Validation of Heat Transfer Characteristics between Impinging Circular Air Jet and Flat Plate by Artificial Neural Network

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***Abstract*:-An experimental investigation is performed to study the effect of the finned (rough) surface on the local heat transfer coefficients between the impinging circular air jet and flat plate. Reynolds number is varied between 7000 and 30000 based on the nozzle exit condition and jet to plate spacing between 0.5 to 6 times of the nozzle diameter. The fins used are in the form of cubes of 2 *mm* size spaced at a pitch of 5 *mm* on the target plate. The increase in the heat transfer coefficient up to 56.54% depending on the shape of the fin, nozzle plate spacing and the Reynolds number is observed in case of cubical finned surfaces as compared to the smooth surface. Experimental results are validated using Artificial Neural Network (ANN) in the present study. The perfect validation of experimental results has been observed.**

***Keywords*: - Impinging air jet, Heat transfer augmentation, finned surface.**

Nomenclature

D	Diameter of the nozzle exit (m)
h	Heat transfer coefficient ($W/m^2.K$)
Nu	Nusselt number (Dimensionless)
r	Radial distance from the stagnation point (m)
Re	Reynolds number based on nozzle exit condition (Dimensionless)
Z	Nozzle plate spacing (m)

I. Introduction

Impinging jets have received considerable attention due to their inherent characteristics of high rates of heat transfer besides having simple geometry. Various industrial processes involving high heat transfer rates apply impinging jets. Heat transfer rates in case of impinging jets are affected by various parameters like Reynolds number, nozzle plate spacing, radial distance from stagnation point, Prandtl number, target plate inclination, confinement of the jet, nozzle geometry, curvature of target plate, roughness of the target plate, low scale turbulence intensity i.e., turbulence intensity at the nozzle exit. Gardon and Cobonpue [1] have reported the heat transfer distribution between circular jet and flat plate for the nozzle plate spacings greater than 2 times the diameter of jet, both for single jet and array of jets. They have used specially designed heat flux gauge for the measurement of local heat transfer rates from a constant wall temperature plate. Gardon and Akfirat [2] studied the effect of turbulence on the heat transfer between two dimensional jet and flat plate. They also studied effect of multiple two-dimensional jets on the heat transfer distribution [3]. Baughn and Shimizu [4] and Hrycak [5] have conducted experiments of heat transfer between round jet and flat plate employing different methods of surface temperature measurement. Lyttle and Webb [6] have studied the effect of very low nozzle plate spacing (Z/D < 1) on the local heat transfer distribution on a flat plate impinged by a circular air jet and found that in the acceleration range of the nozzle plate spacing (Z/D < 0.25), maximum Nusselt number shifts from the stagnation point to the point of secondary peak with the effect being more pronounced at higher Reynolds number. Review of the experimental work on impinging jets is done by Martine [7], Jambunathan *et. al.* [8] and Viskanta [9]. Hansen and Webb [10] have studied the effect of the modified surface on the heat transfer between impinging circular nozzle and the flat plate. They have found that for the pyramidal, short square and intermediate square fins, there is an increase in the average Nusselt number value by 12 to 23 % and reduction in the value of the average Nusselt number by 4 to 38 % for the other types of fins studied. Chakroun *et al.* [11] have studied the effect of surface roughners, in the form of cubes, on the heat transfer between impinging jets and flat plate. They have reported the heat transfer augmentation up to 8 to 28 %. However their data reflects the average Nusselt number variation rather than local data because of the large thickness of the target plate used.

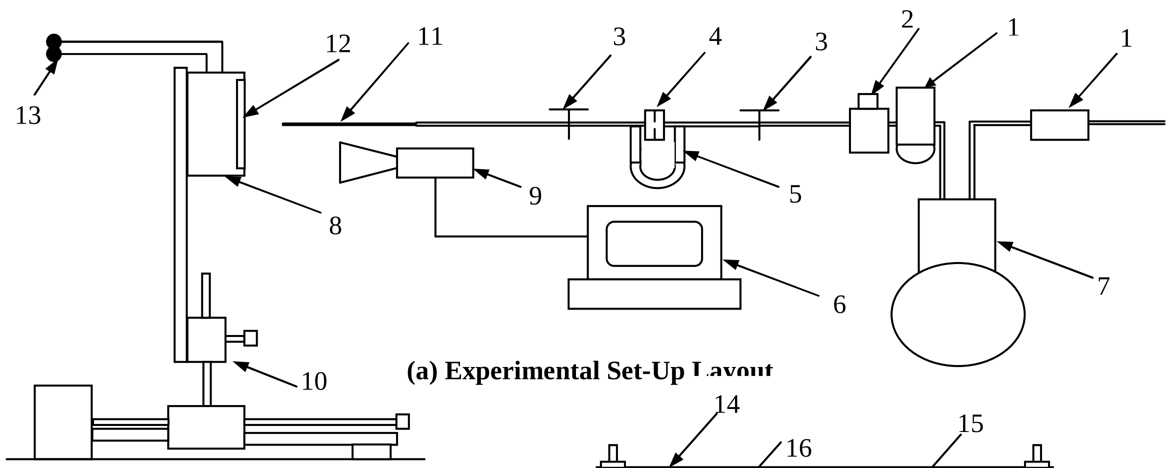
Literature review suggests that there is lack of information on local heat transfer data in case of the finned surfaces, therefore, the aim of the present work is to study the effect of the finned surface on the local heat transfer coefficients between the impinging circular jet and flat plate. The experimental parameters are the Reynolds number varied between 7000 and 30000 based on the nozzle exit condition and jet to plate spacing of 0.5, 1, 2, 4, 6 times the nozzle diameter. Different configurations of finned surfaces studied in the present work are cubical fins. Similarly, experimental results are validated using Artificial Neural Network in the present study.

II. Experimental Test Set Up

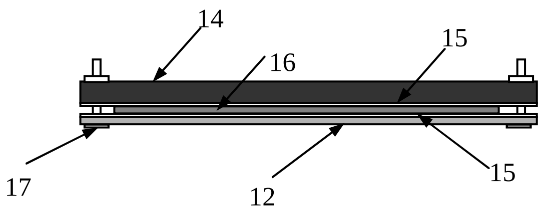
The experimental set up layout is depicted in Fig.1 (a). Air jet is supplied by a three-cylinder two-stage air compressor through a calibrated orifice flow meter. Air filter and Pressure regulator are installed upstream of the orifice flow meter to filter the air and to maintain the downstream pressure at a value of 4 ± 0.05 bar. The flow rate is controlled by two needle valves, one on each side of the orifice flow meter. The nozzle, which directs the air until it impinges upon a heated target plate, is constructed with a 7.35 mm inner diameter aluminum pipe of length to diameter ratio of 83, which is almost same as that of Lyttle and Webb [6]. This length is sufficient to ensure fully developed flow over the Reynolds number range investigated. The end of the nozzle pipe is machined perpendicular to the nozzle axis. The impinging plate is constructed using 1*mm* thick stainless steel plate of size $120\,mm \times 120\,mm$. Nichrome heater of size $100\,mm \times 100\,mm$ is packed between impinging plate and a Bakelite support plate with mica sheets in between to isolate the impinging plate from the heater, electrically. To ensure that the impinging plate makes perfect contact with the heater, the impinging plate and Bakelite support are tightened together with nuts and screws as shown in Fig.1 (b). This assembly is then insulated from all sides except the impinging surface by ceramic wool and thermocol to reduce the back and side conduction losses. A two-dimensional traverse system is used to locate the target plate at a given position with respect to nozzle exit.

Electric power is supplied to the heater through variac. The voltage and the current are measured by digital panel meters. The range and the accuracy of these meters are 0 to 200± 0.5% Volts and 0 to 5± 0.5% amps respectively. A K-type Chromel-Alumel thermocouple junction is soldered on the target plate at its extreme end to know about the steady state. The output of the thermocouple is measured by 'Agronic' milivoltmeter. Similarly, thermocouples are attached on cooled surface along the flow direction from the stagnation point to measure wall temperature distribution.

Different configurations of surfaces in the form of fins, studied in the present work are cubical fins. Cubical fins are of 2 *mm* size spaced at a pitch of 5 *mm* on the target plate as shown in Fig. 2. The increase in the heat transfer area for these surfaces is 56.54% as compared to smooth surface.



(a) Experimental Set-Up Layout



(b) Test Section Assembly

1 – Air filter, 2 – Pressure regulator, 3 – Needle valve, 4 – Orifice flow meter, 5 – Water manometer, 6 – Digital temperature indicator and selector switch, 7 – Reciprocating compressor, 8 – Target plate assembly, 9 – Volt meter and Ammeter ,10 – 2D Traverse system, 11 – Nozzle pipe, 12 – Target plate, 13 – Leeds for heater, 14 – Bakelite support, 15 – Mica sheet, 16 – Heater, 17 – Screw and nut assembly.

Fig. 1 Experimental Test Set-Up

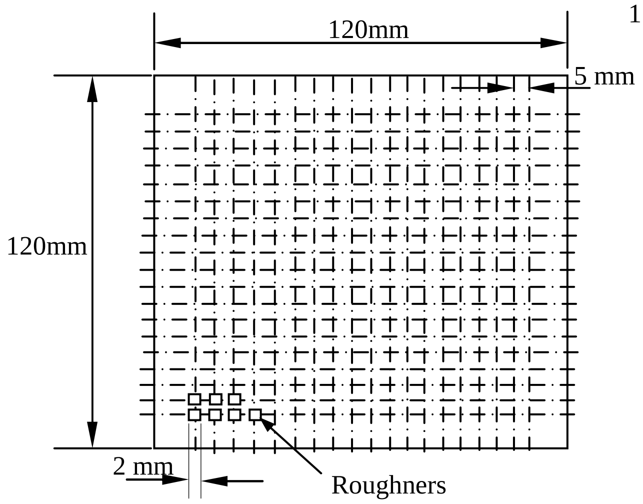


Fig. 2 Details of the Finned (Rough) Surfaces

III. RESULTS AND DISCUSSION

1. Comparison of the Nusselt Number for Smooth and Cubical Finned Surfaces at Reynolds Number of 23000 for different Z/D's:

The experimental evaluation has been carried out for the rough flat surfaces. The rough flat surface under consideration is the cubical finned surface. Figures 3 (a) to 3(e) give the comparison of the Nusselt number for smooth and cubical finned surfaces at different Z/D's and Reynolds number of 23000. The experimental evaluation for cubical finned surface depicts an increase in heat transfer as compared to that of smooth plate for all the values of Z/D's. The increase in the heat transfer on the cubical finned surface is because of the increase in the swirl i.e. superposition of tangential velocity component onto the axial flow, which affects the turbulence characteristics of the flow. Comparing the percentage increase in the Nusselt number at the stagnation point for different nozzle plate spacing, it is observed that increase in Nusselt number for cubical finned surfaces is higher for Z/D of 1 and 6 as compared to all other nozzle plate spacing. For Z/D of 6, as shown in figure 3(e), the maximum Nusselt number is obtained in the vicinity of stagnation point and it decreases monotonically in the radial direction. For Z/D of 6, the maximum Nusselt number is 181.34 for cubical finned surface whereas it is 157.01 for the smooth surface at the stagnation point .The maximum Nusselt number for Z/D of 6 is because of maximum centerline turbulent intensity at this Z/D. Impinging jet for Reynolds number of 23000 is turbulent at the nozzle exit itself and its turbulent intensities are further increased as a result of continuous entrainment of the surrounding air into the wall jet. This entrainment-originated turbulence goes on increasing as the Z/D increases which in turn enhances the heat transfer rate. It is also believed that the circulation around the roughness elements and increase in turbulence intensity are the causes of heat transfer augmentation.

2. Comparison of the Nusselt Number For Smooth and Cubical Finned Surfaces at Z/D Of 6 and for Different Reynolds Number:

Figure 4 (a) to figure 4(f) shows the comparison of the Nusselt number for the smooth and the cubical finned surface at a Z/D of 6 for different Reynolds number of 7000,12500,16000,21500,25000 and 30000. It is observed that the increase in the heat transfer for cubical finned surface as compared to the smooth surface is higher in the stagnation region as compared to wall jet region for the entire range of Reynolds number .Increase in the Reynolds number causes increase in heat transfer in the entire range of Reynolds number for cubical finned surface. It is observed from the figure 4 (f) that the maximum Nusselt number is found to occur at the stagnation point with a value equal to 201.83 for cubical finned surface and 181.29 for the smooth surface at Reynolds number of 30000 as compared to minimum Nusselt number of 56.63 for cubical finned surface and 44.34 for the smooth surface at Reynolds number of 7000 as shown in the figure 4 (a).Comparing the entire range of Reynolds number for cubical finned surface, it is observed that the percentage increase in Nusselt number increases from the Reynolds number of 7000 to 16000 and thereafter percentage increase in Nusselt number is lower for further increase in Reynolds number. Figures show that the maximum Nusselt number occurs in the stagnation region and decreases with increasing radial distance. This behavior is noticed for all other values of Reynolds numbers. For low Reynolds number, roughness shows the small effect on heat transfer. The flow over the plate for a low Reynolds number can be considered as aerodynamically smooth where the roughness effect is so small that the flow behaves as if the wall was smooth. But, at higher Reynolds numbers, the flow changes from hydraulically smooth to the fully rough where the roughness so dominates the momentum transport to the wall that the viscous effects are negligible. Roughness increases the swirl i.e. superposition of the tangential velocity component onto axial flow, which can markedly affect the flow and turbulence characteristics of the flow. As the degree of swirl increases, the jet spread, the rate of entrainment of the surrounding fluid, and the rate of the jet velocity decay are all increased, resulting in higher heat transfer.

3. Validation using ANN :

The experimental results are validated using Artificial Neural Network (ANN) in the present study. The observed values are compared with predicted values obtained by Artificial Neural Network for cubical finned surface at various parameters. Details of the validation is as follows.

(i) Figure 5(a) shows the RMSE values for cubical finned surface at Re = 23000 and Z/D = 6.It is observed from the figure that validation using ANN gives the correlation coefficient (R^2) as 0.999.

(ii) Figure 5(b) shows the RMSE values for cubical finned surface at Z/D = 6 and Re = 16000. As the percentage increase in Nusselt number is higher at Re = 16000, this optimum condition is considered for validation using ANN for this case. It is observed from the figure that validation using ANN gives the correlation coefficient (R^2) as 1.00.

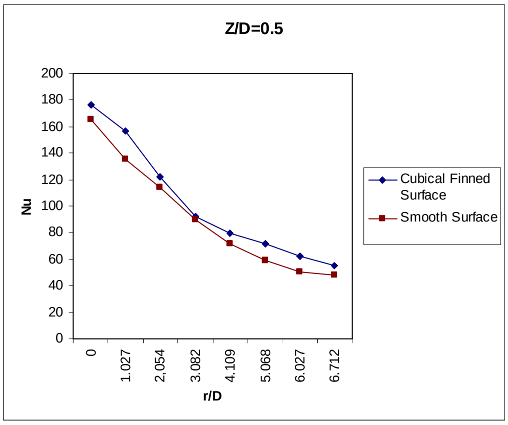


Figure 3 (a)

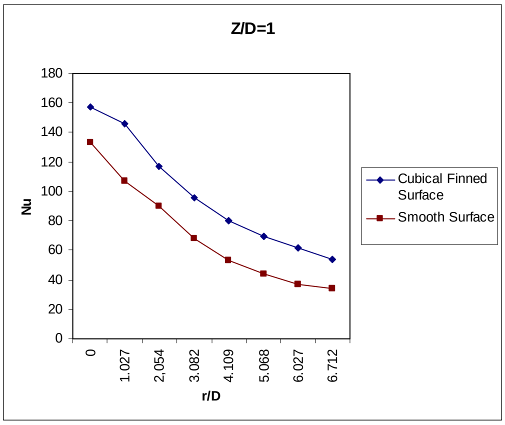


Figure 3(b)

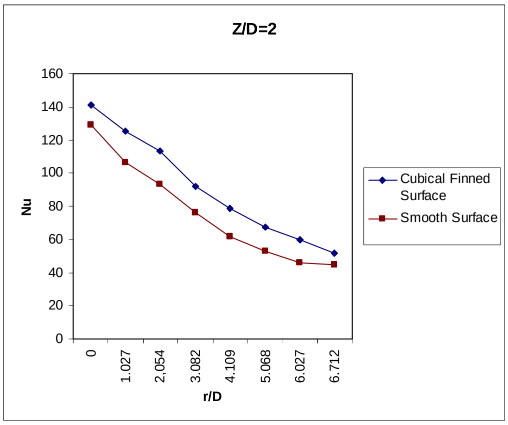


Figure 3 (c)

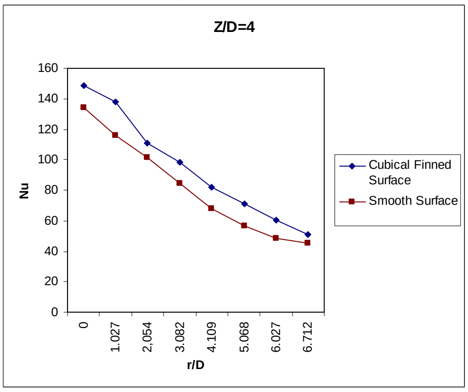


Figure 3 (d)

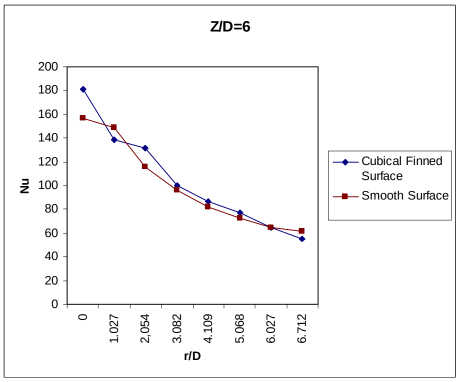


Figure 3 (e)

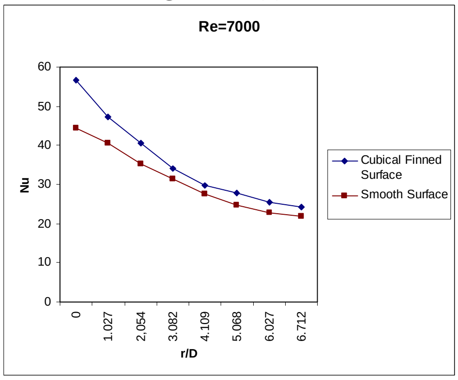


Figure 4 (a)

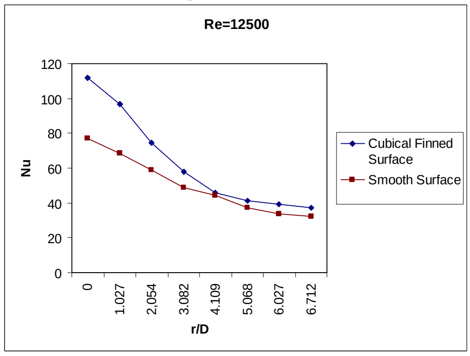


Figure 4 (b)

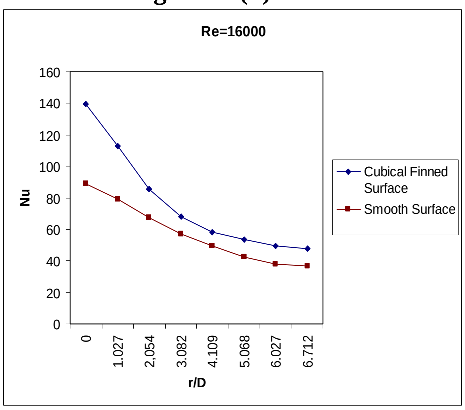


Figure 4 (c)

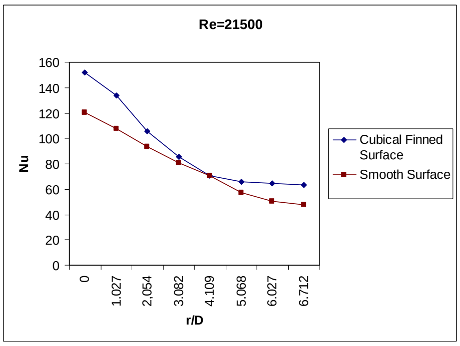


Figure 4 (d)

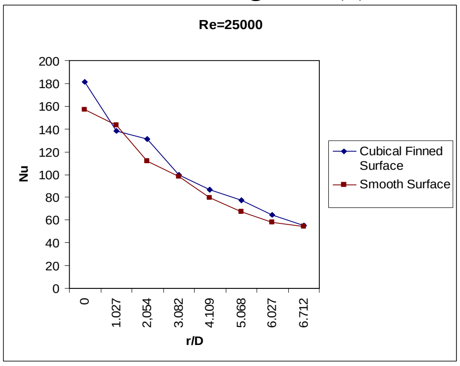


Figure 4 (e)

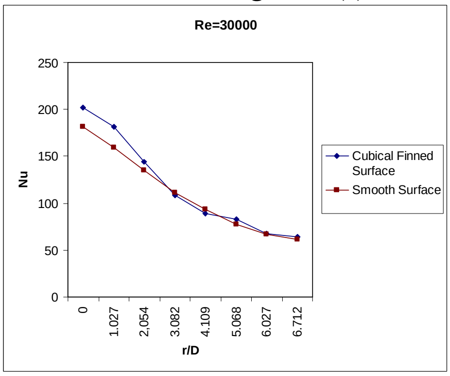


Figure 4. (f)

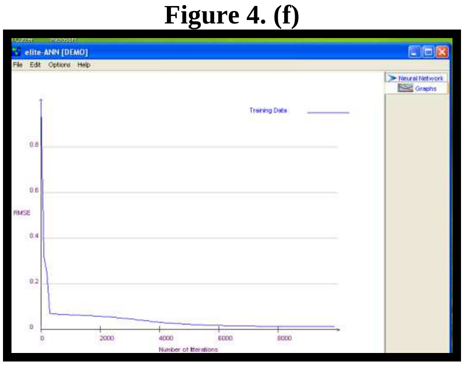


Figure 5(a) : RMSE value for Cubical Finned Surface at Re = 23000, Variable Z/D

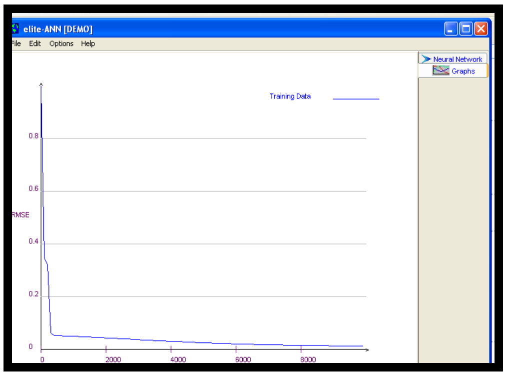


Figure 5 (b) : RMSE value for Cubical Finned Surface at Z/D = 6, Variable Re (Optimum at Re=16000)

CONCLUSIONS

The conclusions from the present research work of heat transfer characteristics of the impinging jets on various surfaces at various parameters are explained as follows

1. From the comparison for the Nusselt number for smooth and cubical finned surface at different Z/D's and Reynolds number of 23000, it is observed that increase in heat transfer is higher for Z/D of 1 and 6 as compared to all other nozzle plate spacings at the stagnation point. For the entire range of Z/D's, the maximum Nusselt number is obtained in the vicinity of stagnation point and it decreases monotonically in the radial direction. The maximum Nusselt number observed is 181.34 for cubical finned surface whereas it is 157.01 for the smooth surface at the stagnation point. The percentage increase in Nusselt number for cubical finned surface as compared to the smooth surface is 6.67 % to 18.01 % for the various values of Z/D.
2. Similarly, from the experimentation at Z/D of 6 and different Reynolds numbers of 7000 to 30000, it is observed that the maximum Nusselt number is found to occur at stagnation point with a value equal to 201.83 for cubical finned surface and 181.29 for the smooth surface at Reynolds number of 30000 as compared to minimum Nusselt number of 56.63 for cubical finned surface and 44.34 for the smooth surface at Reynolds number of 7000. The percentage increase in Nusselt number for cubical finned surface as compared to smooth surface is from 11.32 % to 56.54 % at the stagnation point for the entire range of Reynolds numbers.
3. After training the Artificial Neural Network, it is found that every case of experimental results is in good agreement with the predicted values obtained by ANN. The coefficient of correlation (R^2) for every case tested is found to be in the range of

0.999 to 1, which suggest the validation of experimentation carried out in the present work.

REFERENCES

1. R. Gardon and J. Cobonpue, Heat Transfer Between a Flat Plate and Jets of Air Impinging on It, *Int. Developments in Heat Transfer, ASME* (1962) 454-460.
2. R Gardon and C.Akfirat, The Role of Turbulence in Determining the Heat Transfer Characteristics of Impinging Jets, *Int. J. Heat and Mass Transfer*, 8 (1965) 1261- 1272.
3. R Gardon and C. Akfirat, Heat Transfer Characteristics of Impinging Two Dimensional Air Jets", *J. of Heat Transfer*, 88 (1966) 101-108.
4. J. W. Baughn and S. Shimizu, " Heat Transfer Measurements From a Surface With Uniform Heat Flux and an Impinging Jet", *J. of Heat Transfer*, 111 (1989) 1096-1098.
5. P. Hrycak, Heat Transfer from Round Impinging Jets to a Flat Plate, *Int. J. Heat Mass Transfer*, 26 (1983) 1857-1865.
6. D. Lytle, B. W. Webb, Air Jet Impingement Heat Transfer at Low Nozzle Plate Spacings, *Int. J. Heat and Mass Transfer*, 37 (1994) 1687-1697.
7. H. Martine, Heat and Mass Transfer between Impinging Gas Jets and Solid Surfaces, *Adv. Heat Transfer*, 13 (1977) 1-60.
8. K. Jambunathan, E. Lai, M. A. Moss, B.L. Button, A Review of Heat Transfer Data for Single Circular Jet Impingement, *Int. J. Heat and Fluid Flow*, 13 (1992) 106- 115.
9. R. Viskanta, Heat Transfer to Impinging Isothermal Gas and Flame Jets, *Experimental Thermal and Fluid Science*, 6 (1993) 111-134.
10. L. G. Hansen, B. W. Webb, Air Jet Impingement Heat Transfer from Modified Surfaces, *Int. J. Heat Mass Transfer*, 36 (1993) 989-997.
11. W. M. Chakroun, A. A. Abdel-Rahman, S. F. Al-Fahed, heat transfer augmentation for air jet impinged on a rough surface, *Applied Thermal Engineering*, 18 (1998) 1225-1241.
12. N. Gao, H. Sun. and D Ewing, Heat Transfer to Impinging Round Jets with Triangular Tabs, *Int. J. Heat Mass Transfer*, 46 (2003) 2557-2569.
13. D. H. Lee, J. Song, Myeong Chan Jo, The Effect of Nozzle Diameter on Impinging Jet Heat Transfer and Fluid Flow, *J. of Heat Transfer*, 126 (2004) 554-557.