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"FIN DESIGN OF AIR COOLED HEAT EXCHANGER USED IN ROTARY SCREW COMPRESSOR"

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Abstract: The performance of any compressor is depends on many parameter like pressure, temperature and application of compressor, The efficiency of compressor is mainly depends on heat generation and heat rejection from system, there are variety of heat exchanger used for heat removal like water cooler, air cooler, plate and bar, shell and tube etc. the present work is aim to focus on air-cooled performance enhancement via fin design. The fin is one of the critical components of heat transfer proper design and location of fin on heat exchanger can improve a performance lots. The aim of work is to improve the heat exchange rate of current cooler without changing the parameters like air flow, air velocity. The variety of fin like plain fin, circular fin, herringbone etc. will be verified against geometrical arrangement like circular pitch, rectangular pitch, triangle pitch for performance enhancement and same can be variey with help of CFD. The new cooler fin design was suggested based on the Computational analysis of cooler which was indicating a circular fin with Al as a fin material will give us a better heat rejection and lower pressure drop for cooling air flow. The cooler sample model of cooler was manufactured with help of heat exchanger manufacturer and the cooler performance was measured in industrial test setup having wind tunnel the setup is having all the facilities to measure Velocity, temperature and pressure drop across the wind tunnel also the ambient air temperature can be vary with help of submerge heater and oil tank which can be precisely measured and controlled. The latter stage of work shows the comparisons of both results and we have observed the gap of near about 7.875 % in both results which is far enough to accept as per literature review, the gap analysis was carried out to identify why the gap was observed in both case.

Key word: Thermal analysis, fins geometry, CFD, Wind tunnel

1. INTRODUCTION

1.1Why Is After coolering Required?

- The compressed air discharged from an air compressor is hot Compressed air at these temperatures contains large quantities of water in vapor form. As the compressed air cools this water vapor condenses into a liquid form.
- Reduce compressed air moisture level
- Increase system capacity
- Protect downstream equipment from excessive heat
- Coolers are usually sized with a CTD (2.7°C, 5.5°C, 8.3°C, or 11°C). This means that the compressed air temperature at the outlet of the after cooler will be equal to the cooling medium temperature plus the CTD when sized at the specified inlet air temperature and flow

Air cooled after coolers use ambient air to cool the hot compressed air. The compressed air enters the air cooled after cooler. The compressed air travels through either finned tubes or corrugated aluminum sheets of the after cooler while ambient air is forced over the cooler by a motor-driven fan. The cooler, ambient air removes heat from the compressed air

2. LITRATURE REVIEW

Dong H.Lee **et al.** [1]the present study investigated the air-side heat transfer performance of a heat exchanger with perforated circular finned tubes. The air-side convective heat transfer coefficients for 2-hole and 4hole PCFT cases increased by 3.55% and 3.31% respectively pressure drop by 0.68% and 2.08%, respectively. The fin factor for the 2-hole PCFT case was 5.19, whereas that for the 4-hole PCFT case was 1.59,

M. Zeng **et al.**[2]In this study, the influence of various design parameters on the heat transfer and flow friction characteristics of a heat exchanger with vortex-generator fins is analyzed by numerical method The results show that the intensity of heat transfer can be greatly increased with the increase of the vortex-generator attack angle, the length and the height,

Minsung Kim **et al**,[3]In this study, the performance of air-cooled cross type PHEs to replace large capacity of open-loop cooling towers is investigated. Double-wave PHE the air-side heat transfer performance approximately 50% higher than single wave PHE, but requires 30% more pressure drop

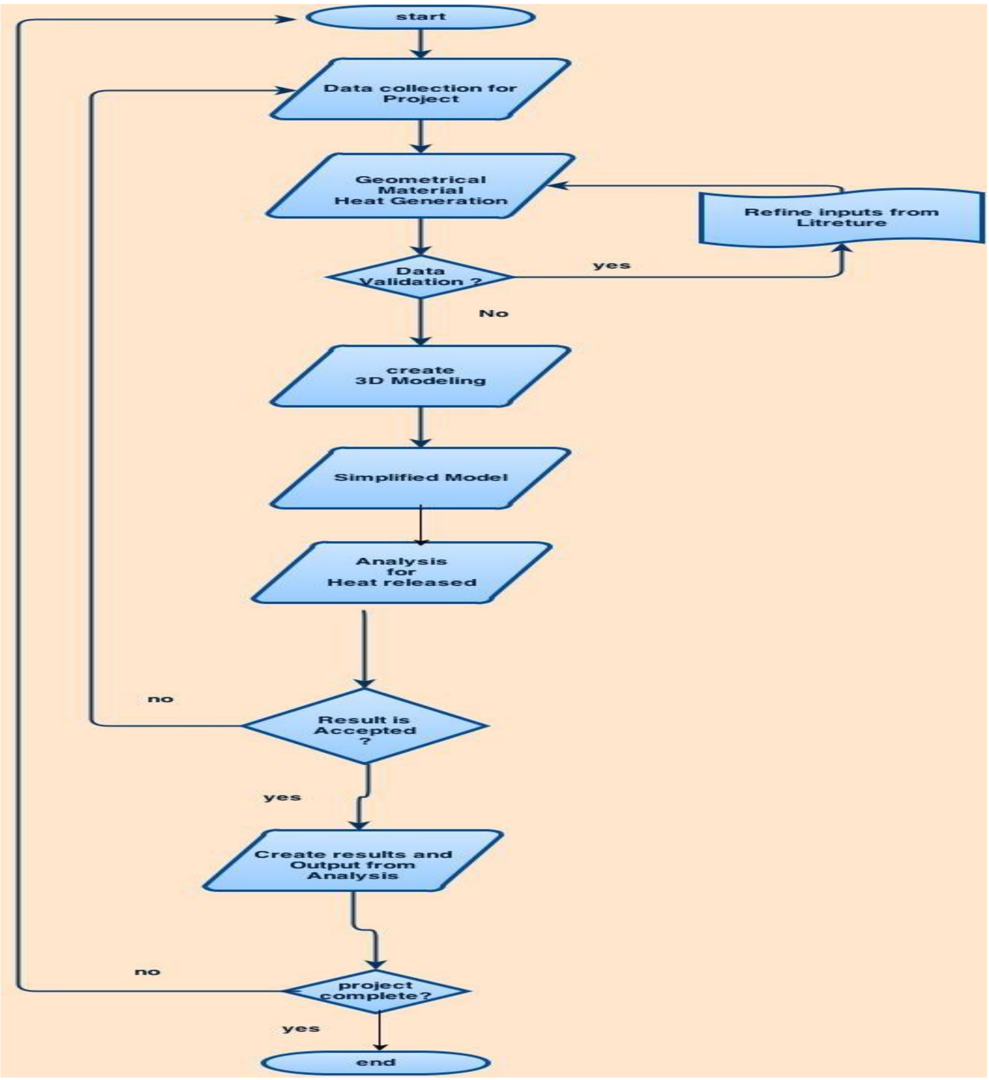
Seth A. Lawson **et al**.[4]Experiments were conducted to determine the independent effects of span wise and stream wise spacing on heat transfer and pressure loss through multiple row arrays of pin fins. If the stream wise spacing was larger than the span wise spacing, the heat transfer was maximized farther upstream in the array.

L.H. Tanget al. ,[5]In the present study, the air-side heat transfer of five kinds of fin-and-tube heat exchangers have been experimentally investigated The larger attack angle, higher length and smaller height ofvortex generators will lead to better overall performance of heat exchangers with VGs .

Kumbhar D.G **et al.** [6]It is observed that heat transfer rate increases with perforations as cor-pared to fins ofsimilar dimensions without perforations. It is noted that in case of triangular perforations optimum heat transfer is achieved. It is also concluded that heat transfer rate is different for different materials or heat transfer rate changes with change in thermal conductivity

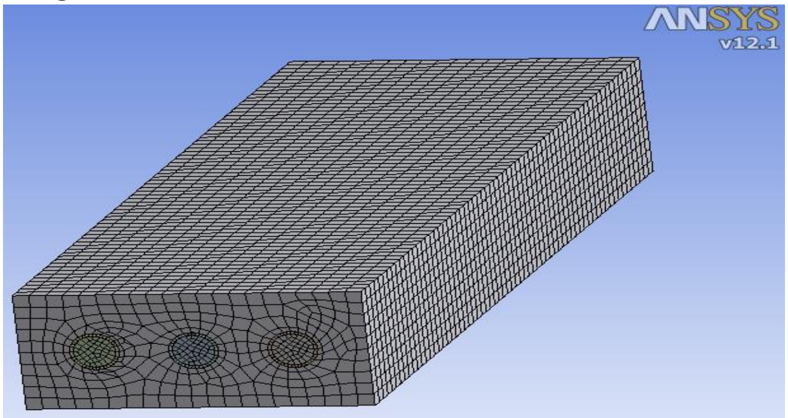
Ian J. Kennedy**et al.** [7]**t**he main conclusion of the study is that inclination leads to a small increase in the thermal performance of the ACHE of approximately 0.5% for the optimum inclination angle of 30,

3. FLOW CHART OF METHODOLOGY



4. EXISTING MODEL SPECIFICATION OF AIR COOLER

- Cooler type – Aluminium bar and plate
- Cooler passes- single
- Max. Working Pr-14bar
- Max. Working fluid temp. - 109°C
- Ambient air temp. range – 2°C to 49°C
- Compressed air inlet mass flow – 0.4096 kg/s
- Compressed air inlet temp. – 101.7 °C
- Compressed air inlet Pressure – 7.9 bar
- Compressed air outlet temp. - < 54 °C
- Compressed air ΔP - < 0.14bar
- Rejected Heat Load – 32.25 KW
- Cooling air inlet temp. Ambient
- Max. Length over Manifolds- 610 mm and Max. Over Width- 450 mm



MESH MODEL

Domain	Nodes	Elements	Wedges	Hexahedra
1	18792	14706	0	14706
2	35612	29184	114	29070
3	25056	12312	0	12312
All Domains	79460	56202	114	56088

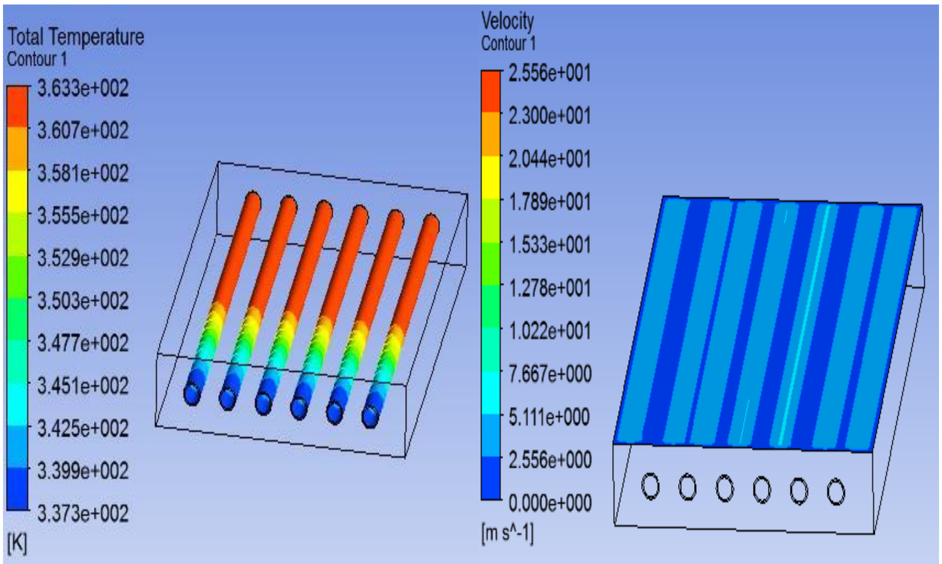
Mesh Result

5.BOUNDARY CONDITIONSARE MENTIONED INTHETABLE BELOW:

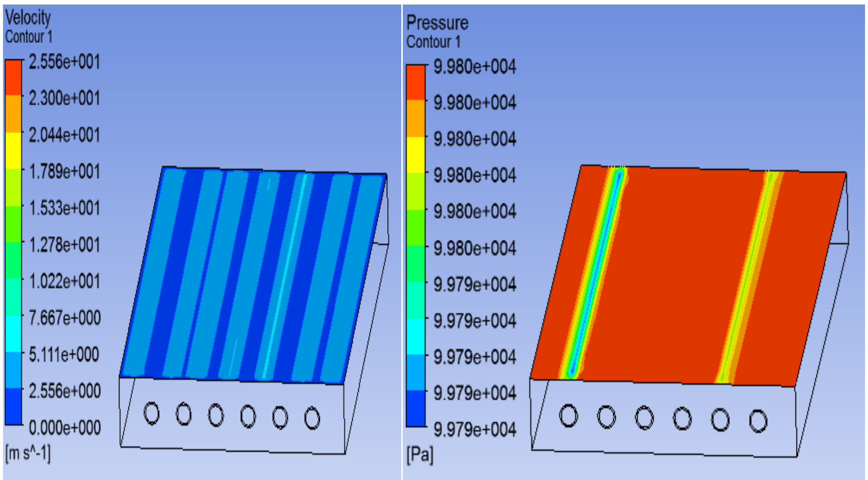
No	Boundary conditions	Value
1	Inletat Bottom surfaceof the cooler	Air mass flow rate0.3328 kg/s at 40°C &99987 pa
2	Outlet at Top surfaceof the cooler	Air velocityat5.5 m/s

3	Coolerinside[inlet Port]	Compressed air at 8 barpressure& 85°C
4	Coolerinside[Outlet Port]	Compressed air at 0.0532kg/s ≈100 CFM
5	Geometry	Symmetrical for cooler so onlysingle coreshould beconsidered for Effectiveanalysis.
6	Ambient Air	40°C

6.CFD ANALYSIS



TemperaturevariationofCircular fins **AirvelocityatCooleroutlet**



AirvelocityatCooleroutletAirPressureat Cooleroutlet

7. Heat Transfer Calculation

As theHeat transfer equation,**Q = U* A*LMTD**

Where, Q =Heat transferred, Watt(J/s),

U= Overall Heat transfer Co-efficient W/m$^{2\circ}$ c

= For Forced Convection of Air = Taking 100 W/m$^{2\circ}$ c

A =Heat transfersurface area(cooler), m$^{2=}$(0.490 * 0.335)m^2

LMTD=LogMean TemperatureDifference °C $\frac{(T1-t2)-(T2-t1)}{ln\frac{(T1-t2)}{(T2-t1)}}$

Heat release, Q = m CΔT

The governing equations are continuity, momentum and energy equations, which are derived from fundamental principles of heat and fluid flow. The equations are posed to implement assumptions are made

The compressibility, the gravitational force and the dissipating heat caused by viscosity are neglected, the continuity, momentum ad energy equations for a fully developed 3D flow heat transfer are: Continuity Equation, Momentum Equation (Navier-stokes Equation), and Energy Equation.

The Continuity Equation

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{dw}{dz}=0$$

$$\nabla\left(\rho \underset{V}{\rightarrow}\right) = 0$$

Simplified Momentum Equation (Navier-stokes Equation),

X-momentum: $\nabla\left(\rho \underset{V}{\rightarrow}\right) = -\frac{\partial p}{\partial x}+\frac{\partial \tau xx}{\partial x}+\frac{\partial \tau yx}{\partial y}+\frac{\partial \tau zx}{\partial z}$

Y-momentum: $\nabla\left(\rho \underset{V}{\rightarrow}\right) = -\frac{\partial p}{\partial x}+\frac{\partial \tau xy}{\partial x}+\frac{\partial \tau yy}{\partial y}+\frac{\partial \tau zy}{\partial z}+\rho g$

Z-momentum: $\nabla\left(\rho \underset{V}{\rightarrow}\right) = -\frac{\partial p}{\partial x} + \frac{\partial \tau xz}{\partial x} + \frac{\partial \tau yz}{\partial y} + \frac{\partial \tau xx}{\partial z} + \rho g$

Energy Equation

$\nabla\left(\rho \underset{V}{\rightarrow}\right) = -\mathrm{p}\nabla \underset{V}{\rightarrow} + \nabla\,(\mathrm{k}\nabla\mathrm{T}) + \mathrm{q} + \psi$

Turbulence is taken care by Shear Stress Transport (SST) k-ώ model

$\frac{\partial (pk)}{\partial t} + \frac{\partial (kui)}{\partial xi} = \frac{\partial}{\partial xj}\left[\frac{\partial k}{\partial xj}\right] + \mathrm{Gk} - \mathrm{Yk} + \mathrm{Sk}$

8. CFD RESULT

TABLE- 1 PLATE FIN

THICKNESS	1.25mm THICKNESS CASE 01			1.5 mm THICKNESS CASE 02		
MATERIAL	**AL 6062**	**Cu**	**90–CU/10 Ni**	**AL 6062**	**Cu**	**90–CU/10 Ni**
NODAL Temp.˚C	94	96.8	95	94.5	97.12	96.12
Heat loss (w)	8.51	**9.38**	8.82	9.05	8.68	8.56
CORRECTED LMTD	125	127.8	127	126.3	128	127.6
COOLINGAIR VELOCITY (M/Sec	5.4	**5.4**	5.4	5.32	5.32	5.32
PRESSUREDROP ACCORSSFIN(Pa)	2.12	2.12	2.12	2.32	2.32	2.32

TABLE – 2 TRIANGLULAR(ORPLATEANDBAR) FIN

THICKNESS	1.25mm THICKNESS CASE 01			1.5 mm THICKNESS CASE 02		
MATERIAL	**AL 6062**	**Cu**	**90–CU/10 Ni**	**AL 6062**	**Cu**	**90–CU/10 Ni**
NODAL Temp.˚C	94.58	97.32	95.60	94.98	96.45	96.23
Heat loss (w)	8.89934	9.69892	9.19988	8.99478	9.920232	9.17498
CORRECTED LMTD	129	130.23	129.67	131	133.45	132.45
COOLINGAIR VELOCITYSTREAM LINE(M/Sec	5.42	5.42	5.32	5.67\	5.67	5.67
PRESSUREDROP ACCORSSFIN(Pa)	2.34	2.34	2.34	2.45	2.45	2.45

TABLE- 3 CIRCULAR FIN

THICKNESS	1.25mm THICKNESS CASE 01	1.5 mm THICKNESS CASE 02

MATERIAL	**AL 6062**	**Cu**	**90–CU/10 Ni**	**AL 6062**	**Cu**	**90–CU/10 Ni**
NODAL Temp.˚C	95.23	97.45	96.08	95.92	98.19	97.23
Heat loss (w)	8.51	9.38	8.82	**8.67**	9.48	8.97
CORRECTED LMTD	125	127.8	127	126.45	128	127.9
COOLINGAIR VELOCITYSTREAM LINE(M/Sec	5.34	5.34	5.34	**5.27**	5.27	5.27
PRESSUREDROP ACCORSSFIN(Pa)	2.34	2.34	2.34	2.43	2.43	2.43

Outofallabovepossiblecombinationwecanobservedthatsomeoffinsaregoodinterms of LMTD someof are inPressureDropacrossthefinhencethefinalselectionbecamevery difficult. To havereasonablesolutionof this quizwehaveformeda Selectionmatrixbasedon manufacturer'sopinion andoperationparameters.The firstconsiderationforselectionoffin wouldbeaHeatLoss (W), thenCoolingAirVelocity StreamLine(M/Sec)thenPressuredrop across the fin(Pa).Basedonthisselectionmatrixwecanconcludethatthe **PlatefinhavingCuasfinbasematerial withfinthicknessof1.25mmisbetter**, butthecoolercorematerialis AL hencethemanufacturing processtojoinALwithCU becamechallenges hencethoughitisan optimumvalueweshouldnot beconsidersamefortheEvolution.Theextavailablesolutionas permatrixandmanufacturingacceptancesis **AL6062withcircularfinhaving1.50mm thickness**should beselected.

Experimental set-up of Wind tunnel

9.Comparison Analytical AND Experimental Results

MATERIAL	**CFD RESULTS**	**EXPERIMENTAL RESULTS**	**GAP IN AVLUE**	**GAP IN %**

NODAL TEMPERATURE°C	95.92	NOT MEASURED	NA	NA
HEAT LOSS (KW)	8.67	7.98	0.69	7.958478
CORRECTED LMTD	126.45	115.9	10.55	8.343219
AIR VELOCITY (M/Sec)	5.27	5.15	0.12	2.27704

10.CONCLUTION

This table shows the value of Nodal temperature can be measured in wind tunnel as cooler is totally enclosed in the sheet metal shroud to avoid any air flow leakages. The heat loss from the CFD was observed as 8.67 KW while same was observed on wind tunnel as 7.98 KW which indicates the gap in cooler is about 0.69 W which is equal to 7.9584 % and same is 8.34 % in case of LMTD. The air velocity was observed as 5.15 M / sec which is having gap of 2.12 5 with references to CFD results.
The gap in computational and experimental results are within limit of 10 % hence we can conclude that the method adopted for computational analysis is up to the required level and same cooler can be consider for the manufacturing with adding design safe factor for application.
The gap in actual and prediction was occurred due to numbers of reasons and some of prime are

- Presence of Humidity in actual test set up cannot be simulated in CFD
- The CFD is an approximate approach which itself indicates that the actual results may vary in some limit.
- The temperature measured are having minor offset in both case as in 3D model we can measured temp as actual required points but while in actual test set up we cannot do same due to mounting limitation of sensor hence some diff in temperature is obvious
- The cooling air was diverted through blower and duct towards the cooler and due to frequency variation there is change in speed and ultimately in cooling flow and velocity. While it is considered as constant in case of CFD
- The cooler was consider without brazing joint efficiency in 3D model while in actual condition the Fins are brazed to cooler core hence minor temp as well as heat rejection diff in actual location which results to overall diff in both the values.

11.NOMENCLATURE

Q = Heat Transfer Rate, W
Cp = Specific Heat, J/kgK
T1 =Hot Fluid inlet temperature °C

T2 =Hot Fluid outlet temperature,°C

t1 = Cold Fluid inlet temperature, °C

t2 = Cold Fluid outlet temperature,°C

U = overall heat transfer coefficient, W/m^{2}K
Ai= Area of Inner Pipe, m^2
ΔT = LMTD
K =Thermal Conductivity Of Fluid, W/mK
V =Velocity Of Fluid, m/s
Δp = Pressure Drop, pa

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