TRANSIENT NATURAL CONVECTION IN A LIQUID DURING COOLING

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An experimental study was made for an analysis of cooling of a liquid under conditions of natural convection.

In an earlier study [1] the method of computer-aided numerical integration was found useful for analyzing the problem of a cooling-down liquid with the buildup, the development, and the decay of natural convection also taken into account. A regular convection mode of cooling has been established which, together with the well-known regular Kondrat'ev mode [2], makes it possible to determine the thermophysical properties of liquids.

In a continuation of the author's research on transient convection [1,3-5], here the convective effect in cooling-down liquids has been studied experimentally.

The trends in the cooling of a liquid were examined in horizontal vessels of square cross sections 40 × 40 mm or 20 × 20 mm. Their length, 200 mm, by far exceeded their width. Their upper and lower walls, through which heat transfer was made to take place, were 0.5-mm-thick polished aluminum plates, while the lateral walls were 15-mm-thick plates made of acrylic glass and additionally heat-insulated by an asbestos interlayer.

For heating a test liquid inside a vessel to the necessary initial temperature T_1, the latter was placed between a pair of horizontally lying flat heat exchangers of adequate capacity. After complete preheating of a liquid, its vessel was quickly transferred, by means of a mechanical device, to between two other heat exchangers both at the same but a much lower temperature T_0. The transferring time was approximately 1-2 sec. The horizontal dimensions of all heat exchangers were much larger than those of a vessel. Water served as the heat carrier in these heat exchangers and its temperature checked accurately within 0.02°C with two ultrathermostats. Calculations of the mean coefficient of heat transfer at the interface of a vessel and a heat exchanger, at a given flow rate of the heat carrier, indicated that the Biot number for all the test liquids had varied within the 400-650 range, i.e., that boundary conditions of the first kind had almost been achieved. The absence of temperature gradients on the surfaces of the heat exchangers was verified by thermocouple measurements.

The temperature in the test liquids was measured with seven differential copper-Constantan thermocouples having junctions 0.03 mm in diameter. Each of these seven thermocouples was pulled horizontally across a vessel between its lateral walls so as to line up all junctions in one vertical row uniformly spaced over the height of a vessel. Such an arrangement made it possible to shift the "string" of thermocouple junctions through any distance away from a lateral wall and thus place them with the zones of ascending and descending streams respectively.

The readings of all thermocouples were recorded simultaneously with a model N-115 loop oscillograph. The temperature was measured with a 0.5°C accuracy. During tests with only one thermocouple its signals were transmitted to a photocompensation amplifier and from there to a type KSP recording potentiometer, the circuit of the latter having been modified slightly for higher sensitivity. The temperature measurements were in this case accurate within 0.05°C.

For visual observation and photographing of the emergent convection streams, small amounts of aluminum powder of the 5-10 μm fraction were added to a test liquid. The illumination system, which produced a plane-parallel light beam, was capable of sweeping any section of the vessel cavity either laterally or longitudinally. Pictures were taken immediately after the start of a test and in definite intervals of time so that the pattern of

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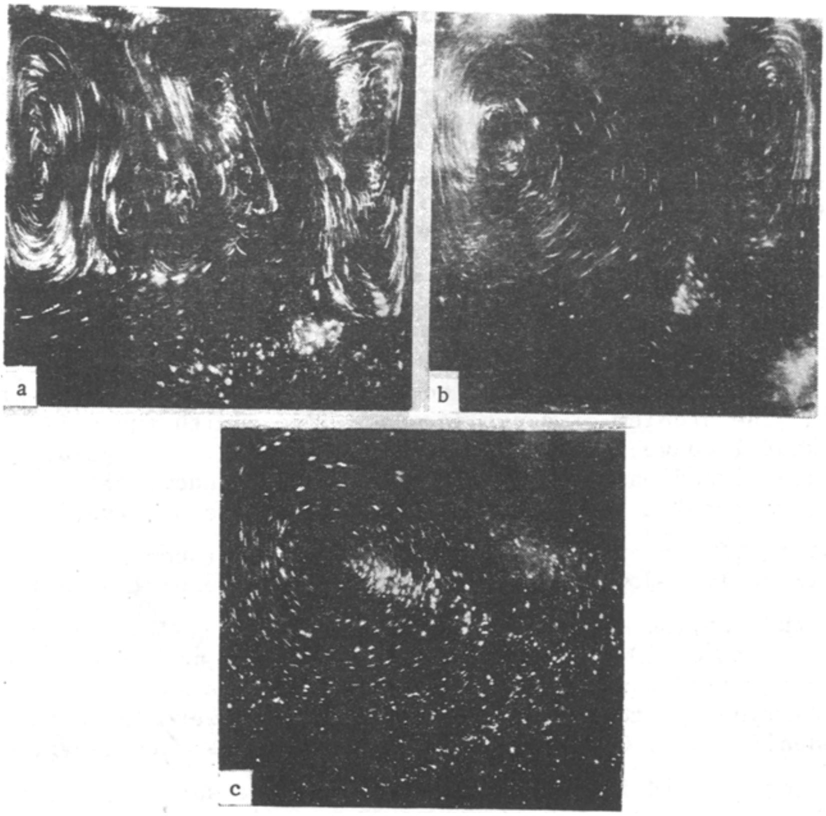

Fig. 1. Photographs of the convection pattern taken across a vertical transverse section of the cavity at various instants of time, with glycerin as the test liquid: (a) 0.5 min after the start of a test (in evidence is a transient multivortex pattern); (b) after 8 min (in evidence is a bivortical pattern, with an ascending stream at the center and descending streams along the walls); (c) after 25 min (in evidence is a univortical pattern, with the liquid flowing upward along the right-hand wall and flowing downward along the left-hand wall).

convection buildup and development could be tracked. Photographs of the section in which the thermocouples were located made it possible to compare the flow pattern with the temperature field.

In order to perform the experiment over wide ranges of values of the Rayleigh number and of the Prandtl number, it was necessary to select substances with widely different thermophysical properties: glycerin, castor oil, transformer oil, ethylene glycol, orthoxylene, and nonyl, octyl, heptyl, isopropyl, ethyl alcohols.

Inasmuch as even a negligible amount of a gas dissolved in a liquid could influence the test results, measures were taken to ensure filling of a vessel with degassed liquids only. Prior to pouring, a test liquid was heated to 40-50°C and held there for at least one hour under vacuum so as to let any dissolved gases escape to the fullest extent possible.

After a liquid has been heated up and its vessel then brought in contact with a "cold" pair of heat exchangers, the temperature of the liquid is lower in its upper and lower regions than in the middle. Conditions necessary for convection to occur are realized only in the upper region, where the direction of the temperature gradient is opposite to the direction of gravity. After a layer of a certain thickness has cooled down, convection builds up with the upper region of the vessel. The layer within which convection flow takes place is quickly expanding downward so as to occupy an increasing part of the vessel. Convection spreads deeper at a fast rate, "explosively" after an induction period [3]. The induction period in all the test liquids was approximately 8-12 sec, i.e., much longer than the time needed to transfer a vessel from the "hot" heat exchangers into the space between the "cold" heat exchangers.

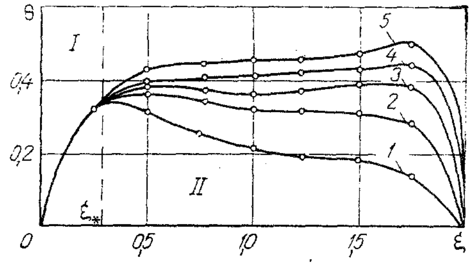

Fig. 2. Temperature profiles in a vertical transverse section of the cavity at the instant of time N_{Fo} = 0.3, with transformer oil as the test liquid $N_{Ra,i} = 3.71 \cdot 10^6$ and $N_{Pr,i}$ = 101 (I, conduction region; II, convection region): 1) η = 0.2; 2) 0.4; 3) 0.6; 4) 0.8; 5) 1.0.

Visual observations made and photographs taken transversely indicate that at the very start a liquid flows in an unordered pattern (Fig. 1a). Subsequently its flow pattern changes and a multivortex one with an even number of vortices forms. Each pair of vortices constitutes a convection cell. As the temperature difference decreases, the flow becomes first bivortical (Fig. 1b) and then univortical (Fig. 1c). A comparison based on photographs taken lengthwise indicates a ridge-like flow pattern. It has been found, furthermore, that changes from one flow mode to another occur in a crisis manner with the restructurization time shorter than the period during which any one pattern prevails.

According to the pictures in Fig. 1, the entire space occupied by a liquid can be tentatively divided into two regions. Within the upper region convection takes place, while within the lower region with a positive temperature gradient the liquid remains stationary. The total liquid volume thus splits into two regions with different mechanisms of heat transfer: natural convection in the upper region and conduction in the lower region.

Thermocouple measurements yielded the temperature profiles in both regions. The temperature profiles in transformer oil at the instant of a bivortical flow pattern are shown in Fig. 2. Each curve here corresponds to a certain distance η from one lateral vessel wall ($0 \leqslant \eta \leqslant 2$). The temperature profiles in the lower part of the vessel are almost the same at various sections, while in the upper part they differ depending on the location of the thermocouple "string" in the ascending stream ($\eta \to 1$) or in the descending stream ($\eta \to 0$) respectively. The boundary between the two regions appears here very distinctly.

An analysis of the trend of the temperature profiled with time has revealed that convective flow, as it builds up and spreads deeper, will very shortly (almost within the induction period) shift downward along the boundary between the region with convection and the region without convection. Later on the location of this boundary does not change much further. This means that, although the intensity of convection decreases with time, the region where convection occurs remains almost constant. The location of the boundary can be regarded as quasisteady.

The dimensions of both regions can be estimated, approximately, on the basis of the following considerations. Let convection already be developed and the location of the conduction region remain almost constant in time. Then at the point $\xi = \delta/h$, where δ is the height of the layer without convection and h is half the vessel height, the thermal fluxes moving upward and downward must be equal

$$q_1 = q_2, \tag{1}$$

where $q_1 = \alpha_{eff}(T_\delta - T_0)$ and $q_2 = \alpha(T_\delta - T_0)$. Here T_δ is the temperature at the interface of both regions; T_0, the temperature at both upper and lower boundaries of the cavity; $\alpha = \lambda/\delta$, the coefficient of heat transfer to the lower part of the vessel; and α_{eff}, the effective coefficient of heat transfer to the upper part of the vessel

$$\alpha_{eff} = \alpha_T (1 + 0.07 \mathrm{Ra}_0^{1/3}), \tag{2}$$

in the case of a plane layer [6] with the coefficient of heat transfer by pure conduction $\alpha_T = \lambda/(2h - \delta)$ and the Rayleigh number $N_{Ra,0}$ defined in terms of the temperature differences $T_\delta - T_0$ and the height $2h - \delta$ of the convection layer.

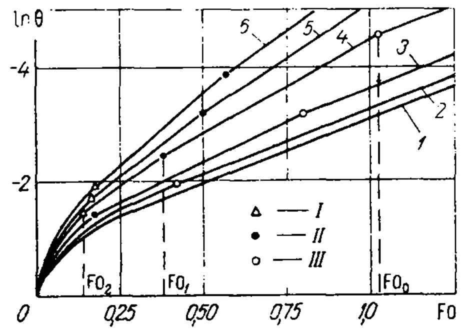

Fig. 3. Logarithm of the dimensionless temperature θ as a function of the Fourier number N_{Fo}, at various values of the initial Rayleigh number $N_{Ra,i}$: 2) $1.35 \cdot 10^5$; 3) $1.49 \cdot 10^5$; 4) $3.71 \cdot 10^6$; 5) $1.21 \cdot 10^7$; 6) $4.80 \cdot 10^7$; curve 1 corresponds to pure conduction, points I correspond to the instant a bivortical pattern has become established, points II correspond to the instant a univortical pattern has become established, points III correspond to the instant natural convection has decayed.

The said Rayleigh number $N_{Ra,o}$ can also be expressed as

$$\mathrm{Ra}_0 = \mathrm{Ra}_i \theta [(2h - \delta)/h]^3,$$

where $N_{Ra,i}$ is the Rayleigh number defined in terms of the initial temperature difference $T_1 - T_0$ and the total vessel height 2h; $\theta = (T_\delta - T_0)/(T_1 - T_0)$ is the dimensionless temperature at the interface.

Letting $\varphi = 0.07\ (\mathrm{Ra}_i\theta)^{1/3}$ and inserting expression (2) into Eq. (1), we obtain an expression for the quantity ξ

$$[\xi/(2 - \xi)][1 + \varphi(2 - \xi)] = 1.$$

The solution to this equation is

$$\xi = (1 + \varphi - \sqrt{1 + \varphi^2})/\varphi. \tag{3}$$

With the aid of expression (3), it is possible to calculate approximately the size of the layer in which heat transfer is effected by conduction. When the initial Rayleigh number varies over a wide range, $N_{Ra,i} = 10^4$–10^7, then the value of ξ varies from 0.2 to 0.5. The dimensionless temperature θ is assumed to be of the order of 1/e.

Thus when heat transfer by convection is much more intensive than heat transfer by conduction, then the effect of convection is manifested in the upper layer of the liquid. The cooling of a liquid layer of height 2h with convection can be formally reduced to the cooling by conduction of a layer of a smaller thickness δ but with asymmetric thermal boundary conditions. Boundary conditions of the first kind are realized at the lower surface of such a fictitiously defined layer and boundary conditions of the third kind are realized at its upper surface. At its upper surface, moreover, the heat transfer coefficient depends on the intensity of convection, i.e., α should be a function of the Rayleigh number.

It follows that any change in the intensity of convective flow in the upper part of a vessel should continuously affect the cooling process in its lower part without convection. A comparison of the temperature–time curve recorded in the experiment at any point by thermocouples with the cooling curve for the same point based on the laws of pure conduction will yield the necessary information describing the trend of natural convection. For this purpose, then, several tests were made with a fixed position of a thermocouple within the region of conductive heat transfer at the distance ξ = 0.25 from the lower boundary of the cavity.

The results of this experiment with several substances are shown in Fig. 3, evaluated in the form of curves depicting the logarithm of the dimensionless temperature θ as a function of the Fourier number N_{Fo}. Curve 1 corresponds to pure conduction. This curve has been obtained by calculation from data in [7] for the given thermocouple location. The other curves lie above curve 1, in the order of increasing values of the Rayleigh number $N_{Ra,i}$. It is quite evident here that a higher value of the Rayleigh number $N_{Ra,i}$ corresponds to a faster cooling of a liquid. As the cooling time can be regarded, for instance, the time within which the dimensionless temperature drops to 3% of the initial temperature of the liquid. The dependence of the cooling time N_{Fo-} on the magnitude of the Rayleigh number $N_{Ra,i}$ and on the magnitude of the Prandtl number $N_{Pr,i}$, within their respective ranges $10^4 < N_{Ra,i} < 10^7$ and $5 < N_{Pr,i} < 5 \cdot 10^3$, can then be described as

$$Fo_{-} = 1.22 (Pr_i/Ra_i)^{0.06}.$$

The relation between cooling time with natural convection N_{Fo-} and cooling time with conduction only $N_{Fo,T}$ can be described by the ratio

$$Fo_T/Fo_{-} = 0.98 (Ra_i/Pr_i)^{0.06}.$$

For transformer oil with $N_{Ra,i} = 3.7 \cdot 10^6$ and $N_{Pr,i} = 10^2$, for instance, the ratio of $N_{Fo,T}$ to N_{Fo-} is equal to 1.92, i.e., convection reduces the cooling time to one half.

We will now proceed to determine the transient characteristics of natural convection. Each curve in Fig. 3 can be divided into a series of straight segments. This indicates that convection in the upper region of the cavity decays not monotonically with time but by passing through some number of precisely defined levels within each of which cooling proceeds at a constant rate. A comparison of thermal measurements with visual observations will make it possible to correlate each straight curve segment with a corresponding definite flow mode. Transition from one straight segment to another signifies a restructurization of the flow pattern.

The last straight segments of curves 2, 3, 4 are parallel to the curve for pure conduction. This is possible only when the convection in the upper part of the vessel is so weak as to have no effect on the heat transfer. As the instant at which such a mode of heat conduction sets in can be regarded the end of the convection decay period $N_{Fo,o}$. An evaluation of the data in criterial form yields the relation

$$Fo_0 = 0.148 Ra_i^{0.28} Pr_i^{-0.2} \quad \text{for} \quad 5 \cdot 10^4 \leqslant Ra_i \leqslant 3 \cdot 10^6, \quad 10 \leqslant Pr_i \leqslant 10^3.$$

It has not been possible to establish a mode of pure conduction for curves 5 and 6, because convection is still significant even at extremely small temperature differences.

The two straight curve segments preceding the purely conductive mode of cooling (curve 4) correspond respectively to bivortical and univortical flow modes. The time $N_{Fo,1}$ separating these two modes can be expressed as a function of the initial Rayleigh and Prandtl numbers, namely

$$Fo_1 = 0.036 Ra_i^{0.45} Pr_i^{-0.45}.$$

Therefore, the dependence of ln θ on N_{Fo} reveals several time zones. The first zone $0 < N_{Fo} < N_{Fo,2}$ is nonlinear and corresponds to short periods. Convection nucleates and develops with a transient pattern forming (Fig. 1a). The second zone $N_{Fo,2} < N_{Fo} < N_{Fo,1}$ corresponds to a ridge-like bicellular pattern (Fig. 1b). It has not been possible to express the time $N_{Fo,2}$ in the form of a criterial relation. The third zone $N_{Fo,1} < N_{Fo} < N_{Fo,o}$ corresponds to univortical flow (Fig. 1c). Finally, at $N_{Fo} < N_{Fo,o}$ the purely conductive mode of cooling becomes established.

Characteristically, not each curve in Fig. 3 passes through all time zones. The establishment of one or another flow pattern after nonlinear cooling depends largely on the initial value of the Rayleigh number $N_{Ra,i}$. Experiments have shown that univortical flow remains the sole pattern beginning from the critical value of the initial Rayleigh number $N_{Ra,i}$ at which convection begins and up to $N_{Ra,i} = 1.5 \cdot 10^6$ (curve 2 in Fig. 3). Within the range $1.5 \cdot 10^5 < N_{Ra,i} < 3.5 \cdot 10^5$ there first becomes established a bivortical pattern, which then changes into a univortical one. Finally, at $N_{Ra,i} > 3.5 \cdot 10^5$ there occurs a successive transition from an unordered three-dimensional flow to a univortical two-dimensional flow.

The limits within which one or another pattern prevails during the cooling process must be characterized by the instantaneous value of the Rayleigh number $N_{Ra,o}$. This value must be referred to the maximum temperature difference within the cavity and the height of the convective layer. Measurements with the thermocouple "string," and also visual observations, have made it possible to determine the critical values of the Rayleigh number corresponding to restructurizations of the flow patterns. The lower Rayleigh number, which determines the threshold of decay of natural convection, has been found to be 860 ± 40. The value of the Rayleigh number $N_{Ra,o}$, which separates bivortical flow from univortical flow, is 8850 ± 150. The transition point from multivortical flow to bivortical flow could be determined much less accurately. It is 38,000 ± 2000.

It is interesting to note that these critical values of the Rayleigh number are much lower than those obtained in other studies [8-10] under steady conditions with heating from below. Such a drop in the values can be attributed to the fact that, because of the temperature being the same at both upper and lower boundaries, convection does not penetrate through the entire depth of the cavity but takes place only within its upper region and thus produces a free lower surface. Furthermore, the processes in this study were highly transient. Such a transiency, together with the inertia of convective flow, should also lower the critical values of the Rayleigh number. The transient characteristics of the cooling process and the decay of natural convection, which have been examined here in liquids, can be found helpful in several thermotechnical applications. One of them is the methodology of measuring the thermophysical properties of liquids and gases. Eliminating the effect of convection is a major problem in the practice of such measurements. It seems very worthwhile, therefore, to consider the feasibility of measuring the thermal properties of liquids without eliminating natural convection but, instead, taking it into account in the analysis of the cooling trends.

Let us return to the relation between ln θ and N_{Fo} (Fig. 3). Beginning at time $N_{Fo,o}$, cooling of a liquid is effected by conduction. The linearity of these curve segments, in the selected coordinates, proves that the dimensionless temperature θ is an exponential function of the Fourier number N_{Fo}, i.e., that a regular cooling mode [2] becomes established in a liquid. The thermal diffusivity a of a liquid can be determined from the slope of this curve segment.

The other two straight segments, by analogy with those in [1], can be regarded as corresponding to regular convective cooling modes. The first of them is characterized by a bivortical flow pattern, the second one is characterized by a univortical flow pattern. Regular convective cooling modes can be realized only in the lower part of a vessel, within the region of conductive heat transfer. The effect of convection, however, is manifested at the upper free surface and determines the heat dissipation associated with any one flow mode. Consequently, the cooling rate depends on the intensity of convection and thus on the initial Rayleigh number $N_{Ra,i}$

$$k_1 = d \ln \theta / d\,\mathrm{Fo} = 1.18 \log \mathrm{Ra_i} - 2.608, \quad (4)$$

$$k_2 = d \ln \theta / d\,\mathrm{Fo} = 0.883 \log \mathrm{Ra_i} - 2.066, \quad (5)$$

where k_1 and k_2 are the cooling rates corresponding respectively to the first and the second regular convective cooling modes.

The value of $N_{Ra,i}$ can be easily calculated according to expressions (4) and (5) from experimental data on k_1 and k_2. The dimensionless complex $g(T_1 - T_0)h^3/a$ in $N_{Ra,i}$, with a defined according to the theory of regular conduction, is known. Thus, knowing $N_{Ra,i}$, one can calculate the complex $\beta/\nu(T)$. With the temperatures T_0 and T_1 matched correspondingly, and with the coefficient of volume expansion known, one can thus determine the temperature dependence of the kinematic viscosity $\nu(T)$.

In this way, the presence of regular convective cooling modes in a liquid makes it possible to experimentally determine its thermophysical properties as functions of the temperature.

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INVESTIGATION OF THE CHARACTERISTICS OF A LOW-TEMPERATURE HEAT PIPE WITH A CRIMPED RETICULAR WICK

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A practical method of simulating weightlessness conditions for ground tests of a low-temperature heat pipe using Freon-22 is developed and checked experimentally.

The use of a wick with a reticular mesh enables one to produce heat pipes with a high heat transmission and low thermal resistance [1].

Possible deviations in the dimensions of the cells of the mesh or their contamination by solid inclusions do not have any considerable effect on the characteristics of the heat pipes. In view of the fact that the channels of the wick are open in vapor space, the establishment of the operating ability of the pipe after drying the wick (e.g., by the action of dynamic forces) is not limited by collapse of the vapor bubbles, as in arterial pipes, and occurs in a short time.

A crimped mesh enables one to construct most simply a capillary structure of the channel type in a long ($L/D \geqslant 50$) heat pipe, both straight and curved. Due to the elasticity of the crimped mesh it is possible to obtain satisfactory contact between the wick and the inner surface of the tube over its whole length.

The crimped mesh forms longitudinal open channels facing the vapor space and closed channels facing the body of the pipe. A feature of this wick is the hydraulic coupling between its channels due to the permeability of the mesh. For a fairly large heat pipe diameter and coarse wick channels not all the channels are filled with liquid under gravitational conditions, even when there is no thermal load. Hence in such heat pipes, when tests are made under gravitational conditions, there is always excess liquid which fills the lower part of the vapor channel, which henceforth will be called a "pool." The dimensions of the pool vary depending on the temperature and the power supplied and have a considerable effect on the thermal characteristics of the pipe when it has been tested. At the same time, when operating the pipe under conditions of weightlessness, the liquid completely fills all the channels of the wick, and there is no pool, and possible excess liquid (due to drying of the channels, excess priming, and an increase in the specific volume when the temperature increases) accumulates at the end of the condensation zone.

The problem therefore arises of determining reliable characteristics of a heat pipe intended for operation under conditions of weightlessness when it has been tested under gravitational conditions.

The problem is solved by determining the value of the slope of the pipe when it is tested under ground conditions, for which the heat-transfer properties of the pipe are not higher than under weightlessness conditions. The slope is determined by a method worked out on a model of the heat pipe with a wick whose channels lie in one plane.

1. Operating Process under Weightlessness Conditions. Consider a heat pipe with a

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