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# THERMAL PROTECTION OF AN AREA FROM AN INCIDENT GAS JET

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*Results are offered from an experimental study of the decrease in thermal effect of a jet upon a normally oriented surface produced by introduction of cold air or water into the central portion of the jet.*

As was shown in [1], one of the possible methods for reducing the effect of a hot gas jet on a normally oriented surface is to feed a coolant (cold air or water) from out of the surface in the vicinity of the critical point (Fig. 1a). In the present study we will consider a method for thermal effect reduction in which the jet itself includes a mechanism for thermal protection of the surface upon which it impinges. We will consider supply to the central portion of the jet of either cold air, cold water, or both (Fig. 1b). Both methods shown in Fig. 1 are intended to form a fan-shaped protective jet along the surface.

The arrangement of the experimental configuration is evident from Fig. 1b. A jet of hot air (having traversed an electrical heater) passes through a constricting nozzle and impinges on a barrier in the form of a thin stainless steel disk with back side coated by thermal insulation material. The cooling agent is fed through a coaxial tube located on the nozzle axis.

During the experiments the following quantities were measured: mass flow rate G, temperature $T_0$, and pressure $P_0$ of the hot air, mass flow rate $G_c$ and temperature $T_c$ of the cooling agent, and temperature distribution $T_w$ over the disk, across the diameter of which thermocouples were installed symmetrically with respect to the center. The hot junctions of the thermocouples were soldered into copper cylinders (2.5 mm diameter) riveted into holes through the disk. The experimental results can be presented in the form of dimensionless wall temperature: $\theta = (T_w - T_c)/(T_0 - T_c)$, where the coolant (water or air) temperature is equal to the temperature of the surrounding medium, $T_c = T_i$.

The experiments were performed under the following conditions: nozzle diameter $D_n = 21$ mm, outer diameter of air and water supply tubes equal to $d_1 = 8$ mm and $d_2 = 3$ mm respectively (wall thickness 1 mm), relative distance from main nozzle mouth to disk $\bar{H} = 1.7$-$6$, relative distance from mouth of coaxial coolant supply tube to disk $\bar{h} = 0.15$ for coolant supply near the disk, or $\bar{h} = 1$ for coolant supply at nozzle mouth, hot air pressure reduction in nozzle $P_0/P_i = 1.7$-$2.5$, braking temperature in nozzle $T_0 = 590$-$860$ K, relative coolant flow rate $\bar{G}_{c\,max} = 1.4\%$ for water, and $\bar{G}_{c\,max} = 15\%$ for air.

The effectiveness of disk cooling for cooling air supply toward the critical point is evident from Fig. 2, which shows the functions $\theta = f(\bar{r}, \bar{G}_c, \bar{H}, \bar{h}, P_0/P_i, T_0)$. With increase in $\bar{G}_c$ the maximum disk temperature $\theta_{max}$ decreases, while its location moves away from the critical point. In particular, the following is evident: upon increase in $P_0/P_i$ the quantity $\theta_{max}$

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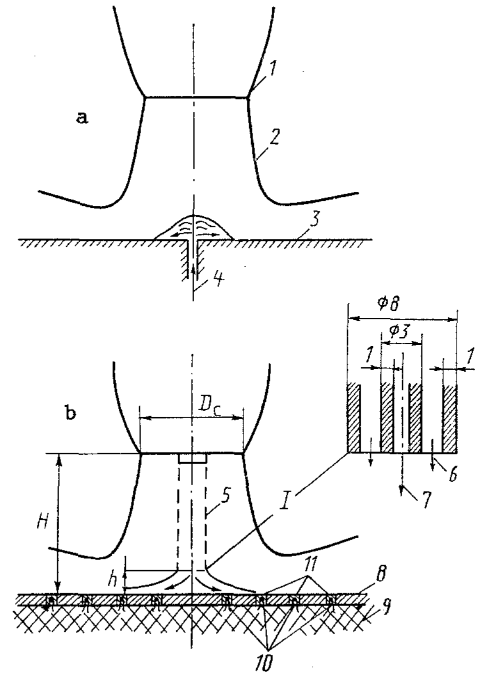


Fig. 1. Thermal protection methods: a) method studied in [1] (1, nozzle; 2, jet; 3, surface; 4, coolant supply (cold air or water)); b) method of present study (5, coaxial coolant supply tube (air – 6, water – 7); 8, thin stainless steel disk; 9, thermal insulation layer; 10, thermocouples; 11, brass cylinders).

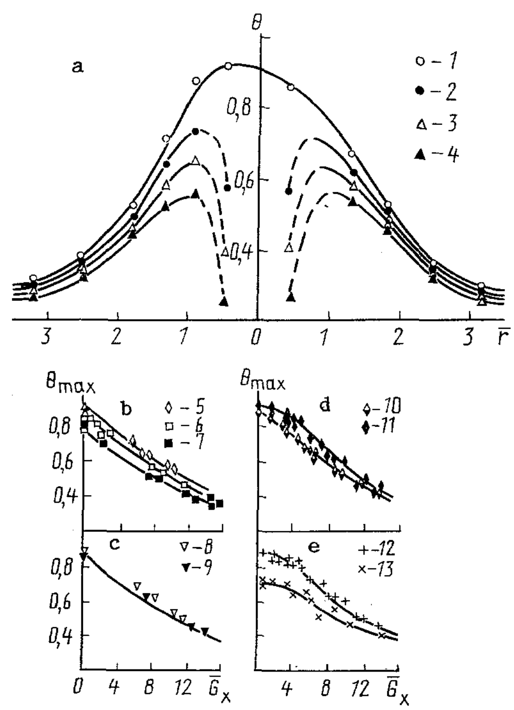


Fig. 2. Surface dimensionless temperature along jet flow radius for air coolant: a) $P_0/P_i$ = 2.5, $\bar{T}_0$ = 2.7, $\bar{H}$ = 1.7, $\bar{h}$ = 0.15 (1, $\bar{G}_c$ = 0; 2, 5.5%; 3, 7.7; 4, 10.2); b) effect of $P_0/P_i$ on $\theta_{max}$ (5, $P_0/P_i$ = 2.5; 6, 2.1; 7, 1.7); c) effect of $\bar{T}_0$ on $\theta_{max}$ ($P_0/P_i$ = 2.1) (8, $\bar{T}_0$ = 1.8; 9, 3); d) effect of $\bar{h}$ on $\theta_{max}$ ($P_0/P_i$ = 2.1) (10, $\bar{h}$ = 0.15; 11, 1); e) effect of $\bar{H}$ on $\theta_{max}$ ($\bar{h}$ = 1, $P_0/P_i$ = 2.1) (12, $\bar{H}$ = 1.7; 13, 3.5).

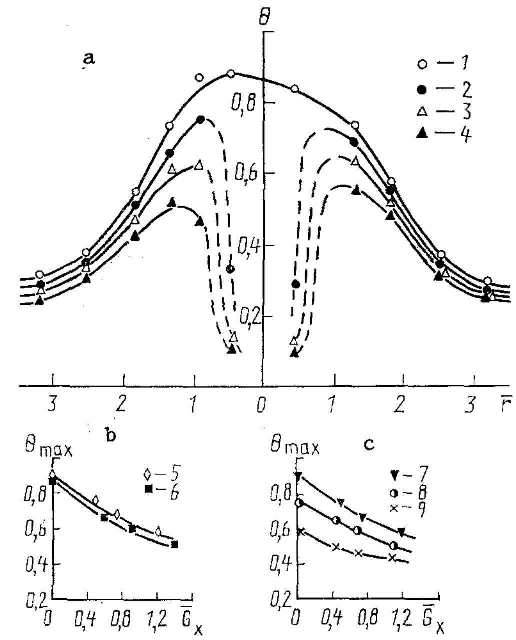


Fig. 3. Surface dimensionless temperature along jet flow radius for water coolant: a) $P_0/P_i = 2.5$, $\bar{T}_0 = 2.7$, $H = 1.7$, (1, $\bar{G}_c = 0$; 2, 0.5%; 3, 0.75; 4, 1.2); b) effect of $P_0/P_i$ on $\theta_{max}$, $\bar{T}_0 = 2.7$, $\bar{H} = 1.7$ (5, $P_0/P_i = 2.5$; 6, 2.1); c) effect of $\bar{H}$ on $\theta_{max}$ (7, $\bar{H} = 1.7$; 8, 3.5; 9, 6).

also increases, while for change in the parameter $\bar{T}_0 = T_0/T_c$ from 1.8 to 3.0 the function $\theta_{max} = f(\bar{G}_c)$ does not change. With cooling air supply at the nozzle mouth (h = H) the effect at low values of $\bar{G}_c(\bar{G}_c < 5\%)$ is markedly less than for cooling air supply in the direct vicinity of the disk surface ($\bar{h} = 0.15$); however with increase in $G_c$ the effectiveness of the two methods gradually becomes the same. Upon removal of the main nozzle from the surface ($\bar{H} = 3.5$) the value of $\theta_{max}$ decreases significantly, and although the effectiveness of cooling at $\bar{G}_c < 4\%$ is very low, the function $\theta_{max} = f(\bar{G}_c)$ lies below the analogous curve for $\bar{H} = 1.7$.

When water was fed to the critical point the effect was significantly nonuniform over the disk surface, probably due to the difficulty of establishing a uniform initial spreading of the liquid film. The nonuniformity decreased upon injection of the water at the nozzle mouth $\bar{h} = 1$. The effectiveness of cooling for this case is shown in Fig. 3. It is evident that with use of water, producing the same effect requires an order of magnitude lower coolant flow rate than with air. The parameter $P_0/P_i$ affects $\theta_{max} = f(\bar{G}_c)$ in the same manner as in the case of air. With increase in $\bar{H}$ the values of $\theta_{max}$ decrease, but the character of $\theta_{max} = f(\bar{G}_c)$ is maintained.

Thus, the studies performed show the quite high efficiency of the proposed cooling technique over a wide range of geometric and gas dynamic parameters. Thus, a coolant expenditure of ~1% (for water ) or ~10% (for air) produces a reduction in maximum temperatures by 30-40%, and in contrast to the method proposed in [1], the effectiveness can be maintained upon displacement of the nozzle along the surface.

## NOTATION

$T_0$, hot gas temperature at nozzle output; $T_c$, coolant (cold air or water) temperature; $T_w$, surface temperature; $T_i$, ambient temperature; $\theta = (T_w - T_c)/(T_0 - T_c)$, dimensionless surface temperature; $\theta_{max}$, maximum dimensionless surface temperature; $\bar{T}_0 = T_0/T_c$, ratio of hot gas and coolant temperatures; $P_0$, total hot gas pressure; $P_i$, ambient pressure; G, hot gas flow rate; $G_c$, coolant flow rate; $\bar{G}_c = G_c/G$, ratio of coolant and hot gas flow rates; $D_n$, main nozzle diameter; H, distance from nozzle section to surface; h, distance from end of coolant supply tube to surface; r, distance from disk center to point of temperature measurement; $\bar{r} = r/D_n$, relative distance from point of temperature measurement; $\bar{H} = H/D_n$, relative distance from nozzle end to surface; $\bar{h} = h/H$, relative distance from tube end to surface.