EFFECTS OF HEAT-TRANSFER RATE AND COOLANT FLOW PARAMETERS ON HEATING AND COOLING RATES

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It is shown that change in the heat-transfer coefficient over a wide range does not alter the heating or cooling rate, which is determined by the heat transfer along the flow.

If a body is cooled or heated in a medium whose temperature varies along the length of the body, one gets the following system of equations:

$$\frac{\partial t}{\partial \tau}+W\frac{\partial t}{\partial x}\mp\frac{4\alpha}{\rho_g c_g D}(T-t)=0, \tag{1}$$

$$\frac{\partial T}{\partial \tau}-a\frac{\partial^2 T}{\partial x^2}-a\frac{\partial^2 T}{\partial y^2}\pm\frac{\alpha}{\rho_m c_m \delta}(T-t)=0. \tag{2}$$

One can neglect the second and third terms in (2) in many cases, namely for hollow cylindrical bodies whose size l along the flow is substantially larger than the transverse linear dimension such as the thickness of the jacket or tube δ, provided that this is made of material of high thermal conductivity ($\mathrm{Bi} \equiv \alpha\delta/\lambda \ll 1$); but even with this substantial simplification, it is still very difficult to solve (1) and (2).

The complicated form to the solution [1, 3] does not enable one to determine with certainty what are the possible ways of accelerating the cooling or heating, nor to identify the basic criteria for processing measurements. Moreover, the form of the system and the solution would seem to indicate that the heat-transfer coefficient is the most important factor as regards the cooling rate for the entire range of parameters. It is sufficient to note here, for example, the generally accepted form for the dimensionless variables in this type of problem:

$$s\equiv\frac{\alpha}{\rho_g c_g D}\frac{x}{W};\quad b\equiv\frac{\alpha}{\rho_m c_m \delta}\left(\tau-\frac{x}{W}\right).$$

Results published on this would seem to indicate that the proposed solutions are the sole ones possible on the basis of the conditions; however, a substantially simplified solution is possible for a large practical region of variation in the independent parameters, and this facilitates the analysis.

It has previously been shown [4, 6] that the range of practical interest in the parameters for channels or tubes for carrying low-boiling liquids up to temperatures of 80-90°K is that in which the cooling rate is determined by the relation between the specific heats of the flow and the object to be cooled, while major changes in the heat-transfer coefficient do not have any substantial influence in the cooling rate.

This region is characterized by the ratio

$$\frac{\pi d\alpha z}{G_g c_g}\equiv \mathrm{St}\frac{F}{f}>20. \tag{3}$$

In this region, the heat-transfer rate is determined by the total thermal capacity of the channel and the scope for carrying away heat in the flow.

The wall temperature change is defined by

$$\frac{T-T_{\mathrm{in}}}{T_0-T_{\mathrm{in}}}=\exp\left[-A\frac{G_g c_g \tau}{\pi d\delta\rho_m c_m z}+B\right]. \tag{4}$$

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The basis for this form for the result is derived from various pieces of indirect evidence, and so one cannot obtain a decision on how far the solution applies for other processes.

To extend these results to other processes, we describe briefly the sequence in which the experiments were performed.

A warm pipeline ($T_0 \cong 300$°K) was supplied with helium gas cooled to 80-90°K, or else with liquid nitrogen; the inlet temperature was kept constant throughout the experiment, while the temperatures at various points along the length were monitored. The heat transfer was convective in the presence of forced motion, and in the case of liquid nitrogen cooling the heat-transfer coefficient was influenced by the film boiling.

We examined not only pipe cooling but also results on other processes that substantially extended the range of heat-transfer mechanisms.

In recent years, much use has been made of transfer of cryogenic liquids from storage vessels to other vessels at short distances, the transfer being provided by a pressure difference exerted on a vapor cushion in the vessel, which is provided by supplying a gas under pressure. This gas may itself be the vapor of the liquid or else an inert gas supplied at the environmental temperature. In the latter case, the gas replaces the displaced liquid, and complex heat and mass transfer processes are involved, in which the gas is cooled and the vessel wall is heated.

The temperature variations in gas and wall can be described by a system of the type of (1) and (2); natural convection is the transfer mechanism if the expulsion is slow. The results from numerous experiments [5] show that the heat-transfer rate from gas to wall is expressed by an equation of the type of (4); in that case also, variation in the heat-transfer coefficient α from 5 to 30 $W/m^2 \cdot deg$ has no effect on the wall heating rate or gas cooling rate. Although numerous measurements were made, the conditions for the processes in all cases corresponded to (3).

The same type of result was obtained on cooling localized masses, as in the operation of cooling systems based on throttling of compressed gases [7]. In such a system, some of the precooled gas is fed directly to the specimen to be cooled and bypasses the heat exchanger. The cold gas is passed through a throttle to the specimen, which may take the form of a thin disk in an insulated vessel. The heat-transfer coefficient is then determined by the flow conditions, and it was 700-4000 $W/m^2 \cdot deg$.

Before we consider the reasons for the similarity between the various processes, we must consider a very simple calculation.

If we assume that the cooled object is a lumped mass of high thermal conductivity, and also that the gas temperature equals the mean temperature of the gas between input and output, then we get the following form for the internal problem neglecting the heat influx from the surroundings [6]:

$$\frac{T - T_{in}}{T_0 - T_{in}} = \exp(-P\tau), \tag{5}$$

where

$$P = \frac{\alpha F}{M_m c_m + \alpha F \dfrac{M_m c_m}{2G_g c_g}} = \frac{2G_g c_g}{M_m c_m} \cdot \frac{1}{1 + \dfrac{2G_g c_g}{\alpha F}}. \tag{6}$$

It is simple to show that the second factor in (6) can be neglected for $\alpha F/G_g c_g > 20$.

Now we consider the solution of (1) and (2). This we consider for pipeline cooling, and we note that the $\partial t/\partial \tau$ term in (1) can be neglected because it is small when the time to cool the tube at a given point is much greater than the time taken to fill the tube up to that point. This corresponds to the majority of cases of practical importance, and it can be demonstrated rigorously by reducing (1) to dimensionless form and estimating the dimensionless coefficients to the terms containing the variables.

We can therefore suppose that the cooling is described satisfactorily by the system

$$W \frac{\partial t}{\partial x} - \frac{4\alpha}{\rho_g c_g D}(T - t) = 0, \tag{1'}$$

$$\frac{\partial T}{\partial \tau} + \frac{\alpha}{\rho_m c_m \delta}(T - t) = 0. \tag{2'}$$

To estimate the order of the factors in (1), we consider an equation describing the instantaneous wall and flow temperatures at various instants; this can be derived by a simple transformation from (1):

$$\frac{dt(\tau)}{T(\tau)-t(\tau)} = \frac{4\alpha}{W\rho_g c_g}\,\frac{dx}{D}. \tag{1''}$$

It is not possible to integrate (1") exactly, but some important conclusions can be drawn. The integral of the left side is the ratio of the temperature change along the channel from input to output to the lateral temperature difference; the integral of the right side is simply four times the product of St by the ratio of the channel length to diameter. If we specify that $4\,\mathrm{St}\,l/d \gg 1$ within the relevant region, we can assume that the difference in temperature between wall and flow is much smaller than the temperature drop along the channel, and then we can assume $t \cong T$ in determining the stored heat in the pipeline. This assumption is in complete agreement with experiment. For instance, when a cryogenic liquid is displaced from a vessel, the wall temperature ceases to change at all points when no more liquid is taken, and attains a constant value.

The exact value of $\mathrm{St}\,l/d$ at which we can assume $T \cong t$ is dependent on the required accuracy.

Then (1") and (2") can be put as

$$\frac{\partial T}{\partial x} - \frac{4\alpha}{W\rho_g c_g}\,\frac{1}{D}(T-t) = 0, \tag{1'''}$$

$$\frac{\partial T}{\partial \tau} + \frac{\alpha}{\rho_m c_m \delta}(T-t) = 0,$$

$$T(0,\ x) = T_0,\ T(\tau,\ 0) = T_{in}, \tag{2''}$$

or

$$\frac{\frac{\partial T}{\partial \tau}}{\frac{\partial T}{\partial x}} = -\frac{W\rho_g c_g D}{4\rho_m c_m \delta}. \tag{7}$$

The solution thus amounts to integrating (7) with respect to τ and x:

$$\frac{T-T_{in}}{T_0-T_{in}} = -\exp\left[\frac{\rho_g c_g D}{4\rho_m c_m \delta}\,\frac{W\tau}{x}\right]. \tag{8}$$

Experiments have shown that (8) becomes an equation of (4) type on account of the effects of the initial parts of the channel and the response time of the surface over a finite length. The only complex independent variables are the ratios of the overall specific heats of the body and flow together with the homochronicity number.

The solution applies for numerous internal problems when the heat-transfer rate along the flow is substantially less than the heat-transfer rate transverse to the body.

A characteristic feature of these problems is that there is no change in the heating or cooling rate when the heat-transfer coefficient is varied. The cooling rate can be increased for a given size and given thermophysical parameters of the body by increasing the gas density (e.g., by increasing the pressure), by increasing the specific heat (using another coolant), or by raising the linear speed of the coolant.

NOTATION

t, T	are the temperature of coolant (heat transfer agent) and body;
x, y	are the coordinates along flow and normal to it;
ρ_g, ρ_m	are the specific density of heat transfer agent and body;
c_g, c_m	are the specific heat of heat transfer agent and body;
D, δ	are the linear dimensions of body (e.g., diameter and thickness of relatively thin cylindrical bodies);
a	is the thermal diffusivity of body;
W	is the linear flow velocity;
l	is the length of the body;
G	is the mass flow rate of coolant (heat transfer agent);
St	is the Stanton number;

F, f are the heat transfer surface and cross-sectional area of channel;

α is the heat transfer coefficient.

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