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Experimental Investigation of the Impacts of Laminar and Turbulent Impinging Jet Flows on the Convective Heat Transfer in a Metal Plate

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ABSTRACT

Dissipation of heat has become a common reason for the thermal runway and malfunction of various electronic devices. To effectively use these devices, it is mandatory to extract heat from the dissipating parts. Impinging jet flows (IJF) have been used for cooling purposes for decades. The jet impingement technique contains a pressurized fluid and is designed to spray the fluid with the help of a nozzle on the surface of the disc or plate which is being heated. However, recent advances on the effective heat transfer using metal foams are being studied. This research work focuses on the contribution of laminar and turbulent flows from IJF in extracting heat from the surface plate in the presence of geometrically diverse metal foams. The factors under consideration are H/D_j, Heat flux, thickness of the metal foams, and diameter of the metal foams. This research is purely experimental, and the experimental setup consists of mainly compressor, pressure gauges, rotameter, electric resistor, and thermocouples. The flows are controlled through rotameter and are kept in the range of Re from 800 to 2000 for laminar and from 3000 to 23000 for turbulent region. The outcomes of the results are extremely useful in engineering applications and the data can be used to design an effective heat exchanger.

1. INTRODUCTION

Thermal management has become an important task for all heat-dissipating devices to perform the optimum and processes. It will also increase durability, and reliability in performing operations. Several cooling techniques have already been in practice of which air cooled are always considered to be the most suitable in overall performance which are further categorized natural and forced convection. However, the removal of dissipated heat by convective air cooling has already acquired the highest potential of efficiency and cannot be further developed without any novel discovery of finest configuration of the setup. Thus, research is going on in various techniques including micro evaporation cooling, liquid cooled cold plates, heat pipes, spray cooling, microchannel, with nanofluids and with impinging jet flows (IJF). Given that plenty of cooling methods are available of which, impinging jet flow (IJF) has met the requirement of providing high heat removal rates, which makes it potentially a better solution for thermal management of equipment in comparison to other methods [1-4]. Similarly, most of the literature is dominated with the results related to free jet impingement application whereas, a small portion of the work is done to evaluate heat transfer in a confined jet [5].

There are several configurations of jets that are currently in use such as the use of single jets, rows of jets, arrays of jets, single slots, and rows of slots etc. The impinging of these jets on the target surface has been studied frequently. IJF is very important in terms of cooling of various technological practices such as with the cooling of solar PV collectors, turbine blades, batteries, electronic equipment etc. The main benefit of using IJF is its capacity to create highly localized heat & mass transfer rates because of producing thin boundary layer around the flow stagnation region. It is stagnation point where most of the velocity loses its momentum and gains thermal energy from the plate. IJF directs the air flow to the targeted surface with simple design, less plumbing & installations and comparatively at a lower operational cost. It also consumes less power, it is easier to control and flow the air in the direction where it is needed and it does the required job effectively [6, 7].

The flow behavior of the IJF is very well known and is explained extensively in the literature. The main attributes of the flow are. significantly There are three distinct flow regions that have been portrayed for impinging jet hydrodynamics. Figure 1 shows these three regions: (1) Region I is called Free jet region. It starts from the nozzle exit to the end of the potential core. The velocity at the potential core remains constant and equal to the nozzle exit velocity. The length of the potential core generally ranges from 6 to 7 times the diameter of the circular nozzle. (2) Region II is known as stagnation region in which the jet strikes the plate, loses its kinetic energy, and is then diverted towards the axial direction. (3) Region III is called wall jet region where the fluid moves towards axially and gradually decreases its velocity [8].

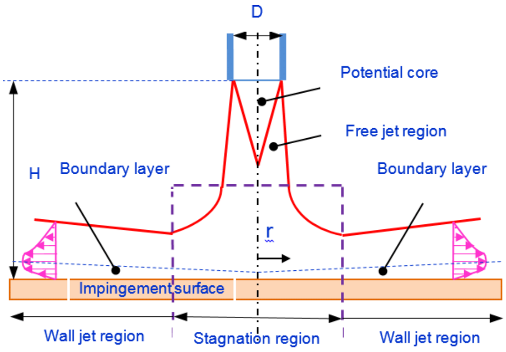


Figure 1. Thermal physical behavior of fluid flow in IJF

Laminar jets do not have the capacity to support much in the removal of the heat as it is highly dependent on the turbulency and transport rates, while laminar are in streamlines. Therefore, more of the research is dominated by the turbulent jet. It is mentioned in the literature that if IJF employed together with Porous Media (PM) with high permeability, porosity, and thermal conductibility, such as aluminum foams, can improve removal of the heat from a targeted surface The factors which affect the heat transfer rates with respect to experimental investigations includes jet geometry such as jet diameter (D_j), jet orientation, H/D_j ratio, thermal, structural, and geometrical parameters of the PM used, Reynolds number (Re), flow parameters such as heat transfer coefficient, material properties such as heat transfer coefficient, jet temperature etc.

Also, porosity, permeability, and pores per inch (PPI) of MF are important factors through which high-performance for heat transfers can be achieved. The metal foam cell structure for aluminum open foam greatly decreases the thermal resistance [9, 10]. Among MFs, aluminum (al) MF has been proven to be the best for the cooling of high-power electronics. Aluminum is considered ideal in this regard because it is abundant, cheap, and easily available. In an experiment the overall heat transfer coefficient of al-MF has proved to be about 25% greater than that of conventional finned array heat exchangers. With turbulent flows is possible to gain heat transfer enhancement of 2-4 times using porous media, and that enhancement is because of the micro turbulent mixing flows in the porous media.

To further enhance heat transfer using existing techniques for such applications, porous materials i.e. MF have appeared to be a very viable, feasible, and efficient heat dissipation addition to reach the objective. Various applications of use of porous media for heat transfer enhancement are available such as in heat exchangers and cooling [11-14]. And numerically PM has been studied for convective cooling by various scholars. Performance of thin metal foams are also subjected to the arrangement of the jets on the substrate. Some scholars reported a significant enhancement in heat transfer compared to other similar setups where single jet impingement was used or when thick blocks of foam were placed in a duct. Also, it is found that most of the heat transfer occurs in a thin layer of foam adjacent to the heated surface [15-17].

To increase effectively heat transfer phenomena and to enhance the Nu distribution over the surface plate is to control the fluid with the use of IJF for effective distribution of Nu patterns. control of fluid may avoid excessive heating or cooling also. Regarding porosity of metal foam and its significance in affecting heat transfer participation area per unit volume compared to others in convection heat transfer enhancement is recognized in the literature. At a given Re, decreasing the porosity may enhance the heat transfer. One of the experimental analysis exhibits that an increase in the pore density results in the increase in Nu. However, some authors believe that the opposite is true as this negative proportionality is also convinced by several other authors. A numerical study with PM over metal fin heat exchanger exhibits that increasing Re and decreasing porosity has led to increment in the heat transfer rate. Some authors pointed out the impact of plate attributes on heat transfer rates. The amount of time taken by fluid stays over the plate depends on the surface roughness, friction, etc. which can result in a decrease in the total interaction time of the fluid with the plate [18-20].

Since the literature is already sufficient and saturated with the use of impinging jet, however, it comes of understanding that the addition of metal foams (MFs) to enhance heat transfer is getting grounds. The heat transfer rates of IJF with PM suggest that for a certain parametric range the addition of porous media may or may not enhance heat transfer rate. Thus, this experimental research work is done to analyze various combinations of the parameters in a way to further improve heat transfer rate.

2. EXPERIMENTAL SETUP

Impinging jet flows (IJF), in this setup, works by flowing pressurized fluid, air in our case normally on the surface of the target plate which is being heated with the help of electrical resistor. It is necessary to understand and learn that how heat is being transferred on the plate. Ten J type thermocouples are fixed to the setup: five of the thermocouples are fixed to the targeted plate on different radial positions as well as at different distances from each other as shown in Figure 2.

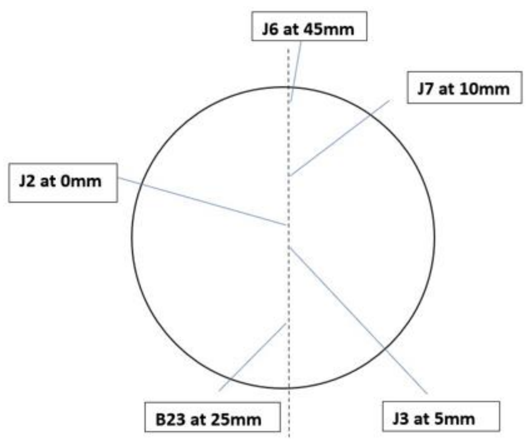


Figure 2. Radial position of the thermocouples on the target plate

While five other thermocouples are attached at other positions of the setup to understand overall heat transfer phenomena such as to measure ambient temperature, temperature before rotameter, temperature after rotameter and most importantly at the tip of the exit tube which measures temperature of the jet (T_j). Figure 3 shows the position of the metal foam at the center of the target plate of thickness 3 mm during current experiments.

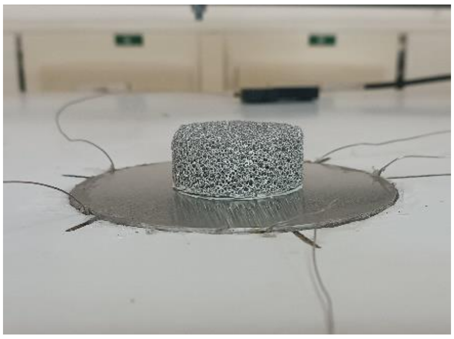


Figure 3. Position of the MF over target plate

The setup as shown in Figure 4, consists of mainly: 1) Pressure to enhance fluid pressure. 2) Pipe or duct. 3) Rotameters, which maintain and control dynamic pressure of air in the duct. 4) Pressure gauges to measure pressure and pressure difference. 5) An Aluminum Target plate which is to be cooled. 6) A long impinging jet tube of 1 meter in length normal to the plate with inlet nozzle diameter of 14 mm. 7) J type thermocouples, connected at different positions of the plate and the setup. 8) An ice point for reference junction for the measurements. 9) An AGILENT 34980 multifunction Data logger and acquisition hardware. 10) A PC, to be used to acquire and display data. 11) Agilent software to read & support data acquisition from data logger to computer. 12) An electrical resistance RS Components IT 245-590 18 Ohm powered by AGILENT E3633A to supply electric heat flux to the plate.

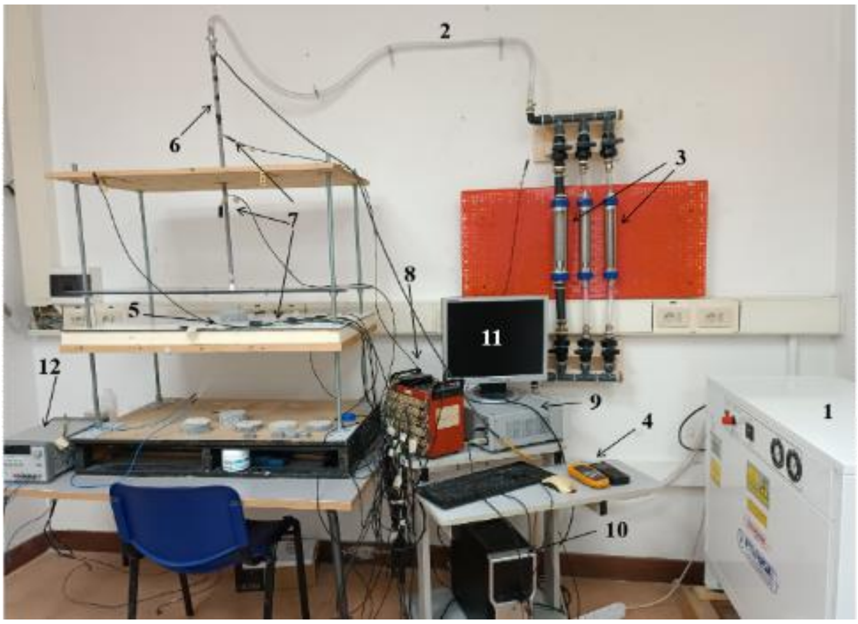
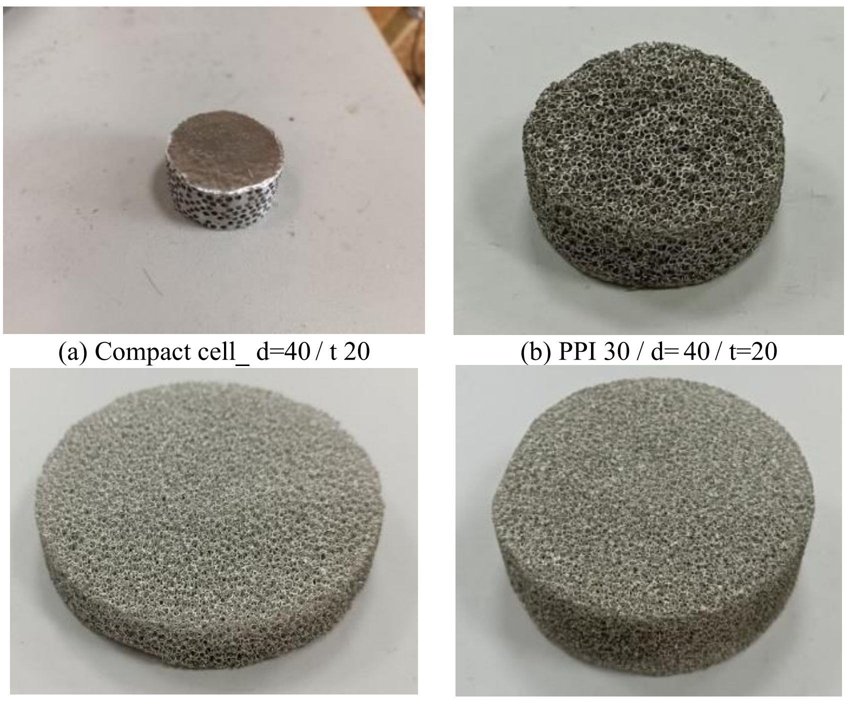


Figure 4. Experimental setup for IJF experiments with necessary equipment



(a) Compact cell_ d=40 / t 20

(b) PPI 30 / d= 40 / t=20

(c) PPI 30 / d=100 / t=20

(d) PPI 30 / d=100 / t=40

Figure 5. Different metal foams for experimental analysis

In this experiment, the kind of metal foams subjected to experiment based on their properties of pores is per inch (PPI); 30 PPI, but of different geometries as shown in Figure 5.

The thickness of metal foams is varied from 20 mm to 40 mm while the diameter is varied from 40 mm to 100 mm difference. The general parameters used in these experiments include the density of the air, viscosity of the air, thermal conductivity of the air and the range of the fluid flow as shown in Table 1 below.

Table 1. General properties of the inputs for experiments

ρ	μ	h Wm^{-2}	Laminar Region	Turbulent Region
1.15	1.87×10^{-5}	721	799-2000	3000-24000

Each metal foam is fixed with a thin film of aluminum foil for uniform temperature distribution from the plate and is attached to the plate using small amount of thermal paste to fix the metal foam with the plate. To reduce heat losses, an insulation material polystyrene block and fiberglass is attached to the surrounding and bottom heated plate to minimize heat losses.

3. METHODOLOGY

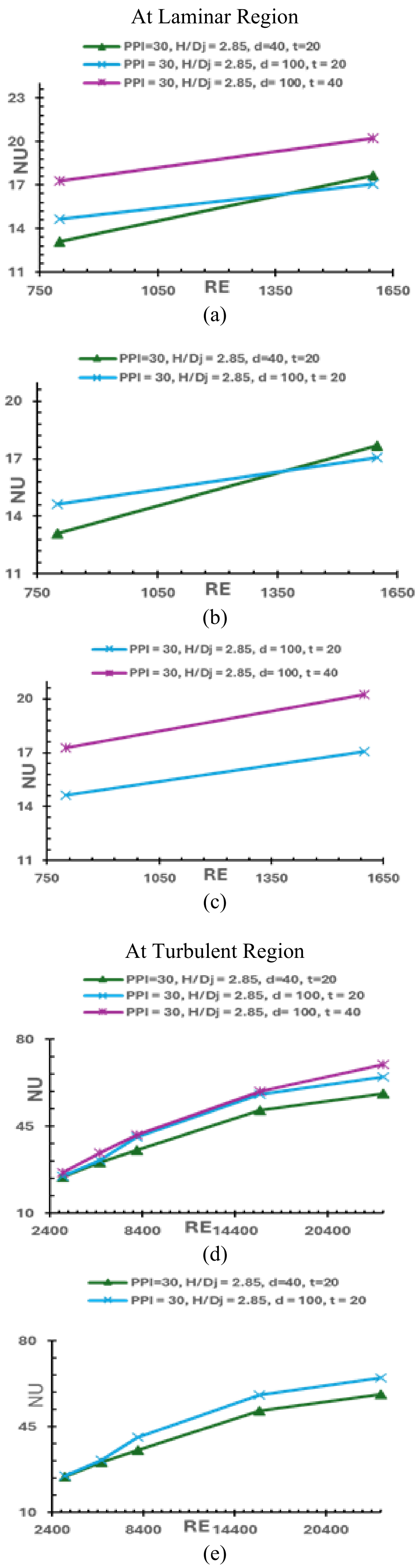
To observe the effects of geometry and structure of the MF in heat transfer coefficient using IJF, the research experiments have been classified into five diverse experimental arrangement as shown in the Table 2; each with two different values of laminar flow and as well as five different values of turbulent flows. It is possible to design different applications and augmentations as per requirement and the nature of the applications use. For example, to examine the use of a single circular jet to impinge on a surface and cool it down with and without the presence of porous media. Fluid impinges over the surface plate in a confined impinging jet configuration. The diameter of the circular inlet nozzle is D_j and Tw is the temperature at the surface plate, which provided through a constant heat flux of 721 while the reference temperature is equal to ambient, T_o. The other configuration is designed in such a way that the heated surface plate is covered with porous media of very diverse geometrical structure.

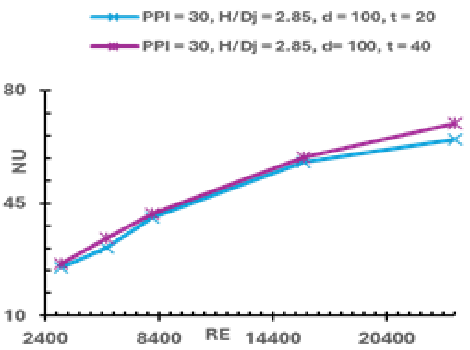
Table 2. Various configuration of experimental setup

Sr.No.	H	D_j	H/D_j	d	t	PPIs
1.	40	14	2.857	--	20	CC
2.	40	14	2.857	40	20	C_{pf}
3.	40	14	2.857	40	20	30
4.	40	14	2.857	100	20	30
5.	40	14	2.857	100	40	30

4. RESULTS AND DISCUSSIONS

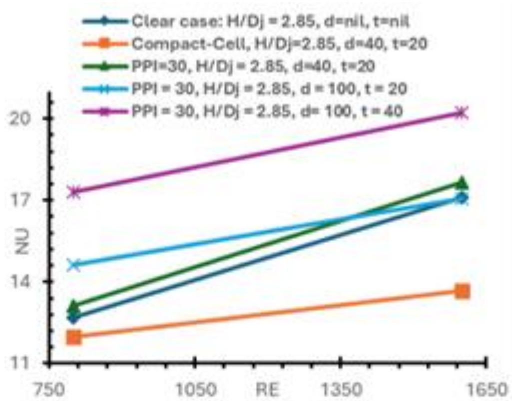
The results are categorized into three formats. According to H/D_j ration, according to d (diameter of the MF) and according to t (thickness of the MF). The results below are displayed according to flow regions. Heat transfer enhancement in the metal foams is the combination of both convective heat transfer via IJF and conductive heat transfer by MF from the target surface. Increasing the thermal conductivity ratio results in heat transfer enhancement from the hot wall.



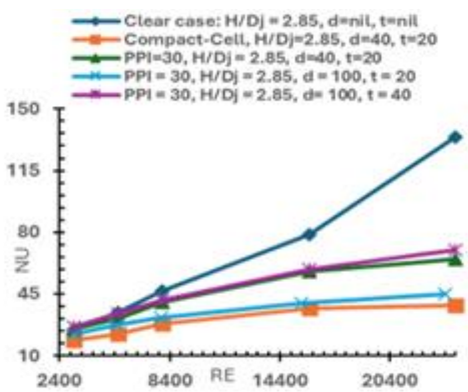


(f)

Figure 6. Results of experimental investigations



(a)



(b)

Figure 7. Overall experimental result for all the cases together

Figure 6(a) and (d) represents that generally for the given Re at the laminar phase and turbulent phase: greater the diameter would increase the average value of Nu whereas, when greater thickness of the MF used, there isn't any significant improvement in the Nu enhancement. Figure 6(b) and (e) shows the effect of diameter of metal foam on the Nu in laminar and turbulent regions while Figure 6(c) and (f) shows the impact of thickness of metal foam on Nu in the laminar and turbulent flows. This is because turbulent stage in, the greater thickness of the MF causes resistance in flow while greater diameter provides greater area of heat dispersion from the hot metal plate. However, if thickness of the MF increases may not increase the Nu at turbulent while at laminar flow it enhances Nu a bit. It means that at laminar stage when the flow is streamlined, the flow could easily pass through the pores of the MF even with that of larger thickness and could greatly extract the heat from the metal foam pores. Because in general increasing the pore density could increase the Nu.

Overall, MF of 30 PPI with 100 mm diameter and 40 mm thickness has greater value of Nu in the Laminar while the compact cell has less. On the other hand, at the turbulent stage, clear case (without MF) shows very high values of Nu while compact cell remains at the lowest as in Figure 7(a) and (b). This is because a compact cell is supposed to have less permeability as well as porosity and behaves more like a solid rather than porous which hinders heat transfer enhancement. While in clear case it is because there isn't any barrier which restrict the flow to strike the target plate directly, but this heat transfer is purely convective as there is no MF to support conducive heat transfer.

5. CONCLUSION

It is concluded that geometry of MF has definite effects on the Nu enhancement at the given conditions. It is to be noted that these results are concluded by keeping the H/D_j ratio, constant. Changing the parameters may affect the results. Finally, it is to be said that at any conditions the laminar and turbulent jets affect differently and could possibly have different results on the given conditions.

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NOMENCLATURE

D_j	Diameter of the jet
PM	Porous Media
MF	Metal Foam
IJF	Impinging jet flows
d	Diameter of the MF
t	Thickness of the MF
q_w	Heat flux, Watt
T_j	Temperature of the jet
T_w	Temperature of the heated plate
V_j	Velocity of the Jet
Nu	Nusselt number
Re	Reynolds number
k	Thermal conductivity, $Wm^{-1}K^{-1}$

Greek symbols

μ	Dynamic viscosity of air, kg. $m^{-1}.s^{-1}$
ρ	Density of the air at 298K, Kgm^{-3}

Subscripts

w	Wall
j	Jet
wa	Wall average