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TRANSIENT PRESSURE MEASURING METHODS. TRANSIENT PRESSURE TRANSDUCER DESIGN AND EVALUATION

PRINCETON UNIV NJ

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I. GENERAL

This technical report is the second in the series on transient pressure measurement. The program has developed from the requirement for improved dynamic pressure measurements in the combustion instability program operated under the Department of Aeronautical Engineering at Princeton University. The present technical report covers basic transducer design, including the vibratory system, heat transfer characteristics, application of such specialized transducers to rocket chambers, and a description of a static and dynamic calibration system. The shortcomings of present transducers, such as lack of heat transfer ability and limited frequency response are discussed, and are the target of the tests to be described. All the information gathered in these reports is of immediate interest to the combustion instability program, but will also be of more general interest to various facilities involved including research and development testing, and transducer manufacturers, several of whom have been cooperative in this program.

A third report will be published shortly on comparative results of tests made on several commercially available transducers, whereas the first report covered the use of transducers at the end of tubing, to measure dynamic pressures.

II. TRANSDUCER DESIGN

In order to appreciate the limitations of a practical transient pressure transducer and the problems facing a transducer designer, the transducer will be considered first as a classic vibratory system. Having thus considered the overall dynamic response to a driving function, the portions constituting the mass, compliance, and damping will be separately considered. The basic design limitations of frequency response, accuracy, and cooling will be covered in light of their mutual interdependence.

III. THE VIBRATORY SYSTEM

In any study of the dynamic response of a body to a driving function, the subject of harmonic motion is bound to be of primary interest. A pressure transducer, subjected to a varying pressure, will respond as a whole in full accordance with the physical laws governing any such system. Providing the slope of the driving function is flat enough, response will be the steady-state response. When the slope of the driving function becomes steeper, however, the response of the vibratory system will depart from the steady-state response, with the result that the system (or transducer) output then injects its own signal in the form of spurious response, phase shift, etc.

In Figure la a typical vibratory system is shown. Any body can vibrate in response either to an external force or to internal forces generated by an interchange of energy from one element to the other. The response of a transducer to an external oscillating pressure can be considered "forced vibration" while that response, caused by the passage of a step function of pressure is "free vibration". Since it is apparent that the "ringing" caused by a step function will eventually die out, the

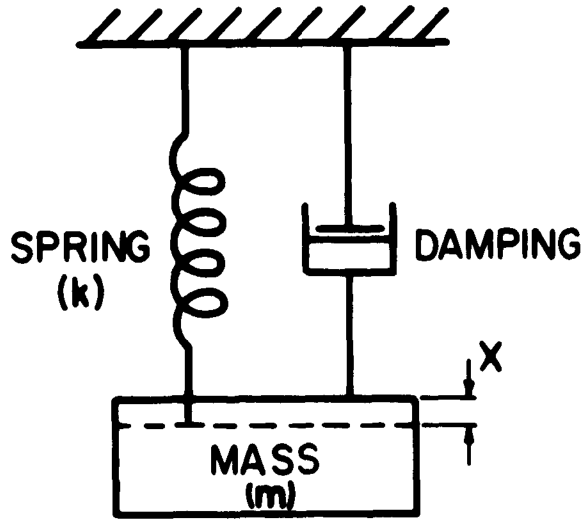

FIGURE 1a - SECOND ORDER VIBRATORY SYSTEM
SINGLE DEGREE OF FREEDOM

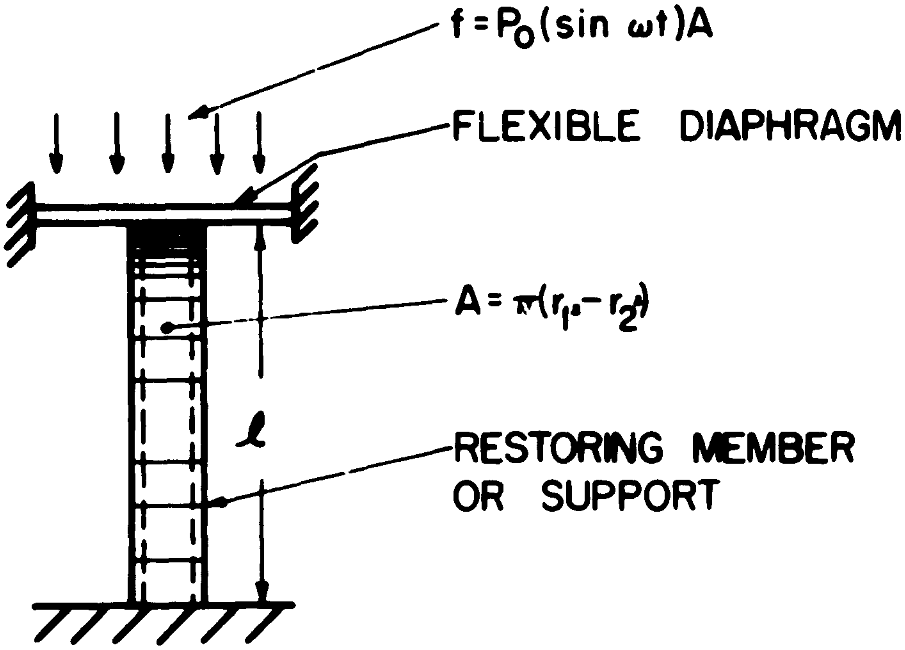

FIGURE 1b - SECOND ORDER VIBRATORY SYSTEM
MANY DEGREES OF FREEDOM

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system of Figure la has damping shown, is a "Second-Order System" and has a natural frequency which is a function of all three components.

Variations on this simple system immediately present themselves in a practical case. The most common, when investigating a transient pressure transducer, is that of Figure lb. When a transducer is to be subjected to the high rates of heat transfer present in a rocket chamber complexity of construction because of cooling requirements results in several masses and several compliant elements. These are unfortunately interconnected, and so interact with one another, usually at different natural frequencies. It is often possible, however, to analyze the behavior of a transducer sufficiently accurately, without considering the several degrees of freedom that are present.

A. <u>Analysis of the Vibratory System</u>

Fortunately, most periodic vibrational responses resulting from such systems as shown in Figure la and lb are harmonic and so can be analyzed in a reasonably simple manner (Ref. 1).

The basic equation for harmonic motion, neglecting damping is:

$$\frac{d^2x}{dt^2} + \omega^2 X = 0 \tag{1}$$

where

$\omega = 2\pi f = \dfrac{2\pi}{\text{period}}$

M = mass

x = displacement

The mass, when displaced a distance x will be restored by a force Kx by the spring and Equation (1) becomes:

$$M\left(\frac{d^2x}{dt^2}\right) + K X = 0 \tag{2}$$

The general solution of the differential Equation (2) must contain two constants of integration. If the general solution, $x = A \sin\omega t + B \cos\omega t$ is substituted into Equation (2), the result is:

$$\left(-\omega^2 + \frac{K}{M}\right)x = 0$$

Obviously, a solution for any value of x is:

$$\omega^2 = \frac{K}{M}$$

$$\omega = \sqrt{\frac{K}{M}} \quad \text{or} \quad F = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \qquad (3)$$

Equation (3) is the simplest method of describing natural frequency.

The consideration of the foregoing has been from a harmonic standpoint. When these vibratory systems are disturbed, they will freely oscillate, at a frequency determined by Equation (3) continuously. When damping is added as in a practical case, the result is damped free oscillation shown in Fig. 2. This cannot be considered harmonic motion except for small values of damping, since adjacent cycles are not equal. The frequency then must be corrected for the damping, if such correction is appreciable. The type and amount of damping should therefore be studied.

Damping is the dissipation of the energy in the vibratory system from cycle to cycle and consists of viscous damping, coulomb or frictional camping, and solid damping. One or more types are always present in any practical system, although in a transient pressure transducer, damping is small.

Viscous damping exerts a force proportional to the velocity.

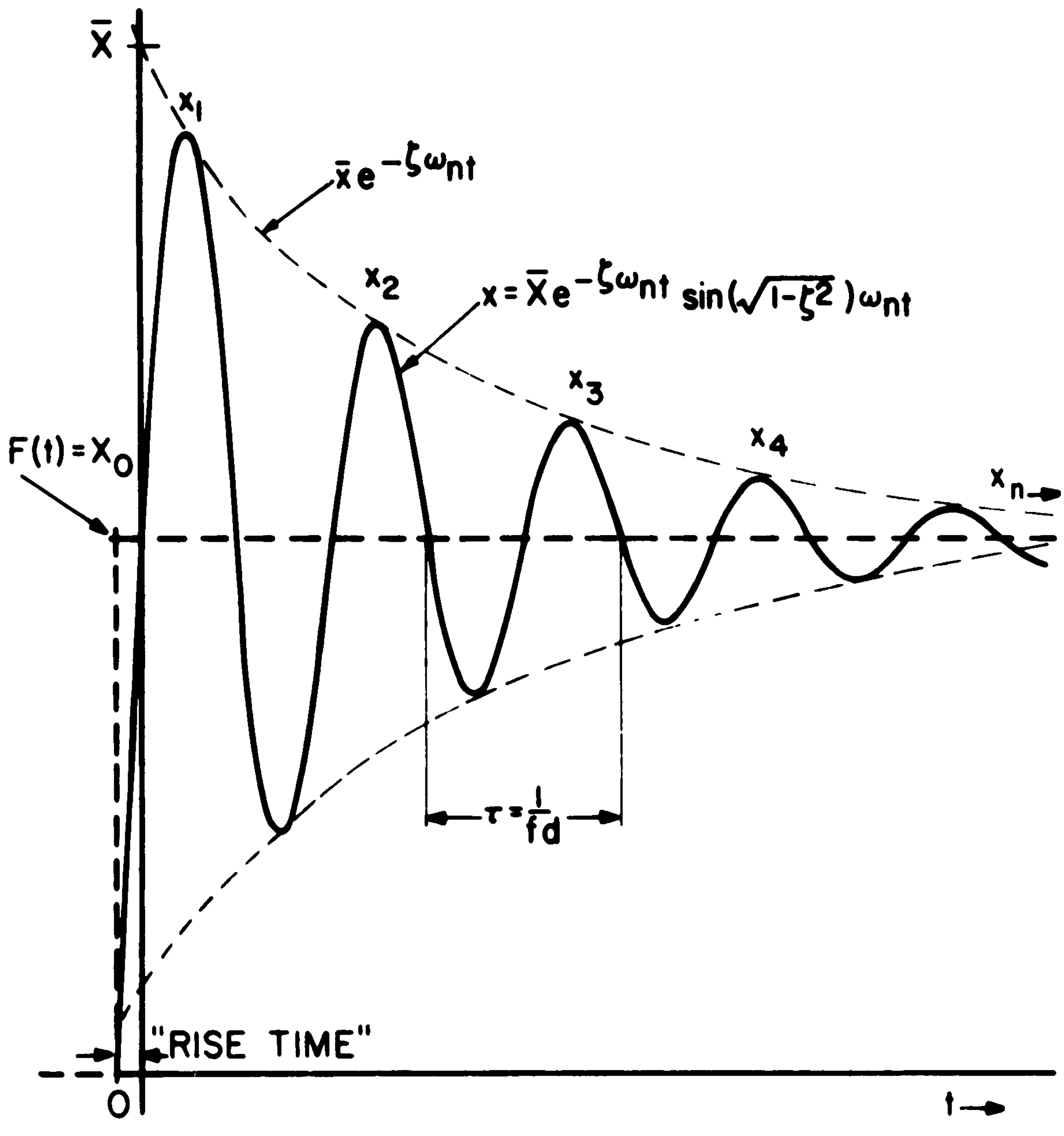

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FIGURE 2

Viscous damping is analogous to the dashpot, a body moving at moderate speed through a fluid:

$$F = -cv$$

where c = a constant related to damping

v = velocity of the moving part

Coulomb damping is caused by the sliding of dry surfaces. It is determined by the equation of kinetic friction:

$$F = -\mu F_n$$

where μ = coefficient of friction

F_n = the normal force between elements

Coulomb damping is essentially constant and is the force which results in the complete cessation of oscillation when the final stages of damped free oscillation are reached when amplitude and therefore velocity sensitive or viscous damping is a small effect.

Solid damping is due to the friction within the material itself, primarily in the material of which the restoring member is constructed. Solid damping force is independent of frequency, but proportional to the amplitude of oscillation,

$$F = \gamma x K$$

where γ is a constant of the material and is nondimensional.

In a complex vibrational assembly of fast response, such as a transient pressure transducer, the value of K must be large, to achieve the necessary speed of response. It is, therefore, extremely difficult to insert viscous damping in a transient transducer. The use of extremely viscous oil or grease can add a small amount of damping. Coulomb damping exists when there is slipping or sliding contact between two elements in the transducer. Solid damping, because of the low values of viscous damping obtainable becomes an important factor in the overall damping of a transducer, although it normally is a small damping force in itself.

When damping is extremely large the result of a disturbance in the vibratory system is a slow return to equilibrium and no oscillation. When damping is reduced to the point where oscillation just does not occur, the condition of "critical damping" exists. When damping is still further decreased, a damped oscillation occurs and the practical case applying to pressure transducers pertains.

Solid and viscous damping both result in an exponential decay of oscillation in a system damped below critical. A convenient way of defining the damping is in terms of the logarithmic decrement, δ ;

$$\delta = \ln \frac{X_1}{X_2} \quad \text{or} \quad \frac{X_1}{X_2} = \varepsilon^{\delta} \tag{4}$$

where $\frac{X_1}{X_2}$ = ratio of successive cycles

The nondimensional ratio, ζ , is commonly used and represents the ratio of the damping constant of a particular case to critical damping.

$$\zeta = \frac{C}{Cc} \tag{5}$$

ζ is known as the "damping factor."

For small values of damping,

$$\zeta \cong \frac{\delta}{2\pi} \tag{6}$$

To determine damping from the response of a transient transducer under damped free vibration, Fig. 3 was plotted, ζ vs cycles

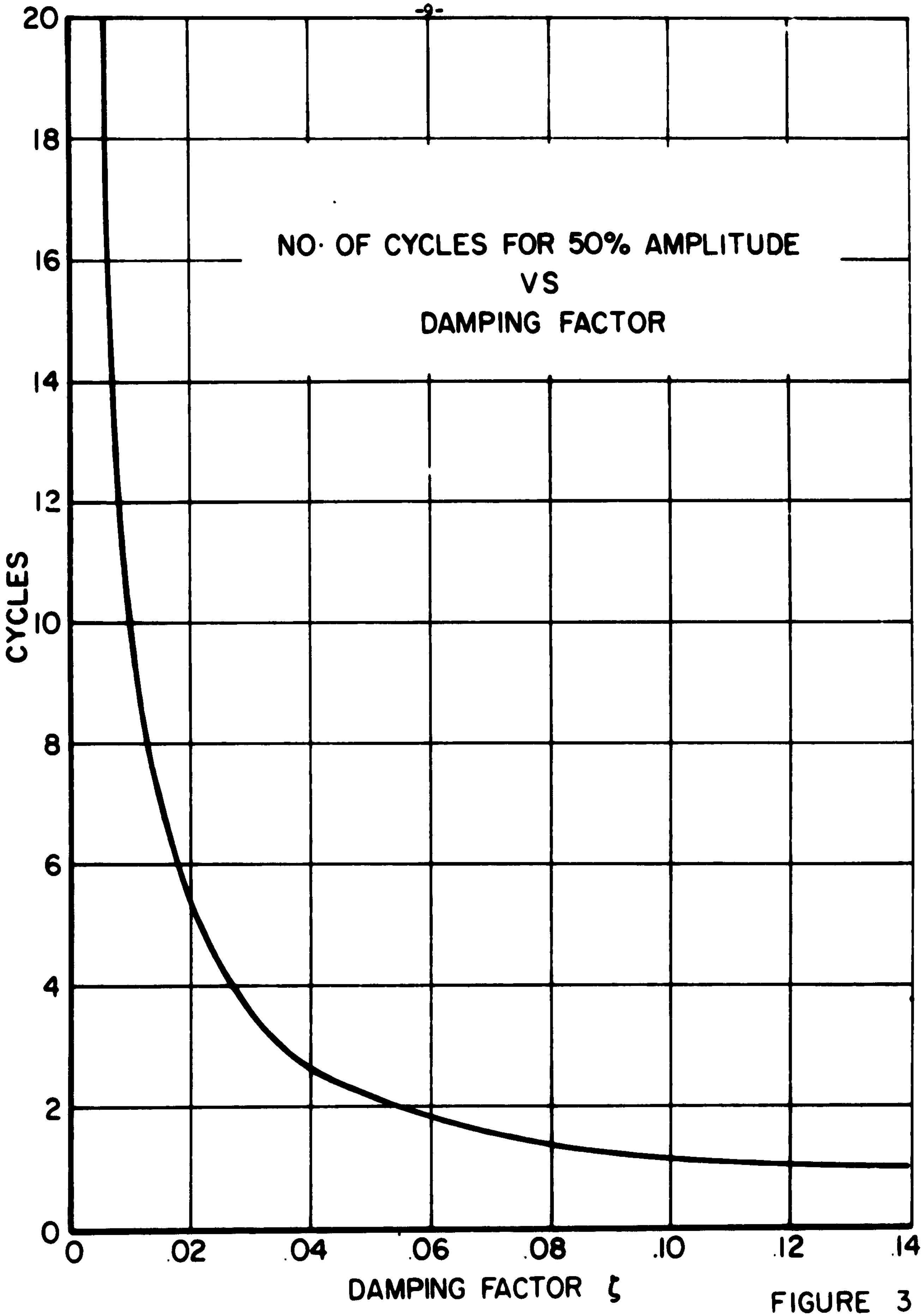

FIGURE 3

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for 50% decay:

$$\delta = \frac{1}{n} \ln \frac{X_o}{X_n}$$

$$= 2\pi \zeta = \frac{1}{n} \ln 2$$

$$\zeta n = \frac{.693}{2} = .110 \qquad (7)$$

Fig. 3 is a convenient way to determine damping from a photograph of transducer response to a step function.

It is not necessary to discuss higher damping factors because transient pressure transducers exhibit uniformly low damping characteristics.

The assumption that the combination of viscous and solid damping, still of a fairly low total value of ζ, will act as purely viscous damping is a fair approximation. For a free, viscous damped oscillation, therefore, the equation will be:

$$x = X \varepsilon^{-\zeta \omega_n t} \sin \left(1 - \zeta^2\right) \omega t \qquad (8)$$

Note that the damped natural frequency is $\sqrt{1-\zeta^2}$ times the undamped frequency.

Previous discussion has referred primarily to the case of a free oscillation. When a transducer is used to measure dynamic pressures, continual excitation of the vibratory system of the transducer occurs. To permit analysis, assume the excitation to be harmonic:

$$m \frac{d^2x}{dt^2} + C \frac{dx}{dt} + Kx = Fo \sin \omega t \qquad (9)$$

The oscillation observed as a result of the force, is

$$x = \bar{X} \sin(\omega t - \phi) \tag{10}$$

where $\bar{x}$ = amplitude of oscillation

ϕ = phase angle by which oscillation lags driving force

Substituting Eq. (10) into Eq. (9) the following equations appear which may be physically evaluated:

$$\bar{X} = \frac{Fo}{\sqrt{(k - m\omega^2)^2 + (c\omega)^2}} \tag{11}$$

$$\phi = \tan^{-1}\left(\frac{c\omega}{k - m\omega^2}\right) \tag{12}$$

These may also be reduced further by substitution of Eqs. (3), (5), $C_c = 2m\omega_n$, into Eqs. (11) and (12). The nondimensional term, A, the response ratio and ϕ are thereby described in terms of $\frac{\omega}{\omega_n}$, the ratio of the driving frequency to the natural frequency. (These equations are plotted in Fig. 4a and 4b.)

$$\frac{X}{X_{st}} = A = \frac{1}{\left(1 - \left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\zeta\frac{\omega}{\omega_n}\right)^2} \tag{13}$$

$$\phi = \tan^{-1} \frac{\left(2\zeta\frac{\omega}{\omega_n}\right)}{\left(1 - \frac{\omega}{\omega_n}\right)^2} \tag{14}$$

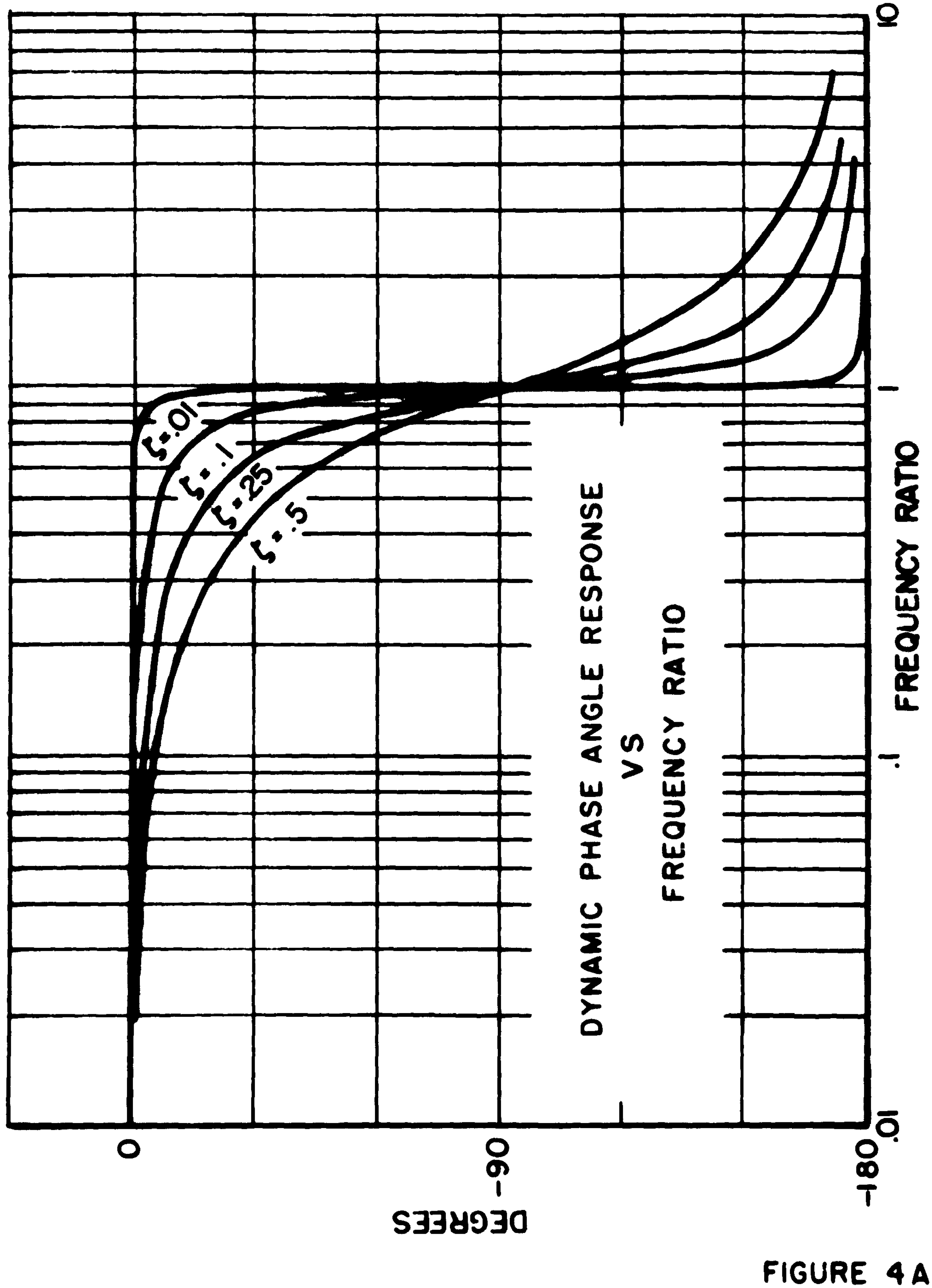

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FIGURE 4A

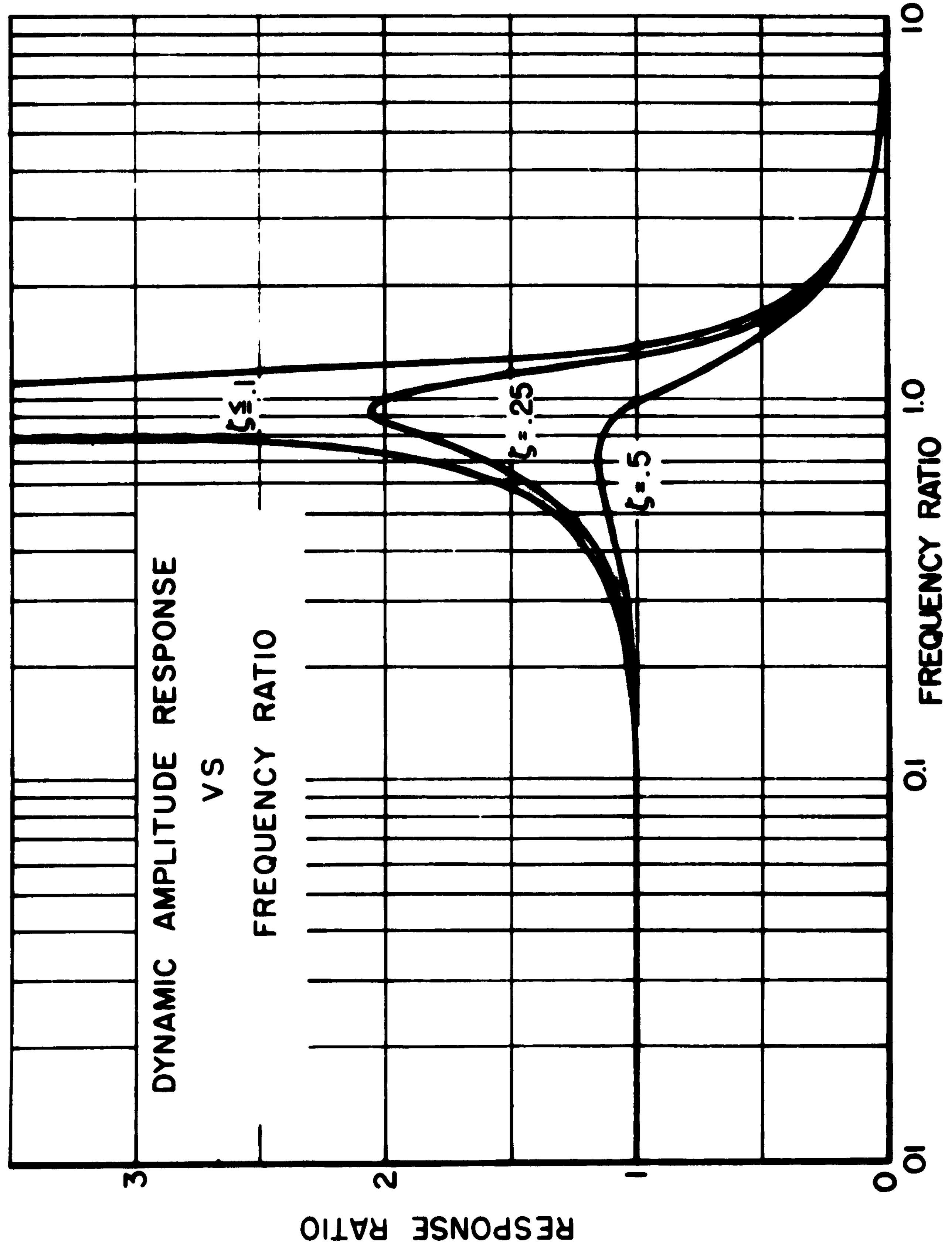

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FIGURE 4B

In the treatment so far, only a simple vibratory system, with a single degree of freedom has been investigated. Design of transducers, and estimation of their response prior to their fabrication can often be accomplished by the methods so far described. The difficulty of the design analysis is immeasurably increased when the more realistic cases are attacked, so from this point derivations, and approximations, such as by Rayleigh's method, Ref. (2) are omitted.

A liquid cooled pressure transducer usually will consist of a single or double diaphragm, of a more or less compliant nature, supported at or near the center. It may, however, be supported at the edges and be stiff enough to resist the applied pressure itself.

When considering such a transducer, where the entire vibratory system consists of a flat diaphragm, with edges clamped, and no initial tension, resonant frequency is Ref (3):

$$fn = 1.945 \frac{tc}{d^2} \tag{15}$$

where

t = thickness

c = velocity of sour.d in diaphragm material

d = diameter

When a compliant diaphragm, supported by a stiff column is involved, the support will represent a number of degrees of freedom itself as it vibrates in a longitudinal mode, with one end fixed and the other free:

$$fn = \frac{n}{4l} \sqrt{\frac{AE}{\rho}} \tag{16}$$

where n = 1,3,5,7 etc-odd integers

l = length of rod or tube

A = cross-sectional area

E = Young's modulus

ρ = mass/unit length

This results in the situation wherein two oscillatory modes, probably of different frequencies, exist together.

In estimating the performance of a transducer to a dynamic excitation, Ref. (4) usually sufficiently accurate information is available when the assumption of a simple second order system is made. When two or more vibratory modes exist at different frequencies, it becomes evident in the form of beats in the step function response of the transducer. The result is a distortion of the damped oscillation of Fig. 2, and difficulty arises in estimating damping. Elimination of one frequency, either by filtering or by estimating its effect on the damped oscillation will serve to permit analysis. In cases where one frequency is predominant, assumption of a second order system is not much in error.

The response of the transducer to a dynamic pressure may also be affected by the impedance match of the diaphragm to the medium applying the dynamic pressure. Transducers for rocket chamber applications, however, are normally designed for large steady state pressures and the gas density, being therefore large, coupling in the higher pressure ranges such as 1000 psi is adequate to minimize difficulty from this cause Ref.(5). The relatively small active diaphragm diameter of modern transducers also tends to minimize difficulty on this account.

IV. HEAT TRANSFER AND COOLING

A. Design Parameters

In this section, the criteria affecting the ability of a flush diaphragm transducer to survive and operate in a rocket chamber will be covered. The techniques described will be familiar to rocket chamber designers, but apparently have not been applied to the transducer designs presently available.

The prime requirement for a transducer diaphragm is that it act as a pressure seal and simultaneously be able to maintain its operating temperature below that at which failures might occur due to melting or stress, or a combination of these. In the case of a dynamic transducer diaphragm, its low inertia necessitates a light, strong design, but requires also a construction having extremely low thermal inertia. It is therefore vulnerable to peak values of heat flow of extremely short time duration. A larger margin between operating temperature and melting point of the diaphragm material than is employed in the relatively heavy chamber walls is therefore imperative.

A safe maximum temperature of 1200°F for the commonly used diaphragm materials, such as stainless steel, nickel, or inconel, will be specified. Thus, with chamber temperatures ranging from 4000°F to 8000°F, the resulting high temperature differential constitutes a heat driving potential far in excess of that employed in general engineering designs. Furthermore, the high turbulence and recirculation resulting from the highly active combustion causes abnormally large values of convective heat transfer rate and values of 15 BTU/sq.in/sec, are common with values to 25 BTU/sq.in/sec occurring under unstable burning conditions.

To evaluate the design parameters for transducers, the method of proceeding will be detailed. Figure 5 shows a diagram in which the heat flow q is shown opposite the temperatures encountered in its passage from the rocket chamber into the coolant. The boundary layers on each side of the diaphragm are shown, as well as the temperature rise through the diaphragm. If temperature stability has been reached, q into the transducer must equal q into the coolant and the heat transfer from the gas in the chamber to the coolant is:

$$\frac{Q}{A} = q = hc\,(Tdl - Tl) = K\left(\frac{Tcd - Tdl}{t}\right) = hg\,(Tc - Tcd) \qquad (17)$$

where Q = total heat transfer rate

A = area

q = heat transfer rate (BTU/sq.in/sec)

Tc = chamber gas temperature ^{o}F

Tcd = diaphragm temperature, chamber side, ^{o}F

Tdl = diaphragm temperature, liquid side, ^{o}F

Tl = coolant temperature ^{o}F

hc = film coefficient, coolant

hg = film coefficient, gas

k = thermal conductivity of diaphragm material

To determine the cooling capacity of a particular transducer design, it will be necessary to evaluate each of the terms in Equation (17). To accomplish this, consider first the boundary layer between the diaphragm and the coolant:

$$q = hc\,(Tdl - Tl) \qquad (18)$$

where hc = heat transfer coefficient

Tdl = inner diaphragm temperature

Tl = coolant temperature

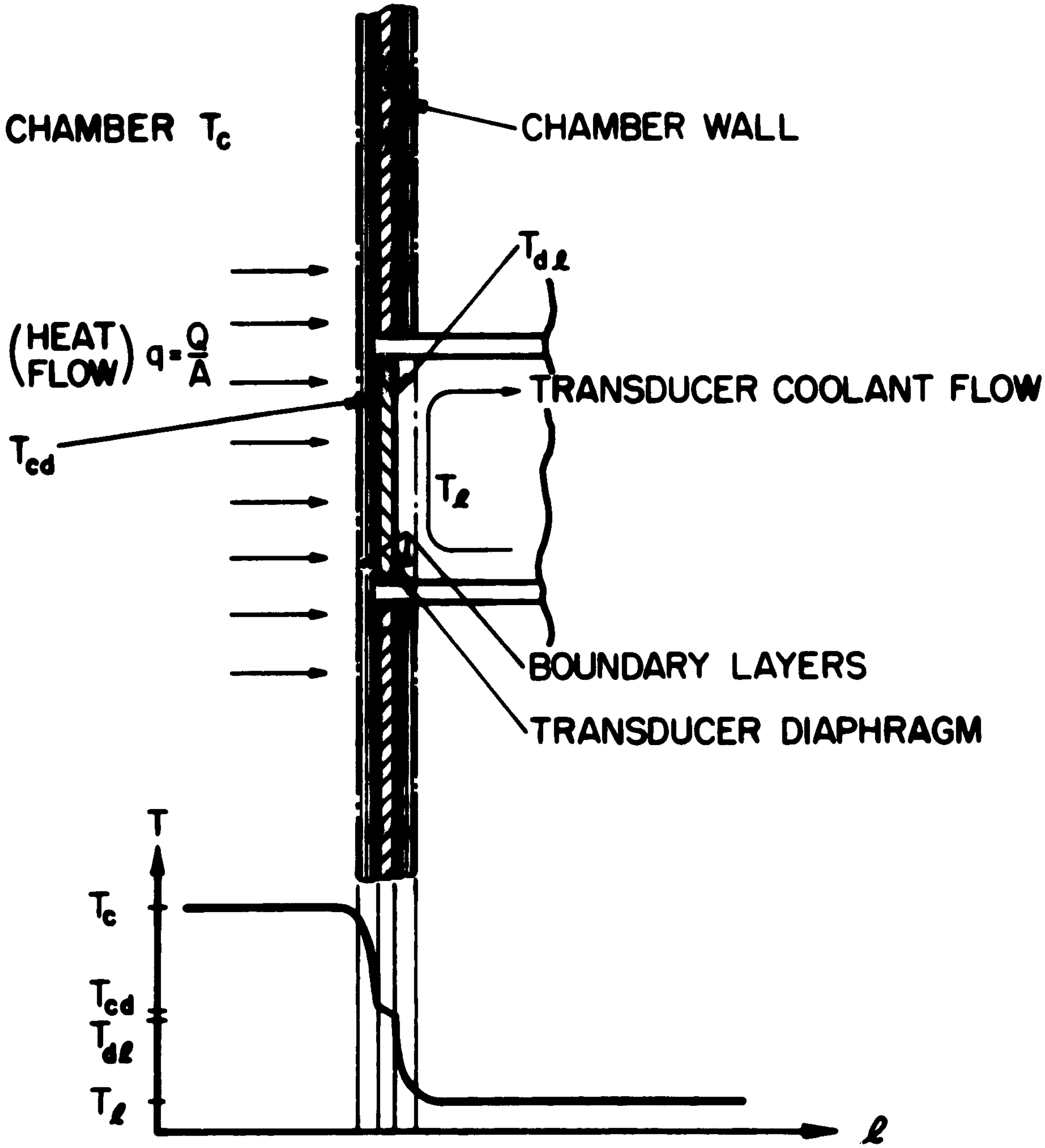

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FIGURE 5

The evaluation of hc has been the object of much investigation and experimentation. The coolant bulk characteristics, the flow conditions, and the character of the wall surface all affect its value. When considering heating of a liquid, flowing through a tube of cylindrical cross section at low heat flows the following relationship will give approximately correct values for fully developed turbulent flows:

$$hc = .0225 \frac{K}{D} \left(\frac{DV\rho}{\mu}\right)^{.8} \left(\frac{c\mu}{K}\right)^{.4} \tag{19}$$

where

K = thermal conductivity of the fluid

D = diameter of the stream

$\left(\frac{DV\rho}{\mu}\right)$ = Reynolds number

$\left(\frac{c\mu}{K}\right)$ = Prandtl number

This is the classic expression used in nonboiling heat transfer calculations, as in the case of a cooling tower, or condenser (Ref.6).

The case of a transducer to be used in a rocket chamber is a far different one, since the heat flux is sufficiently high that heat transfer will probably take place with the pressure of the coolant below its critical pressure, the coolant pressure being limited to some reasonable value determined by the diaphragm configuration. This results in the phenomenon of nucleate boiling wherein large quantities of microscopic gas bubbles serve to agitate and mix the coolant so that temperature gradient is flat across the flow profile.

Figure 6 shows a family of curves relating q to various conditions of the coolant. It will be observed that as q increases for a given coolant velocity and pressure, the coolant temperature increases along the nonboiling lines determined by Equation (18) until the nucleate

HEAT TRANSFER TO H_2O NON-BOILING ~ SIEDER-TATE EQ
NUCLEATE BOILING ~ BASED ON EXPERIMENT
UPPER LIMITS ~ GUNTHER JPL PR (FOR 60°F BULK TEMP.) 4-120

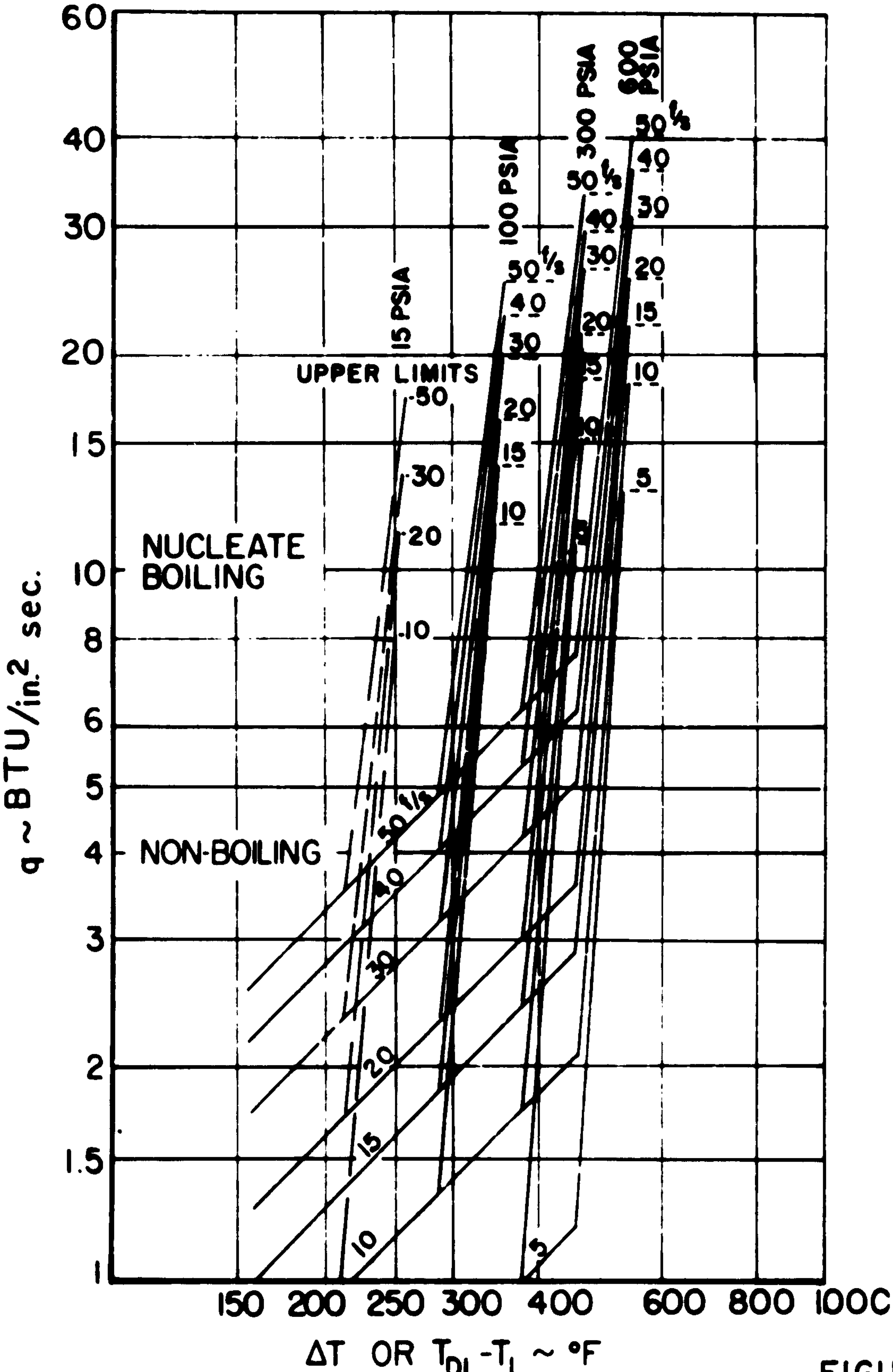

FIGURE 6

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line departs from the nonboiling line. At this point very little further temperature increase occurs, so that T_{dl} is essentially fixed, between a fairly low value of q up to the value where the generation of steam blocks further cooling.

Temperature change across the diaphragm, $(T_{cd} - T_{dl})$, is a simple calculation of solid conductivity, except that the temperatures encountered are sufficiently large that the appropriate values of thermal conductivity must be chosen:

$$T_{cd} - T_{dl} = \frac{qt}{K} \tag{20}$$

where t = thickness

K = thermal conductivity in appropriate units

Figure 7 shows the temperature drop vs q for 304 stainless steel and for nickel.

In the rocket chamber, the calculation of heat flow again is governed by Equations (18) and (19). However, in a rocket chamber large temperature gradients exist near the wall, and as a result the properties of the gas are rapidly changing with distance from the wall. The conditions of turbulence, the point at which a transducer is installed relative to the distance from the nozzle, the propellent combination, and the possible presence of high frequency instability all introduce large errors in the establishment of the coefficient, h_g. The designer of a transducer, furthermore, has no control and very often no way of obtaining information as to the expected application of the new design. Since the value of q is the figure most likely to determine the success or failure in the application of a given design, and is also likely to be known to the designer directly, it will be sufficient for this purpose to assume heat input,

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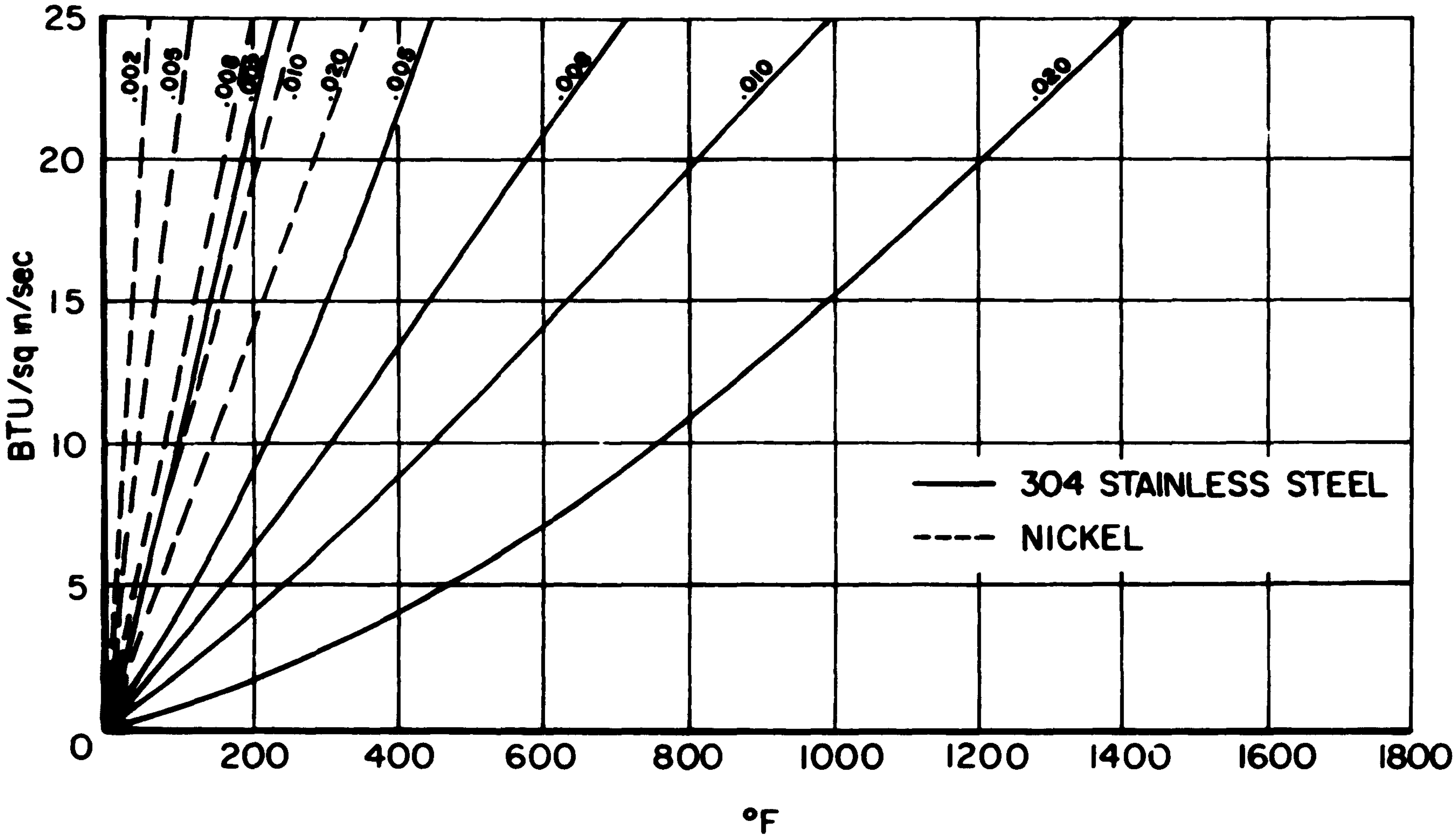

TEMPERATURE DIFFERENTIAL VS HEAT FLOW

FIGURE 7

independent of the value of coefficient.

The value of q in a rocket chamber can vary over wide limits ranging from 0.5 BTU/sq.in/sec in the gas generator, to 25 BTU/sq.in/sec using the more active propellants accompanied by combustion instability. It is important to observe that Tc is only one of several factors that determine the cooling requirements of a transducer. It is probably best, then to consider only the relative effect on q of the several factors affecting it.

With a given set of operating conditions, Bartz has shown, (Ref.7), a five-fold increase in q as the point of measurement is moved through the contraction of the chamber into the nozzle. It is obvious, therefore, that only under the most unusual conditions would one desire to install a pressure transducer in the nozzle. The value of q is also greatly affected by unstable combustion. It has been shown (Ref.8), that fully developed combustion instability may cause a three-fold increase in q . Obviously the requirements for dynamic pressure measurement are greatest when combustion instability is present or suspected, and so capability of the transducer must be determined in the most stringent manner. The combustion temperature will exert some effect on q because of the essentially fixed value of Tcd resulting from the material limitations, and so propellant combinations, such as hydrogen-flourine, having Tc = 8000°F will generate heat fluxes about 150% those of alchol-oxygen systems, if it is assumed that the diaphragm is at its maximum temperature in both cases. In general, also, higher energy propellants appear to exhibit greater tendencies toward combustion instability, so as the difficulty of attaining a suitable design increases, so too does the need for such a design. The specifications set up under this contract to establish the goal toward which the design of a transducer should be aimed give ultimate q as 25 BTU/sq.in/sec. It is felt that this value can be reached in practice and the following section will demonstrate how such a design can be accomplished.

B.. Practical Design

Let the value of q for the point in the rocket chamber wall where the flush mounted transducer will be located be 25 BTU/sq.in/sec, and T_c = 5500.

Since all heat transfer must eventually end up in the coolant, it is best to establish the required coolant pressure and velocity first. A reference to Figure 6 will show that a heat flow of this magnitude represents rather a fearsome challenge. A flow velocity of at least 50 ft/sec at 100 psia is necessary at the point in the exposed diaphragm where velocity and pressure are lowest. Furthermore, even these values are marginal, and any momentary increase in q or interruption of coolant flow will result in immediate burnout. For sake of the design, however, take these values. At this design point, $T_{dl} - T_l$ = 360° from Figure 6. If bulk temperature is assumed to be 60°F, T_{dl} = 420°F. A diaphragm thickness and material must now be chosen. At 25 BTU/sq.in/sec, using stainless steel, temperature limited to 1200°F, the maximum temperature rise permitted across the diaphragm is $1200^\circ - 420^\circ = 780^\circ$F. The thickest material of which a diaphragm can be formed, and not exceed 1200°F is (by interpolation) .0025" Therefore, let the diaphragm be formed of this thickness 304 stainless steel. The gas side temperature will then be 1200°F at 25 BTU/sq.in/sec. The procedure outlined is, of course, somewhat undependable because of the several assumptions necessary. It may be useful, however, to make this first [illegible] so as to determine the effect of design changes. It must also be understood that the above calculations, and estimates apply only to a point on a homogeneous wall. Furthermore, it is important to note that this is the poorest cooled and weakest point. Stagnation of flow at any point in the coolant passage, cavitation, or isolation of any point from the coolant, due to the passage configuration will cause a point of limited heat transfer capability.

Referring to Figure 6, it is apparent that increases in coolant flow velocity are more effective, other factors being equal, than changes in coolant pressure. It is important, on the other hand, to observe the effect of back pressure in the lower pressure ranges. It is probable that 200 psi is a reasonable maximum coolant pressure to maintain. Since this will require a considerably higher coolant supply pressure, further increases will require more elaborate pumping equipment than would be practical. A velocity in excess of 50 ft/sec will result in excessive pressure drop or an excessive flow rate. The absolute limit, then, of q transferred into water appears to be about 30 BTU/sq.in/sec, where the value of Tdl-Tl = 400°F which leaves an uncomfortably small factor of safety, to allow for cavitation, and points of stagnated flow. The next step to investigate is the diaphragm construction. Since 1200°F is the maximum temperature permitted in the diaphragm a drop across the diaphragm of 800°F limits diaphragm thickness of stainless steel severely, but if pure nickel were used, a drop of only 200° will occur at .008 thickness and 25 BTU/sq.in/sec. The larger thermal conductivity of nickel, as compared to stainless steel, limits the temperature rise in the diaphragm for the same heat flux.

The original value of hg can be derived from the initial conditions:

$$hg = \frac{q}{Tc - Tcd} = \frac{25}{5500-1200} = .0058 \text{ BTU/sq.in/sec/}^{\circ}\text{F}$$

When the diaphragm material is altered, no change in h_g will result so;

$$q = .0058\ (5500-600) = 28.4\ \text{BTU/sq.in/sec}$$

A small net increase in heat transfer results, because of the larger gas-side temperature gradient.

There are two facts apparent, however. The first is that the diaphragm may now be made thicker, and more durable (at some sacrifice, however, in frequency response due to added mass). This has the advantage, if the diaphragm is .015" thick (see Fig.7), of permitting a certain amount of transverse heat flow away from hotter areas in the diaphragm into the cooler portions, thereby tending to equallize temperatures across the diaphragm. The other fact is that if some material, such as a ceramic, can be placed as a coating on the diaphragm a very different set of figures is indicated. Figure 8 shows the temperature drop across three thicknesses of zirconium oxide and aluminum oxide. If a coating is sprayed on the diaphragm, the large temperature drop permitted in the ceramic layer decreases the temperature difference across the composite diaphragm. The actual drop in the nickel can now be neglected, the temperature on the liquid side is essentially constant as shown by Figure 6, so at 25 BTU/sq.in/sec the temperature drop in the ceramic, coated .002" thick is 3150°. T_{cd} is 3150 + 350 = 3500°.

If h_g and T_c are constant, while the surrounding wall is receiving a heat flow of 25 BTU/sq.in/sec, the diaphragm receives a lower value because of its far higher surface temperature.

To determine this value of q, successive approximations between the heat flow equation, and Figure 8 are made:

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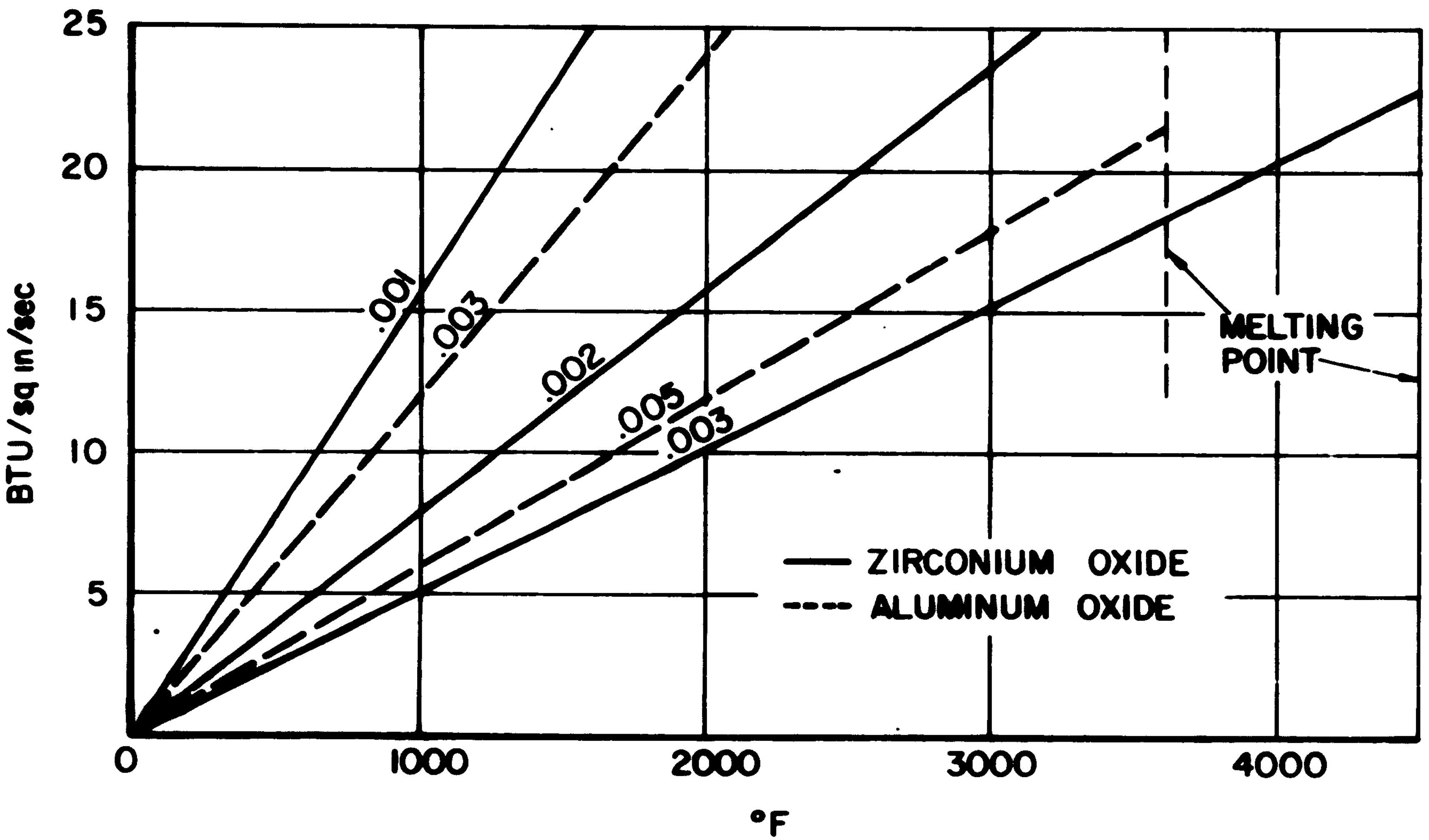


TEMPERATURE DIFFERENTIAL IN CERAMIC COATINGS, VS q

FIGURE 8

Assumed q	Temp.drop Fig.8	Tcd	.0058 (5500-Tcd)
25	3150	3500	11.6
11.6	1500	1850	21.2
21.2	2650	3000	14.5
14.5	1850	2200	19.3
19.3	2440	2790	16.2
16.2	2050	2400	18.0

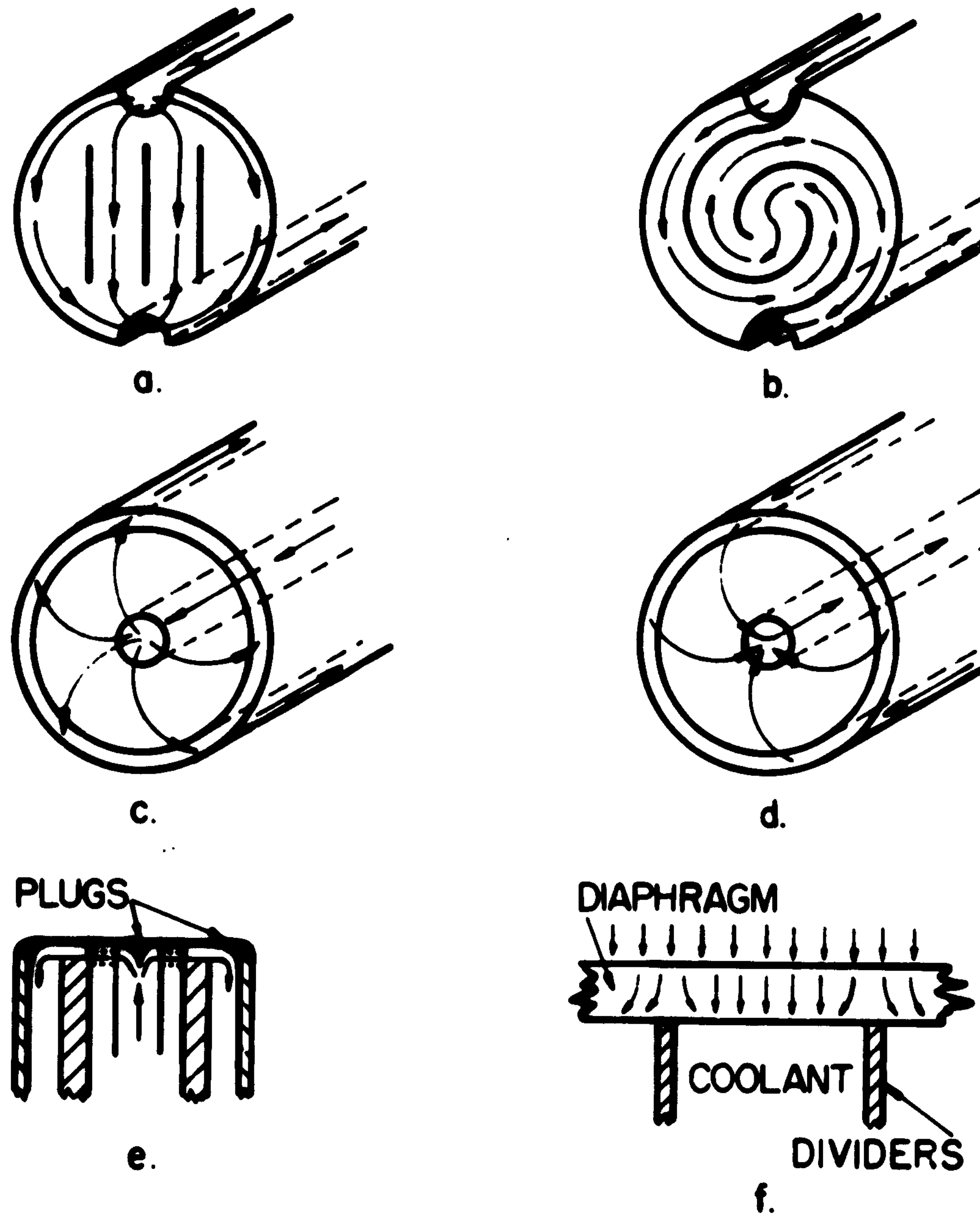
The results indicate about 17 BTU/sq.in/sec into the diaphragm. The use of zirconium oxide permits a much more conservative use of the coolant heat transfer. No presently known transducer diaphragm is so constructed. Existing transducers, having 15 BTU/sq.in/sec heat transfer ability, can, by ceramic treatment .003" thick operate to 22 BTU/sq.in/sec, calculated by similar methods.

The ceramic treatment with aluminum oxide permits a somewhat thicker coating, which would be more durable. The use of a nickel diaphragm as a base will permit a thicker material to be used so a rougher surface permitting better adherence of the ceramic can be attained. The reduction to practice of this technique as applied to a pressure transducer is a subject for future work.

It is evident from the foregoing that a flush diaphragm transducer must have low pressure drop in its connecting tubing, uniform high velocity at all points in the diaphragm area where the heat is concentrated and back pressure applied to the outlet so that no point in the exposed section falls below the predicted minimum pressure. Of importance is the fact that coolant velocity must be high at all points exposed to the heating. The introduction of coolant without having stagnant areas is complicated by the fact that the water enters in a direction normal to the diaphragm surface, actually representing a nodal point in the velocity distribution.

Although extreme turbulence can result in raising the overall pressure drop in the transducer, it can help at these points to eliminate the hot spot. Another method of achieving the necessary increase of heat transfer capacity is to provide small plugs, Figure 9e, of gold or other high, conducting material, to transfer the heat to the coolant where it is moving and fill in the spots where it is stagnant.

Figure a-d, shows four methods of cooling a flush diaphragm. These are the only methods available of supplying coolant to the diaphragm surface. They all have nodal velocity areas and are subject to burnout in these spots. Figure 9a, the linear cooling results in low velocity around the edges, and also low velocity at the inlet and outlet. Figure 9b, spiral cooling, usable only with double diaphragm transducer can have constant velocity except at the inlet and outlet, providing that the passage area is kept constant when the spiral dividers between diaphragms are installed. It has fairly high pressure drop. In Figure 9c and 9d, coaxial cooling is shown. This is the least desirable, because of the decreasing velocity as distance from the center increases, unless baffles behind the exposed diaphragm are installed with decreasing spacing at the edges to keep velocity constant.



TYPICAL COOLING PASSAGE CONFIGURATION

JPR 1205

FIGURE 9

V. TRANSDUCTION

So far no reference to the conversion of the dynamic displacement of the transducer elements into electrical signals has been made. A number of methods exist to accomplish this end. They all have advantages and disadvantages, and exhaustive discussion of each is available from a number of sources. Only a sufficient discussion will be included here to demonstrate working principle and explain evaluation technique.

A. Strain Gage

The strain gage transducer is undoubtedly familiar to those working with transducers (Ref.9). The strain gage is normally used in a Wheatstone bridge circuit with one, two, or four active arms. The sensitivity of a strain gage described as:

$$\frac{\Delta R}{R} \div \frac{\Delta L}{L} = \text{gage factor}$$

The gage factor is a function of the wire of which the gage is fabricated normally on the order of 2.

The resistance of the gage should be high enough so that lead wires do not constitute an appreciable portion of the total. About 350 ohms per leg seems common. The temperature coefficient of resistivity must be low for transducers subjected to the severe environment of rocket chamber testing, and compensation for zero drift with temperature must be allowed for by complementary gages being located in the same ambient condition. This is accomplished in many transducers by winding wire both axially and circumferentially to form a bifilar winding on the outside of a tube which is then loaded by the force from the diaphragm in an axial direction, Fig.10a. The windings can be divided so that there are four

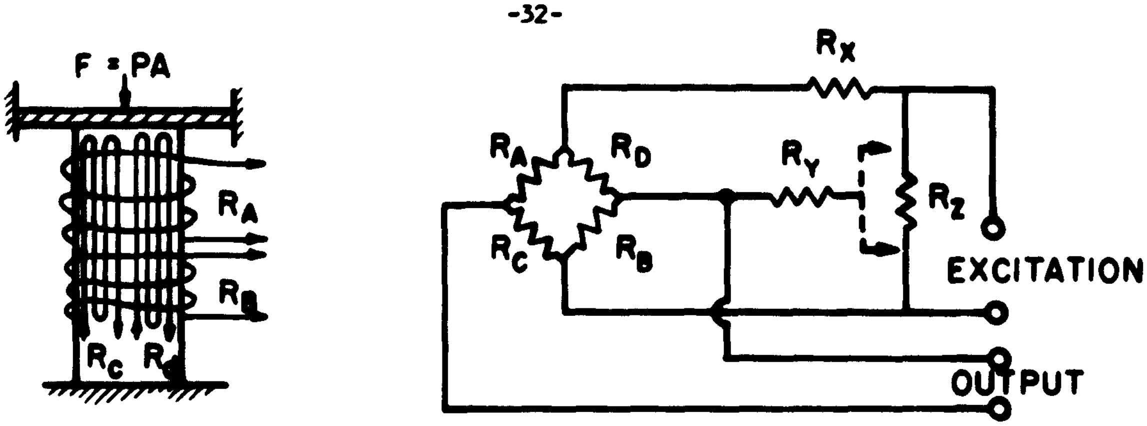


FIGURE 10a - STRAIN GAGE

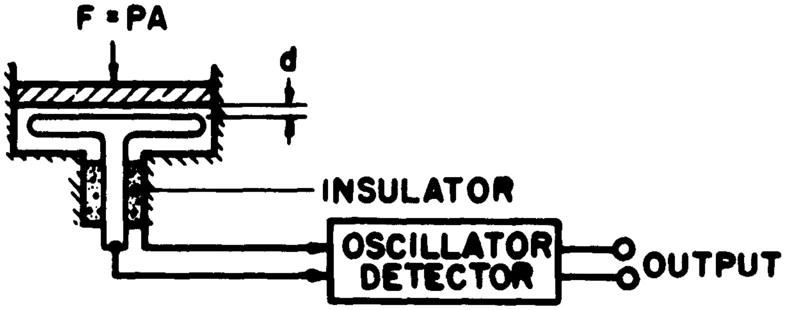


FIGURE 10b - CAPACITIVE

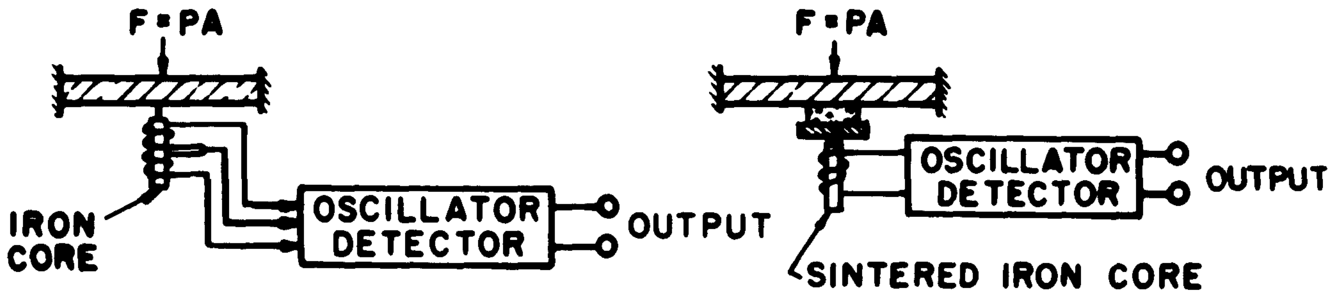


FIGURE 10c - VARIABLE RELUCTANCE

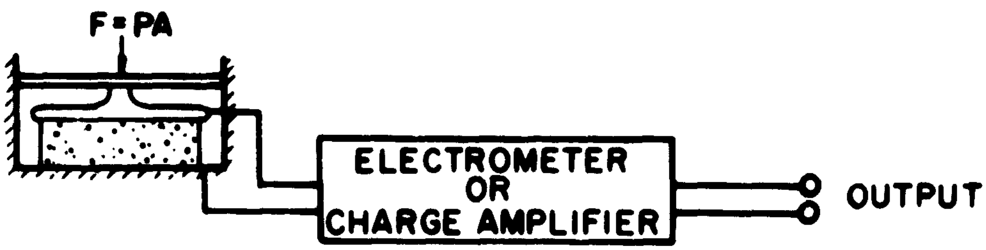


FIGURE 10d - PIEZOELECTRIC

METHODS OF TRANSDUCTION

legs to complete the bridge. When a tube is so loaded, all four arms are active, but the axial compressional strain does not equal the circumferential expansion strain by the factor μ where μ is Poisson's ratio, about .25. It therefore is apparent that the strain gage bridge does not have four equally active arms but two that decrease by $-\Delta \ell$ with a certain compressive strain, and two which increase by $.25\Delta\ell$.

The greatest disadvantage of any strain gage pressure transducer is its relatively small voltage output. The voltage output is, naturally, dependent upon the input excitation, which in turn is ultimately limited by the safe power handling capacity of the gages but often is limited to some lower value because of drift due to excessive temperature change. Usually the full scale output voltage is less than 50 mv. When dynamic pressures are being measured, a wide band-pass is necessary in the recording system, which includes the frequency spectrum of many of the common high intensity interferences resulting from power line noise and switching transients. Also the requirements often indicate a higher sensitivity for channels recording dynamic pressure data, and the signal-to-noise ratio obtainable under such conditions, even when the most caréful shielding, AC isolation, and grounding has been done, is very often rather low. The stability and repeatability of a strain gage system is excellent, however.

Circuitry of bridge and compensation for changes due to temperature in Young's modulus, and adjustment to exact zero balance may be seen in Fig.10a.

It will be seen that Rc and Rd decrease with pressure while Ra and Rb increase. With change in temperature, however, all change equally, maintaining bridge balance. Resistance Ry is supplied at manufacture with the transducer to make the final zero balance trim. Resistance Rx is in series with the bridge supply, and since it is a resistance having a large temperature coefficient is cut to value to compensate for changes with temperature in Young's modulus of the restoring element. Rz is simply to trim the input resistance of the bridge to some fixed value. Because of its simplicity and ruggedness, the strain gages method is practically the standard of the field.

The use of unbonded gages is usually impractical in a transducer designed for measuring dynamic pressures because of large vibrational sensitivity, and difficulty in attaining sufficiently high frequency response. Damping by means of oil filling renders some improvement, but in general, the results with such transducers subjected to vibration are very poor.

Efforts have been made to achieve a larger output from design of the elements of the supporting structure, (not a notable success) and through the use of silicon semi-conductor strain gages, Ref.10. Semi-conductor strain gages are capable of output on the order of volts, having an uncompensated gage factor of about 130, compared to 2 for wire gages. As the zero drift is relatively higher than wire gages, the compensation required to achieve proportional stability results in factors of about 50, still a considerable improvement in sensitivity.

The increased output of the semi-conductor gages will permit greatly improved signal-to-noise ratios in a long transmission line. Unfortunately, however, the greater output is accompanied by a proportionately greater thermal coefficient, and so the use of semi-conductor gages to deliver an equivalent output to wire gages with a less sensitive and stiffer structure (as required for faster response) is not likely to be successful at this time at least.

One obvious advantage of semi-conductor gages is the availability of complementary units, which through suitable fabrication can exhibit approximately equal positive and negative gage factors under the same stress. This permits truly compensated bridge circuitry having four equally active arms.

B. Capacitive Transducers

The diaphragm used in a capacitive transducer is one plate of an air dielectric capacitor, the interelectrode spacing of which varies in response to a variation in pressure, see Fig.10b. This in turn causes the capacitance, represented by the transducer, to change with pressure. The capacitance of such an assembly is

$$C = .224 \frac{KA}{d} \tag{21}$$

where

K = dielectric constant = 1 for air

d = spacing (inches)

A = area (sq.in.)

As may be seen, from (21) if Δd is proportional to $\frac{1}{\Delta P}$, which is a good assumption for small displacements, ΔC bears a hyperbolic relationship to pressure and, unless d is large compared to Δd serious non-linearity will occur. As the diaphragm area is small, C must be small and to obtain a reasonable value for ΔC the spacing must also be small.

Improvement in linearity at no sacrifice in output will result from the use of a composite dielectric. The air gap is partially filled with mica, K = 8, of a suitable thickness (Ref.11).

The advantage of this type of transducer is the relative simplicity and ease of manufacture, particularly for the coolant passages, and the great ruggedness of a transducer so constructed. When combined with appropriate detector equipment, a large output with good signal-to-noise ratio is developed. The disadvantages are its inherent non-linearity, its requirement for specialized signal conditioning equipment, and the fact that, since the diaphragm also represents the restoring element, larger drifts due to heat flow and resultant temperature change,in both zero and sensitivity are bound to occur then when the restoring element is remotely located, isolated from large temperature excursions.

Owing to the inherently low value of ΔC in response to pressure, the sensitivity of the capacitance measuring unit is necessarily high and consequently a high value of zero drift within the detector-oscillator system is common in transducer systems of this type. The cable connecting the transducer to the detector, although operating at a low impedance is likely, also, to result in large values of zero shift with cable temperature and will cause variation of output with vibration.

C. Variable Reluctance

A number of transducers are presently available, using variable reluctance transduction, none, however with flush diaphragm, fast response construction. A brief discussion, for the sake of completeness, is in order, nevertheless, because the present state of the art permits such high excitation frequencies that a transient response of 30KC or better is practical. A subminiature distance detector* is commercially

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*Bently D152

available and has been incorporated into the design for an experimental transducer, of great simplicity and ruggedness, although reduction to practice is presently incomplete. It appears from preliminary experiment to permit relatively high response accompanied by small size, and inexpensive fabrication. An excitation rate of 1.0 megacycle is used and consequently, transient response is not limited by excitation frequency.

Advantages of such a variable reluctance system are the large output with good linearity available, the ruggedness and relatively low output impedance, and simple and inexpensive transducer construction. Disadvantages are the necessity of an excitation generator and detector, (which in the case of the experimental system aforementioned is solid state and small, but requires 15vDC, well regulated) the effect of large masses of magnetic material in the vicinity of the transducer, the fact that it is sensitive in some degree to magnetic fields, and difficulty of compensation for zero drift. (Several small variable reluctance transducers of the cavity type are available, which can give limited transient response, but no flush, water-cooled diaphragm transducers operating on this principle are available commercially).

Two configurations of variable reluctance displacement sensors are shown, in Fig.10c, the differential type and the single ended type. The latter was incorporated in an experimental transducer and voltage displacement curves for various diaphragm materials are shown in Fig.11.

D. Piezo Electric

The generation of potential as a result of a force applied to a crystalline dielectric has long been known as a method of measuring pressure fluctuations. It has only been a fairly recent development, however, which made a really practical transducer system possible, which

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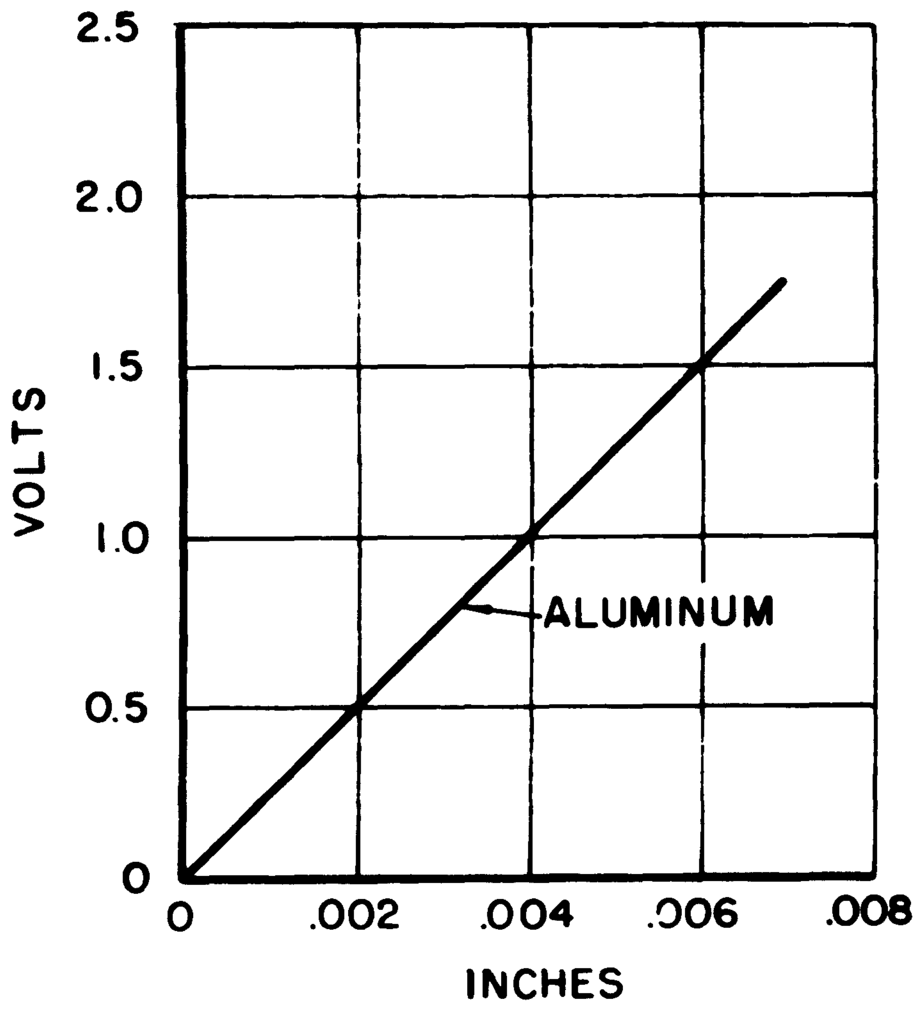
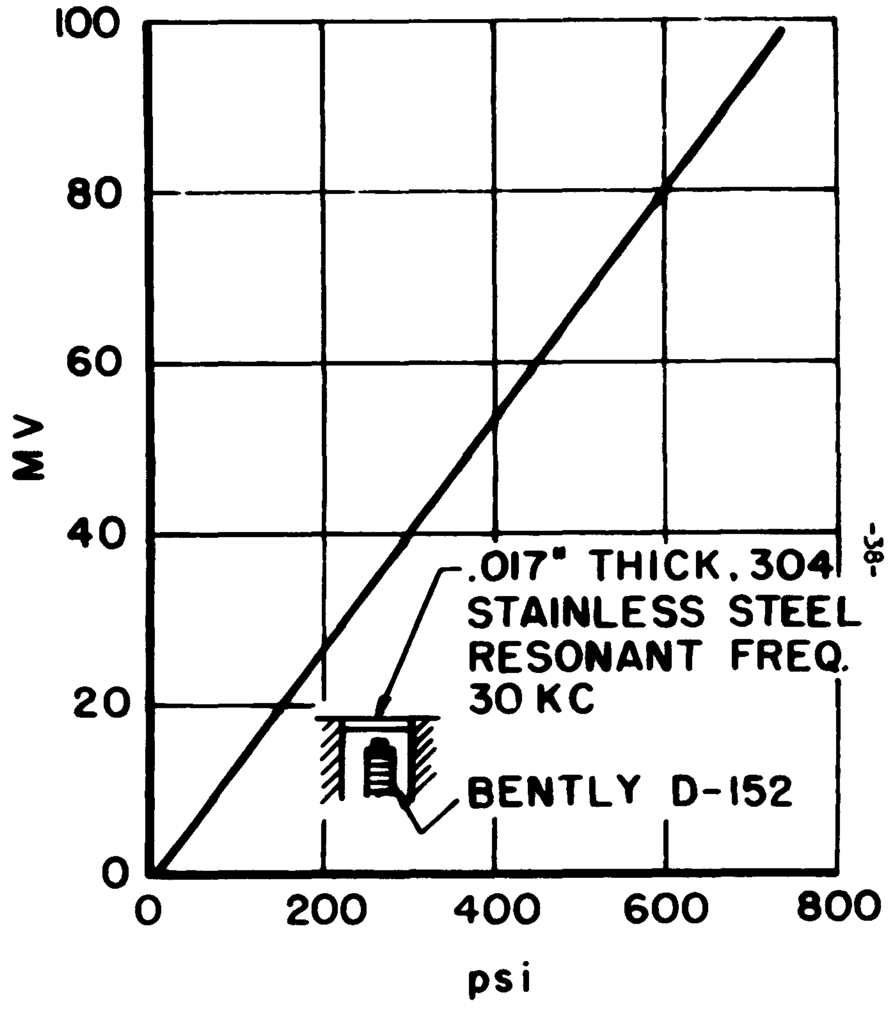


BENTLY D-152 DISTANCE DETECTOR
OUTPUT (V) VS DISTANCE



FLUSH DIAPHRAGM TRANSDUCER
OUTPUT (mv) VS PRESSURE (PSI)

FIGURE 11

can measure steady-state as well as fluctuating pressure.

The use of quartz crystals has been almost universal in transient pressure transducers of the type being discussed, and so although barium titanate, Rochelle Salt, and "ADP" are other active cyrstalline materials, these are all thermally less stable, and have relatively poorer insulating qualities than quartz.

Quartz crystals have asymmetrical charge distributions and when the lattice is distorted, the displacement of the internal charges produces external charges of opposite but equal value which are detected by connections to electrically conducting plates to the outside of the crystal, as shown in Fig. 10d.

A quartz element is a slab or "cut" from a crystal. The magnitude and polarity delivered by a quartz slab is determined by the relation existing between the direction of cut and the major axes of the crystal from which it is taken. The output magnitude from a quartz crystal is a function primarily of the quartz itself, and so, once finished and polished, very little further change in charge sensitivity is to be expected.

Either voltage or the charge may be measured by means of the speciallized equipment normally used with the quartz transducers. When a force, F, is applied to a crystal the charge so generated is:

$$Q = dF$$

where d is a constant of the quartz from which the slab was cut.

$$d = 2.25 \times 10^{-12} \frac{\text{coulomb}}{\text{newton}}$$

A voltage E results:

$$E = \frac{Q}{C}$$

where C is the capacitance between the electrodes

Obviously, since an imperceptible change in C can result from application of a force, the output voltage must be directly proportional to Q.

Quartz crystals as normally cut and incorporated in transducers (e.q. EBL-Liu, and Kistler) have a very large output voltage when force is applied. To utilize the voltage effectively, a device having a large input impedance is essential, since very small amounts of energy are present in the measuring element.

The use of electrometer type amplifiers, which have an input impedance of 10^{14} ohms, results in a sufficiently large time constant so steady state readings are possible, providing proper techniques of transducer construction, maintenance, and installation are observed. Such techniques are necessary to maintain insulation resistance at a value comparable to the input impedance of the amplifier. A time constant on the order of an hour or more can be expected under the correct operating conditions.

The capacitance of a small quartz element is on the order of 5 pfd. When a length of cable is added, it is essentially in parallel with the transducer and consequently a charge generated by a pressure will result, because of the added capacitance, in attenuated voltage output. The cable length permissible is therefore limited. To overcome this difficulty an interesting variation on the electrometer voltage amplifier is the charge amplifier.

The charge amplifier is a high gain dc voltage amplifier having capacitive feedback from the low output impedance to the high input impedance. The output voltage occurring when a charge appears at the transducer is coupled back to the input circuit, resulting in a net voltage at the input of the amplifier of essentially zero. The change in charge from the transducer is now transferred into the feedback capacitor and so the net voltage across it is equal to the charge divided by its capacitance. It is then a simple matter to read this voltage and produce a proportional voltage at the output of the amplifier. The cable length, since no net change in charge on it has occurred, does not now affect the sensitivity of the system.

The advantages of a quartz transducer are its stability in sensitivity, its ruggedness, its small size, high voltage output, simplicity, and ability to withstand exposure to temperatures as high as 600°F. (Temperatures in this range cause temporary deterioration of the insulation resistance, but no permanent harm). The transient response of a quartz transducer is extremely good, since the measuring element is so stiff as compared with other types. The disadvantages of a quartz transducer are the very high internal impedance required to permit steady state measurement and calibration, therefore, the requirement of special electrometer or charge amplifiers to permit really satisfactory results, and the necessity for carefully installed special cabling, kept clean and dry at all times. It is impossible to achieve really long-time stability of zero, although sensitivity is constant because of the aforementioned inherent charge sensitivity being a constant of the quartz material.

VI. PRACTICAL APPLICATION OF TRANSIENT PRESSURE TRANSDUCERS IN ROCKET CHAMBERS

Previous sections have discussed various facets of the flush diaphragm transducer and associated equipment. It is imperative that those who will put the transducers to practical use have this information available not only in general, but in detail as required for the particular design chosen. It is equally important that the chamber operating conditions be known before the test as it is therefore often possible to take steps to minimize the failures which so often occur and which have given the flush diaphragm transducer a wide reputation of undependability in practice.

Transducers of the particular type under discussion are always a compromise, with the requirement for extremely efficient cooling being of utmost importance. Such transducers are necessarily elaborate in their internal design, and are often subject to high hysterisis or other measuring errors. Good design and test practice would dictate that only transient pressures be recorded by means of such transducers. Normally, the heat flows into the transducer, even with water cooling, are of such values that large excursions of temperature occur, causing excessive errors which cannot be tolerated if the steady state operating parameters are to be recorded simultaneously. A transducer of the cavity type, being isolated by pressure passages of such length that no temperature change occurs will invariably give more accurate steady state data. Inaccuracies in data from flush diaphragm transducers fall into two categories, steady state errors and dynamic errors. Although the causes of these are interrelated, they will be discussed separately.

A. Steady State Errors

One universal complaint regarding the performance of these transducers is the change in sensitivity which results from heating. In all designs of flush diaphragm devices, the diaphragm itself represents between 10% and 100% of the overall restoring force of the transducer. In strain gage and piezoelectric transducers, the measuring element constitutes the larger portion, but in the variable reluctance or capacitive type, all the restoring force is exposed to the full heating present at that point in the chamber wall. Young's modulus for the materials normally employed in diaphragms, such as 304 stainless steel, decreases with temperature at about 1% for each 40°F. The decrease in the value of the portion of restoring force exerted by the diaphragm is therefore 1% per 40°F. Transducer sensitivity is affected in direct proportion to the portion of total stiffness represented by the stiffness of the diaphragm. In those transducers having isolated measuring or strain elements, the measuring element also is not immune to the temperature rise occurring due to diaphragm heating and consequently it too changes sensitivity, although the temperature rise is limited to a lower value because of isolation and cooling. Strain gage transducers normally are compensated for temperature induced sensitivity changes by means of resistances having large resistance-temperature coefficients, but these are not necessarily subject to the identical short-time temperature variations seen by the diaphragm and measuring element, and so can only compensate for slow changes in ambient temperature. Sensitivity changes due to heating of the transduction occur also, such as may be caused by change in gage factor with temperature, or change in temperature of the dielectric in a capacitive transducer.

Zero drift, when transducers are used only for dynamic pressure measurement is not as serious as sensitivity drift which will affect both steady state and dynamic data. It is caused by thermal expansion of the diaphragm resulting in warping, and by thermal expansion of the transducer body resulting in a change in preload on the measuring element by the diaphragm restoring force. In a capacitive transducer, the result of case expansion is to increase the air gap, causing a negative drift.

Zero drift in the measuring element can be minimized either through compensation (as between bridge arms in the strain gage element) or temperature stabilization by the incoming coolant flow.

One extremely likely cause of sensitivity error and/or zero drift is overtightening the transducer upon installation. The measuring element in many transducers is supported by the case and stresses are set up between the support of the measuring element and the diaphragm, causing changes of preload.

Coolant pressure may cause both sensitivity changes and zero drift because of distortion of the diaphragm. The usual effect of increased coolant pressure is to cause a zero drift in the direction of positive pressure because of the swelling of a double diaphragm and also a decrease of sensitivity caused by increased diaphragm stiffness. Some transducers, such as Norwood 118 have direct coolant pressure dependency, and output must be so corrected. Others have almost no error caused by coolant pressure.

B. Dynamic Errors

Dynamic errors result in the distortion of the recorded waveform, or the introduction of a nonperiodic component in the output.

By far the most serious error in transducer output is caused by the inability of the vibratory system to follow a driving function faster than some (small) fraction of its resonant frequency. This was discussed under Section III, and obviously is a limitation of any mechanical device. The use of filters, properly matched to the transducer response can permit a transducer to reproduce an electrical signal most nearly identical to the driving function without ringing or overshot. Other electronic devices, permitting somewhat better response, will be covered in a later section. Amplitude and phase errors so introduced may be corrected for through the use of curves such as Fig. 4, if necessary, but use of a transducer of adequate frequency response is the proper solution.

Other dynamic errors occur which are not related to the transducer natural frequency, however. The vibration to which all transducers flush mounted on the chamber wall will inevitably be exposed, causes electrical output which may or may not be related in its source to the pressures seen at the diaphragm. If related, it can cause what appears to be distortion since it is in synchronism with the pressure fluctuations, or if not, a nonperiodic component will occur, resulting in lack of periodicity of the entire recorded pressure waveform.

Response of several transducers to such vibrational excitation is listed in a later section.

Another dynamic error is observed when the pressure driving function exceeds the linear range of the transducer, or the electronics associated with it. In its usual form, the extremities of the waveform are rounded off or clipped sharply, but occasionally a larger overshoot than can be accounted for by pressure alone or even an apparent negative pressure larger than an absolute vacuum will appear.

When coolant flow is increased to the necessary value, occasionally noise is caused by cavitation in the coolant passages. Usually it is possible to eliminate the noise by increasing coolant back pressure while maintaining flow rate through increased inlet pressure.

Electrical noise, including power supply, switch transients, amplifier noise such as microphonics in the electronics, and others, affects the output to the recorder of those transducers having low output levels. Methods of eliminating these are covered in Ref. 12.

Combustion pressure fluctuations, the result of the non-homogeneity of the propellents on injection and mixing, combined with any periodic pressure fluctuations in a waveform which is not examined over many cycles may appear to have no overall periodicity. When the distortions, such as ringing of the transducer, and electrical noise are considered and eliminated, careful analysis is necessary unless there is large unstable pressure amplitude of a periodic nature.

C. Burnout

Many transient transducers are offered for use in rocket chambers which are unable in spite of any reasonable coolant flow to withstand the necessary heat transfer rates encountered. When these fail, it is not unexpected. Failure of otherwise adequate transducers however occurs owing to several causes which can usually be eliminated.

The coolant pressure and flow into a transducer must be extremely dependable. In Section IV, the important relationship to ultimate heat transfer rate of coolant flow and pressure was covered. Coolant flow must be maintained regardless of water main pressure and so should either be provided from a pressure tank or a pump of adequate capacity.

Filters and strainers must be provided which have .01" mesh or smaller. The normal water systems have excessive rust and particle flow for use without filters, and will clog the coolant passages.

Oil film, inhibitor or antifreeze in coolant can set up a boundary layer in the coolant passages which can cut heat transfer rate into the coolant by as much as 50%.

Certain propellant combinations, and operational parameters can permit a highly oxidizing atmosphere to exist in parts of the rocket chamber.

The materials of which diaphragms are constructed, such as stainless steel are not able to resist locally severe oxidation in the presence of heat under these conditions. Since some diaphragms are thin already, it is possible to have pin-holes develop which will result in immediate burnout if appreciable coolant loss occurs. (This is particularly prevalent in solid propellant chambers.) The use of zinc chromate jelly or heavy silicone grease will sometimes prevent burnout when it occurs at the starting conditions, while gold plating the diaphragm has been successful in a greater number of cases.

Burnout is often caused by improper installation. A transducer should never project into the chamber since the projecting edge will reach full stagnation temperature through elimination of its boundary layer and the edges are always marginally cooled. All transducer installations should be carefully checked to prevent this, as repeated installation or over-tightening can crush the gasket and permit the transducer to project. Ocassionally it is possible to prevent burnout by recessing the transducer somewhat. It has been found that recessing 1/32" reduces heat transfer by 10%-20%.

Any leakage past the transducer will overheat the sides of the body where there is little or no cooling and can result in burnout through external failure of the transducer.

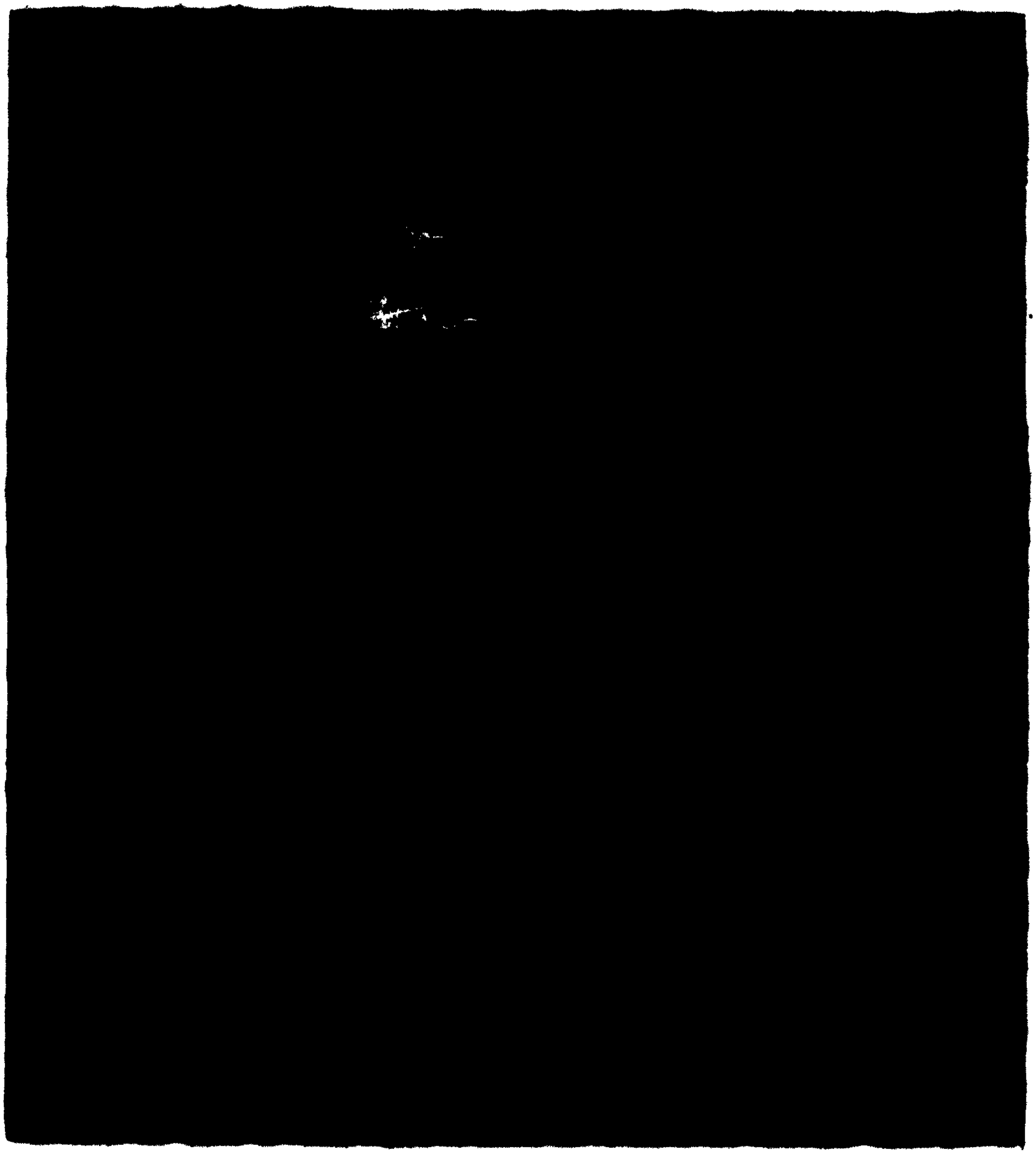
VII. EQUIPMENT FOR TRANSDUCER CALIBRATION AND EVALUATION

To describe the performance of any transducer, regardless of type or operating principle, there are two basic sets of parameters to be described, steady state and dynamic. It is the intent of this section to describe the tests and specific apparatus required to perform these tests. In addition to the normal set of data required, evaluation of the heat transfer ability of particular designs must also be included as an essential portion of the tests. Sections follow covering each of these.

A. Steady State Calibration

The obvious first step in checking performance of a transducer as received from the manufacturer is to obtain its steady state transfer function. Since only evaluation of transducers specifically intended for dynamic use is contemplated, calibration facilities are not as elaborate and sensitive as required to calibrate steady state transducers. The calibration installation to be described is accurate to 0.25%.

The calibration bench is shown in Figure 12, and its block diagram is shown in Figure 13. The source of pressure is at the right, where two gages and two pressure regulators provide known calibration pressures. (When higher accuracy is required a dead weight tester is available.) In the center section, an assembly was fabricated using a bridge power supply, and a Brown "Electronik" precision indicator interconnected as shown in Figure 14. Although this calibration device was primarily designed to operate with strain gage transducers, it is used easily with other types of transducers through suitable voltage dividers. The span of the indicator is 10 m.v. and five cascaded ranges, each of 10 m.v. permits a maximum input of 50 m.v. This is equivalent to a scale length of 25 feet, and the smallest increment is .01 m.v. .01% resistances were used throughout. Accuracy when used with strain gage transducers is assured by matching the slide wire voltage directly against a portion of the bridge voltage through a precise ratio



TRANSDUCER STATIC CALIBRATOR

FIGURE 12

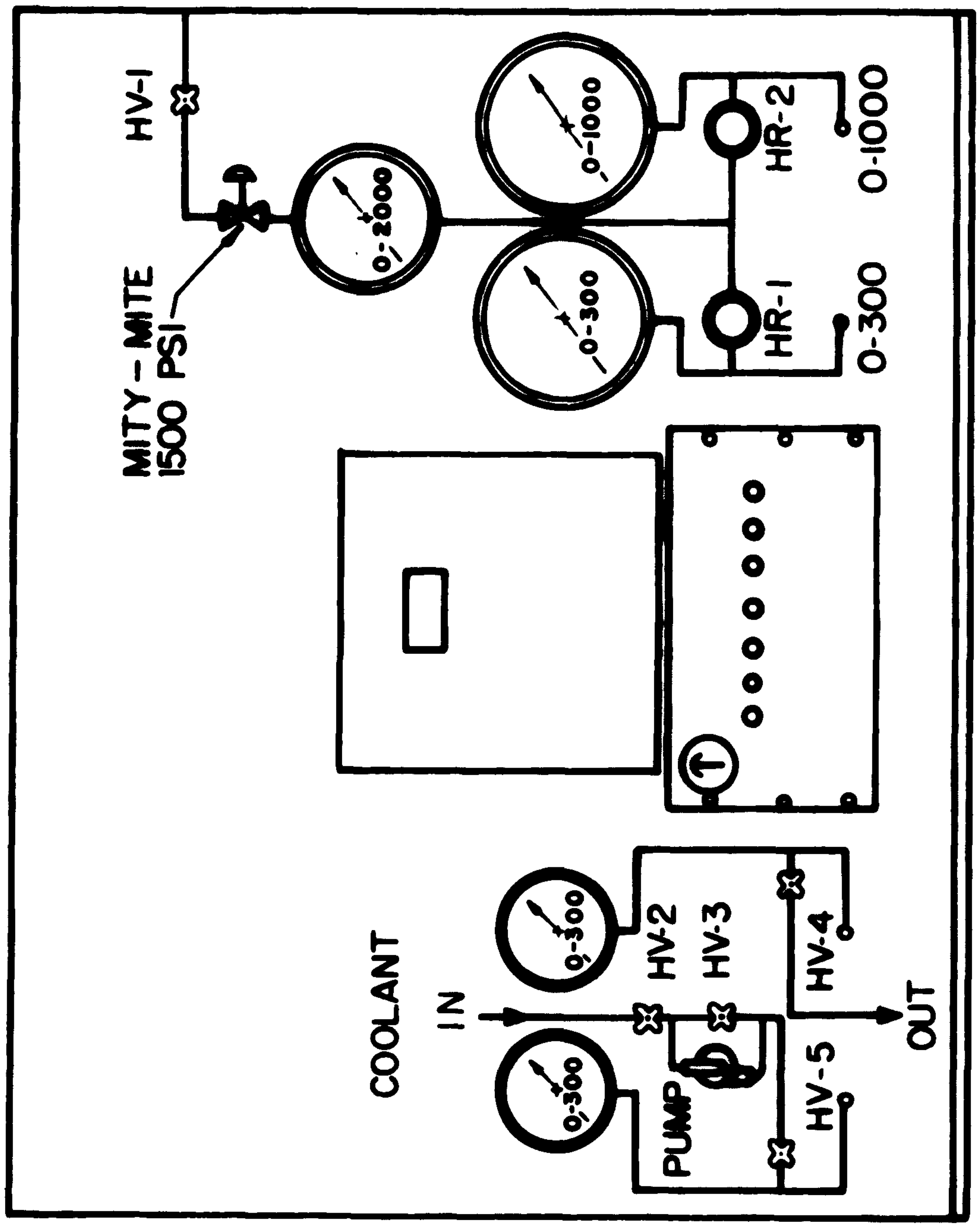


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FIGURE 13

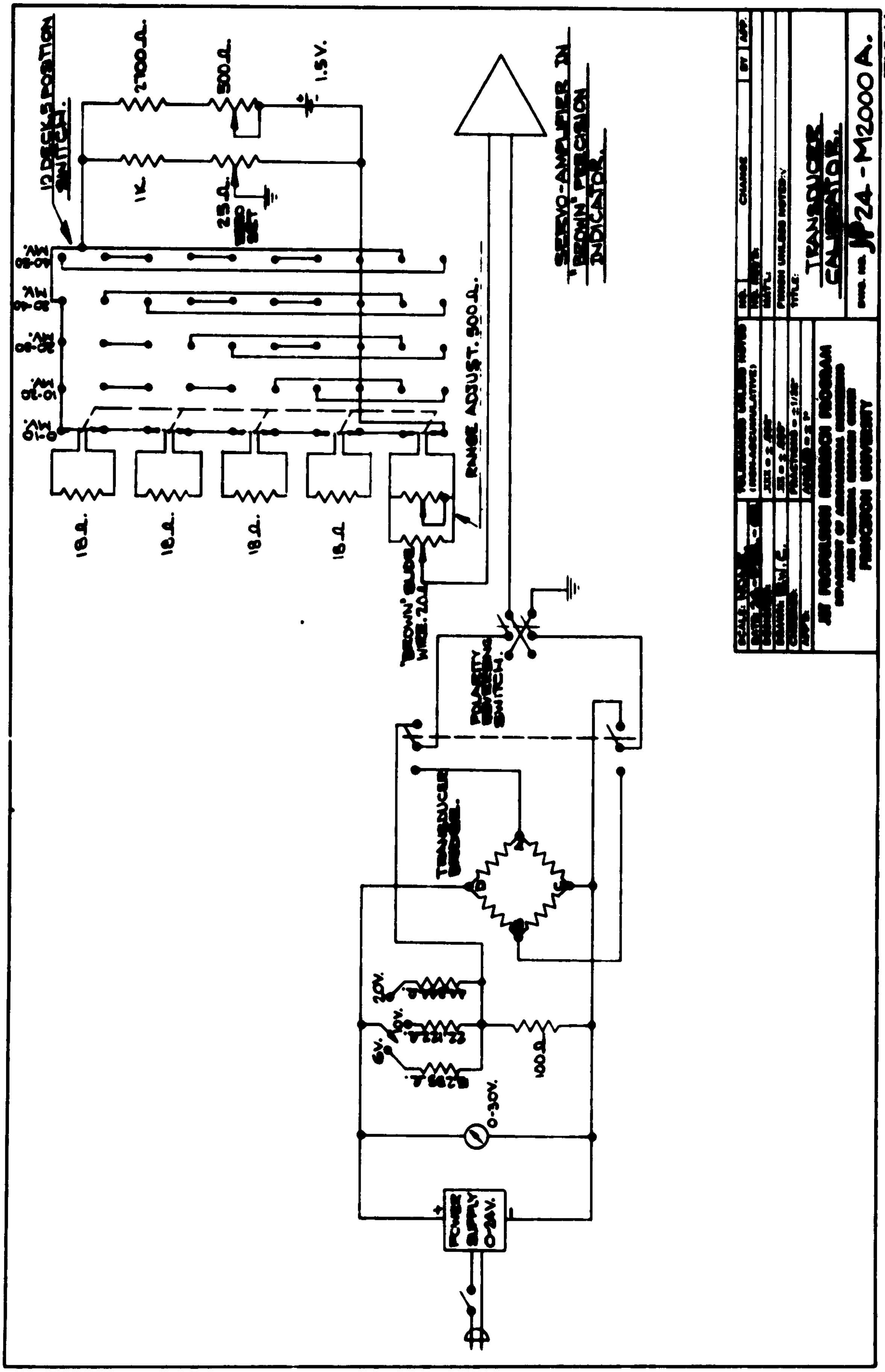
12 DECK 5 POSITION SWITCH.
2700 Ω.
500 Ω.
1.5V.
1K.
25 Ω.
ZERO SET
0-10 MV.
10-20 MV.
20-30 MV.
30-40 MV.
40-50 MV.
18 Ω.
18 Ω.
18 Ω.
18 Ω.
RANGE ADJUST. 500 Ω.
"BROWN" SLIDE WIRE. 20 Ω.
SERVO-AMPLIFIER IN "BROWN" PRECISION INDICATOR.
POLARITY REVERSING SWITCH.
TRANSDUCER BRIDGE.
20V.
10V.
6V.
100 Ω.
0-30V.
POWER SUPPLY 0-24V.
JET PROPULSION RESEARCH PROGRAM
PRINCETON UNIVERSITY
TRANSDUCER CALIBRATOR.
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FIG.14.

of resistance. An exact measurement of bridge excitation voltage is therefore unnecessary.

On the left side of the panel, a coolant supply is provided where water pressure and flow up to 200 psi and 350 lbs/hr is available to measure the interdependence of output and coolant conditions. Accuracy of the calibration bench is limited by the pressure gauges used, but by means of the dead weight tester, accuracy is 0.1% checked and verified against calibration standards elsewhere.

B. Dynamic Calibration

To define the characteristics of a transient pressure transducer, apparatus is required which will permit the evaluation of the analytical expressions defining the transducer transfer function. This is responsible for the inequality existing between the driving function and the analog signal which is the transducer output. If the transfer function can be accurately determined for a transducer, the reliability of the output can be predicted, and the output can sometimes be compensated to render a reasonably exact duplicate of the driving function.

As defined in Section III, the output ratio A and phase ϕ is:

$$A = \frac{1}{\left(1-\left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\zeta\frac{\omega}{\omega_n}\right)^2}$$

$$\phi = \tan^{-1}\frac{\left(2\zeta\frac{\omega}{\omega_n}\right)}{\left(1-\frac{\omega}{\omega_n}\right)^2}$$

These values are plotted in Figures 4a and b. They are valid only in the cases where transducers perform predictably as second order systems with a single degree of freedom. Fortunately most well-designed transducers act in this manner and it is therefore feasible to represent the frequency response with fair accuracy by these curves. The determination of damping and natural frequency now must be accomplished.

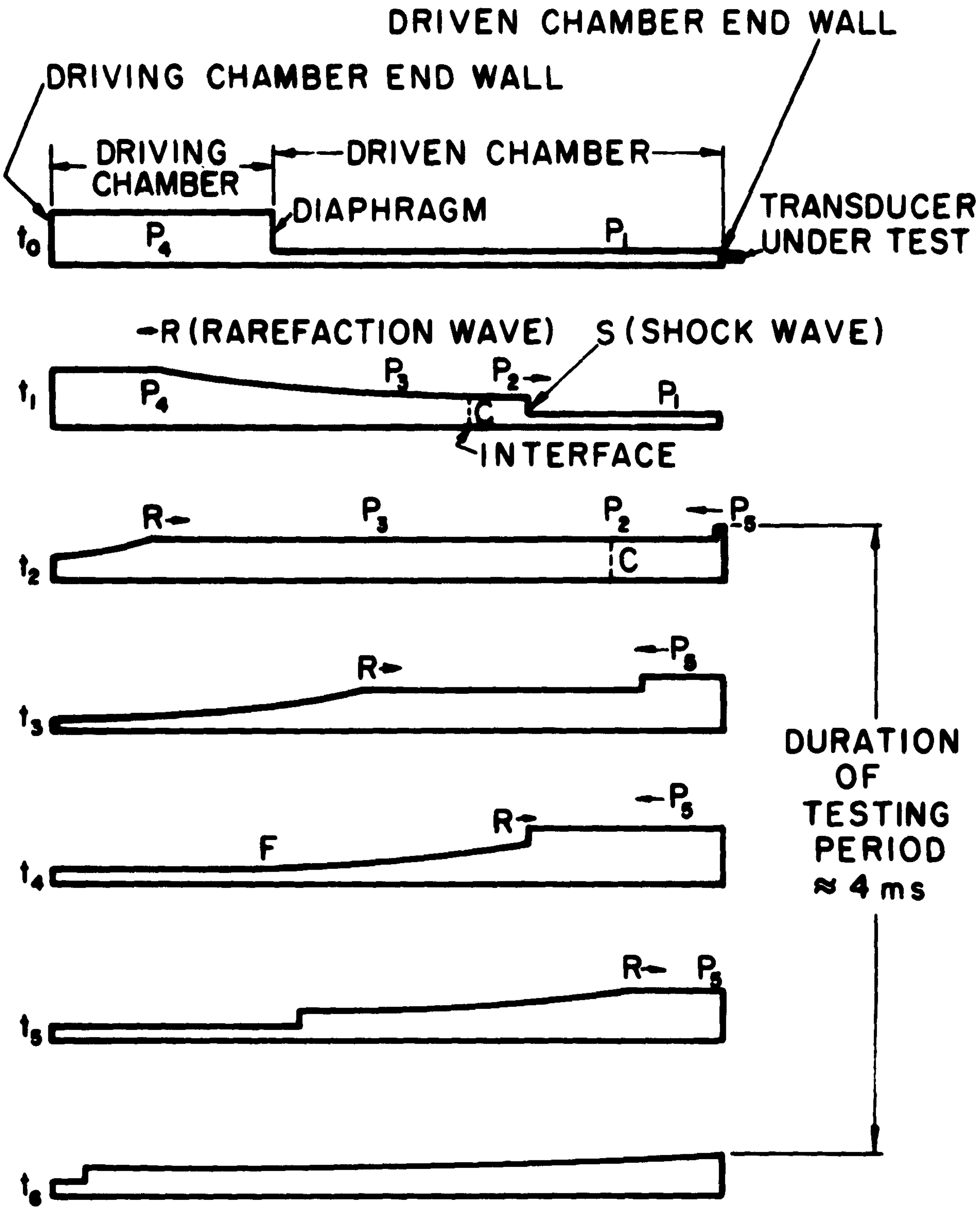
C. The Shock Tube

The requirement for a piece of apparatus which can deliver a step-function of pressure of -edictable shape usually results in a decision to utilize some variation of a shock tube. The shock tube, first demonstrated by Vieille in France in 1899, is an obvious principle, but less obvious is the method by which a controlled pressure step function can be generated and applied to the evaluation of a transducer design.

The shock tube consists of two sections of pipe, separated from one another by a diaphragm which serves to permit a difference of pressure between one section and the other. In Figure 15 a shock tube is schematically represented, and successive pressure histories at times $T = 0$ through $T = T6$ represent the progression of shock waves in the indicated directions. The theoretical and practical definition of these conditions has been extensively covered in the literature, and only those parameters directly related to the evaluation of transducers will be discussed.

The important characteristics of the pressure history for the testing of transducers are the sharpness of the pressure rise, the value of P_5 and the duration of P_5 before either the reflected rarefaction wave or P_3 reaches the end wall. The rise time of the initial normal shock front is extremely small. Irregularities in the transducer diaphragm, however, permit a broadening of the impingement of the shock front. Seldom is there a larger rise time than 1μsec.

The value of P_5 is predicted on the basis of experiments by several researchers to be related to the value of P_4. Although the



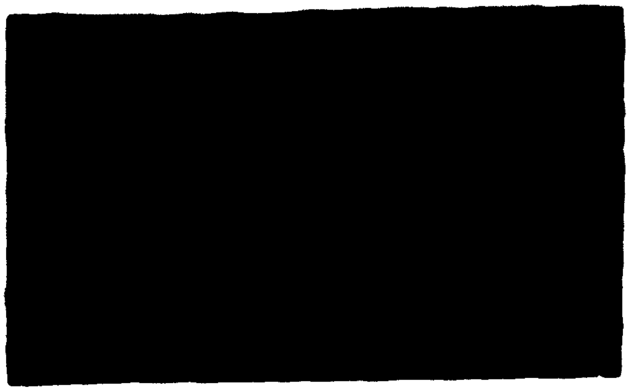
PRESSURE DISTRIBUTION IN A SHOCK TUBE
AT INTERVALS AFTER BURST OF DIAPHRAGM

FIGURE 15

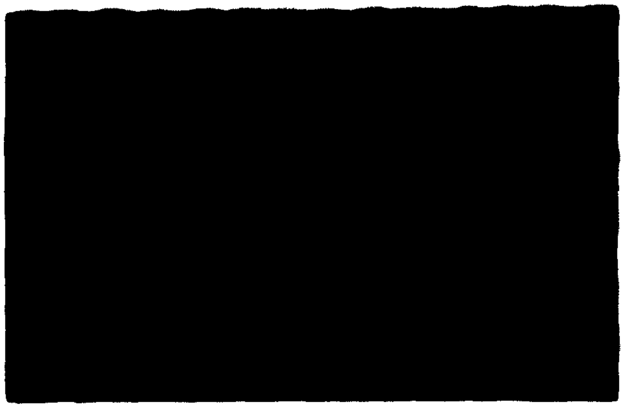
theoretical value varies according to the velocity of the initial shock wave, P_5 seems to be more affected by the tube geometry, wall roughness, etc. and is approximately constant at $^{P_5}/_{P_4} = .75$ in the case at hand.

The duration of steady pressure at the end wall is determined ultimately by the period required for the reflected rarefraction wave R to arrive. The total length L of the tube determines this time. It is necessary in the original design to permit sufficient testing time, depending on the characteristics of the transducers to be tested, so that the damped oscillations resulting from the pressure step can die down appreciably. To attain this limiting time, it is important to choose values of P_4 and P_1 which will permit passage of the reflected shock P_5 through the interface, labelled C , which is the boundary between the driving gas and the driven gas. Since the shock Mach number, Ms, is determined by the ratio of $^{P_4}/_{P_1}$ and the temperature, pressure, and composition of the driving gas may be predicted easily, it is possible to set up driving gas conditions, so that there is a non-reflecting condition at the interface, and verify this operation by measurement of Ms. For values of Ms between about 3.5 and 4.0, there is little or no limitation of testing time due to reflection from the interface. A curve of $^{P_4}/_{P_1}$ vs Ms is shown in Figure 17. Figure 16a, b, and c show pressures at the end wall of the tube for conditions of Ms high, correct, and low.

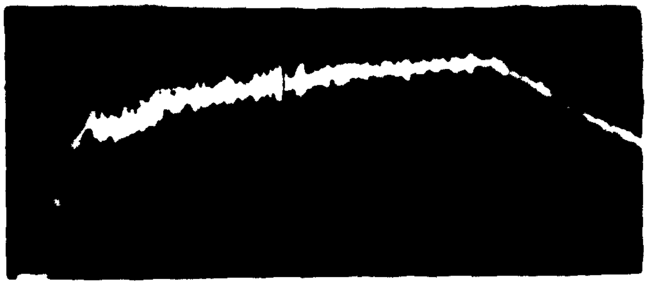
In the shock tube employed for the work here reported, no diaphragm bursting mechanism was employed. A series of diaphragms of varying thickness made of aluminum was used satisfactorily. Figure 18 shows the burst pressure using a rounded lip clamping plate, as shown.



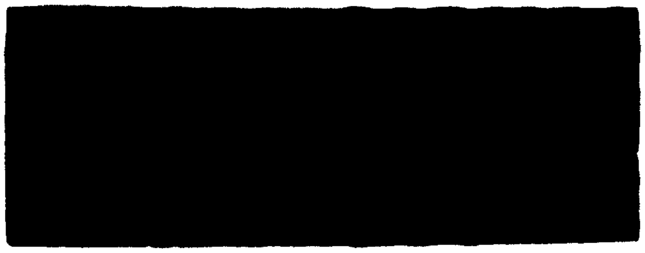
a. Output of thin film gauges upper trace - upstream gauge lower trace - downstream gauge.



b. Thin film gauges of Fig. 16a after amplification and rejection of low frequency components by amplifiers shown in Fig. 21.

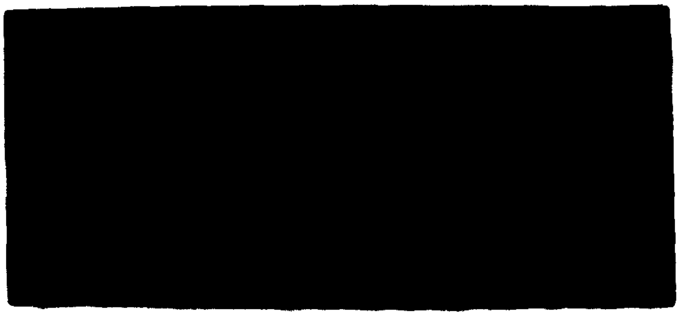


c. Reflected shock pressure, M_s = 5.2

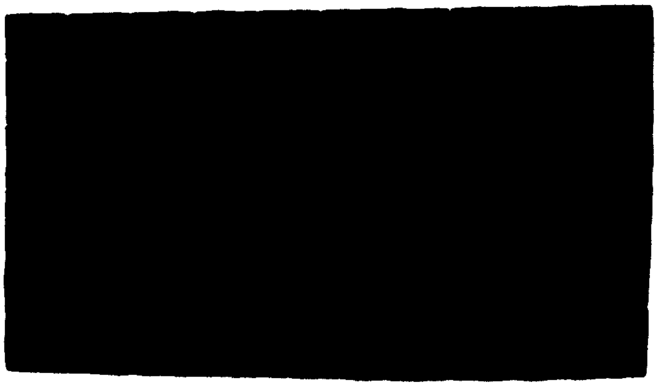


d. Reflected shock pressure, M_s = 3.5

e. Reflected shock pressure, M_s = 2.5



f. Typical transducer pressure trace
(Dynisco PT49A - 15 psi shock)



g. Upper Trace. Kistler 601 Transducer
Lower Trace. Axial vibration-200g/div.
(300 psi shock)

Sweep Speeds for Fig. 16a-g

a,b	-	50 μs/cm
c,d,e,f	-	200 μs/cm
g	-	20 μs/cm

FIGURE 16. Waveforms Observed in Shock Tube Testing

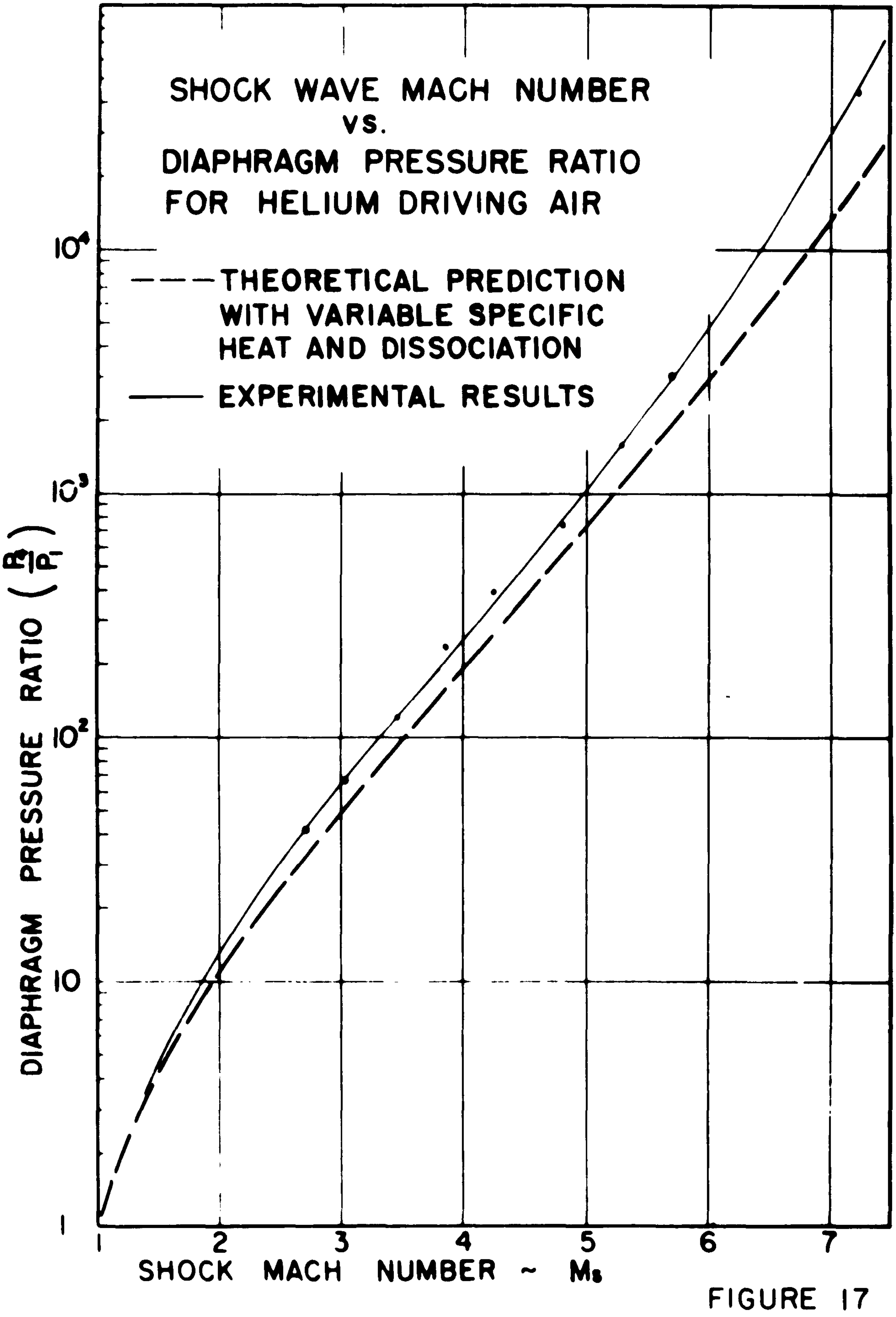


FIGURE 17

JPR 227

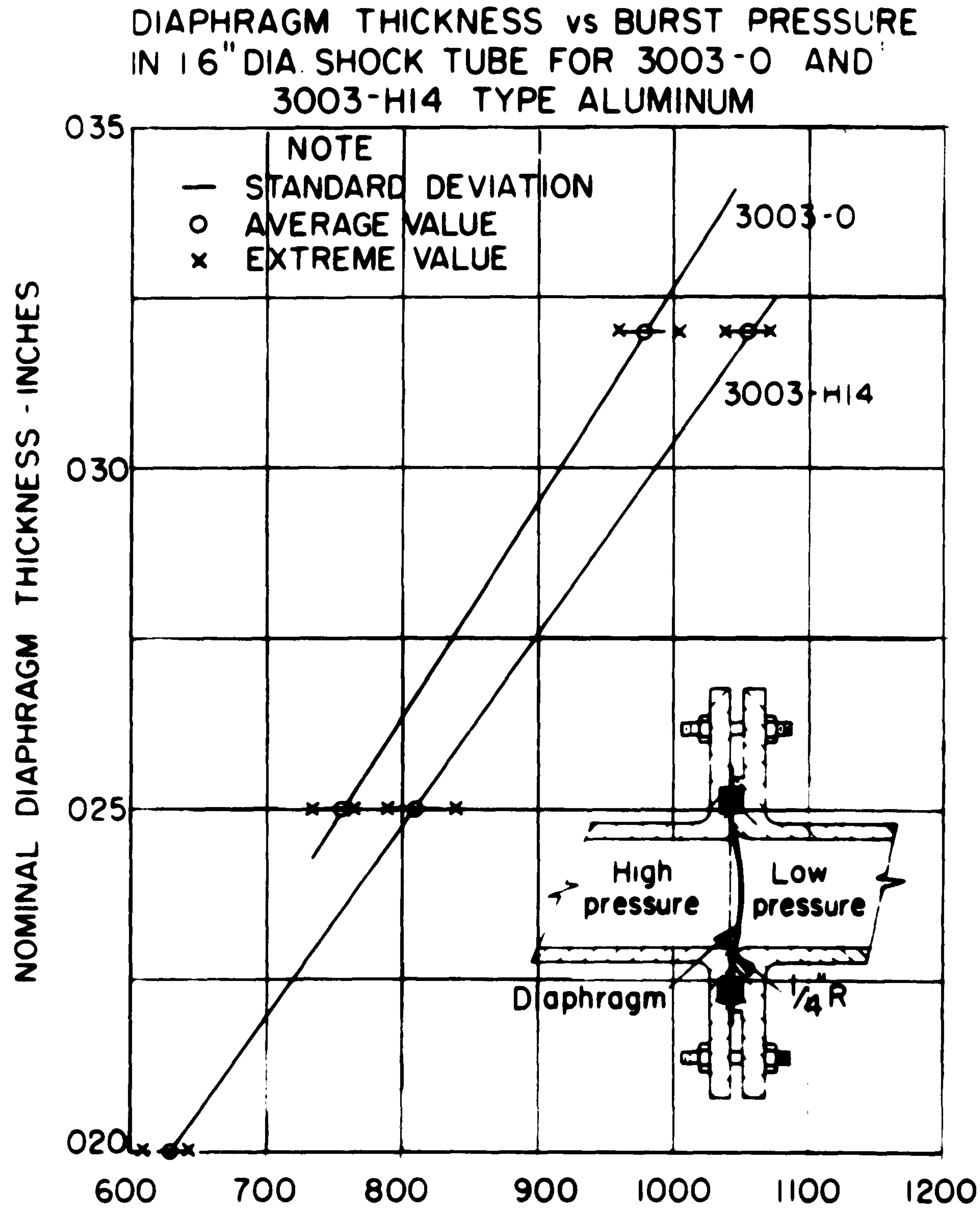


FIGURE 18

JPR 224

In addition to the aluminum diaphragms, considerable use was made of Mylar sheet, .002" thick, which burst at 65 psia.

Normally, helium driving air was used to operate the shock tube. It was found particularly important to purge each section between tests with the appropriate gas in the interest of attaining consistent results.

In the block diagram, Figure 19, the operational details will be noted. A pressure gauge and appropriate valving for the gas supply are connected to the pressure tap on the driving end, while a vacuum pump, manometer, and valving to gas supply complete the driven section pneumatic connections. Adapters for the installation of a number of different transducers, and a double flange diaphragm holder complete the mechanical details others of which may be seen in either Figure 19 or Figure 20 a photograph of the tube. It will be noticed that the entire tube is free to move axially except where restrained near the testing section. The vibration caused in the test section by the diaphragm burst and subsequent shock wave results in the generation of an electrical signal by certain transducers sensitive to vibration. It was found that rigid mounting of the test section resulted in smaller vibration magnitude but considerably higher frequency than when suspended by springs. As the springs were weakened, it was possible to lower the frequency sufficiently to essentially eliminate interference from this source, normally referred to as "ground shock".

To sense the arrival of the shock wave early enough to permit initiation of the oscilloscope sweep several methods were tried, including two commerical "high response" thermocouples neither of which were adequate. The final solution was found to be platinum thin film

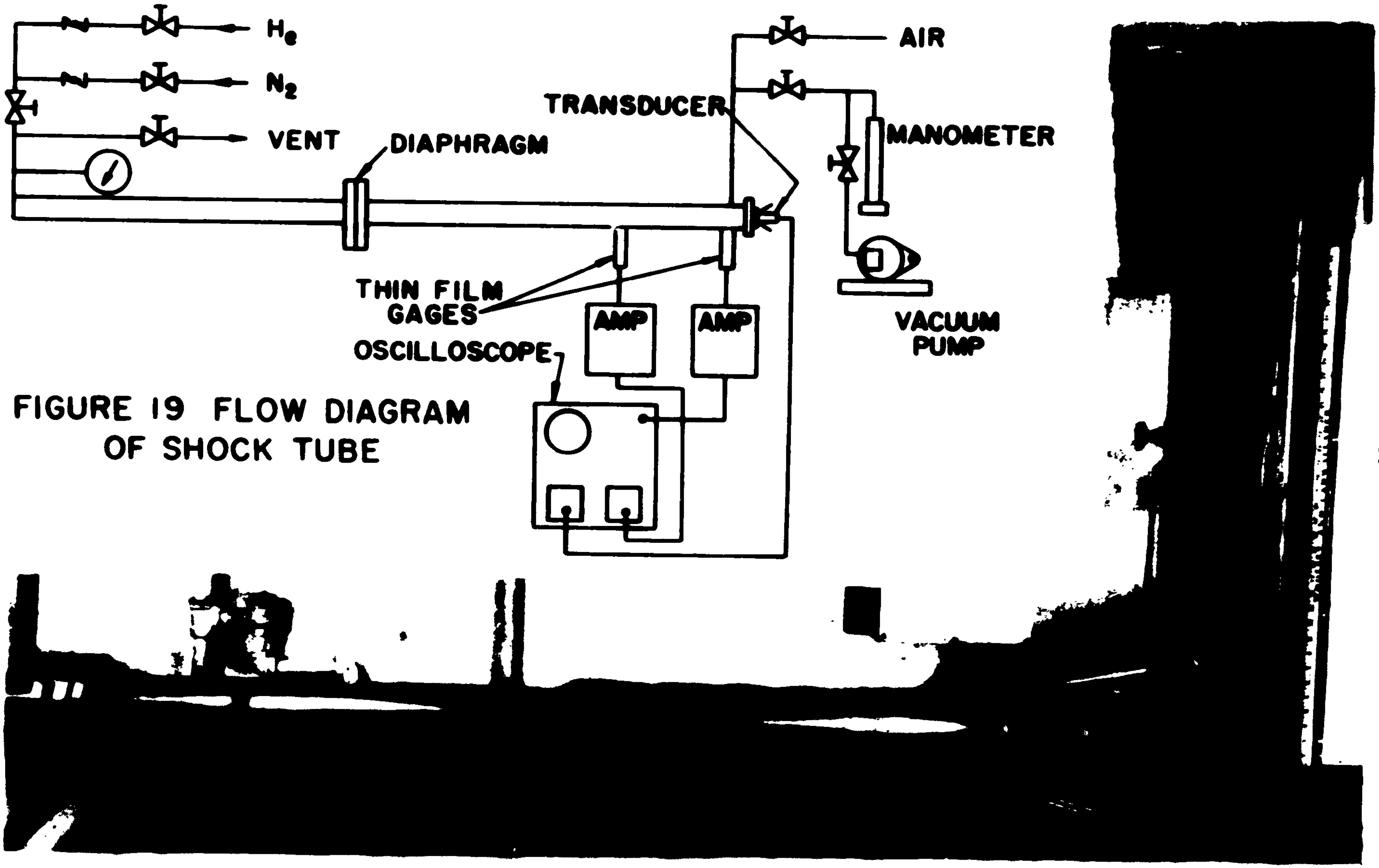


FIGURE 19 FLOW DIAGRAM OF SHOCK TUBE

FIGURE 20 INSTRUMENTATION SHOCK TUBE

gauges, used by McAlevy, Ref.13. These are deposited platinum, of molecular thickness on glass, used as a resistance thermometer to sense the temperature rise at the passage of the shock. It will be noted from Figure 19 that two such gauges are installed, the first quite close to the test section, the second one foot upstream. The velocity of the shock wave may be measured directly by measuring the time required to travel the one foot separating the two gauges.

The rise time of these gages is under 1 μsec, and the output about 3 millivolts into the amplifiers when used with the circuit shown in Figure 21. Figure 16d shows a typical output waveform. The amplifier permits a large signal, on the order of 9 volts, to be applied to the external trigger terminal of the oscilloscope. It was designed to permit good rise time coupled with low frequency AC rejection. It is equipped with a gain control so that its gain can be reduced when required to reduce its sensitivity to stray electrical transients, rather a severe problem prior to addition of the control.

To determine the response of a transducer to driving functions approaching the transducer natural frequency, the determination of damping and natural frequency is done from the photograph of the pressure step signal. Normally damping is sufficiently low to neglect its effect on natural frequency. It is, therefore, a relatively simple matter to count the cycles on a known time base and thereby determine the natural frequency. Damping is also determined from the shock tube photograph by means of the hyperbolic curve plotted in Figure 3, the damping factor vs number of cycles to decay to 50% amplitude.

D. Sine Wave Modulator

In Tallman, Ref.4, a more complicated method of deriving

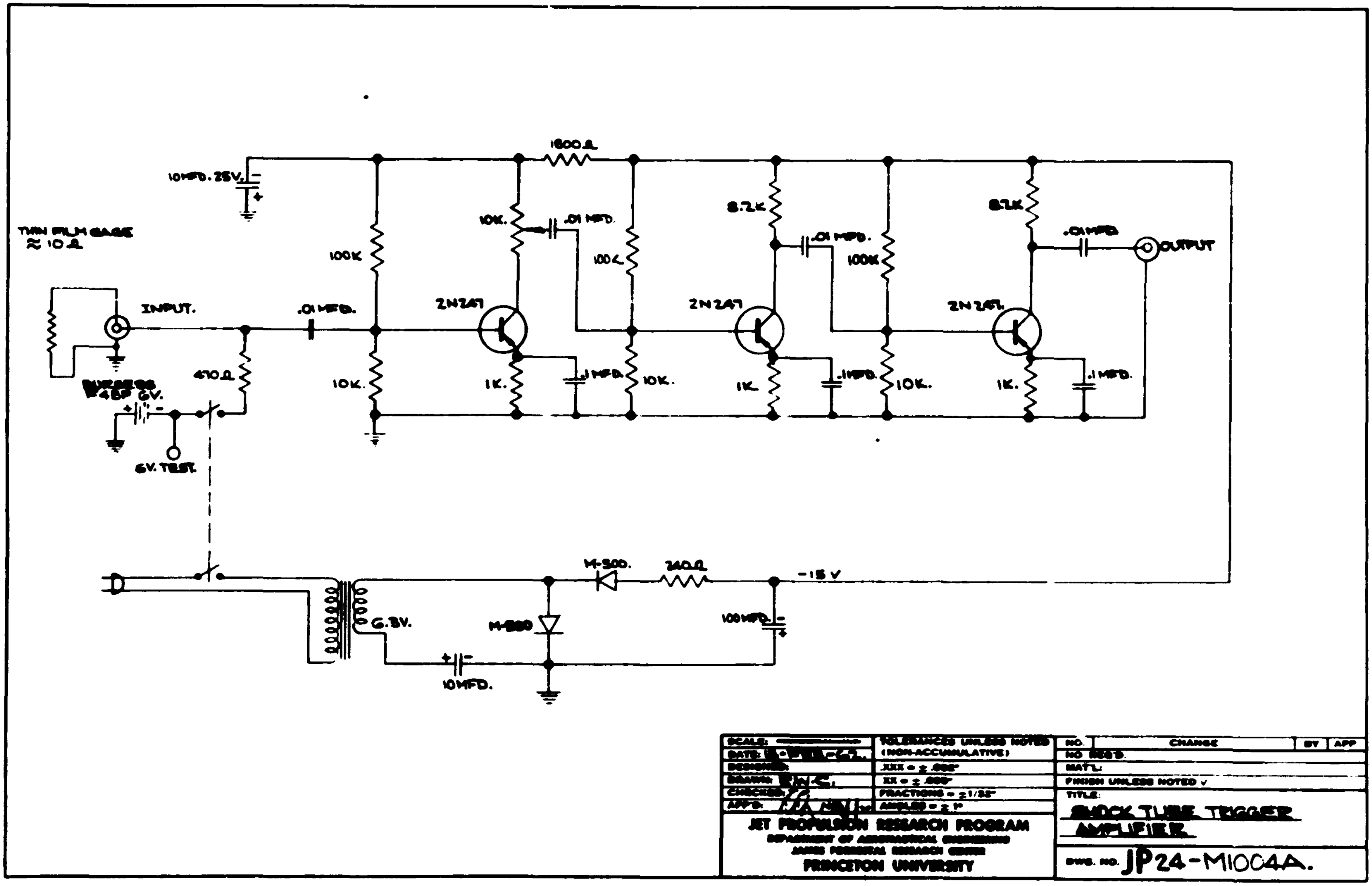
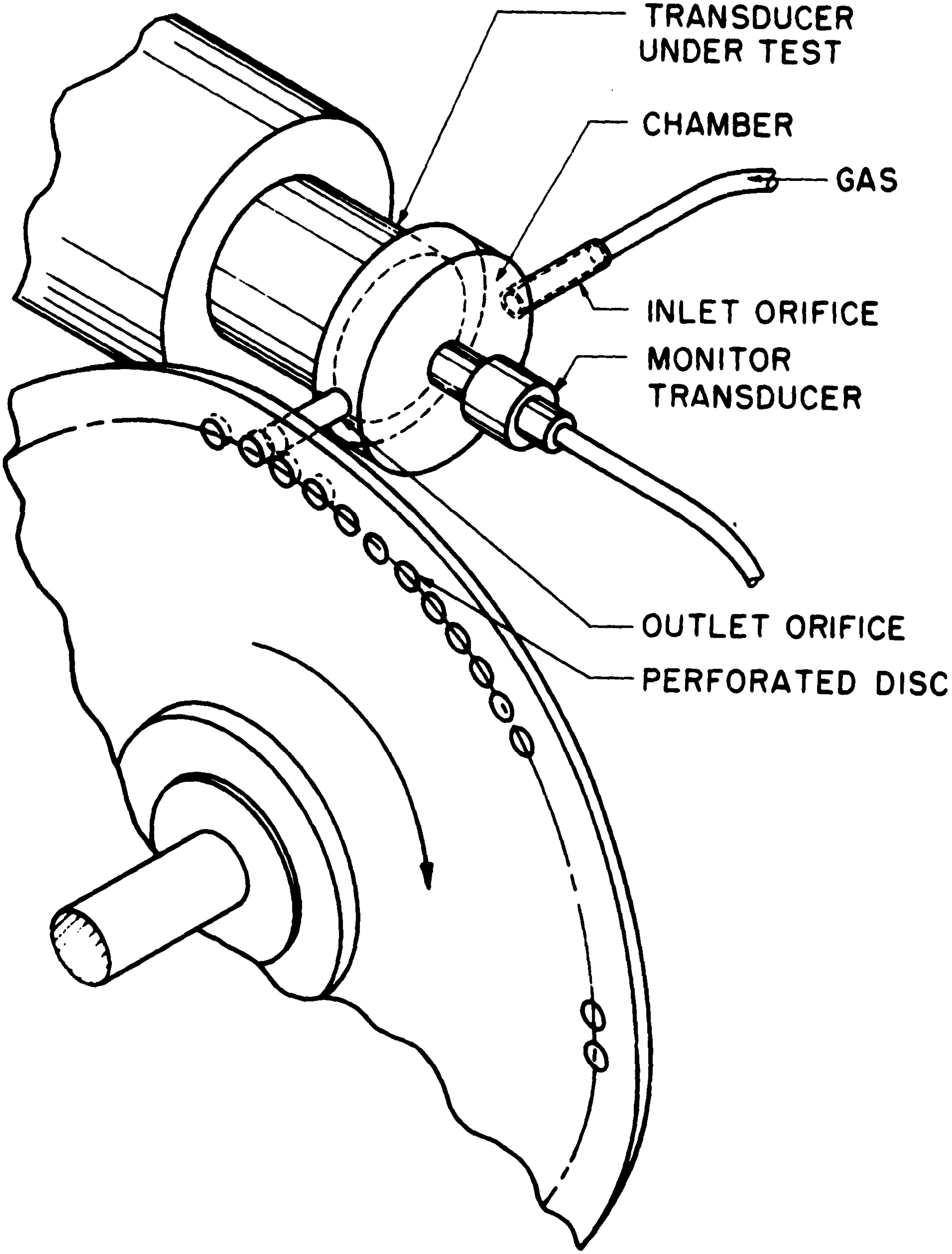

THIN FILM GAGE
≈ 10 Ω
INPUT.
470 Ω
4.5V 6V.
6V. TEST.
10 MFD. 25V.
.01 MFD.
100K
10K.
2N247
10K.
.01 MFD.
1600 Ω
1K.
.1 MFD.
100K
10K.
2N247
8.2K
.01 MFD.
1K.
.1 MFD.
100K
10K.
2N247.
8.2K
.01 MFD.
1K.
.1 MFD.
OUTPUT
6.3V.
10 MFD.
M-500
M-500.
240 Ω
-15 V
100 MFD.
SCALE:
DATE:
DESIGNED:
DRAWN:
CHECKED:
APPD:
TOLERANCES UNLESS NOTED (NON-ACCUMULATIVE)
XXX = ± .005"
XX = ± .005"
FRACTIONS = ± 1/32"
ANGLES = ± 1°
JET PROPULSION RESEARCH PROGRAM
DEPARTMENT OF AERONAUTICAL ENGINEERING
JAMES FORRESTAL RESEARCH CENTER
PRINCETON UNIVERSITY
NO.
CHANGE
BY
APP
NO REQ'D.
MAT'L:
FINISH UNLESS NOTED √
TITLE:
SHOCK TUBE TRIGGER AMPLIFIER
DWG. NO. JP24-M1004A.

FIG. 21.



JPR 1207

PRINCIPLE USED IN SINE WAVE MODULATOR

FIGURE 22

Reference 14 describes in detail the various modes of oscillation encountered in cylindrical chambers. When $L/D > 1$, as in the sine wave modulator, it would be expected to observe purely transverse modes of oscillation if frequency is sufficiently high. It is essential for predictable operation to limit all testing to frequencies well below the lowest to be encountered.

The first tangential mode is the lowest frequency predicted by Reardon in Ref. 13:

$$f = \frac{S_1 (C)}{d} = \frac{1.84\ (37900)}{.75} = 29,600 \text{ cps}$$

where

S_1 = Bessel function number = 1.84 for 1st tangential mode

C = 3160/ft/sec = 37900

d = chamber diameter

A configuration was designed where mechanical rigidity and convenience of operation were considered. Figure 23 shows an assembly drawing of the sine wave modulator. A micrometer thread to adjust clearance between wheel and outlet orifice was provided. The transducer under test compresses one side of the chamber. A monitor transducer, an SLM quartz gauge having a resonant frequency of 140KC is mounted flush in the opposite wall. The inlet orifice is choked at times when gas is flowing and is 1/10 in. diameter. The outlet orifice is 1/8" as are the 72 holes in the wheel, spacing being 1/8" also. When smaller transducers are tested, a cylindrical shim is used to decrease chamber diameter, thus permitting a higher frequency to be used before encountering the lowest resonant conditions.

The device was constructed and was found to give excellent sine wave response at the lower frequencies, up to about 12KC. A photograph

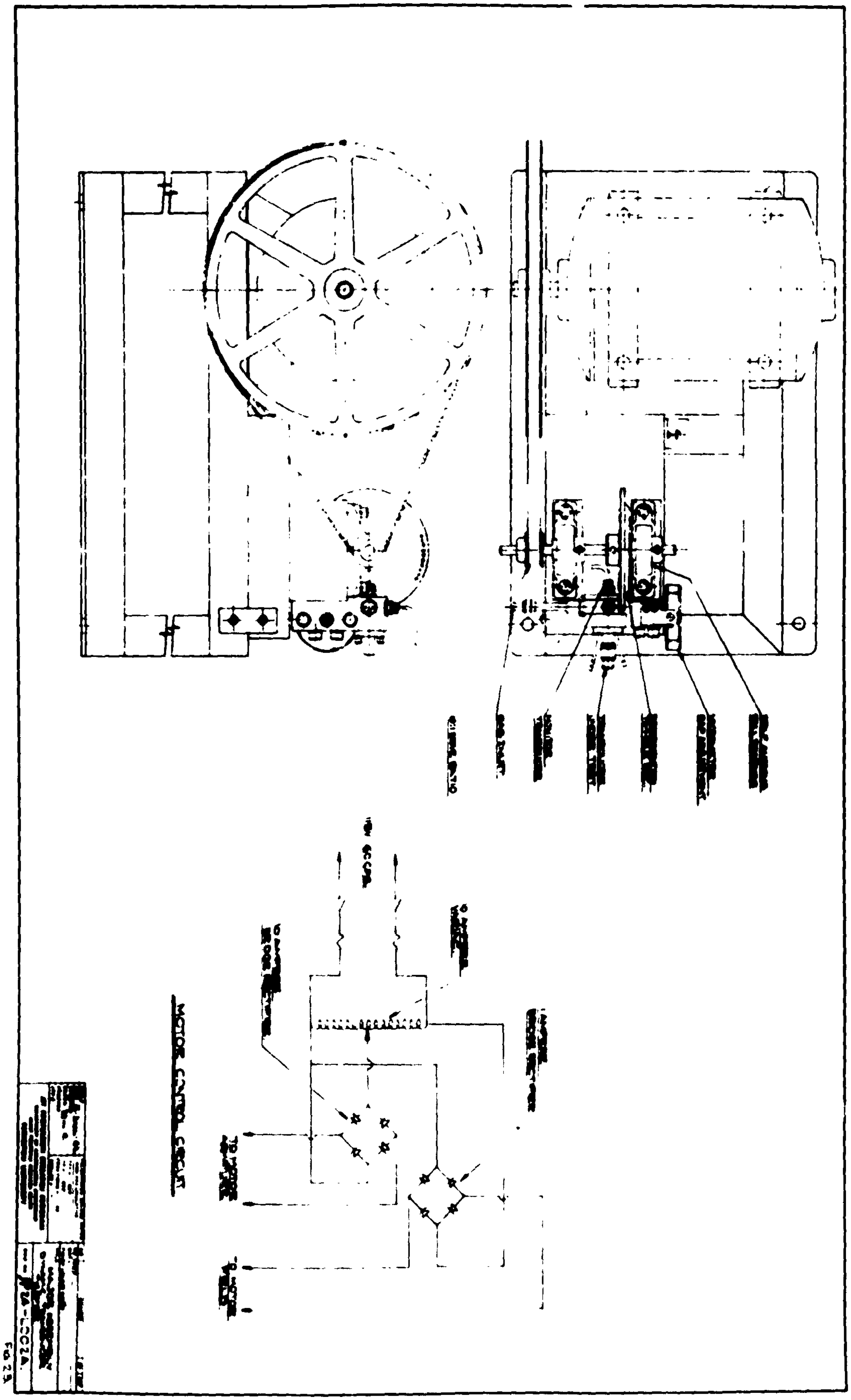


Fig 23

of the recording installation is shown in Figure 24, and a photograph of the modulator is shown in Figure 25.

The customary method of use requires the unknown transducer and the monitor transducer to be recorded on a two beam oscilloscope simultaneously. At various frequencies RMS outputs from each are then recorded. After some considerable use it was noted that the single traces used in obtaining the photographs were not necessarily repetitive over a number of cycles and the cause was found to be a speed irregularity caused by the variable speed drive used to rotate the wheel. In addition, amplitude variation was noted and the wheel, being measured, was found to be untrue in an axial direction, which resulted in spurious modulation at each revolution of the wheel. A new wheel, made to more accurate specifications and a shunt wound D.C. variable speed motor were incorporated. These measures essentially eliminated the difficulties.

The effect of turbulence noise generated in the orifices is present at higher gas flows;i.e.,when the outlet orifice is uncovered. This is minimized by operating the wheel as close as possible to the orifice for minimum leakage and by using the lowest pressure applied which will keep the inlet orifice choked.

The experimentally determined response curve is shown in Figure 26. It will be noted that as the frequency increases amplitude decreases considerably. It is possible to improve the apparent waveform by filtering below the resonant frequency of the monitor transducer. Figure 26 shows a series of these photographs.

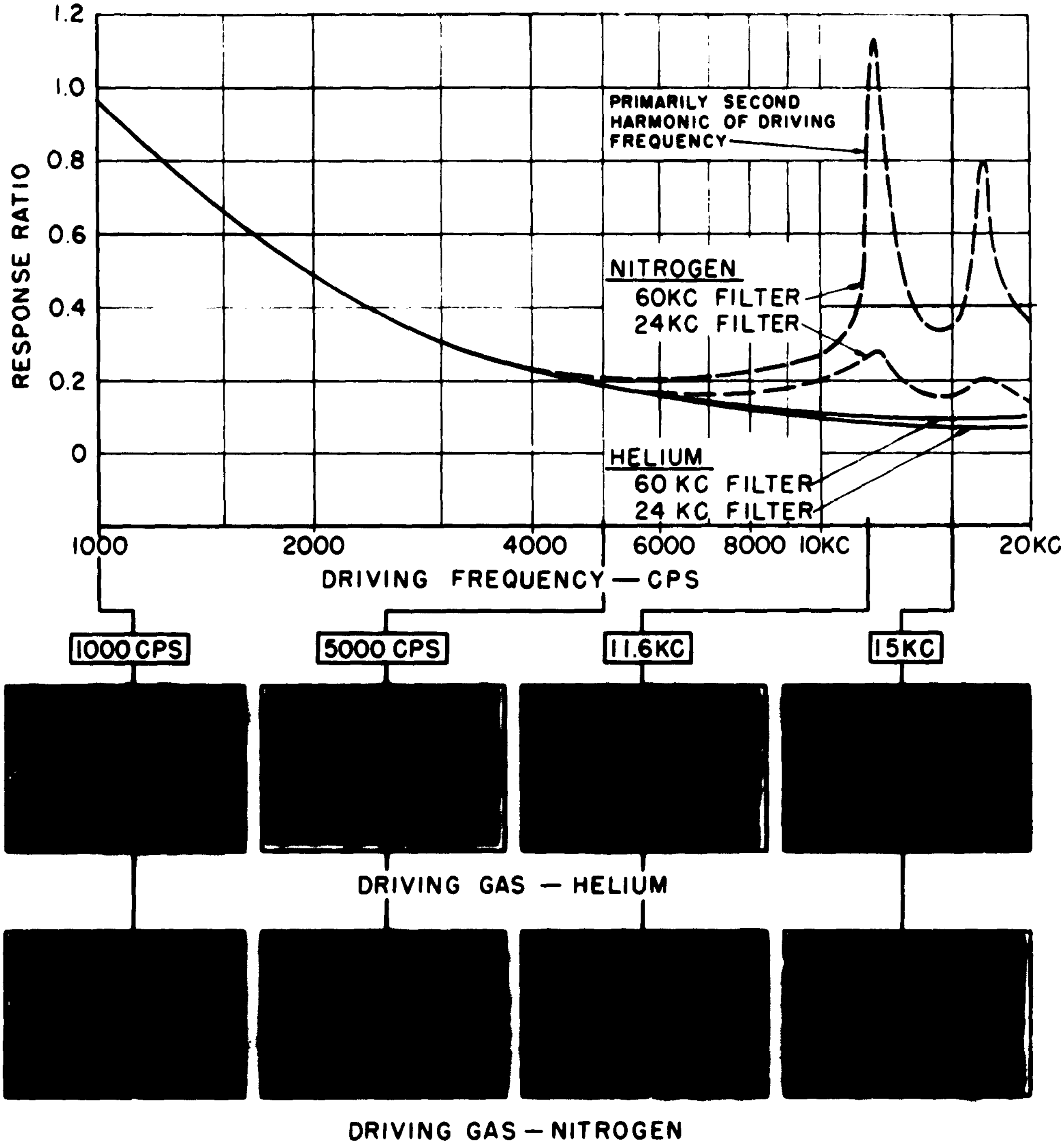
A much more extensive treatment of the chamber waveforms is included in an MSE thesis on the analysis to be published shortly. In this experimental investigation, a set of three Kistler transducers were

SINE WAVE MODULATOR
RECORDING APPARATUS

FIGURE 24

SINE WAVE MODULATOR

FIGURE 25



(UPPER TRACES ON EACH PHOTOGRAPH — FILTERED AT 24 KC
LOWER TRACES ON EACH PHOTOGRAPH — FILTERED AT 60 KC)

PERFORMANCE OF SINE WAVE MODULATOR

FIGURE 26

installed in the wall usually taken by the transducer under test. A wide range of waveforms are displayed and analyzed using gases with lower values of C (Freon 12 and nitrogen) as well as helium, so that the chamber could be examined in several modes of oscillation.

E. Heat Transfer Determination by Nondestructive Testing

To determine ability of a transducer to withstand large values of heat transfer rates into the diaphragm, it is necessary to perform tests at a much lower and controllable value to avoid burnout and ultimate complete destruction of the transducer without rendering any usable data. Section IV of this report covers the theory of heat transfer in transducers. If it is possible to locate a point in Figure 6 by a reasonable heat transfer rate and diaphragm temperature with a given cooling pressure and flow, it can be seen that for the given pressure and flow, it is possible to extrapolate through the nucleate boiling range to the point of burnout, thereby predicting at what value of q burnout will occur.

The use of temperature sensitive paint, applied to the transducer diaphragm which is then heated to a fairly low value until temperature indication by the paint is reached can establish diaphragm surface temperature. At the same diaphragm temperature, a measurement of q into the diaphragm may be made (Ref.8).

$$q = \frac{Q\Delta T}{A}$$

where

Q = flow rate lb/sec

ΔT = temperature rise of coolant ^{o}F

A = active area of diaphragm - m^2

q = heat transfer rate BTU/sq.in/sec

This test is presently being conducted and although results appear encouraging, little data is ready for use as of this date. A photograph, Figure 27 shows the transducer undergoing this test.

Because of the relatively low q , coolant temperature rise is small and a sensitive yet fast response ΔT measurement is necessary. It was decided to use thermisters and an extremely small type, Veco 31A3 , was obtained and connected into a bridge circuit to measure ΔT across the transducer. The instrument used to read out the data was a Brown Electronik strip chart recorder having a full scale span of 8mv, equivalent to about 12°F full scale. The copper mounting is water-cooled and protects the body of the transducer from the heat source. Heat transfer into the body of the transducer from the copper mounting accounts for less than 10% of the total observed temperature rise across the transducer, as demonstrated by the fact that ΔT returns immediately to a value under 10% of the former reading upon removal of the heat source in spite of the fact that the heavy copper mounting has much slower temperature response. This is in agreement with Condomines' finding in Ref.8.

In the liquid propellant combustion instability program, conducted concurrently, a special unstable rocket motor is undergoing test. Figure 28 shows a specially prepared chamber section for transducer evaluation up to 12 BTU/sq.in/sec. The method of testing for ultimate transducer heat transfer ability is to operate the rocket chamber at an unstable condition and lower coolant flow rate in steps to the point of damage, thereby establishing the ability of the transducer to withstand heat fluxes at a given coolant flow rate.

NON DESTRUCTIVE HEAT TRANSFER TEST

FIGURE 27

HEAT TRANSFER TEST SECTION

FIGURE 28

VIII. CONCLUSION

In a later technical report, full details will be reported on several available transient pressure transducers, these details to be derived from the methods, procedures and apparatus described herein.

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Department of Aeronautical Engineering
Guggenheim Laboratories for the Aerospace Propulsion Sciences

1 September 1961

Specifications For Transient Pressure Transducer

1. Application

The subject transducer is to be primarily designed for applications in the transient pressure measurement of rocket combustion chambers, in research and development testing, where the most extreme environmental conditions are to be expected, including extreme heat flux, radiation (in certain cases), and vibration.

2. Mechanical Design

a.) General

The transducer will be used under conditions of extreme heat, moisture, vibration, etc. Its active area, to assure its ability to read fast pressure perturbations must be mounted flush in the chamber wall where the measurement is to be made. Because of space limitations between cooling passages, as in a cooled rocket chamber, the diameter of the transducer shank leading to the active area must be limited to .250 inches or slightly more. Also, the transducer body should be small because of the space limitations placed by other apparatus nearby. The mounting method should be that which would require the smallest volume, i.e., either flange or thread mounting, possibly a universal mounting with a removable flange and also a machine thread which would permit either type of mounting.

b.) Material

All metal portions to be fabricated from stainless steel or other corrosion resistant material. The diaphragm, being thin and subject to heat, should be particularly corrosion proof, as any appreciable amount of oxidation would affect the transducer markedly.

Active Area: #304 or #316 stainless steel or equivalent.

Body & Shank--Stainless steel or other corrosion resistant material.

c.) Dimensions

See Drawing

d.) Mounting

Removable flange, see drawing.
Sufficient force to attain pressure seal and/or thermal expansion of mounting shall not affect zero or sensitivity of transducer.

e.) Coolant Passages

Pressure rating of cooling passages to be at least as high as operating pressure of transducer. Coolant lines to have flare or equivalent connections.

f.) Electrical Connections

Electrical fittings to be completely water and moisture proof. Electrical connectors to be highest quality, hermetic, MS type, preferably at the end of a flexible conduit.

3. Thermal Design

a.) General

The transducer designed in accordance with the specifications will be subjected to temperatures of 5000°F and heat fluxes of up to 25BTU/sq in/sec. In order to achieve the necessary cooling ability, the region of nucleate boiling will be employed. Control of the coolant velocity and pressure drop will be extremely critical and will represent a considerable and very important contribution to the total design problem.

b.) Heat Transfer Capability.*

1.) Normal operation: 5000°F, 15BTU/sq in/sec
2.) Extreme conditions: 7500°F, 25BTU/sq in/sec

c.) Coolant Pressure Rating

Maximum pressure rating equal to the transducer operating pressure plus the coolant pressure drop. Normal operation at 100 psi inlet pressure.

*Several sections of the specifications are divided into two sets of conditions, as follows:
Normal operation: Less than 15BTU/sq in/sec--normal coolant pressures--compliance with all specifications expected.
Extreme Conditions: 15-25BTU/sq in/sec--coolant pressures up to maximum--some relaxation of specifications tolerated.

d.) Effect of Coolant Pressure on Zero and Sensitivity

	Zero	Sensitivity
1.) Normal operation (normal coolant pressures)	0.25%	0.1%
2.) Extreme conditions (increased coolant pressures)	2%	1%

4. Transmission Properties

a.) General

Several methods of generating the electrical output from the transducer have been investigated. Because of the small diaphragm area exposed to the pressure, it is difficult using conventional strain gaging techniques to derive a suitable output voltage while keeping the compliance low enough. Such methods as variable capacitance, variable reluctance, semi-conductor strain gages, piezo electric, etc., which will provide the necessary output either separately or when combined with another circuit element will not be rejected, providing that compliance with specifications is essentially complete.

An output from such a system of a magnitude at least equivalent to a standard strain gage device, namely 3 mv/volt F.S., with appropriate compensation for zero and sensitivity change with temperature would appear essential.

b.) Electrical Output

3 mv/volt excitation for full scale pressure.

c.) Accuracy

Total error, all causes, ±1% F.S. static calibration.
Note: (Following tolerances are percent of full scale)

1.) Non-linearity - departure from straight line through zero, 0.25% or less.
2.) Hysterisis ±.255
3.) Zero drift ±1%, normal operation
±2%, extreme conditions
±1%, 100°F case temperature rise
4.) Sensitivity ±1%, normal operation
±2%, extreme conditions
±.5%, 1000°F case temperature rise

d.) Damping

20% of critical, minimum

(It would be most desirable to have the damping variable from 0 to 64% of critical by insertion of the correct damping medium during fabrication.)

e.) Sensitivity to Vibration

±1% F.S. for 5000g below 10KC on all three axes.

f.) Pressure Rating

0-1000 psi, with overload capability as follows:

200% F.S. without permanent deformation
400% F.S. without damage, other than zero shift

g.) Frequency Response

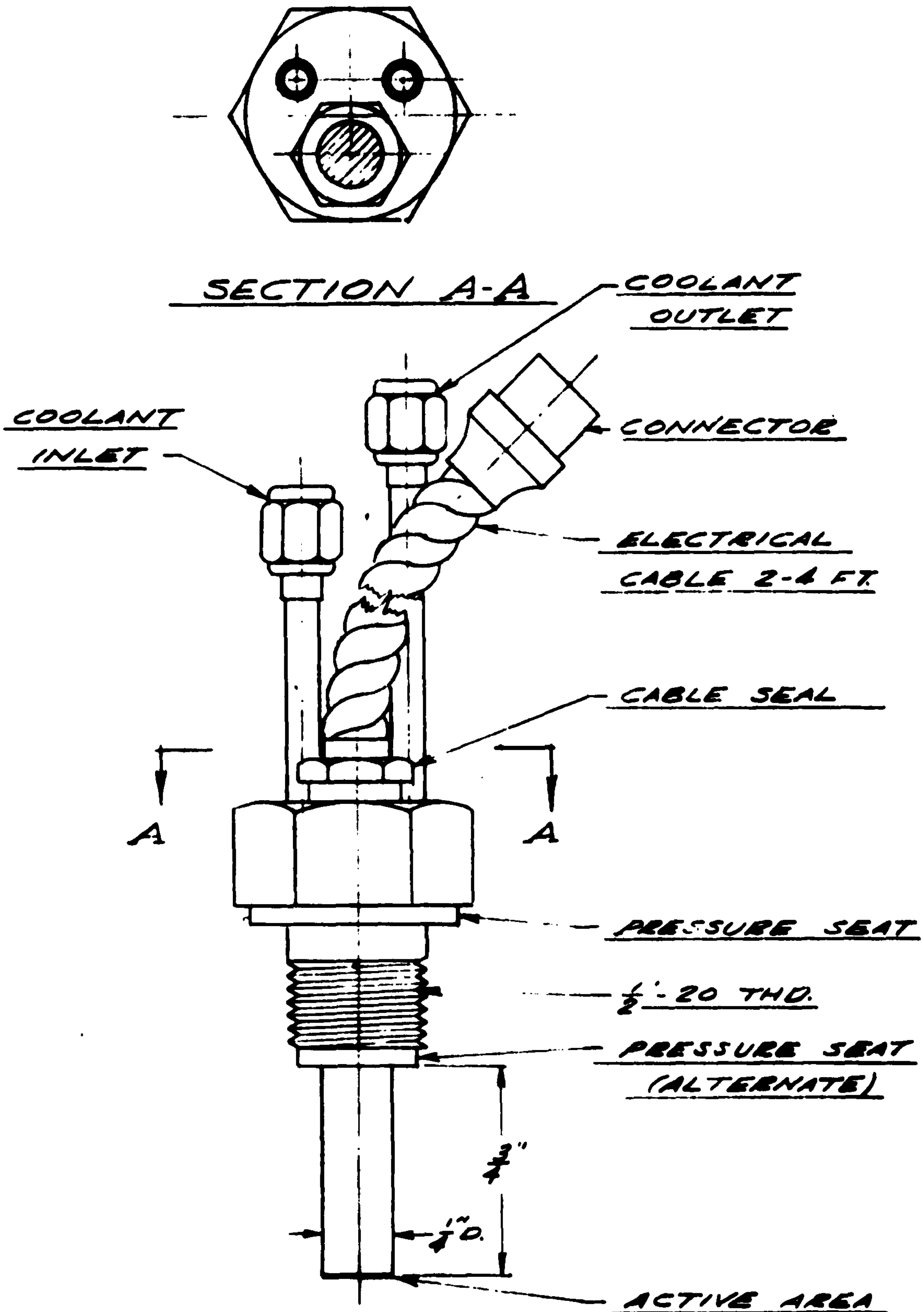
0-15KC (±10%) to sine wave.
Rise time 20 microseconds ro step function.
Resonant frequency sufficiently high so that when combined with damping specification, 4d, response will be essentially as noted.

5. Miscellaneous

a.) Transducer should be so constructed that burnout of diaphragm will not permit pressure to escape through coolant tubes or body.

b.) Connector should have sufficient spare pins to permit connection for shutdown fuse in case of diaphragm burn-out, temperature or end-of-line compensation for voltage, etc.

c.) These specifications are a guide to the design of a practical transducer of the type necessary to determination of transient pressures present in rocket propulsion research, particularly under conditions of unstable burning. Departure from certain of the specifications will be negotiated, but the general requirements are the consensus of many researchers who were contacted on this subject.



DOUBLE SIZE

MINIATURE PRESSURE TRANSDUCER