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ANALYSIS OF A HEAT TRANSFER DEVICE FOR MEASURING FILM COEFFICIENTS

by R. A. Medrow
University of Missouri
Rolla, Missouri 65401

and

R. L. Johnson, W. R. Loomis, and L. D. Wedeven
Lewis Research Center
Cleveland, Ohio 44135
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ABSTRACT

A heat transfer device consisting of a heated rotating cylinder in a bath was analyzed for its effectiveness to determine heat transfer coefficient of fluids. A time dependent analysis shows that the performance is insensitive to the value of heat transfer coefficient with the given rig configuration.

INTRODUCTION

In the broadest sense, the motivation for this investigation lay in the effects of heat transfer upon lubrication. The specific area of concern was the observed lubricant behavior immediately downstream of a concentrated contact found for example between a ball and raceway of an aircraft engine bearing. Here, in the oil film wake, fluid separation and striations had been observed (1). This behavior has been suggested (2) as a possible cause of some bearing failures which appeared to be more related to improper cooling rather than lubrication. As suggested in (2) trapped gases between the surface and lubricant can prevent effective heat removal.

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*Associate Professor of Mechanical Engineering, Department of Mechanical and Aerospace Engineering, University of Missouri - Rolla, Rolla, Missouri 65401.

In addition, there are fluids, such as polyphenyl ethers, being considered for advanced lubrication applications that have poor wetting characteristics. It is speculated (1) that the apparent inadequate lubrication of these types of fluids may be the result of heat transfer deficiencies rather than poor lubrication properties.

The exploration of this subject could follow several paths. The availability of the apparatus described in the following section offered the possibility of an experimental investigation of the heat transfer and wetting characteristics of lubricating fluids. The objective of this work is to determine the suitability of this existing apparatus to this end.

TEST APPARATUS

Figure 1 shows a general view of the heat transfer apparatus available. It consists of a hollow stainless steel cylinder which rotates about a vertical axis. The top view of the cylinder is shown in Figure 2. The cylinder is heated at the top by means of an induction coil and cooled at the bottom by the test fluid in which it was partially immersed. The fluid is contained within a glass dewar. During operation, fluid temperatures were measured continuously at three locations within the fluid as shown in Figure 1. Although the cylinder rotational speed could be varied it was held constant at 1660 rpm during all experimental runs.

DEVELOPMENT OF EXPERIMENT AND ANALYSIS

This section is devoted to the means whereby the existing apparatus could best be used as a heat transfer device for determining the heat transfer coefficients of lubricating fluids.

Development of Specific Objectives

It is clear from Figure 1 that the data to be extracted from the system consists largely of temperature information from the three submerged thermocouples. The manner in which this data was to be used was derived from the experimental objective: to answer the question of the effect of wetting upon heat transfer in the system available.

The plan developed to accomplish this, proposed conducting tests with two classes of fluids: wetting and nonwetting. Values of the film, or heat transfer, coefficient were then to be obtained for each fluid tested through use of a model of rig performance.

Before it is possible to explain the use to be made of such data, it is necessary to consider some general facts concerning heat transfer coefficients.

A heat transfer coefficient is normally sought when the fluid behavior is too complex for analytical formulation. It is defined by the equation

$$h = \frac{\dot{Q}}{A\ \Delta T} \tag{1}$$

where

$\dot{Q}$ heat transfer rate from surface to fluid

A surface area

ΔT some characteristic temperature difference

It should be noted that the use of a heat transfer coefficient in a transient process is normally considered to be a questionable matter. However, it is frequently done. The comparatively slow rate of change of temperature involved in the present situation suggested that it would probably be a satisfactory step, and the results will be seen to support this conclusion.

Heat transfer coefficients are generally found experimentally. As in many other areas it is the development of empirical correlation equations arising from dimensional analysis which allows one to generalize from such experimental values.

For a fixed rig geometry operated under steady state or slow transient conditions, dimensional analysis leads to the result that Nu (the Nusselt number) should be a function only of Pr (the Prandtl number) and

Re_r (the rotational Reynolds number) if wetting <u>is</u> <u>not</u> a factor. Here,

$$Nu = \frac{hD}{k} \tag{2}$$

$$Pr = \frac{\mu Cp}{k} \tag{3}$$

$$Re_r = \frac{WD^2\rho}{2\mu} \tag{4}$$

where

h	heat transfer coefficient
D	specimen cup diameter
k	dynamic conductivity of the test fluid
μ	dynamic viscosity of the test fluid
Cp	specific heat of the test fluid
W	angular velocity of cup
ρ	density of the test liquid

If wetting is a factor of significance in the situation it follows that Nu cannot then be expressed as some experimentally-derived function of only Pr and Re_r. Thus, the objective developed was to determine whether or not a correlation valid for wetting fluids was also valid for nonwetting ones. If it were found to be valid one could conclude with some confidence that, in this one situation, wetting was not important.

Development of a Test Sequence

The prerequisites for determination of heat transfer coefficients are knowledge of A, ΔT and $\dot{Q}$. Of these, A is easily determined by direct observation. So too is the fluid temperature, which, along with the surface temperature, is used to determine ΔT. Thus, surface temperature and $\dot{Q}$ are seen to be the quantities which must be determined indirectly. The test sequence developed rested upon a desire to relate $\dot{Q}$ to the time rate of change of fluid temperature in as direct a fashion as was possible.

Taking as a control volume the region in which the test fluid was present as a liquid, the conservation of energy requires that

$$
\begin{aligned}
&\dot{Q} + (\text{rate of conduction heat transfer to the dewar}) \\
&\quad + (\text{rate of convective heat transfer to the air}) \\
&\quad - (\text{test fluid vapor enthalphy}) \times (\text{rate of mass evaporation of test fluid}) \\
&\quad + (\text{rate at which cylinder does work on liquid}) = (C_p d/dt)(mT_f) \qquad (5)
\end{aligned}
$$

where m is the mass of the test fluid and T_f is the temperature of the fluid. The equation assumes that the entire process proceeds slowly enough so that the fluid temperature is essentially uniform throughout the fluid at each instant in time.

Experiments conducted some years ago with a similar apparatus were run until, from the fluid point of view, steady state conditions were obtained. Normally such conditions make the interpretation of experimental results a simpler process. During the present investigation, however, such a course was impossible for two reasons. $\dot{Q}$ would then depend solely upon the values of the four written-out terms in equation (5), only the last of which can be estimated with any accuracy. In addition, the first results of the investigation verified the initial guess that specimen cup temperatures above fluid level might well reach values at which lubricant combustion could occur.

It was decided, therefore, to conduct experiments so as to cause the term on the right hand side of equation (5) to be the dominant one, while also keeping the changes in fluid temperature small. One consequence of the last consideration was to make possible reasonable estimates of the contribution of the written-out terms of equation (5).

The sequence of operation followed is shown in Figure 3. With the cylinder rotating and $\dot{Q} = 0$ (power off) a small rate of increase in temperature was observed. Since tests were commenced with the test liquid at room temperature (T_o), conduction and convection losses should be negligible, as should the enthalphy flux due to evaporation. Thus, the initial rate of temperature increase is very closely related to the work rate. The work rate thus obtained was assumed to remain constant throughout the entire test.

After a period during which power was applied to the induction coil, rotation was continued. During this period the fluid temperature passed through a maximum and started to fall. With $\dot{Q}$ equal to zero in this interval the rate of temperature decrease is directly proportional to the sum of the written-out terms in previous equation (5). The rate of energy transfer due to conduction, convection, and evaporation at the final fluid temperatures, T_{max}, is then directly proportional to the sum of the magnitude of the two temperature slopes shown on Figure 3. During the entire test this rate of energy transfer was assumed to be equal to the rate at T_{max} times the quantity $(T - T_p)/(T_{max} - T_p)$ where T is the instantaneous fluid temperature and T_p is the fluid temperature at the time power is applied to the induction coil.

Development of an Analytical Model of Rig Performance

Initially, the two lumped systems shown in Figure 4 were analyzed analytically. Details of the systems are shown on Figure 4. In both cases the temperature of each "lump" was assumed to be uniform throughout at each instant in time. Neither system was considered to be a highly accurate model of the actual system. Each, however, contributed information useful in justifying the development of a more elaborate model along certain lines.

Of greatest significance was the fact that the assumption of a time-independent heat transfer coefficient, used in both models, produced results in agreement with the experimental findings of the earlier experiments. To determine a heat transfer coefficient in a system like this it is necessary to adjust the values of $\dot{Q}$ and h used in the model until the model's predicted behavior agrees with that actually found. These simple models predicted the existence of a nearly linear portion of the temperature-time curve, the slope of which is defined below and indicated on Figure 5(b).

$$SLOPE = \frac{\dot{Q}}{m_c C_c + m_f C_f} \tag{6}$$

where

m_c mass of cylinder

m_f mass of fluid

C_c specific heat of cylinder

C_f specific heat of fluid

This slope is predicted to be directly proportional to $\dot{Q}$, as is the maximum temperature, T_{max}, reached by the fluid, also shown on Figure 5(b). The first of the current tests, with water, yielded agreement to within a few percent between values of $\dot{Q}$ obtained by each of the measurements of actual performance suggested by the analytical behavior shown on Figure 5.

With this encouragement a more accurate, numerically-solved, model of the cylinder was formulated and coded. This model was no more than elaboration of the second analytical model, in that it allowed the specimen cup to be divided into more than two segments. The performance of this model, in comparison with the actual experimental results for water, immediately illustrated some shortcomings of certain assumption used to this point.

Specifically, the predicted temperature rise within the fluid lagged considerably behind that actually observed. The $\dot{Q}$ values used in the model were those obtained by measurement of both slope and maximum temperatures. This lag was found for all values of the heat transfer coefficient used, including some well above any to be anticipated in a non-boiling situation.

What now appears to be the cause of this lag was found by examining the temperatures predicted for the portion of the cup within the induction coil. That the coil itself remains quite cool during operation was clear from the formation of condensed moisture on the coil. For this to occur the coil must operate at a temperature such that it radiates a negligible amount of energy compared to that emitted by the upper portion of the cup. Estimates of this latter rate were made using the results of the first numerical model. They showed that, at the end of the heating portion of the test, the rate of radiant heat transfer to the coil from the cup might be as much as 40% of the rate of heat generation induced in the cup by the coil. Thus, the question of a time-dependent net $\dot{Q}$ had to be considered. In order to obtain an immediate look at the possible significance of radiant heat transfer upon the fluid temperature, the original program was modified to include this effect. The method whereby this was accomplished is identical with that described later in connection with the discussion of the final model. Figure 6 demonstrates quite clearly the impact of radiative transfer. The input parameters $\dot{Q}$, h, fluid properties, etc., are the same for both curves, except for the factor of radiation.

The high temperatures in the cup also suggested that conduction into the cup support shaft might be important enough to force consideration of this possibility.

The considerable range in cup temperature during the course of a test further suggested the necessity of exploring the question of a time-dependent electrical coupling between the coil and the cup. With the aid of reference (3) it was determined that the heat generation rate in the cup might be on the order of 25% greater at the end of the heating period than it was at the beginning. Because of this, the equations given in reference (3) were used to develop a simplified analytical relationship between coupling efficiency (η) and the average cup temperature in the heated section. Figure 7(a) shows the general character of this relationship. Figure 7(b) shows how the effective $\dot{Q}$ increased with respect to the initial value ($\dot{Q}_o$) in one of the actual tests as a result of this temperature dependent coupling efficiency.

The large temperature difference across the small angular region between cup and coil pointed to the advisability of considering convective as well as radiant heat transfer. This was accomplished by means of the results for a fairly similar situation reported in reference (4).

The final configuration of the cup used in all subsequent calculations is shown in Figure 8. As in the previous model the thin cylindrical shell portion of the cup is subdivided into a number of rings. From 12 to 96 such subdivisions were used. Because of the thinness of the shell, temperature in the cup was assumed to be a function only of time and the axial location, independent of radius.

Because of its large mass and radial temperature variation, inclusion of the support shaft into the calculations posed several problems. It was decided to carry out the calculation in two ways with the intention of bounding the effect of conduction between cup and shaft. In the first way the inner surfaces of the upper cup elements were assumed to be adiabatic, an assumption equivalent to saying that there was no conduction between cup and shaft. In the second series of calculation two additional elements were introduced. The first of these is the ring-shaped element with a square cross-sectional area shown at the top of the cup in Figure 8. The second is composed of the four bars (see Fig. 2) which connect the cup to the shaft. When these elements were included, the root (inner) temperature of the bars was held constant at T_i. The effect of this was to force the shaft to act as a sink; while energy would be conducted into the shaft whenever the bar temperature was above T_i, none of this energy would ever be conducted back into the bar during the period after power shut-down. These two methods of handling the problem of the shaft are thus seen to be bounds upon the actual participation of the shaft in the dynamic performance of the system.

The equation governing the performance of the various systems (cup elements and test fluid) were obtained by writing expressions for the conservation of energy for each type of element.

General cylinder element:

(rate of heat conduction in) - (rate of heat conduction out)

+ (net rate of radiative heat transfer in)

+ (rate of convective heat transfer in)

+ (heat generation rate due to induction coil) = $m_c C_c \, dT/dt$ (7)

In an explicit form this becomes, for the i^{th} element,

$$k_{i,i-1} A_{cd}\left(\frac{T_{i-1} - T_i}{\Delta x_{i,i-1}}\right) - k_{i,i+1} A_{cd}\left(\frac{T_i - T_{i+1}}{\Delta x_{i,i+1}}\right)$$

$$+ \epsilon_i A_r \left(T_{rs}^4 - T_i^4\right) + hA_{cv}(T_{cs} - T_i) + \dot{Q} = m_i C_i\left(\frac{T_i' - T_i}{\Delta t}\right) \quad (8)$$

where

$k_{i,j}$	thermal conductivity at $T = 0.5\,(T_i + T_j)$
A_{cd}	area available for conduction between elements
A_r	element surface area seen by coil
A_{cv}	element surface area available for convective heat transfer
$\Delta x_{i,j}$	distance from center of i^{th} element to center of j^{th} element
ϵ_i	total emissivity of element surface
h	appropriate heat transfer coefficient
m_i	mass of i^{th} element
C_i	heat capacity of i^{th} element
Δt	size of time step

T_i	temperature of i^{th} element at time t
T_i'	temperature of i^{th} element at time $t + \Delta t$
T_{rs}	coil surface temperature at time t
T_{cs}	temperature appropriate to the convective situation at time t

Fluid element:

(rate of convective transfer in)

- (rate of energy loss to surroundings)

+ (rate at which cylinder does work on fluid) = $m_f C_f dT_f/dt$ (9)

In an explicit form this becomes

$$hA_{cv} \sum_{se} (T_i - T_f) - \dot{Q}_L + \dot{W} = m_f C_f \left(\frac{T_f' - T_f}{\Delta t} \right) \qquad (10)$$

where the additional variables are

T_f	fluid temperature at time t
T_f'	fluid temperature at time $t + \Delta t$
$\dot{Q}_L$	sum of first three written-out terms in equation (5)
m_f	mass of fluid
C_f	specific heat of fluid
$\sum_{se}$	denotes summation over all submerged elements (see Fig. 8)

When the equations are written in an explicit manner each element temperature at time $t + \Delta t$ is determined entirely by temperatures at time t.

The outer cup elements were divided into three groups: heated, air gap and submerged (see Fig. 8). The left-handed side of the general element equation contains five terms. All of these terms had nonzero values only for elements in the heated length. There, h corresponded to a value obtained from reference (4) and T_{cs} was the coil temperature. In the other two groups both ϵ_i and $\dot{Q}$ were zero. In the air gap group h was also zero, while in the submerged group h was the current trial value for the heat transfer coefficient being sought. In this latter group of elements T_{cs} was T_f.

The term used to represent the radiative heat transfer is appropriate for the interchange between the outer surface of the cup and the inner surface of the coil, given the geometries involved and the values of the radiative properties of the surfaces. No account was taken of the interchange between cup elements via the mutual visibility of their inner surfaces. The practical effect of having done so would have been equivalent to the introduction of a $\dot{Q}$ dependent upon axial position within the coil, a refinement deemed unnecessary.

RESULTS AND CONCLUSIONS

The first portion of this section is devoted to establishing the extent to which the analysis developed in the previous section agrees with the experimental observations. The second portion explores the ramifications of this agreement for the present rig.

Comparison of Theory and Results

Figures 9 through 12 demonstrate the agreement obtained between theory and experiment for the four tests conducted. Here, as in the following Figures, all temperatures plotted are departures from T_p. The omitted test numbers (1, 4, and 6) were used to designate the no-power tests used to obtain values for the work rate and heat loss terms. For reasons to be explained later the value used for h was, in all cases, 1500 Btu/hr ft^2. With this value fixed, $\dot{Q}_o$ was adjusted until satisfactory agreement had been reached.

The last two tests (5 and 7) were conducted with identical settings on the induction heating unit using respectively a synthetic paraffinic and polyphenyl ether oil. It is gratifying to note that the values of $\dot{Q}_o$, determined by an automatic search procedure purely on the basis of the fit between theory and experiment, were quite similar. For test 5, the analytical results are those for $\dot{Q}_o = 10.2976$ Btu/min, while those for test 7 are for $\dot{Q}_o = 10.6841$ Btu/min.

Figures 13 and 14 show some interesting analytical results concerning temperature within the specimen cup. The first of these shows quite clearly the extreme variation in cup temperature at the end of the power-on portion of test 2. Such behavior is typical of that found in all tests. Figure 14 shows, for the same test, the temperature histories of two heated elements, or nodes.

Based upon these results, it is reasonable to conclude that the analysis is of satisfactory accuracy in that it is capable of correctly predicting the fluid temperature history. All physical properties used were taken from standard sources, and were not adjusted to enhance the extent of the analytical/experimental agreement.

Consequences and Conclusions Concerning Rig

Despite the extent of the agreement between theory and experiment, no conclusions of value can be reached concerning the quantity of interest, h. This follows from the fact that the predicted temperature history of the fluid is most insensitive to the value of h used.

Figure 15 demonstrates this quite clearly. It contains the results of six different computer runs made after the one used in Figure 9. In each of these six runs one of the conditions or input parameters used to obtain the results of Figure 9 was changed. Each such change led to a different fluid temperature history. Because these histories were all quite similar, the results plotted in Figure 15 show the differences, as functions of time, between the results of each of these six additional runs and the "basic" run. The "basic" run is the one plotted on Figure 9, the conditions and parameters of which are shown on Figure 15. As this Figure shows, an h value of 500 Btu/hr ft^2 (one-third of the "basic" value) produces less change in the results than does the estimated contribution of convection between the cup and the induction coil. A comparable statement can be made concerning the relative impacts of an increase in ϵ from 0.7 to 0.8 and this two-thirds reduction in h. The predicted history is also seen to be more sensitive to the extent of conduction losses into the shaft or a one-eighth inch change in the air gap value than it is to a reduction in h from 1500 to 150 Btu/hr ft^2.

Having established that the predicted rig performance is most insensitive to the value of h, it remains to demonstrate why this is the case. Figure 16 supplies the answer. Here we see plotted the difference between the cylinder temperature, T_c, and T_p as a function of submerged length only for three different h values. These results are for test 2 just at the end of the power-on period. As the curves and the table on Figure 16 demonstrate, the lower the value of h used, the greater the value of $\overline{\Delta T}$. Here, $\overline{\Delta T}$ is the average temperature of the submerged portion of the cup minus the fluid temperature. The rate of change in fluid temperature is very nearly directly proportional to $\mathring{Q}$ to the fluid, which in turn is equal to $Ah\,\overline{\Delta T}$. Thus, the fluid response is dependent upon the product of h and $\overline{\Delta T}$. The last entry in the Table on Figure 16 shows that this product is nearly independent of the value of h.

If the average temperature of the cup was known this fact would pose no problem. Unfortunately, for all three h's, the submerged portion temperatures are too low to allow use of a pyrometer, the only temperature measuring technique possible without a complete redesign of the apparatus.

As a matter of interest, one general observation concerning cup temperature should be made. Based upon the analytical calculations, as much as 92% of the submerged portion of the cup might be at a temperature below that of the fluid. Thus, a major part of this portion of the cup would be heated by the fluid and not by conduction down through the cup.

The data obtained and the analysis performed lead to the conclusion that further investigation involving the present rig are quite likely to be of little value. In the interest of seeing whether or not any simple qualitative differences could be found among the fluids tested, one additional series of calculations, shown in Figure 17, was made. Here, all of the test results obtained have been plotted dimensionlessly. The time used to make the horizontal axis dimensionless is the length of the power-on interval, Δt. The reference temperature used for the other axis is the maximum temperature predicted for constant $\dot{Q}$ by the simple analytical models first considered. Such differences as appear are not considered significant.

REFERENCES

1. Johnson, R. L.: Discussion. Interdisciplinary Approach to Friction and Wear. NASA SP-181, 1968, pp. 410-416.
2. Fisher, J.: Cooling Failures Often Give Lubes a Bad Reputation. Metal Working News, Jan. 27, 1969, p. 19.
3. Simpson, P. G.: Induction Heating Coil and System Design. McGraw Hill, 1960.
4. Bjorkland, I. S.; and Kays, W. M.: Heat Transfer Between Concentric Rotating Cylinders. J. of Heat Transfer, vol. 81, Series B, No. 3, Aug. 1959, pp. 175-186.

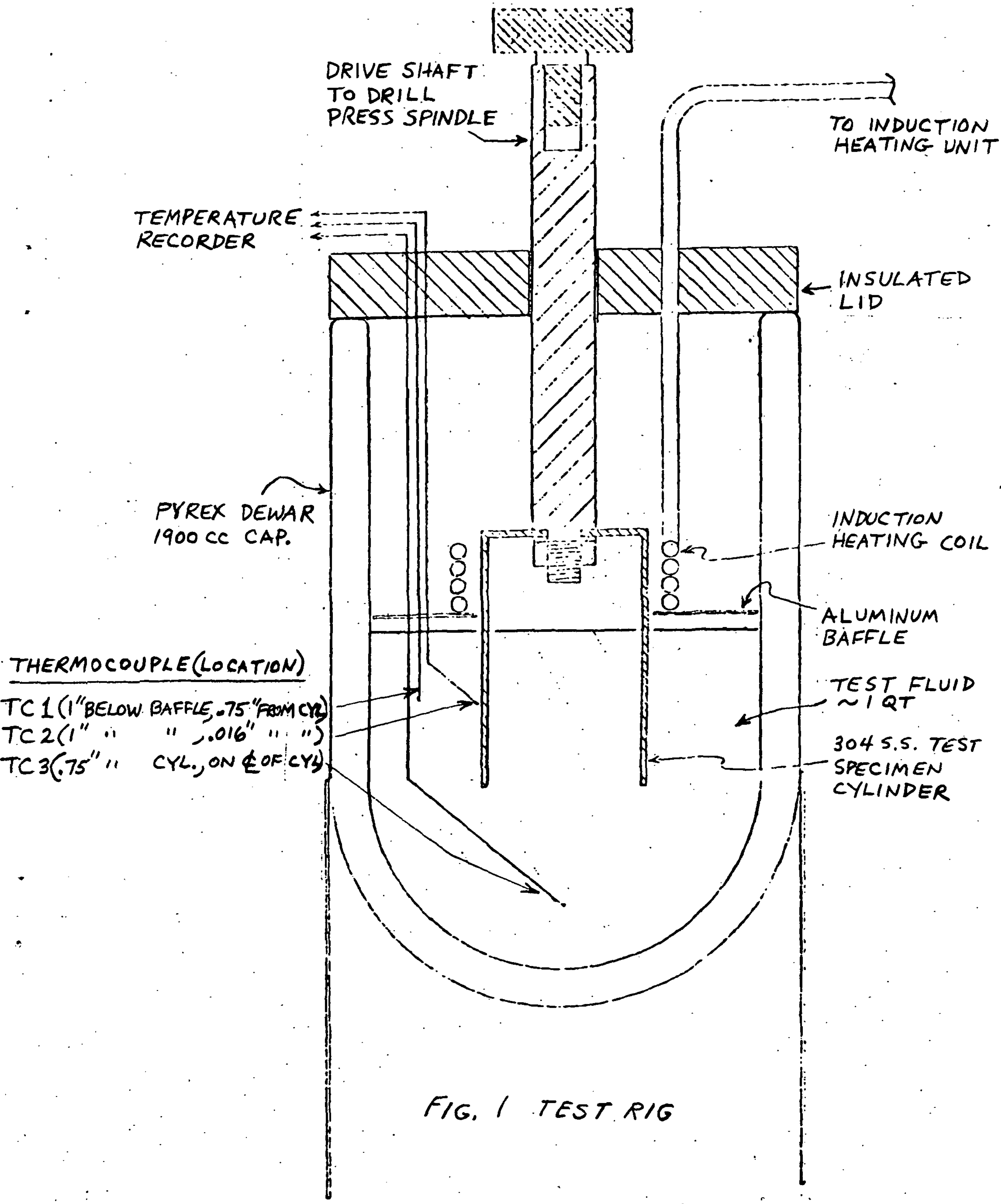


FIG. 1 TEST RIG

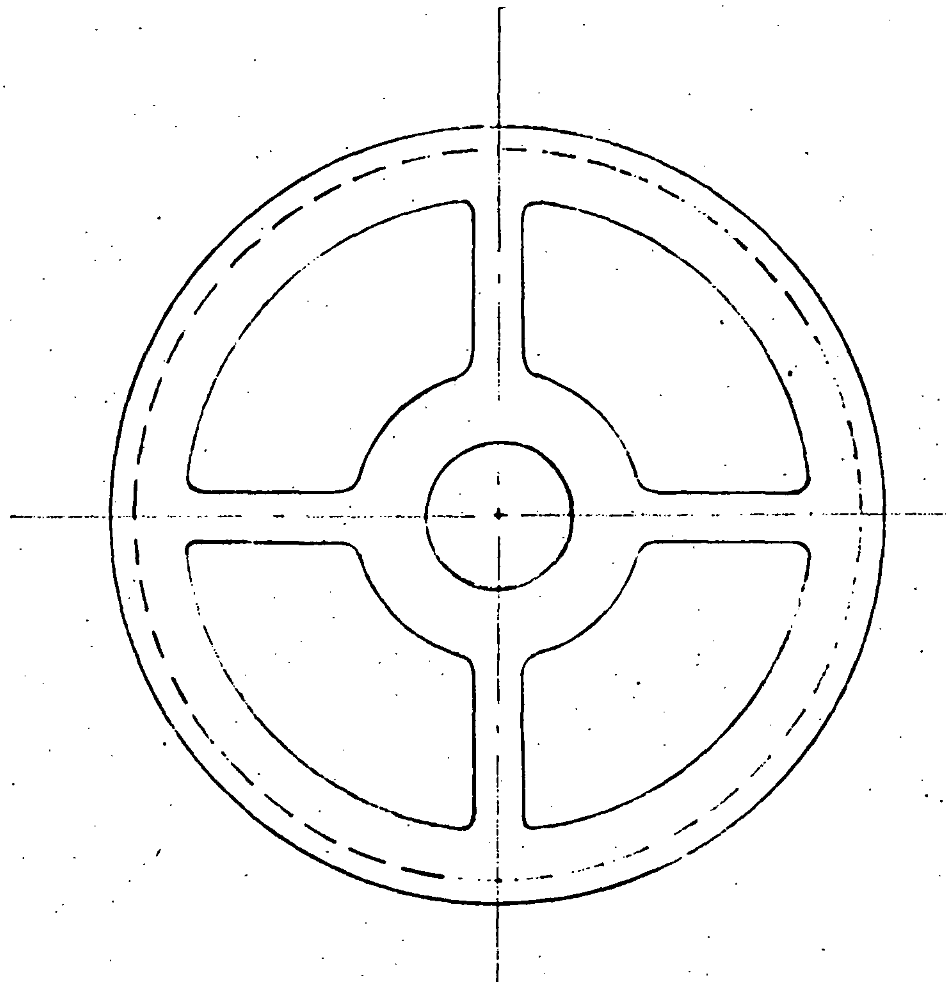


FIG. 2 TOP VIEW OF SPECIMEN CYLINDER

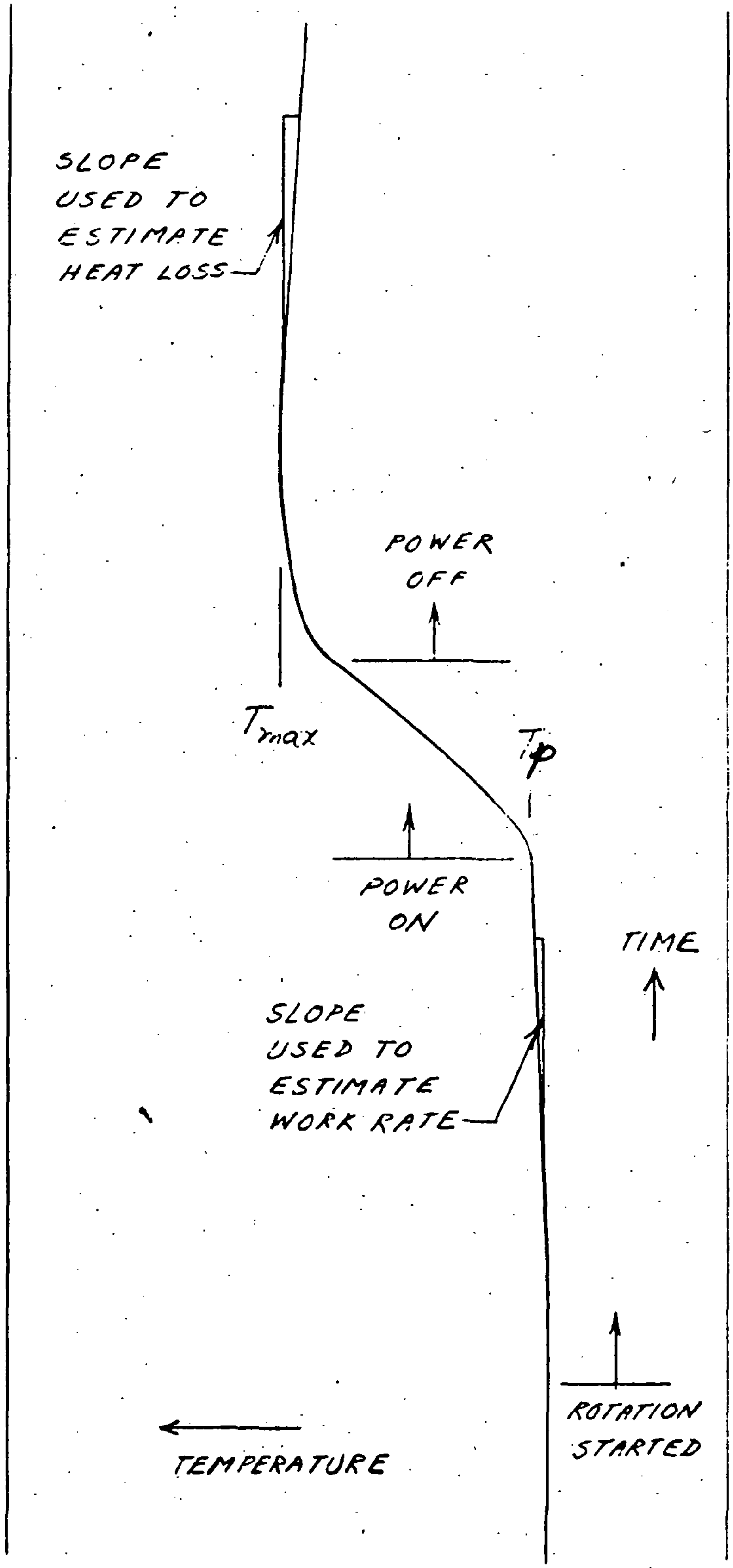


FIG. 3 TEST PROFILE

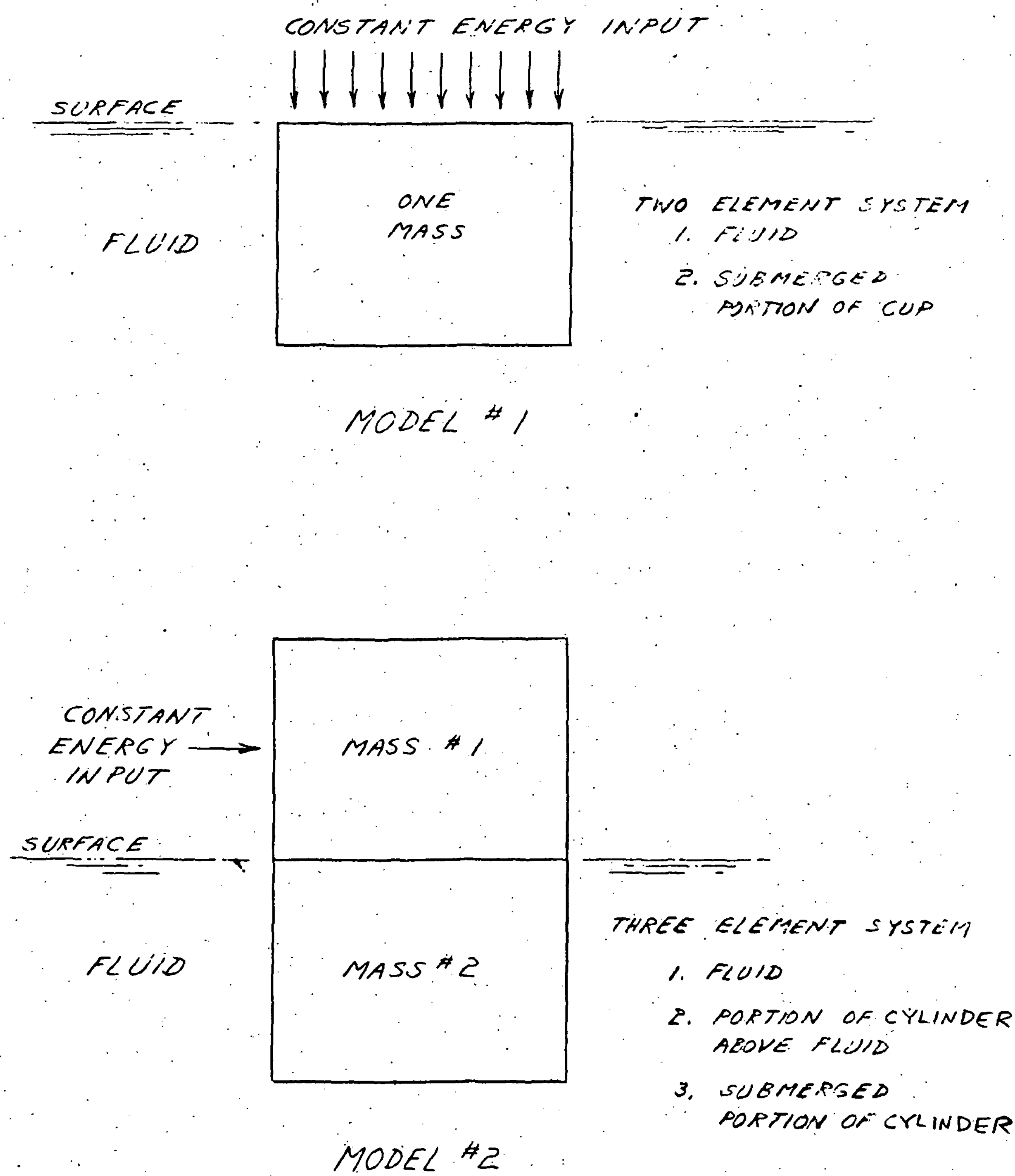
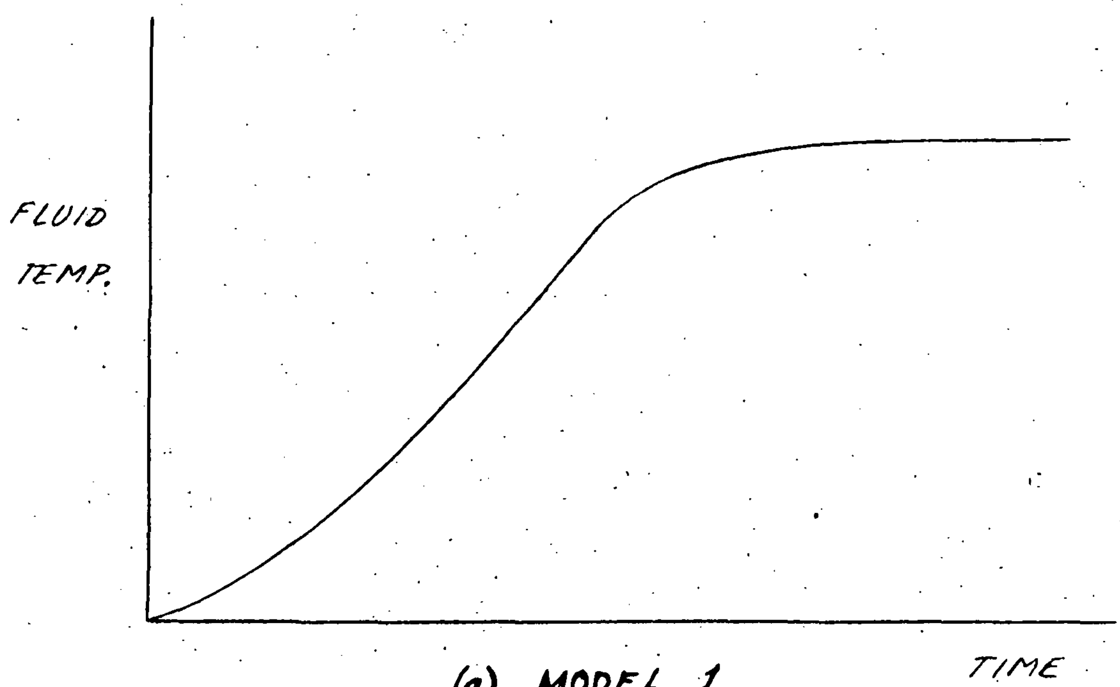
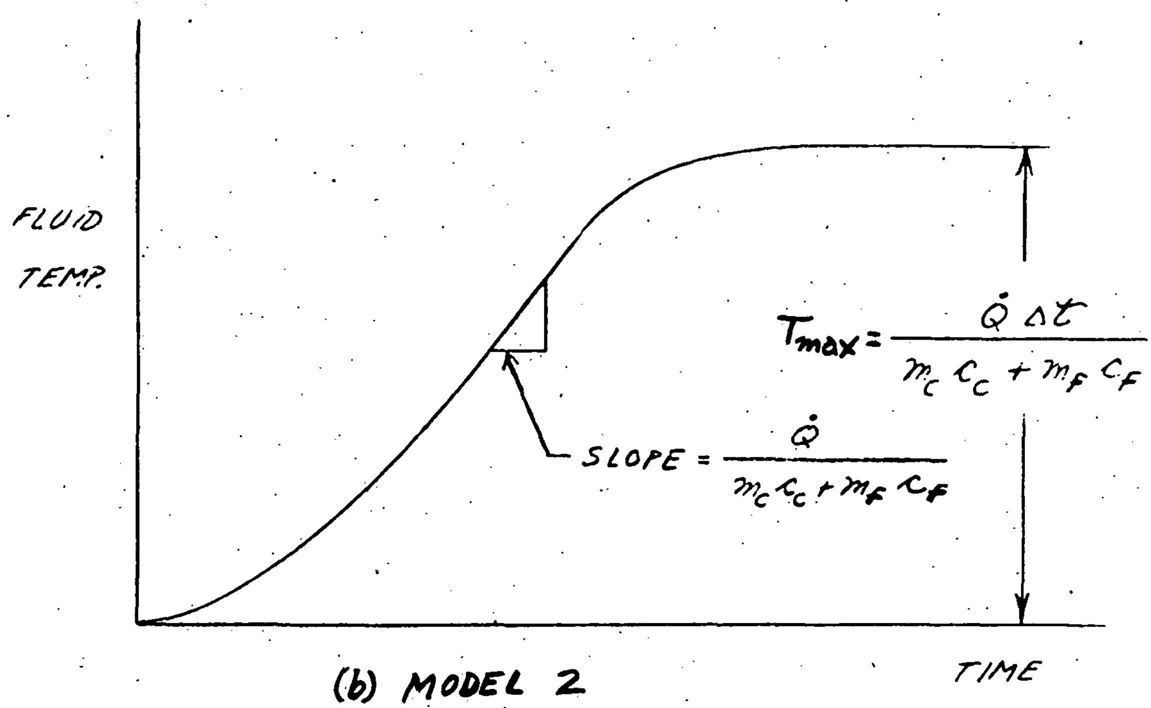


FIG. 4 MODELS 1 AND 2



(a) MODEL 1



(b) MODEL 2

FIG. 5 RESULTS OF MODELS 1 AND 2

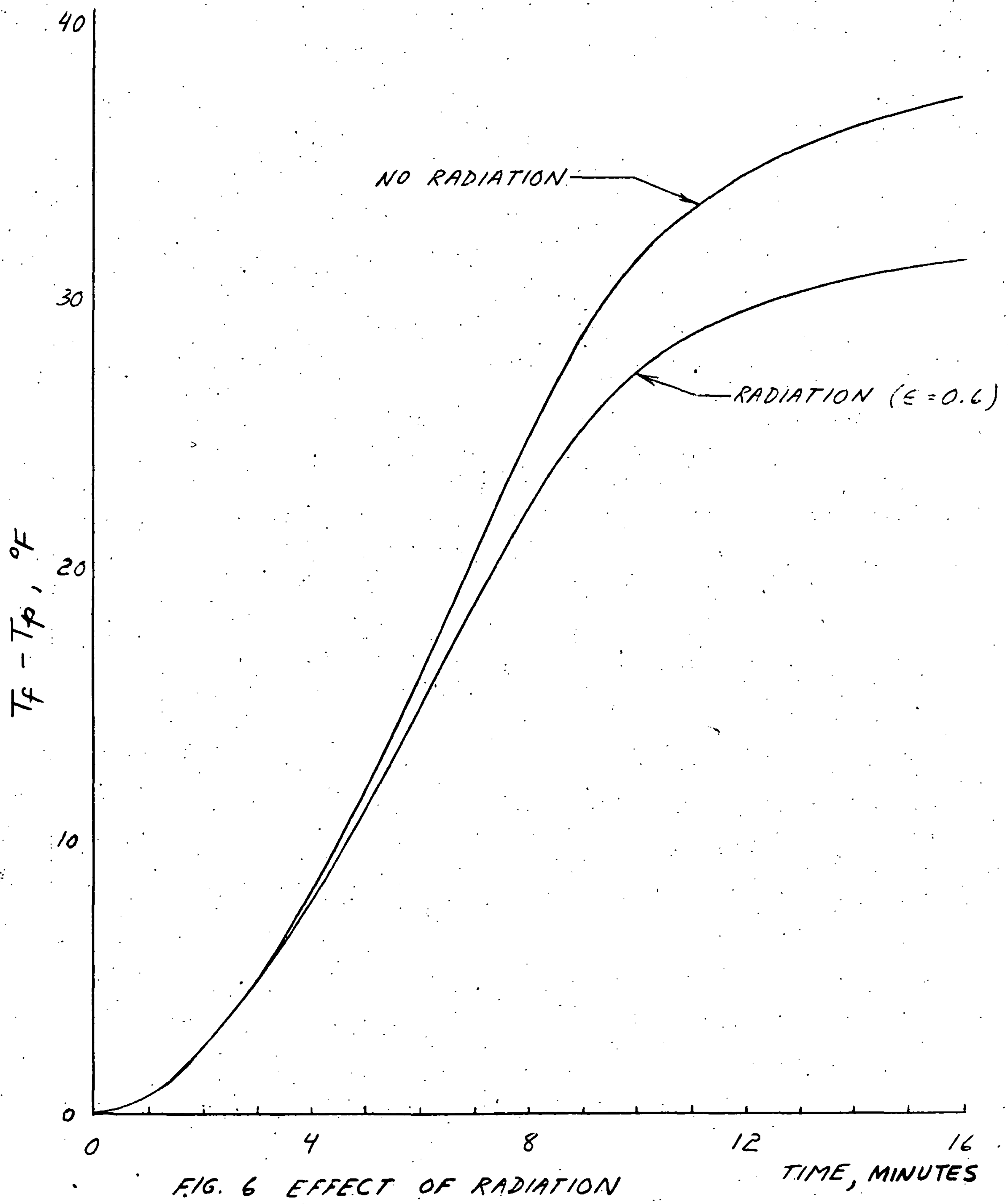


FIG. 6 EFFECT OF RADIATION

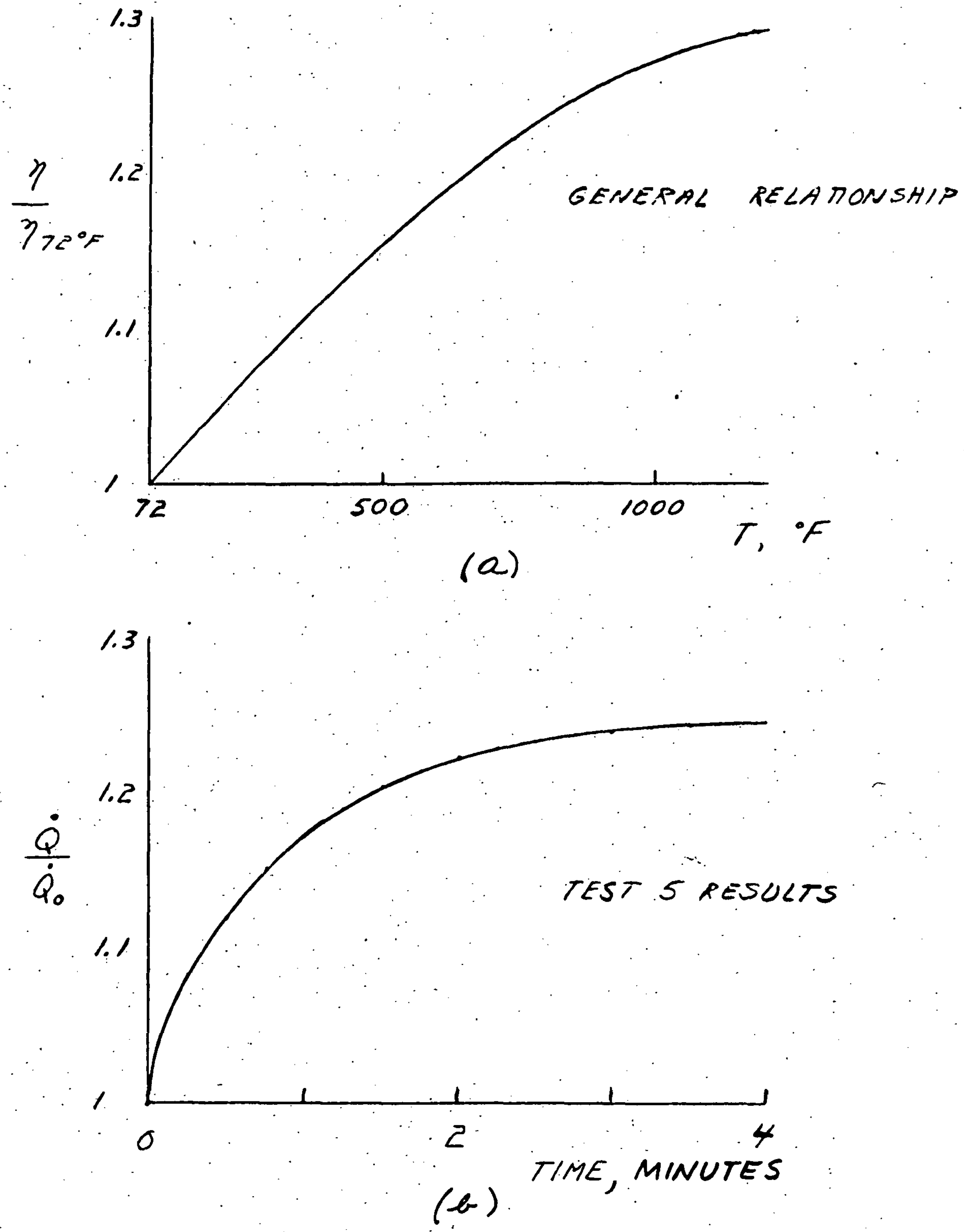


FIG. 7 COUPLING EFFICIENCY

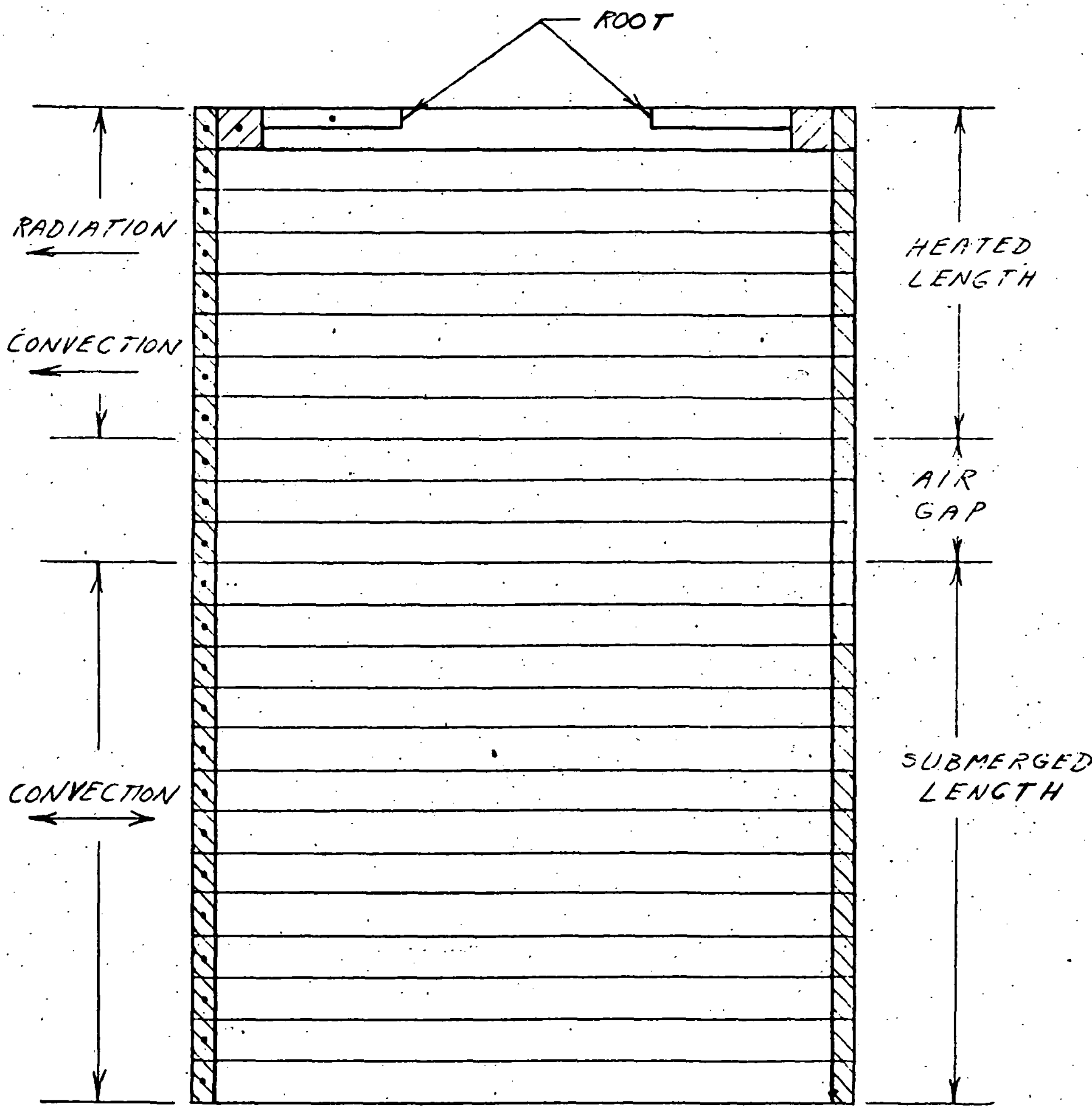


FIG. 8 SKETCH OF ANALYTICAL MODEL

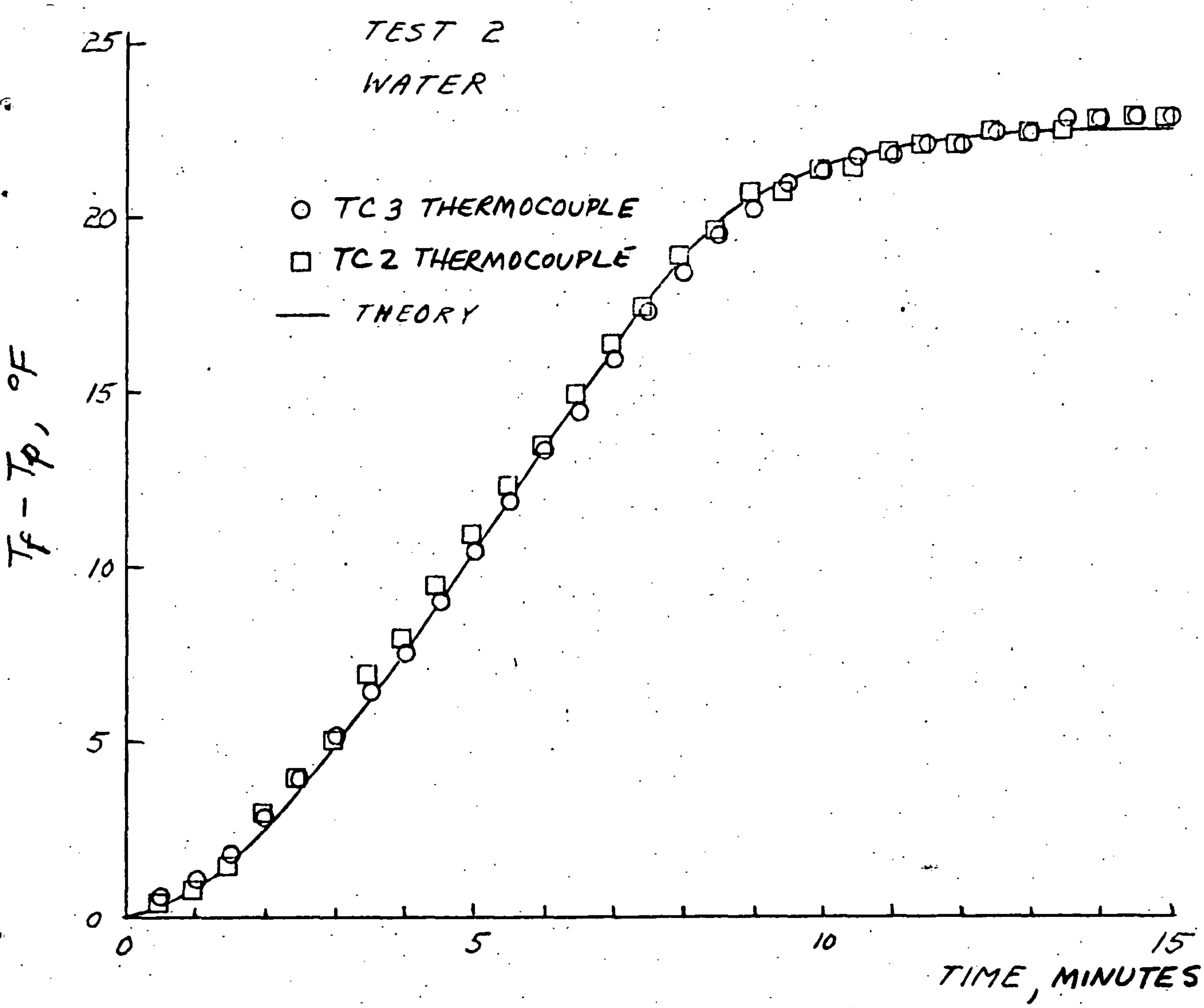


FIG. 9 RESULTS, TEST 2

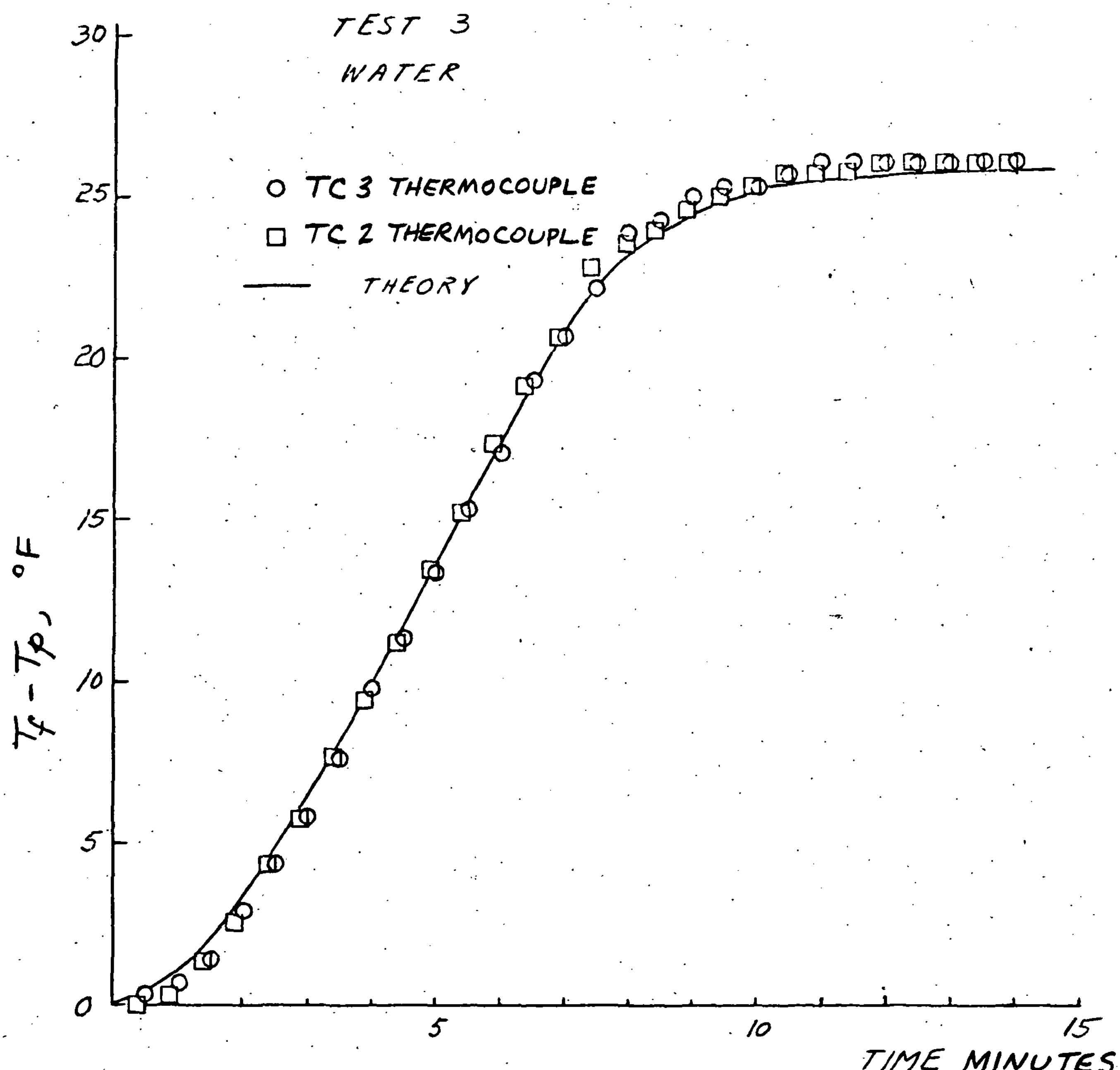


FIG. 10 RESULTS, TEST 3

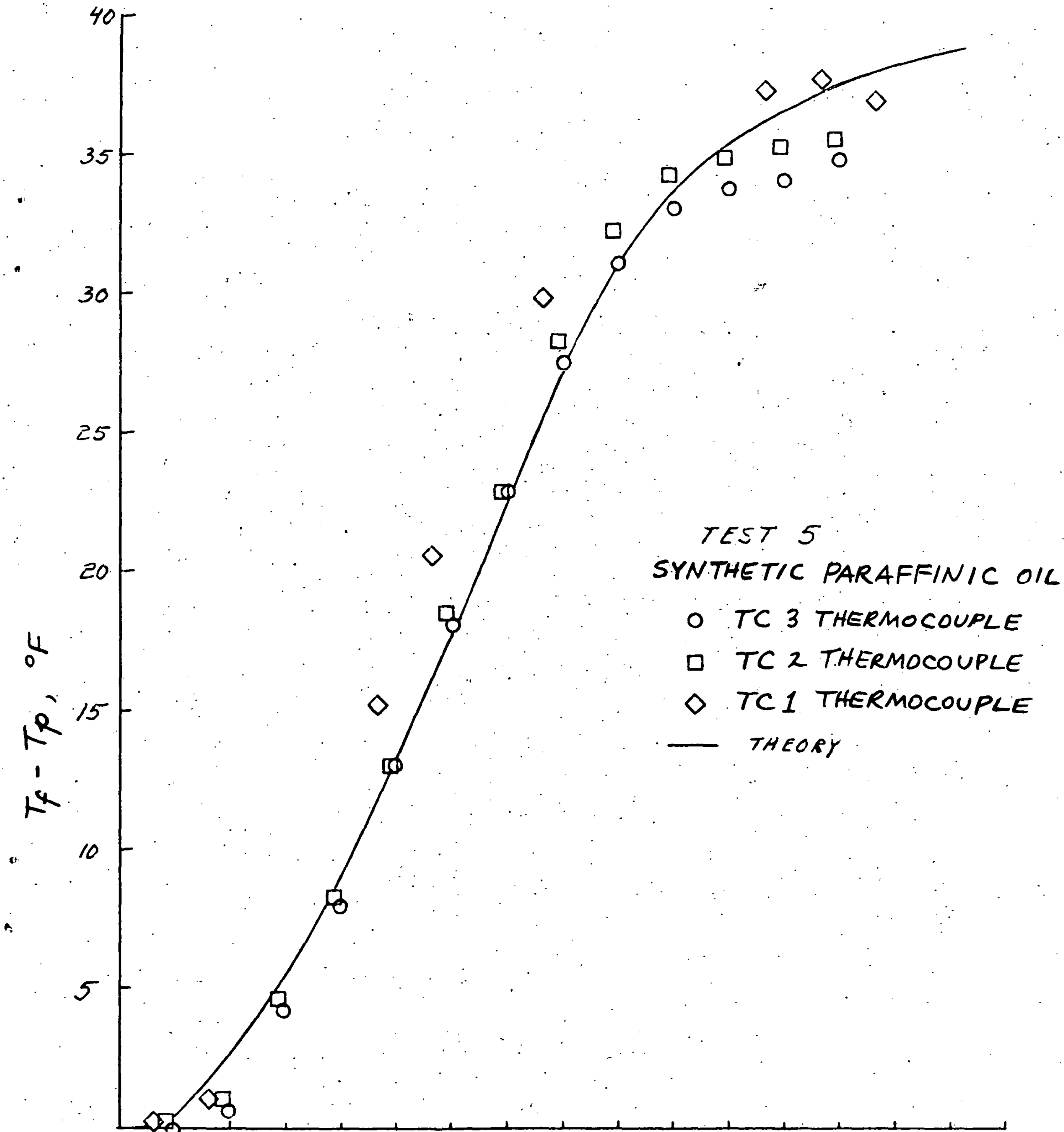


FIG. 11 RESULTS, TEST 5

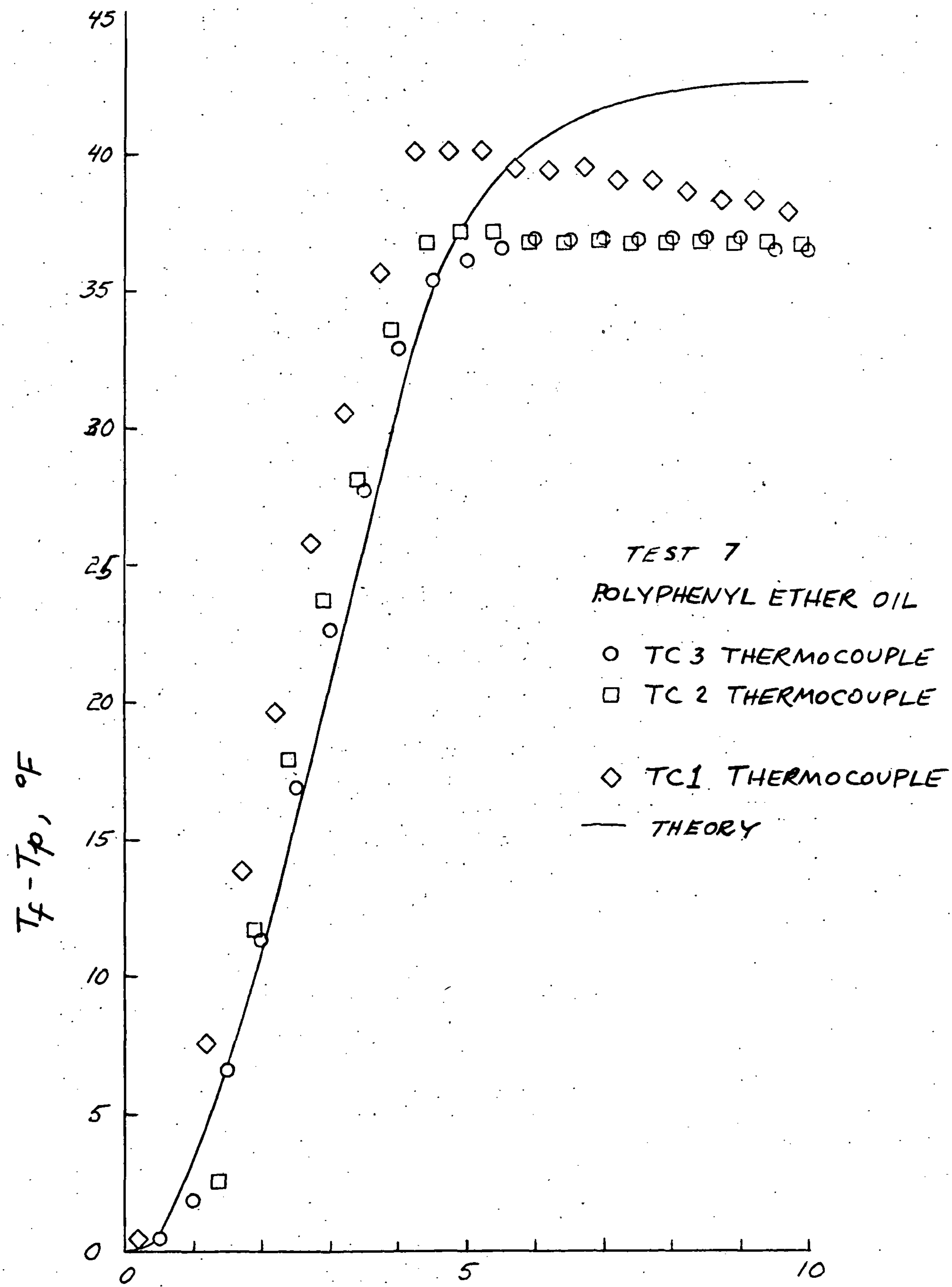


FIG. 12 RESULTS, TEST 7

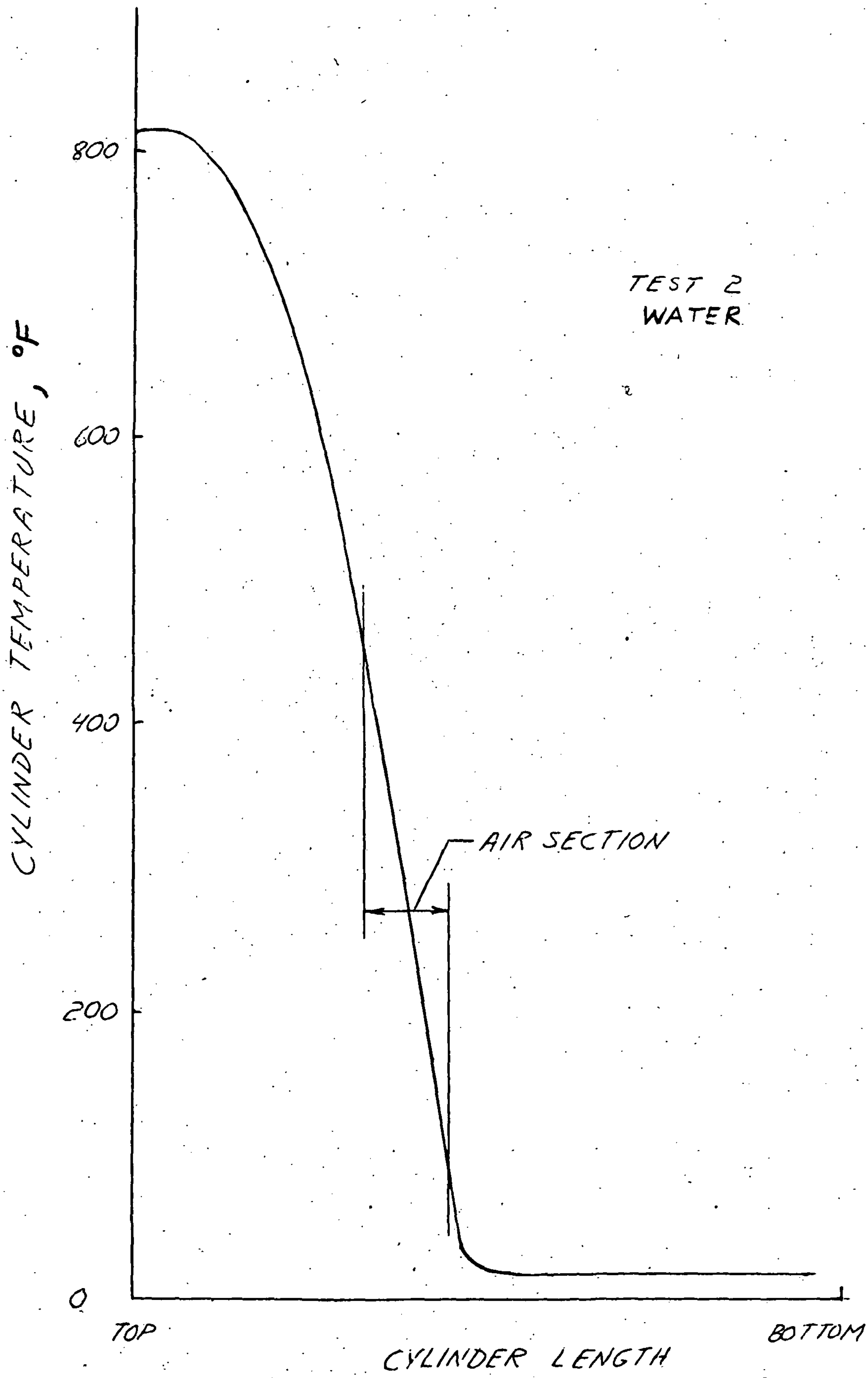


FIG. 13 TYPICAL CUP TEMPERATURE PROFILE

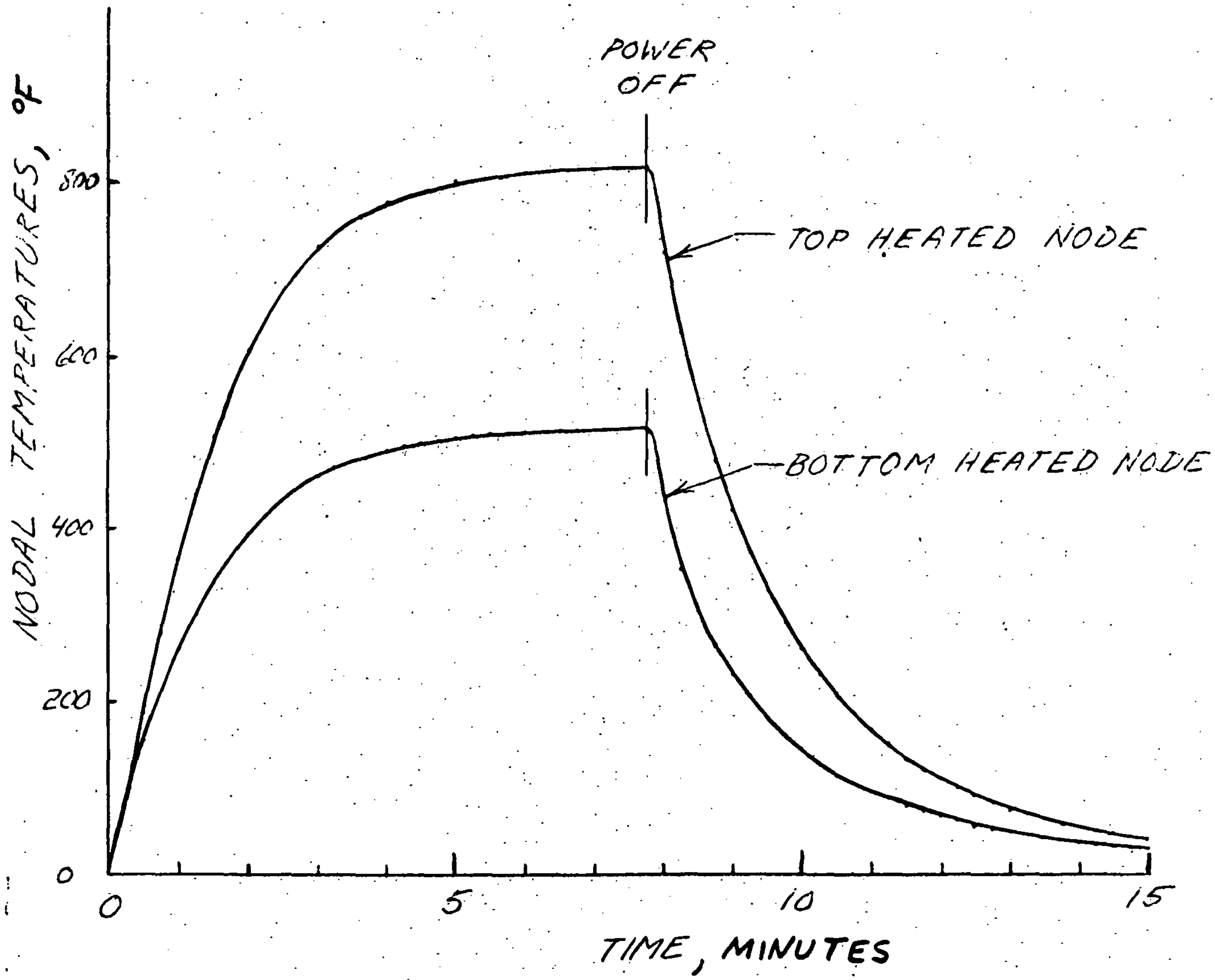


FIG. 14 CUP TEMPERATURE VERSUS TIME

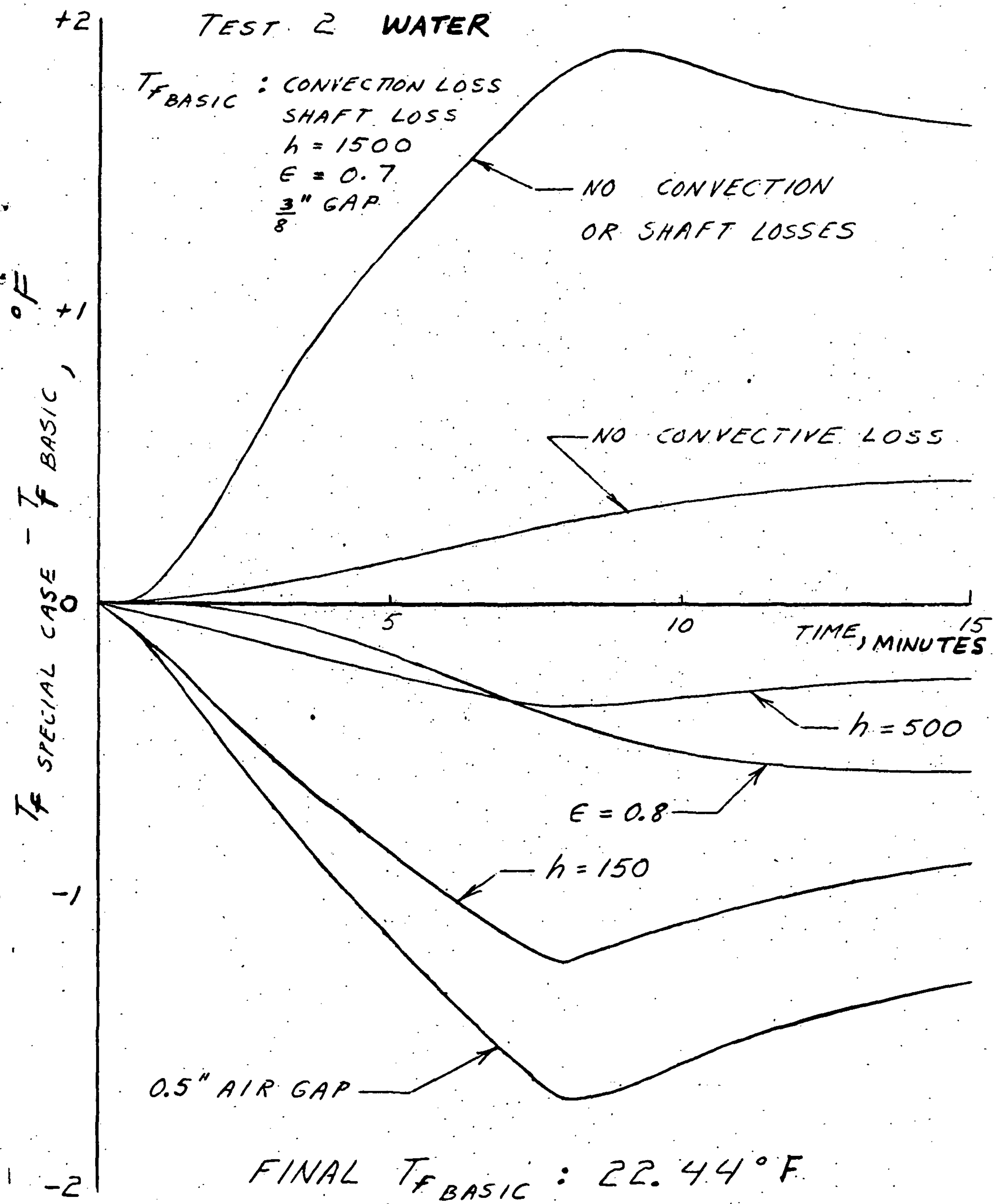


FIG. 15 FLUID TEMPERATURE DIFFERENCES FROM VARIOUS CAUSES

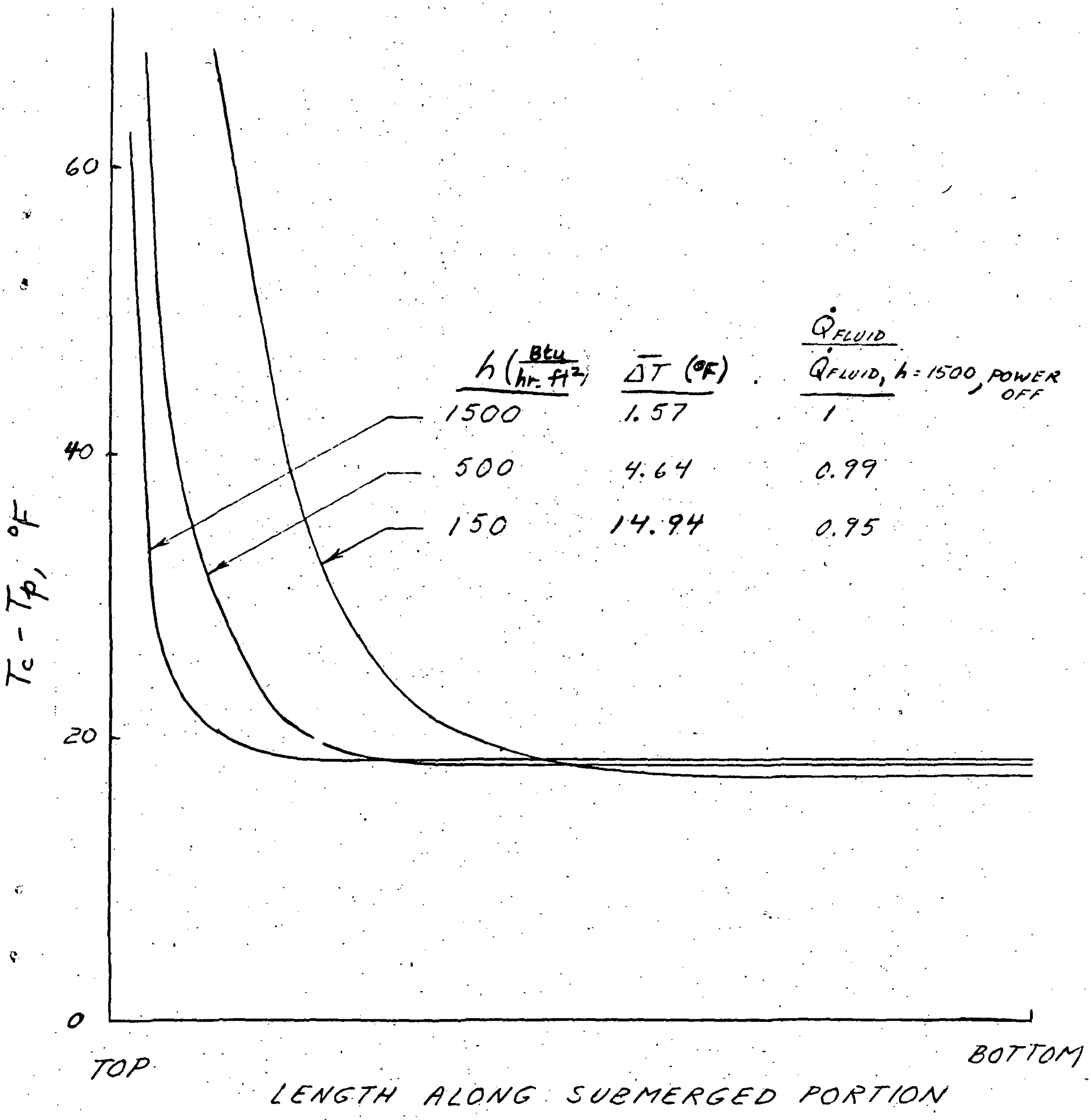


FIG. 16 CUP TEMPERATURE PROFILES FOR VARIOUS h VALUES.

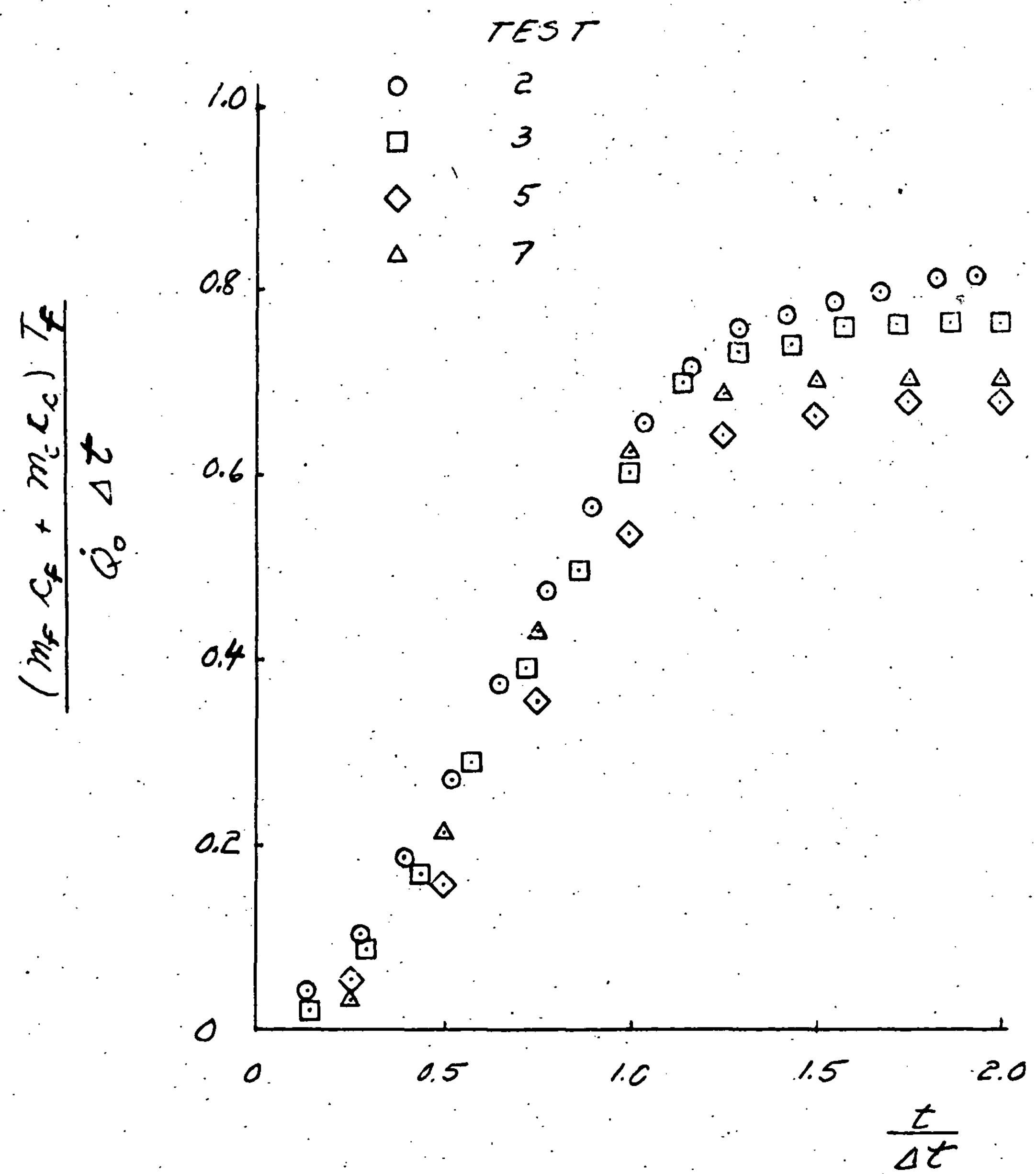


FIG. 17 DIMENSIONLESS PLOT OF ALL RESULTS