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Energy Procedia 101 (2016) 1095 – 1102

Energy Procedia

71st Conference of the Italian Thermal Machines Engineering Association, ATI2016, 14-16 September 2016, Turin, Italy

Experimental investigation on fluid dynamic and thermal behavior in confined impinging round jets in aluminum foam

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Abstract

In this paper an experimental investigation is carried out on impinging jets in porous media with the wall heated from below with a uniform heat flux. The fluid is air. The experimental apparatus is made up of a fun systems, a test section, a tube, to reduce the section in a circular section. The tube is long 1.0 m and diameter of 0.012 m. The test section has a diameter of 0.10 m and it has the thickness of 10, 20 and 40 mm. In the test section the lower plate is in aluminum and is heated by an electrical resistances whereas the upper plate is in Plexiglas. The experiments are carried out employing aluminum foams with 5, 10 and 40 PPI and three thickness over the heated circular plate. Results are obtained in a Reynolds number range from 500 to 1500 and wall heat flux from 500 W/m2 to 1400 W/m2. Results are given in terms of wall temperature profiles, local and average Nusselt numbers, pressure drops, friction factor and Richardson number. Moreover, to evaluate the improvement due to the presence of the metal foam, it is necessary a quantitative methodology. In this work an energy performance ratio is employed to compare the performances of surface with and without foams in terms of heat transfer coefficients and pressure drops. Preliminarily experimental results has confirmed that the use of the porous medium improves the heat transfer promoting the heat dissipation of the surface with high efficacy but determines an increase in pressure drops.

Peer-review under responsibility of the Scientific Committee of ATI 2016.

Keywords: impinging jet; porous media; round jets; metal foam.

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Peer-review under responsibility of the Scientific Committee of ATI 2016.
doi:10.1016/j.egypro.2016.11.149

1. Introduction

Impinging jets are applications very useful for engineering applications because they achieve high heat transfer coefficients both for cooling or heating applications. A particular improvement for impinging jet are the metal foams, because they have high thermal conductivity, low weight, good strength, noise damping, shock resistance [1]. Leong and Jin [2] accomplished an experimental investigation on an oscillating flow through a rectangular channel filled with open-cell metal foam. They analyzed the metal foam with different PPI and collected data on pressure drops and velocities. The results showed that the oscillating flow in an open-cell metal foam was governed by the kinetic Reynolds number Re based on a hydraulic diameter and the dimensionless flow displacement amplitude AD_h. Hollworth and Durbin [3] investigated the impinging jet for electronic cooling. A numerical study about the heat performance of metal foam has accomplished in Dukhan et al. [4]. This analysis showed that the temperature decreases exponentially through the foam in the flow direction. Marafie et al. [5] numerically studied the non-Darcian effects in a metal foam block with a confined slot jet in mixed convection regime. The results showed that the Nusselt number increases with decreasing of dimensionless height of the foam block up to 0.05, below which the Nusselt number decreases. Kuang et al. [6] studied the effects of foam height and foam distance from the jet under an axial fan flow. Shih et al. [7] accomplished an experiment on the heat transfer characteristics of aluminum foam heat sink under an impinging jet. They found that the increase of the Nusselt number along the aluminum foam heat sink is caused by the reduced convective resistance at the solid-gas interface and by the increased velocity near the heat generation surface. Imraan and Sharma [8] numerically studied the heat transfer of jet impingement in a frost free refrigerator for different Reynolds number. They found that the slot jet impinging produces heat transfer rate comprised between those for the corresponding uniform cross-flow and slot jet impingement on a non-confined cylinder. Di Bella et al. [9] studied the jet dynamics in a round jet impinging on a foamed porous media. They found that the penetration of the impinging flow into the porous media is significantly affected by permeability. Huang and El Genk [10] accomplished an experimental analysis on heat transfer of an impinging jet on a flat surface to determine the values of the local and average Nusselt numbers, in particular for low values of Reynolds number and jet spacing. This paper analyzed experimentally the air flow and the heat transfer of an impinging jet on an aluminum foam with 40 PPI, heated from below with a uniform heat flux. The aim is to obtain values of friction factor and for three different configurations. The metal foam analyzed have 10, 20 and 40 mm of thickness. The Reynolds number varies from 5100 to 15300 and the wall heat flux is imposed to 510 W/m^2, 765 W/m^2, 1020 W/m^2 and 1400 W/m^2. The results are presented in term of wall temperature profiles, air temperature, local and average Nusselt numbers, pressure drop, friction factor and Richardson numbers.

Nomenclature

AD_h	dimensionless flow displacement amplitude
D	Diameter (m)
EPR	efficiency
f	Friction Factor
Gr	Grashof number
H	Height slot (mm)
k	Thermal conductivity ($Wm^{-1}K^{-1}$)
Nu	Nusselt number
P	Pressure
q	heat flux (Wm^{-2})
PPI	pores per inch
Re	Kinetic Reynolds number
Ri	Richardson number
T	Temperature (°C)
V	air velocity (ms-1)
Greek symbols	
P	density (kgm-3)

ε	porosity
μ	kinematic viscosity (kgm-1s-1)
Δ	gradient
ν	dynamic viscosity
Subscripts	
s	clean

2. Experimental setup

The experimental apparatus includes the test section, instruments and fan system. The room air is putted into the device by means of a fan system positioned upstream. The flow is controlled by a valve and it is regulated by a rotameter Krohne type VA40S the flow rate was regulated (Fig.1a).

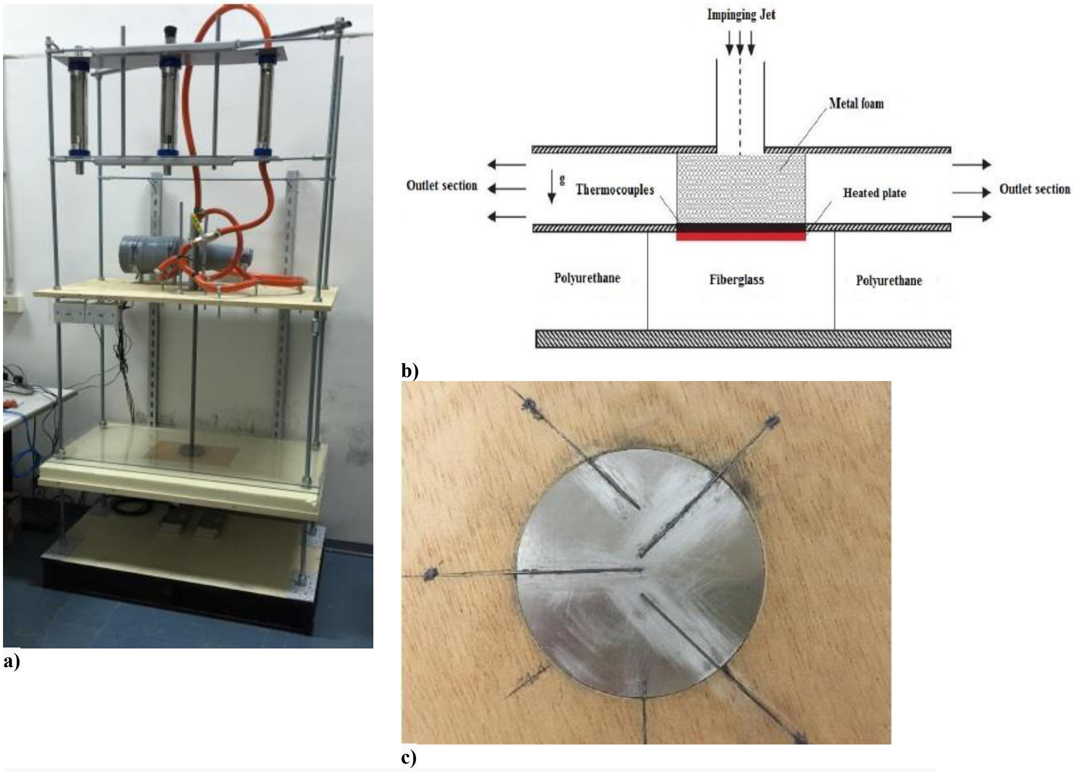


Fig.1. Experimental apparatus: a) fan system; b) sketch of impinging jet of metal foam; c) thermocouples position

The flow enters in a duct with a diameter of 12 mm and 1 m long. After, the jet impinges the upper surface of metal foam. Under the metal foam an aluminum plate (3 mm of thickness) is heated from below by means of an electrical resistance regulated by a power supply AGILENT E3633A. to reduce heat losses, a polystyrene block and fiberglass is affixed to bottom face of heated plate (Fig. 1b). The test section is confined between two Plexiglas plates of 4 mm of thickness. Three different thickness of metal foam are studied: 10 mm, 20 mm and 40 mm with a PPI equals to 40.

An Isotech instrument mod. 938 ice point with an accuracy of ±0.04 °C was used as thermocouple reference junctions. Five thermocouples were deployed along the diameter of the aluminum plate and other two in the outlet section of impinging jet in order to measure the air jet temperature (Fig.1c). An AGILENT 34980A multifunction measurement unit and a computer were used for the data acquisition. To acquire the data of pressure a digital manometer tube has been used. The tests has been performed varying the Reynolds number from 5100 to 15300. The average and local Nusselt number were determined as functions of parametric values of the Reynolds and Richardson numbers:

$$Nu_i = \frac{q_c D}{(T_{wi} - T_o) k_{air}} \tag{4}$$

the subscript i indicates the radial location on the wall (index w) at which the temperature is being measured. T_o is a reference temperature. It is the inlet air temperature of the test section. D is the diameter of the impinging jet, q_c is the wall heat flux and k_{air} is the thermal conductivity of the air. The average Nusselt number was calculated by:

$$Nu = \frac{q_c}{T_w - T_0} \frac{D}{k_{air}} \tag{5}$$

The Nusselt number results will be plotted as a function of Reynolds, Richardson and Grashof numbers, defined:

$$\mathrm{Re} = \rho V D \mu^{-1} \tag{6}$$
$$Gr = g\beta(T - T_0) H^3 \mu^{-2} \tag{7}$$
$$Ri = Gr\,\mathrm{Re}^{-2} \tag{8}$$

Where H is the height of the metal foam (10,20 and 40 mm) Δp is the pressure drop calculated as the difference between upstream and downstream pressure of the test section. The friction factor is:

$$f = \Delta P(\rho V^2 0.5)^{-1} \tag{9}$$

To evaluate the heat transfer improvement due to the presence of the metal foam the EPR has been evaluated comparing the performances of surface with and without foam:

$$EPR = \overline{Nu} Nu_s^{-1} \left(f f_s^{-1} \right)^{-0.33} \tag{10}$$

With Nu_s and f_s are respectively the average Nusselt number and friction factor of the smooth surface (without the metal foam).
Analyses of the uncertainties on the measured parameters are ± 0.3°C for temperature and ± 1% for pressure. The uncertainties of Nusselt number, Reynolds number, friction factor and EPR have been evaluated with the Kline and McClintock [19] methodology. It has been estimated that the uncertainty of the Nusselt number is 2.48%, the Reynolds number is 4.26%, EPR is 4.85% and friction factor is 3.13%.

3. Results

The results have been presented as a function of Reynolds number. Three different configurations are presented and a comparison of temperature profile between case clean and case with heat flux at 1020 W/m^2 is accomplished.

Fig. 2 shows that the ratio D/H is determinant. In fact, at low D/H values, the metal foam determines a minimal wall temperature at Reynolds number equal to 10200 and 15300. Besides, the metal foam has been generated a ΔT equal to 5-7 °C respect to the clean case.

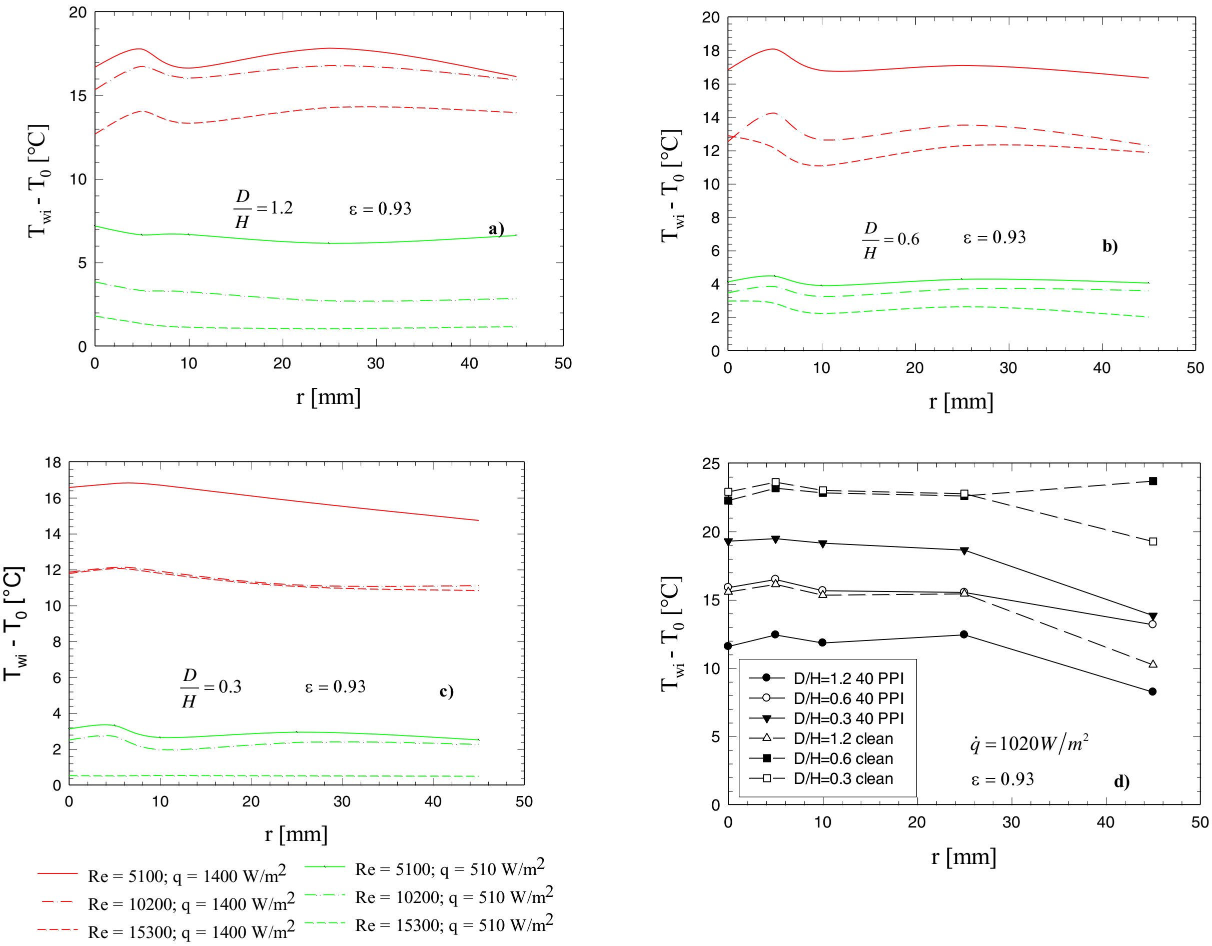
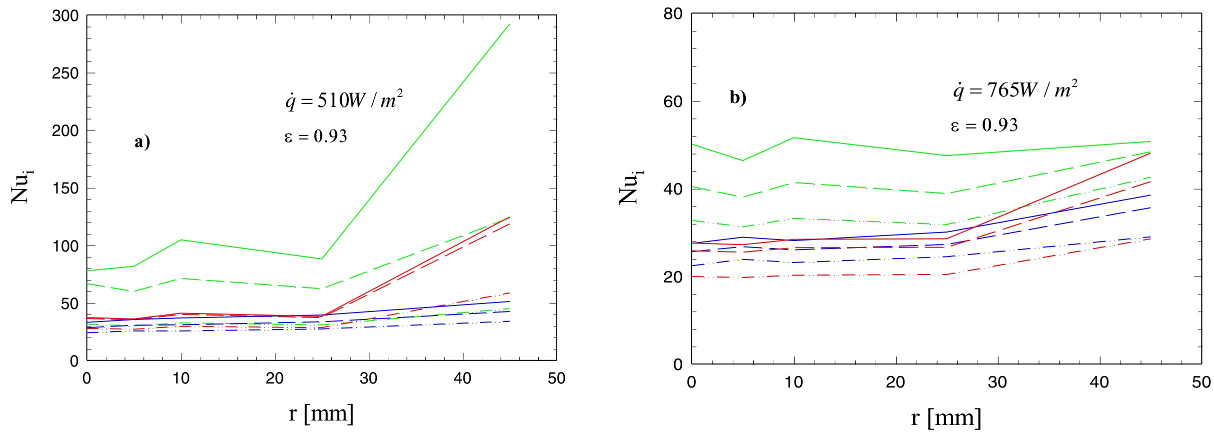


Fig. 2. Temperature profile: a) D/H = 1.2; b) D/H = 0.6; c) D/H = 0.3; d) comparison between case clean and with metal foam.

Fig. 3 (a-d) compares local Nusselt number as a function of r for different configurations. Fig. 3 shows that the convective heat transfer increases with the mass flow rate, in fact, the local Nusselt number is higher for high values of Reynolds number. Therefore, for q = 510 W/m^2 and for r = 45 mm the convective phenomena are predominant.



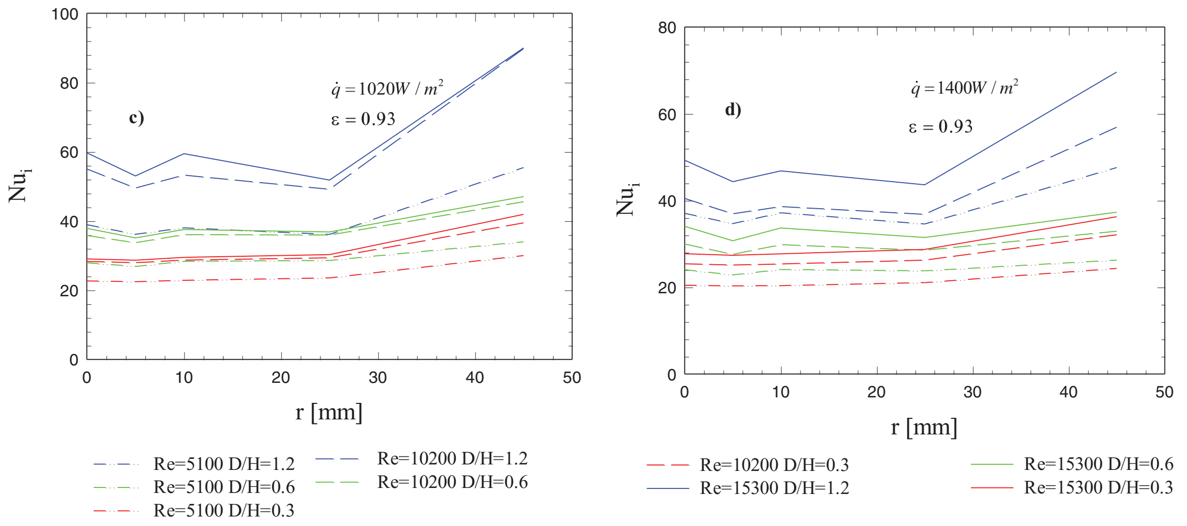


Fig. 3. Local Nusselt number: a) $\dot{q}$ = 510 W/m^2; b) $\dot{q}$ = 765 W/m^2; c) $\dot{q}$ = 1020 W/m^2; $\dot{q}$ = 1400 W/m^2

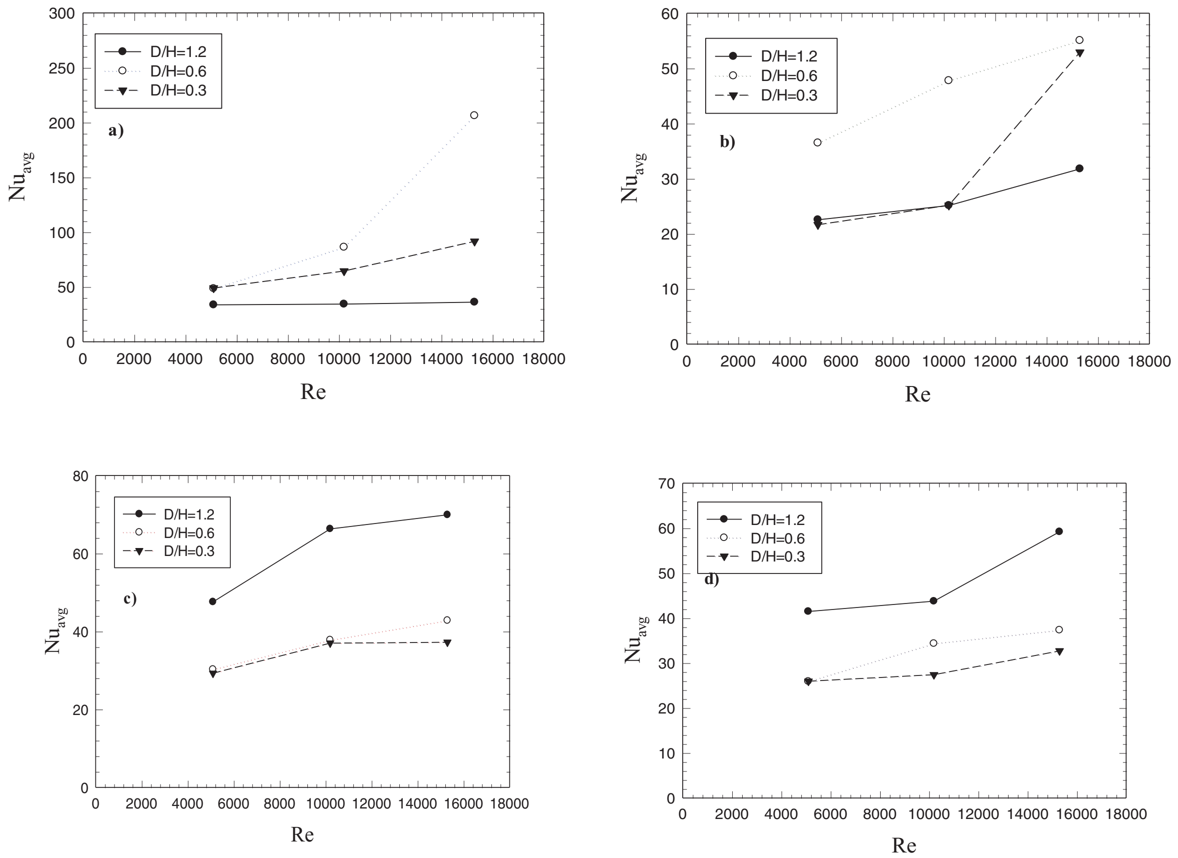


Fig. 4. Trend of the ratio Nu_{avg} as a function of Reynolds number for three different configurations: a) $\dot{q}$ = 510 W/m^2; b) $\dot{q}$ = 765 W/m^2; c) $\dot{q}$ = 1020 W/m^2; $\dot{q}$ = 1400 W/m^2.

Fig. 4 (a,d) shows clearly that increasing the Reynolds number the average Nusselt number increases. Figure 5 (a-b) shows pressure drop and the friction factor as a function of Reynolds number.

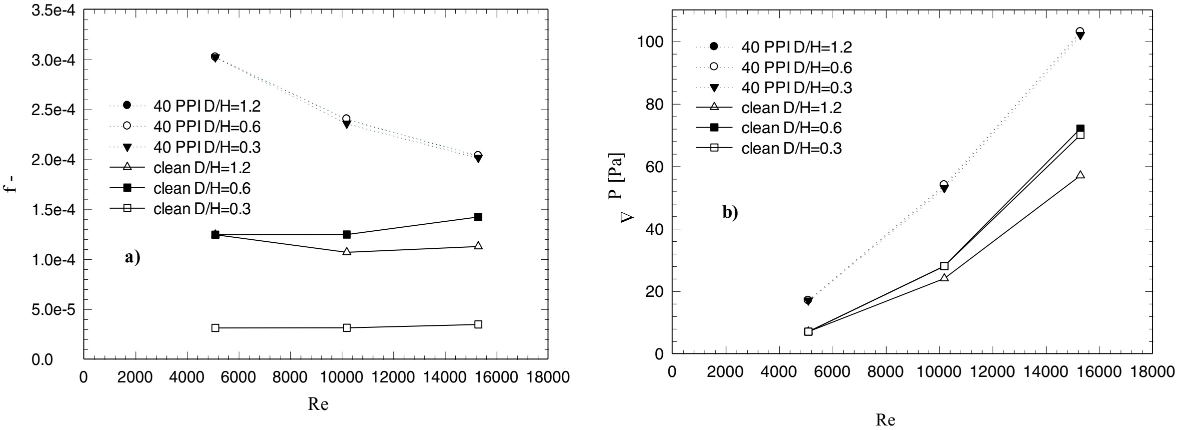


Fig. 5. a) Friction factor and b) pressure drop as a function of Reynolds number.

It is evident that an increase of pressure drop increases the velocity of the flux. It is evident that the trend of friction factor decrease with Reynolds number. The higher values of friction factor occur for Reynolds number equal to 5100 with metal foam.

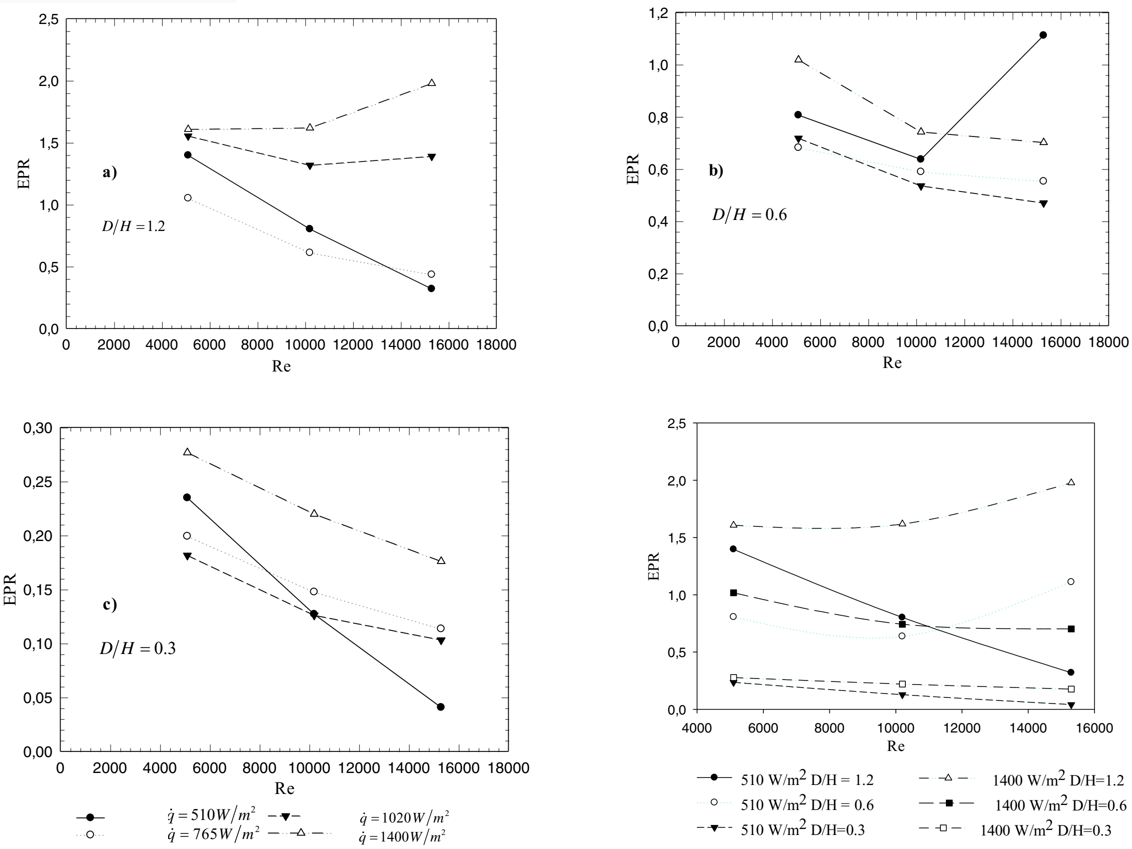


Fig. 6. EPR as a function of Reynolds number: a) D/H = 1.2; b) D/H = 0.6; c) D/H = 0.3; d) comparison between two different heat flux.

Figure 6 shows the trend of EPR as a function of Reynolds number and the ratio D/H. It has been demonstrated that the higher efficiency is for higher values of heat flux for all configurations. Besides, the ratio D/H = 1.2 has higher values of EPR at lower values of Reynolds number, instead, for higher values of Reynolds number, the EPR decreases for all configurations except for D/H equal to 1.2 and wall heat flux equal to 1400 W m^{-2}.

5. Conclusions

Main conclusions of these experimental investigations are:

- The use of metal foam improved the heat transfer impinging jet on the heated wall. The increase of Reynolds number determined an increased of Nusselt number and it was detected that for high D/H, equal to 1.2, and low wall heat flux values, the average Nusselt numbers are the highest respect to the other D/H values.
- It was demonstrated that the metal foam increases the values of friction factor and decreases the ΔP values for all configurations.
- It was demonstrated that metal foam with 20 mm of thickness has higher efficiency of metal foam with 10 and 40 mm of thickness at higher Reynolds number. At lower values of Reynolds number the metal foam 10 mm thick have efficiency higher of those with larger thickness.

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