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Experimental Investigation of Impingement Cooling Thermal Performance for Leading Edge blade of Gas Turbine

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ABSTRACT

This paper is an experimental investigation to study the impingement cooling method in leading edge of gas turbine blade by using an infrared camera technique , the two configurations were investigated to studying the effects of blowing ratio in range (0.5 , 0.8 ,1 , 1.5 ,2) and variation of space between the jet and target at h/d (1 , 2 , 3) on the film cooling effectiveness, heat transfer in inner and outer surface of the target and discharge coefficient , the results obtained indicated that increasing blowing ratio leads to increasing the film cooling effectiveness , heat transfer coefficient for both side of target and discharge coefficient, whine increasing h/d leads to a decreasing in the film cooling effectiveness, while the discharge coefficient increasing.

KEYWORDS: film cooling effectiveness, blowing ratio, impingement system cooling.

NUMCLATURE

A_h : hole area ΔP : pressure difference
BR : blowing ratio ρ : air density
C_d: discharge coefficient ρ_m : air densityof mean stream
d : hole diameter ρ_j : air densityof jet
$\bar{h}$: averageheat transfer coefficient $\bar{\eta}$: film effectiveness coefficient
I R : infrared camera N_h : the number of jet holes
$T_{orifice}$: air temperature at inlet orifice U_m : mean stream air velocity
T_m: temperature of mean stream air U_j : mean stream air velocity
T_j : temperature of air jet T_w : surface wall temperature
T_{out} : temperature at outlet section T_∞: temperature of mean stream air
h / d : ratio of space between target and jet to hole diameter

INTRODUCTION

Gas turbine plays a vital role in the new today industrialized society, and as the increasing demands for power, therefor power output and thermal efficiency of gas turbines must also increase. Land-based industrial gas turbine is normally operated continuously over long operational time , The thermal efficiency and specific power of gas turbine can be improved by increasing the turbine inlet temperature, operation a high-temperature caused a thermal failure to prevent that risk film cooling is commonly used [1], The thermal efficiency and

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specific power of gas turbine can be improved by increasing the turbine inlet temperature, operation with high-temperature caused a thermal failure of some turbine parts [2], for preventing that risk cooling methods are commonly used. Impingement cooling is one method involved injection of coolant on the internal surface of blade gas turbine (as example) to absorb heat gained from hot gases.

The objective of impingement cooling is to achieve internally high heat transfer between the turbine blades and coolant flow , In this high-capacity cooling form the convection cooling, the cooling air clashes on the inner surface of the walls by high-velocity , this permitting an increased amount of heat to be transferred to the cooling air from the material surface. This cooling method can be restricted to desired sections of the components to maintain even temperatures over the entire surface. For example, the leading edge of a blades, vans are need to be cooled more than the mid chord section and trailing edge, so the gas is impinged.[3]

As stated in the literature for impingement system, the optimum jet angle which gives high heat transfer is achieved at 90º, where the maximum momentum of flow is strike on the target plate [4], Also, it shows that the effects of hole length to diameter (L/D) was not significant for development of impingement jet when L/D is greater than unity [5]; therefore in present work, the study was constrained to L/D=1, Serhankucuka [6], heat transfer coefficients for the flat jets are greater than heat transfer coefficients for the radial jets, Jason's O' grady[7] In this study it is numerical investigation that comparison between one row of jet impingement jet that result the enhancing in heat transfer coefficient in multiple jet rows, the researcher conducted that because of cross flow affecting in the multiple jets case.Luai M. Al-Hadhrami et al [8] Jet impingement heat transfer is dependent on several flow and geometrical parameters. The jet impingement Nusselt number is presented in a functional form as follows:

$$Nu = f\left[Re, \left({}^{x}/_{d}\ , {}^{Z}/_{d}\right), out\ flow\ orintation\right] \quad (1)$$

GORDON E. ANDREWS et al. [9] The results showed that the increase in surface average impingement heat transfer, relative to that of a smooth wall was small. The main effect of the slotted ribs was to change the influence of cross flow. T. Ravi Teja& S. Krishna Chaitanya [10], this paper focuses on the Mach Number Effects on Impingement heat transfer have varied the jet Reynolds number with the Mach number, they concluded that at low Mach numbers (<0.2) the influence on heat transfer, by the Mach number, could be neglected.EmadElnajjar et al [11] two heat transfer configurations (central-jet and side-jet) of jet impinging on a semicircular surface are experimentally studied. The results indicate that central-jet impingement shows a 50% improvement in heat transfer with minimal difference in pumping pressure compared to the side-jet for the same range of Reynolds numbers. M. E. Taslim&D. Bethka [12] This investigation dealt with impingement on the leading edge of an airfoil in the presence of cross-flows the presence of the external cross-flow reduces the impinging jet effectiveness both on the nose and sidewalls, Andrew C. Chambers et al [13] This paper presents a computational and experimental investigation into the use of shaped elliptical or elongated circular impingement holes designed to improve the penetration of the impinging jet across the coolant passage the results In all cases, a 16% increase in the Nusselt number on the impingement target surface Andrew Schroder et al [14] An experimental test facility has been designed, constructed, and commissioned for studying the convective heat transfer of an array of 55 impingement jets with a constant heat flux boundary condition, The results show that the time averaged Nusselt number increases with increasing jet Reynolds number in both the local and span wise averaged values A. J. UJAM et al [15] Studies were carried out to determine the Convective Heat Transfer Coefficient, *h*, under multiple jets of impinging cold air on a target heated flat plate. Results obtained were reduced and analyzed by evaluating for the dependence of *h* on the impingement jet diameter, Jet Reynolds Number, and the coolant mass flow rate *G*. However, there had been more works in impingement cooling on flat plate otherwise there were few researches on the curvature surface , The aim of the present study is to investigate the effect of nozzle arrangement for array of impinging jets in case with different jet-to-target distances. The in-line and staggered arrangement were experimentally conducted. The temperature distribution on the impingement surface was investigated using a infrared camera and the heat transfer coefficient distribution was evaluated the current study experimental, addition estimated discharge coefficient.

2. Experimental Setup:

For consideration the experiment of this study, a low speed open duct jet test rig was designed and built at University of Technology, Mechanical Engineering Department. The performance of the test rig conforms to the recommendations given by Refs [12] according to [13]. The basic idea that will be achieved in designing the test rig should have low turbulence, low pressure gradient, and high system efficiency .The apparatus consists of two open loops mainstream and secondary flow loops, the two loops were combined together to get the final shape ofthe test rig. The present test rig was used to investigate steady state of impingement cooling systems, Figure (1) illustrates the image facility, and figure (2) shows scheme of the rig .

2 – 1 Mainstream flow loop:

The mainstream flow loop has the ability to run with operating conditions as the air temperature of 30 ˚C and maximum speed of 10 m/s at the test, the main loop includes the following components: air blower, settling chamber, diffuser, contraction, figure (3) and bent duct and static pitot tube figure (4):

2 – 2 Secondary flow loop:

The secondary flow loop system supplies a hot air flow to the test section. It contains the following components: a high pressure air blower, air heaters, and plenum, as shown in the figure (5). The system is proved a maximum air flow rate and temperature at the test section of (250m^3/h) and (120 ˚C), respectively.

2 – 3 Test section configuration:

The test section is designed to suit the impingement test. The experiments were performed in an open duct is showed in the figures (6) which illustrates the test section configuration for impingement, The impingement experimental set-up contains the secondary flow that passes through the impingement part (pipe is perforated with impingement holes) and impacts the inner surface of the test target, while the mainstream flow works as a resistive film on the outer surface of the target, then both flows leave the test section the figures (7 , 8 , 9) illustrate schemes of plenum , figure (10) shows target, and figure (11)shows the dimension of test section .

2 – 4 Measuring Devices:

i – Temperatures sensing devices:

Five thermocouples type K and temperature recorder type BTM – 4208SD were used in this study to measure the secondary and mainstream flows temperatures at different locations. For secondary flow loop, one thermocouple was used to measure $T_{orifice}$ in the entrance of the orifice meter; the second thermocouple was used to measure $T_{injection}$ in the plenum close to the injection holes, T_{stream} was a mean stream temperature . The $T_{orifice}$ was used to calculate the mass flow rate of the secondary flow. The other two thermocouples were used to measure T_{out} in the exit of the test section exit, respectively. The temperatures reading T_s, T_{out}, and $T_{injection}$ were used to calculate the heat transfer coefficient and film effectiveness of impingement cases. All thermocouples and reader were calibrated at Central organization for standardization and quality control.

ii – Orifice meter:

The secondary flow rate was measured by using 50 mm diameter orifice plate. The orifice plate is made according to BS1042 [19].

iii – Pitot-static tube:

The mainstream velocity was measured by using standard ellipsoidal nosed Pitot- static tube with curved junction (N.P.L Standard).

iv – IR camera:

Thermography is the use of an infrared imaging and measurementcamera to"see" and "measure" thermal energy emitted from an object.Thermal [20], or infrared addition, is electromagnetic radiation that is not visible because its wavelength islonger than that of visible light. Unlike visible light, in the infrared world, everythingwith a temperature above zero emits heat.

Matheology:

i – heat transfer coefficients:

The measured thermodynamic properties available are the plenum chamber entry temperature (T_{in}), the jet temperature (T_j), the exit secondary flow temperature (T_{out}), the mainstream flow (T_∞), target plate thermal conductivity (K), mass flow rate of the secondary flow (m.) and the target outer surface temperature distribution(T_{wo}). The principles of the present heat transfer calculation are supposed that the energy transfers from hot flow to a surface of target and travels into the material of target in a one-dimensional pathway normal to the surface. The heat loss from the impingement jet is entirely removed by the mainstream flow that flowing over the outer surface of the target. The test section outer side is insulated to minimize the energy loss to the surrounding and resultant uncertainty in the heat into the fluid. Also, the internal heat loss by radiation is very small and can be neglected. Figure (12) illustrates the location of thermocouples and the heat transfer flow logic.

Therefore, the total heat lost from the impinging jets flow by mainstream

flow can be calculated as follows.

$$Q_L = m \cdot C_p \cdot \Delta T \quad (1)$$

Where $\Delta T = T_j - T_{out}$

T_j :is the measured jet total temperature.

T_{out} :represents the average outlet secondary flow temperature (cross flow exit temperature).

Since the test section is insulated completely, the heat transfer process can be regarded as adiabatic process, thus the heat lost removed from the test target by the mainstream flow in convection is equal to the heat transfer through the wall by conduction as:

$$Q_L = Q_{conv.} = Q_{cond.}$$

$$Q_L = K\, L\pi\, \frac{\Delta T}{\ln(Rout)-\ln(Rin)} \quad (2)$$

Where , $\Delta T = (T_{win} - T_{wout})_{av.}$

Then,

$$(T_{win})_{av.} = \frac{Q_L \cdot [\ln(Rout) - \ln(Rin)]}{k \cdot \pi \cdot L} + (T_{wout})_{av.}$$

The average heat transfer coefficient ($\overline{h_{in}}$) of the inner target surface is given by:

$$\overline{h_{in}} = \frac{Q_L}{A\left((T_{fluid})_{av.} - (T_{win})_{av.}\right)} \quad (3)$$

Where

$$(T_{fluid})_{av.} = \frac{T_j + T_{out}}{2}$$

And the average heat transfer coefficient ($\overline{h_{out}}$) of the outer target surface is given by:

$$\overline{h_{out}} = \frac{Q_L}{A\left((T_{fluid})_{av.} - (T_{wout})_{av.}\right)} \quad (4)$$

$$(T_{fluid})_{av.} = \frac{T_\infty + T_{wout}}{2}$$

ii – Cooling effectiveness:

Wall cooling effectiveness is estimated to check the activeness of the cooling systems, effusion, impingement/effusion and impingement. In general, the film cooling effectiveness (η) is a dimensionless temperature ratio that describes the relationship between the local temperature of the cooled surface to the coolant and mainstream temperatures, T_S and T_∞.

iii– Discharge coefficient (C_d):

Discharge coefficient (C_d) is a parameter that is used to express the performance of a hole, and is commonly defined as the ratio of the amount flow going through a hole, or number of holes, to the predicted flow based on the compressible flow for a specific physical area [59].Both the hole geometry and flow condition are restricts the discharge coefficient values Discharge coefficient (C_d) is calculated by applying the definition of (C_d) then.

$$C_d = \frac{\dot{m}}{A_h \cdot N_h \cdot \sqrt{2 \cdot \rho \cdot \Delta p}} \quad (5)$$

iv – Blowing ratio (BR):

To study the effect of the secondary flow variation amount with fixed values of mainstream flow, a blowing ratio (BR) is introduced. (BR) is defined as the ratio of the mass flow rate per unit area of the secondary flow to the mass flow rate per unit area of the mainstream flow. BR is given by the following equation:

$$BR = \frac{\rho_S \cdot U_S}{\rho_\infty \cdot U_\infty} \quad (6)$$

3 – Experimental Results and Discussion:

The IR images of outer target surface are showed in figures (13),(14) for configuration of inline and stagger array holes arrangement respectively at different**(BR)** in the range (0.5 , 1 , 1.5 ,2) and different **(h / d)** in the range (1,2,3) respectively,The variation average film cooling effectiveness **($\overline{\eta}$)** with blowing ratio **(BR)** were presented in figure (15) and (16) for inline and stagger holes array at different space **(h/d = 1 , 2 ,3)** the results showed that **($\overline{\eta}$)** was increased with increasing in (BR) back to increase in amount of secondary flow while **($\overline{\eta}$)** was lower with increasing **(h/d)**The narrowspacing between jet arrays and target plate induced a highly constrain flowthat leads to high momentumof jet impingement and cross flow and additional backpressureThepresence of a strong cross flow at low h/d tends to disturb the impinging jetpattern at the stagnation region and enhance the surface boundary layer thatdegrades heat transfer rates, contributing to create low wall coolingeffectivenessGenerally, better enhancement was observed at**(h/d = 1),**As stated before, the array inline holes best result than stagger array holes because the cross flow created by the jets didn't disturbthe next jet that stands in its pathway for inline arrangement while instaggered configuration, the cross flow disturbed the downstream jets, figures (17,18) for inline and stagger holes arrangementrespectively which are presented variation of heat transfer coefficient **(h_{in})** at inner target surface with **(BR)** at different **(h / d)** because the same reason of increasing **($\overline{\eta}$)** this agreement with [20]. figures (19), (20) for inline and stagger

holes arrangement respectively that are appeared mutation of heat transfer coefficient**(h $_{out}$)** at outer target surface with **(BR)** at different **(h/d)** which were noted small variation like constant in almost since heat transfer coefficient is effected clearly by velocity (or Re $_{mainstream}$) but small changeable with variation in the surface temperature, and like that figures (21) , (22) for inline and stagger holes arrangement respectively they are illustrated discharge coefficient **(Cd)** variation with **(BR)** at different **(h/d)** , the enhancement in **(Cd)** with increases **(BR)** and **(h/d)** because increase in the amount of secondary flow and decrease the flow resistance . The narrow spacing between jet arrays and target plate induced a highly constrain flow that leads to high momentum of cross flow and additional backpressure that agreement with [11].

4 – Conclusions:

In this paper is studied that effect of blowing ratio, hole jet arrangement, distance between jet and target on impingement cooling in leading edge of gas turbine for wall cooling effectiveness, heat transfer coefficients and discharge coefficient.

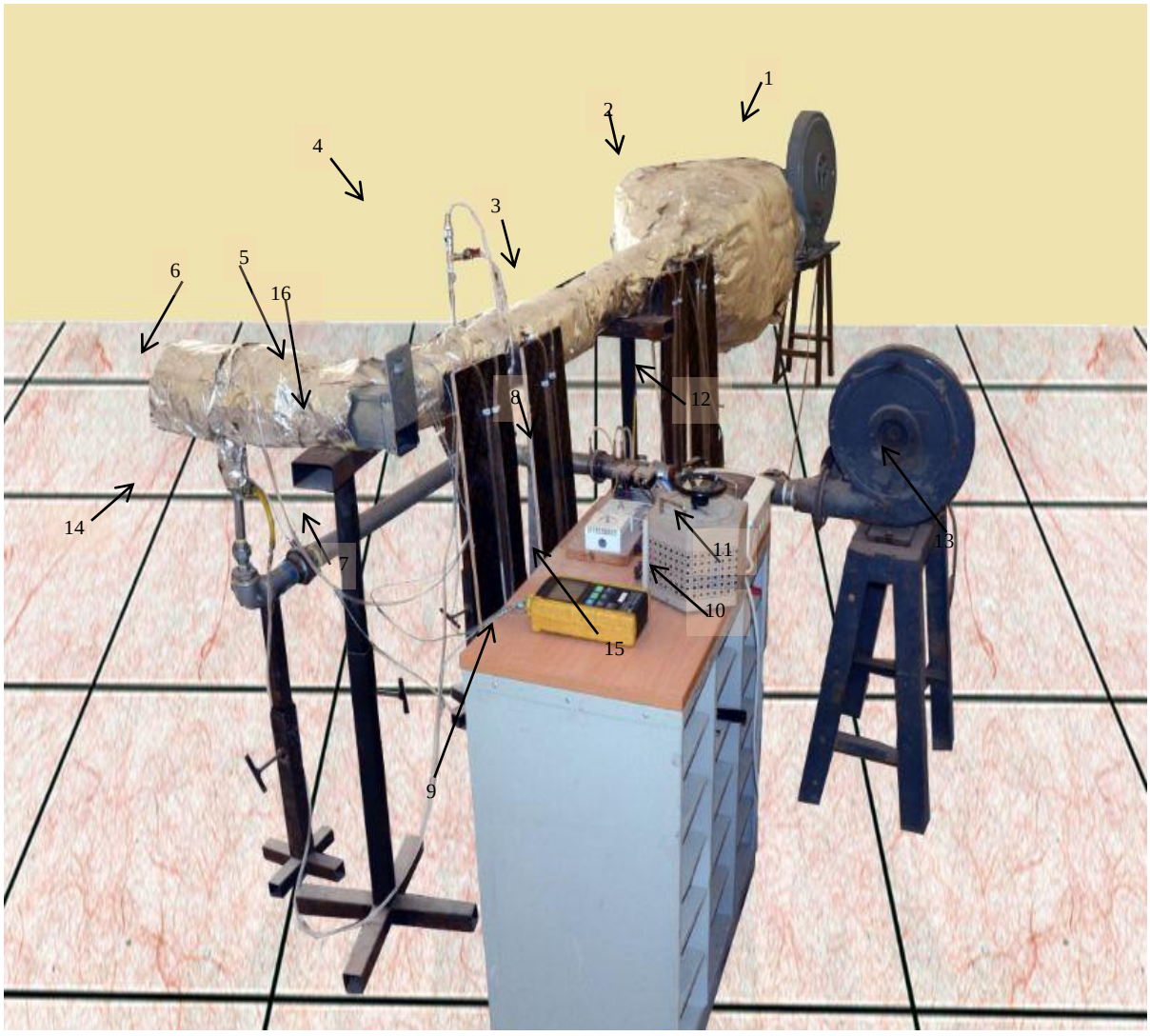

Fig. 1: The test rig

1 – Mainstream flow blower 2 – Settling chamber 3 – Straight duct 4 – Pitot tube 5 – Elbow duct 6 – Test section 7 – Secondary flow heater 8 – Orifice flat plate meter 9 – Temperature recorder 10 – vairic 11 – Control box 12 – Barometer 13 – Secondary flow blower 14 – Plenum 15 – Electronic voltage regulator 16 – Camera window

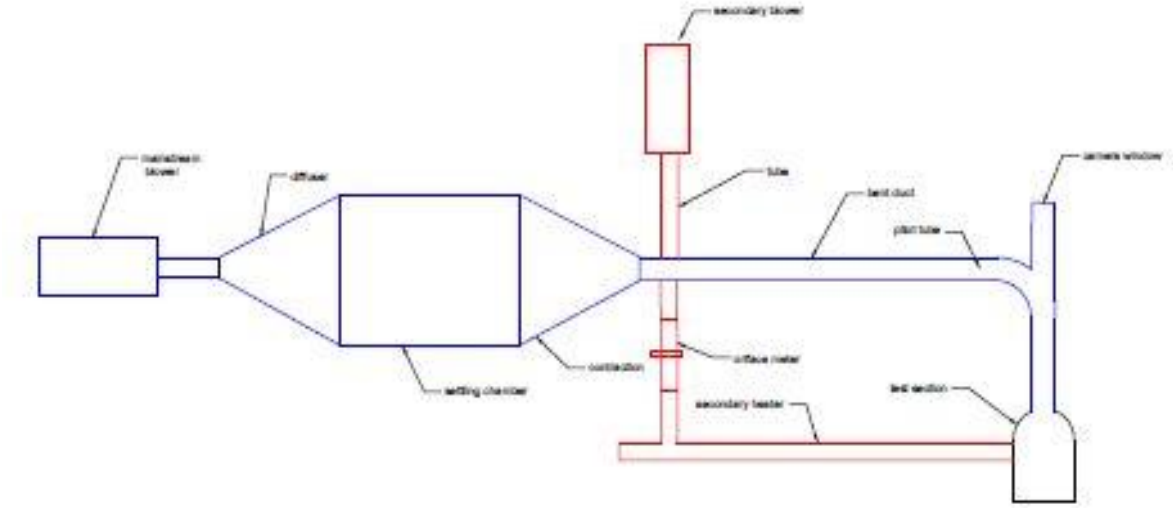

Fig. 2: Rig's scheme

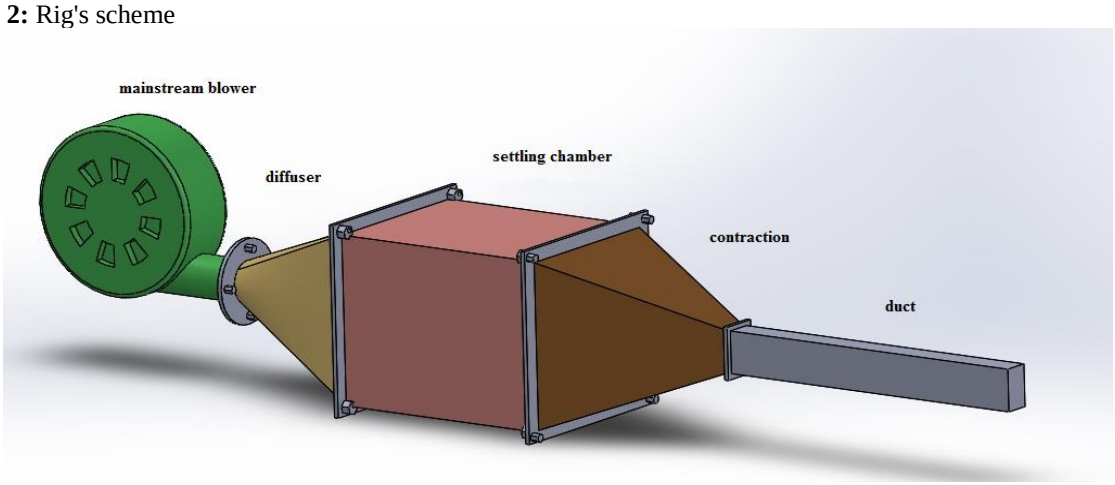

Fig. 3: Mean stream flow loop

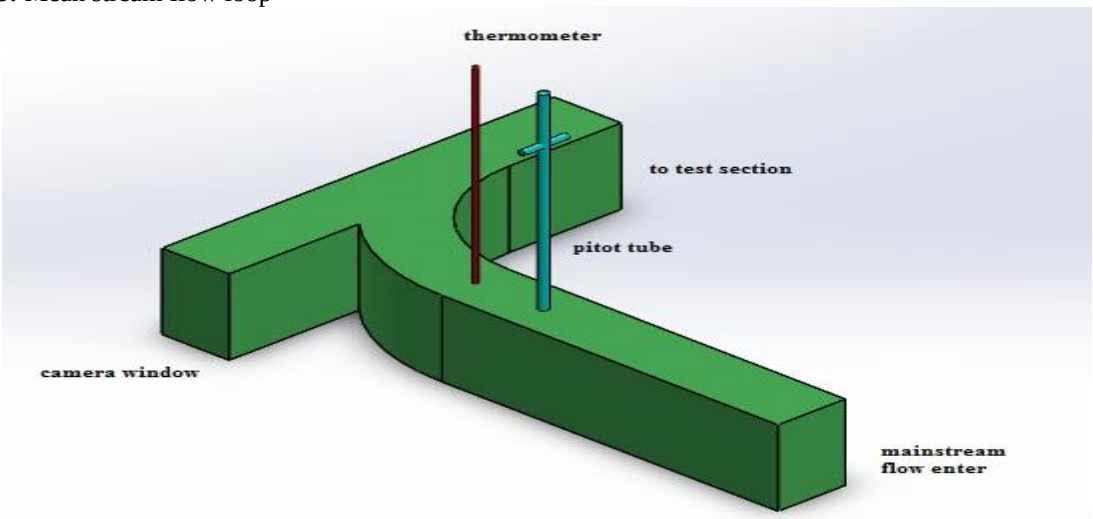

Fig. 4: Bent tube

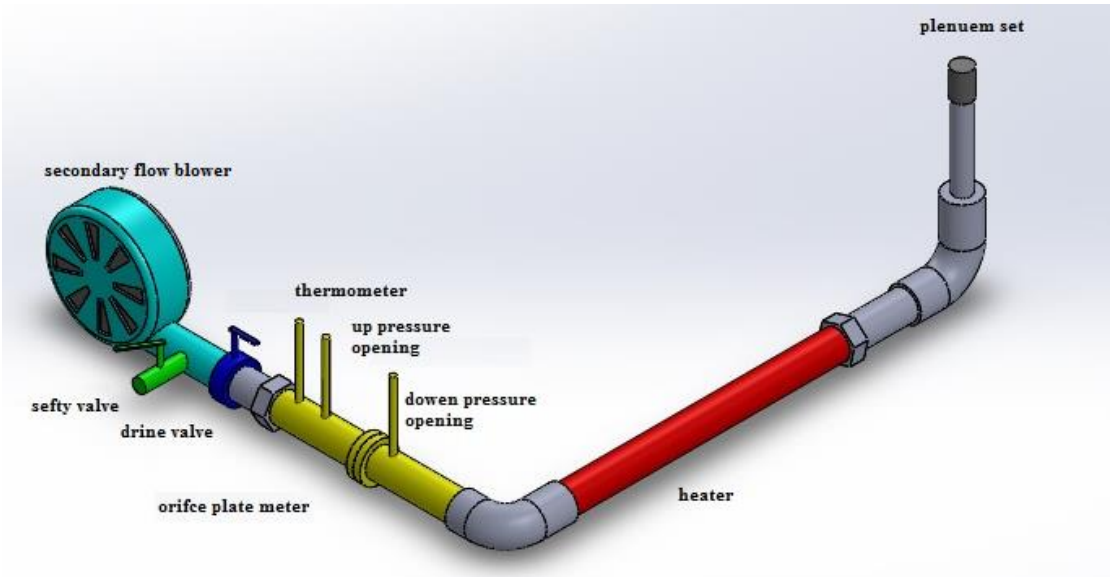

Fig. 5: Secondary flow loop

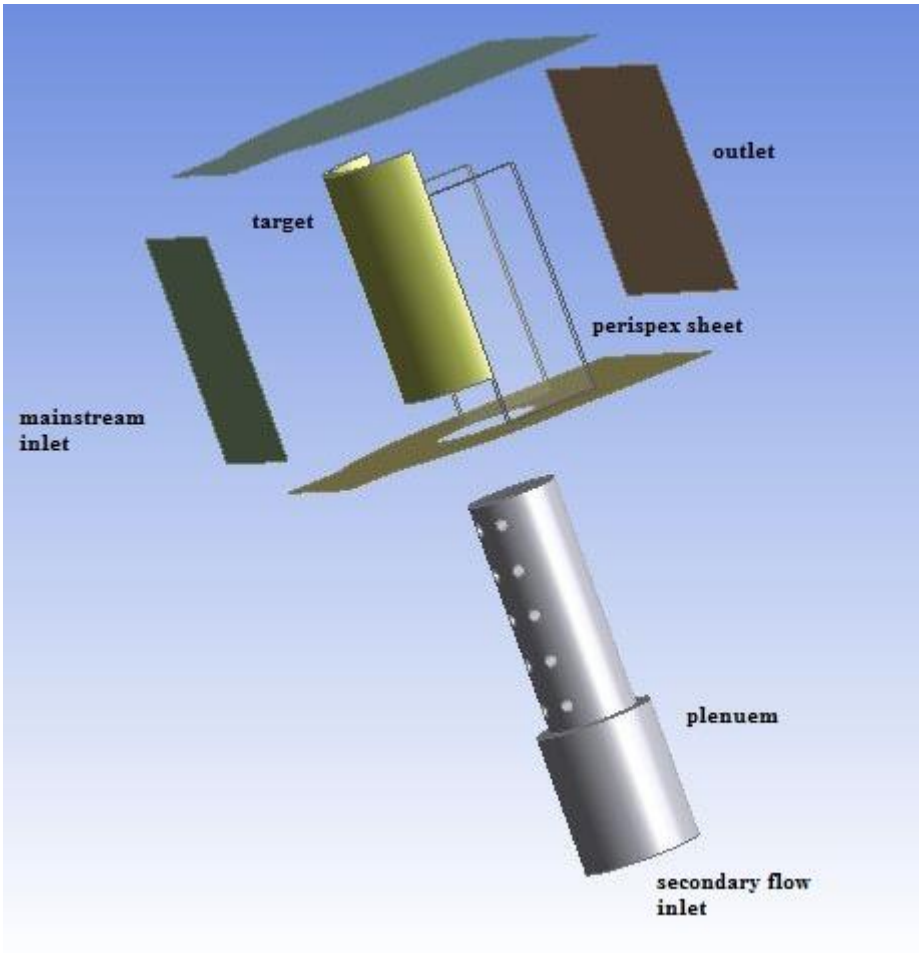

Fig. 6: Test section configuration

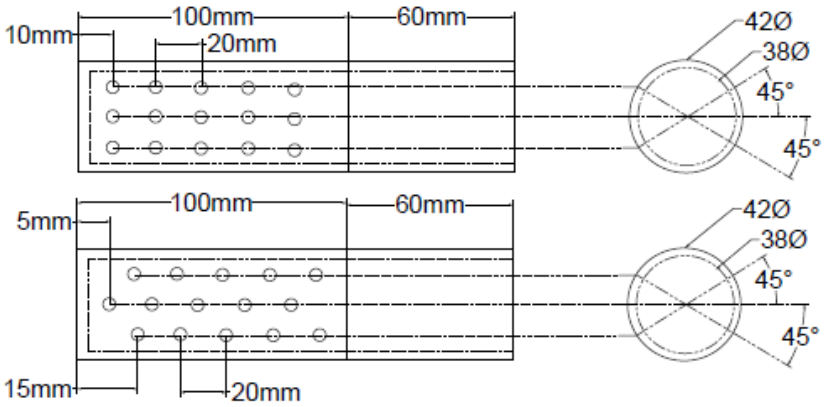

Fig. 7: inline and stagger impingement holes which h/d=1

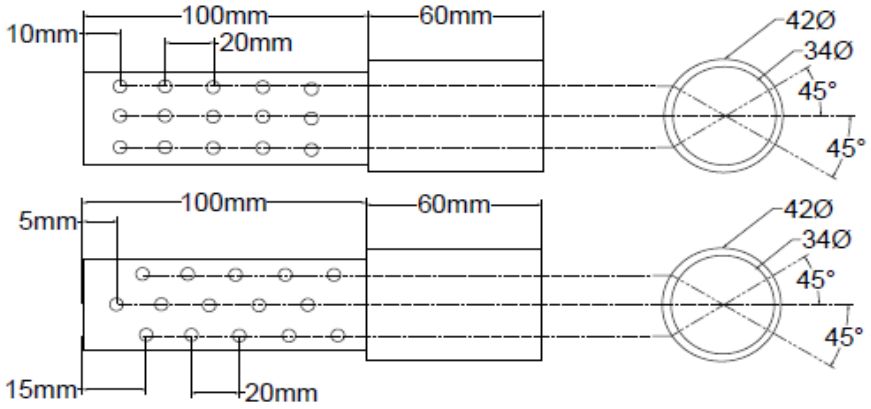

Fig. 8: inline and stagger impingement holes which h/d=2

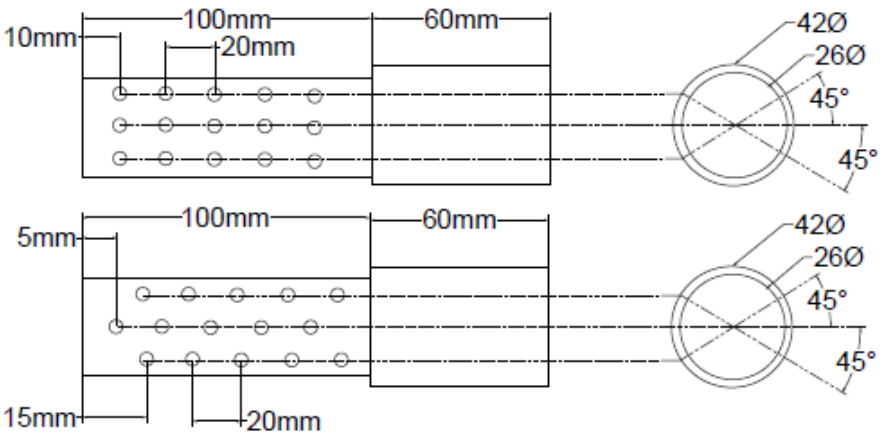

Fig. 9: Inline and stagger impingement holes which h/d=3

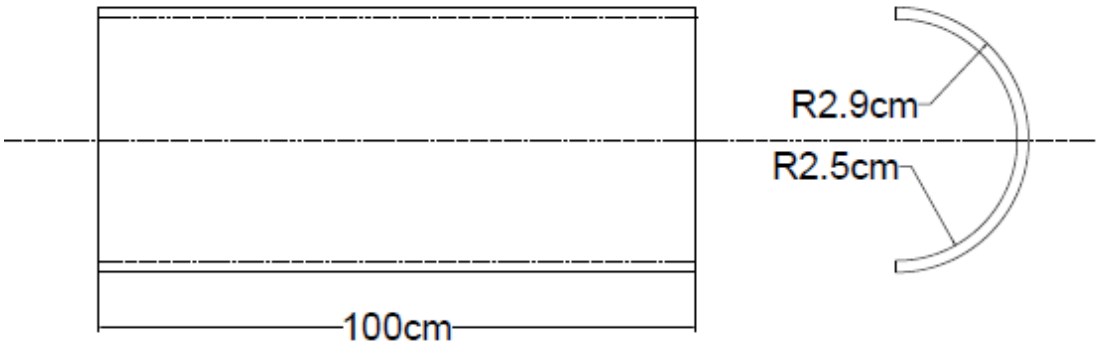

Fig. 10: Illustrate the target

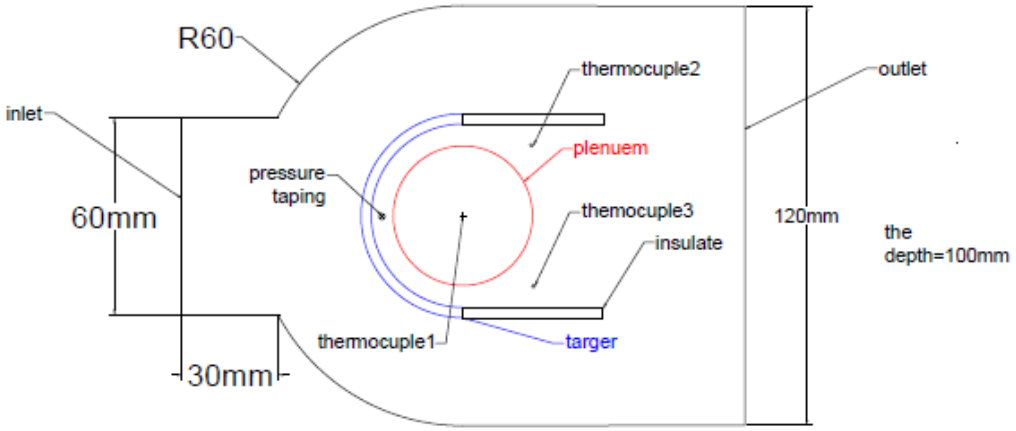

Fig. 11: dimension of test section

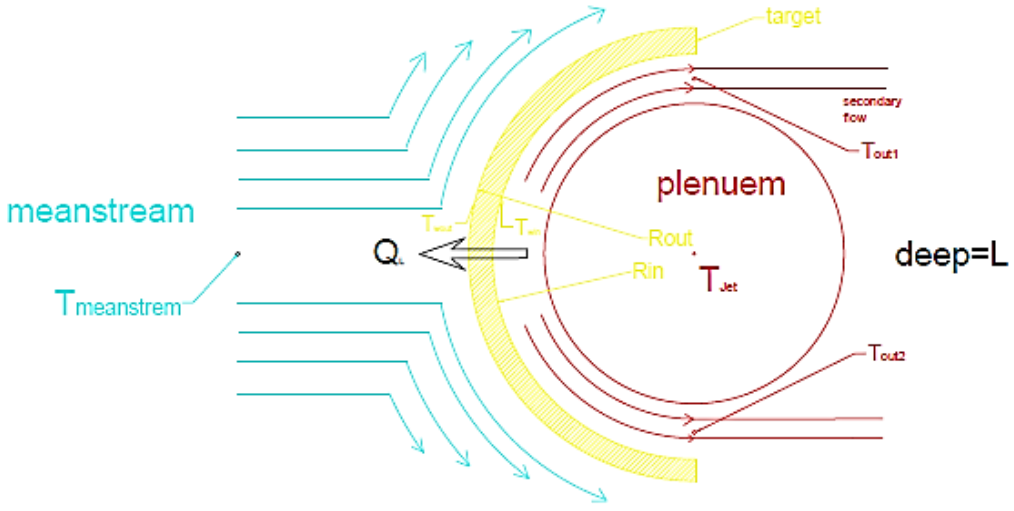

Fig. 12: Heat transfer data flow logic sketch

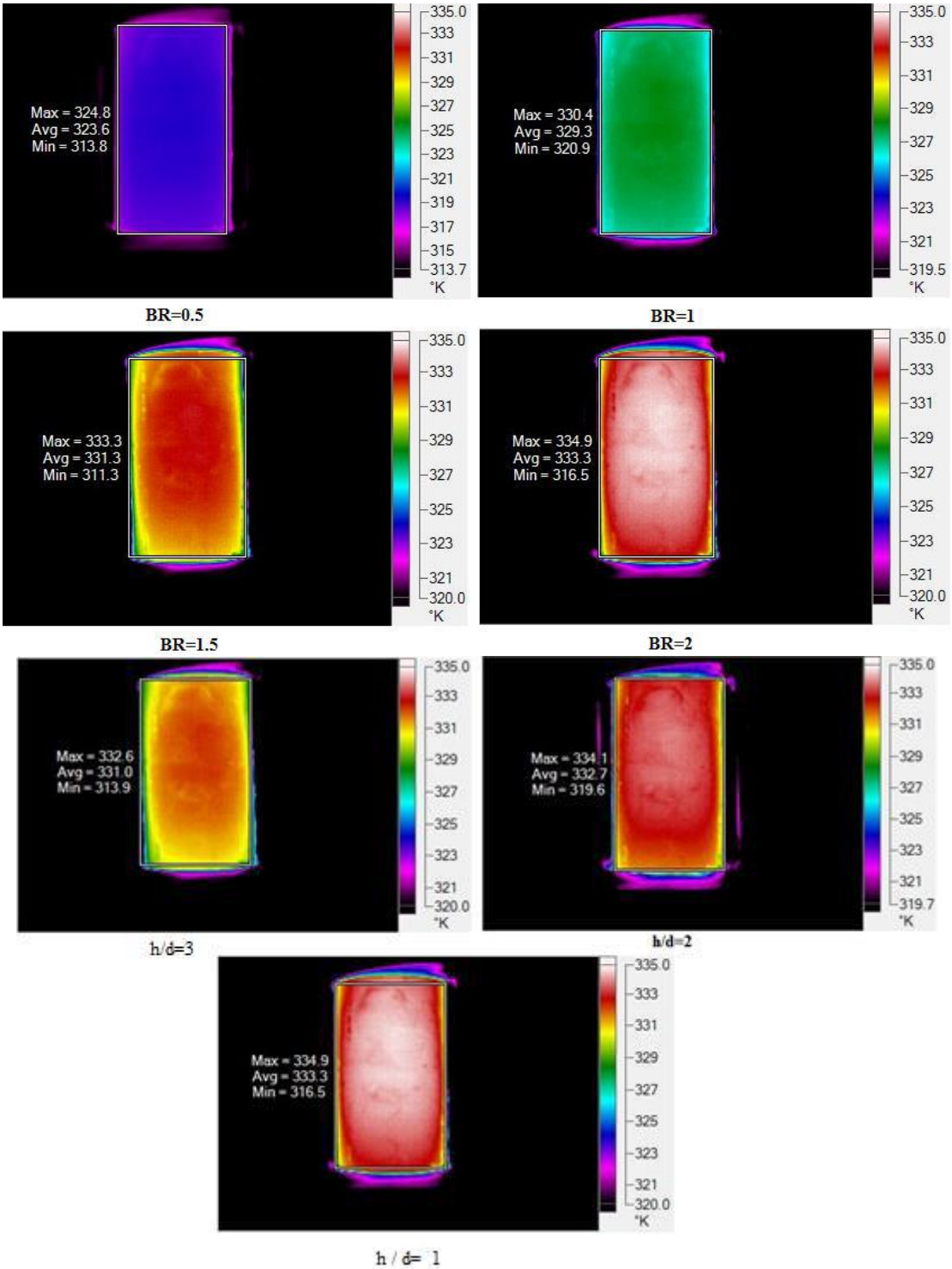

Fig. 5–13: IR images for inline jet arrangement at h/d=1 and different (BR) and at BR=2 with different (h/d)

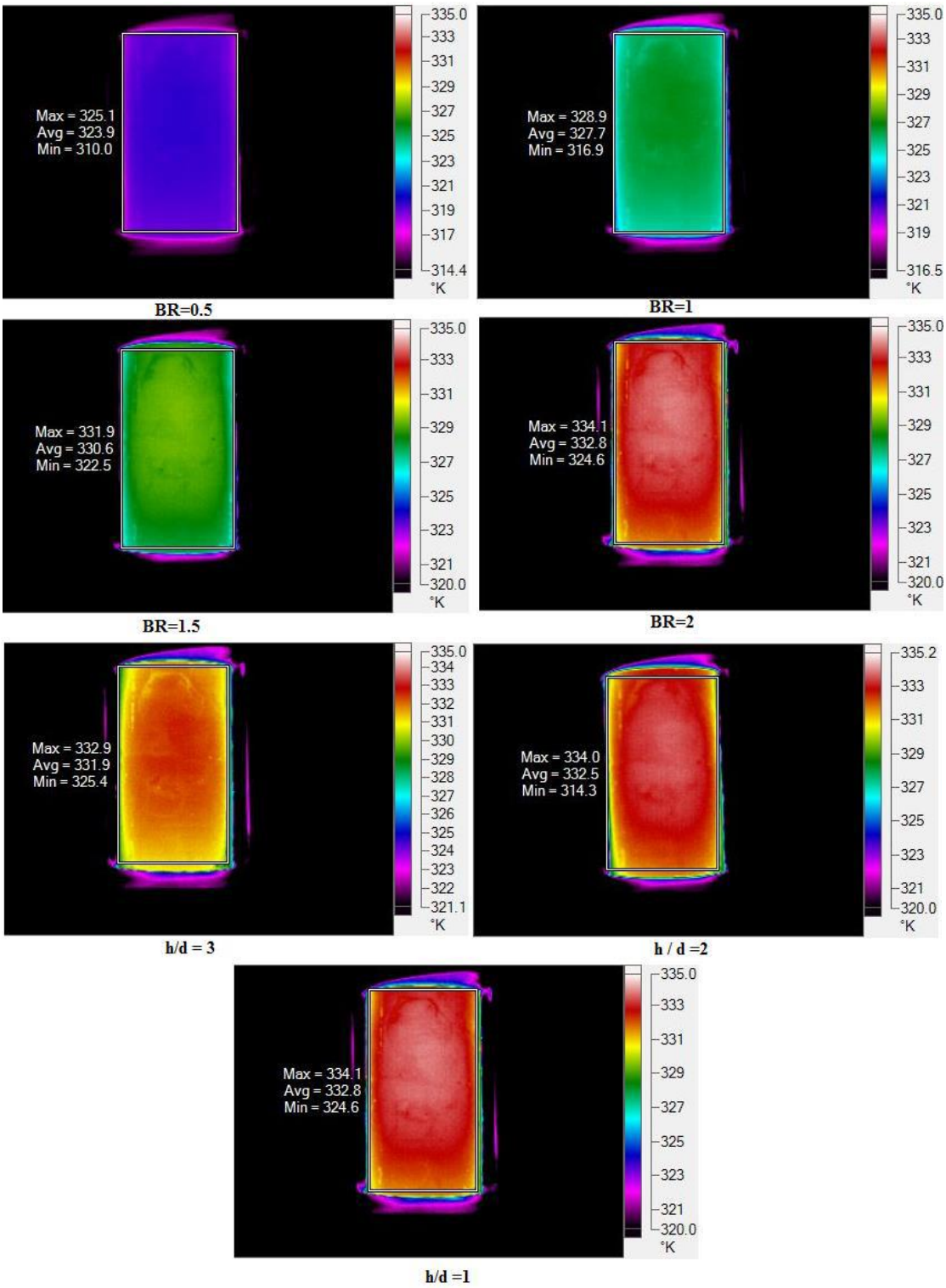

Fig. 5-14: IR images for stagger jet arrangement at h/d=1 and different (BR) and at BR=2 with different (h/d)

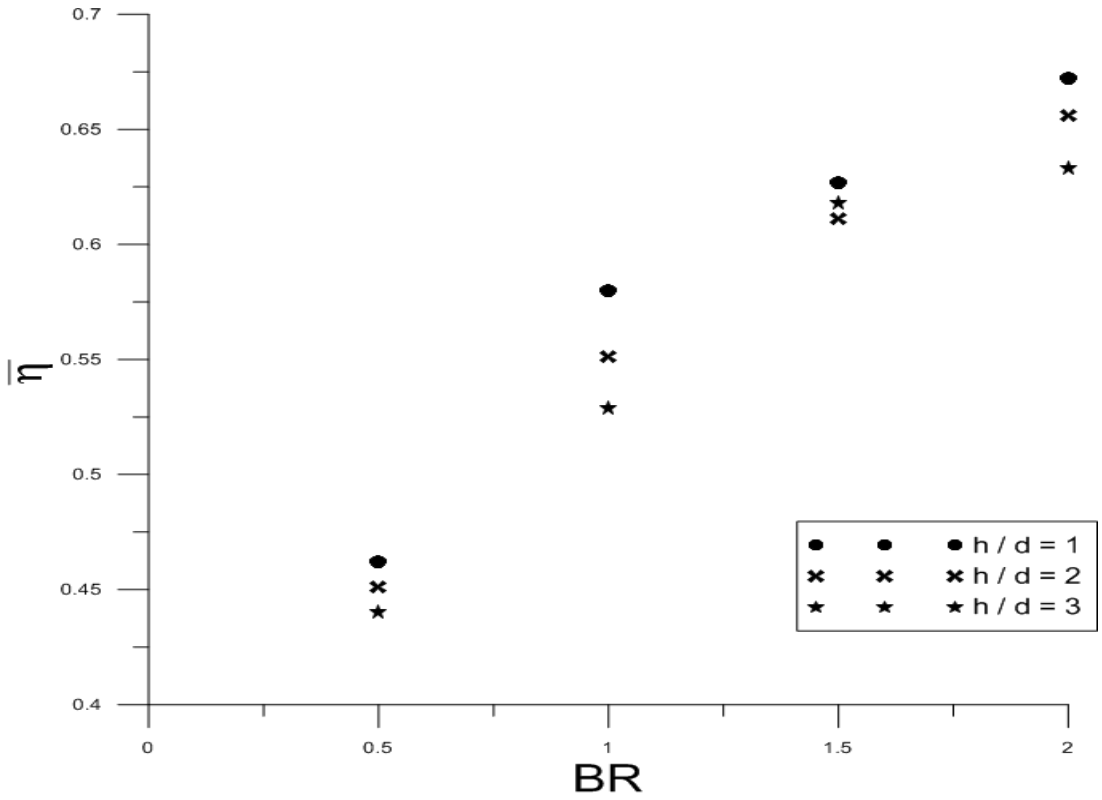

Fig. 15: variation of film cooling effectiveness for inline arrangement array with blowing ratio

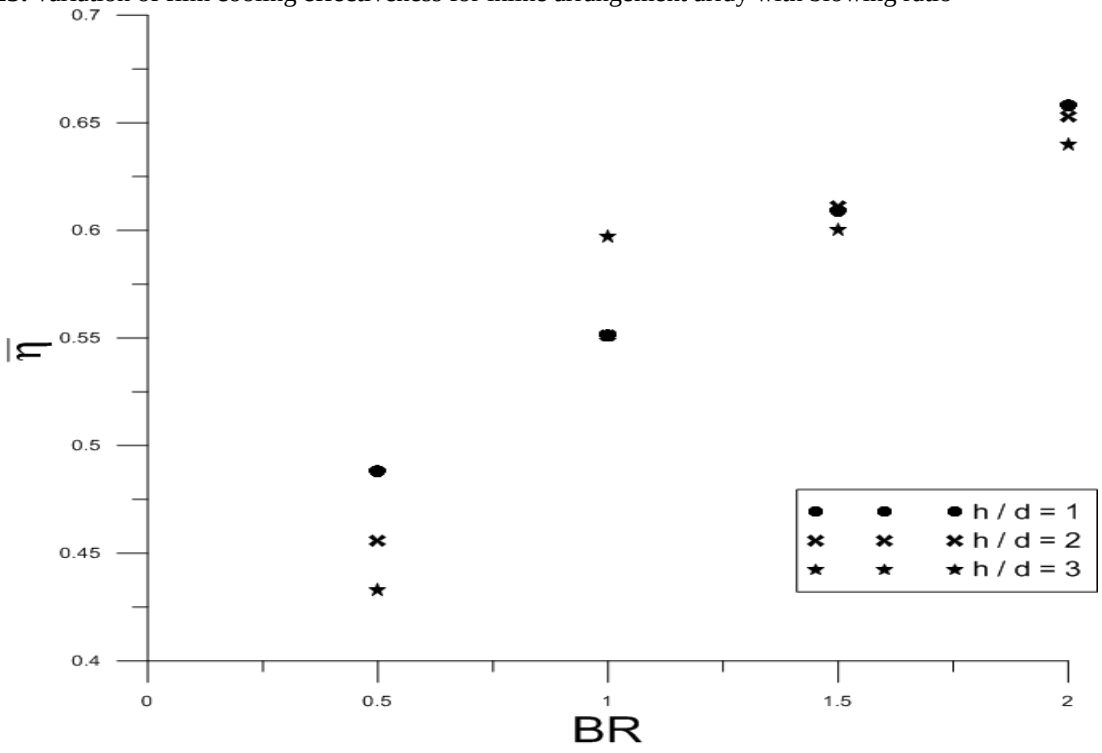

Fig. 16: variation of film cooling effectiveness for stagger arrangement array with blowing ratio

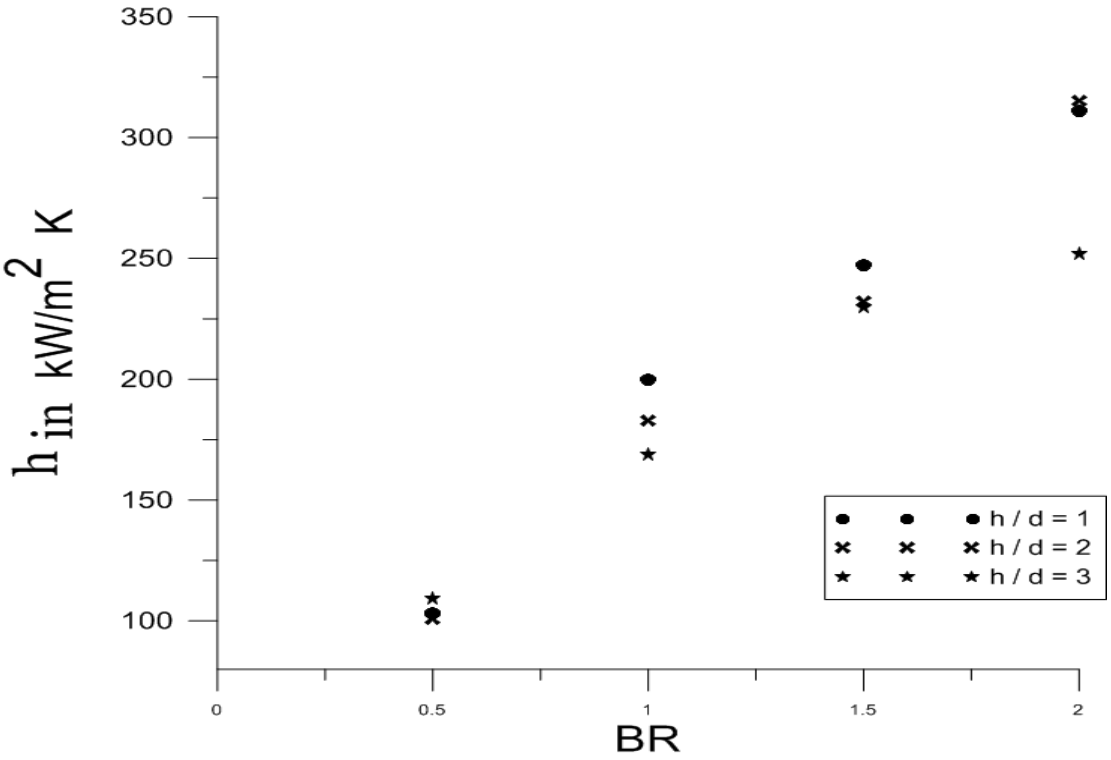

Fig. 17: h $_{in}$ for inline arrangement array with BR

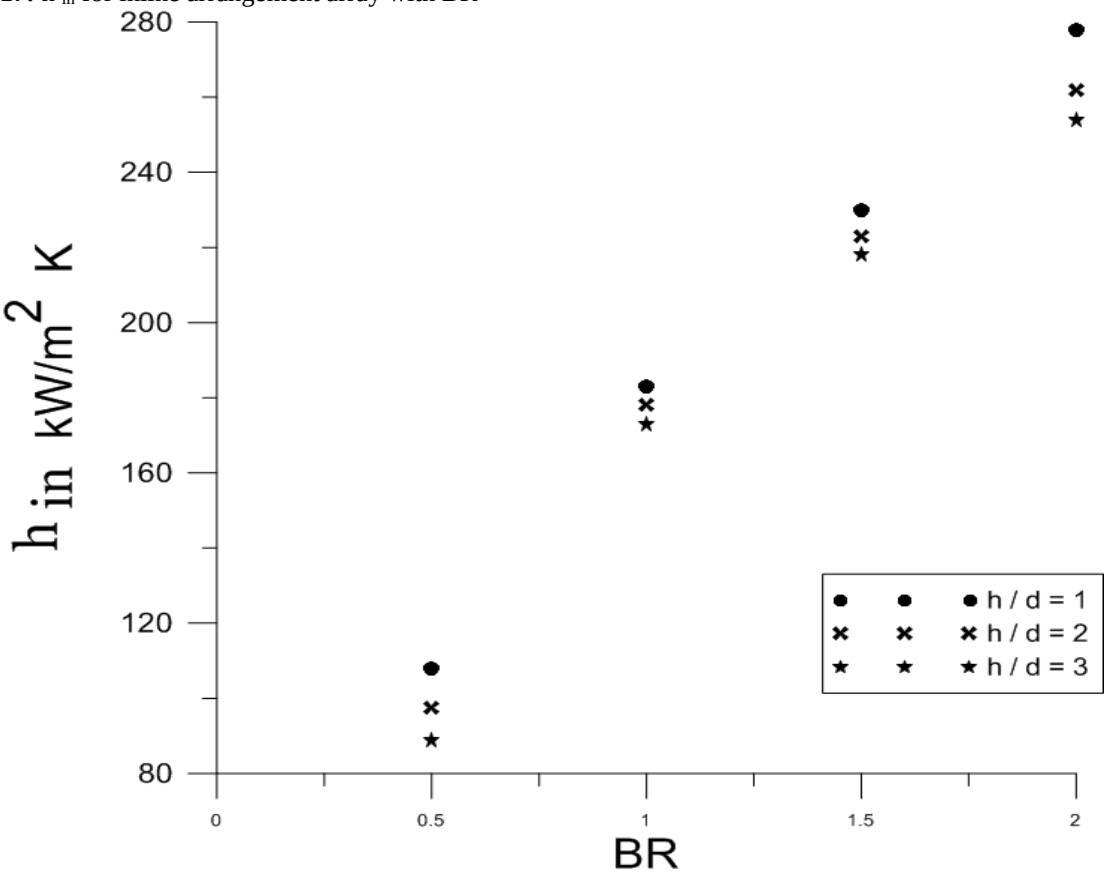

Fig. 18: h $_{in}$ for stagger arrangement array with BR

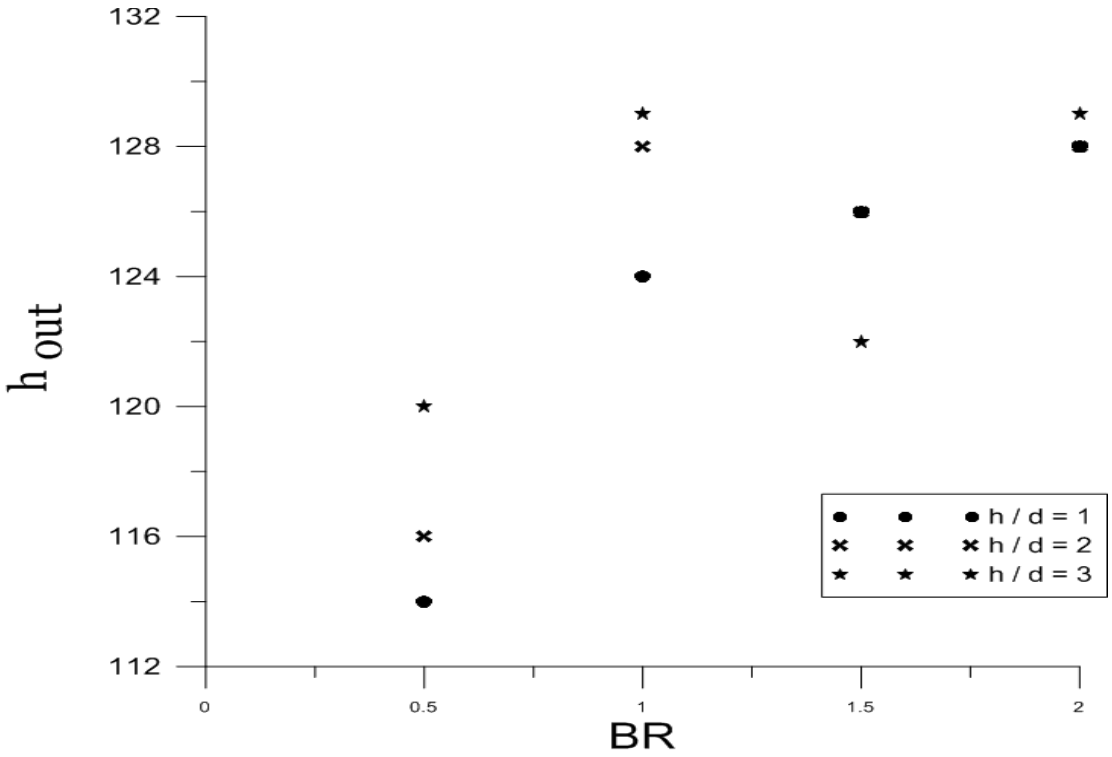

Fig. 19: h $_{out}$ for inline arrangement array with BR

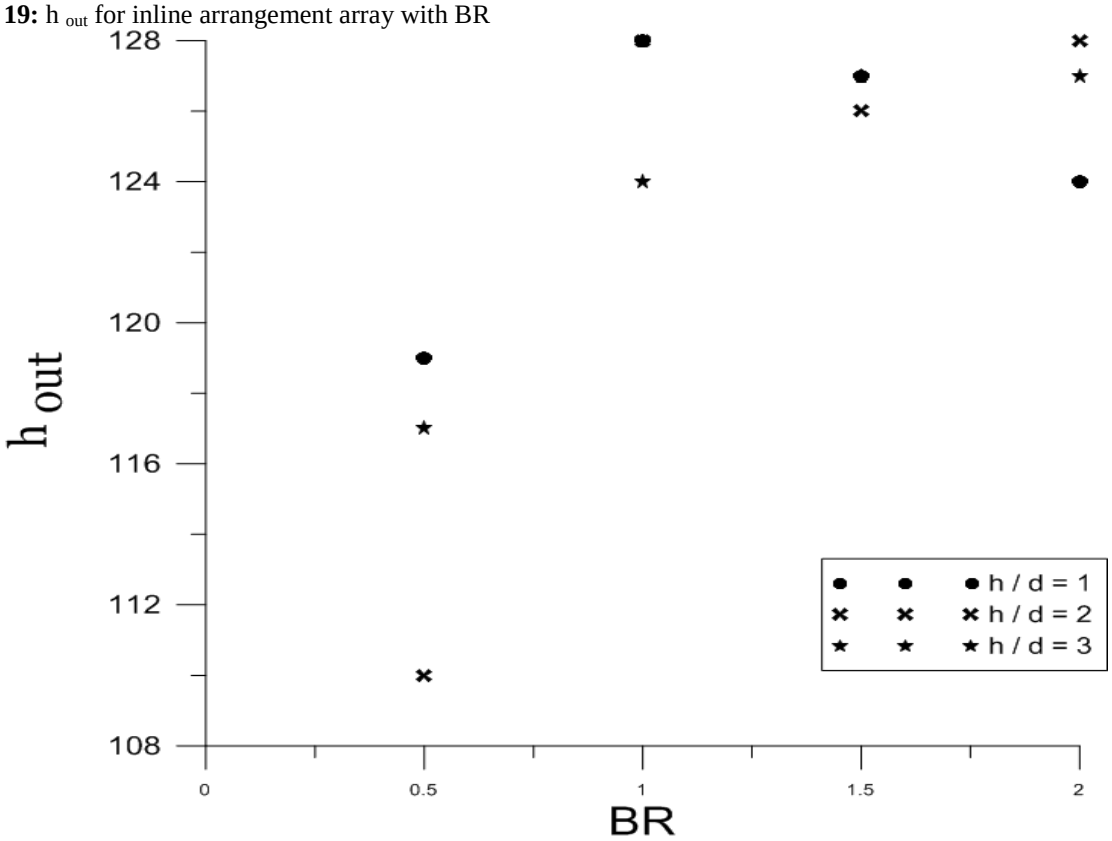

Fig. 20: h $_{out}$ for stagger arrangement array with BR

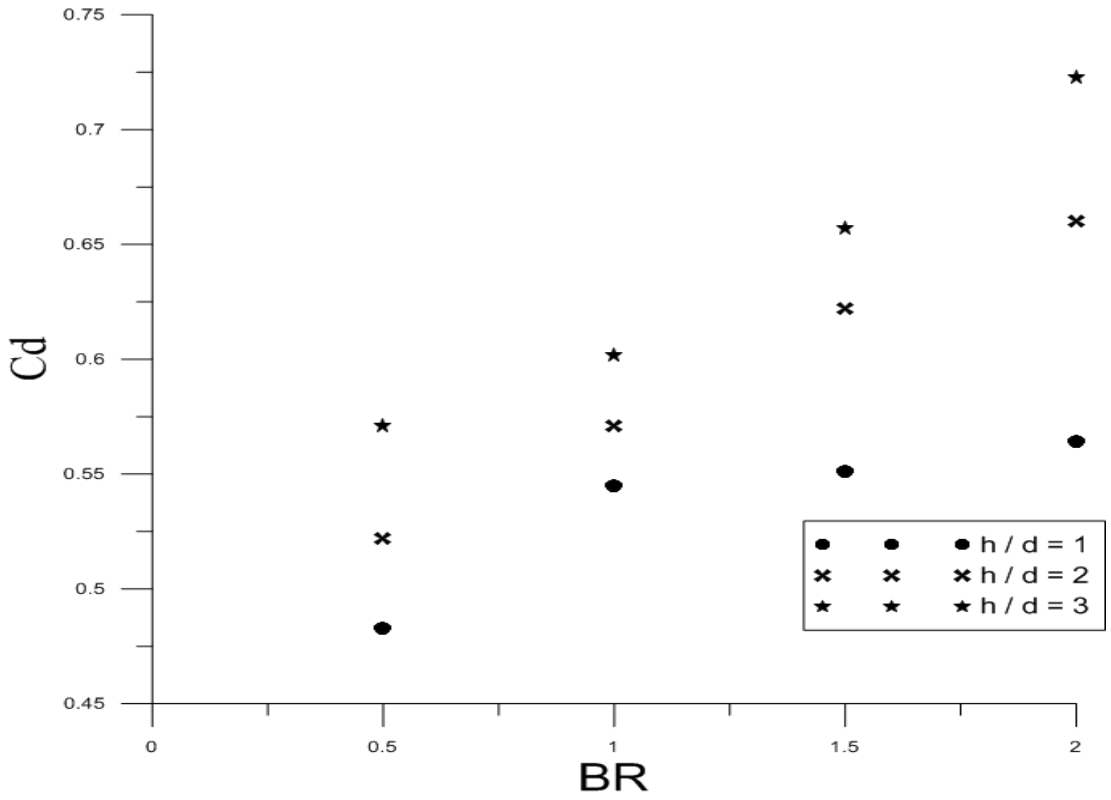

Fig. 21: C_d for inline arrangement array with BR

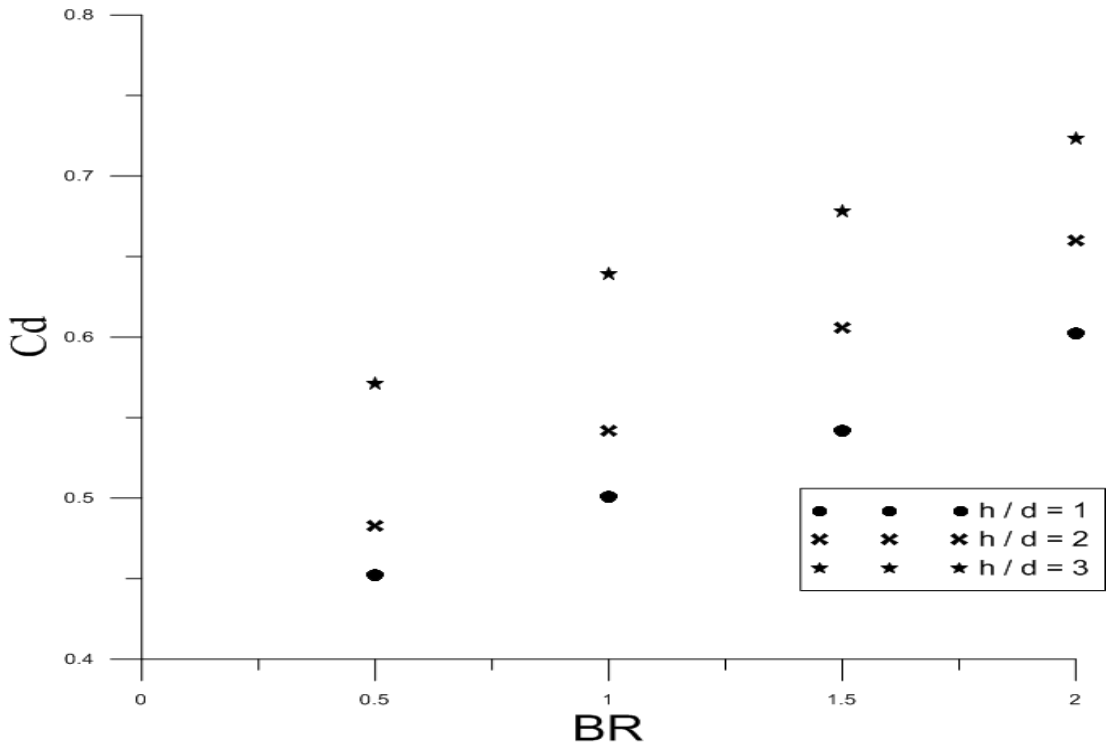

Fig. 22: C_d for stagger arrangement array with BR

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