Journal of Aeronautical and Automotive Engineering (JAAE)
p-ISSN: 2393-8579; e-ISSN: 2393-8587; Volume 3, Issue 1; January-March, 2016 pp. 33-37

http://www.krishisanskriti.org/Publication.html

Performance of Micro-channel Heat Exchanger in Automotive Radiator

Manoj Kr. Gupta[1] and R.R. Sahoo[2]

[1]*Indian institute of Technology (BHU) Varanasi, Uttar Pradesh*
[2]*Indian institute of Technology (BHU)*
Varanasi, Uttar Pradesh
E-mail: [1]manoj.kgupta.mec14@itbhu.ac.in, [2]rrs_iitbhu@rediffmail.com

Abstract—*Micro-channel heat exchanger (MCHX) has been used in Heating, Ventilation, and Air Conditioning & Refrigeration field due to its higher efficiency heat transfer rate, more compact structure, and lower cost. In this paper Numerical simulation is performed on automotive radiator with micro-channel heat exchanger to calculate pressure drop, pumping power, heat transfer rate, cooling capacity etc. with variation in mass flow rate of water and air .The working fluid which is used in automotive radiator is water and air.*
In this automotive radiator calculated Cooling capacity is around 200 % higher as compare to conventional automotive radiator. Calculated fan power is much less as compare to conventional automotive radiator because of very low pressure drop in air side.

Keywords*: Micro-channel heat exchanger, Numerical simulation, automotive radiator*

1. INTRODUCTION

Heat exchangers play a very important role in energy conversion, transmission, and utilization. Energy efficient heat exchangers can save a large amount of energy by improving power conversion efficiency, this can lead to reductions in the size, cost and greenhouse gas emissions. Energy consumption by the HVAC (heating, ventilation and air conditioning) sector in most industrialized countries accounts for one-third of total energy consumption [1].Research is being conducted throughout the world with a view to providing energy efficient, compact, and less expensive heat exchangers. A heat exchanger consists of heat transfer elements such as a core or a matrix containing the heat transfer area, and fluid distribution elements such as headers, manifolds, tanks, nozzles, or pipes. The surface area, which is in direct contact with both fluids, is called the primary or direct surface. The extended surface area is called the extended, secondary, or indirect surface [2].

Heat exchangers can be classified in many ways by geometric conFig. uration of the heat transfer surface, flow arrangement, and other considerations. Two considerations are always taken into account, in addition to other factors, in heat exchanger design compactness and heat transfer rate. Conventional compact heat exchangers having an area density of about $700 m^2/m^3$ is prepared by extending the heat transfer area, but extension of the heating surface area is limited. As such, it does not meet increasing heat transfer demands very effectively. Over the years, a lot of research has been conducted proposing numerous suggestions for increasing the heat transfer rate. Tuckerman and Pease (1981) introduced a micro-channel heat sink for removal of heat from integrated circuits. Narrow flow passages, especially micro or meso scale, have been popular among researchers in recent decades. As $Nu = hD_h/k_f$, for fully developed laminar flow in duct with a constant cross-sectional area, an increase in "h" can be achieved by reducing D_h (micro-channels) or increasing the thermal conductivity of fluid, k_f (nano-fluids, higher thermal conductivity than base fluids). But nano-fluids are very costly and are hazardous. Reducing the hydraulic diameter (D_h) is a possible means of increasing the heat transfer coefficient (h value). Thus, micro-channel ($D_h \leq 1mm$) is considered to be a promising technology in the heat exchanger field. Modern manufacturing industries are capable of producing micro-channels to facilitate the mass production of micro-channel heat exchangers. Micro-channel heat exchangers have achieved great attention in the research field due to their much higher area density, lower fluid quantity, very compact size, and higher heat transfer rate and energy efficiency.

2. MATHEMATICAL MODELING

2.1 Assumptions

- Steady state heat dissipation.
- Uniform heat transfer coefficient over the full face surface.
- Thickness of fin is small compared to other dimensions.
- The heat transfer along the length is negligible.
- Temperature and velocity at the entrance of the radiator core on both air and coolant sides are uniform.
- There are no phase changes in all fluid streams.
- Properties of coolant as well as air assumed to be constant and considered on average temperature.

2.2 Air side calculations

a. Air heat capacity rate,

$$C_a = W_a c_{p,a}$$

b. Core mass velocity of air,

$$G_a = \frac{W_a}{A_{fr}\sigma_a}$$

c. Heat transfer coefficient,

$$h_a = \frac{j_a G_a c_{p,a}}{Pr_a^{2/3}}$$

d. Correlation for the calburn factor for plate fin,

$$j_a = \frac{0.174}{Re_a^{0.383}}$$

e. Reynolds number expression for plate,

$$Re_a = \frac{G_a D_{ha}}{\mu_a}$$

Where air velocity is given by

$$u_a = \frac{G_a}{\rho_a}$$

2.3 Water side calculations

$h_w = Nu_w k_w / D_{h,w}$

$Nu_w = 0.023 Re_w^{0.8} Pr_w^{0.3}$

$Re_w = G_w D_{hw} / \mu_w$

$Pr_w = \mu_w C_{pw} / k_w$

(a) Heat capacity rate , C_w can be expressed as :

$$C_w = W_w C_{pw}$$

(b) Heat exchanger effectiveness for cross-flow unmixed fluid, ε can be expressed as Eq.:

$$\varepsilon = 1 - \exp[C^*](NTU)^{0.22}[\exp[-C^*(NTU)^{0.78}]-1]$$

Where, $C^* = C_{minimum}/C_{maximun}$

$$NTU = U_a A_{fr,a}/C_{min}$$

(c) Overall heat transfer coefficient, based on air side can be expressed as Eq. where wall resistance and fouling factors are neglected.

$$1/U_a = (1/h_a) + (1/h_w)$$

(d) Pressure drop can be expressed as :

$$DP_w = 2G_w^2 f_w H / (\rho_w D_{h,w})$$

Where,

$$f_w = 64/(Re_w)$$

(e) Pumping Power can be expressed as : $P = V_w * DP$

(f) Total heat transfer rate can be expressed as :

$$Q = \varepsilon C_{min}(T_{w,in} - T_{a,in})$$

2.4 Specification of automotive radiator

The micro-channel heat exchanger in the current study is made up of fifteen extruded multiport aluminum slabs having 68 circular channels in each slab which is shown in the Fig. 2.2. Each slab (Fig. 2.1) has 2 mm height and channel's inner circular diameter set at 1 mm, and the airside frontal area is 304 mm x 304 mm.

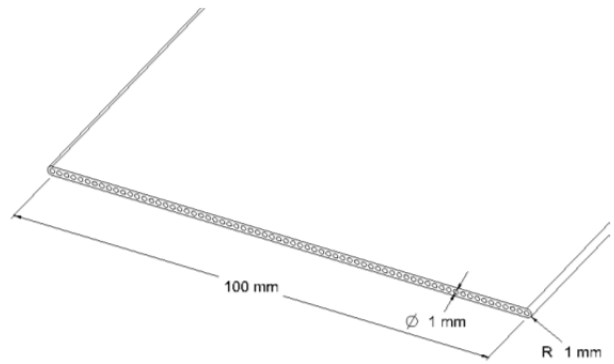

Fig. 2.1: Single micro-channel heat exchanger slab

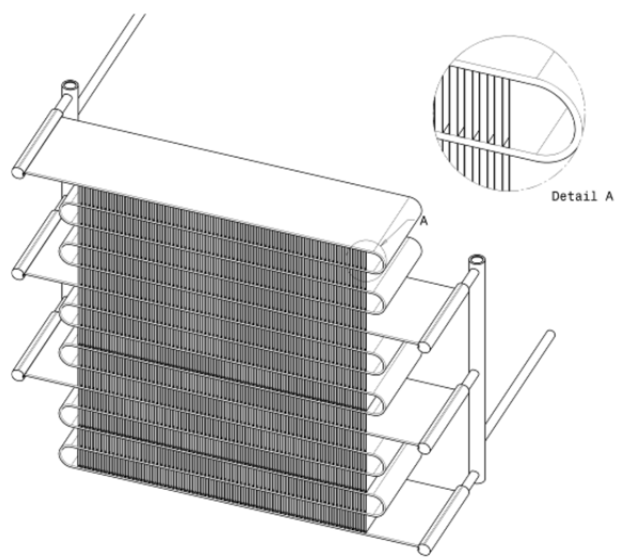

Fig. 2.2[4]: Micro-channel heat exchanger core

Table 2.1: Specifications of MCHX core (air-side)

Hydraulic diameter ($D_{h,a}$)	3.49×10^{-3} m
Fin type	Rectangular
Fin height (F_h)	16×10^{-3} m

Fin spacing (F_s)	2×10^{-3} m
Fin thickness (F_t)	0.1×10^{-3} m
Fin density (F_d)	12 fins per 25.4mm
Minimum free flow area ($A_{min,a}$)	70.9×10^{-3} m
Frontal size ($A_{fr,a}$)	0.304m × 0.304m
Air flow Length (L_a)	100×10^{-3} m
Total fin area ($A_{f,a}$)	$7.88m^2$
Total heat transfer area in air side (A_a)	$8.13m^2$
Contraction ratio (σ_a)	0.818

Table 2.2: Specifications of MCHX (liquid side)

No. of slabs (N_{MS})	15
No. of channels per slab (N_{CS})	68
Micro channel diameter ($D_{MS}=D_{h,w}$)	1×10^{-3}m
Micro channel slab height (S_h)	2×10^{-3}m
No. of flow circuits	3
No. of flow passes per circuit	5
Total flow passes	15
Total heat transfer area in water-side (A_w)	**2.3** $\times 10^{-3}m^2$

3. RESULTS AND DISCUSSIONS

3.1 Effect of coolant mass flow rate and air velocity on cooling capacity

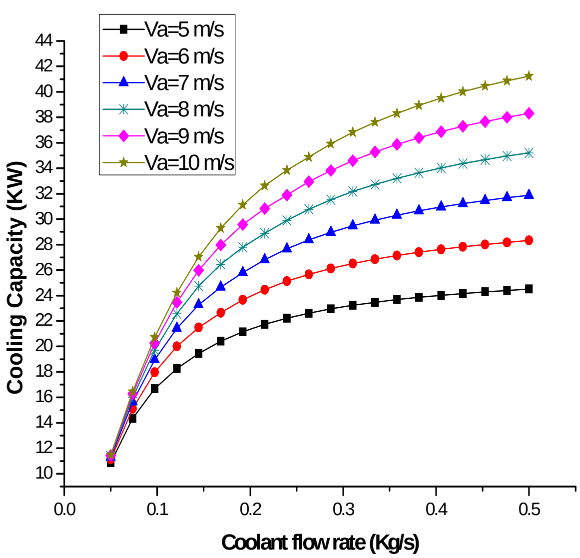

Fig. 3.1: Effect of coolant flow rate on cooling capacity

Fig. 3.1 shows how the cooling capacity increases with coolant flow rate. Cooling capacity also vary with air velocity.(5-10 m/s).with increase in air velocity and mass flow rate of coolant, there is enhancement in overall heat transfer coefficient which is responsible for increase in cooling capacity.

3.2 Effect of coolant flow rate on coolant side pumping power and pressure drop

Fig. 3.2 represents the variation of pressure drop across heat exchanger with coolant flow rate. From this Fig. pressure drop is increased with increase in mass flow rate of coolant which is due to increase in friction factor. Due to increase in pressure drop with coolant flow rate, pumping power increases in the same manner.

In this automotive radiator coolant side pressure drop is much higher than conventional radiator. Air velocity has no effect on coolant pressure drop.

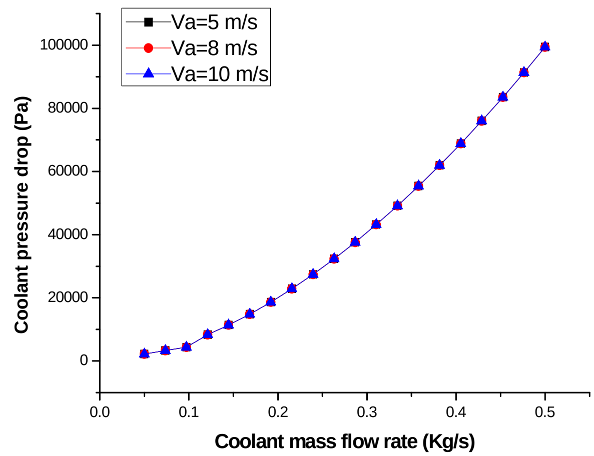

Fig. 3.2: Effect of coolant flow rate on coolant side pressure drop

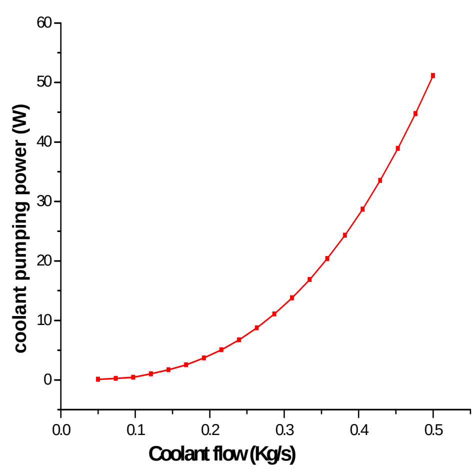

Fig. 3.3: Effect of coolant flow rate on coolant side pumping power

3.3 Effect of coolant flow rate and coolant Reynolds number on Overall heat transfer coefficient

Fig. 3.4 &3.5 shows that overall heat transfer coefficient increases with coolant flow rate and also with coolant side Reynolds number. It is because of coolant side heat transfer coefficient is highly increased. There is no effect of air velocity on overall heat transfer coefficient with coolant flow rate.

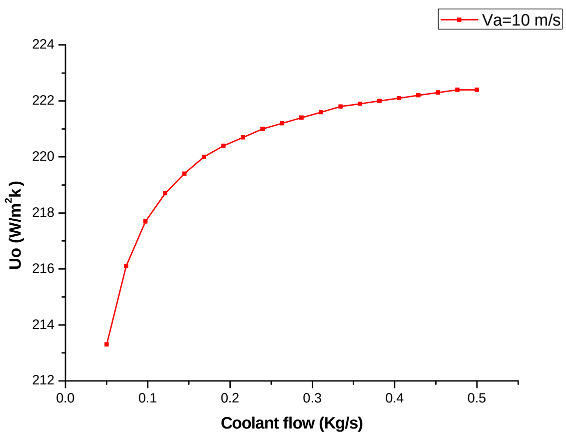

Fig. 3.4: Variation of overall heat transfer coefficient with coolant flow rate

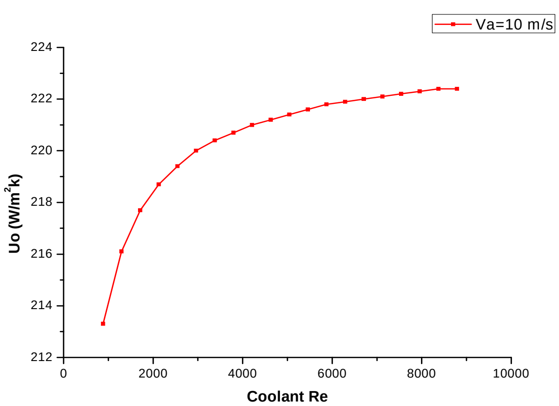

Fig. 3.5: Variation of overall heat transfer coefficient with coolant Reynolds number

3.4 Effect of air velocity on cooling capacity

Fig. 3.6 shows variation of cooling capacity with air velocity and coolant flow rate. Cooling capacity is high for higher value of air velocity and also for higher value of coolant mass flow rate.

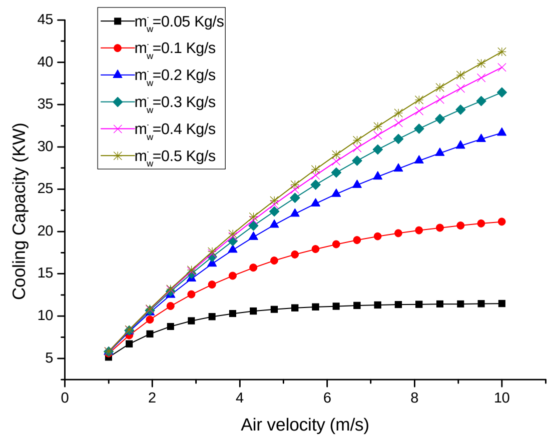

Fig. 3.6: Variation of heat transfer rate with air velocity

3.5 Effect of Air pressure drop with air velocity

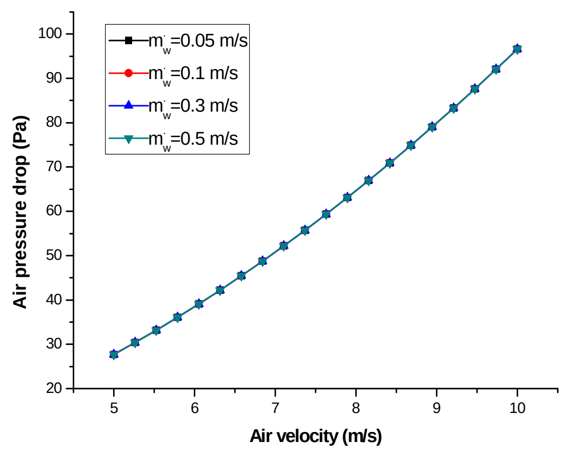

Fig. 3.7: Variation of air side pressure drop with air velocity

Fig. 3.7 represent the variation of air pressure drop with air velocity. With increase in air flow rate friction factor increases and due to which air side pressure drop is also increases.

3.6 Effect of air velocity on fan power

Fig. 3.8 represent the variation of fan power with air flow rate. It shows that fan power increases with air velocity. Friction factor at air side increases with air velocity, due to which air pressure drop (Fig. 3.7) is, also increases and resultant to this fan power increases.

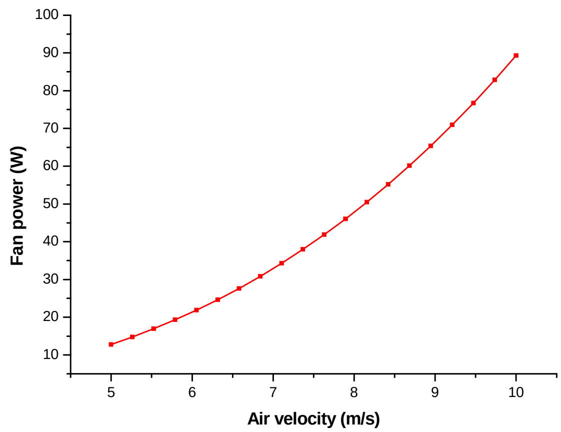

Fig. 3.8: Variation of fan power with air velocity

3.7 Effect of air velocity on overall heat transfer coefficient

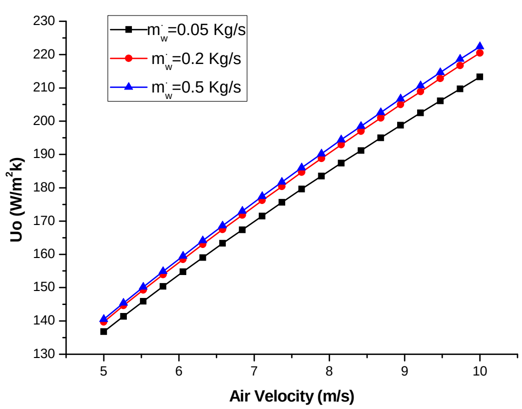

Fig. 3.9: variation of overall heat transfer coefficient with air velocity

Fig. 3.9 represents that overall heat transfer coefficient linearly increases with air velocity. There is a slight variation in Uo with mass flow rate of coolant.

4. CONCLUSION

The aim of this study was to determine the heat transfer, and key parameters of micro-channel heat exchanger performance such as heat transfer rates, Nusselt numbers, Reynolds numbers, friction factors, overall heat transfer coefficient, effectiveness, and NTU. The effects of air and water flow rate on these key parameters were examined.

- Cooling capacity increases (11.5 to 41.5 KW) with the increase in coolant flow rate (0.05 to 0.5 Kg/s). In this radiator cooling capacity is around 200% higher as compare to conventional automotive radiator.
- Coolant Heat transfer coefficient is increases (4.3-27.5 KW/m^2k) with increase in coolant flow rate (0.05 to 0.5 kg/s).
- Overall heat transfer coefficient is increases (140-225 W/m^2k) with increase in air velocity (5 to 10 m/s). In this radiator U is slightly high as compare to conventional automotive radiator.
- Coolant Pressure drop and pumping power (0.12-52 W) required increases with the increase in water mass flow rate. In this radiator coolant side pressure drop is around 400% more as compare to conventional automotive radiator.
- Air side pressure drop (28 to 97 Pa) increases with increase in air velocity (5 to 10 m/s). This is much less as compare to conventional automotive radiator.

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