Design, Testing & Experimental Analysis of Hybrid Oil Cooler with Finned Tube Heat Exchanger And Heat Pipe

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Abstract- *Hydraulic circuits are most commonly used power sources in industry. The progress in recent years has offered high efficiency and reliable hydraulic component, yet hydraulic cooling circuit design is often neglected part of the development. One aspect of the hydraulic circuit design is prevention of overheating of hydraulic oil. Letting oil temperature rise beyond particular limit can increase the power consumption reduce the life of a system due to poor lubrication, higher internal leakage, and higher risk of cavitations and damage component. Keeping temperature down also help ensure the oil and other components last longer. Excess heat can degrade hydraulic oil, form harmful varnish on component surface and deteriorate rubber and elastomeric seal. Operating within recommended temperature ranges increases a hydraulic system's efficiency and helps in improving equipment productivity. Being efficient equipment it has more machine uptime and fewer shutdowns, which reduces periodic servicing and maintenance or repair costs. Considering the benefit of cooler offer, it's apparent that accurately sizing them is paramount concern for design engineers. Presently hydraulic oil cooler used is shell and tube type oil coolers with water as the cooling medium, this is extremely bulky and running cost is high hence it is needed to be replaced by another system that will be air cooled to make provision for lower space consumption and cost reduction. The proposed solution comes as finned tube turbine diffuser used in conjunction to a conventional radiator or a liquid to air cooler which is fan assisted. The combined system has been tested such that the turbine diffuser was used with three different sizes of heat pipe to dissipate the heat load for the diffuser part. The proposed system was thus developed as hybrid one with the downsized radiator used as the primary heat exchanger and the turbine diffuser heat exchanger as the secondary one. The project work included the test and trial on all three systems and results thus obtained were further compared and statistical analysis was done using Minitab software to validate the results using Anova technique.*

Keywords- hydraulic oil cooler, turbine diffuser, heat pipe, finned tube heat exchanger, combined assembly.

I. INTRODUCTION

It has been observed that overheating is main problem among all the hydraulic systems there are various reasons for overheating of hydraulic oils A hydraulic system's heat load is equal to the total power lost (PL) through inefficiencies and can be expressed as:

PL (total) =PL (pump) +PL (valves) +PL (plumbing) +PL(actuators)

With the high temperature operating conditions the oil may drop down its viscosity below optimum values for the system components for which they are designed depending upon the viscosity grade. There are two simple ways to overcome of overheating problems in hydraulic systems: either increase heat dissipation or decrease heat load. Few of the major causes of overheating are; undersized system piping, worn components, undersized reservoir etc. to overcome the excess heat, on precautionary basis, check oil level in reservoir if it is less fill it. Check for air flow around the reservoir tank in order to get the proper circulation of air. Check for dust, dirt or any external layers on walls of the heat exchanger because that may become barrier in heat transfer.

PROBLEM STATEMENT:

The present radiator system poses problems of overheating an increasing the capacity of cooling employing a larger radiator is not possible due to cost and space considerations.The secondary problem faced is that of maintenance. Due to scaling at high tempratures the radiator requires frequent cleaning which means more down time of machine. The overall the problem statement is that the earlier system is to be replaced by a more effective cooling system that will take lesser space and reduce the problem of overheating and frequent clogging of radiator.

SOLUTION:

The solution to the above problem is to develop an innovative method of cooling by employing a hybrid system. One system that will use heat pipes along with the conventional radiator system. The heat pipe extracts latent heat from the oil hence the heat transfer is multi folds that of the conventional methods. Hence the heat pipes are to be used. The problem of clogging occurs at elevated temperatures above 70 degrees hence by employing the heat pipe we ensure that temperature of oil entering the radiator system does not exceed 55 to 60 degrees thus the root cause of scaling and subsequent clogging of the radiators will be solved. More over as major part of the heat load is taken by the heat pipe system size of the radiator will fall by a considerable margin.

OBJECTIVES:

1. Selection of heat pipe system for given heat load (60%) and radiator system (40%) for the oil cooling application.
2. Thermal analysis of spiral radial fin heat sink in use for all three heat loads of three heat pipes to be used in setup.
3. Test and trial on system to determine the Heat transfer ability and overall heat transfer coefficient,
4. Statistical analysis and comparison of the three heat pipe system to propose the optimal system for given application.

II. LITERATURE REVIEW

[1] Fin-Tube Heat Exchanger Optimization, by Piotr Wais

In this paper authors have stated that Fins are commonly used in extended surface exchangers. Conventional fin-tube exchangers often characterize the considerable difference between liquids' heat transfer coefficients. In a gas-to-liquid exchanger, the heat transfer coefficient on the liquid side is generally one order of magnitude higher than that on the gas side. To minimize the size of heat exchangers, fins are used on the gas side to increase the surface area and the heat transfer rate between the heat exchanger surface and the surroundings. Both the conduction through the fin cross section and the convection over the fin surface area take place in and around the fin. When the fin is hotter than the fluid to which it is exposed then the fin surface temperature is generally lower than the base (primary surface) temperature. If the heat is transported by convection to the fin from the ambient fluid, the fin surface temperature will be higher than the fin base temperature, which in turn reduces the temperature differences and the heat transfer through the fin. Exchangers with fins are also used when one fluid stream is at high pressure. The temperature value is limited by the type of material and production technique. All above causes that finned tube heat exchangers are used in different thermal systems for applications where heat energy is exchanged between different media. Applications range from very large to the small scale (tubes in heat exchangers, the temperature control of electronic components). The subject, which is investigated in the chapter, is inspired by the increasing need for optimization in engineering applications, aiming to rationalize use of the available energy. The performance of the heat transfer process in a given heat exchanger is determined for different fin profiles, considering the fluid flow as a variability often neglected for the fin optimization. The optimization task, defined in the chapter, is to increase heat transfer rates and reduce the fin mass by means of changing the shape of the fin. The fin shape modification influences not only the mass of the heat exchanger, but also affects the flow direction that causes the temperature changes on the fin contact surfaces. The air flow is considered in all 3D models. The numerical outcome of heat transfer coefficient is compared to the results received from the empirical equation for the fin-tube heat exchanger of uniform fin thickness. The correlation function is cited and the procedure how to verify the models is described. For modified fin shapes, mass flow weighted average temperatures of air volume flow rate are calculated in the outlet section and compared for different fin/tube shapes in order to optimize the heat transfer between the fin material and the air during the air flow in the cross flow heat exchanger

[2] Review of Literature on Heat Transfer Enhancement in Compact Heat Exchangers byK. M. Stone

This paper features a broad discussion on the application of enhanced heat transfer surfaces to compact heat exchangers. The motivation for heat transfer enhancement is discussed, and the principles behind compact heat exchangers are summarized. Next, various methods for evaluating and comparing different types of heat transfer enhancement devices using first and/or second law analysis are presented. Finally, the following plate-fm enhancement geometries are discussed: rectangular and triangular plain fins, offset strip fins, louvered fins, and vortex generators

[3] Design, development and testing of cross flow circular heat pipe to cooling of oil by Vishnupant J. Sargar, Pruthveeraj D. Mali

Heat pipes are very flexible device to effective thermal control. It can easily be implemented for heat removing of hydraulic oil. In power pack application, hot oil is passed through four modules of the heat pipe. In heat pipe, heat is removed from oil and cold fluid is returned to the reservoir. The working fluid as water in the heat pipes respond to the application of heat by boiling, changing into a gaseous state, and transporting the heated gas to the pipe ends exposed to air. Ambient air is drawn over the exposed pipe by using blowers, causing the working fluid to condense and return to the heat source (the hot oil) to repeat the process. As long as there is a temperature difference between the oil and the ambient air, the heat pipes will remove heat from the oil, thus cooling is done. In the case of Hydraulic oil coolers- heat is removed from oil into the ambient air through heat pipes and fins. This paper reviews mainly heat pipe developments for cooling of oil for various applications like power pack, roasting pulp and chemical industries etc

[4] Heat pipes in modern heat exchangers Leonard L. Vasiliev

Heat pipes are very flexible systems with regard to effective thermal control. They can easily be implemented as heat exchangers inside sorption and vapour-compression heat pumps, refrigerators and other types of heat transfer devices. Their heat transfer coefficient in the evaporator and condenser zones is 103– 105 W/m2 K; heat pipe thermal resistance is 0.01–0.03 K/W, therefore leading to smaller area and mass of heat exchangers. Miniature and micro heat pipes are welcomed for electronic components cooling and space two-phase thermal control systems. Loop heat pipes, pulsating heat pipes and sorption heat pipes are the novelty for modern heat exchangers. Heat pipe air preheaters are used in thermal power plants to preheat the secondary–primary air required for combustion of fuel in the boiler using the energy available in exhaust gases.

Gap in Literature:

After careful study of literature, it is found that though various methods of reducing oil temperature by use of various devices have been put up in the literature. But the specific problem solving using a combination of heat pipes system in conjunction to the conventional radiator system is not yet discussed. There is a good chance of making one such system that will specifically solve the oil overheating problem by using the effective nature of the heat pipes and then also will be successful in reducing the size of the overall system Innovativeness & Usefulness

Findings from literature review:

1. Use of heat pipes to improve heat transfer ability
2. Use of Spiral radial fins to remove heat effectively from the condenser end of heat pipe which is in contact with air
3. Device is designed to combine the use of heat pipe system and conventional radiator system
4. Reduced size as compared to earlier system
5. Lower weight and compactness.

III. DESIGN AND DEVLOPMENT OF EXPERIMENTAL SYSTEM

Description of Concept:

As discussed earlier the net heat load on the cooling set-up is divided into two parts and thus two separate cooling mechanisms will be used in series, namely the Turbine diffuser heat pipe heat exchanger and the finned tube heat exchanger or the radiator

A) **Turbine diffuser heat pipe heat exchanger with spiral radial fins**

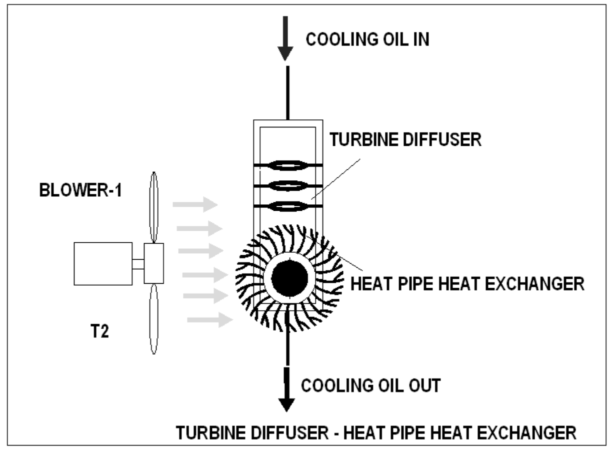


Figure. No. 1 composite assembly of heat pipe

In this set-up the hot stream of oil from the gear pump is directed onto a series of pivoted disks that act like turbine. Impingement of the jets on the disk will make them to rotate and thereby disperse the jet into fine droplets, thus heat is dispersed and the inner wall of the diffuser will absorb this heat. The turbine diffuser carries a cylindrical flat heat pipe at the base. The Evaporator end of the heat pipe is embedded into the diffuser and the condenser end is fitted with spiral radial fins. Thus heat from the diffuser chamber is carried by the heat pipe to the fins that dissipate it to the forced stream of air by the blower-1. Thus substantial part of the heat from the oil is removed in this first stage of the cooling system

The above system dissipates maximum heat in minimum space and can be oriented just behind the radiator unit. The 12 volt DC blower helps the forced convective heat transfer over the spiral radial fins that offer the maximum surface area in minimum space. The relatively cold oil is now directed to second stage of cooling ie, the finned tube heat exchanger (radiator).

B) Radiator System

The finned tube heat exchanger or the radiator is the second stage cooler of the system. The hot oil from the diffuser setup enters the radiator at one end, passes through a series of flat finned tubes arranged inside the radiator. Forced cooling is achieved using a 12 volt DC blower. In order to test and trial the above setup is arranged as a test rig where in the gear pump is used for circulation of the oil. The oil tank with heater (230 volt immersion type) is provided with a thermostat controller to maintain the oil temperature in given range. The temperature measurement is done at three locations as indicated in the schematic where as flow measurement is done using a measuring beaker and stop watch.

SPECIFICATIONS

- Heat pipes used Diameter 20 mm , 25 mm & 32 mm
- Material of heat pipe = Copper
- Working fluid Water
- Wick structure is sintered copper
- Material of Spiral radial fins = Aluminium
- Heater = 1000 watt

The heat generation amount in the customer's machine is known, ie the power consumption of the machine

2 hp = 1500 watt

The heat generation amount can be determined based on the power consumption or output of the heat generating area i.e. the area requiring cooling within customer's machine. Output (shaft power, etc)

W: 1500[W]
Heat generated (Q) = P- (W X η)

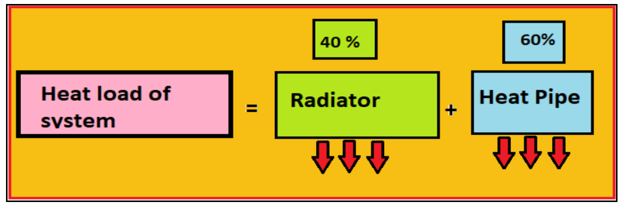
In our case we shall use an efficiency of 0.7 , assuming that 30 % of the energy supplied to the system is converted to heat due to in efficiency of the machine resulting into heat generated by the system which will further go to heat up the oil and result into overheating of the system

Q = 1500 **-** (1500 x 0.7)
= 1500 **-**1050 = 450 watt
Q = 450 watt

Considering 20% factor of safety Cooling capacity of the device required to maintain the safe limit of oil temperature

Q $_{act}$ = Q x 1.2 = 450 x 1.2
Q $_{act}$ = 540 watt

As discussed in the problem definition section we shall reiterate that the net heat load will be shared by two systems



Thus heat load shared by the individual system will be given as Heat load for Radiator System

Q $_{act}$ X 0.4 = 540 X 0.4 = 216 watt

Q –Radiator System = 216 watt

Heat load for Heat Pipe System
= Q $_{act}$ X 0.6 = 540 X 0.6 = 324 watt

Q –Heat pipe System = 324watt

Problem for Heat Pipe System
Heat load on the heat pie system = **324 watt**
Temperature increase deg =

$$\frac{\text{heat generation rate (watt/min)}}{\text{(oil specific heat (kj/kgK) X (oil flow rate (lb/min))}}$$

Specific heat of oil =**2.34 (kj/kgK)**

Maximum Allowable Temperature rise = **20 °C ----** target value

Flow rate through system = 324 / 2.34 x 10^3 x 20 = **0.007 Kg/sec**

Flow rate through system = 0.415 kg/min = **0.52 lpm**

Thus using reverse engineering considering
mass flow rate of 1 lpm ,

the minimum capacity required by the het pipe system will be:
Wattage of Heat pipe = (0.824 /60) *2340 *20 = 642 **watt**
Factor of Safety = 642/324 =**1.928**

Thus the heat pipe system selected is having a factor of safety =**2**

Thus to be on a safer side and still achieve good results with given system of input we shall assume that a heat pipe capable to transfer 200 watt and above will be suitable for the application.

Table. No. 1 Heat Transfer Capability for selected Heat Pipe

DIAMETER	20°C	30°C
20 mm	210 WATT	240 watt
25 mm	227 watt	246 watt
32 mm	235 watt	254

- The power handling figures are for heat pipe working in horizontal position.
- Length 75 mm long
- evaporator length 26mm
- condenser length 24mm
- adiabatic length =20mm
- Sintered copper powder

Temperature Sensors

-PT-1OO, ø6 mm, 2m LENGTH, TEMPERATURE sensing element

-Display is multi channel temperature indicator

Pipe size	20mm	25mm	32mm
Length	75 mm	75 mm	75 mm
Wattage	240 watt	246 watt	254 watt
Wick	sintered copper	sintered copper	sintered copper
Working fluid	Water	Water	Water

Figure. No. 2 Heat sink

Measurement Mass Properties

Material : Aluminium

Displayed Mass Property Values

Volume = 45749.346346060 mm^3

Area = 106970.325514622 mm^2

Mass = 0.121693261 kg

Weight = 1.193404297 N

Radius of Gyration = 31.397539329 mm

Centroid = -0.000855740 -0.001402973 13.000000000 mm

Detailed Mass Properties

Analysis calculated using accuracy of 0.990000000

Information Units kg – mm

Density = 0.000002660

Volume = 45749.346346060

Area = 106970.325514622

Mass = 0.121693261

Figure No. 3 experimental set up

IV. THERMAL ANALYSIS

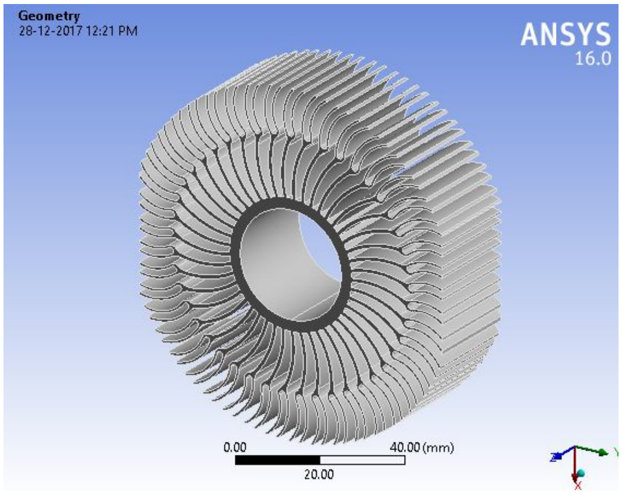


Figure 4 3D Geometry of Spiral Radial Fins for 20 mm heat pipe

Solid model of the fin structure and heat exchanger was developed using Unigraphix Nx-8 software, using the Cylinder, block, Pattern, Boolean operations, etc, commands. Once the model was developed the drafting was done to make the working drawings and then step203 file was developed using the export command. This exchange file was used to generate the geometry in ANSYS software modeler by use of Import external geometry file command. One the geometry was imported the Generate command was used to transfer geometry to the solver model section

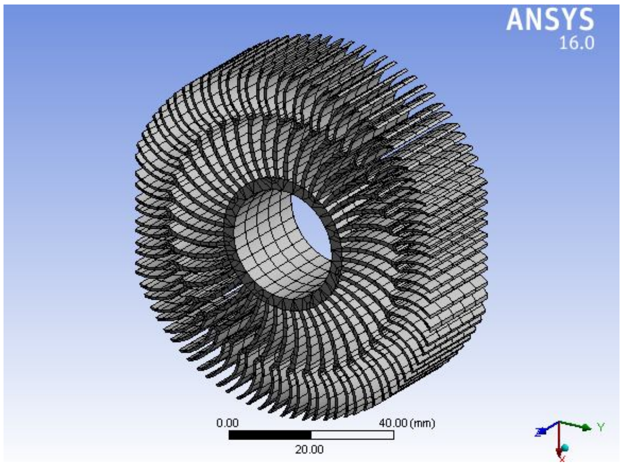


Figure 5 Nodes and element in Spiral Radial Fins for 20 mm heat pipe

Ansys Workbench free mesher was used to generate the mesh, first the meshing was done and then the generate command was used to apply it, details of the mesh parameter set by the workbench have been displayed in the table below:

Statistics	
Nodes	48441
Elements	6920
Mesh Metric	None

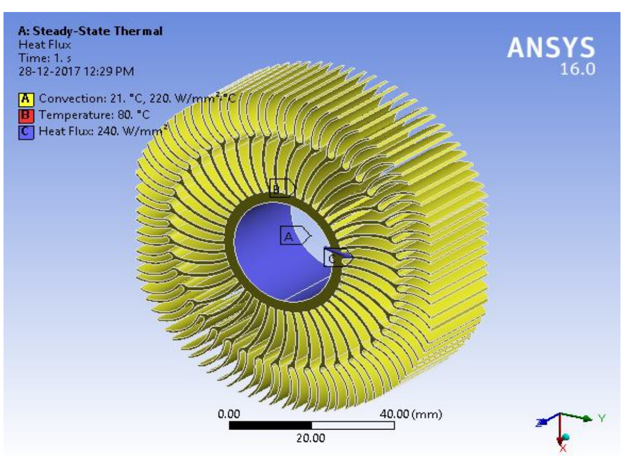


Figure 6 Details of steady state thermal distribution for Spiral Radial Fins for 20 mm heat pipe

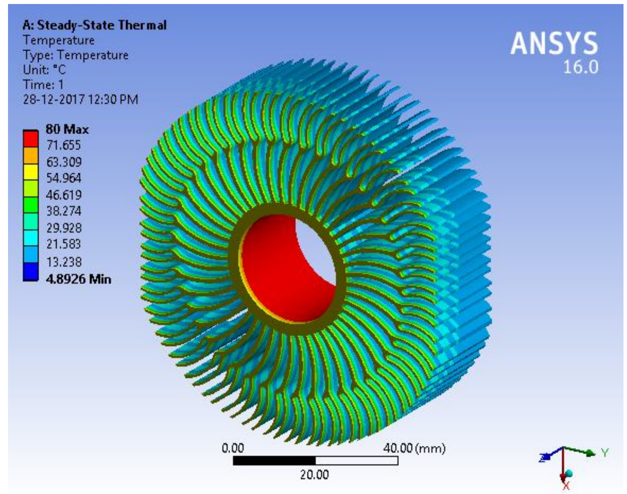


Figure 4.4 Temperature Variation Contour

Temperature variation contour is obtained when given boundary condition are applied to the problem. The results obtained from ANSYS software simulation for temperature contour is shown in figure . From figure we can say that as fluid entered into aluminium block firstly it comes in contact with inner member therefore maximum temperature exists there. After that it is entered the fin section where heat is dissipated.

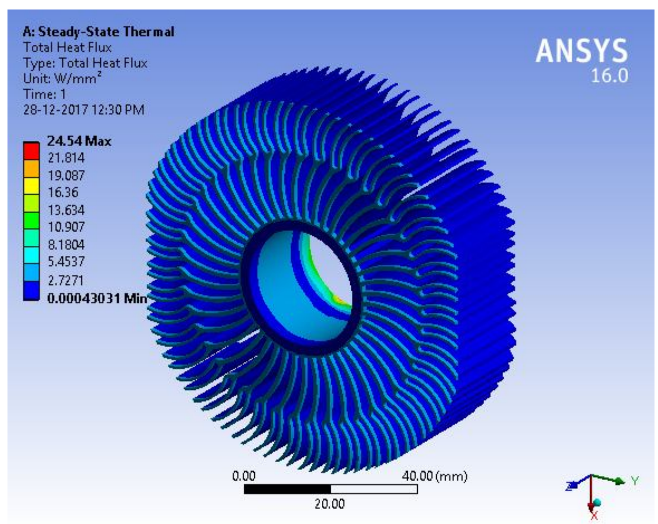


Figure 4.5 The heat flux carried by the fins

The figure above gives the heat flux carried by the fin system which is close to 24.54 watt very close to the required heat dissipation ability of the system

V. TEST INVESTIGATION

Test & Trial On 20 mm diameter heat pipe
Mass Flow Rate of Hot Oil
Mass Flow Rate of Oil
Table no 3 Mass Flow Rate of Oil

Sr.No	Volume in Beaker (ml)	Time (Sec)	Mass Flow Rate (Kg/sec)
1	50	38	0.001201
2	50	32	0.001427
3	50	27	0.001691
4	50	23	0.001985
5	50	19	0.002403
6	50	15	0.003043

Table no 4 Temperature Readings of Oil and Air

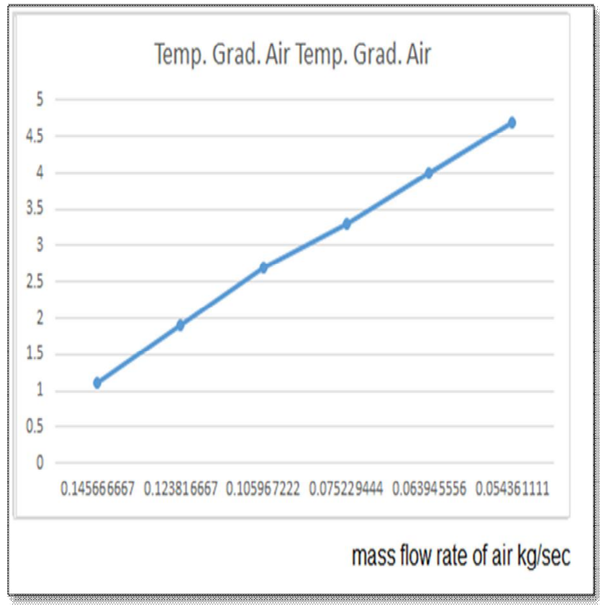
Sr. No.	Cold air Inlet Temp. (T_{ci}) °C	Cold air outlet Temp(T_{co}) °C	ΔTC Air °C	Hot oil Inlet Temp. (T_{hi}) °C	Hot oil outlet Temp.(T_{ho}) °C	ΔTH OIL °C
1	28.5	29.6	1.1	80	63	17
2	28.5	30.4	1.9	80	60.5	19.5
3	28.5	31.2	2.7	80	58.2	21.8
4	28.5	31.8	3.3	80	56.4	23.6
5	28.5	32.5	4	80	52.1	27.9
6.	28.5	33.2	4.7	80	50	30

Table no 5. Result Table with Various performance Parameters

SR.NO	mCpΔT (Hot Oil) KW	mCpΔT (cold Air) KW	U =W/ m²k
1	0.027485	0.58558	49.08054
2	0.037808	0.65950948	50.95359
3	0.051244	0.60703323	64.21546
4	0.070155	0.48387578	78.2979
5	0.091362	0.43700393	95.96875
6	0.120719	0.40974687	114.9704

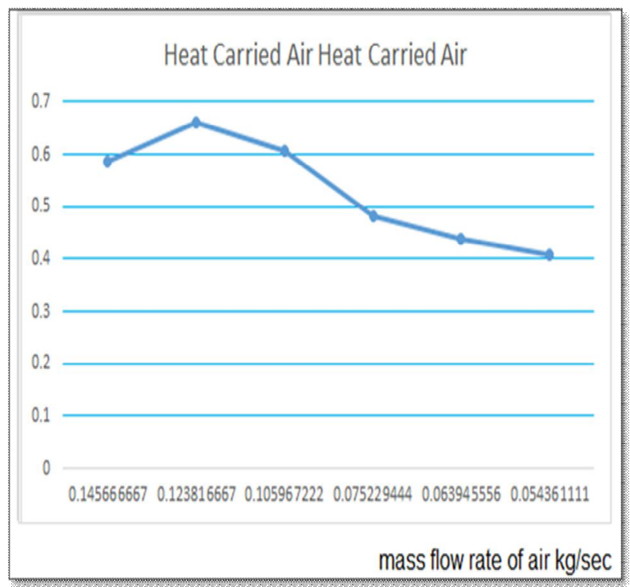
GRAPHS :

Temprature gradientof air vs mass flow rate of air



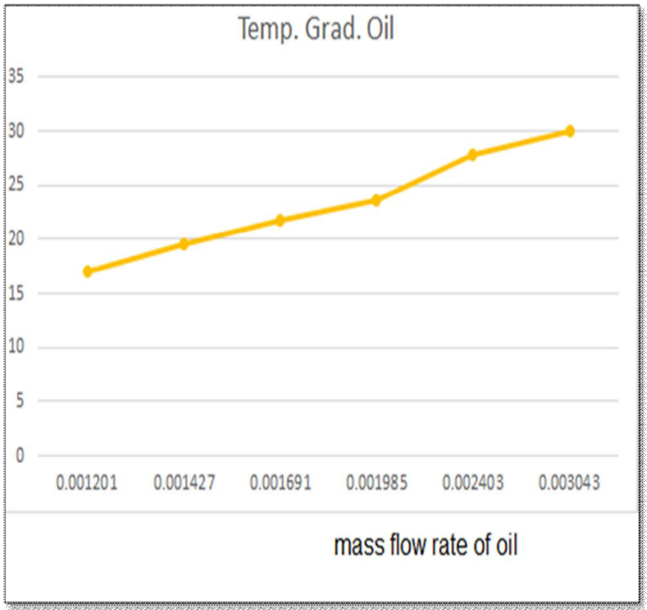
Graph No 1 Temperature gradient of air increases with increase in mass flow rate of oil

Heat transfered to air vs mass flow rate of air



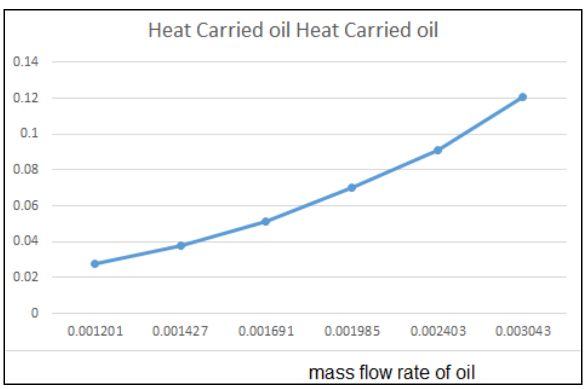
Graph No 2 Heat transferred to air increases to a certain limit and then it drops hence the optimal mass flow rate is around 0.125 kg/sec

Temprature gradient of oil vs flow rate oil



Graph No 3 Temperature gradient of air increases with increase in mass flow rate of oil

Heat transfered to oil vs mass flow rate of oil



Graph No 4 Heat carried by oil increases with increase in mass flow rate of oil

Optimization of Results for 20, 25, 32 mm diameter heat pipe using Mini Tab by Taguchi method for Design of Experiment: Document of Analysis

Selection of Process Parameters

Based on the experimental results discussed above important parameters have been selected to analyses their effect size using Taguchi's design of experiment technique. In present work, three input parameters namely Size, Mass flow rate of oil, Mass flow rate of air have been investigated.

Table No 6 Process variables and their levels

Parameter	Level-1	Level-2	Level-3
Size	20	25	32
Mass flow rate of oil	13.2	13.6	13.85
Mass flow rate of Air	5.6	6	6.42

Selection of Orthogonal Array and Parameter Assignment:

The orthogonal array forms the basis for the experimental analysis in Taguchi method. The selection of orthogonal array is concerned with the total degree of freedom of process parameters. The degree of freedom for the orthogonal array should be greater than or at least equal to that of the process parameters. Thereby, a L9 orthogonal array having degrees of freedom equal to 8 is considered in present case.

Taguchi Analysis: Overall HTC versus Size, M_oil, M_air
Response Table for Signal to Noise Ratios Larger is better

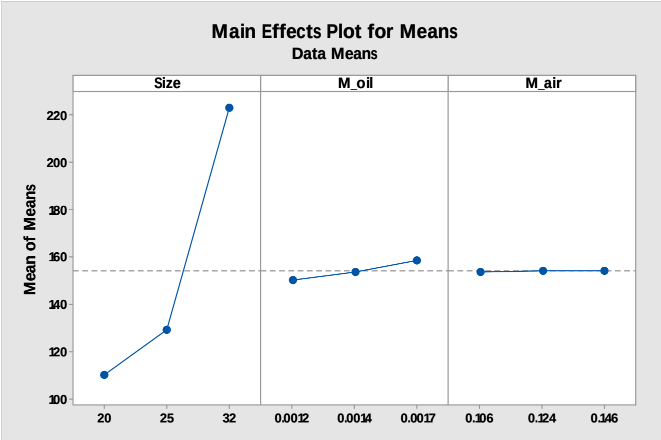
Table No 7 Response Table for Signal to Noise Ratios

Level	Size	M_oil	M_air
1	40.85	43.12	43.31
2	42.23	43.32	43.33
3	46.98	43.62	43.42
Delta	6.13	0.50	0.12
Rank	1	2	3

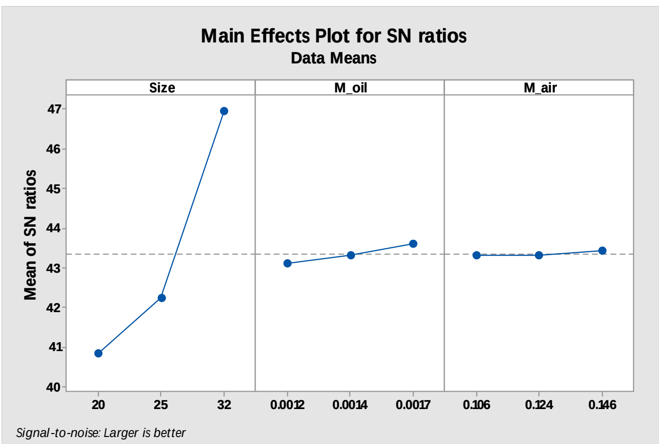
Response Table for Means

Table No 8 Response Table for Means

Level	Size	M_oil	M_air
1	110.3	150.3	154.0
2	129.3	154.0	154.7
3	223.3	158.7	154.3
Delta	113.0	8.3	0.7
Rank	1	2	3



Graph No 5 Main Effects Plot for Mean



Graph No 6 Main Effects Plot for SN ratios

VI. RESULT ANALYSIS AND DISCUSSIONS

The experiments were conducted by using the parametric approach of Taguchi's method. Using Taguchi approach, only main effect of individual parameters have been evaluated. The effects of individual parameters, on the output characteristics namely stress have been discussed in this section. Experimental data have been converted into signal to means as suggested by Taguchi method. "Larger is the better"

Selection of Optimum Level of Parameters

The least variation and the optimal design are obtained by means. Lower means, more stable the achievable quality (To sun et al., 2004). Figure 7.4 shows the means plots for thickness (t). It is clear from Figure 5.lowest means third level of Size (32), THIRD level of mass flow rate of oil (0.017 kg/sec)mm third level of flow rate of air (6.3) mm Therefore, the optimal setting of process parameters which yield maximum surface finish is **A3B3C**

Results of Statistical Analysis:

Table no 6.1 Results of Statistical Analysis

Optimal Parameter	Value
Size	32 mm
Mass flow rate of Oil (kg/sec)	0.017 kg/sec
Mass flow rate of air (kg/sec)	0.146

Experimental Results for Ansys Analysis:

Based on the experimental layout depicted in Table above the tests are performed in random order output and the Permutations under L4 orthogonal array are displayed below.

Table No 6.2 Experimental results for ansys analysis

Size	M_oil	M_air	Overall HTC
20	0.0012	0.106	106
20	0.0014	0.124	109
20	0.0017	0.146	116
25	0.0012	0.124	127
25	0.0014	0.146	129
25	0.0017	0.106	132
32	0.0012	0.146	218
32	0.0014	0.106	224
32	0.0017	0.124	228

VII. CONCLUSION

1. The Heat pipe system was designed and developed for retrofitting 3 sizes of heat pipes to be used in conjunction to the conventional radiator system
2. Heat pipe system was developed by suitable selecting 20, 25 and 32 mm heat pipes, spiral radial fins were used to increase the condenser area of heat pipes
3. Spiral radial fins appropriately selected and designed and analysis revealed that close to 30 watt power capacity of the sink ensures that maximum heat will be transferred from the exhaust gases to the cabinet.
4. The Experimental study revealed that temperature gradient of the air increases with increase in mass flow rate of air
5. The Experimental study revealed that temperature gradient of the oil increases with increase in mass flow rate of oil
6. The Experimental study revealed that overall heat transfer coefficient of the system increases with increase in mass flow rate of air
7. 32mm heat pipe system is the most effective with maximum heat transfer coefficient of 240 W/m2K

8. Design of experiment using Taguchi method revealed that 32 mm heat pipe with mass flow rate of air as 0.146 kg/sec and mass flow rate of oil as 0.017 kg/sec is the optimal condition to use the heat exchanger.

9. The application of heat pipe system successfully reduced the space and cost of system required to attain the given cooling rate.

10. By application of the 32mm heat pipe we can also eliminate the conventional radiator system through adding heat pipe systems in series.

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