Proceedings of Global Power and Propulsion Society
ISSN-Nr: 2504-4400
GPPS Hong Kong 2023
October 17-19, 2023
www.gpps.global

GPPS-TC-2023-0016

HEAT TRANSFER CHARACTERISTICS OF CORRUGATED WALL IMPINGEMENT WITH CROSSFLOW

Ryuta Ito
Mitsubishi Heavy Industries, Ltd.
ryuta.ito.px@mhi.com
Takasago, Hyogo, Japan

Satoshi Mizukami
Mitsubishi Heavy Industries, Ltd.
satoshi.mizukami.xa@mhi.com
Takasago, Hyogo, Japan

Atsuya Sakata
Mitsubishi Heavy Industries, Ltd.
atsuya.sakata.cx@mhi.com
Takasago, Hyogo, Japan

Ryosuke Mito
Mitsubishi Heavy Industries, Ltd.
ryosuke.mito.tm@mhi.com
Takasago, Hyogo, Japan

Keijiro Saito
Mitsubishi Heavy Industries, Ltd.
keijiro.saito.45@mhi.com
Takasago, Hyogo, Japan

Satoshi Hada
Mitsubishi Heavy Industries, Ltd.
satoshi.hada.32@mhi.com
Takasago, Hyogo, Japan

ABSTRACT

To improve the performance of a gas turbine, it is important to cool hot gas path parts with a small amount of air, and it is effective to reuse the cooling air until it heats up sufficiently. However, if the cooling air is reused in impingement cooling, there is a concern that the heated air used for upstream cooling will interfere with the impinging jet downstream, degrading its cooling performance. Therefore, the corrugated wall impingement, which can discharge the heated crossflow without adversely affecting the impinging jet downstream, is useful. In this paper, the heat transfer coefficient and fluid reference temperature (adiabatic wall temperature) of corrugated wall impingement are experimentally investigated to confirm the heat transfer characteristics.

INTRODUCTION

In order to realize a sustainable society, it is essential to promote renewable energy and achieve the efficient use of fossil fuel energy in the power generation field. Gas turbine combined cycle is expected because it has high efficiency and quick responsibility for output fluctuation of renewable energy, and it is required further efficiency enhancement. It needs a higher turbine inlet temperature and reduction of cooling consumption which requires designs with a higher temperature capability. A reduction of the cooling air consumption can be achieved with reusing the cooling air until it heats up sufficiently.

However, if the cooling air is reused in impingement cooling, which is widely used technique to reduce thermal load of hot gas path parts in gas turbines, there is a concern that the heated air used for upstream cooling interferes with the impinging jet downstream, degrading its cooling performance. Kercher and Tabakoff studied the effect of crossflow on jet impingement heat transfer for flat wall impingement, which is widely used today, and the Nusselt number decreased rapidly because the strong crossflow disrupts and pushes over downstream jets (Kercher et al., 1970). Therefore, it is required to develop a structure that widens the crossflow area while keeping the jet-to-target spacing as a flat plate impingement, and to incorporate the corrugated wall design, which traps the crossflow in the corrugations between impingement jets to reduce crossflow effects on downstream, can be one of the solutions. Some research has been conducted to reduce this detrimental effect of crossflow by installing the new structures such as corrugated wall impingement and some of them provided better heat transfer coefficient than flat wall impingement (Esposito et al., 2007; Kim et al., 2021). But degrading heat transfer coefficient is not the only concern for impingement cooling with large crossflow. When hot gas path parts exposed to high thermal load is cooled with a small amount of air, the cooling air heats up, and the high temperature crossflow disrupts the downstream jet. In the case of flat plate impingement, it has been reported that the cooling performance decreases by increasing the fluid temperature at which the high temperature crossflow impinges on the target wall (Florschuetz, 1987). In order to apply corrugated wall impingement to the actual machine, it is necessary to investigate the overall cooling performance, so the reference temperature as well as the heat transfer coefficient should be confirmed. The present study focus on the corrugated wall design. Experimental study was carried out to investigate both heat transfer coefficient and reference temperature of corrugated wall impingement of jet spacing X/D=12.5 and jet-to-target spacing Z/D=3.5 under Reynolds number Re=10,000. In this study, parameters such as X/D, Z/D, and Re were not changed, but these parameters can be studied in the future to understand the detailed heat transfer characterestics.

METHODOLOGY

A flow diagram of test facility is shown Figure 1. Experimental apparatus consists of compressor, valves, orifice flowmeters, air heater, and test section which is contained in the pressure vessel. There are two air lines in this test facility. One is impingement air line, and the other is crossflow air line. The compressor supplies air of each line to the test section and valves modulate the mass flow rate. The mass flow rate is calculated from the pressure, the temperature in front of the orifice flow meter, and the pressure difference between the front and rear side of the orifice flow meter. Impingement air enters the inlet chamber, and it is injected into test section through impinging jet hole. Crossflow air enters the inlet channel after passing through an air heater to control the crossflow temperature. The pressure is measured by pressure gauge and the temperature is measured by thermocouples for both impingement air and crossflow air to confirm the Reynolds number and crossflow ratio (CR) at inlet chamber or channel. Impingement air and crossflow air interferes with each other in the test section, and flow to the exit.

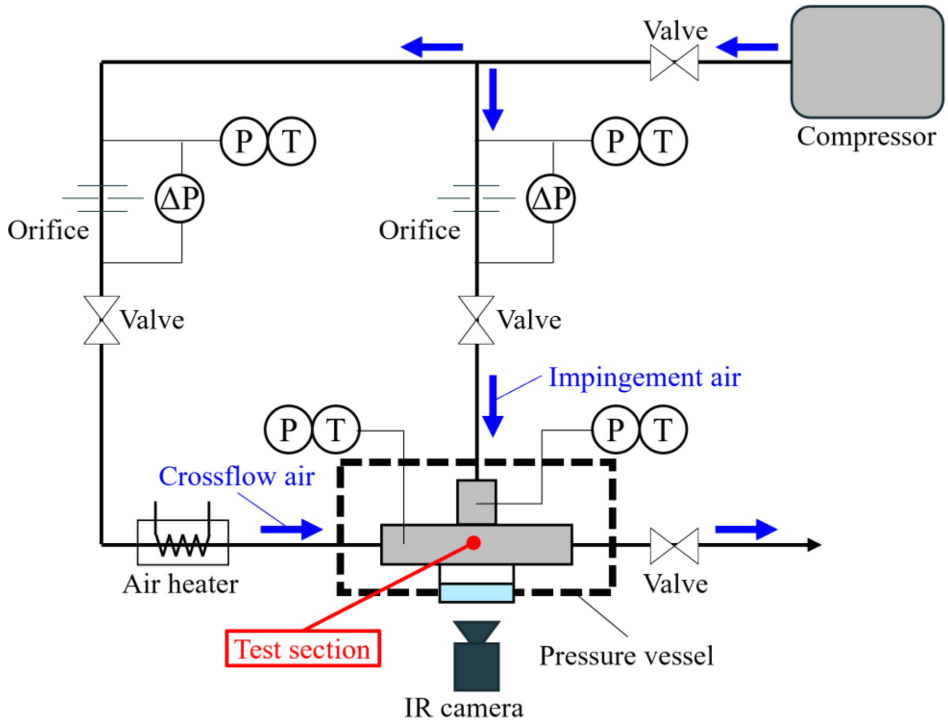


Figure 1 Flow Diagram of Facility

A schematic view of experimental apparatus is shown in Figure 2. Impingement jet air is supplied through corrugated wall test piece and impinges to target plate. Crossflow air is supplied through bellmouth inlet. Experimental parameters and geometrical parameters of corrugated wall impingement are shown in Table 1. The jet Reynolds number is 10,000 and CR is from 0 to 0.3. CR represents the crossflow velocity (U_{cr}) relative to the jet velocity (U_j), where U_j is calculated from total mass flow rate of impingement air and all hole areas in the jet plate, and U_{cr} is calculated from mass flow rate of total crossflow air plus impingement air upstream of evaluation region and the crossflow channel area. The jet spacing and jet-to-target spacing are denoted by X/D and Z/D, respectively. Dimensions of X/D and Z/D are 12.5 and 3.5, respectively. The corrugated wall height (H/D) is 9.

The thickness of target plate made by hastelloy is very thin (0.25mm) and it can be heated uniformly by electricity. In this experiment, the test section including the corrugated wall test piece and the thin-film heater are housed in a pressure vessel because the experiment is carried out under high pressure condition. The pressure vessel is also equipped with the air supply and exhaust system and an infrared transmission window flange for temperature measurement by IR camera.

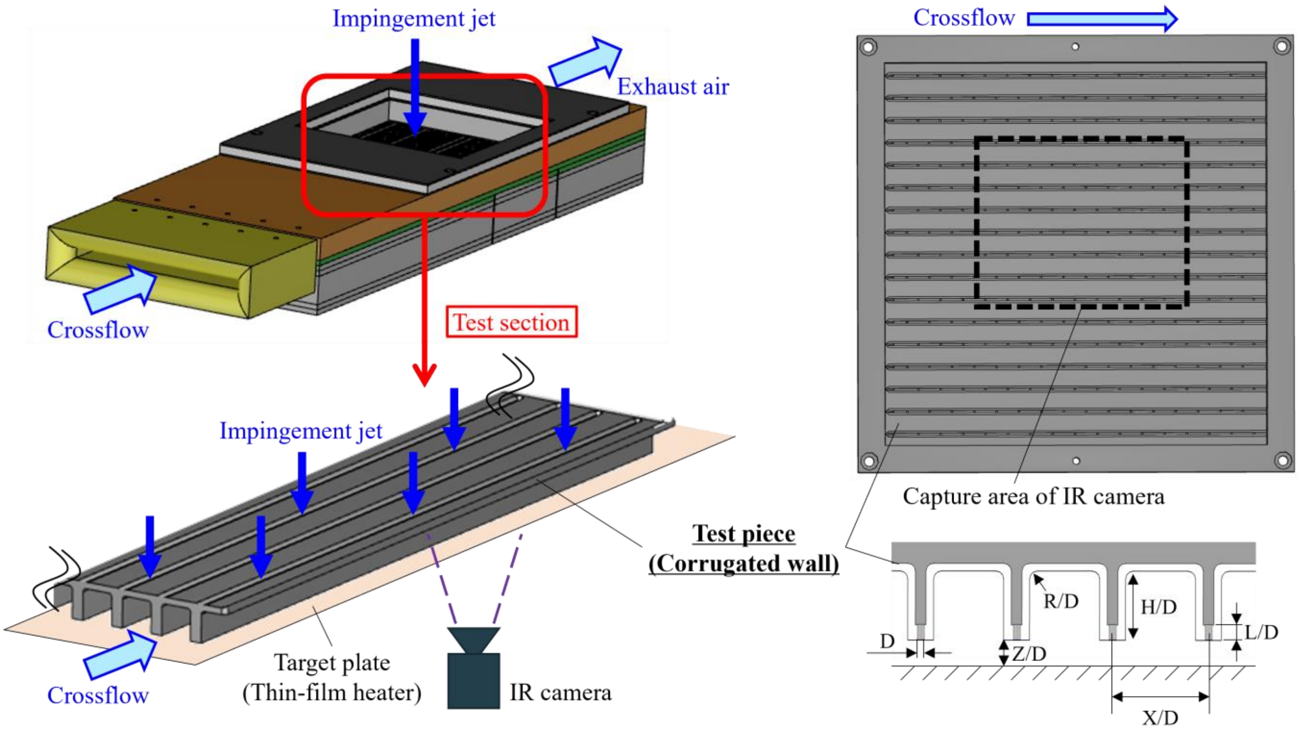


Figure 2 Test Section and Corrugated Wall Test Piece

Table 1 Experimental Parameters and Geometrical Parameters of Corrugated Wall Impingement

Jet Reynolds number [-]	Re_D	10,000
Crossflow ratio [-]	CR	0～0.3
Jet spacing [-]	X/D	12.5
Jet-to-target spacing [-]	Z/D	3.5
Corrugated wall height [-]	H/D	9
Jet hole length [-]	L/D	2
Fillet radius [-]	R/D	1

The test method is shown in Figure 3. In the present study, two kinds of experiment were conducted to investigate the heat transfer characteristics of corrugated wall impingement. The first test is conducted to obtain heat transfer coefficient for the corrugated wall impingement. The heat transfer coefficient distribution was measured by a steady state method using a thin-film heater and an IR camera. Impingement air and crossflow air without heating by air heater supplied into test section. The thin-film heater heated uniformly by electricity, and the temperature distribution on the thin-film heater surface was captured with an IR camera from the back of target plate after the temperature became steady. CaF_2 (calcium fluoride) was used for the infrared transmission window in terms of the wavelength range and intensity of transmission. The heat transfer coefficient, α is calculated as follows:

$$\alpha = \frac{q_t}{T_w - T_j} \qquad (1)$$

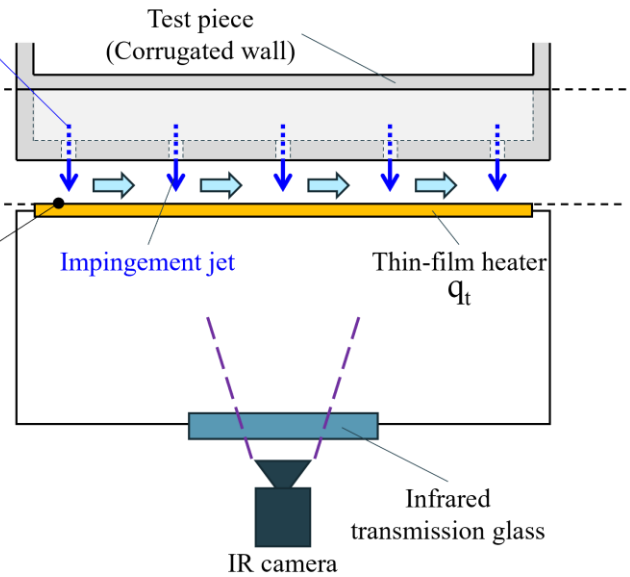
where q_t, T_w, T_j are heat flux of thin-film heater, target wall temperature, and impingement jet temperature, respectively. The heat transfer coefficient is converted into the dimensionless form of the Nusselt number, Nu given in equation (2).

$$\mathrm{Nu} = \frac{\alpha D}{k} \qquad (2)$$

The second test is conducted to measure the fluid temperature difference influence factor, η defined as follows:

$$\eta = \frac{T_{ref} - T_j}{T_{cf} - T_j} \qquad (3)$$

where T_{ref}, T_j, T_{cf} are fluid reference temperature, impingement jet temperature, and crossflow temperature, respectively. Since the temperature order differs greatly between the test condition and the actual machine condition, dimensionless number η, which is similar to film cooling effectiveness, was introduced to estimate the effect in the actual machine. The fluid reference temperature is adiabatic wall temperature on the target wall. The effect of crossflow temperature can be evaluated quantitively by introducing η. The high η means the reference temperature is close to the crossflow temperature. In order to confirm the effect of the crossflow temperature on the target wall, impingement air and crossflow air with heating by air heater supplied into test section and impinged on the thin-film heater which was turned off. The reference temperature distribution on the target wall was captured with an IR camera after the temperature became steady, and η is calculated by equation (3).



	Test 1	Test 2
Evaluation target	α	η
Thin film heater	ON	OFF
Air heater	OFF	ON

Figure 3 Test Method

Experimental uncertainties in Reynolds number, wall temperature, heat transfer coefficient, and fluid temperature difference influence factor were ±2%, ±0.4℃, ±6%, and ±7%, respectively. Maximum contribution to Reynolds number, heat transfer coefficient, and fluid temperature difference influence factor came from mass flow rate, wall temperature, and adiabatic wall temperature respectively.

RESULTS AND DISCUSSION

Heat transfer coefficient (Test1)

Results are presented for jet Reynolds number of 10000. Figure 4 shows detailed heat transfer coefficient distribution for CR of 0.03, 0.12, 0.22, and 0.32. The crossflow is from left to right. CR=0.03 is no initial crossflow condition. The pattern of heat transfer coefficient distribution does not change significantly if CR is less than 0.32.

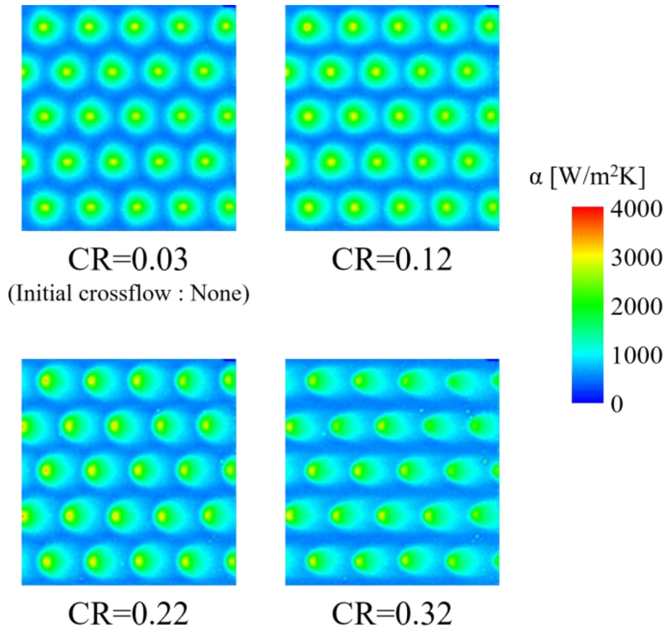


Figure 4 Detailed Heat Transfer Coefficient Distribution of Corrugated Wall Impingement

Figure 5 presents nomalized Nusselt number Nu/Nu0 for each CR case. Nu0 is area averaged Nusselt number for CR=0.03, that means crossflow is almost zero. Nusselt number does not change until CR exceeds 0.1. When CR become 0.3, area averaged Nusselt number decrease by only about 8% although stagnation Nusselt number is lower (Figure 4). This is because the Nu number in the wall jet region, which occupies a larger area than the stagnation point, is slightly increased by the high crossflow velocity. The crossflow effect for heat transfer coefficient is very small at corrugated wall impingement because it traps the crossflow in the corrugations between impingement jets to reduce crossflow effects on downstream.

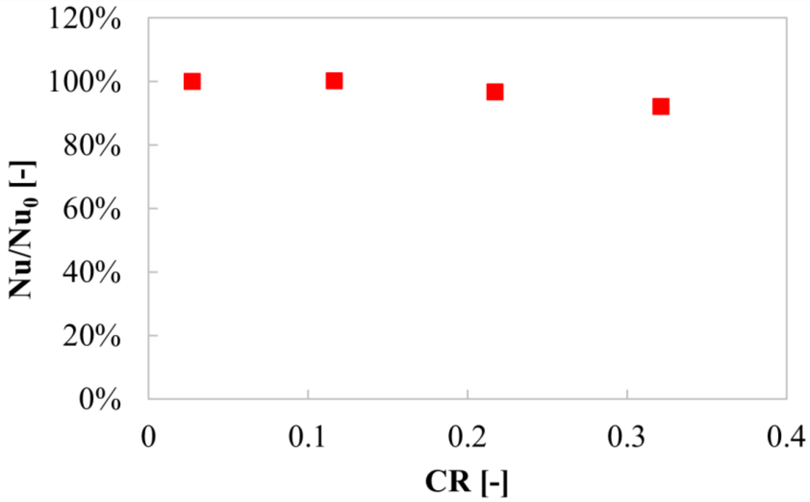


Figure 5 Normalized Nusselt number for Corrugated Wall Impingement

Fluid temperature difference influence factor (Test2)

Detailed wall temperature distribution are shown in Figure 6. Measurements were conducted by changing crossflow temperature and CR to investigate those effect. As a result, the higher crossflow temperature and CR cause the higher wall temperature. It means crossflow temperature has detrimental effect for corrugated wall impingement cooling and the effect becomes more serious when CR is higher. Figure 7 presents fluid temperature difference influence factor for each CR case. Even if CR is small, crossflow temperature has some effect for cooling performance. When CR exceeds 0.3, fluid temperature difference influence factor almost achieve 0.8. This means that the effect of crossflow temperature must be taken into account in the design for corrugated wall impingement.

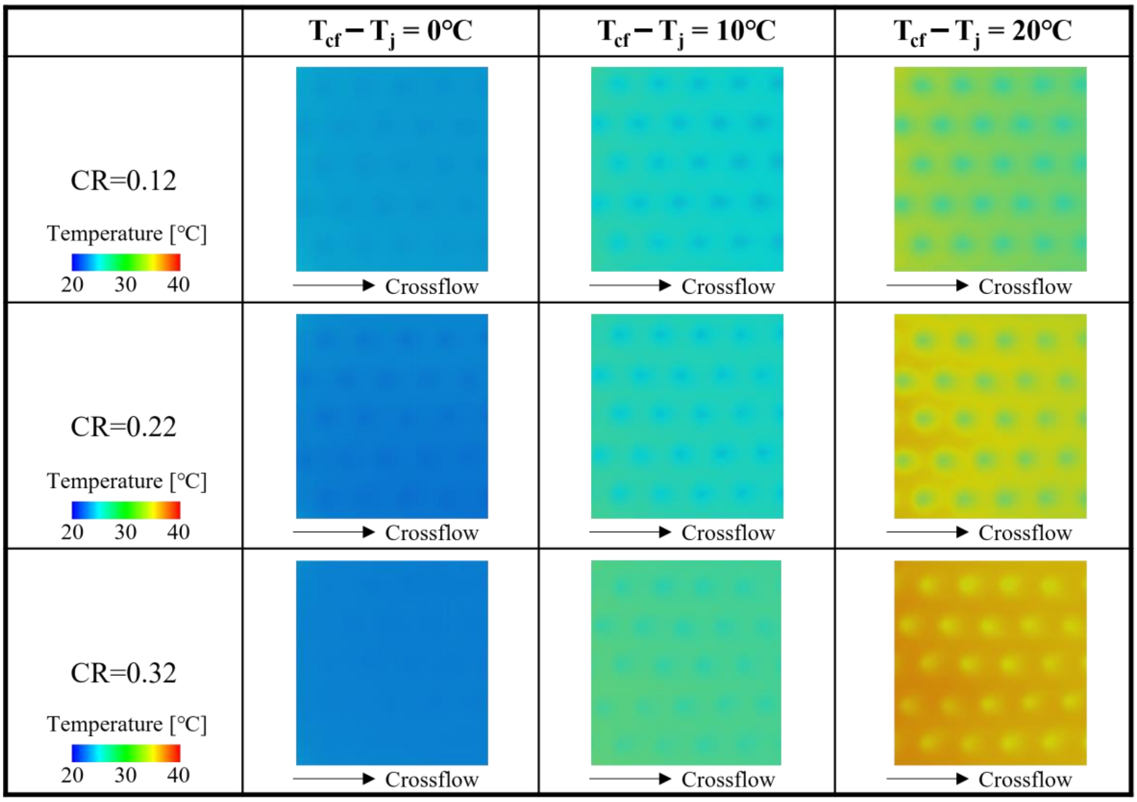


Figure 6 Detailed Wall Temperature Distribution for Corrugated Wall Impingement

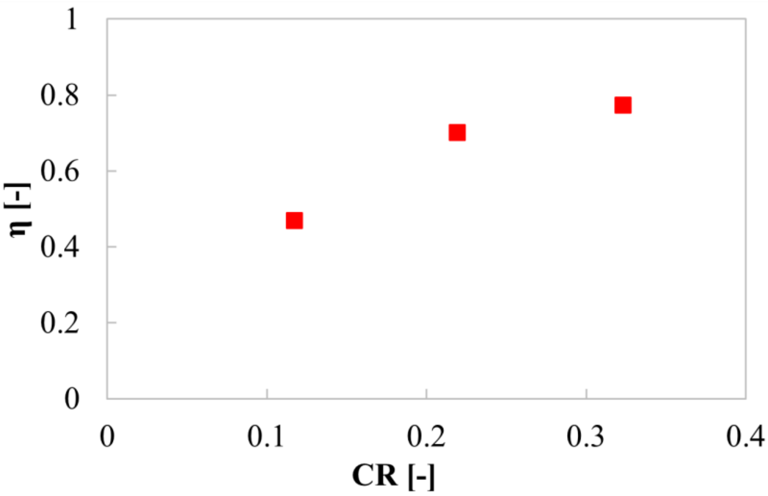


Figure 7 Fluid Temperature Difference Influence Factor for Corrugated Wall Impingement

CONCLUSIONS

It is important for clean power generation to enhance the efficiency of gas turbine and it needs a higher turbine inlet temperature and reduction of cooling consumption. A reduction of the cooling air consumption can be achieved with reusing the cooling air until it heats up sufficiently. When the cooling air is reused in impingement cooling, crossflow effect may be large. Therefore, the corrugated wall design, which traps the crossflow in the corrugations between impingement jets to reduce crossflow effects on downstream, is one of the most effective solutions and the present study focus on the the heat transfer characteristics of corrugated wall impingement.

Experimental study was conducted to confirm the heat transfer coefficient and fluid temperature difference influence factor for corrugated wall impingement. The heat transfer coefficient does not change significantly even if CR becomes large. However, The fluid temperature difference influence factor has detrimental effect for corrugated wall impingement cooling and the effect becomes more serious when CR is larger. Although the temperature order of the actual gas turbine condition is different from the test condition, crossflow temperature effect should be represented by dimensionless number η, which is obtained at low temperature condition if the flow field is the same. This means that the effect of crossflow temperature must be taken into account in the design for corrugated wall impingement.

NOMENCLATURE

CR	Crossflow ratio (=U_{cr}/U_j)
D	Jet hole diameter
H	Corrugated wall height
k	Thermal conductivity of fluid
L	Jet hole length
P	Pressure
Nu	Nusselt number
Nu_0	Nusselt number without crossflow
Pr	Prandtl number
q_a	Heat flux from air heater
q_t	Heat flux from thin-film heater
R	Fillet radius
Re_D	Jet Reynolds number
T	Temperature
T_{cf}	Crossflow temperature
T_j	Impingement jet temperature
T_{ref}	Fluid reference temperature
T_w	Wall temperature
U_{cr}	Crossflow velocity
U_j	Jet velocity
X	Jet spacing
Z	Jet-to-target spacing
α	Heat transfer coefficient
η	Fluid temperature difference influence factor defined as equation (3)

REFERENCES

Esposito, E. I., Ekkad, S. V., Kim, Y. and Dutta, P. (2007). Comparing Extended Port and Corrugated Wall Jet Impingement Geometry for Combustor Liner Backside Cooling. *ASME PaPer No. GT2007-27390*.

Florschuetz, L. W. and Su, C. C. (1987). Effects of Crossflow Temperature on Heat Transfer Within an Array of Impinging Jets. *ASME Journal of Heat Transfer*, 109, pp. 74-82.

Kercher, D. M. and Tabakoff, W. (1970). Heat Transfer by a Square Array of Round Air Jets Impinging Perpendicular to a Flat Surface Including the Effect of Spent Air. *Journal of Engineering for Power*, 92(1), pp. 73-82.

Kim, T., Jung, E. Y., Bang, M., Lee, C., Moon, H. K. and Cho, H. H. (2021). Heat Transfer Measurement for Array Jet Impingement with Castellated Wall. *ASME PaPer No. GT2021-58895*.