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AN INVESTIGATION OF THERMAL TRANSIENTS AT A SOLID-FLUID INTERFACE

SUMMARY

This report discusses analytical and experimental heat-transfer studies of cyclic temperature oscillations in a fluid medium and their effect on the temperature of an adjacent solid boundary. These studies are intended to aid in the analysis of thermal stress caused by cyclic temperature fluctuations.

The results of experimental measurements of thermal amplitude attenuation for an effectively infinite wall are presented together with heat-transfer analyses for two cases with finite walls.

NOMENCLATURE

A	area, ft^2
a	plate thickness, ft
b	h/k_w, ft^{-1}
C	cosh am
c	cos am
D	diameter, ft
d	constant
h	film coefficient of heat transfer, Btu/hr-ft^2-^oF
k	thermal conductivity, Btu/hr-ft-^oF
L	entrance length, ft
m	$\sqrt{\frac{\omega}{2\alpha}}$, ft^{-1}
q/A	heat flux, Btu/ft^2-hr
S	sinh am
s	sin am
t	temperature, ^oF
x	distance inside wall, ft
α	thermal diffusivity, ft^2/hr
ϵ	phase shift, radians
θ	temperature difference $t - t_m$, ^oF
ϕ	phase shift, radians
τ	time, hr
ψ	phase shift, radians
ω	angular velocity, hr^{-1}
Δ	amplitude of environment temperature, ^oF

Subscripts

e environment

m mean

w wall

Dimensionless Parameters

am $= a\sqrt{\frac{\omega}{2\alpha}}$

m/b $= (k_x/h)\sqrt{\frac{\omega}{2\alpha}}$

$\eta = \dfrac{\text{amplitude of temperature oscillation at surface of plate}}{\text{amplitude of temperature oscillation in environment}}$

$\delta = \dfrac{\text{amplitude of temperature oscillation at position } x \text{ in plate}}{\text{amplitude of temperature oscillation at surface of plate}}$

INTRODUCTION

It has been demonstrated that the combination of a nonuniform fluid temperature profile and an unstable velocity field will generate transient environmental temperature fluctuations which cause cyclic thermal stresses in the duct walls.[1] If the temperature of the boundary layer is sufficiently above that of the wall, and if the instabilities are appreciable, the magnitude of the temperature oscillations at the fluid-wall interface may be great enough for these thermal stresses to be significant. Therefore, an experimental program to investigate thermal transients and their effects on corrosion and material strength was initiated. Three interrelated areas were concurrently studied:

1. High-temperature thermal cycling - (investigations in the frequency range 0.01 - 1.0 cps).
2. High-temperature - high-frequency thermal cycling - (investigations in the frequency range 1.0 - 10.0 cps).
3. High-frequency thermal-cycling water tests.

The water tests were designed to develop instrumentation and thermocouple techniques for the high-frequency thermal-cycling test and to verify the theoretical calculations of attenuation of thermal oscillations at a solid-fluid interface for use in stress calculations. The interface attenuation has been studied analytically for the three boundary conditions: (I) adiabatic with infinite wall,[3] (II) adiabatic with finite wall,[4] and (III) bath condition with finite wall.[2, 6] The only conclusive experimental results to date have been for Case I.

ANALYSIS

A case of considerable practical interest is that in which a fluid with an oscillating mean temperature flows through a duct of circular cross section. For simplicity, it is assumed in the following analyses that this temperature variation is sinusoidal. Further, mathematical studies by Platus and Meghreblian[2, 6] have shown that the effects of a cylindrical geometry are unimportant when the inner radius of the tube is greater than one characteristic length, $1/m$. Thus, assuming unidirectional heat flow, the pertinent differential equation is

$$\frac{\partial^2\theta(x,\tau)}{\partial x^2} = \frac{1}{\alpha}\,\frac{\partial\theta(x,\tau)}{\partial\tau} \tag{1}$$

This equation has been solved for the three boundary conditions given previously.

Case I - Infinite Adiabatic Plate

The following additional assumptions are made:

(1) the physical properties are independent of temperature,

(2) the film coefficient of heat transfer, h, is constant and defined by the equation

$$h = \frac{q}{A\theta_{x,e}} \quad , \text{ at } x = 0$$

where $\theta_{x,e} = t_x - t_e$ is the instantaneous temperature excess of a point on the wall over the environment temperature t_e (requiring that q/A vary directly with $\theta_{x,e}$), and

(3) no net heat losses or gains exist.

Let the temperature of the environment fluctuate according to the equation

$$\theta_e = \Delta \cos (\omega\tau)$$

where Δ is the amplitude of the temperature excess, $(t - t_{e,m})$. Then, with the introduction of the dimensionless parameter,

$$\frac{m}{b} = \frac{k_w}{h} \sqrt{\frac{\omega}{2\alpha}} \quad ,$$

the integration of equation 1 leads to the following temperature distribution in the plate:[3]

$$\theta_{x,\tau} = \Delta \, \eta\delta \cos (\omega\tau - mx - \epsilon) \tag{2}$$

where

$$\eta = \sqrt{\frac{1}{1 + 2\, m/b + 2\, (m/b)^2}} \; ^*$$

$$\epsilon = \tan^{-1} \frac{1}{1 + \frac{b}{m}}$$

$$\delta = e^{-mx}$$

Case II - Finite Adiabatic Plate

With the same assumptions as those for Case I and allowing the temperature of the environment to fluctuate according to the equation

$$\theta_e = \Delta \sin (\omega\tau) \quad ,$$

the integration of equation 1 results in the following expression for the temperature fluctuations at the interface, $x = 0$:[4]

$$\theta_{0,\tau} = \Delta \, \eta \sin (\omega\tau + \phi) \tag{3}$$

*Note that η is the ratio of thermal amplitude at the surface to that of the environment.

where

$$\eta = \frac{\cosh 2\,am + \cos 2\,am}{\left[\gamma^2 + \left(\frac{m}{b}\beta\right)^2\right]^{1/2}}$$

$$\phi = \tan^{-1}\left(\frac{m}{b}\frac{\beta}{\gamma}\right)$$

$$\gamma = \frac{m}{b}(\sinh 2\,am - \sin 2\,am) + (\cosh 2\,am + \cos 2\,am)$$

$$\beta = \sinh 2\,am + \sin 2\,am$$

To determine the temperature distribution through the plate, it is assumed that a slab of thickness equal to 2a exists with the surfaces at $x' = a$ and $x' = -a$ maintained at temperature $\theta_{(0,\tau)} \sin(\omega\tau + \phi)$ for $\tau > 0$. The point $x' = 0$ can then be considered as the insulated back side of a thin wall whose surface temperature fluctuates as in equation 3. Then, the integration of equation 1 leads to the following temperature distribution in the plate:[5]

$$\theta_{x',\tau} = \delta\theta_{0,\tau} \sin(\omega\tau + \phi + \psi) \tag{4}$$

where

$$\delta = \left[\frac{\cosh 2\,mx' + \cos 2\,mx'}{\cosh 2\,ma + \cos 2\,ma}\right]^{1/2}$$

$$\psi = \mathrm{ARG}\left[\frac{\cosh mx'(1+i)}{\cosh ma\,(1+i)}\right]$$

Since

$$x' = a - x$$

$$\delta = \left[\frac{\cosh 2\,m(a-x) + \cos 2\,m(a-x)}{\cosh 2\,ma + \cos 2\,ma}\right]^{1/2}$$

$$\psi = \mathrm{ARG}\left[\frac{\cosh m(a-x)(1+i)}{\cosh ma(1+i)}\right]$$

On combining equations 3 and 4, the solution for the wall becomes:

$$\theta_{x,\tau} = \delta\eta\Delta \sin(\omega\tau + \phi + \psi) \tag{5}$$

Case III - Finite Plate with Bath Conditions at the Free Face of the Plate

It is assumed that

(1) the physical properties are independent of temperature,

(2) the film coefficient of heat transfer is constant, and

(3) the temperature of the free side is equal to zero.

If the temperature of the environment is allowed to fluctuate according to the equation

$$t_e(\tau) = t_e(1 + d \cos \omega\tau) \quad ,$$

then the integration of equation 1 leads to the following equation for the temperature at the face of the plate:[6]

$$t(0,\tau) = \frac{at_e}{1 + a} + dt_e\eta \sin(\omega\tau + \phi) \tag{6}$$

where

$$\eta = \sqrt{\frac{2m^2/b^2\ (S^2C^2 + s^2c^2) + 2m/b\ (SC + sc)(S^2 + s^2) + (S^2 + s^2)^2}{2m^2/b^2\ (S^2 + c^2) + 2m/b\ (SC + sc) + S^2 + s^2}}$$

$$\theta = \tan^{-1} \frac{m/b\ (SC + sc) + S^2 + s^2}{m/b\ (SC - sc)}$$

and

$$S = \sinh am$$

$$C = \cosh am$$

$$s = \sin am$$

$$c = \cos am$$

EQUIPMENT

An experimental system was designed to test these analytical solutions. A flow diagram of the test apparatus is shown in Figure 1. Two bellows-sealed reciprocating pulse pumps, 180 deg out of phase, were used to force the oscillations by pneumatically injecting alternate slugs of hot and cold water into the common stream such that the total flow rate was constant. These pumps feature Scottish yoke drive mechanisms to obtain a sine wave output. A U. S. Motors Vari-Drive was used to drive the pumps in the frequency range from 0.7 - 10 cps. The operating conditions are shown in Table I. A baffled 50-gallon hold-up tank was used to deaerate the hot water supply. The flow rate of both the hot and cold streams were measured by Fisher-Porter Rotameters.

The test section was a 0.470-in. I.D., 0.250-in. wall thickness Inconel tube with an entrance length of 24 in. (L/D) = 50). This entrance length was sufficient to insure established flow. Instantaneous stream temperatures were measured with a 36-gauge Inconel-nickel wire thermocouple. This thermocouple was experimentally determined to have a time constant (the time required for the junction temperature to reach 63.2 per cent of the amplitude of a step-temperature change) of less than 3.5 milliseconds and to attain 95 per cent of this change in less than 15 milliseconds. Frequencies up to 30 cps could be accurately followed. The thermocouple junction was positioned approximately mid-stream and in the same plane as the thermocouple used in obtaining the tube inside surface temperature.

Inconel-nickel "gunbarrel" thermocouples obtained from Midwest Research Company were used to measure the wall temperature fluctuations at the inside

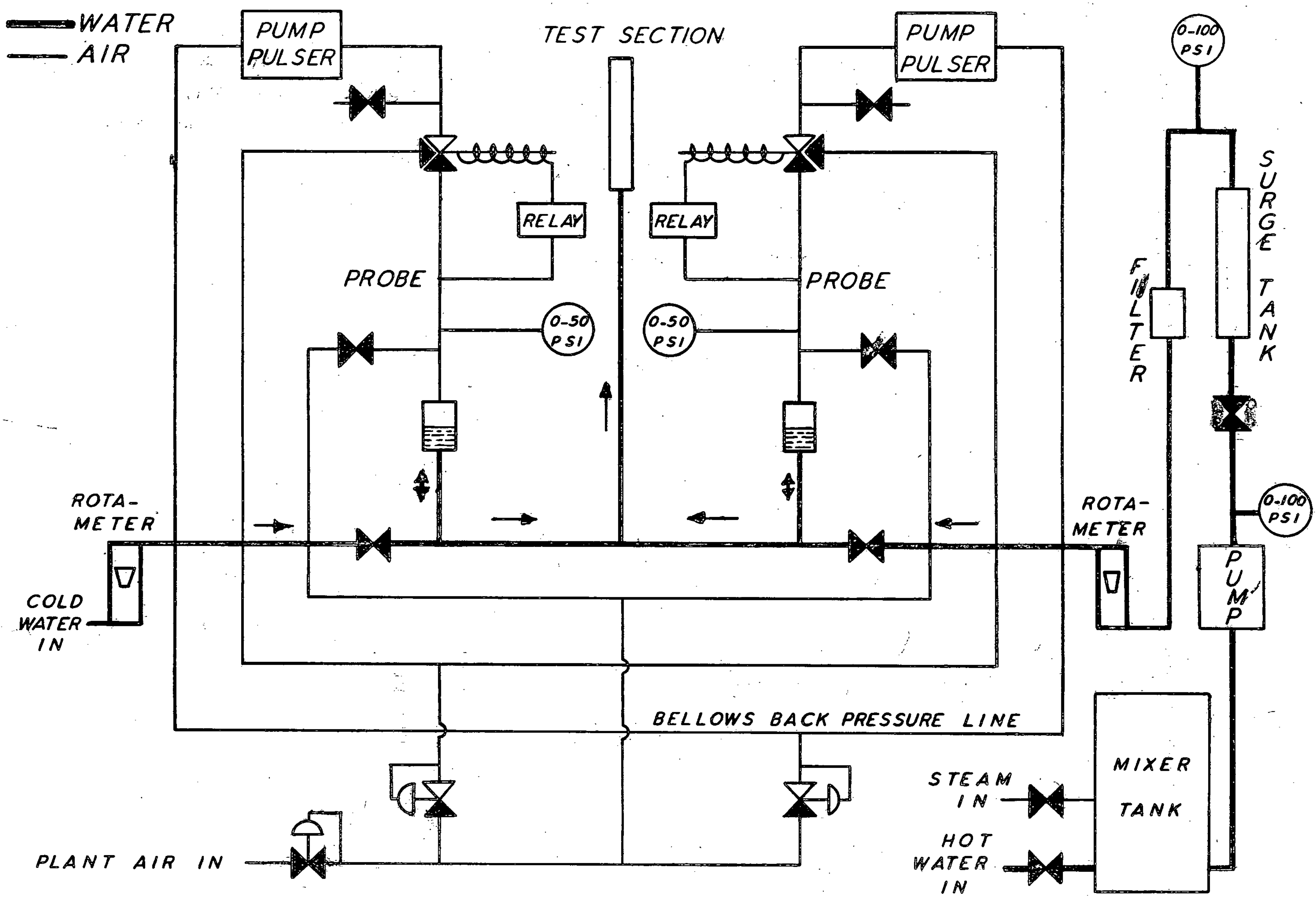

FIG. 1 — FLOW DIAGRAM OF HIGH FREQUENCY THERMAL CYCLING WATER TEST

surface of the Inconel test piece. These thermocouples were shrunk fit into the wall of the tube and the tube reamed such that the end of the thermocouple was flush with the tube inside wall. The inside surface of the tube was then coated with a thin ($\sim$1 micron) layer of nickel by vacuum evaporation to form the thermocouple junction. Figure 2 illustrates the thermocouple construction and installation. Since Inconel comprised 98 per cent of the volume of the thermocouple, it was believed that the nickel wire and coating had negligible effect on the measured temperatures. Due to its low mass, the response of this wall thermocouple was essentially instantaneous. The temperatures were recorded using a Kay Lab preamplifier (Model III D.C.) in conjunction with a Brush amplifier (Model BL-550, High-Gain D.C.) to drive the Brush oscillograph (Model 202) from the low-level thermocouple output signals.

PROCEDURE

The thermocouples (both wall and mid-stream) were calibrated at the hot and cold stream temperatures by operating the system isothermally at both the hot and cold water temperatures. Under such isothermal conditions all the thermocouple readings should be equal, and where variation existed, a correction factor was determined. Flows and temperatures were then set so that the mean bulk temperature of the mixed hot and cold streams was equal to that of the ambient to insure adiabatic conditions.

The η (wall attenuation) ratio was obtained directly by measuring the amplitude of the wall thermocouple reading on the chart and dividing by the amplitude of the stream thermocouple reading, with appropriate correction factors.

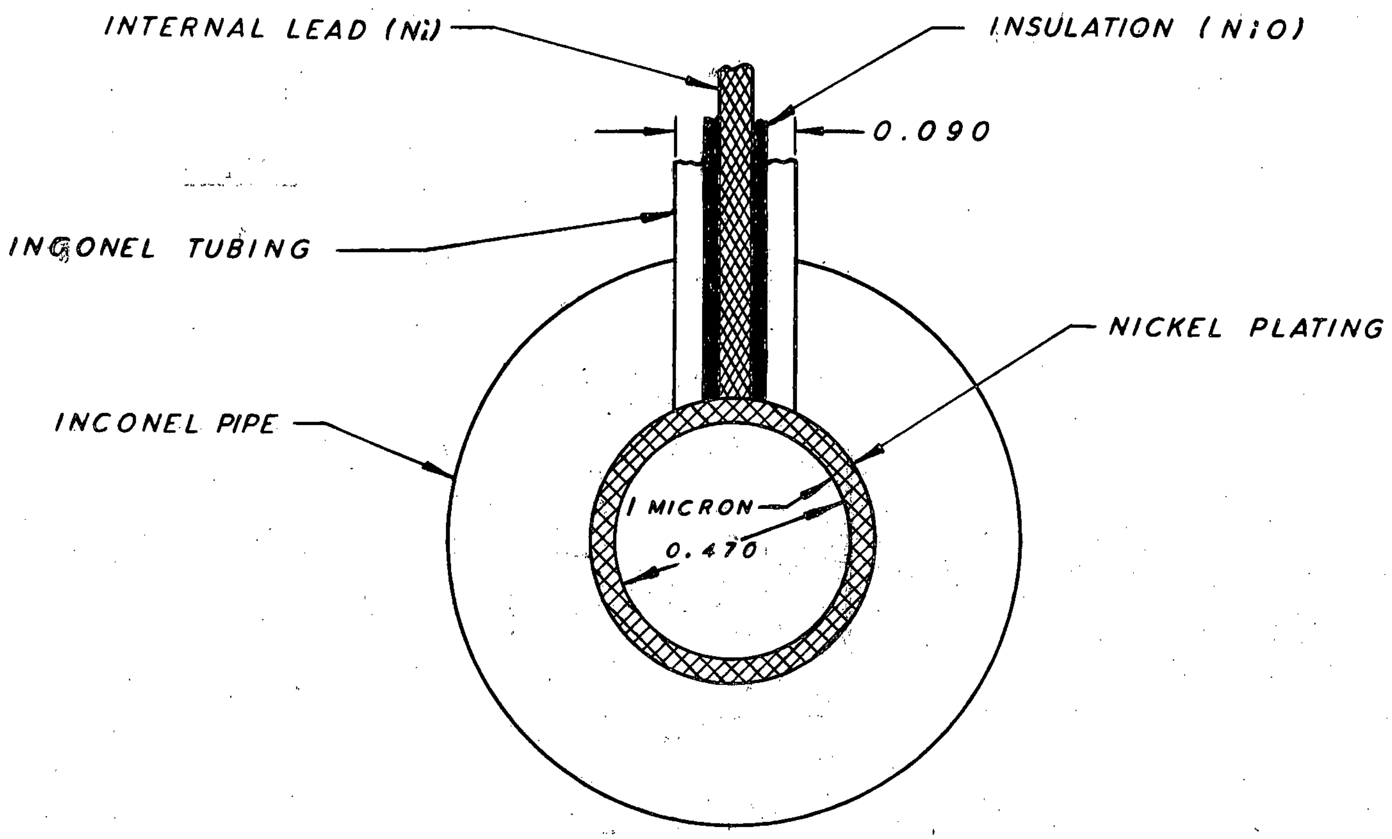

FIG. 2 — DETAIL OF "GUNBARREL" THERMOCOUPLE INSTALLATION

TABLE I

	RUN NO. I	RUN NO. II	RUN NO. III
MIXED FLOW RATE (GPM)	8.25	7.7	18.0
MEAN FLUID TEMPERATURE (F)	80	80	80
HOT FLUID TEMPERATURE (F)	100	100	100
COLD FLUID TEMPERATURE (F)	60	60	60
MEAN REYNOLDS NUMBER	64,000	99,100	143,700
MEAN HEAT TRANSFER COEFFICIENT	2,675	3,700	5,000

OPERATING CONDITIONS

RESULTS AND DISCUSSION

The experimental results are shown in Figures 3, 4, and 5 in which η is plotted as a function of frequency for three values of the film coefficient, h.* A comparison is made with the theoretical curve derived for Case I (thick-walled, adiabatic). The flow was turbulent in all experiments. The results seem to indicate that the Jakob equation (equation 3) applies with reasonable accuracy, with the greatest deviations occurring at the lower frequencies. The three sets of data indicate the excellent precision of the experimental measurements. Since the effect of temperature on the heat-transfer coefficient is quite pronounced, the maximum allowable Δt was set at $40^{\circ}F$. This insured a variation in h of only $\pm$ 5 per cent about the mean.

It is proposed to make similar experimental measurements to check the theory of Cases II and III (finite wall), since these are of practical importance.

CONCLUSIONS

The validity of the theoretical analysis for attenuation of thermal oscillations by an effectively infinite solid wall has been experimentally confirmed for a particular solid-liquid system under conditions of fully developed turbulent flow. The "gunbarrel" thermocouple technique appears to be an excellent one for measurement of instantaneous surface temperatures in thick metal walls. Its adaptability to high-temperature systems has been briefly investigated.[7] Further work is indicated to determine the validity of the finite wall analyses.

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*h is calculated from the Dittus-Boelter equation:

$$N_{Nu} = 0.023\ N_{Re}^{0.8}\ N_{Pr}^{0.35}$$

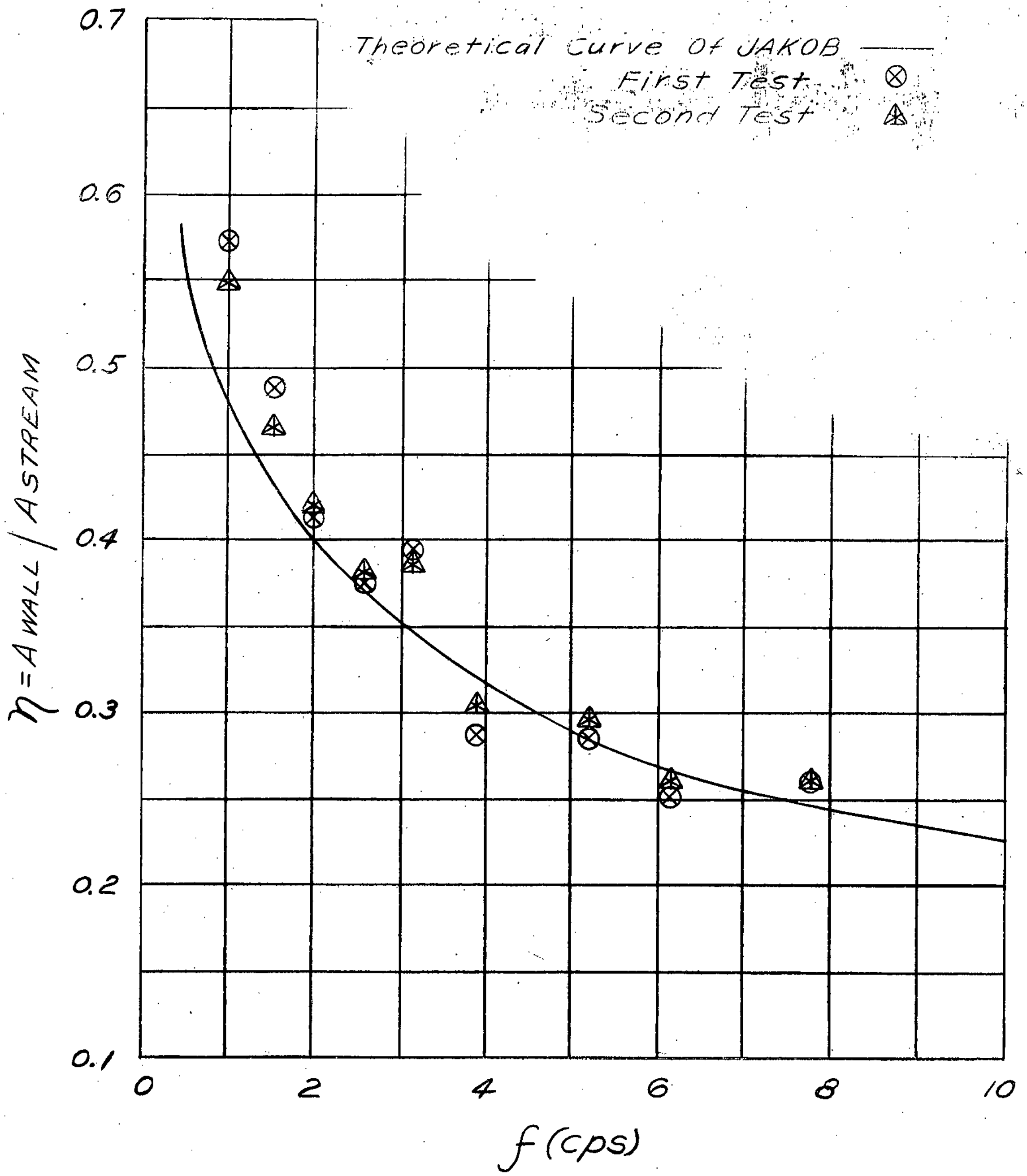

FIG. 3- RATIO OF WALL TEMPERATURE AMPLITUDE TO THE FLUID TEMPERATURE, η, AS A FUNCTION OF THE FREQUENCY, f, OF THE THERMAL CYCLE IMPOSED ON THE FLUID FOR A THICK-WALLED INCONEL TUBE WITH A FILM HEAT TRANSFER COEFFICIENT OF 2675 BTU/HR -FT^2-$^{\circ}$F

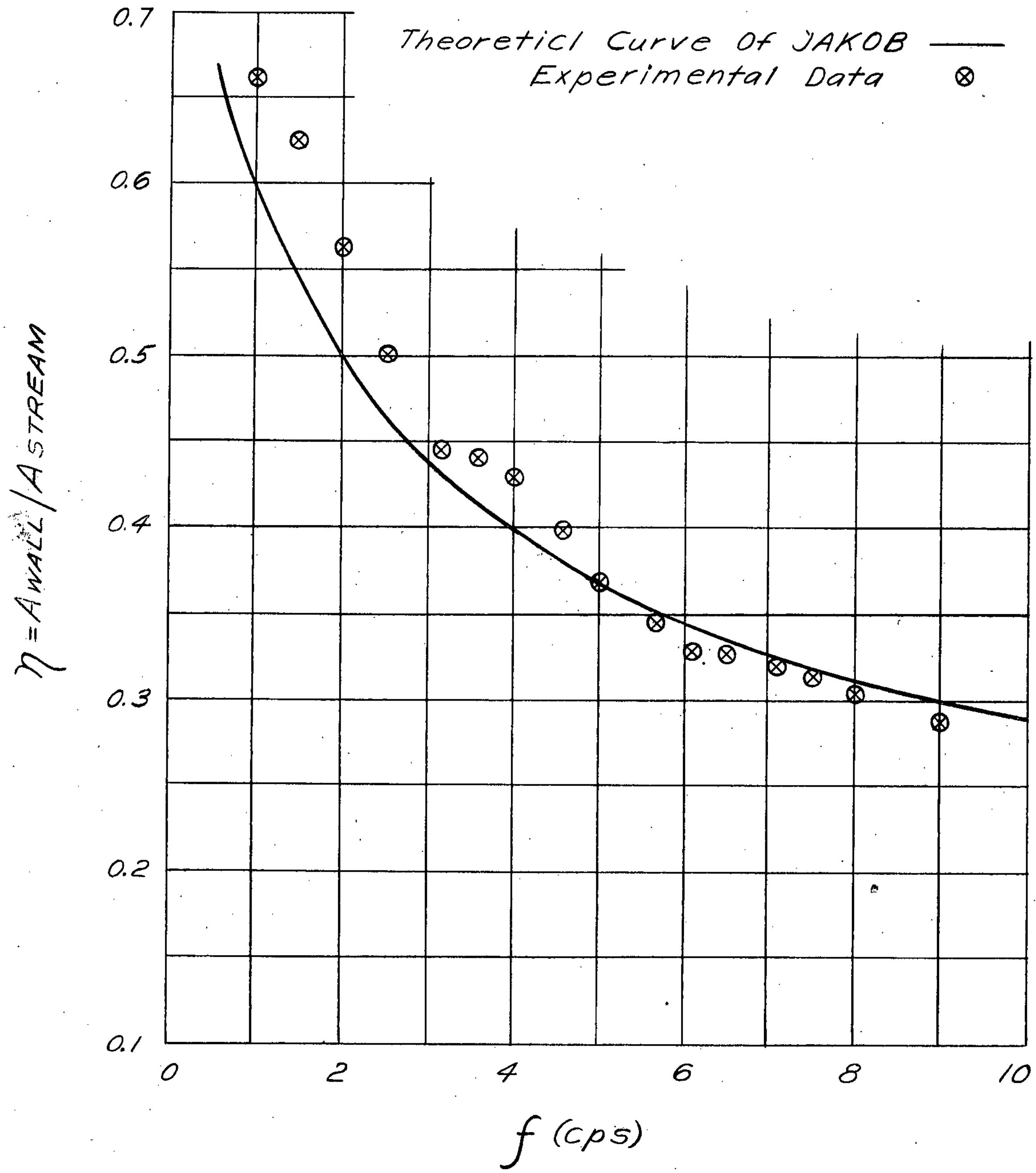

FIG. 4 - RATIO OF WALL TEMPERATURE AMPLITUDE TO THE FLUID TEMPERATURE, η, AS A FUNCTION OF THE FREQUENCY, f, OF THE THERMAL CYCLE IMPOSED ON THE FLUID FOR A THICK-WALLED INCONEL TUBE WITH A FILM HEAT TRANSFER COEFFICIENT OF 3700 BTU/HR-FT^2-$^{\circ}$F

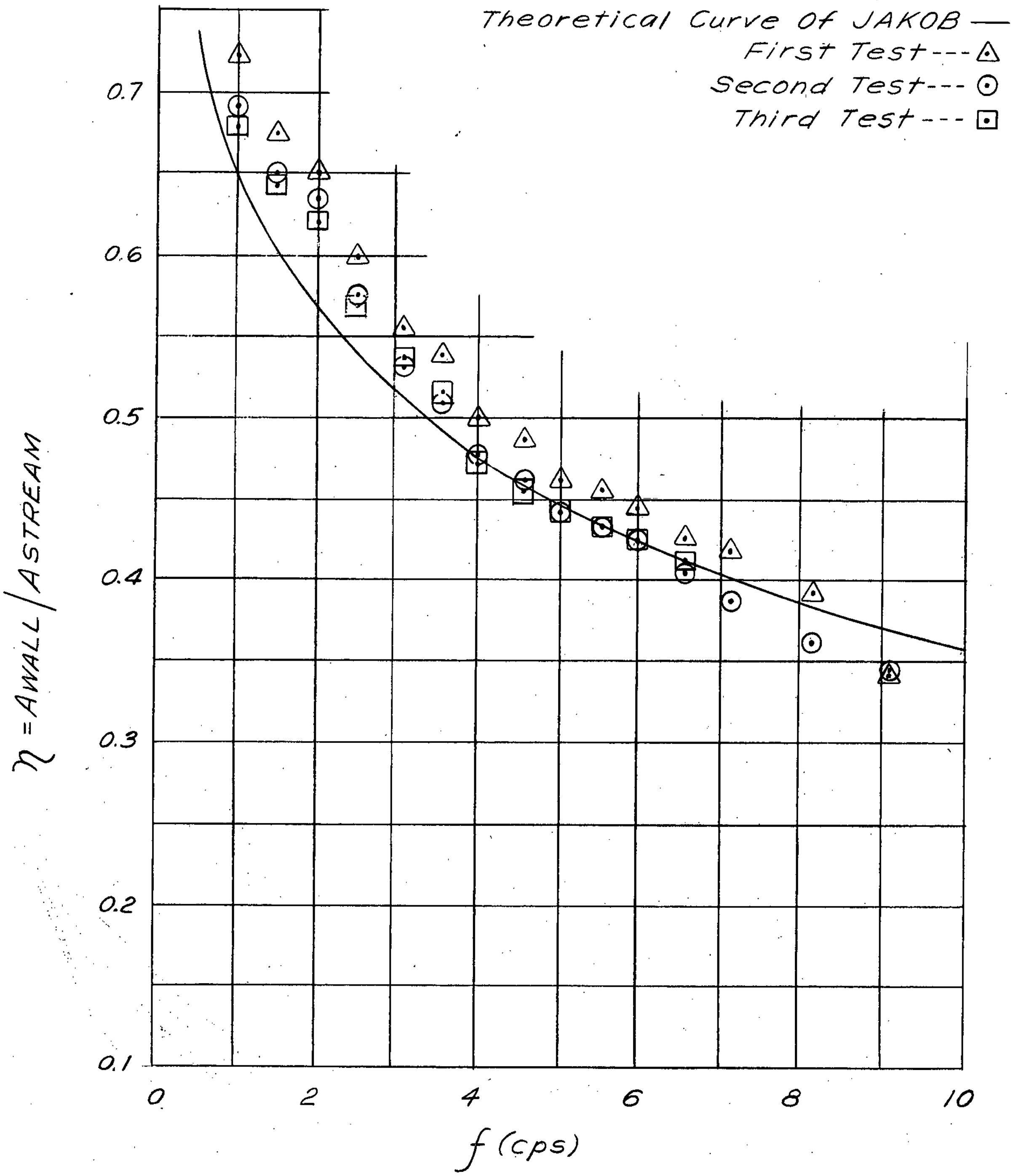

FIG. 5 - RATIO OF WALL TEMPERATURE AMPLITUDE TO THE FLUID TEMPERATURE, η, AS A FUNCTION OF THE FREQUENCY, f, OF THE THERMAL CYCLE IMPOSED ON THE FLUID FOR A THICK-WALLED INCONEL TUBE WITH A FILM HEAT TRANSFER COEFFICIENT OF 5000 BTU/HR-FT^2-$^{\circ}$F

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