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Impact of Channel Geometries and Flow Patterns On Micro-Channel Heat Sink Performance

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***Abstract:** Demand for greater capability of electronic devices in very small volume for compactness has affected huge augmentations in heat indulgence at all stages of device, electronic wrapping, test section and system. Latest cooling systems are hence needed to eliminate the released heat while maintaining compactness of the device. The micro-channel heat sink (MCHS) is ideal for this situation as it consists of channels of micron size which offers an extended surface area to volume ratio of approximately 15.294 m^2 / m^3 compared to 650 m^2 / m^3 for a typical heat compact exchanger. A comprehensive review has been done for consequence of heat flux (q^o), mass flux (G), vapour quality (x) and channel geometries at flow patterns and heat dissipation of MCHS. The study show that to increase the rate of heat transfer by using different channel geometries like converging-diverging, segmented etc. compared to conventional rectangular micro-channels has given better cooling effect The Flow patterns like bubbly, slug flow are associated with nucleate boiling dominated in low vapour quality and annular flow also given the significant effect on heat transfer in higher vapour quality region.*

***Keywords:** Flow Patterns, Micro-channel geometries, Nucleate boiling, Heat Transfer Coefficient (h).*

I. INTRODUCTION

Latest technology prompts the issue of high heat flux due to the reduced size of devices. Therefore, there is a need to eliminate high heat generated from closely packed devices such as supercomputer chips, nuclear fission & fusion reactors, laser diodes for their persistent long-life execution. Heat flux drainage varies according to different applications. For compact integrated circuits (ICs) & laser mirror the. highest, ,value of qo, recorded, is 102 W/cm2 [1], for aeronautics, and VLSI ,,,103 W/cm2 required to be dissipated[1] and, for fusion reactor, and defense applications, 104 W/cm2 , ,,heat flux , removal , is recorded,[2]. A MCHS is an right choice to dissolve heat fluxes up to 103 W / cm2 [1] as it provides a heat rejection area to volume ratio of approximately 15,294 m2 / m3 compared to 650 m2 / m3 for a closely packed heat exchanger [3]. Micro- channels act as flow paths for the working fluid in dimensions ranging from 10 to 1000 micrometres [4]. MCHS can be considered as one stage and two phases. In a mono-phase MCHS rrefrigerant sustain its mono-phase liquid state throughout for a constant flow of heat and a high flow rate. However in a two-phase MCHS fluid approaches its boiling point at low flow rate. Two-stage MCHS is ideal for eliminating large amounts of heat in a narrow space which is needed in latest applications. Two Phase MCHS offers the advantages of channel dimensions and the boiling of the fluid. In addition to this convection during boiling requires less pumping capacity & less flow rate to blow away equal heat in comparison to single-phase. The cooling fluid when flows through a MCHS follows different flow patterns, such as a bubble flow [5-12], a flow of mucus [5-7, 10-14], a ring flow, an outflow flow [10-11], a plug flow [12- 13], wavy flow [15], stratified flow [15], etc. as demonstrated in Fig.1 that affected the heat transfer of MCHS. Flow patterns in different channel geometries other than conventional rectangular micro-channels are different, which has a remarkable influence on heat transfer of MCHS. An inclusive review of the flow patterns and impact of different channel geometries on heat transfer & flow patterns of the MCHS has been carried out.

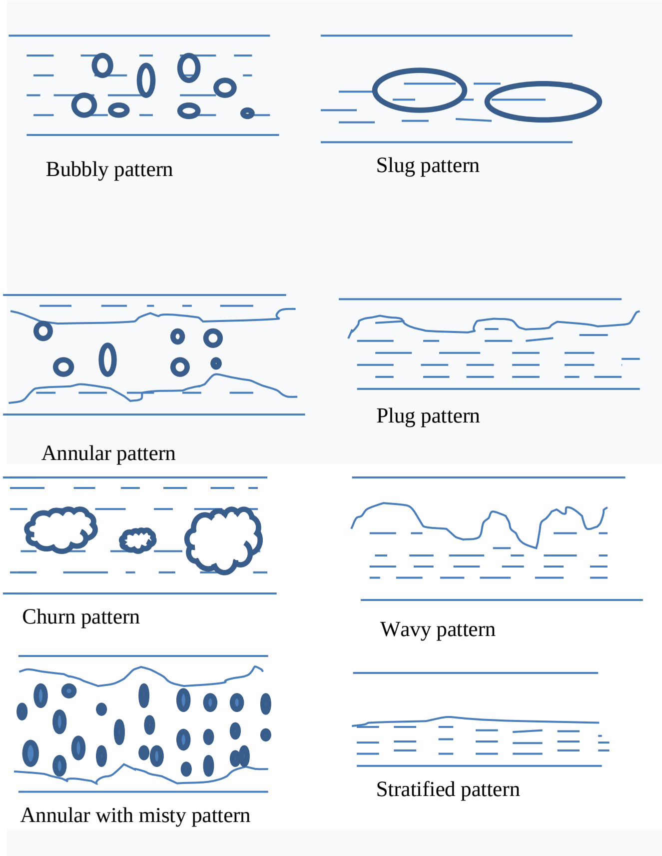

Fig. 1 various flow patterns appeared during boiling of working fluid in MCHS [1]

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II. IMPACT OF HEAT FLUXES, MASS FLUXES & VAPOUR QUALITY ON FLOW PATTERN AND HEAT TRANSFER IN MCHS

Qu & Mudawar, 2003, specified an unforeseen shift to annular flow close to zero vapour quality and bare that the presiding mechanism of heat transfer in forced convection evaporation analogous to the annular flow for water flow at G = 135 to 402 kg/m^2s & saturation temperature (T_{sat}) = 30 ° C, 60 ° C, and at the P_{out} = 1.17 bar [6], which can be clearly seen in Fig. 2. T. Dong et al. 2008 saw a similar trend for the hot R-141b flow in 50 parallel rectangular channels of 60 µm width and 200 µm of boiling nucleation depth dominated by low vapour quality, while at high-temperature convective steam vapour dominates. Thus when x was less than 0.4 flow may not develop to a complete ring flow [18], as shown in Figure 3. Megahed and Hassan, 2009 conducted tests for 21 straight rectangular channels using FC-72 as coolant. While Testing for G = 341 - 531 kg / m^2s and q^o = 60.4 to 130.6 kW / m^2 they noted three major patterns bubbly, slug & annular flow [19]. Alam et al. 2013 noted that heat transfer on the micro-gap is much better in lower G & high q^o condition and because of slug and annular boiling supremacy during flow boiling of deionized water [20]. Tamanna and Lee, 2015 have observed that few bubbles nucleate, separates and outspread quickly & lastly glides on the heated area at low value of q^o for the straight micro-gap heat sink. When heat flux surges fluid between slug shrinks & expanded bubble occupies micro-gap spaces which establish annular flow during the flow of deionized water [21]. Kuznetsov and Shamirzaev, 2010 observed that during boiling of R-21 in 10 microchannels heat transfer enhances with increment in heat flux and the dominating mechanism was nucleate boiling for bubble flow. At higher value of q^o saturation occurs and heat transfer remains constant in slug and intermittent regimes. For intermittent regimes, the heat transfer coefficient vaguely dependens on steam quality whereas it declines with increment of steam quality in the annular flow region [7]. Hsu et al. 2015, tested the two phase boiling of HFE7100 for 440µm hydraulic diameter inclined multiport micro channels. Tests were carried out horizontally at G = 100-200 kg/ m^2s and inclination of -90, -45, 0, 45, 90^0. Results demonstrated that in general heat transfer coefficient of the upstream equipment is more comparable to the downstream arrangement if the delivered heat flow is 25 kW/m^2, due to the controlled convective source, but at the 40 kW/m^2 it can be observed that the nucleation source have a incredible consequence on mechanism of heat transfer [22]. Agostini et al. 2008 has executed experiments on two horizontal micro-venting ducts with tube diameters of 509 & 790 µm for the R-134a flow characteristics at G= 200 to 1500 kg / m^2s, T_{sat} = 300 ° C and at a vapour quality (x) of 2% to 19%. It has been found that the prolonged bubble velocity respective to the uniform stream enhances with increment in the length of the bubble to reach the maximum, as well as increasing the diameter of the channel and increasing the mass velocity [23]. Jang et al. 2008 has observed the heat transfer features of FC-72 in circular microchannels by changing G, $T_{sat,}$ and x. The results show that low steam quality includes both mono phase flow and nucleate boiling [24]. Impact of various parameters when fluid flows through the MCHS as observed by different authors are presented in Table 1. Most of the authors concluded that at low value of x monophase flow & bubble nucleation occurs while at high value of x corresponding to higher value of q^o convective boiling with annular pattern detected and also when value of q^o increases shifting of flow occurred from bubble to annular pattern with local dry-out.

Table- I: Flow boiling results observed by different authors at different input parameters and working fluid.

Authors	Fluid	MCHS geometry (W,H,D, & D_h) (µm)	Input parameters	Flow Results
Kim and Mudawar, 2012 [5].	HFE7100.	rectangular channels W= (123.4, 123.4, 235.2, 259.9), H= (304.9, 526.9, 576.8, 1041.3), N= (24,24,11,11).	G=(2200-5550), (1280-3210), (1330-3350), and(670-1683) kg/m^2s , q^o= (0-560), (0-580), (0-640), (0-664) kW/m^2 P_{out}= 1.138 bar.	Bubble, slug & annular pattern.
Qu and Mudawar, 2003 [6].	Deionized water.	Rectangular channels W= 231, H=713, N=21.	G= 135-402 kg/m^2s, T_{in}= 30^0C and 60^0C, P_{out}= 1.17 bar	Nucleate boiling: bubble & slug flow. forced convection boiling: annular flow.
Hardt et al. 2007 [29].	2-Propanol and water.	Square channels COP channels (50x50), Silicon channels (30x30) .	T_{in} = 82^0C and 93^0C, V_{in}= 0.5-1 m/s.	COP channels: film evaporation with continuous acceleration, film evaporation preceded by eruptive evaporation. Silicon channels: Stationary menisci, Parallel oscillations, chaotic oscillations and nucleate boiling.
Chang and Pan, 2007 [8].	Deionized water.	Rectangular channels D_h= 86.3µm, N=15.	Flow rate (Q)= 0.01 to 10 ml/min.	For Stable case: bubble, slug & slug or annular Patterns. For Unstable case: onward or inverted slug/annular flows.
Alam et al. 2013 [20].	Deionized water.	Rectangular D_h= 190.34 &, 385.70.	Saturation temperature(T_{sat}) = 86^0C, G= 400-1000 kg/m^2s and q^o= 0-85 W/cm^2.	Curbed slug and annular boiling dominance.

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Tamanna and Lee, 2015 [21].	Deionized water.	Depth at inlet= 200 µm & expanding depth= 300 & 460µm.	G= 400-1000 kg/m^{2}s and q^o= 0-85 W/cm^2.	Bubbles nucleate, detach and expand quickly at low heat flux for straight microgap. Confined annular flow pattern for high heat flux.
Agostini et al. 2008 [23].	R-134a.	D= 509 & 790.	G= 200 to 1500 kg/m^{2}s, x= 0.02-0.19, $T_{sat=}$ 30^{0}C.	Elongated bubbles.
Jang et al. 2008 [24].	FC-72.	Circular channels.		Low vapour qualities associated with monophase phase flow and bubble nucleation.
Deng et al. 2015 [38].	Ethanol.	Rectangular re-entrant channels D_h=786.	T_{insub}= 10^{0}C and 40^{0}C, G= 125, 200, and 300 kg/m^{2}s.	Flow patterns shifted from bubble to slug flow, intermittent flow & annular flow also local dry-out ensued as the heat fluxes increased.
Shafeie et al. 2013 [26].	Water.	Cylindrical, D=80, H= 90,180, and 500.	Re_{max}= 200 and 1000.	Forced convection boiling.
Hsu et al. 2015 [22].	HFE 7100.	Inclined channels, D_h= 440µm, inclination with horizontal= -90, -45, 0, 45, 90^0.	G= 100-200 kg/m^{2}s.	Nucleate boiling.
Keepaiboon and Wongwises, 2015 [11].	R-134a.	Single rectangular channel, D_h= 680.	q= 7.63–49.46 kW/m^2, G= 600–1400 kg/m^{2}s, and T_s= 23–31^{0}C.	Six different patterns are dictated: bubble , bubble-slug , slug , throat-annular, churn & annular flow.
Dong et al. 2008 [18].	R-141b.	Rectangular channels, (W,H)= (60,20), N=50.	G= 400-980 kg/m^{2}s and q=40-700 kW/m^2.	x< 0.4, the flow might not grows to a complete annular pattern.
Choi and Kim, 2009 [12].	Water and Nitrogen-gas.	Five different types of rectangular channels (D_h=141, 143, 304, 322 and 490).	Liquid velocity (V_L)= 0.06-1.0 m/s and Gas velocity (V_G)= 0.06-72 m/s	Bubbly flow, slug bubble flow, elongated bubble flow, transition flow and liquid ring flow.

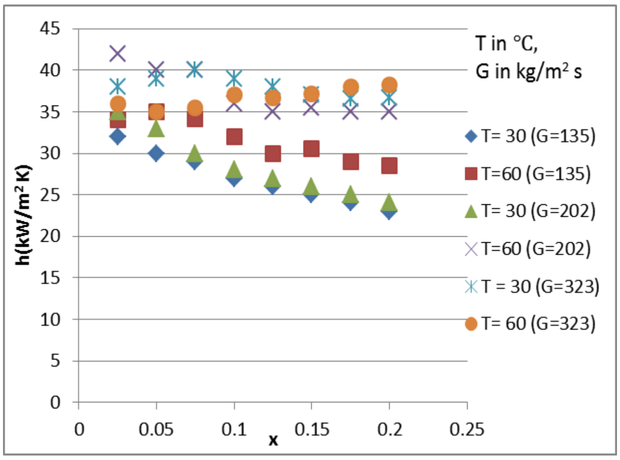

Fig. 2 Influence of vapour quality on heat transfer coefficient at different mass fluxes [5].

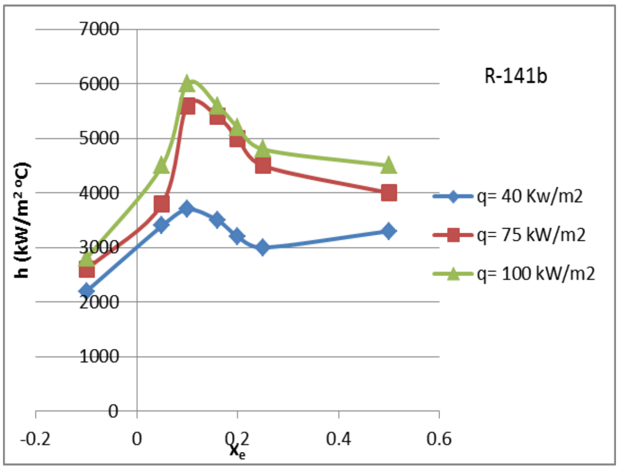

Fig. 3 Effect of vapour quality on heat transfer coefficient at different heat fluxes [18].

III. CHANNEL GEOMETRIES EFFECT ON FLOW PATTERN & HEAT TRANSFER IN THE MCHS

Kroeker et al. 2004, has observed that circular channels dissipate more heat when compared to rectangular micro-channels with same Reynolds number & their hydraulic diameter (D_h) [25]. The physical and mathematical model developed by Na and Chung, 2011, to forecast the behaviour of two-phase water flow in a MCHS which consists of circular channels has observed an annular flow pattern for uniform heat flow condition [26]. Shafeie et al. 2013, has observed that heat dissipation of finned MCHS is less to optimal straight MCHS at medium and high pumping power under laminar forced convection of water. However, finned radiators work slightly better than optimal straight MCHS for low pumping power [27]. Liu and al. 2011 has observed the heat dissipation of high pin fins with Re = 60 to 800 for demineralized water. Researches were carried out for micro square pin fin cross- sectional area of 559 µm × 559 µm and 445 µm × 445 µm . Results revealed that the heat transfer rate could approach 299.98 W / m^2 at 57.225 L / h flow rate and at an elevated temperature of 73.4^{0}C for 445 µm × 445 µm, [28]. Hardt et al. 2007 audited the process of infection of cyclo olefin polymers (COP) and square silicon micro-systems using 2-Propanol and water as working fluids. COP channels of 50 µm × 50 µm were flat, and silicon billets of (30 µm × 30 µm) were rough. Two different modes are considered for COP channels: first, film vaporization with uniform acceleration of the meniscus and other is the film vaporization pre-exist by uncontrolled vaporization. In silicon micro-channels, four different types of modes are identified: stationary menisci, parallel oscillations, chaotic oscillation, and nucleate boiling [29]. Peles et al. 2005 observed that minimum heat resistance can be achieved when a pin fin heat sink is used when compared to traditional MCHS [30]. Prajapati et al. 2015 have juxtapose the features of dimeneralized water boiling in three different geometries of micro-channels (uniform, divergent & segmented channels). The tests are carried out at a sub-cooling liquid state of the inlet with a mass of the coolant, and the heat flows vary in the range of G = 100 to 350 kg/m^2 s & q^o = 10 to 350 kW/m^2, respectively. Outcomes revealed that segmented channels display highest value of h compared to other two channel configurations due to higher stability found in segmented channels for overall range of G [31]. which can be observed from Fig. 4 Chakravarthii et al. 2017 found that converging diverging microchannels maintain higher value of h compared to straight microchannels because recirculating zones in converging diverging microchannel increased the disturbance of solids boundary in channel surface and increases value of h for monophasic liquid flow with G = 001232 to 0.01848 kg/s and q^o =10-50 W/cm^2 [32]. Dehghan et al. 2015 checked that the Nu and convection heat transfer increases with tapering. MCHS has the optimal heat transfer at the width to tapered width ratio equal to 0.5 applying finite volume methods (FVM) in the laminar region [33]. Megahed, 2011, investigations attributed the characteristics of FC-72 to 45 cross-linked micro-channels with D_h = 248μm over a value of G = 99-290 kg/m^2s, q^o = 7.2-104.2 kW/m^2 and x = 0.01-0.71. it has been observed that the first regime is the slug, the cross-linking pathways also show that the bubbles nucleated and propagated on cross-links surface in tangential direction at inlet because effect of circulation created in those regions [34]. Zhang et al. 2013, has conducted an experiment on the lotus porous copper heat sink with water as fluid and observed, heat transfer in pores increases due to convection as water flow rate increases [35]. Hung et al. 2013 showed that with large Reynolds number and large porous configuration thermal performance could be improved using 3D models of porous MCHS with different geometries such as rectangle, outlet expansion, trapezoidal, thin rectangular blocks and sandwiches type, etc. [36]. Alfaryjat et al. 2014 has investigated numerically the heat transfer for 26 parallel channels of 90- 130 μm x 200-300 μm in operating parameters (Tin = 17^0C, Re = 100-1000, and q^o = 500 kW/m^2). Prevailing equations and 3D steady state heat transfer applied for three channels (hexagonal, circular, and rhombus) have been solved by Finite volume method (FVM). results indicated that the hexagonal MCHS with minimum D_h has maximum pressure drop & heat transfer coefficient compared to another configurations [37]. Deng et al. 2015 showed that re-entry micro-channels (REM) show a significant increase in 2-phase h in case of high subcooling & medium to high heat flux compared to conventional rectangular micro-channels (RM) at the same D_h with two refrigerants (demineralized water and ethanol) with inlet sub-cooling 10 ° C, 40 ° C & G = 200 to 300 kg / m^2s [38]. as observed in Fig. 5 Sakanova et al. 2015 have investigated heat transfer of MCHS using the corrugated channel structure. Influence of corrugated amplitude, wave length, volumetric flow rate & fraction of various nanofluids are shown. Three different class of nanofluids have been applied with a volume concentration of 1% to 5%. It has been observed that if the purest form of water is used then MCHS heat transfer has improved significantly compared to the conventional channel [39]. Vinodhan & Rajan 2014 has carried out computational experiments for heat transfer & flow pattern in four new nnel MCHS configurations to compare it with a traditional heat sink. Higher heat transfer and Nusselt number rates were achieved in new designs due to the presence of many regions of development flow [40]. Kuppusamy et al. In 2015, studied the influence of the triangular micro-mixers adjusted between main current channels in MCHS. For simulation, a unit wall with simple MCHS and a triangular microcomputer (MTM) is selected. Results revealed that the heat transfer of MTM depends on changes in all geometric parameters [41]. Vafai & Zhu 1999 revealed that double-layer MCHS is a major improvement over a traditional single-layer MCHS because less temperature rise of the base surface has been noticed competed to a single layer heat sink [42]. Balasubramanian et al. 2011 have conducted a test for straight and expandable micro channels having same dimensions & similar working conditions using demineralized water as a refrigerant. Results demonstrated that the expanding MCHS has preferable heat transfer over the regular MCHS also heat transfer coefficient in expandable channels is nearly consistent while straight channel heat transfer coefficient shows large variation throughout the range of G due to better bubble stability [43] which is demonstrated in Fig.6. Mohammed et al. 2011 observed that the MCHS zigzag has less temperature and a higher value of h between straight, sinuous and undulating channels for the same cross-sectional area of MCHS when numerical simulations were performed using 3D model FVM for water [44].

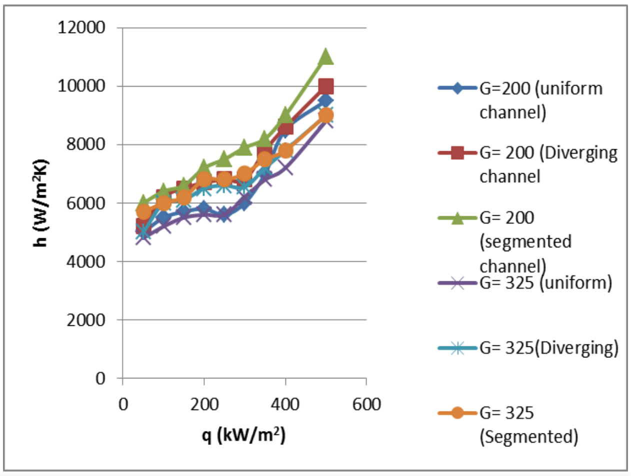

Fig. 4: Influence of heat flux on heat transfer coefficient for the uniform. Diverging & segmented channels [31].

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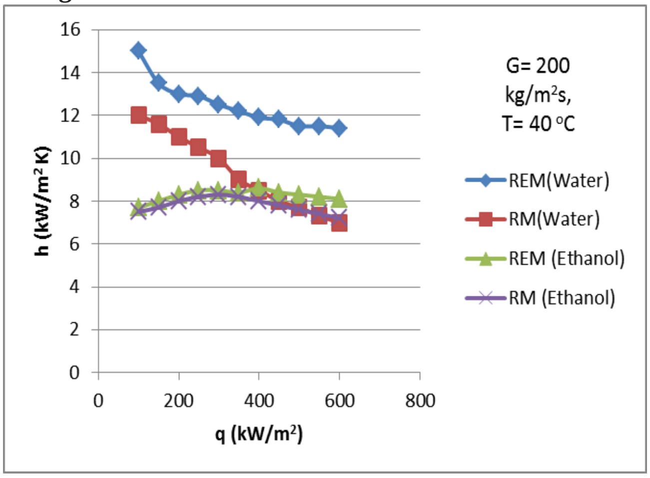
Fig.5: heat flux variation with heat transfer coefficient



for water & ethanol [43].

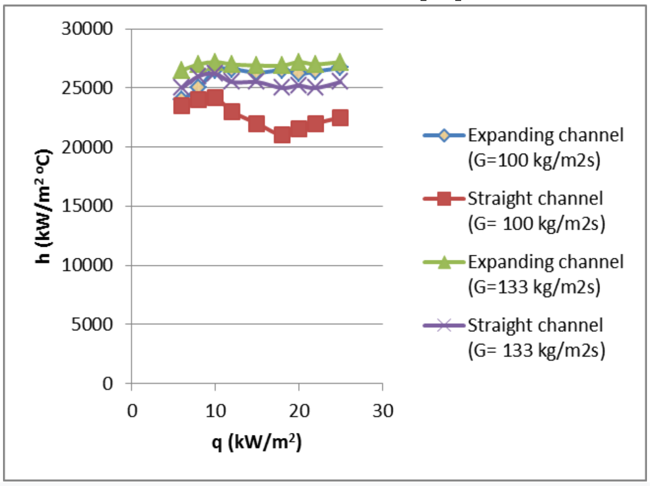


Fig.6: heat flux impact on heat transfer coefficient for straight & expanding channels [38].

IV. CONCLUSION AND FUTURE SCOPE

An inclusive review has been done on channel geometries and influence of governing parameters on flow patterns which affects heat transfer of MCHS.

- Higher value of h can be obtained by using different micro-channel geometries as approximately 37% increment found in segmented channel, 26 % in expanding channels and 50% increment in re-entrant channels. Higher value of h increases the pressure drop hence further investigations are needed to develop the correlations between pressure drop and h.
- MCHS performance can also be enhanced by using porous design and micro pin fins with larger Reynolds number because it provides lower thermal resistance. Experimental investigations are required to analyse MCHS performance by using nanofluids in porous design and micro pin fin heat sinks.

• At low value of x nucleate boiling conquered while at high value of x convective boiling dictates. Nucleate bubbling is commonly identified with bubble and slug streams though convective dissipation is commonly identified with annular stream. In nucleate boiling region h varies with heat flux but in general shows less sensitiveness to G and x and in convective flow boiling regime h is a function of G & X but independent of q°. Further investigations are needed for a wide range of x.

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