A HEAT TRANSFER ANALYSIS FROM A POROUS PLATE WITH TRANSPIRATION COOLING

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Present study is focused on improving heat transfer from a porous plate by cooling of air with transpiration cooling. Effects of Reynolds number of the air channel flow and particle diameter on cooling effectiveness of porous plate and efficiency of system were investigated experimentally. It was observed that increasing Reynolds number of 15.2% causes a decrease of 6.9% on cooling efficiency of the system and a decrease of 8.6% on cooling effectiveness of porous plate. Decreasing particle diameter causes a significant decrease on surface temperature and an increase on cooling effectiveness of porous plate. Difference of cooling effectiveness of porous plate from d_p = 40-200 μm is 12%. Verification of this study was also shown by comparing experimental results of this study with literature.

Key words: *experimental, heat transfer, porous plate, transpiration cooling*

Introduction

Transpiration cooling, compared with passive and ablative protection, is presumably the most efficient way to enable the structure withstanding of high heat loads and thermal management. This method has been used to protect solid surface exposed to high-heat-flux, high temperature environments such as liquid rocket thrusters, water oxidation technology, hypersonic vehicle combustors, gas turbine blades, and the nose of aerospace vehicles during the re-entry phase of their flight to atmosphere. By using transpiration cooling not only solid surface can be protected but also temperature of hot gas can be reduced. So it can also be used to cool air. In transpiration cooling process: coolant fluid is injected into a porous matrix from the opposite side of heat flux, conduction and convection processes occurs while passing pores, finally the coolant causes to a thin film layer on the hot side surface to reduce the heat flux coming into the porous matrix and to cool the hot gas stream. The outlet temperature of the hot gas can also be reduced by evaporation of the coolant.

The porous material in transpiration cooling can be of different types. Stainless steel plates, some ceramics and porous plastics can be used as a porous media. The structure generally has many pores with diameter changing between 5 μm and 200 μm and can be made of different shaped particles, porous plate thicknesses can be change between 1 and 10 mm or more.

Because this method has high heat transfer potential, studies on transpiration cooling are recently increasing. Some experimental investigations on transpiration cooling can be found in the literature. Jiang *et al.* [1] investigated turbulent flow and heat transfer in a rectangu-

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lar channel without and with transpiration cooling experimentally and numerically. They used bronze particles as porous wall and air as coolant. Porosity was 0.45 mm and particle diameter was 0.1 mm. Their results showed that the transpiration cooling greatly increases the boundary-layer thickness and reduces the wall skin friction, the wall temperature and the convection heat transfer coefficient. Liu *et al.* [2] investigated the transpiration cooling mechanisms for thermal protection of a nose cone experimentally and numerically. Their results showed that the injection rate strongly influenced the cooling effectiveness. The increase of Reynolds number significantly reduced the cooling effectiveness. Coolant physical properties, especially specific heat, most strongly influenced the cooling effectiveness. Liu *et al.* [3] investigated the flow and heat transfer characteristic of transpiration cooling with particle diameters d_p = 40 μm and 90 μm experimentally and numerically. In their study their parameters were solid matrix thermal conductivity, injection rate and particle diameter. They showed that the cooling effectiveness increased with increasing injection rate and the cooling performance for the porous wall with the smaller particle diameters was better. Arai and Suidzu [4] investigated experimentally effects of the porous ceramic coating material such as permeability of cooling gas, thermal conductivity and adhesion strength. They showed that porous ceramic coating has superior permeability for cooling gas and transpiration cooling system for gas turbine. Wang *et al.* [5] investigated the effect of different mainstream temperature, Reynolds numbers, and coolant injection ratio on transpiration cooling of the wedge shape nose. They obtained that the average temperature over the transpiration area falls with an increase in the coolant injection ratio, whereas the average cooling effectiveness rise. Langener *et al.* [6] investigated transpiration cooling applied to flat C/C material under subsonic main-flow conditions. They showed that thickness of porous material and main fluid total temperature did not affect the cooling efficiency. The coolant's specific capacity was the most effective parameter for the cooling efficiency. He *et al.* [7] investigated performance of evaporative cooling with cellulose and polyvinyl chloride (PVC) corrugated media experimentally. Their results showed that the cooling efficiency of the cellulose media vary from 43% to 90% while the cooling efficiency of the PVC media are 8% to 65% depending on the medium thickness and air velocity.

Some investigations on transpiration cooling and its applications were also performed numerically in the literature. Polezhaev [8] numerically investigated transpiration cooling for critical high temperature turbine blade. He showed that transpiration gas-cooled blade concept had a potential to achieve the goal of 60% thermal efficiency for power plant. Trevino and Medina [9] also obtained numerical results for transpiration cooling of a thin porous plate in a laminar convective flow. They presented the asymptotic solution for the thermally thin wall regime and they showed that numerical results were in a good agreement with the asymptotic solution close to the asymptotic limits. Andoh and Lips [10] presented an analytical approach method to determine the fluid-flow and the heat transfer characteristics in the porous material. They obtained that an increase of 90% of the value of volumetric heat transfer coefficient involves a decrease from 45% to 54% of the surface temperature. Liu *et al.* [11] investigated numerically the influence of the thermal conductivity and porosity. Their results showed that both the porosity and the thermal conductivity variations had a substantial effect on the temperature distribution. The reduction of the thermal conductivity and the decrease of the porosity reduced the influence of the outside boundary conditions to a thin layer. He and Wang [12] studied on a simulation about 1-D and steady-state transpiration cooling problem with phase change numerically. Their results showed that the thickness of the two-phase region does not vary with heat flux, when the plate is in vapor, two-phase and liquid phase, while the thickness is mainly influenced by coolant mass flow rate. Huang *et al.* [13] investigated cooling of a sintered porous struts using transpiration cooling in scramjets numerically. Their results showed that increasing the coolant blowing ratio improves the cooling effectiveness. The surface temperature decreased as the material thermal conductivity increased. Shi and Wang [14] studied on an analytic solution of a local thermal non-equilibrium model to optimize a porous structure which consists of two layered media. Their results indicated that the thermal conductivity and porosity of the second layer near the hot side is very important for the hot side temperature. Song *et al.* [15] investigated the cooling enhancement by applying evaporative cooling to an air-cooled finned heat exchanger numerically. Their results showed that when the surface is covered with thin water film, the temperature difference between the channel wall and the air becomes smaller due to the water evaporation from the fin surface and latent heat dissipation. Hsyan *et al.* [16] investigated the evaporation of liquid on steady, 2-D laminar convection flows in porous media numerically. Their results showed that the latent and sensible heat of mixed convection is greater than that of forced convection. Maity [17] investigated the flow and heat transfer of a thin liquid film over an unsteady porous sheet with suction or injection numerically. His results showed that the film thinning rate decreased with increase of the porosity of the porous medium. The temperature of the liquid film increased with increasing of suction velocity and decrease with increase of injection velocity when the sheet is heating.

These previous investigations can be grouped into two categories. The first group focused on analyses of the boundary-layer with turbulent or laminar flow. The second group focused on effectiveness of transpiration cooling at harsh, high pressure and temperature, condition.

However, there are few studies of controlling the temperature of hot gas stream and surface to enhance heat transfer by using transpiration cooling. The objective of this study is to investigate cooling effectiveness of a porous plate, and cooling efficiency of a system with air as a hot gas stream and water as a coolant, to figure out the influence of Reynolds number of hot gas stream and particle diameters of porous plate experimentally. Important results were obtained to understand effects of these parameters on the system cooling efficiency. The results of this study were also verified with results from the literature.

Materials and methods

The present study is mainly focused on improving heat transfer by cooling of air with transpiration cooling. In order to do this, an experimental investigation of effects of Reynolds number of hot gas stream and particle diameters of porous plate on local wall temperature and cooling effectiveness along the surface of a porous flat plate inside a rectangular channel with air as a hot gas stream and water as a coolant was performed. A porous media was chosen to form a thermal barrier with low thermal conductivity as well as an active cooling plate by evaporating water from the surface of porous media. A schematic of the experimental facility, which consists of an air blower, flow meter for air and water, a computerized data acquisition system, a test section, a power unit, an absolute and differential manometer and a calibrator thermometer, is shown in fig. 1.

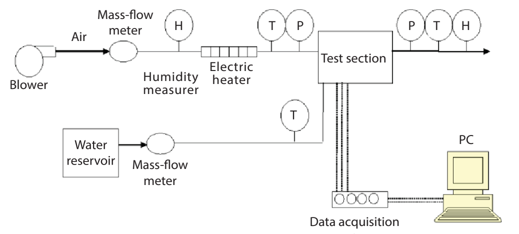

Figure 1. Schematic of experimental apparatus

The power unit includes variac and parallel connected air heaters. The test section is composed of a polycarbonate rectangular channel and a porous plate. Details of the test section are shown in fig. 2.

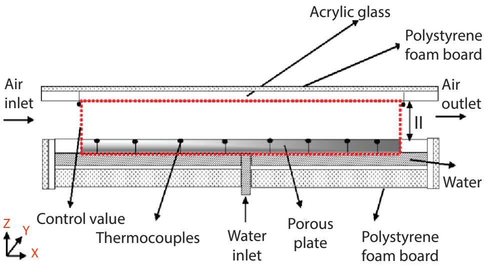

Figure 2. Details of the test section

Table 1. Thermal properties of porous plate

Thermal properties of porous plate	
Coefficient of thermal expansion [$\times 10^{-6}$ K^{-1}]	130-200
Specific heat [Jkg^{-1}K^{-1}]	1900
Thermal conductivity [Wm^{-1}K^{-1}]	0.42-0.51
Upper working temperature [°C]	55-95

The channel is made of polycarbonate sheet. Dimensions of test section were arranged as 220 × 880 × 10 mm (width, length, high). Hydraulic diameter of the channel is 19.1 mm. The porous plates (high performance polyethylene) were arranged and located over water channel in the test section. The porous plates with low thermal conductivity were chosen to protect the main surface from hot air as a thermal barrier. Thermal properties of porous plate were presented in tab. 1.

Temperatures at the centers of porous plate were measured using calibrated *T*-type thermocouples inserted through 2 mm holes inside the thickness of the porous plate. Calibrated thermocouples were located in the middle of porous plates in these arrangements (6 cm, 16 cm, 28 cm, 38 cm, 50 cm, 60 cm, 72 cm, and 82 cm). Conventional temperature measurements methods were used to evaluate cooling efficiency of the system. Two electric heaters were used to support enough power to the system to increase air temperature to the needed level. This electric heaters were connected parallel and they supplied 280-620 W power to the system according to power we need. A variac was used to arrange voltage to support needed power for the system. The power supplied was monitored by using multimeters for the control of current and voltage. Then flow meter of air and water were added to the system. A calibrated air-flow meter was used for measurement of mass flow rate of air. A micro flow mater (measurement range between 5 mL/min to 80 mL/min) and a micro pump (measurement range between 3 mL/min to 20 mL/min) were used for measurement of mass flow rate of coolant (water) for experimental study. For each particle diameter (40 µm, 50 µm, 100 µm, and 200 µm) porous plates were prepared again and tested for different parameters. Air inlet temperature is 77 °C. Polystyrene foam board was used to insulate the top side of the channel with a thickness of 50 mm (k = 0.032 W/mK). The ambient temperatures in the experiments varied between 20 °C to 24 °C. The system was considered to be steady-state when surface temperatures and the inlet and the outlet fluid temperatures of water and air were all within ±0.1 ºC. Air velocity was used to calculate Reynolds number given in eq. (1):

$$\mathrm{Re} = \frac{\rho V_{\mathrm{air}} D_h}{\mu} \tag{1}$$

where V_{air} is the inlet velocity of air, D_h – the hydraulic diameter of the channel, μ – the dynamic viscosity, and ρ – the density of air.

Data reduction

In this application there is a porous plate in a rectangular channel, this porous plate is wetted by water of which reservoir temperature is T_{water} = 22 ºC. Air enters in the channel at different velocities. Dry air (relative humidity of air, ϕ = 0) was used as a hot gas stream. In this application, all Reynolds numbers were higher than 3000 so it can be assumed that it is a turbulent flow. To understand the effect of water-flow rate, as a coolant on heat transfer, optimum water-flow rate, by assuming all water was evaporated by surface heat flux, was calculated analytically at the condition of test section. So energy balance equation on the surface of porous plate:

$$\dot{m}_{\text{air}}(h_{\text{air,out}} - h_{\text{air,in}}) = \dot{m}_{\text{water}}(h_{\text{water,in}}) - \dot{m}_{\text{water,liq}}(h_{\text{water,liq,out}}) - \dot{m}_{\text{water,vap}}(h_{fg}) - q_{\text{lost}} \tag{2}$$

where $\dot{m}_{\text{air}}$ is the mass-flow rate of air, $h_{\text{air,out}}$ – the outlet enthalpy of air, $h_{\text{air,in}}$ – the inlet enthalpy of air, $\dot{m}_{\text{water}}$ – the total amount of mass-flow rate of water, $h_{\text{water,in}}$ – the inlet enthalpy of water, $\dot{m}_{\text{water,liq}}$ – the mass-flow rate of liquid water (liquid water without phase change), $h_{\text{water,liq,out}}$ – the outlet enthalpy of liquid water, $\dot{m}_{\text{water,vap}}$ – the mass-flow rate of evaporated water (evaporated water with phase change), h_{fg} – the latent heat of water (can be called as evaporative heat flux), and q_{lost} – the heat lost to the environment from insulation. The q_{lost} is less than 1% of total energy input so it can be assumed as negligible.

To express protection level of surface from hot gas, cooling effectiveness of the porous plate is identified. The cooling performance for various test sections, expressed by Liu *et al.* [2], was compared based on the cooling effectiveness of porous plate. So cooling effectiveness of porous plate, Huang *et al.* [18]:

$$\eta = \frac{T_{\text{surface}} - T_{\text{air,in}}}{T_{\text{water}} - T_{\text{air,in}}} \tag{3}$$

where T_{surface} is the local surface temperature of porous plate on the hot gas side, $T_{\text{air,in}}$ – the local average main stream temperature which was assumed to be the mainstream inlet temperature of air, and T_{water} – inlet temperature of water.

To express cooling performance of the system for cooling of hot air, cooling efficiency of the system is identified. Cooling efficiency of the system is generally used to determine the performance of evaporative cooling systems. It represents how close the exiting air gets to the state of saturation. The definition of cooling efficiency of the system was given by He *et al.* [7]:

$$\eta_{\text{sys}} = \frac{T_{\text{air,in}} - T_{\text{air,out}}}{T_{\text{air,in}} - T_{\text{avg,surface}}} \tag{4}$$

where $T_{\text{avg,surface}}$ is the average surface temperature of porous plate, $T_{\text{air,in}}$ inlet temperature of air and $T_{\text{air,out}}$ is outlet temperature of air. Moffat [19, 20] gave a description of errors' sources by using uncertainty in the planning of experiments. Besides that, Root-Sum-Square (RSS) method was used to predict the uncertainty by Caggese *et al.* [21] and Fechter *et al.* [22]. Table 2 shows errors and uncertainty levels of measured parameters in the calculation of effectiveness of porous plate, efficiency of the system and the Reynolds number.

Table 2. Experimental uncertainties

Parameter	Units	Value	Errors	[%]
V_{air}	[ms^{-1}]	3.8	± 0.114	± 3
$(T_{\text{air,in}} - T_{\text{surface}})$	[°C]	$T_{\text{air,in}}$ = 77 °C T_{surfce} = 54 °C	± 0.1	± 0.4
Re	[–]	Re = 3580	113.34	3.16
η	[–]	η = 0.791	0.0268	3.4
η_{sys}	[–]	η_{sys} = 0.4874	0.0313	6.4

In the experiments, the velocity measurements were accurate within ±3%, and the temperature measurements were accurate within ±0.1 °C. The uncertainty of Reynolds number, effectiveness of porous plate, and efficiency of the system for the studied parameters is within ±3.16%, ±3.4%, and ±6.4%, respectively.

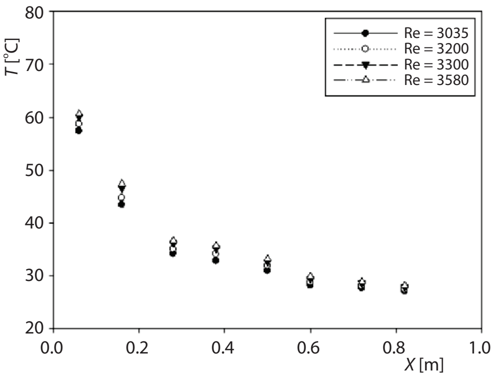

Figure 3. Surface temperature variations for different Reynolds numbers

Results and discussions

Effect of Reynolds number of hot gas stream (air)

Experiments were conducted for different Reynolds number of the air channel flow for $T_{\mathrm{air,in}} = 77$ ºC, particle diameter $d_p = 200$ μm and $\dot{m}_{\mathrm{water}} = 0.000083$ kg/s. Velocity profile of hot gas stream (air) were chosen in the turbulent flow regime with low Reynolds number, because it is planned to see the effect of porous plate and coolant on heat transfer on the hot side of porous plate precisely. For different Reynolds numbers, measured surface temperatures were shown in fig. 3.

Effect of Reynolds number of the air channel flow can be seen clearly at the inlet region of porous plate. But by cooling of air along the porous plate causes a gradual decrease on difference of surface temperature up to the end of porous plate. Temperature difference is minimum at the outlet region of porous plate. So there is not a significant difference on surface temperature and cooling effectiveness at the end of porous plate for different Reynolds numbers used. The reason of this is the choice of Reynolds numbers in a narrow gap, to see effect of Reynolds number near to the optimum value. Surface temperature values changed between 60 ºC and 27 ºC. Surface temperature is decreasing sharply up to $x = 0.28$ m but after that point there is not a prominent difference on surface temperature. Cooling efficiencies of porous plate for different Reynolds numbers were shown in fig. 4.

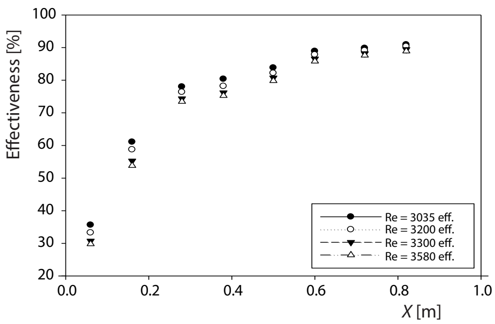

Figure 4. Cooling effectiveness of porous plate for different Reynolds numbers

Cooling effectiveness of porous plate is changing between 30% and 90% along the porous plate. Cooling effectiveness increases up to $x = 0.28$ m, after this point it changes only slightly. Increase of 15.2% on Reynolds number of the air channel flow causes a decrease of effectiveness of the porous plate of 8.6%. So, we could observe that increasing Reynolds number of the air channel flow causes an increase on surface temperature, and a decrease on cooling effectiveness of porous plate. Cooling efficiencies of the system for different Reynolds numbers were shown in fig. 5. Increasing Reynolds number causes a slight decrease on efficiency of

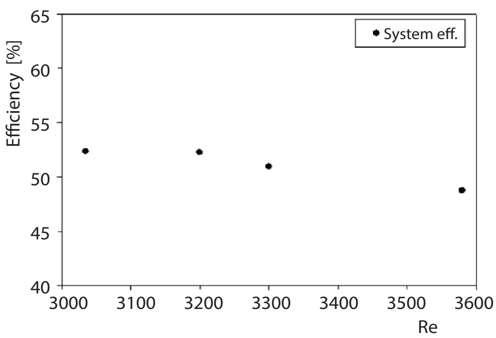

Figure 5. Cooling efficiency of system for different Reynolds number

the system. Increase of 15.2% on Reynolds number causes a decrease of efficiency of the system of 6.9%. Decreasing trend of the efficiency of the system also increases gradually between Re = 3035 and 3580.

The reason of this decrease on the efficiency of the system is that increasing air velocity (while increasing Reynolds number) causes a decrease on heat convection between hot air and water on the surface of porous plate. So, outlet temperature of air increases more than surface temperature. By that way cooling performance of the system to cool hot air also decreases.

Effect of particle diameter

In this study effect of particle diameters on heat transfer from the surface is especially investigated. It is known that heat transfer occurs from the cavities of particles of surface. It is thought that if particle diameter of porous plate is decreased, surface area can be extended, so an increase on heat transfer can be obtained. Extanding surface by decreasing particle diameter was shown in fig. 6.

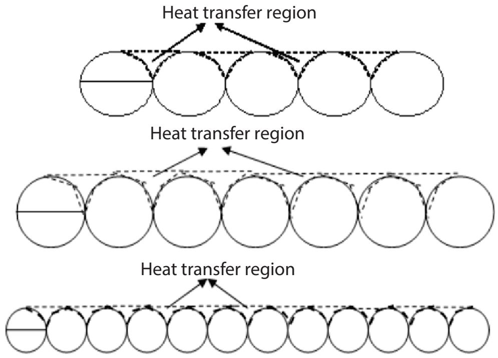

Figure 6. Extanding surface by decreasing particle diameter

But there should be a limit for decreasing particle diameter of surface. So experiments were conducted for different particle diameters as d_p = 40 μm, 50 μm, 100 μm, and 200 μm for Re = 3300, $T_{\text{air,in}}$ = 77 ºC and water flow rate $\dot{m}_{\text{water}}$ = 0.000083 kg/s. Surface temperature was measured for different particle diameters. Measured temperature values were shown in fig. 7. It can be seen that decreasing the particle diameter causes a significant decrease on surface temperature. This was interpreted with the fact that, decreasing particle diameter causes an increase on surface area for evaporation and heat transfer. Effect of particle diameter on surface temperature can be detected easily at the inlet region of porous plate, but this difference decreases slightly to the end of the porous plate.

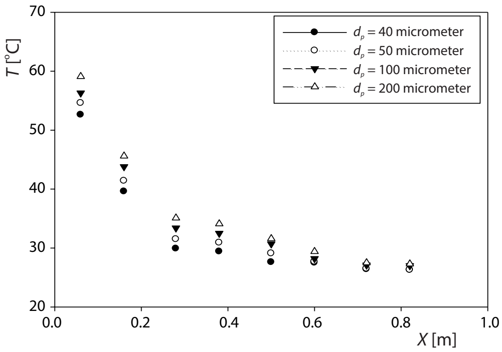

Figure 7. Surface temperatures for different particle diameters

One can see that decreasing particle diameter causes an increase on cooling effectiveness of porous plate. Beyond the point of x = 0.6 m there is not a significant change among different particle diameters. Difference of cooling effectiveness of porous plate from d_p = 40 μm to d_p = 200 μm is 12%. But this difference decreases to 2% from inlet region end of the porous plate. Cooling effectiveness of porous plate for different particle diameters was shown in fig. 8.

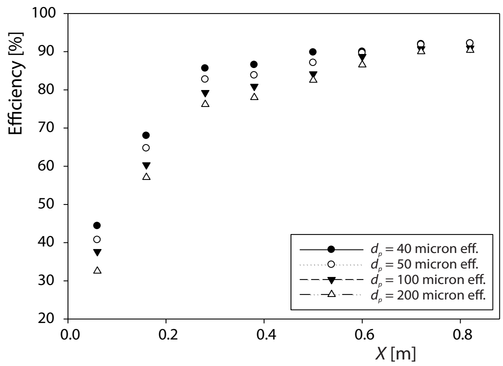

Figure 8. Cooling effectiveness of porous plate for different particle diameter

Verification of experimental results

Experimental results of this study were compared with the study of Jiang *et al.* [1]. Verification of experimental results is shown in fig. 9. Both studies are about transpiration cooling, but main deference between these two studies is that Jiang *et al.* [1] applied air as the hot gas stream and coolant. However, in present study the hot gas stream is air and the coolant is liquid water. The comparison was done by using dimensionless parameters x/L, ratio of distance of measured point to the total length of porous plate, and h_x/h_{avg} ratio of local convection heat transfer coefficient to average heat convection coefficient. It can be seen that flow characteristics are very similar, in spite of different boundary conditions. A significant temperature decrease and a prominent heat transfer increase can be obtained at the inlet region of porous plate. Since in the present study a phase change occurs, heat transfer increase is bigger than in the study by Jiang *et al.* [1]. For both results, temperature decrease occurs suddenly at the inlet region of porous plate, but after that point there is not an important increase on heat transfer, because of the decrease of film temperature on the surface of the porous plate. At both studies increasing blowing ratio has not a significant effect on heat transfer, especially at the end of the porous plate.

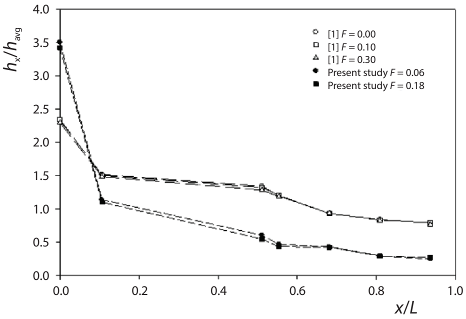

Figure 9. Verification of experimental results

Conclusions

The present study is focused on developing structured solid surface geometry to improve heat transfer by cooling of air with transpiration cooling. Effects of particle diameters and Reynolds number of the air channel flow on local wall temperature and cooling effectiveness along the porous plate inside a rectangular channel with air as a hot gas stream and water as a coolant were investigated experimentally. Different from the literature transpiration cooling is used as an air cooling mechanism at this study. The following conclusions can be obtained from the experimental results;

Increasing Reynolds number of the air channel flow causes an increase on surface temperature and a decrease on cooling effectiveness of porous plate. Increasing Reynolds number of 15.2% causes a decrease of 6.9% on cooling efficiency of the system.

Decreasing particle diameter causes a significant decrease on surface temperature and an increase on cooling effectiveness of porous plate. The reason of this, decreasing particle diameter causes an extension on surface area for evaporation and heat transfer.

Verification of this study was done by comparing results of this study with findings by Jiang *et al.* [1]. Similar heat transfer and fluid-flow behavior was observed. Effect of boundary conditions and phase change can also be seen by this comparison. Phase change effect can be seen especially at the inlet region of porous plate.

Investigating effects of nanofluids, Kilic and Ali [23], as a coolant of transpiration cooling on heat transfer for different geometries may be a research area for future studies.

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Nomenclature

D_h – hydraulic diameter of the channel, [mm]
F – blowing ratio
$h_{\mathrm{air,out}}$ – outlet enthalpy of air, [kJkg^{-1}]
$h_{\mathrm{air,in}}$ – inlet enthalpy of air, [kJkg^{-1}]
h_{surface} – enthalpy of water at surface, [kJkg^{-1}]
$h_{\mathrm{water,in}}$ – inlet enthalpy of water, [kJkg^{-1}]
$h_{\mathrm{water,out}}$ – outlet enthalpy of water, [kJkg^{-1}]
h_{fg} – latent heat of water, [kJkg^{-1}]
k – conduction coefficient, [Wm^{-1}K^{-1}],
Re – Reynolds number, (= $V_{\mathrm{air}}D_h\nu^{-1}$)
V_{air} – air velocity, [ms-1]
V_c – coolant velocity, [ms^{-1}]

Greek Symbols

η – cooling effectiveness of porous plate
η_{sys} – cooling efficiency of the system
μ – dynamic viscosity, [kgm^{-1}s^{-1}]

Subscript

exp. – experimental
in – inlet

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