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ANALYTICAL INVESTIGATION OF TWO LIQUID COOLING SYSTEMS FOR TURBINE BLADES

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RESEARCH MEMORANDUM

ANALYTICAL INVESTIGATION OF TWO LIQUID COOLING SYSTEMS FOR TURBINE BLADES

By Thomas W. Jackson and John N. B. Livingood

SUMMARY

A simplified analysis was made to determine flow characteristics and heat transfer in the turbulent and laminar regions of two turbine liquid cooling systems. In the first system, called the straight-through-flow system, the cooling fluid enters the turbine wheel, flows through coolant passages, and is subsequently discharged from the wheel. The second system employs recirculation of the cooling fluid, and is referred to as the loop system. In the loop system, the cooling fluid enters the turbine wheel, flows through coolant passages, and then part of the liquid leaves the turbine while the remainder is recirculated through the coolant passages with the entering coolant. Relations for friction pressure drops and heat-transfer coefficients determined for tube flow are used in the analysis.

Nondimensional charts which simplify calculations of heat-transfer coefficients and coolant temperatures are presented.

A numerical example which compares the straight-through-flow system with the loop cooling system is also presented. For a blade temperature of 1000° F, an increase in effective gas temperature of 350° F could be obtained by use of the loop cooling system instead of the straight-through-flow system. Several other possible liquid cooling systems are described and discussed.

INTRODUCTION

Although liquids are more effective coolants than gases, because of their physical characteristics, a completely satisfactory liquid cooling system for turbines has not yet been evolved. Investigations of liquid cooling have been conducted for several years at the NACA Lewis laboratory. A summary of design information (reference 1) and experimental results on two liquid-cooled turbines (reference 2) are available. Two turbines used for research at the NACA Lewis laboratory and reported in reference 2 incorporated only the simplest possible natural- and forced-convection systems. Both turbines incorporated a free liquid surface within the rotor at a large radius to minimize the lengths of the water columns and, therefore, the hydrostatic pressures encountered in the blades. A theoretical investigation (reference 3) shows that with appropriate design a combination of forced- and natural-convection influences can improve the heat-transfer coefficients in the blade-coolant passages.

Natural-convection pumping forces combined with forced-convection circulation may provide the designer with several types of cooling systems. One system, the straight-through-flow system, results from the extension of the liquid columns in a simple forced-convection turbine to the axis of rotation. A large pumping force results from the density difference in the radial passages between the cold entrance flow and the hot discharge flow from the rotor blades. This pressure or pumping force may be utilized to overcome the high pressure drop occurring in small blade-coolant passages. Another cooling system, the loop system, will result if the hot and cold columns are connected at the axis of the turbine. This interconnection allows the large natural-convection pumping forces to recirculate the coolant many times through the rotor, thereby increasing the heat-transfer coefficient for a given coolant flow rate through the turbine.

A simplified analysis was made to compare the effectiveness of the natural-convection pumping system with and without internal recirculation. In this analysis, heat flow, blade geometry, and coolant-temperature rise are fixed. Blade temperature and effective gas temperature become dependent variables.

It should be emphasized that this analysis serves only to outline the over-all characteristics and potential effectiveness of each system. The analysis does not directly solve the designer's immediate problem of evaluating the blade-coolant-passage configuration and required coolant flow for best performance of any given turbine. The analysis does, however, suggest further refinements in cooled-turbine design to overcome limitations in current systems.

The loop circuit analyzed herein was first proposed by E. R. G. Eckert.

GENERAL DESCRIPTION

Experimental results obtained from two liquid-cooled turbines operated at the NACA Lewis laboratory appear in reference 2. A sketch of one of these turbines, a 12-inch diameter forced-convection water-cooled aluminum turbine, is shown in figure 1. The design of this turbine limited the coolant flow rate in the blade-coolant passages to the laminar range. A typical blade-to-water heat-transfer coefficient for this turbine is 2000 Btu/(hr)(sq ft)($^{\circ}$F). A larger heat-transfer coefficient could be obtained if the flow rate were increased. However, this increase would result in an excessive coolant flow.

Another liquid-cooled turbine, a 14-inch diameter natural-convection water-cooled steel turbine, was operated at the NACA Lewis laboratory. A sketch of this turbine is shown in figure 2. For this turbine, a typical blade-to-water heat-transfer coefficient is 600 Btu/(hr)(sq ft)($^{\circ}$F). This heat-transfer coefficient is much lower than would be expected for the high Grashof numbers encountered in this turbine. In reference 3, by means of a theoretical calculation, it is shown that this lower value is probably the result of the turbulent free-convection boundary layer choking the small-diameter coolant holes. To obtain high heat-transfer coefficients with simple natural convection, the holes should be fairly large in diameter. Turbine blades cannot accommodate holes of large diameter near the leading and trailing edges and consequently other methods must be found to increase the heat-transfer coefficients in small-diameter holes in these locations. It is also shown in reference 3 that the available natural-convection pumping forces can be utilized to increase the heat transfer in a small hole by connecting it at the blade tip to a large-diameter hole, provided that a common coolant reservoir exists in the blade base.

Past theoretical and experimental investigations of liquid cooling point to a turbine-cooling method which is simple and which has possibilities of improved effectiveness.

If a section of a turbine wheel as shown in figure 3, with water entering tube A and leaving tube B is considered, it will be found that, because of differences in the densities of the water in legs A and B and the intense centrifugal forces in a rotating turbine, a coolant flow is generated. The water in leg B is hotter than that in A because it picked up heat in flowing through the turbine blade. Obtaining the same coolant flow for a similar nonrotating system would require a pressure difference of several hundred pounds per square inch between the exit and entrance sections. Because of the intense natural-convection forces in a rotating turbine, the coolant will probably have to be throttled at point C in order to limit the coolant flow to a desired value. In order to make use of the excess pressure difference generated in the cooling system, which is shown in figure 3, a loop circuit is set up using a one-way check valve between the coolant exit and entrance tubes at the rotor hub. In this system, shown schematically in figure 4, (symbols appear in appendix A), the high pressure differences are consumed in recirculating the coolant many times through the loop. No great pressure differentials exist between the coolant entrance and exit from the turbine. For a given coolant flow through the turbine, higher heat-transfer coefficients, therefore, are obtained by means of the loop circuit than by straight-through flow, and sealing problems due to high pressure differences may be eliminated.

The problem of forced-convection cooling using the free-convection forces for pumping (fig. 4) will be analyzed for a simplified system. Equations will be obtained giving the flow and heat transfer for the loop circulation operating with both laminar and turbulent flow. These equations will be used in a numerical example, and various liquid cooling systems using modifications of the systems shown in figures 2, 3, and 4 will be discussed.

ANALYSIS

Turbulent-Flow Heat-Transfer Coefficients

The loop cooling system is shown schematically in figure 4. The fluid enters the turbine at point A, travels from A to B and then through N tubes in each blade of length l_b to point C. From C the fluid travels to D where it divides, one portion going through check valve E back to A to be recirculated and the other portion leaving the system at F.

In the system considered all pressure drop due to friction will be assumed to take place in the small-diameter holes used in the turbine blades. The coolant passages through the turbine disk are assumed to be sufficient in number and large enough in diameter to make the pressure drop through them negligible. This assumption is valid for most cases because the holes in the turbine blades are much smaller in diameter and consequently their pressure drop is much higher than the holes leading from the rotor hub to the blade base. However, before applying the results of the calculations presented herein, an order-of-magnitude calculation should be made for the pressure drops in the system being considered.

Analysis of the system shown in figure 4 requires a relation between the friction pressure drop in the blade and the buoyancy forces between the inlet and outlet coolant passages. For a tube of length l_b with turbulent flow the pressure drop is (reference 4)

$$\Delta p = 0.316 \frac{1}{\left(\frac{uD}{\nu}\right)^{1/4}} \rho \frac{u^2}{2} \frac{l_b}{D} \tag{1}$$

In the system considered a pressure difference also exists due to the buoyancy forces created by the difference in temperature between the inlet leg AB and the outlet leg CD. This pressure difference causes the coolant to circulate through the system. The pressure difference is

$$\Delta p = \omega^2 \rho \beta \; (t_3 - t_2) \frac{L^2}{2} \tag{2}$$

Assuming the pressure drop along DEA to be negligible, and equating equations (1) and (2) gives

$$\omega^2\beta\ (t_3-t_2)\ L^2 = 0.316\ \frac{1}{\left(\frac{uD}{\nu}\right)^{1/4}}\ u^2\ \frac{l_b}{D} \tag{3}$$

In equation (3), t_2 is an unknown. If an energy balance is made of the system, and the specific heat of the coolant is assumed to be constant

$$w_1t_1 + w_2t_3 = (w_1+w_2)\ t_2 \tag{4}$$

from which t_2 is found to be

$$t_2 = \frac{w_1t_1 + w_2t_3}{w_1 + w_2} \tag{5}$$

Substituting equation (5) for t_2 in equation (3) and introducing a Grashof number

$$Gr = \frac{L\ \omega^2\beta\ (t_3-t_1)\ D^3}{\nu^2}$$

gives

$$0.316\ \left(\frac{uD}{\nu}\right)^{7/4} = Gr\left(\frac{w_1}{w_1 + w_2}\right)\frac{L}{l_b} \tag{6}$$

In equation (6) the velocity u and the weight flow w_2 are unknown. Using the equation for turbulent heat transfer in tube flow (reference 5)

$$\frac{Nu}{Pr^{0.4}} = 0.023\ Re^{0.8} \tag{7}$$

and the relation for Reynolds number as a function of weight flow

$$w_1 + w_2 = N\ Re\ \frac{\pi D}{4}\ \mu g \tag{8}$$

and substituting in equation (6), the following relation is obtained

$$\frac{Nu}{Pr^{0.4}} = 0.0345 \left[Gr \left(\frac{L}{l_b}\right)\left(\frac{w_1}{D\mu gN}\right)\right]^{0.291} \tag{9}$$

Equation (9) gives the heat-transfer coefficient for a given weight flow using the recirculating loop system. If the system does not contain the crossover leg DEA, thereby becoming a straight-through-flow system, w_2 in equation (6) is zero and the heat-transfer equation is

$$\frac{Nu}{Pr^{0.4}} = 0.039 \left[Gr \left(\frac{L}{l_b}\right)\right]^{0.457} \tag{10}$$

Equation (10) gives the heat-transfer coefficient for a straight-through-flow system which is operating with a given temperature rise of the coolant. This system is considered to be unthrottled and consequently will pump the maximum amount of coolant for the pressure difference available. This pressure difference is due to the buoyancy forces created by the difference in coolant temperatures between the inlet and outlet passages from the turbine.

Laminar-Flow Heat-Transfer Coefficients

If the system is rotating slowly enough to have a $Gr(L/l_b)$ number which is less than 242,000, laminar flow is obtained. A $Gr(L/l_b)$ of 242,000 is equivalent to a Reynolds number of 2300. Reynolds number as a function of $Gr(L/l_b)$ can be obtained by equating equations (7) and (10).

By use of the equation for pressure drop through a tube of length l_b with laminar flow (reference 3)

$$\Delta p = 32\ \mu \frac{ul_b}{D^2} \tag{11}$$

and the equation for laminar-flow heat transfer (reference 5)

$$Nu = 1.62 \left(Re\ Pr \frac{D}{l_h}\right)^{1/3} \tag{12}$$

the following relation for the heat-transfer coefficient with recirculation is found in a manner similar to that used for turbulent flow.

$$\frac{Nu}{\left[Pr\left(\frac{D}{l_h}\right)\right]^{1/3}} = 0.843\left[Gr\left(\frac{L}{l_b}\right)\left(\frac{w_1}{D\mu gN}\right)\right]^{1/6} \tag{13}$$

If the system does not contain the crossover leg DEA, the heat-transfer equation is

$$\frac{Nu}{\left[Pr\left(\frac{D}{l_h}\right)\right]^{1/3}} = 0.405\left[Gr\left(\frac{L}{l_b}\right)\right]^{1/3} \tag{14}$$

Equation (14) gives the heat-transfer coefficient for a straight-through-flow system which is operating with a given temperature rise of the coolant. This system is unthrottled and is allowed to pump as much coolant as possible.

If the coolant passages in the turbine blades make sharp bends at the blade tips, the laminar boundary layer will be interrupted at the bends; then l_h in equation (12) would represent only one half of the length of the coolant passage. In reference 5, l_h represents the heated length of a straight tube. In the analysis given herein l_h is not considered to include sharp bends and represents the total length of the coolant passage exposed to the hot combustion gases. For convenience, l_h and l_b are indicated in figure 4 as not including the crossover passage lengths at the blade tip. In reality these sections are included in the heating and pressure-drop lengths.

Temperature and Weight Ratios

For design purposes it is desirable to know the weight flow and temperature ratios of the coolant in terms of calculable parameters. From equation (5) an equation relating the weight flows and temperature is obtained

$$\frac{w_1 + w_2}{w_1} = \frac{t_3 - t_1}{t_3 - t_2} \tag{15}$$

By use of equations (7), (8), and (9), the following equation is obtained

$$\frac{w_1 + w_2}{w_1} = 1.31\left[Gr\left(\frac{L}{l_b}\right)\right]^{0.364}\left(\frac{D\mu gN}{w_1}\right)^{0.636} \tag{16}$$

Equation (16) gives the ratios of temperatures and weight flows for turbulent flow in terms of known quantities.

For laminar flow, by use of a procedure similar to that used for turbulent flow, the relation for the temperature and weight-flow ratios is

$$\frac{w_1 + w_2}{w_1} = 0.111\left[Gr\left(\frac{L}{l_b}\right)\left(\frac{D\mu gN}{w_1}\right)\right]^{1/2} \tag{17}$$

Blade Temperature for Given Coolant Flow and Temperature Rise

Determination of the blade temperature for a given heat input is of special importance to the designer. In the above derivations, the heat input equal to $w_1 c(t_3 - t_1)$ was assumed to enter at the tip of the blade. This assumption was made in order to simplify the analysis and allow the coolant temperature to change instantaneously from t_2 to t_3 at the blade tip. In reality, the heat would enter over a given heating length l_h with a temperature difference between the blade and the cooling fluid. Because the weight flow and temperature rise of the coolant are independent variables, the blade temperature that fulfills the equilibrium heat flow for the specified conditions must be determined from the heat-balance equation

$$Q = w_1 c(t_3 - t_1) = H_i \pi D l_h N\left(t_B - \frac{t_3 + t_2}{2}\right) \tag{18}$$

The quantity $(t_3 + t_2)/2$ is used in equation (18) for the average temperature of the coolant flowing through the heating length l_h. For the condition which will be normally encountered in turbines, the arithmetical average temperature difference is approximately equal to the logarithmic mean temperature difference, which is commonly used in tube flow.

Substitution of equation (5) in equation (18) and simplification results in the following expression for blade temperature

$$\frac{t_B - t_3}{t_3 - t_1} = 0.318\left(\frac{D}{l_h}\right)\left(\frac{Pr}{Nu}\right)\left(\frac{w_1}{D\mu gN}\right) - 0.50\left(\frac{w_1}{w_1 + w_2}\right) \tag{19}$$

Equation (19) holds for both laminar and turbulent flow.

In equation (19), t_B is the blade temperature at the coolant-passage surface. The blade temperature on the outside surface of the blade would depend on the amount of heat transferred, the thermal conductivity of the blade material, and the configuration of the blade.

In the subsequent numerical example, the outside-blade-surface temperature was calculated assuming a blade-wall thickness of 0.24 inch and parallel surfaces. (See appendix B for details.) In reference 6, methods of calculating blade temperatures for other wall configurations such as wedges, concentric circles, and so forth are given. When the outside-blade-surface temperature is found, the effective gas temperature can be determined from the heat flow, the outside surface area, and the gas-to-blade heat-transfer coefficient.

RESULTS AND DISCUSSION

Nondimensional Charts

The analytical expressions are presented as nondimensional charts in figures 5 and 6. Equations (9) and (10) for turbulent flow are plotted in figure 5(a). In figure 5(a) the heat-transfer coefficient variations with $Gr(L/l_b)$ which represents primarily the rotation and temperature rise of the coolant, and $w_1/D\mu gN$ which represents primarily the coolant flow are shown. Equation (10) has a slope of 0.457 and represents a cooling system which does not contain a crossover leg. The family of lines representing equation (9) has a slope of 0.291 and represents a system containing the crossover leg DEA (fig. 4).

For a loop system with a fixed weight flow, a value of $Nu/Pr^{0.4}$ can be determined for each value of $Gr(L/l_b)$, and hence, as a range of $Gr(L/l_b)$ is covered, a series of values of $Nu/Pr^{0.4}$ results. The solid curves in figure 5(a) represent such values for fixed weight flows. For an unthrottled straight-through-flow system, a value of $Nu/Pr^{0.4}$ can be determined for each value of $Gr(L/l_b)$. The dashed curve in figure 5(a) represents such a system. At a given Grashof number, this system will pump a specific weight flow of coolant. This weight flow is equal to that represented by the line of equation (9) which intersects the line of equation (10) at the given Grashof number. If the straight-through-flow system is limited to a fixed weight flow, it becomes a throttled system and the factor $Gr(L/l_b)$ has no significance in so far as the determination of $Nu/Pr^{0.4}$ from figure 5(a) is concerned. The value of $Nu/Pr^{0.4}$ for a throttled-straight-through system for a fixed weight flow w_1 can be obtained from figure 5(a) by determining the point of intersection of the solid line representing the given weight flow with the dashed line.

Similar curves for laminar flow, which represent equations (13) and (14), are plotted in figure 5(b).

The $Gr(L/l_b)$ numbers in figure 5 which indicate the transition zone between laminar and turbulent flow compare to Reynolds numbers for a straight-tube flow of 2300 and 10,000.

The temperature ratios and weight flows, equations (16) and (17) for turbulent and laminar flow respectively, are plotted in figure 6. These curves are used in determining the number of circulations the fluid makes in the loop and also the temperature at which the coolant enters the turbine blades.

Application of Charts to Numerical Example

In appendix B, sample calculations are made for a 36-inch-diameter turbine using the straight-through-flow and loop cooling systems. Using a specified operating condition, weight flow of coolant, and coolant temperature rise, heat-transfer coefficients, flow rates, temperature ratios, and blade temperatures were found for both systems. In order to illustrate the use of figure 5(a), results of a numerical example are shown in the figure. For a value of $Gr(L/l_b)$ of 4.4×10^8 and an assumed weight flow of 108 pounds per hour per blade, which is equivalent to a weight flow parameter $w_1/D\mu gN$ of 2610, the value of $Nu/Pr^{0.4}$ with the loop system is 110; for the same weight flow and a throttled-straight-through-flow system, $Nu/Pr^{0.4}$ is shown to be 15. For an unthrottled-straight-through-flow system and the same value of $Gr(L/l_b)$ equal to 4.4×10^8, the weight flow w_1 is found to be approximately 5100 pounds per hour per blade. Limitations such as turbine size, excessive radiator capacity requirements, and prohibitive pumping power losses obviously render a coolant flow of such magnitude impractical. If an effective gas temperature is calculated which fulfills the condition of equilibrium at this coolant flow and the other assumed conditions listed in appendix B, a value in excess of 30,000° F is obtained. This value obviously is excessive for turbine application and serves to indicate that use of these figures should be confined to flow ranges and temperature ranges within practical limits. From figure 6(a) the amount of fluid traveling in the loop was found to be 11.8 times the through weight flow. In addition, from figure 6(a) the temperature of the water entering the turbine blades for the loop system was found to be 187° F as contrasted to 50° F for the straight-through-flow system. The temperature of the coolant entering the turbine for both systems is 50° F.

From equation (19) the temperature of the turbine blade at the coolant surface was found to be 232° F for the loop system and 404° F for the straight-through-flow system. For the loop system the blade temperature is only 38° F above the average temperature of the coolant, whereas in the straight-through-flow system the blade temperature is 279° F higher than that of the coolant. For the straight-through-throttled flow, the flow condition was assumed to be turbulent. In reality, the Reynolds number of the flow was only 3320. This Reynolds number is in the transition range; consequently, the heat-transfer coefficient is smaller than that calculated. The smaller heat-transfer coefficient would give a higher blade temperature than indicated here.

The outside-blade-surface temperatures were calculated under certain prescribed assumptions listed in figure 7 and in appendix B, and were found to be 556° and 728° F for the loop and straight-through-flow systems, respectively. From a knowledge of outside-blade-surface temperature and the average outside gas-to-blade heat-transfer coefficient, it is possible to calculate the value of t_e which fulfulls the condition of equilibrium at any specified coolant flow and temperature rise. This value is the maximum permissible for the given conditions and blade temperature. (See appendix B for details.) These calculations were repeated for a range of coolant flow for the assumed conditions listed in the figure and in appendix B, and the variations in blade temperature and the effective gas temperature with coolant flow are shown in figure 7. Since the gas-to-blade heat-transfer coefficient will vary as the effective gas temperature varies, an assumption that a set of constant conditions prevails over an appreciable gas temperature range is not strictly accurate. However, these changes are secondary effects compared with the effect of varying coolant weight flow and the resulting curves indicate a correct general trend.

The break in the temperature curves in figure 7 for the throttled-straight-through-flow system (fig. 3) at a coolant flow rate of approximately 70 pounds per hour per blade is caused by a change in the flow from laminar to turbulent. Below 70 pounds per hour per blade, the coolant flow is laminar and the low heat-transfer coefficient necessitates a large temperature difference to transfer a given quantity of heat. The flow was turbulent in all cases for the loop circulation.

For an allowable blade temperature of 1000° F, which is a reasonable value for nonstrategic materials, figure 7 shows that the effective gas temperature for throttled-straight-through-flow system is approximately 2100° F; for the loop system the effective gas temperature is approximately 2450° F. As shown in figure 7, these values were determined for a coolant weight flow of 108 pounds per hour per blade. This quantity of coolant flow may also be expressed in terms of a coolant-to-gas flow ratio by dividing the coolant weight flow by an estimated combustion gas weight flow. A coolant-to-gas flow ratio of approximately 2.5 percent results.

Another example was calculated for a throttled-straight-through-flow system for a 3500° F effective gas temperature in order to determine the flow rate required to maintain an outside-blade-surface temperature of 1000° F. This selected gas temperature represents a value which can be envisioned in future turbine applications. For the calculation, it was necessary to change the previously used assumptions relative to coolant weight flow w_1 and wall thickness Δx. It was found, for this case, that w_1 equals 417 pounds per hour per blade and Δx equals 0.10 inch. This value of w_1 corresponds to a coolant-flow ratio of about 9 percent.

Design Applications

Many systems using liquids for coolants are possible. The illustrative examples of the straight-through-flow and loop cooling systems used water as the coolant. Fuel or other liquids may feasibly be used as coolants. In some cases, more than one type of cooling liquid may be desirable. The following discussion will consider one-liquid systems and two-liquid systems for application to turbine cooling.

One-liquid systems. - In order to reduce cooling requirements when gas temperatures are not extreme, it may not be desirable to have high heat-transfer coefficients in the coolant passages. In such cases, blade temperatures should be maintained at as high a level as possible. Schematic diagrams of several one-liquid cooling systems appear in figure 8. Systems a, b, and c are similar to the present-day free- and forced-convection arrangements. The forced-convection system using the loop circulation is indicated by d and is analyzed herein. The calculation previously described (fig. 7) indicated that for systems c and d at an effective gas temperature of 1500° F, the outside-blade-surface temperatures would be 780° and 660° F for the throttled-straight-through-flow and loop systems, respectively. Raising the blade temperatures may be desirable to reduce cooling losses. Higher blade temperatures could be obtained if a liquid were used which did not boil or vaporize at low temperatures. High blade temperatures can be obtained using water if the water is allowed to vaporize and be recondensed in some manner in a region outside of the blades. One such arrangement is shown in figure 8(e) where the water is allowed to vaporize in the blade, pass through a pressure relief valve, and then jet into a stream of cold water where condensation takes place. Taking the coolant out of the rotor in the form of steam may not be desirable in aircraft installations because of possible ducting complications. In naval or stationary installations it may be more feasible to remove the steam and utilize it in auxiliary units.

Another system which may permit blade temperatures to be increased is shown in figure 8(f). In this system the cooling circuit creates high circulation in the loop from the axis to the base of the blades. A free-convection cooling system similar to b cools the blades. The crossover holes shown in the blade tip are fully described in reference 3. Because of the high water pressures in the blades due to centrifugal forces, coolant temperatures can be increased. The cooling water which would normally vaporize when the pressure is reduced is cooled to a temperature below 200° F by mixing with cold water in the loop where pressures are appreciable. This arrangement prevents vaporization when the coolant leaves the turbine and the pressures are low.

In future developments of turbines driving supersonic compressors, especially with low strategic-alloy content metals, stresses may make the maintenance of very low blade temperatures desirable. A loop circuit within the blades to obtain low blade temperatures will be very advantageous.

The loop circuit (fig. 8(d)) obviates one inherent difficulty in the simple forced-convection system shown in figure 8(c). As mentioned previously, in this system (fig. 8(c)) the pumping force due to the buoyancy forces is considerable and the flow will have to be throttled to take care of the excess pressure differences in closed cooling systems. The pressure difference creates a difficult sealing problem which should not exist in the loop circuit.

Two-liquid systems. - It may be desirable in some instances to use a two-liquid system for cooling a turbine. For instance, if the turbine blades are to be easily removable, cooling the turbine disk with water or fuel and the blades by means of sodium or some other liquid metal may be feasible. Such an arrangement is shown in figure 9. The sodium would remove the heat from the blades and transport it by free convection to the blade roots where it would be removed by water or fuel operating on the loop principle described in this paper.

CONCLUDING REMARKS

A simplified calculation was carried out for the flow and heat transfer in the turbulent and laminar regions for a loop cooling system in a turbine. Formulas for blade temperatures, heat-transfer coefficients, temperature ratios, and weight flow ratios were obtained. Nondimensional charts were constructed to simplify numerical calculations for the general loop system discussed in this paper.

A sample calculation indicated that for a representative turbine operating with a given water weight flow and water temperature rise the heat-transfer coefficient for the loop circulation was over seven times as great as that using only throttled straight-through flow. The increased circulation due to the loop tends to make the flow more turbulent. In cases where straight-through flow would be in the laminar region, the loop circulation causes the flow to become turbulent with a corresponding additional increase in heat transfer. In the sample calculation, for a blade temperature of 1000° F, the permissible effective gas temperature was approximately 2100° F for the throttled-straight-through-flow system and 2450° F for the loop system.

Because of the high heat-transfer coefficients resulting from the loop circulation, low turbine-blade temperatures are obtained. If raising the blade temperature is desirable, variations of the loop circuit can be used. Several possible one- and two-liquid systems are described and discussed.

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APPENDIX A

The following symbols are used in this report:

A_o outside area of blade, sq ft

c specific heat, Btu/(lb)(°F)

D diameter of tubes used for pressure drop, ft

Gr Grashof number, $\dfrac{L\omega^2\beta(t_3 - t_1)D^3}{\nu^2}$

g acceleration of gravity, ft/sec^2

H_i heat-transfer coefficient between blade and coolant, Btu/(sec)(sq ft)(°F)

H_o heat-transfer coefficient between blade and gas, Btu/(sec)(sq ft)(°F)

k heat conductivity of coolant, Btu/(sec)(ft)(°F)

L radial length from centerline of hub to tip of cooling passage, ft

l_b length of passage for pressure drop, ft

l_h length of passage for heat transfer, ft

N number of passages for calculating pressure drop

Nu Nusselt number, $\dfrac{H_i D}{k}$

Pr Prandtl number, $\dfrac{c\mu g}{k}$

ΔP change in pressure, lb/sq ft

Q heat transferred, Btu/sec

Re Reynolds number, $\dfrac{uD}{\nu}$

t temperature, °F

t_1 temperature of coolant entering turbine, °F

t_3 temperature of coolant leaving turbine, °F

u	mean velocity of coolant in tube of diameter D, ft/sec
w	weight flow of coolant, lb/sec
$\frac{w_1}{D\mu gN}$	dimensionless ratio
Δx	thickness of blade material for heat transfer, ft
β	expansion coefficient, $1/^{\circ}F$
ρ	mass density of coolant, $lb\text{-}sec^2/ft^4$
μ	absolute viscosity, lb-sec/sq ft
ν	kinematic viscosity, sq ft/sec
ω	angular velocity, radians/sec

Subscripts:

1,2, and 3	denote location values in loop circuit
B	denotes blade temperature at coolant passage when used with t, or blade thermal conductivity when used with k
o	denotes gas surface interface blade temperature when used with t_B
e	denotes effective gas temperature when used with t

APPENDIX B

In the numerical calculations, the following conditions were assumed (fig. 4):

$l_b = 10$ in.	$t_1 = 50^\circ$ F
$l_h = 8$ in.	$t_3 = 200^\circ$ F
$L = 18$ in.	$H_o A_o = 25$ Btu/(hr)($^\circ$F)
$N = 3$	$k_B A_o = 1.0$ Btu ft/(hr)($^\circ$F)
$D = 0.125$ in.	$\Delta x = 0.24$ in.
$\omega = 583$ radians/sec	$\beta = 0.0001/^\circ$F
$w_1 = 108$ lb/hr blade	$Pr = 3.57$

Using the assumed conditions, the heat-transfer coefficients, flow rate, temperature ratios, blade temperatures, and effective gas temperatures must be calculated for the loop cooling system and the system where leg AD with check valve E is removed and the flow is throttled.

$$Gr\left(\frac{L}{l_b}\right) = \frac{L\,\omega^2\beta(t_3 - t_1)D^3}{\nu^2}\left(\frac{L}{l_b}\right) = 4.4 \times 10^8$$

and

$$\frac{w_1}{D\mu gN} = 2610$$

From figure 5(a)

$$\frac{Nu}{Pr^{0.4}} = 110 \quad \text{for the loop circuit and}$$

$$\frac{Nu}{Pr^{0.4}} = 15 \quad \text{for the straight-through-flow system throttled to maintain a coolant temperature rise of } 150^\circ \text{ F}$$

From figure 6(a)

$$\frac{w_1 + w_2}{w_1} = \frac{t_3 - t_1}{t_3 - t_2} = 11.8$$

or the amount of fluid traveling in the loop is 11.8 times the through weight flow of coolant. From the above

$$t_3 - t_2 = \frac{150}{11.8} = 12.7^\circ \text{ F} \approx 13^\circ \text{ F}$$

and

$$t_2 = 200 - 13 = 187^\circ \text{ F}$$

The temperature of the water entering the heated section of the loop is therefore 187° F and the temperature rise of the water occurring through the heated section of the loop is 13° F.

Substitution in equation (19) gives

$$\frac{t_B - t_3}{t_3 - t_1} = 0.211 \qquad \text{or} \qquad t_B = 232^\circ \text{ F}$$

for the loop circuit. For the straight-through flow

$$t_B = 404^\circ \text{ F}$$

For the loop circuit, the blade temperature is only 38° F above the average temperature of the coolant $\frac{t_3 + t_2}{2}$, whereas in the straight-through-flow system the blade temperature is 279° F higher than that of the coolant $\frac{t_3 + t_1}{2}$.

The foregoing blade temperatures are those at the coolant passage. With the assumption of a blade wall thickness of 0.24 inch and parallel surfaces, the outside-surface blade temperatures were calculated for the given heat flow from the following equation.

$$Q = w_1 c \left(t_3 - t_1\right) = k_B A_o \frac{t_{B,o} - t_B}{\Delta x}$$

When the outside blade temperatures were found, the effective gas temperatures were obtained by use of the assumed $H_o A_o$ using the following equation.

$$Q = w_1 c \left(t_3 - t_1\right) = H_o A_o \left(t_e - t_{B,o}\right)$$

The results of these calculations for a given rotation and temperature rise of the coolant are plotted for various weight flows in figure 7.

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3. Eckert, E. R. G., and Jackson, Thomas W.: Analytical Investigation of Flow and Heat Transfer in Coolant Passages of Free-Convection Liquid-Cooled Turbines. NACA RM E50D25, 1950.

4. Jakob, Max: Heat Transfer. Vol. I. John Wiley & Sons, Inc., 1949.

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6. Livingood, John N. B., and Brown, W. Byron: Analysis of Temperature Distribution in Liquid-Cooled Turbine Blades. NACA TN 2321, 1951.

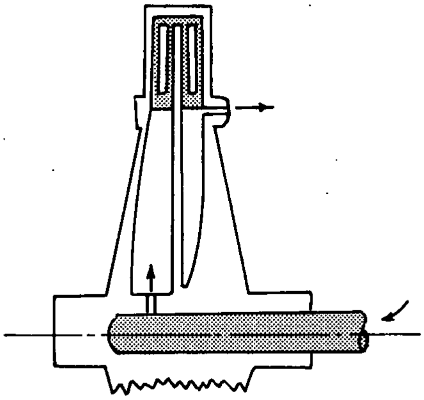


Figure 1. - Forced-convection-cooled aluminum turbine.

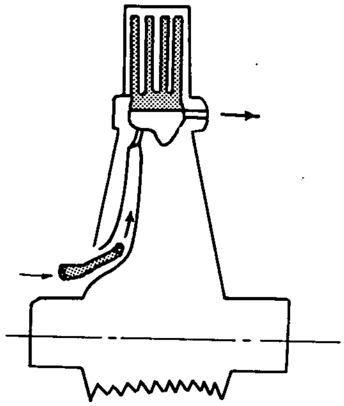


Figure 2. - Free-convection-cooled steel turbine.

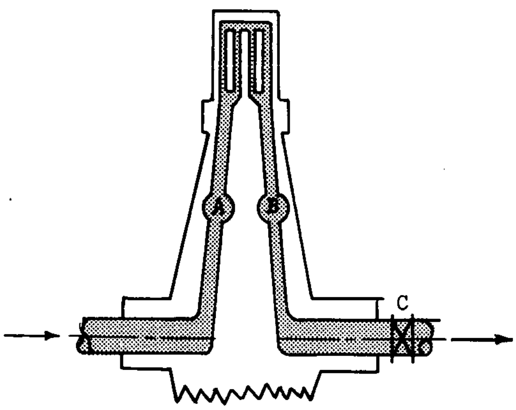


Figure 3. - Suggested liquid-cooled turbine.

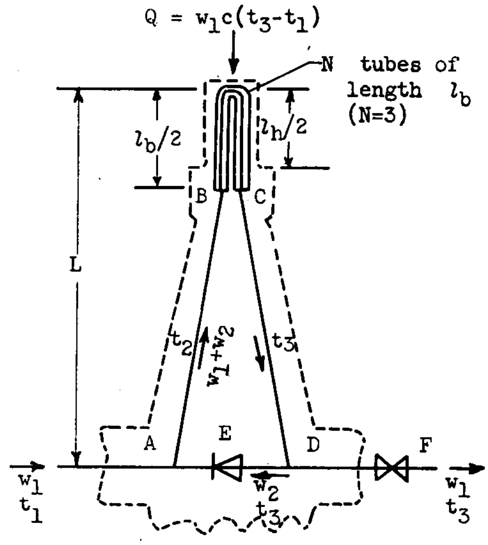
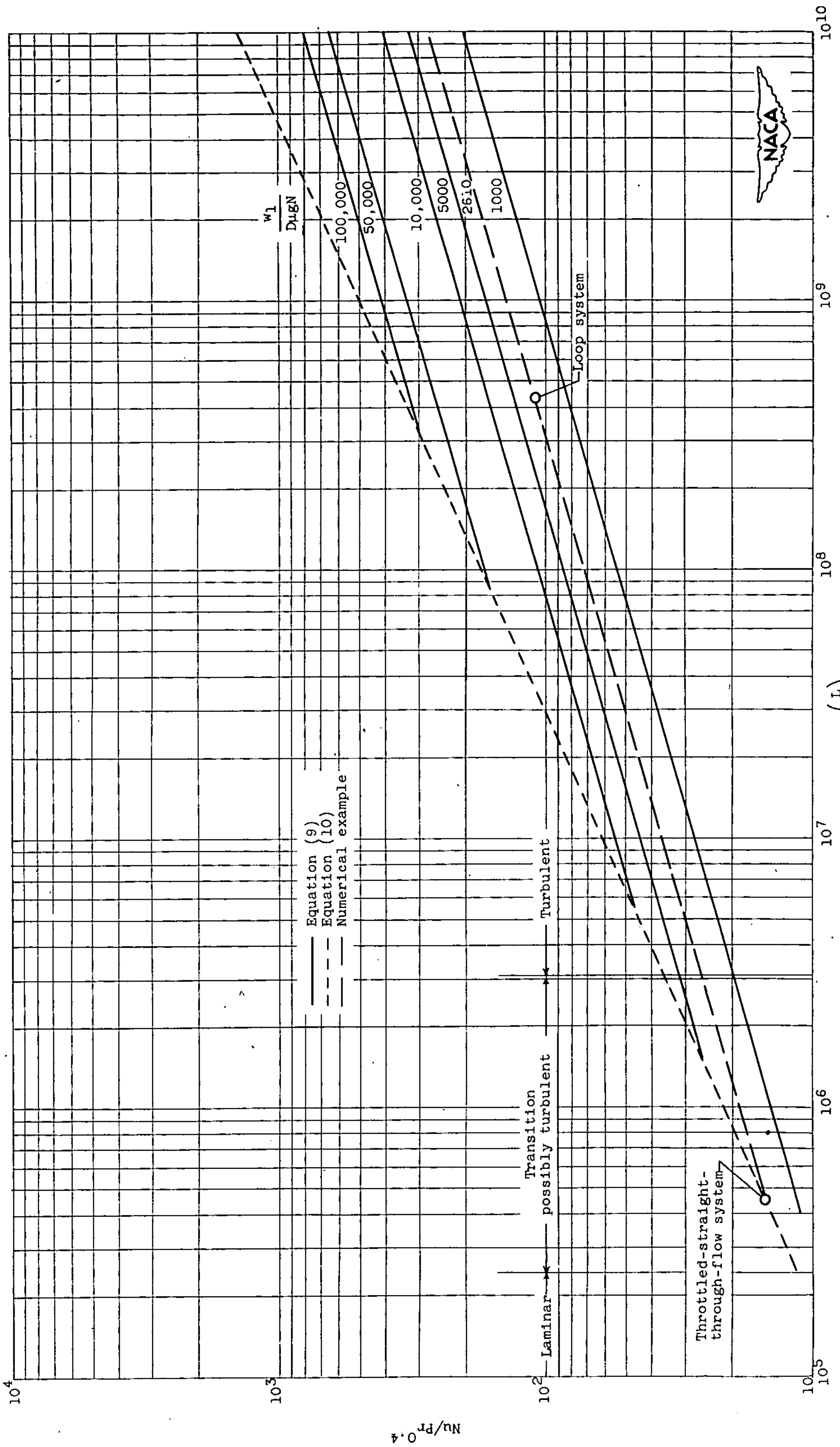
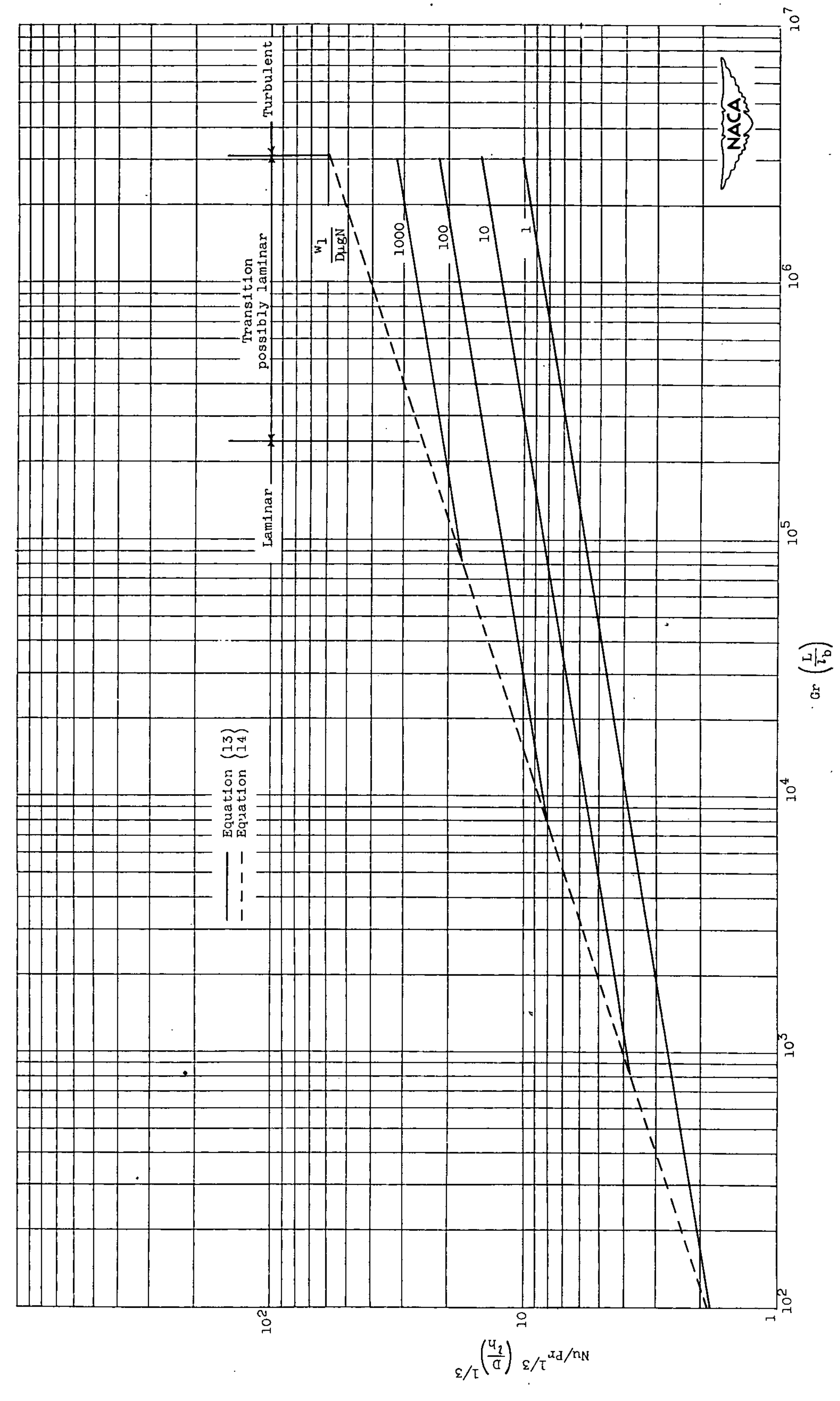


Figure 4. - Schematic drawing of liquid-cooled turbine.



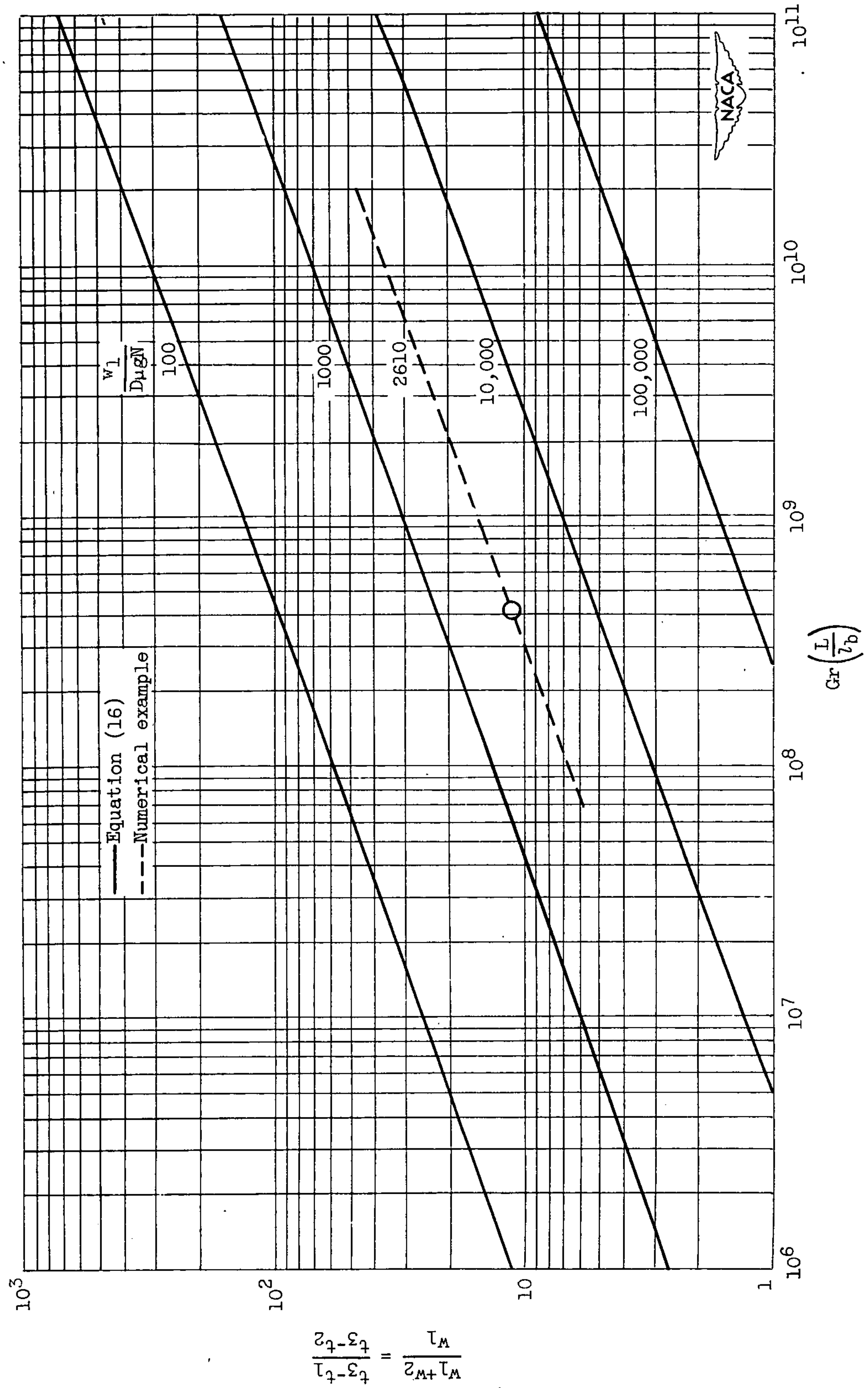
(a) Turbulent flow.

Figure 5. - Heat-transfer coefficients for straight-through-flow and loop cooling systems.



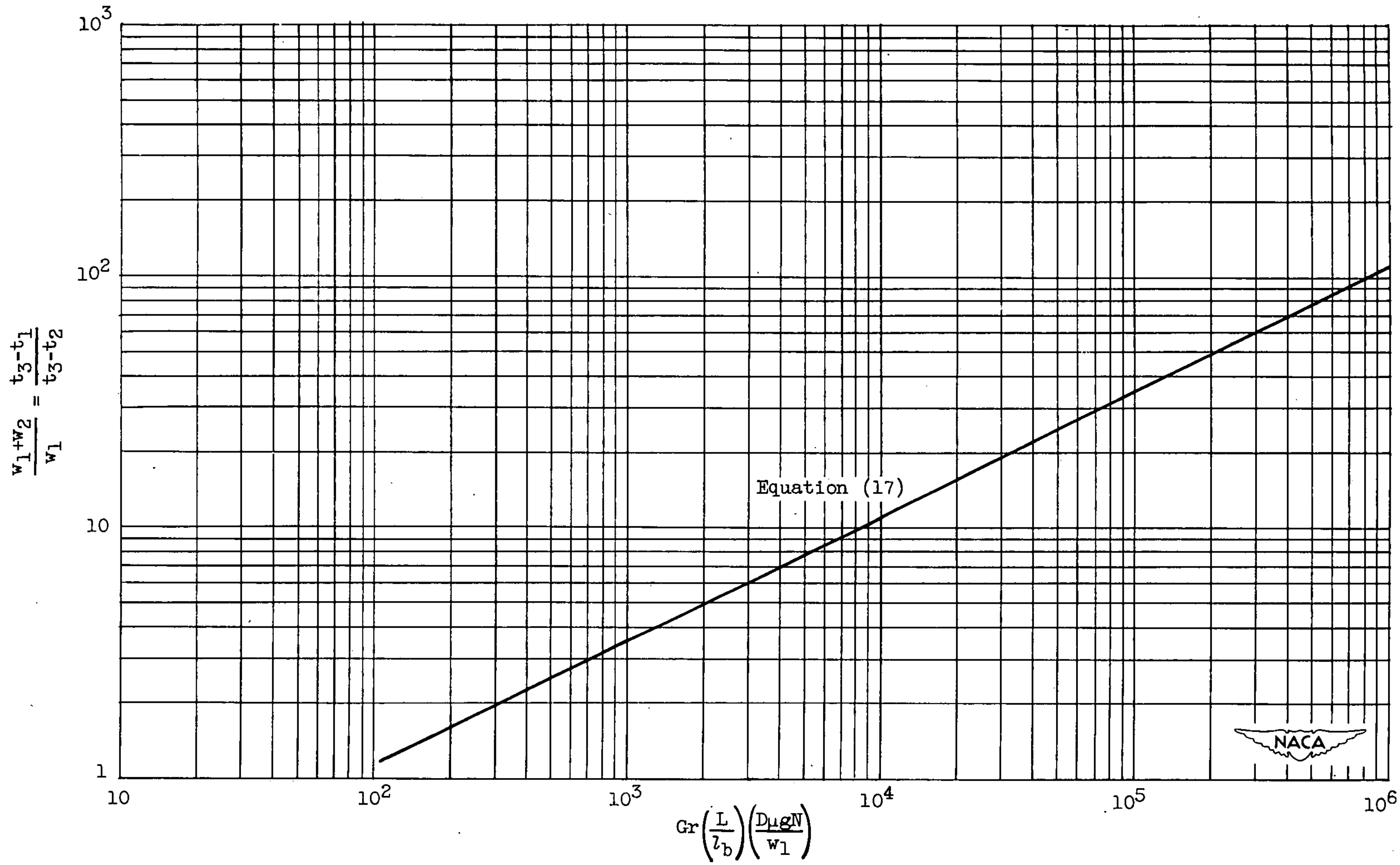
(b) Laminar flow.

Figure 5. - Concluded. Heat-transfer coefficients for straight-through-flow and loop cooling systems.



(a) Turbulent flow.

Figure 6. - Temperature and flow ratios in loop circuit.



(b) Laminar flow.

Figure 6. - Concluded. Temperature and flow ratios in loop circuit.

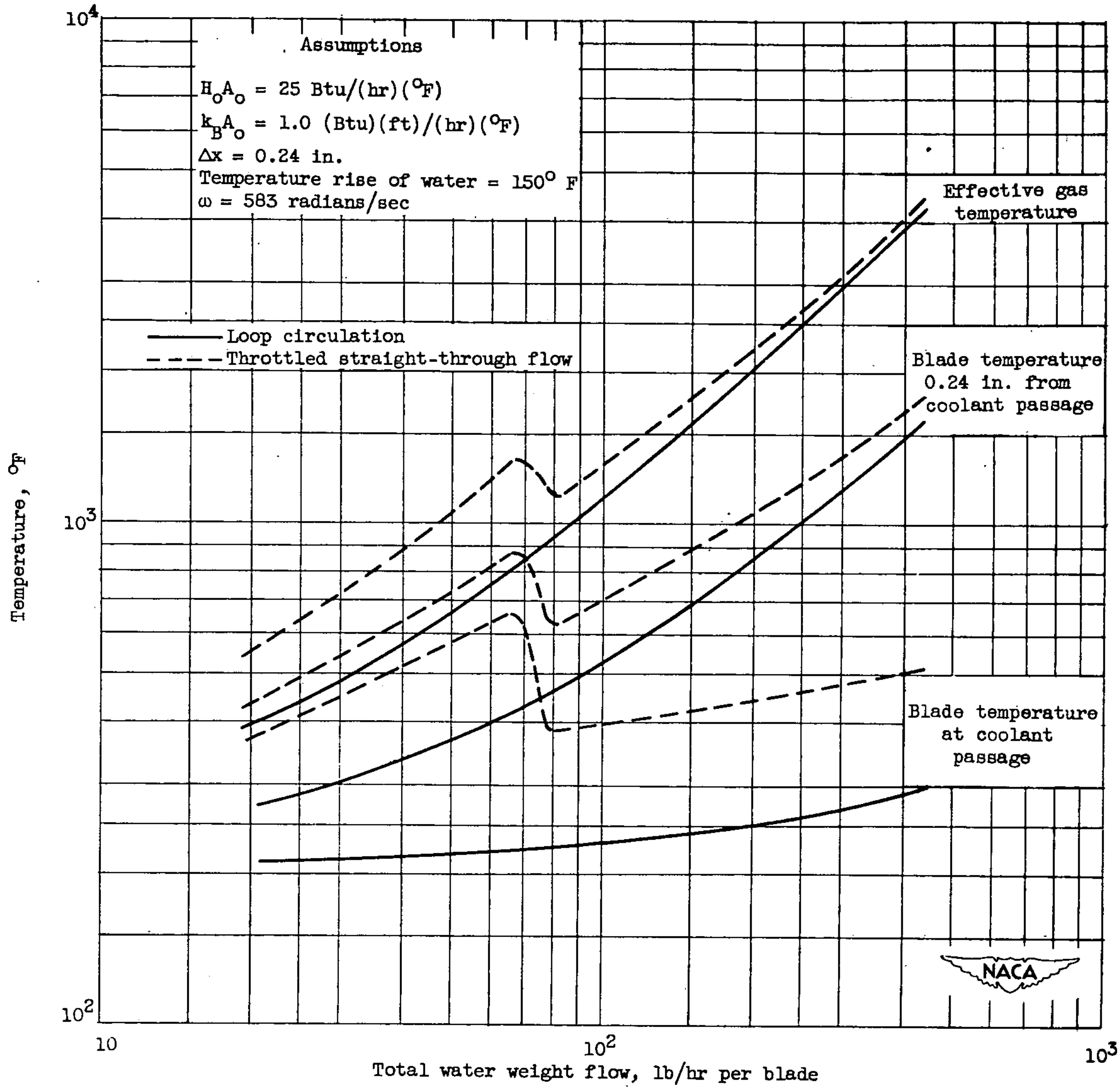
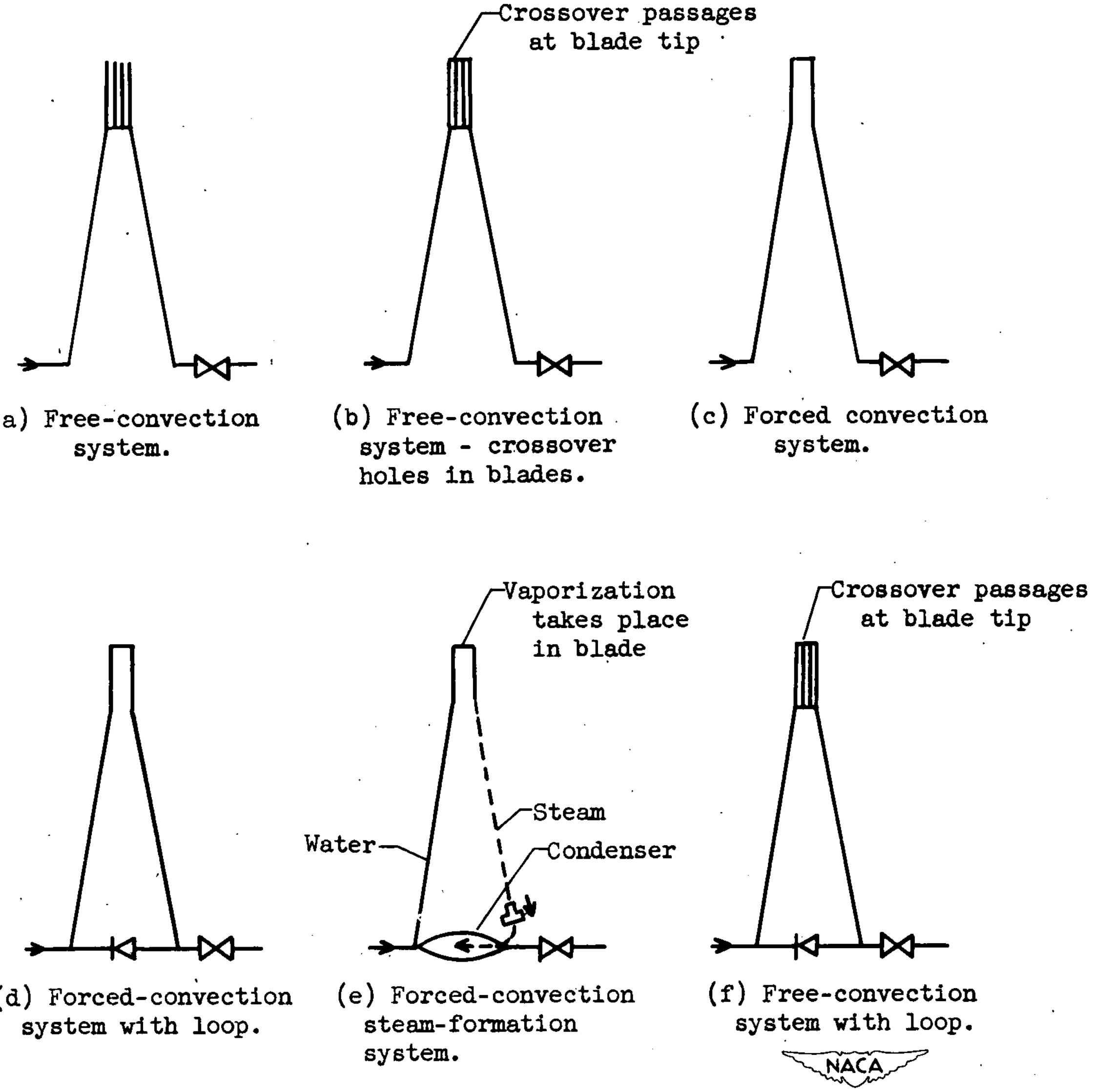


Figure 7. - Comparison of straight-through-flow and loop cooling systems.



(a) Free-convection system.

(b) Free-convection system - crossover holes in blades.

(c) Forced convection system.

(d) Forced-convection system with loop.

(e) Forced-convection steam-formation system.

(f) Free-convection system with loop.

Throttle valve or shut-off valve

Check valve (fluid flows only in direction of arrow)

Pressure relief valve (fluid flows only in direction of arrow)

Figure 8. - Possible one-liquid cooling systems.

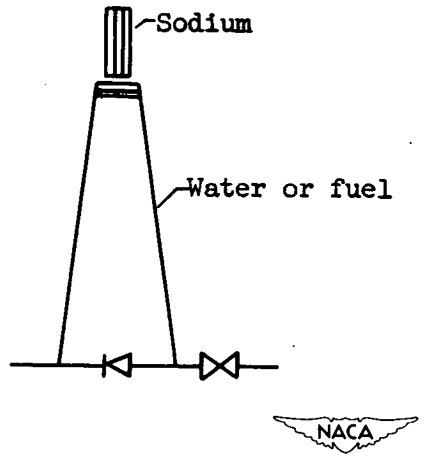


Figure 9. - Possible two-liquid cooling system.

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