313	313
TG(K/ m)	1598 .05	1336 .62	1561 .73	976. 98	772. 238	950.7 68
TF (W/m^2 K)	2397 08	3341 55	2498 77	1465 47	1930 59	1521 23

Fig.20. Temperature gradient contours of Mg with HFC working fluid

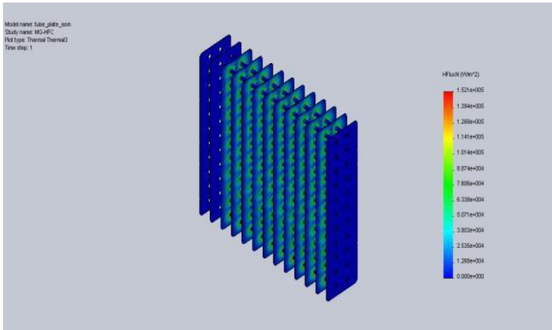


Fig.21. Thermal flux contours of Mg with HFC working fluid

3.3. Optimization of condenser by Changing Thickness

I have optimized the condenser to reduce the volume, by changing the thickness of the fin. By observing optimization results, the optimum thickness value for decreased volume is 1mm.

4. Results and Discussions

THERMAL ANALYSIS RESULT TABLE

	HYDROCARBON	HYDROFLUOROC ARBON

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OPTIMIZATION RESULT TABLE

	HYDROCARBON			**HYDROFLUOROCARBON**		
	AL204	**AL99**	**Mg**	**AL 204**	**AL99**	**Mg**
THICKNESS (mm)	1	1	1	1	1	1
TG(K/m)	1598.05	1336.62	1561.73	976.98	772.238	950.768
TF (W/m^2 K)	239711	334165	249888	146555	193059	152123
VOLUME (m^3)	1158076	1158076	1158076	115807	1158076	1158076

The result obtained from thermal analysis using cosmos
Shown in figures 4 to 21 illustrates that Al99 have the more thermal flux compared to other two materials

5. Conclusion

In this paper I have designed an air conditioner condenser. Present used material for fin is Aluminium alloy 204 and cooling fluid is HCFC.I has modelled the condenser in 3D modelling software Pro/Engineer. To optimize the condenser for best result, thermal analysis is done on the condenser. Analysis is done using fin materials Aluminium Alloy 204, Aluminium Alloy 99 and Magnesium alloy. And also by changing the cooling fluid HCFC and Hydrocarbon.

By observing the thermal analysis results, by using fin material Aluminium alloy Al99, thermal flux is more than by using other two materials. So by using Aluminium alloy A99, the heat transfer rate

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increases. I have optimized the condenser to reduce the volume, by changing the thickness of the fin. By observing optimization results, the optimum thickness value for decreased volume is 1mm.So I conclude that using condenser with fin thickness of 1mm and fin material Aluminium alloy Al99 gives better result.

6. Nomenclature

• Q = cooling load, Btu/hr

• U = Coefficient of heat transfer, Btu/hr.ft².°F

• A = area, ft^2

• CLTD = cooling load temperature difference °F.
The values are determined from tables' available AHSRAE fundamentals handbook

• SC = Shading coefficient

• SCL = Solar Cooling Load Factor

• Ta = Temperature of adjacent space in °F (Note:
If adjacent space is not conditioned and temperature is not available, use outdoor air temperature less 5°F)

• Trc = Inside design temperature of conditioned space in °F (assumed constant)

• N = number of people in space from ASHRAE,

• Q_S, Q_L = Sensible and Latent heat gain from occupancy is given in ASHRAE Fundamentals

• CLF = Cooling Load Factor, by hour of occupancy. See ASHRAE Fundamentals

• W = Watts input from electrical lighting plan or lighting load data

• F_{UT} = Lighting use factor, as appropriate

• F_{SA} = special ballast allowance factor, as appropriate

P = Horsepower rating from electrical power plans or manufacturer's data

Eff = Equipment motor efficiency, as decimal fraction

F_{UM} = Motor use factor (normally = 1.0)

F_{LM} = Motor load factor (normally = 1.0)

• Qin = rated energy input from appliances. See ASHRAE Fundamentals,

• F_r = Radiation factor

• CFM = Infiltration air flow rate

- To, Ti = Outside/Inside dry bulb temperature, °F
- Wo, Wi = Outside/Inside humidity ratio, lb water/lb dry air
- ho, hi = Outside/Inside air enthalpy, Btu per lb (dry air)
- 2545 = conversion factor for converting horsepower to Btu per hour
- Eff_1 = Full load motor and drive efficiency
- Eff_2 = Fan static efficiency
- T_c = Dry bulb temperature of air leaving the cooling coil, °F
- W_c = Humidity ratio of air leaving the cooling coil, lb (water) per lb (dry air)
- h_c = Enthalpy of air leaving the cooling coil Btu per lb (dry air)

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