IJESRT
INTERNATIONAL JOURNAL OF ENGINEERING SCIENCES & RESEARCH TECHNOLOGY

PERFORMANCE ENHANCEMENT OF MOBILE AIR CONDITIONING SYSTEM USING VARIABLE GEOMETRY MICRO CHANNEL HEAT EXCHANGER (MCHX)

A. Ram
Research Scholar, Department of Mechanical Engineering, B.S. Abdur Rahman University, Chennai, India

DOI: 10.5281/zenodo.50415

ABSTRACT

Due to stiff competition among the Original Equipment Manufacturer (OEM's), the Comfort, Fuel efficiency & Safety are the key factors that drive the vehicle business; in that context A/c plays an important part for the comfort of the passengers. However, condenser plays a pivotal role in Power train cooling system & Air conditioning system. Therefore it is a big challenge for the automotive engineer to propose an innovative design that can improve the thermal performance of Condenser without not compromise the package size. The objective of this paper is to further enhance the Condenser performance by using variable geometry on both port & tube control volume and also optimizing the fin parameters for various tube profiles. Selection of refrigerant side pressure drop correlation based on comparison of analytical results with the experimental data (component calorimeter) for the fixed geometry. The mathematical calculation is derived to address two phase density using suitable pressure drop correlations and its effects on refrigerant side heat transfer. Since the effect of air side heat transfer co-efficient plays an important role in overall heat transfer coefficient, the fin optimisation using mathematical calculation and CFD results were carried out. Finally the optimised fin parameters are considered to find out the effect of variable geometry (tube & port) on the overall heat transfer coefficient using CFD & Bench test results.

KEYWORDS: Variable geometry, Calorimeter, Heat transfer co-efficient.

INTRODUCTION

Recently, the compact heat exchanger has been paid more and more attention by HVAC&R (heating, ventilating, air-conditioning and refrigeration) industries and used in some commercial, residential and automotive air conditioning system. Compact heat exchanger i.e. Air cooled cross flow fin- tube heat exchanger is a device that is used in wide variety of applications. Typical among these are automobile radiators, air conditioning evaporators and condensers, charge air coolers, electronic cooling devices and cryogenic exchangers to meet the demand for saving energy and its resources. To reduce the size and weight of heat exchangers, various augmented surfaces have been developed to improve the air side heat transfer performances. Typical fin geometries are plain fins, wavy fins, offset fins, perforated fins, pin fins and louvered fins. For the complex geometries, which are usually set up in a cross flow arrangement, with louvered fin geometry in a compact automotive condenser. In compact heat exchangers, thermal resistance is generally dominant on the air-side and may account for 80% or more of the total thermal resistance. The air-side heat transfer surface area is 8 to 10 times larger than the refrigerant-side. Any improvement in the heat transfer on air-side therefore improves the overall performance of the heat exchanger. Many studies on louver fin have been reported Davenport (1980), through flow visualization study with smoke traces or dye injection techniques, showed that the air flow had two directions depended on Reynolds number based on the louver pitch and maximum air velocity. The air doesn't pass through louvers called duct directed flow at low Reynolds numbers. However, at high Reynolds numbers, the air flow direction became nearly parallel to louvers called louver directed flows. Achaichia and Cowell (1988) further confirmed the phenomena. Based on their researches, some air-side heat transfer coefficient and pressure drop correlations for louver fin proposed based on a huge database (Park and Jacobi, 2009), also including some results of their own experimental studies on heat transfer characteristics of different louver fin geometries by Chang and Wang (1997), Kim and Bullard (2002). However, due to different experimental and data reduction methods, these correlations have significant discrepancy with experimental results. Recently, numerical investigations of the louvered fin array have been widely performed to reduce investigation period.

MATERIALS AND METHODS

An automotive Condenser is made up of a number of tubes, fins, two manifolds and an inlet and outlet connection. Inside the manifolds of the condenser, there will be partitions, which direct the flow of refrigerant into a set path. When creating effective path configuration for the refrigerant total surface area in contact between tube and refrigerant is increased. The fins will provide the large surface area for the cooling medium (air), which will allow the forced convection of the heat given off from the refrigerant via the connected tubes. This heat rejection is vital to the operation of the condenser.

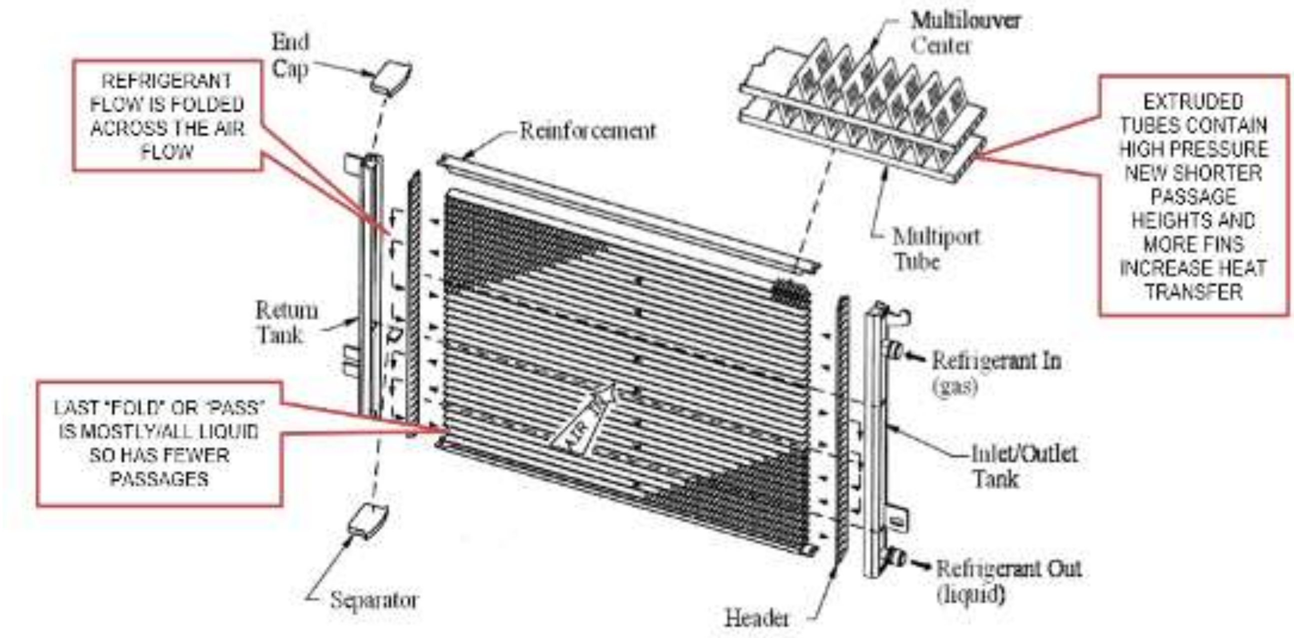


Figure:1

Tube Hydraulic diameter calculation

The geometrical section mainly taken are rectangle, hexagon and oval with the each section has rectangular and triangular ports. The various geometrical section properties like aspect ratio perimeter and hydraulic diameter are calculated for corresponding geometrical section. The formulae for variable geometry are derived from the aspect ratio. The aspect ratio for each design is determined separately and its perimeter, hydraulic diameters are calculated as shown in table-3 .

ISSN: 2277-9655
(I2OR), Publication Impact Factor: 3.785

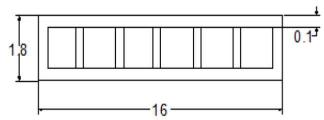
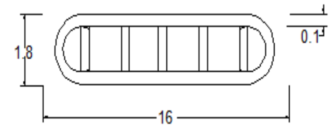
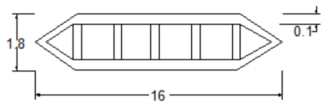
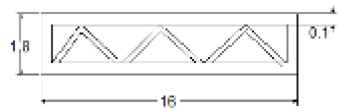
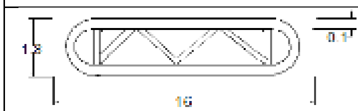
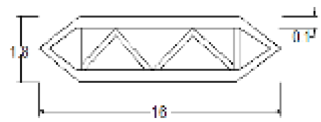
Condenser Variable tube Geometry as shown	
 Rectangular tube with rectangular port	 Oval tube with rectangular port
 Hexagon tube with rectangular port	 Rectangular tube with triangular port
 Curved tube with triangular port	 Hexagon tube with triangular port

Table:3 Tube Hydraulic diameters

Sections	Aspect ratio	Perimeter (mm)	Hydraulic Diameter D_h (mm)
Rectangular tube (Rect-ports)	$r_d = \frac{9.9375}{n} - 0.062$	$P_w = 3.2(1 + r_d)$	$D_h = \frac{3.2 \times r_d}{(1 + r_d)}$
Hexagon tube (Rec-ports)	$r_d = \frac{8.8125}{n} - 0.062$	$P_w = 3.2(1 + r_d)$	$D_h = \frac{3.2 \times r_d}{(1 + r_d)}$
Oval tube (Rec-ports)	$r_d = \frac{8.8125}{n} - 0.062$	$P_w = 3.2(1 + r_d)$	$D_h = \frac{3.2 \times r_d}{(1 + r_d)}$
Rectangular tube (Tri-ports)	$r_d = \left(\frac{2}{\tan\theta}\right)$	$P_{wt} = \left(3 \times N / \tan\theta\right) \times \left(\frac{1}{\cos\theta} + 1\right)$	$D_{ht} = \left(4 \times A / P_{wt}\right)$
Hexagon tube (Tri - ports)	$r_d = \left(\frac{2}{\tan\alpha}\right)$	$P_w = \frac{3}{\tan\alpha}\left[1 + \frac{1}{\cos\alpha}\right]$	$D_h = \frac{4 \times A}{P_w}$
Oval tube (Tri- ports)	$r_d = \left(\frac{2}{\tan\alpha}\right)$	$P_{w1} = 1.97485 + \frac{1.5}{\sin\alpha}$	$D_h = \left(\frac{4 \times A}{P_w}\right)$

Two-Phase Pressure Drop correlations:

A major design consideration for condenser/evaporator operation in air-conditioning and refrigeration applications involves characterizing the two-phase frictional pressure loss. Due to the demand for decreasing the total refrigerant charge required in a refrigeration system, many of the tubes for these heat exchangers are approaching hydraulic diameters on the order of, and less than, 1mm. A very limited amount of work has been performed in the area of two-phase pressure drop modeling in small diameter extruded aluminum micro-channel. Thus, several large

ISSN: 2277-9655
(I2OR), Publication Impact Factor: 3.785

diameter (Di· > 5mm), along with existing small tube, correlations will be compared to data produced from this study. Several of these correlations are based on the separated flow model developed by Lockhart and Martinelli [1949], Friedel(1979), Chisholm(1967).

Locart Martinalli correlation:

$$\left(\frac{dp}{dz}\right)=\varphi_L\left(\frac{dp_F}{dz}\right)_L$$

$$\varphi_L=1+\frac{C}{X}+\frac{1}{X^2},\varphi_G=1+CX+X^2 \qquad (1)$$

Friedel correlation:

$$\left(\frac{dp}{dz}\right)=\varphi_{LO}\left(\frac{dp_F}{dz}\right)_{LO}$$

$$\varphi_{LO}=E+\frac{3.24\times F\times H}{Fr^{0.045}\times we^{0.035}} \qquad (2)$$

Chisholm correlation:

$$\left(\frac{dp}{dz}\right)=\varphi_{LO}\left(\frac{dp_F}{dz}\right)_{LO}$$

$$\varphi_L=1+(Y^2-1)*\left\{Bx^{(\frac{2-n}{n})}(1-x)^{(\frac{2-n}{2})}+x^{2-n}\right\} \qquad (3)$$

Refrigerant Pressure drop for various correlation are calculated and comparing this results with experimental data

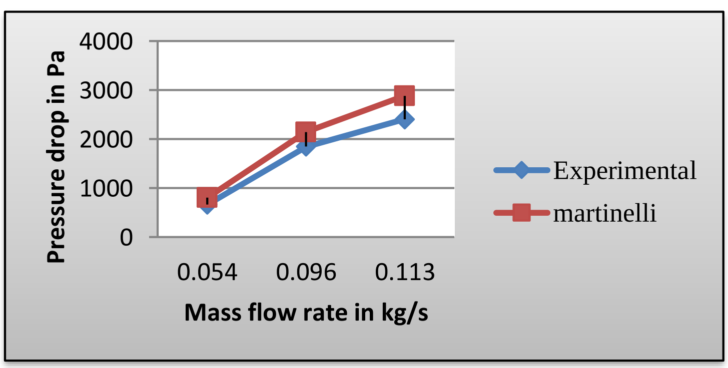


Fig. 2. Calculation versus experimental

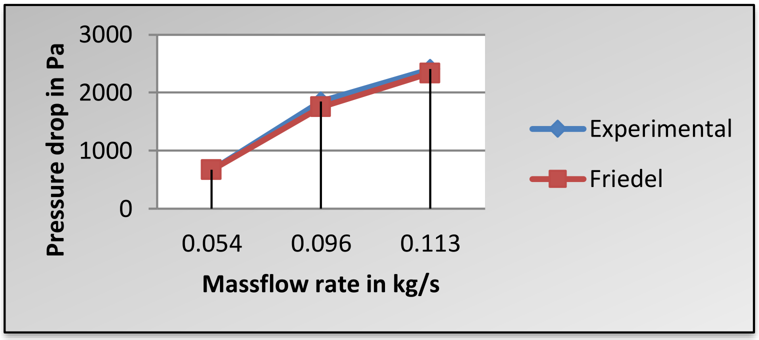


Fig. 3. Calculation versus experimental

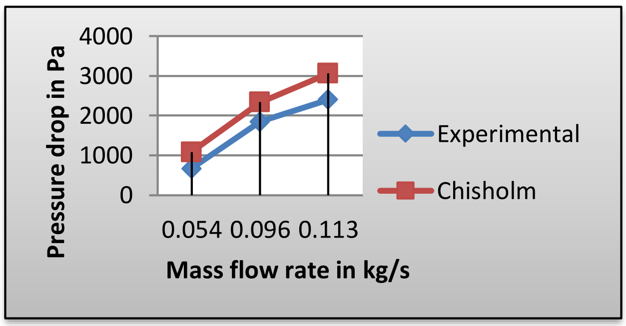


Fig. 4. Calculation versus experimental

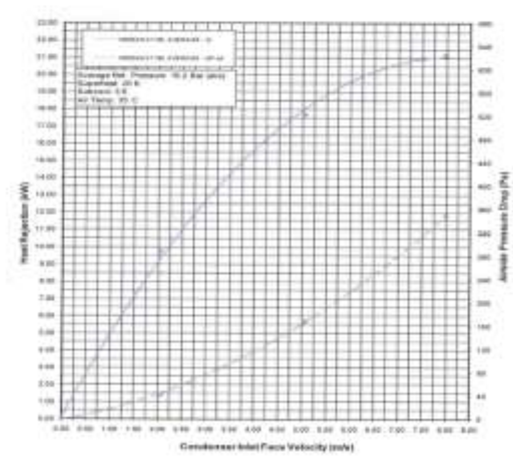


Fig:5 Experimental results

Fig:6 Experimental setup

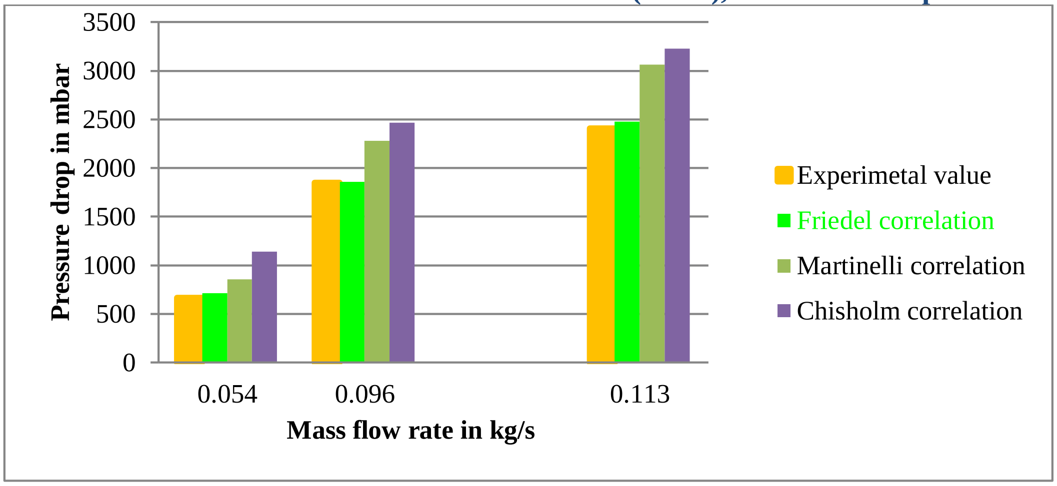


Fig:7 Summary of correlation data

The above table shows that the pressure drop of Friedel correlation is closer to the experimental pressure drop results as shown in figure-7
Since the Friedel correlation gives minimum error, so this correlation is used to find out overall heat transfer co-efficient for variable tube geometry and results are shown in Table-4

Table: 4 Summary of Mathematical results

Heat Transfer Co-efficient (W/m^2K)						
Refrigerant Mass flow rate (Kg/s)	Rec-Rec	Oval-Rec	Hex-Rec	Rec-tri	Oval-tri	Hex-tri
0.054	2558	2557	2637	2625	2723	3024
0.096	3869	3924	4047	4026	4177	4642
0.113	4469	4469	4609	4585	4756	5286

Air velocity (m/s)	Overall heat transfer coefficient (W/m²K)					
	Rec-Rec	Oval - Rec	Hex - Rec	Rec-Tri	Oval - Tri	Hex - Tri
2	324	332	338	325	339	346
5	471	483	490	474	492	502
8	557	570	578	560	581	592

Two-Phase CFD simulation:

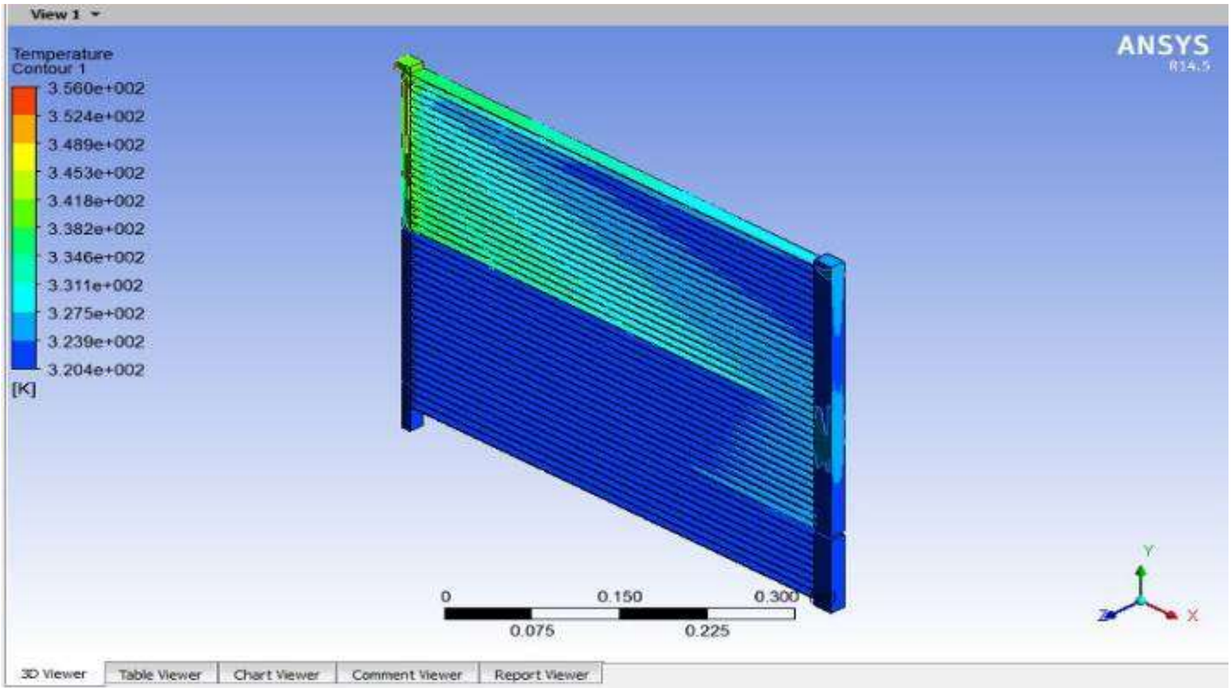


Fig:10 CFD of condenser model

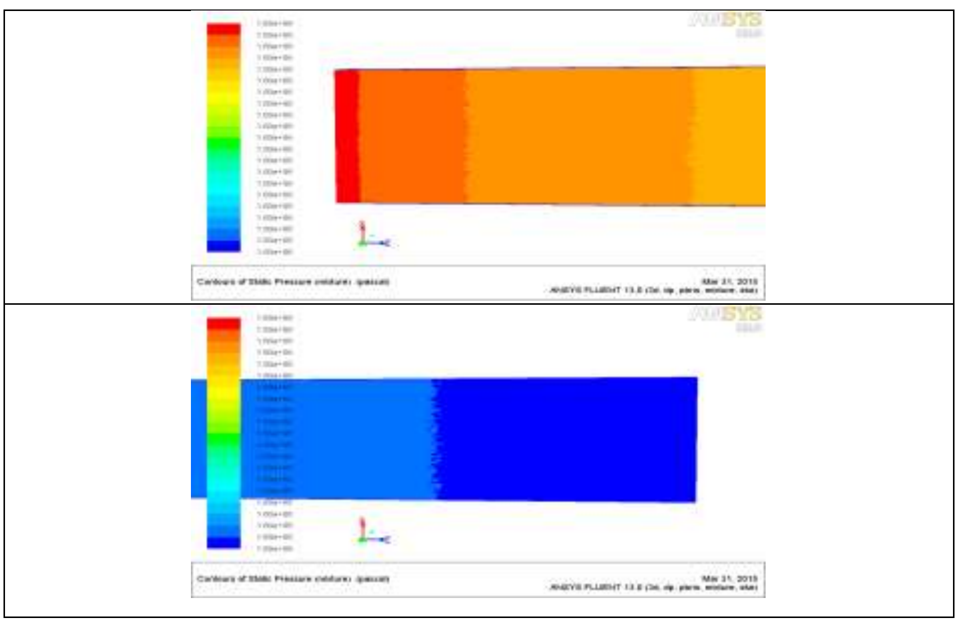


Fig:11 Pressure drop using CFD

ISSN: 2277-9655
(I2OR), Publication Impact Factor: 3.785

Table: 5 Summary of CFD Vs Experimental data

Ref. Mass flow rate (Kg/s)	Outlet temp (°C)	Pressure drop (mbar)		Heat rejection (kW)	
		Experiment	CFD	Experiment	CFD
0.054	44	665	730	9.7	10.7
0.096	43.5	1852	1959	17.6	18.3
0.113	43.5	2408	2674	20.8	21.5

The above table shows 3 to 10% variation between CFD & Experimental results.

Heat transfer and flow friction characteristics:

The heat transfer and flow friction characteristics of a heat exchanger surfaces are commonly expressed in non-dimensional form and are simply referred to as the basic characteristic or basic data of the surface. Various correlations are available in literatures which express the Colburn factor, j and friction factor, f as functions of Reynolds number and other geometrical properties. The correlations of the j and f factors are obtained from chung (2000) correlation data from multi-louvered fins and flat tube heat exchangers with different geometry parameters including the fin pitch, fin height, fin length, fin thickness and louver angle.

Chang Correlation:

$$f_a = 4.97 \times \mathrm{Re}^{\left(0.6049 - \frac{1.0649}{\theta^2}\right)} \times \left\{ \ln\left[\frac{t}{F_p}\right]^{0.5} + 0.9 \right\}^{-0.527}$$

$$f_b = \left(\frac{t_p}{t_h}\right)^{-0.0446} \times \left\{ \ln\left[1.2 + \left(\frac{L_P}{F_P}\right)^{1.4}\right] \right\} \times \theta^{-0.477}$$

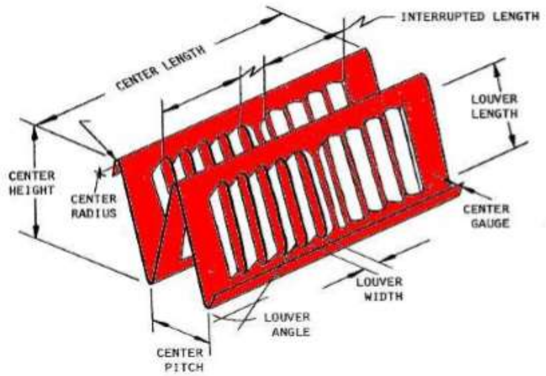
$$f_c = \left[\left(\frac{D_h}{L_P}\right) \times \ln(0.3 \times \mathrm{Re}_{Lp})\right]^{-2.966} \times \left(\frac{F_p}{L_i}\right)^{-0.793 \times \frac{F_P}{F_h}} \quad (4)$$

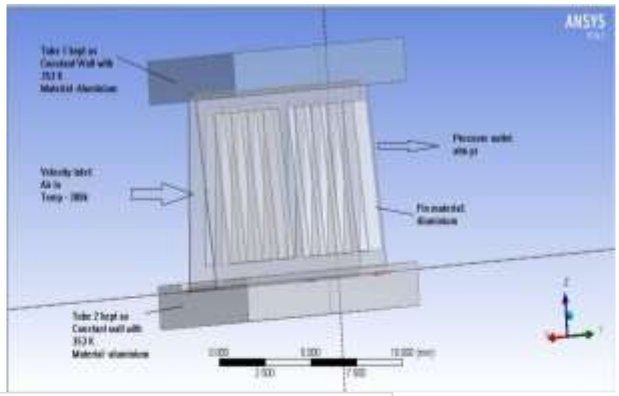
$$f_d = f_a \times f_b \times f_c$$

$$\Delta P = 4 f \frac{f_d}{D_h}\left(\frac{\rho v^2}{2}\right) \quad \frac{\mathrm{N}}{\mathrm{m}^2}$$

Fin Side - CFD Model and Boundary Conditions:

The Automotive air-conditioning condenser is installed in the inlet of wind tunnel with surrounded insulation to protect it from heat loss and air leakage. Tests are performed for face air velocity from 2 to 8 m/s, corresponding to the normal operating range of conditions in automotive air-conditionings. The air inlet temperature is 35˚ for all tests. On the tube side, refrigerant inlet is given as 83˚C temperature. The refrigerant mass flow rate is maintained at 54g/s. The air-side thermal performance data are determined using the effectiveness-NTU method for cross flow heat exchangers with both fluids unmixed and the detailed reduction method can refer to chang's study paper. Air-side pressure drop can be directly measured with static pressure device.





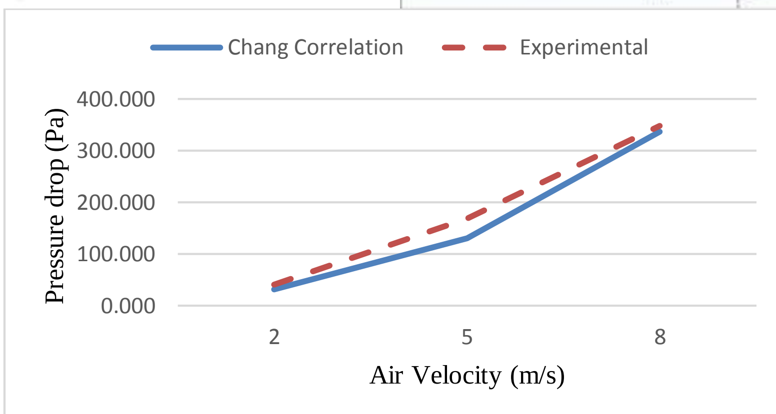


Fig:13 Pressure drop with experimental results and Chang correlation calculation results

Fig.13 presents comparison of experimental results and Chang (2000) correlation calculation results on sample mentioned in above table. It is noticed that the variation tendencies with the air velocity of Pressure drop are well consistent with the Chang correlation calculation results. The discrepancy of pressure drop is within 15% for chang correlation. Therefore, the chang correlation is used in this study can be regarded as reliable and can be used to validate the CFD simulation results.

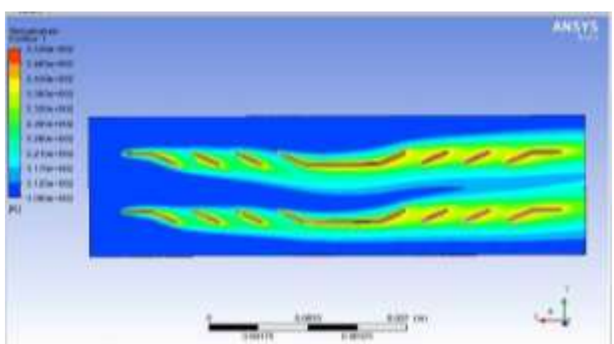


Fig:14 Louvered fin temperature contour

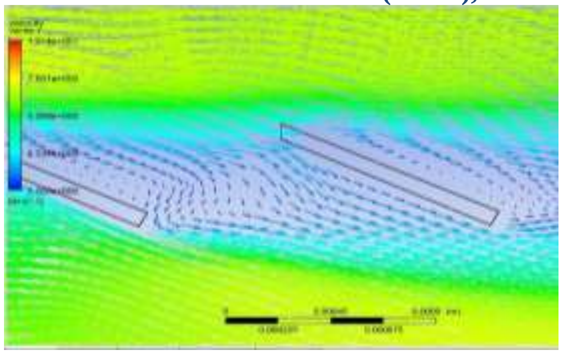
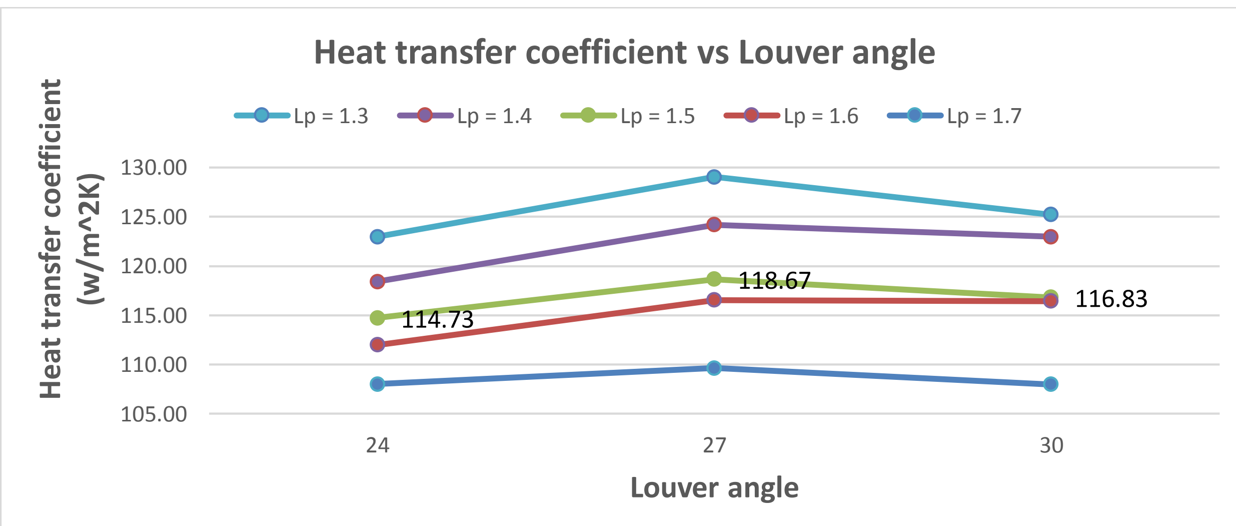


Fig:15 Velocity vector

RESULTS AND DISCUSSION

Fig. 16 show the variations of heat transfer coefficient and pressure drop with louver pitch and louver angle for Fp=1.17mm. The Pressure drop decreases and then increases with increasing louver pitch, and the heat transfer coefficient increases and then decreases with increasing louver pitch. For small louver pitch, the pressure drop of large louver angle is higher. When louver pitch reaches 1.5mm, the pressure drop meet the required value, for both 24o and 27o louver angle. But in the view point of heat transfer co-efficient, 27o louver angle can give a higher heat transfer co-efficient than 24o louver angle. Hence, the combination of 1.5 mm louver pitch and 27˚ louver angle is a better choice for automotive air conditioning condenser.



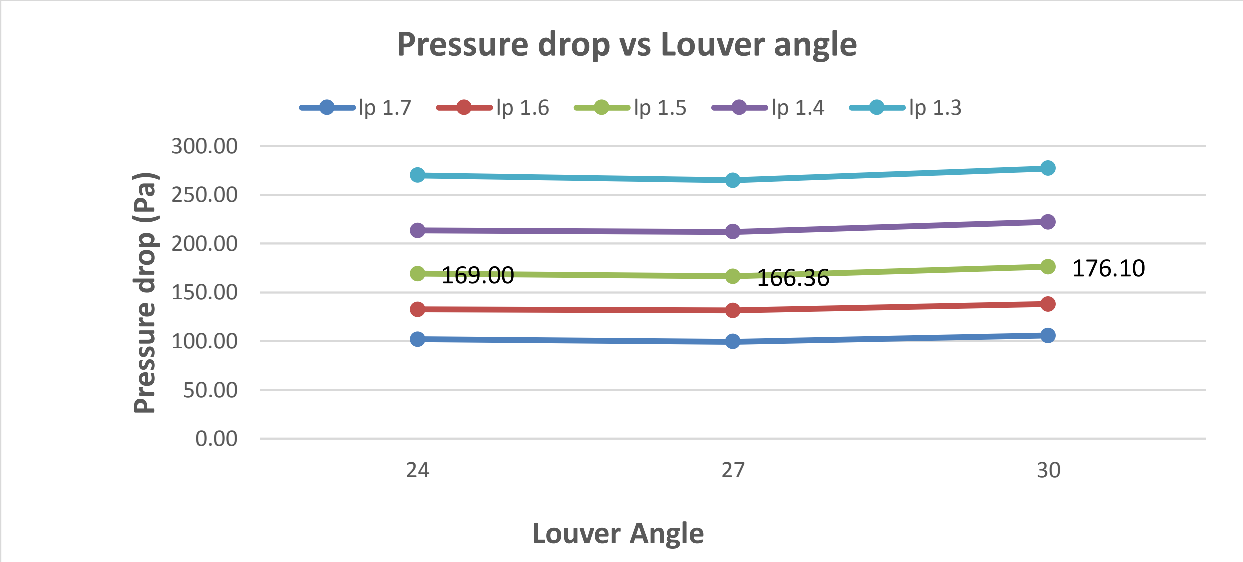


Fig.16: Variations of pressure drop and heat transfer co-efficient with Louver Angle and Louver Pitch

CONCLUSION

In refrigerant side Friedel correlation suits to be best among other correlations taken with deviation of ±10 %.In air side chung correlation is taken with 25%.

From the above illustrations and detailed study the analysis, it is concluded as follows:

Rectangular model has not only low pressure drop but also low heat transfer coefficient

Hexagon model is of high pressure drop and as well as high heat transfer coefficient

In the case of Oval tube model which takes care of the deficiency both in rectangular and hexagon, in such a way that there is neither pressure drop like rectangular tube nor high pressure drop like hexagon tube.

Therefore, in the light of the above mentioned illustrations, it is arrived that oval model is preferable to rectangular and hexagon.

Abbreviations and Acronyms:

- M- Mass flow rate in Kg/s
- m_{flux} -mass flux in Kg/m^2s
- x -Vapor quality
- ρ -Density in Kg/m^3
- ρl -Liquid density in Kg/m^3
- ρg -Vapor density in Kg/m^3
- ρH-Homogeneous density in Kg/m^3
- Dh-Hydraulic diameter in m
- Re_L-Reynolds number for liquid
- Re_G-Reynolds number for vapor
- Re_f -Film Reynolds number
- F_{Lo}-Friction factor for liquid
- F_{Go}-Friction factor for vapor
- µl -Dynamic viscosity for liquid in Ns/m^2
- µg-Dynamic viscosity for vapor in Ns/m^2

- $(\frac{dp}{dz})$ Lo -Frictional pressure drop in $\mathrm{Pa/m}$
- Fr, We, E, F, H-are Fried (1979) correlation dimension less number
- ϕ_L^2 -Pressure drop multiplication factor
- τ -Wall shear stress in $\mathrm{N/m^2}$
- δ^+ and T_d^+ -are dimensionless parameter in the correlation
- $\Pr_L$ -Prandel number for liquid

  h_r-refrigerant side heat transfer coefficient in $\mathrm{W/m^2K}$
- A_f-Frontal flow area of the tube (cut sectional area) in $\mathrm{m^2}$

REFERENCES

[1] Kays, W.M. and London**, A.L.** Compact Heat exchangers, McGraw-Hill, New York (1984)

[2] Achaichia A., Cowell T. A., 1988, Heat Transfer and Pressure Drop Characteristics of Flat Tube and Louvered Plate Fin Surfaces. *Experimental Thermal and Fluid Science*, vol. 1: p. 147-157.

[3] Atkinson K. N., Drakulin R., Heikal M. R., Cowell T. A., 1998, Two- and Three-Dimensional Numerical Models of Flow and Heat Transfer over Louver Fin Arrays in Compact Heat Exchangers, *International Journal of Heat Mass Transfer*, vol.41: p. 4063-4080.

[4] Chang Y. J., Wang C. C., 1997, A Generalized Heat Transfer Correlation for Louver Fin Geometry, *International Journal of Heat Mass Transfer*, vol. 40: p. 533-544.

[5] Tafti D.K., Wang G., Lin. W., Flow transition in a multilouvered fin array. *International Journal of Heat and Mass Transfer* 2000, *43, 901-919.*

[6] LiangYe, Ming Wei Tong , Xin Zeng (2009) , 'Design and analysis of multi Parallel Pass Condensers' , International Journal Of Refrigeration 32 II53- II6I