e-ISSN: 2458-8156

Journal Homepage: http://dergipark.gov.tr/ejens

EJENS

European Journal of Engineering and Natural Sciences

DergiPark AKADEMİK

CODEN: EJENS

Enhancing Heat Transfer from a Porous Plate with Transpiration Cooling

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Publication Info

Paper received:
29 May 2016

Revised received:
15 October 2016

Accepted:
01 March 2017

Abstract

The present study is focused on developing structural solid surface geometry to improve heat transfer by cooling of air with transpiration cooling. Effects of flow rate of water ($\dot{m}_{water}$= 0.000083, 0.000116, 0.000166, 0.000249 kg/s) and particle diameter of porous plate (Dp= 40, 50, 100, 200 μm) on local wall temperature and cooling efficiency of porous plate and the system inside a rectangular channel with air as a hot gas stream and water as a coolant were investigated experimentally. High performance polyethylene as a porous media was used not only to form a thermal barrier but also an active cooling plate by evaporating water from the surface of porous media to cool air. Temperatures were measured by T-type thermocouples. Two electric heaters were used to support enough power to the system. It was observed that increasing water flow rate did not cause a prominent decrease on surface temperature and cooling efficiency of porous plate. The higher injection rates result in further increase of the cooling effectiveness. Cooling efficiency of porous plate changed from 38 to 90 %. Increasing water flow rate as a coolant causes a prominent increase on cooling efficiency of the system. Increasing water flow rate three times causes an increase of 26.4 % on cooling efficiency of the system. Decreasing particle diameter causes a significant decrease on surface temperature. Difference of cooling efficiency of porous plate from Dp=40 to Dp=200 μm decreases from 12% to 2 % from inlet region to end of porous plate.

Key words

Heat Transfer, porous plate, structured surface, transpiration cooling

1. INTRODUCTION

Transpiration cooling is considered as an attractive cooling technique. This method has been used to protect solid surface exposed to high-heat-flux, high-temperature environments such as hypersonic vehicle combustors, liquid rocket thrusters, gas turbine blades, water oxidation technology, the first wall and blanket region in fusion reactors, the nose of aerospace vehicles during the atmospheric re-entry phase of their flight. Transpiration cooling processes involve simultaneously two different heat transfer mechanisms: conduction through a solid plus convection. In transpiration cooling process: fluid coolant is injected into porous matrix in the direction opposite to heat flux, at the same time absorbs the heat conducted into the solid matrix and transports heat flux through the convection passing pores, finally the coolant forms a thin film on the hot side surface to reduce the heat flux coming into the porous matrix and to cool the hot gas stream. The porous matrix in transpiration cooling can be some different types of porous media. Sintered stainless steel walls, metal fibers ceramics and porous plastics can be used as a porous media. The structure generally has a large number of pores with diameter ranging between 5 and 200 μm and can be made of different shaped particles as spherical, porous plate thicknesses can be change between 1 and 10 mm or more.

Interest and research in this topic may even have accelerated in recent years because of its high potential of heat transfer. Many investigators have performed experimental and numerical studies to determine the heat transfer characteristics for transpiration cooling. Jiang et al. [1] investigated turbulent flow and heat transfer in a rectangular channel without and with transpiration cooling experimentally and numerically. They used two-layer k-ε model to calculate the turbulent velocity and thermal characteristics of the main flow. They used sintered bronze particles as porous wall and air as coolant. The nominal particle diameter of the sintered porous wall was 0.1 mm and the porosity was 0.45. Their results showed that the transpiration cooling greatly increases the boundary layer thickness and reduces the wall skin friction and increasing coolant blowing ratio sharply reduced both 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 for various cooling gasses. In their study: the effects of injection rates, model geometry, inlet temperature and Reynolds number of the main stream were studied for air, nitrogen, argon, carbon dioxide and helium. Two-dimensional numerical simulation using the RNG k-ε turbulence model for the main stream flow and the Darcy-Brinkman- Forchheimer momentum equation and thermal equilibrium model were used to compare the general features in the experiments. Their results showed that the injection rate strongly influenced the cooling effectiveness. The increase of the main stream inlet Reynolds number dramatically reduced the cooling effectiveness. And the coolant thermos 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 through sintered porous flat plates with particle diameters dp= 40 and 90 μm experimentally and numerically with dry air as the coolant stream. 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, the temperature distribution on the porous bronze plate was more uniform than that on the sintered stainless steel plates 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. The mixture of 8 wt.% yttria-stabilized zirconium and polyester powders was employed as the coating material, in order to deposit the porous ceramic coating onto Ni-based super alloy substrate in their study. They showed that porous ceramic coating has superior permeability for cooling gas and transpiration cooling system for gas turbine could be achieved by using porous ceramic coating. He et al. [5] investigated new conversation equation for mass, momentum and energy to describe the performances of fluid flow, heat absorption and phase change in porous matrix. Their model's main differences from previous models are firstly, considering the compressibility of vapor in the momentum and energy equations, secondly, adding a term of the momentum transfer caused by liquid phase change into the momentum equation of vapor and liquid phases in two-phase region, finally in the energy equation of two-phase region, taking the variations of temperature and pressure into account. Their results showed that with an increase in heat flux and decrease in coolant mass flow rate the temperature difference over the two-phase region falls and a higher external heat flux or lower coolant mass flow rate will accelerate the phase change process and increase he area of two-phase region and vapor region. Wang et al [6] investigated the effect of different mainstream temperature, Reynolds numbers, and coolant injection ratio on transpiration cooling of the wedge shape nose cone with an equal thickness porous wall using liquid water as coolant. 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 and There is an optimal injection rate, at which the coolant passing through pores with liquid state when the driving force for the coolant injection is the minimum and the cooling effectiveness is high. Tsai et al. [7] investigated the transient cooling process in a sudden-expansion channel with the injection of cold air from the porous bottom wall experimentally. They identified distinct flow features under various rates of coolant injection. They categorized flow features to four cooling patterns; the recirculation pattern, the elevated recirculation pattern, the transpiration pattern and film pattern. Langener et al. [8] investigated transpiration cooling applied to flat C/C material under subsonic main-flow conditions. In their study; main-stream Mach number ranged from Mg= 0.3-0.7 and totals temperature was 523 K. Air, argon and helium were used as coolants. They showed that thickness of sample and main stream total temperature did not affect the cooling efficiency. The coolant used and its specific capacity was the most influential parameter for the cooling efficiency. And with higher Mach number the cooling efficiency decreased. Tsai and Lee [9, 10] investigated the correlation between superheat levels and heat fluxes when used sintered powder structures as wicks. Their parameters were 45 μm, 75 μm, 150 μm of powder sizes and powder shapes of spherical, dendritic. Their results showed that smaller powders structures achieved higher effective thermal conductivities for both powder shapes. Spherical powder structures achieved twice the effective thermal conductivity of dendritic powder ones for each powder size. And at the same superheat level structures of smaller powder size and dendritic powder shape achieved higher heat fluxes. He et al [11] investigated performance of evaporative cooling with cellulose and Polyvinyl Chloride (PVC) corrugated media experimentally. The heat and mass transfer and pressure drop across the two media with three thicknesses (100, 200, 300 mm) were studied. Their results showed that the pressure drop range of the cellulose media is 1.5-101.7 Pa while the pressure drops of the PVC media are much lower with the range of 0.9-49.2 Pa, depending on the medium thickness, air velocity and water flow rate. 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. Most of these previous investigations can be divided into two categories. The first group focused on analyses of the characteristics of the boundary layer, turbulent or laminar flow in transpiration conditions. The second group focused on transpiration cooling effectiveness at high pressure and temperature to simulate practical operational conditions. However, there are few studies of developing structural solid surface geometry and 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 the local wall temperature and cooling effectiveness distribution along the surface of a porous flat plate with air as a hot gas stream and water as a coolant to figure out the influence of mass flow rate of water, and particle diameters of porous plate.

2. MATERIALS AND METHODS

In This study; an experimental investigation of effects of mass flow rate of water, and particle diameters of porous plate on local wall temperature and cooling efficiency 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 done. For this purpose a porous plate was used not only to form a thermal barrier but also an active cooling system by evaporating water from the surface of porous media.

2.1 Experimental Apparatus

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. 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. The channel is made of polycarbonate sheet and its geometry is shown in Fig. 2. Dimensions of test section were arranged as 220x880x10 mm (width, length, high). Hydraulic diameter of the channel is 19.1 mm. Then porous plates (high performance polyethylene) were arranged and located over water channel in test section precisely. 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 this arrangements (6,16,28,38,50,60,72,82 cm).

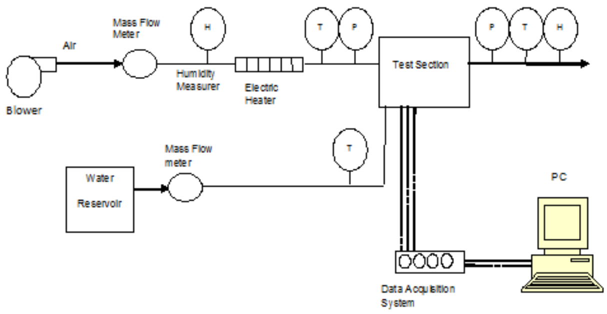

Figure. 1. Schematic of experimental apparatus

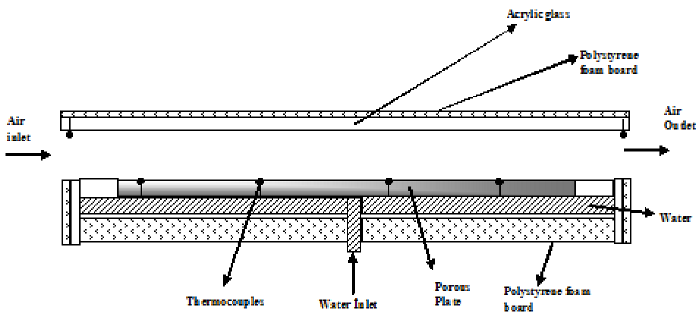

Figure. 2. Details of the test section

Conventional temperature measurements of the porous plate surface 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 heater was connected parallel and they supplied 280-620 W power to the system according to needed power. A variac was used to arrange voltage to support needed power for the system. The total power supplied was monitored using digital multimeters for the control of voltage and current. Then flow meter of air and water were added to the system. For each particle diameter (40, 50, 100, and 200 micrometer) porous plates were prepared again and tested for different parameter. Air temperatures ranged between 47°C and 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/(m.K)). The ambient temperatures in the experiments varied between 20°C to 24°C. The system was assumed to be steady state when variations of the surface temperatures and the inlet and outlet fluid temperatures of water and air were all within ±0.1 ºC.

2.2 Data Reduction

The focus of the present investigation is to develop structural solid surface geometry to improve heat transfer by cooling of air with transpiration cooling. Inlet and outlet temperature of hot stream gas (air) were chosen according to the inlet and outlet temperature of cooling systems. In this application there is a porous plate in a rectangular channel, this porous plate is wetted by water of which reservoir temperature is Twater= 22 ºC. Air enters into the channel with some different temperatures and velocities. For this application dry air (relative humidity of air, x=0) was used as a hot gas stream.

In this application; all Reynolds numbers were bigger that 3000 so it can be assume that it is a turbulent flow. Mean average film temperature in the channel is calculated as;

$$T_{avg} = \frac{T_{airin} + T_{surface}}{2} \tag{1}$$

where Tairin is the inlet temperature of air, Tsurface is the average surface temperature of porous plate. So It can be calculated, as explained by Mills [12], some physical properties of air by using thermodynamics tables as; ρ is the density of air, cp specific heat of air, γ is the kinematic viscosity, Pr is the Prandtl number and Sc Schmidt number. So Stanton number with zero porosity can be calculated as;

$$St^* = 0.0296(\mathrm{Re})^{-0.2}(\mathrm{Pr})^{-2/3} \tag{2}$$

And convective heat transfer coefficient with zero-mas-transfer will be;

$$hc^* = \rho.V_{air}.Cp.St^* \tag{3}$$

where ρ is the density of air, Vair is air velocity, cp specific heat of air and St* is stanton number.

So mass transfer driving force is;

$$B_m = \frac{m_{ev} - m_s}{m_s - 1} \tag{4}$$

where mev is mass fraction of water vapor at channel flow and ms is mass fraction of water vapor at surface.

And mass transfer St number can be calculated as;

$$St^*_{mass} = 0.0296(\mathrm{Re})^{-0.2}(Sc)^{-2/3} \tag{5}$$

So the zero mass transfer limit conductance will be;

$$g^*_{mass} = \rho.V_{air}.St^*_{mass} \tag{6}$$

where ρ is the density of air, Vair is air velocity and St*mass is the mass transfer stanton number. And the mass transfer conductance is;

$$g_{mass} = g^*_{mass}.\frac{\ln(1+B_m)}{B_m} \tag{7}$$

where g^*_{mass} is the the zero mass transfer limit conductance, Bm is the mass transfer driving force. So mass flow rate of evaporated water is;

$$\dot{m}_{ev} = g_{mass}.B_m \tag{8}$$

Then heat transfer blowing parameter is;

$$B_h = \frac{\dot{m}_{ev}.Cp}{hc^*} \tag{9}$$

where $\dot{m}_{ev}$ is the mass flow rate of evaporated water, cp specific heat of air and hc* is convective heat transfer coefficient with zero-mas-transfer. And convective heat transfer coefficient for surface of porous plate is;

$$h_c = h_c^{\ *}.\frac{B_h}{\exp(B_h)-1} \tag{10}$$

So energy balance equation on the surface of porous plate is;

$$\dot{m}_{air}.(h_{airout} - h_{airin}) = \dot{m}_{water}.(h_{waterin} - h_{fg} - h_{waterout}) \tag{11}$$

where $\dot{m}_{air}$ is the mass flow rate of air, h_{airout} is the outlet enthalpy of air, h_{airin} is the inlet enthalpy of air, $\dot{m}_{water}$ is the mass flow rate of water, $h_{waterin}$ is the inlet enthalpy of water, $h_{waterout}$ is the outlet enthalpy of water, and h_{fg} is the latent heat of water.

And convection heat transfer can be calculated as;

$$\dot{q}_{conv} = \dot{m}_{water}.(h_{fg} + h_{waterout} - h_{waterin}) = h_c.A.(T_{air} - T_{surafce}) \tag{12}$$

where $\dot{m}_{water}$ is the mass flow rate of water, $h_{waterin}$ is the inlet enthalpy of water, $h_{waterout}$ is the outlet enthalpy of water, h_{fg} is the latent heat of water, hc is the convective heat transfer coefficient for surface of porous plate, A is surface area of porous plate, Tair is the temperature of air and Tsurface is the avarage surface temperature of porous plate.

Conductıon heat transfer from porous plate to water will be;

$$\dot{q}_{cond} = \dot{m}_{water}.(h_{surfece} - h_{waterin}) \tag{13}$$

where $\dot{m}_{water}$ is the mass flow rate of water, $h_{surface}$ is the enthalpy of water at surface temperature, $h_{waterin}$ is the enthalpy of water at inlet temperature.

So cooling efficiency of the porous plate, shows protecting degree of main surface by decreasing surface temperature with using porous plate, is;

$$\eta = \frac{T_{surface} - T_{airin}}{T_{water} - T_{airin}} \tag{14}$$

And cooling efficiency of the system, shows cooling degree of air with using this system, will be;

$$\eta_{sys} = \frac{T_{airin} - T_{airout}}{T_{airin} - T_{surface}} \tag{15}$$

Moffat [13, 14] presented a general description of sources of errors in engineering measurements and discussion of the use of uncertainty in the planning of experiments. In addition to that, the uncertainty associated with the experimental data is estimated using Root-Sum-Square (RSS) method, to combine the individual uncertainty terms of the independent variables in the calculation of the heat transfer, by Caggese et al. [15] and Fechter et al. [16]. Table 3 shows uncertainty levels of measured parameters and their errors in the calculation of efficiency of porous plate, efficiency of the system and the Reynolds number. In the present experiments, the temperature measurements were accurate within ± 0.1 °C, and the velocity measurements were accurate within ± 2%. The uncertainty of Re, efficiency of porous plate, and efficiency of the system for the ranges of parameters studied under steady-state conditions is within ± 0.7% , ± 3.4% and ± 6.4%, respectively.

Table 3. Experimental uncertainties

Parameter	Units	Value	Errors	%
Vair	m/s	3.35	± 0.2	± 5.9
Tsurface	° C	Tsurface= 33 °C	± 0.1	± 0.3
Re	-	Re =3580	26.57	0.7
η	-	η = 79.1	2,68	3.4
η_{sys}	-	η_{sys} = 48.74	3.13	6.4

3 RESULTS AND DISCUSSION

In this section, experimental results were presented for different water flow rate as a coolant, and particle diameters of porous plate.

3.1 Effect of water flow rate

Experiments were conducted for different water flow rate for Re=3300 and T_{inlet}= 77 ºC. Water flow rates ,applied in experimental, are between 0,000083 and 0.000249 kg/s. Surface temperature was measured for different water flow rates. Measured temperature was shown in Fig.3.

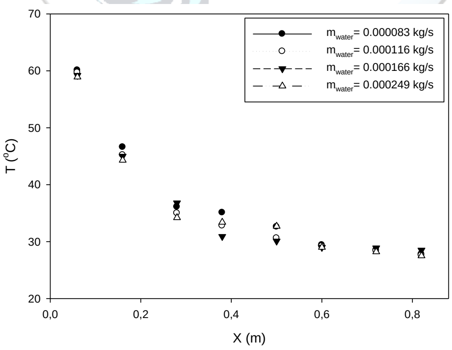

Figure. 3. Surface temperatures for different water flow rate

It was observed that increasing water flow rate did not cause a prominent decrease on surface temperature and cooling efficiency for the water flow rate of experiments. Because flow rate of water is slightly changed to determine the optimum value of water flow rate. The higher injection rates result in further increase of the cooling effectiveness. The difference of cooling efficiency decreased up to the end of the porous plate because air temperature also decreases at that region. Cooling efficiency of porous plate for different water flow rate was shown in Fig.4. It is observed that surface temperature changed from 27 to 56 ºC and cooling efficiency changed from 90 to 38%. If we continue to increase flow rate of water, droplets would occurred on the surface of porous plate and covered the surface. Then amount of evaporated water from the surface decreased. So effect of porous plate on heat transfer, extending surface area, also decreased. Cooling efficiency of the system for different water flow rate was shown in Fig.5.

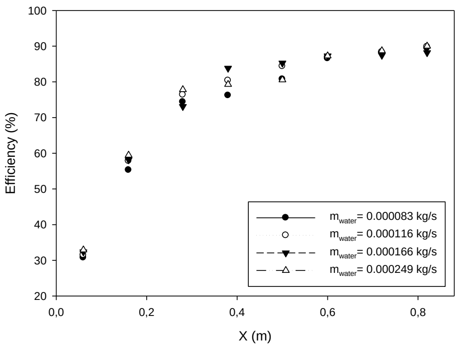

Figure. 4. Cooling efficiency of porous plate for different water flow rate

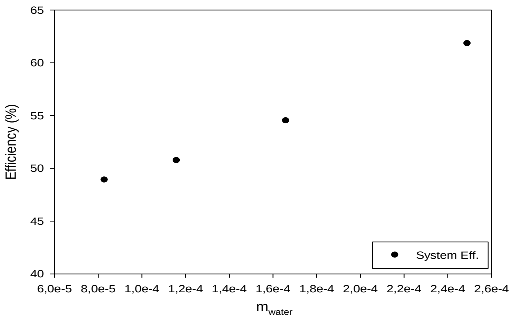

Figure. 5. Cooling efficiency of the system for different water flow rate

Increasing water flow rate as a coolant causes a prominent increase on cooling efficiency of the system. Increasing water flow rate three times causes an increase of 26.4 % on cooling efficiency of the system. Because increasing flow rate of water not only causes an increase on the quantity of evaporated water and amount of heat taken from the air but also decrease of temperature of air and surface.

3.2 Effect of Particle Diameter

In this study effect of particle diameters on heat transfer from the surface is 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 an increase on heat transfer can be obtained. But there should be a limit for decreasing particle diameter of surface. So experiments were conducted for different particle diameter D_p=40, 50, 100 and 200 µm for Re=3300, T_{inlet}= 77 ºC and water flow rate $\dot{m}_{water}$= 0,000083 kg/s. Surface temperature was measured for different particle diameter. Measured temperature values were shown in Fig.6.

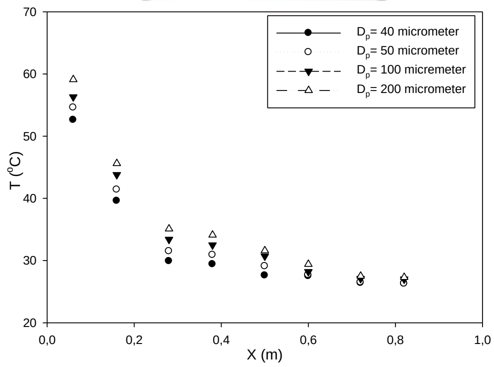

Fig. 6. Surface temperatures for different particle diameter

It can be seen that decreasing particle diameter causes a significant decrease on surface temperature. The reason of this, 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 decrease slightly to the end of porous plate. Cooling efficiency of porous plate for different particle diameter was shown in Fig.7.

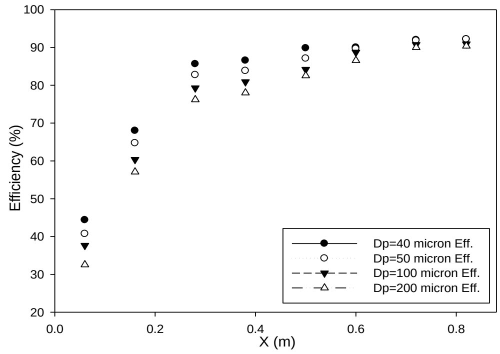

Fig. 7. Cooling efficiency of porous plate for different particle diameter

Beyond the point of x=0,6 m there is not a significant change among different particle diameter. Difference of cooling efficiency from D_p=40 to D_p=200 μm decreases from 12% to 2 % from inlet region to end of porous plate.

4 CONCLUSIONS

The present study is focused on developing structural solid surface geometry to improve heat transfer by cooling of air with transpiration cooling. Effects of particle diameters and mass flow rate of water 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 were investigated experimentally. Surface temperature of porous plate were measured and cooling efficiency of system were calculated for different flow rate of water ($\dot{m}_{water}$= 0.000083, 0.000116, 0.000166, 0.000249 kg/s) and particle diameter of porous plate (D_p= 40, 50, 100, 200 μm).

The following conclusions can be drawn from the experimental results;

Increasing water flow rate did not cause a prominent decrease on surface temperature and cooling efficiency of porous plate for the water flow rate of experiments. Because flow rate of water is slightly changed to determine the optimum value of water flow rate. The higher injection rates result in further increase of the cooling effectiveness. Surface temperature changed from 27 to 56 ºC and cooling efficiency changed from 90to 38 %. Increasing water flow rate as a coolant causes a prominent increase on cooling efficiency of the system. Increasing water flow rate three times causes an increase of 26.4 % on cooling efficiency of the system.

Decreasing particle diameter causes a significant decrease on surface temperature. The reason of this, decreasing particle diameter causes an extension on surface area for evaporation and heat transfer. Difference of cooling efficiency of porous plate from D_p=40 to D_p=200 μm decreases from 12% to 2 % from inlet region to end of porous plate.

Acknowledgment

The financial support of this study by the Scientific Research Council of Turkey (TUBITAK), with the program of postdoctoral scholarship (2219) and University of California Los Angeles Post Doctorate Program (UCLA/USA), is gratefully acknowledged.

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