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Technical Notes

TECHNICAL NOTES are short manuscripts describing new developments or important results of a preliminary nature. These Notes cannot exceed six manuscript pages and three figures; a page of text may be substituted for a figure and vice versa. After informal review by the editors, they may be published within a few months of the date of receipt. Style requirements are the same as for regular contributions (see inside back cover).

Mist Transpiration Cooling System Using Open-Cellular Porous Materials

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Nomenclature

D_s	=	equivalent strut diameter, m
G_w	=	water droplet loading density, $m^3/s/m^2$
q_R	=	irradiation on the upper surface of a heat shield, W/m^2
Re	=	Reynolds number defined by uD_s/ν_f
T_{in}	=	air temperature measured at the inlet of the acrylic test section, K
T_m	=	mean temperature of the porous heat shield, K
T_R	=	equivalent blackbody temperature, K
u	=	mean air velocity in the acrylic test section, m/s
w	=	dimensionless width of a strut with a square cross section
η_T	=	temperature efficiency
ν_f	=	kinematic viscosity of air, m^2/s
π	=	ratio of the circumference of a circle to its diameter
σ	=	Stefan–Boltzmann's constant
τ_o^*	=	optical thickness
ϕ	=	porosity

Introduction

A TRANSPIRATION cooling system[1–6] is well recognized to be a possible means for cooling rocket nozzles and reentry vehicles and protecting structures from high-intensity radiation. For such a system, the convective heating is reduced by fluid injected into the boundary layer, and the radiative heating is partially reflected and partially absorbed by the heat shield, from which a part of the heat is transferred to the injected fluid by convection. Generally, air has been utilized as an injected fluid, but to improve the ability of the heat-absorbing capability of the coolant use of other coolants[7] such as ammonia, water vapor, and water has been proposed. Here, it should be noted that ammonia is a chemically reactive gas, and dissociation occurs in the temperature region less than 1000 K; water vapor is a radiative gas; and water can readily change its phase.

In the present Note, we propose a transpiration cooling system utilizing mist—a mixture of air and water droplets—as an injectant and an open-cellular porous plate[8] as a heat shield and examine its effectiveness as a thermal protection against irradiation by varying several system parameters including water-droplet loading density and Reynolds number in a radiative environment.

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Experiment

A schematic diagram of the present experimental apparatus is shown in Fig. 1. Air was used as a carrier gas for water droplets and was sent out from a blower whose flow rate was measured using a rotermeter. Airflow rate was varied from 120 to 900 Nl/min. The air was blown upward through a circular acrylic pipe of 0.108 m inner diameter and 0.6 m in height. The air temperature measured at the inlet of the acrylic test section using a type-K sheathed thermocouple of 0.0016 m diam ranged from 290 to 301 K. A porous plate was placed on the top of the acrylic pipe as a heat shield. Two kinds of 0.012-m-thick cordierite-alumina ($Al_2\ O_3$/54 wt%, SiO_2/37 wt%, MgO/6 wt%, and the others/3 wt%) open-cellular porous material with different porosity ϕ and pores per inch (PPI) were used as a heat shield: CA#06 ($\phi = 0.89$ and PPI = 6) and CA#13 ($\phi = 0.87$ and PPI = 13). The porous plates used for the experiment were 0.12 m in diameter. The hemispherical reflectivity of the cordierite-alumina[9] was estimated to be 0.78 at 300 K. A spray nozzle was placed at 0.37 m beneath the lower surface of the porous plate. The mass flow rate of water supplied to the nozzle from a water reservoir was varied from 1.67×10^{-5} kg/s to 8.33×10^{-5} kg/s, and the airflow rate required for atomization was less than 5 Nl/min. Variations in the mass of the water reservoir were continuously measured using an electrobalance, which can measure to 0.5 g. The corresponding water-droplet loading density G_w defined by a volumetric flow rate of water per unit cross-sectional area of the acrylic test pipe ranged from 1.83×10^{-6} $m^3/s/m^2$ to 9.11×10^{-6} $m^3/s/m^2$, depending on the mass flow rate of water supplied. Four 250-W infrared lamps were used to heat the upper surface of the heat shield radiatively. The amount of radiant energy from the infrared lamps was measured on the upper surface of the heat shield using a still-type water calorimeter that consists of a 0.1-m-inner diameter and 0.0015-m-thick bakelite disk with 0.003-m-high rim and 1.3×10^{-5} m thick polyethylene film that contains 0.025 kg of water. The voltage of the infrared lamps was kept at 100 V, and the corresponding radiative heat flux was found to be 22.2 kW/m^2. The relative uncertainity in measuring the irradiation was estimated to be ±4%. Because

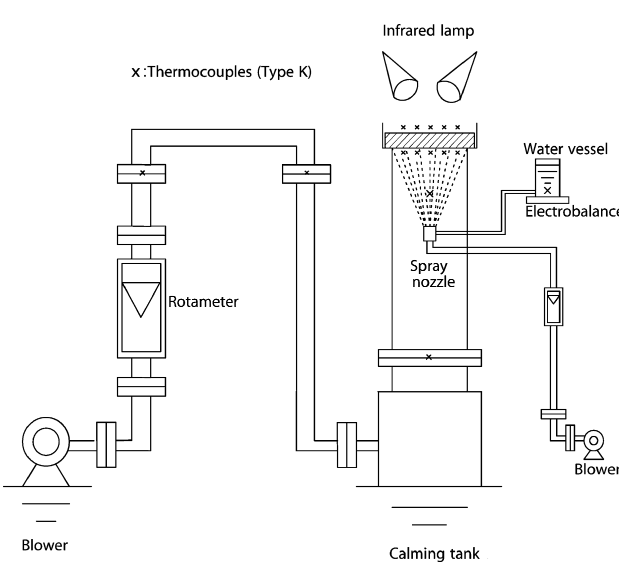

Fig. 1 Schematic diagram of the experimental apparatus.

the mist flowed upward through the test section, the direction of mist flow and radiant heat are opposite to one another. After the infrared lamps were switched on, it took about 60–90 min to attain the steady state of the temperatures of the porous plate. Several type-K thermocouple elements of 3.2×10^{-4} m diam were adhered on the upper and lower surfaces of the porous plate (five for each surface) using alumina cement. The radial temperature variations within the upper surface were found to be from ± 5.2 to ± 12.3 K. The uncertainity in using type-K thermocouples was $\pm 0.4\%$, which is much smaller than the observed radial temperature variations in the present experiment.

Results and Discussion

To evaluate the effectiveness of the present transpiration cooling system, the following temperature efficiency was introduced:

$$\eta_T = (T_R - T_m)/(T_R - T_{\text{in}}) \tag{1}$$

where T_R was evaluated from the relation $T_R = (q_R/\sigma)^{0.25}$, and T_m was given by a simple arithmetic mean of all the measured surface temperatures. Here, the cup-mixing mean temperature of a mixture of air and water droplets is almost equal to an inlet air temperature because a thermal loading ratio between water droplets and a carrier gas is less than 0.03, and thus the contribution of water droplets to a cup-mixing mean temperature of mist could be disregarded. The relative uncertainity in η_T was evaluated to be $\pm 11\%$.

Figure 2 shows variations in η_T against Reynolds number defined by $u D_s/\nu_f$. Here , D_s represents an equivalent strut diameter defined by Eq. (2):

$$D_s = 2w(0.0254/PPI)\Big/\sqrt{\pi} \tag{2}$$

$$w = 0.5 + \cos\left[\tfrac{1}{3}\cos^{-1}(2\phi - 1) + \tfrac{4}{3}\pi\right] \tag{3}$$

As seen from this figure, the temperature efficiency increases with Reynolds number Re and water droplet loading density G_w: the difference between the mean heat shield temperature and an inlet air temperature decreases as Re and G_w increase because the volumetric heat-transfer coefficient between the solid phase and the injected fluid increases with Re and G_w. Moreover, η_T for CA#13 is slightly lower than that for CA#06: this is caused by the fact that the optical thickness CA#13 ($\tau_o^* = 2.86$) is greater than that of CA#06 ($\tau_o^* = 1.22$), and thus the slab absorptance of CA#13 is greater than that of CA#06. Here, the optical thickness τ_o^* was defined by a product of the extinction coefficient of a porous plate and its thickness.

The effect of the water-droplet loading density G_w on the temperature efficiency η_T is shown in Fig. 3. The η_T increases with G_w and tends to be asymptotic to a constant, but η_T can be augmented by only a little addition of water droplets, say, G_w less than 4×10^{-6} (m^3/s/m^2), to a carrier gas, and, when the velocity of a carrier gas u was increased beyond a certain value, mist was trapped by the heat shield and a water film began to form on the lower surface of the shield: this critical velocity was about 0.5 m/s for $G_w = 3.6 \times 10^{-6}$ m^3/s/m^2 and about 1.7 m/s for $G_w = 7.1 \times 10^{-6}$ m^3/s/m^2, irrespective of kinds of the porous material examined. Moreover, use of mist for CA#06 is more effective than that for CA#13.

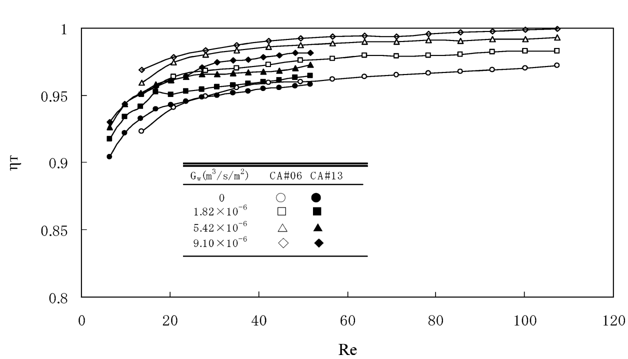

Fig. 2 Variations in the temperature efficiency η_T against Reynolds number Re.

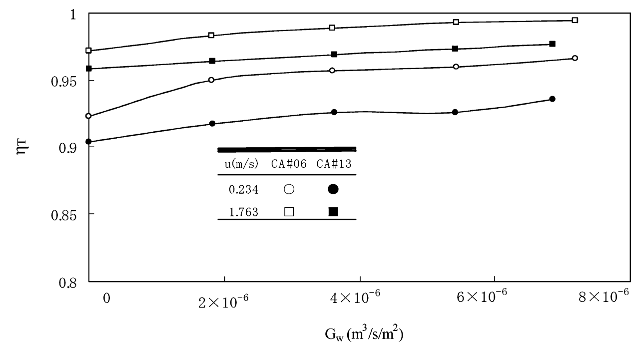

Fig. 3 Variations in the temperature efficiency η_T against the water droplet loading density G_w.

Conclusion

Only a little addition of water droplets to a single gas phase flow, say, water droplet loading density less than 4×10^{-6} m^3/s/m^2, is effective for lowering the mean temperature of the porous heat shield in a transpiration cooling system using an open-cellular porous material.

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