AN INVESTIGATION OF THE PRACTICABILITY OF TUNED PENDULUM TYPE VIBRATION ABSORBERS FOR USE ON ROTARY WING AIRCRAFT

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Approved:

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Meaning of Symbols Used

g = 32.2 ft/sec/sec, acceleration of gravity

KE = Kinetic Energy

m = Mass per unit length of pendulum arm

M = Mass of pendulum bob (assumed to approximate point mass)

$\mathfrak{m}$ = Mass of whole ship, including pendulums

p = $n\Omega$ where n= order of harmonic being considered

Q = Generalized force

t = Time

X = Axis along axis of pendulum hinge offset arm

Y = Axis perpendicular to X and Z

Z = Axis along Axis of drive shaft

ψ = Angle of X-axis from down wind direction, measured positive in direction of rotation

ψ_i = Phase angle of harmonic forces

Ω = Rate of angular rotation of drive shaft = constant (average)

Thrust Pendulum:

a = Length of pendulum pivot offset from drive shaft axis

b = Length of pendulum arm

k_α = Gravity moment of pendulum about hinge/α_{STATIC}

N_0 = Amplitude of thrust unbalance

δ = Assumed deflection of whole ship under vibration

δ_o = Amplitude of vibration of whole ship

α = Angle of flap of pendulum absorber measured from α_{STATIC}, positive downwards

α_o = Maximum angle of flap

α_{ST} = Angle of flap at which centrifugal force of pendulum absorber balances gravity moment of pendulum arm and bob

Torque Pendulum:

c = Length of pendulum pivot offset from drive shaft axis

d = Length of pendulum arm

I_R = Moment of inertia of one blade about Z axis

K_L = Assumed spring constant brought into action by assumed existance of angle of rotation β

L_o = Amplitude of torque unbalance

β = Assumed angle of rotation between rotor blade and X axis

β_o = Maximum angle of rotation between rotor blade and X axis

γ_1 = Angle between d and line through c.g. of pendulum bob and Z axis in plane of rotation

γ_2 = Angle between d and line through c.g. of pendulum arm and Z axis in plane of rotation

ϕ = Angle of lag of torque pendulum absorber, positive in direction of rotation, Ω, measured from X axis

ϕ_o = Maximum angle of lag

Alternating Forces:

a_o = Slope of lift curve of blade airfoil section

C_{Do} = Blade profile drag coefficient

c_r = Blade root chord

c_t = Blade tip chord

C_Q = Torque coefficient = average torque/$\rho \pi \Omega^2 R^5$

C_T = Thrust coefficient = $T/\rho \pi \Omega^2 R^4$

L = Harmonic instantaneous torque

N = Harmonic instantaneous thrust

R = Blade radius

R_t = Blade taper ratio

T = Average thrust

V = Forward velocity, ft/sec.

V_i = Induced velocity at plane of rotation

w = Longitudinal induced velocity variation parameter

$\mathcal{W}$ = Gross weight of ship

z = $R_t - 1$

θ = Blade angle of incidence, radians

ϵ = Profile drag polar constant

λ = $V_i/\Omega R$ for particular ship considered

μ = $V/\Omega R$ for particular ship considered

σ_t = Rotor Solidity = $c_t/\pi R$ (one rotor)

χ = Ultimate angle of rotor slipstream downwash

ψ = Azimuth angle of blades, measured from downwind position in direction of rotation

Ω = Angular rotation of blades (average)

AN INVESTIGATION OF THE PRACTICABILITY OF TUNED PENDULUM TYPE VIBRATION ABSORBERS FOR USE ON ROTARY WING AIRCRAFT

SUMMARY

The tuning and amplitude equations derived herein for undamped pendulum type vibration absorbers may be applied, with slight modifications, equally well to any type of rotary wing aircraft to absorb the respective harmonic unbalanced forces and moments. Four assumptions were made that effect the derivations. One is that the angles involved are small enough so that the sine of the angle is equal to the angle itself, and the cosine equals one. The second is that the concentrated mass of the absorber approximates a point mass. The third is that the system is completely rigid, and the fourth is that the ship itself is in unaccelerated flight. These assumptions are involved in the design of the absorber itself, so that the degree of accuracy attained is dependent on the designer and the practical aspects of the problem to be solved.

The equations derived herein are limited to the case of forward flight at a constant velocity only. Conditions involving roll, pitch, or a combination of the two, and accelerated flight conditions are beyond the scope of this investigation.

A practical application is included in the paper, following the derivations. A first and second harmonic thrust absorber and a first

harmonic torque absorber are designed to indicate a method of attacking a problem for a ship having counterrotating, single bladed, counterweighted rotors without cyclic pitch thrust equalization. No experimental results are included, as none were available at the time of writing.

Methods of mounting the absorbers, the incorporation of centrifugal unloading devices which free the absorbers when the rotors reach operating speed, and position of pendulum stops are not treated in this paper.

PART I. Tuning and Amplitude Equations.

Consider the system composed of the rotor blades and the pendulum vibration absorbers, rotating about the axis of the drive shaft, Z, with a constant angular velocity. Superposed on this rotation are forcing torques of the form, $L_i = L_{io} \cos(pt-\psi_i)$, and forced vibrations of the form, $N_i = N_{io} \cos(pt-\psi_N)$. The former is due to the variation of the forces on the rotors in the plane of rotation, and the latter, to the variation of forces on the rotors in a direction perpendicular to the plane of rotation, acting along the axis of rotation.

Due to the placing of the vibration absorbers on the shaft, very nearly in the rotor plane of rotation, the vibrating forces for which the absorbers are tuned will not be transmitted through the shaft to the ship itself. However, for analytical purposes, assume that the forced vibration, N_i, acting along the shaft causes the whole ship of mass m, to vibrate with a vertical amplitude, S_o, causing its correspondingly tuned pendulum absorber to vibrate at an angle α. Assume that the forcing torque, L_i, causes the rotors to cyclically rotate at an angle β with respect to a line through the pivots of the torsional vibration absorbers rotating at the average rotor angular velocity, causing these absorbers to swing at an angle ϕ from their mean position.

The equations of motion are derived by the use of the LaGrange equations considering the relative motion of the system with respect to the uniformly rotating coordinate system, Ω. This method of arriving at the equations of motion of a moving system is covered in references (2), (3), (4), (5), and (7).

The system is assumed rigid enough so that no bending deflections occur.

A. Thrust absorber, single degree of freedom, flapping hinge only.

From figure 1, page 4, the coordinates of M are:

$$x_M = a + b\cos(\alpha + \alpha_{ST})$$

$$y_M = 0$$

$$z_M = \delta + b\sin(\alpha + \alpha_{ST})$$

differentiating with respect to time, t, yields

$$\dot{x}_M = -b\dot{\alpha}\sin(\alpha + \alpha_{ST})$$

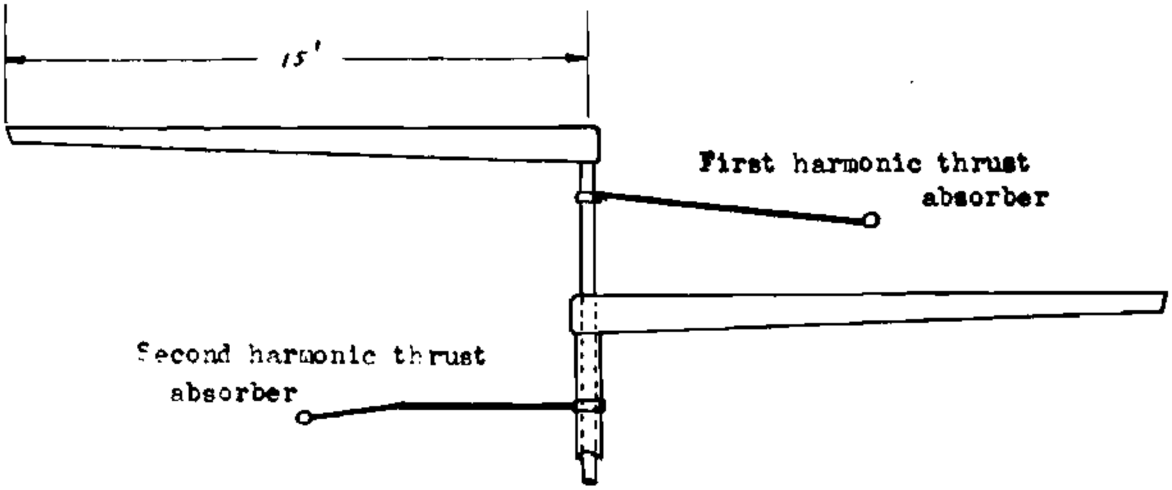
$$\dot{z}_M = \dot{\delta} + b\dot{\alpha}\cos(\alpha + \alpha_{ST})$$

Also from figure 1, the coordinates of mb are:

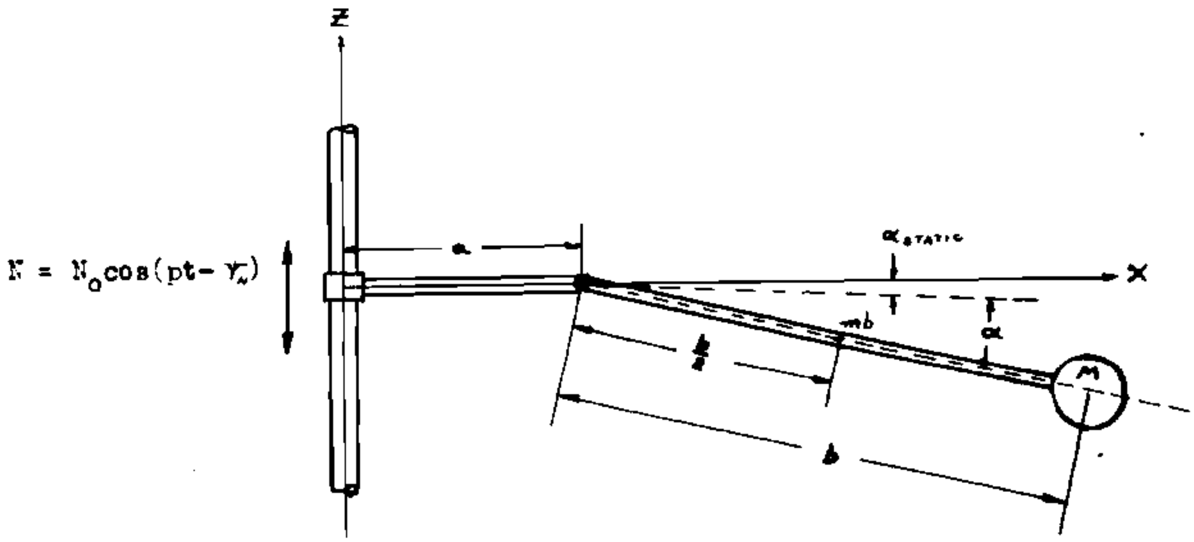
$$x_m = a + (b/2)\cos(\alpha + \alpha_{ST})$$

$$y_m = 0$$

$$z_m = \delta + (b/2)\sin(\alpha + \alpha_{ST})$$



(a) Position of absorbers on drive shaft.



(b) Schematic diagram of absorber.

FIGURE 1. Pendulum Thrust Absorber.

differentiating with respect to time, t, yields

$$\dot{x}_m = -(b/2)\sin(\alpha + \alpha_{ST})\,\dot{\alpha}$$

$$\dot{z}_m = \dot{\delta} + (b/2)\cos(\alpha + \alpha_{ST})\,\dot{\alpha}$$

Therefore

$$v_M^2 = \dot{x}_M^2 + \dot{y}_M^2 + \dot{z}_M^2 = b^2\dot{\alpha}^2 + \dot{\delta}^2 + 2b\,\dot{\delta}\dot{\alpha}\cos(\alpha + \alpha_{ST})$$

$$v_m^2 = \dot{x}_m^2 + \dot{y}_m^2 + \dot{z}_m^2 = (b/2)^2\dot{\alpha}^2 + \dot{\delta}^2 + b\,\dot{\delta}\dot{\alpha}\cos(\alpha + \alpha_{ST})$$

The Kinetic Energy of the system is then

$$KE = (1/2)M\left[b^2\dot{\alpha}^2 + \dot{\delta}^2 + 2b\,\dot{\delta}\dot{\alpha}\cos(\alpha + \alpha_{ST})\right] + (1/2)mb\left[(b/2)^2\dot{\alpha}^2 + \dot{\delta}^2 + b\,\dot{\delta}\dot{\alpha}\cos(\alpha + \alpha_{ST})\right] + (1/2)Mk_A^2\dot{\alpha}^2 + (1/2)mb\,b^2\dot{\alpha}^2/12 + (1/2)\left[\mathcal{M} - (M + mb)\right]\dot{\delta}^2. \qquad (1\text{-}A)$$

The LaGrange equation of motion takes the form

$$\frac{d}{dt}\left[\frac{\partial(KE)}{\partial\dot{q}_i}\right] - \frac{\partial(KE)}{\partial q_i} = Q_i$$

The generalized force with respect to δ is the undesirable force that is to be absorbed and is of the form

$$Q_\delta = N_0\cos(pt - \psi_N). \qquad (2\text{-}A)$$

where $p = n\Omega$ and 'n' represents the order of the harmonic force to be absorbed.

The generalized force with respect to α is considered as an effective spring constant times the deflection angle, α, of the pendulum.

$$Q_\alpha = -K_\alpha\alpha$$

and K_α is defined as

$$K_\alpha = \frac{\text{Gravity moment of pendulum weight about hinge}}{\alpha_{STATIC}}$$

and α_{STATIC} is defined as the angle of deflection at which the centrifugal restoring moment about the hinge line is equivalent to that of the gravity moment.

$$[(M + (1/2)mb]bg = M[a+b\cos\alpha_{ST}]\Omega^2 b\sin\alpha_{ST} + mb[a+(b/2)\cos\alpha_{ST}]\Omega^2(b/2)\sin\alpha_{ST}$$
$$+(1/12)mb\,\Omega^2 b^2\cos\alpha_{ST}\sin\alpha_{ST}.$$

where the first two terms on the right represent the centrifugal torque about the hinge axis of the pendulum bob and pendulum arm respectively due to their rotation about the shaft. The third term represents the centrifugal couple due to rotation of the arm about an axis parallel to the shaft and through its center of gravity tending to move the arm into a plane perpendicular to the axis of rotation. The centrifugal force due to rotation about the hinge axis has no effect as its line of action passes through the hinge axis. Now assuming α_{ST} small so that $\cos\alpha_{ST} = 1$; $\sin\alpha_{ST} = \alpha_{ST}$ and solving for α_{STATIC}

$$\alpha_{STATIC} = \frac{[M + (1/2)mb]g}{\{[M + (1/2)mb]a + [M + (1/3)mb]b\}\Omega^2}$$

and K_α then becomes

$$K_\alpha = \frac{g[M + (1/2)mb]\{[M + (1/2)mb]a + [M + (1/3)mb]b\}\Omega^2 b}{[M + (1/2)mb]g}$$

For which the generalized force with respect to α becomes

$$Q_\alpha = -\{[M + (1/2)mb]a + [M + (1/3)mb]b\}\Omega^2 b\ \alpha \qquad \text{(3-A)}$$

Assuming small oscillations, i.e. $\cos(\alpha + \alpha_{ST}) = 1$; $\sin(\alpha + \alpha_{ST}) = \alpha + \alpha_{ST}$

and that the pendulum bob mass approximates a point mass, the Kinetic Energy equation is

$$KE = (1/2)\left[M + (1/3)mb\right]b^2\dot{\alpha}^2 + (1/2)\mathcal{M}\dot{\delta}^2 + \left[M + (1/2)mb\right]b\,\dot{\delta}\dot{\alpha} \qquad (4\text{-A})$$

Applying LaGrange's equation for the coordinates δ and α gives respectively:

$$\mathcal{M}\ddot{\delta} + \left[M + (1/2)mb\right]b\,\ddot{\alpha} = N_0\cos(pt - \psi_\nu). \qquad (5\text{-A})$$

$$\left[M + (1/3)mb\right]b^2\ddot{\alpha} + \left[M + (1/2)mb\right]b\,\ddot{\delta} = - \left\{\left[M + (1/2)mb\right]a + \left[M + (1/3)mb\right]b\right\}\Omega^2 b\,\alpha \qquad (6\text{-A})$$

Assuming no damping, the variation of α and δ will be of the form:

$$\alpha = \alpha_0\cos(pt - \psi_\nu)$$

$$\delta = \delta_0\cos(pt - \psi_\nu)$$

Substituting in equations (5-A) and (6-A) give

$$-\mathcal{M}\delta_0 p^2\cos(pt - \psi_\nu) - \left[M + (1/2)mb\right]b\,\alpha_0 p^2\cos(pt - \psi_\nu) = N_0\cos(pt - \psi_\nu) \qquad (7\text{-A})$$

$$-\left[M + (1/3)mb\right]b\,\alpha_0 p^2\cos(pt - \psi_\nu) - \left[M + (1/2)mb\right]\delta_0 p^2\cos(pt - \psi_\nu) = - \Omega^2\left\{\left[M + (1/2)mb\right]a + \left[M + (1/3)mb\right]b\right\}\alpha_0\cos(pt - \psi_\nu) \qquad (8\text{-A})$$

Solving equations (7-A) and (8-A) for α_θ and δ_0

$$\alpha_0 = \frac{-\left[M + (1/2)mb\right]N}{\mathcal{M}\left[\left\{\left[M + (1/2)mb\right]a + \left[M + (1/3)mb\right]b\right\}\Omega^2 - \left[M + (1/3)mb\right]bp^2\right] + \left[M + (1/2)mb\right]^2 bp^2} \qquad (9\text{-A})$$

$$\delta_0 = \frac{\left[\left[M + (1/3)mb\right]bp^2 - \left\{\left[M + (1/2)mb\right]a + \left[M + (1/3)mb\right]b\right\}\Omega^2\right]N_0}{\mathcal{M}p^2\left[\left\{\left[M + (1/2)mb\right]a + \left[M + (1/3)mb\right]b\right\}\Omega^2 - \left[M + (1/3)mb\right]p^2 b\right] + \left[M + (1/2)mb\right]^2 bp^4} \qquad (10\text{-A})$$

If the ship does not vibrate, δ_0 must equal zero. Therefore

$$\left[M + (1/3)mb\right]bp^2 - \left\{\left[M + (1/2)mb\right]a + \left[M + (1/3)mb\right]b\right\}\Omega^2 = 0$$

The tuning equation then becomes, solving for 'p'

$$p = \Omega \left\{\frac{\left[M + (1/2)mb\right]a + \left[M + (1/3)mb\right]b}{\left[M + (1/3)mb\right]b}\right\}^{\frac{1}{2}}. \qquad (11\text{-}A)$$

The amplitude equation becomes

$$\alpha_0 = \frac{-N_0}{\left[M + (1/2)mb\right]bp^2} \qquad (12\text{-}A)$$

where α is measured from α_{STATIC}.

B. Torque absorber, single degree of freedom, lag hinge only.

From figure 2, page 9, the coordinates of M are:

$$x_M = c\cos\beta + d\cos(\beta + \phi)$$

$$y_M = c\sin\beta + d\sin(\beta + \phi)$$

$$z_M = 0$$

differentiating with respect to time, t, yields

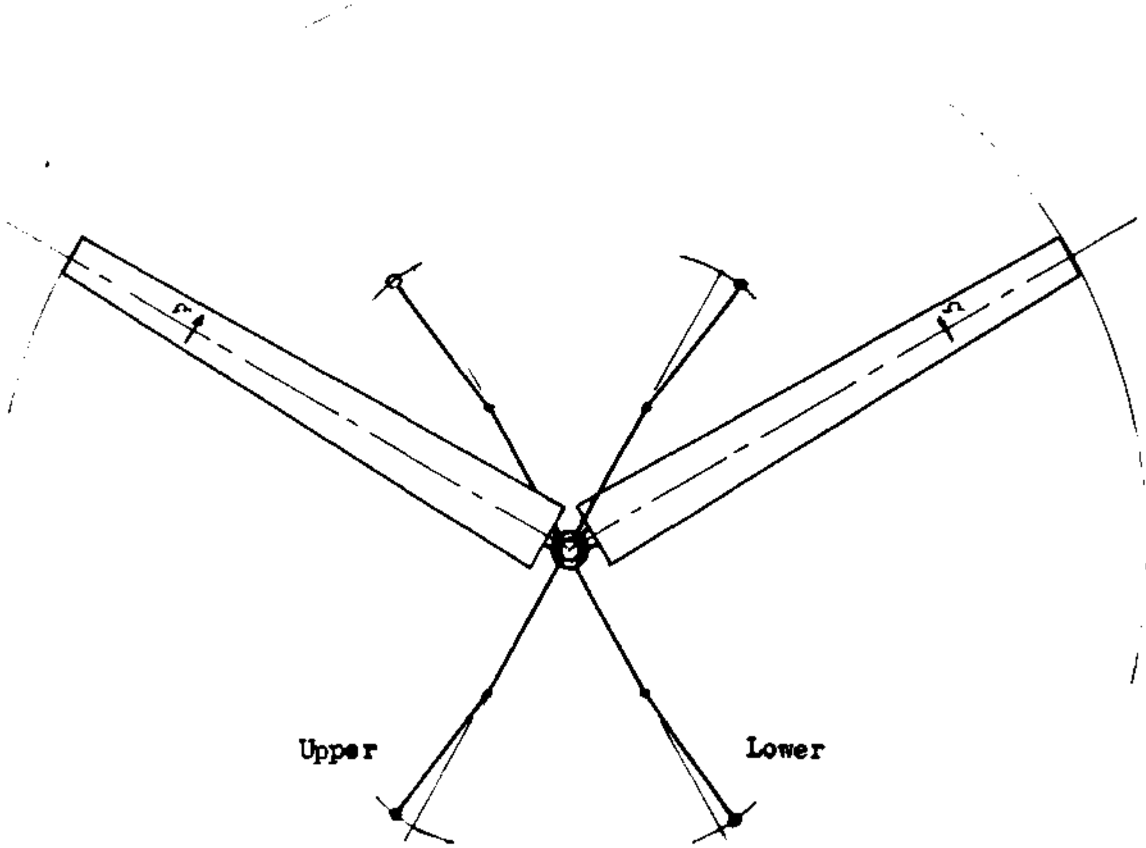
$$\dot{x}_M = -c\,\dot{\beta}\sin\beta - d(\dot{\beta} + \dot{\phi})\sin(\beta + \phi)$$

$$\dot{y}_M = c\,\dot{\beta}\cos\beta + d(\dot{\beta} + \dot{\phi})\cos(\beta + \phi)$$

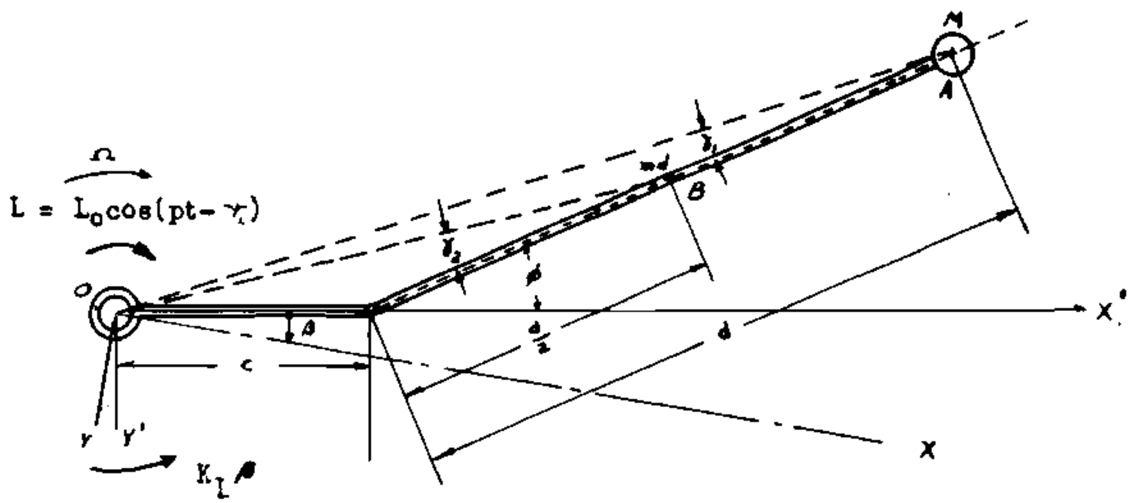
Also from figure 2, the coordinates of md are

$$x_m = c\cos\beta + (d/2)\cos(\beta + \phi)$$

$$y_m = c\sin\beta + (d/2)\sin(\beta + \phi)$$



(a) Position of absorbers on drive shaft.



(b) Schematic diagram of absorber.

FIGURE 2. Pendulum Torque Absorber.

differentiating with respect to time, t, yields

$$\dot{x}_m = -c\dot{\beta}\sin\beta - (d/2)(\dot{\beta} + \dot{\phi})\sin(\beta + \phi)$$
$$\dot{y}_m = c\dot{\beta}\cos\beta + (d/2)(\dot{\beta} + \dot{\phi})\cos(\beta + \phi)$$

Therefore

$$v_M^2 = \dot{x}_M^2 + \dot{y}_M^2 = c^2\dot{\beta}^2 + d^2(\dot{\beta} + \dot{\phi})^2 + 2cd\dot{\beta}(\dot{\beta} + \dot{\phi})\cos\phi$$
$$v_m^2 = \dot{x}_m^2 + \dot{y}_m^2 = c^2\dot{\beta}^2 + (d/2)^2(\dot{\beta} + \dot{\phi})^2 + cd\dot{\beta}(\dot{\beta} + \dot{\phi})\cos\phi$$

The Kinetic Energy of the system is

$$\begin{aligned} KE = {} & (1/2)M\left[c^2\dot{\beta}^2 + d^2(\dot{\beta} + \dot{\phi})^2 + 2cd\dot{\beta}(\dot{\beta} + \dot{\phi})\cos\phi\right] \\ & + (1/2)md\left[c^2\dot{\beta}^2 + (d/2)^2(\dot{\beta} + \dot{\phi})^2 + cd\dot{\beta}(\dot{\beta} + \dot{\phi})\cos\phi\right] \\ & + (1/2)Mk_M^2(\dot{\beta} + \dot{\phi})^2 + (1/2)md\,d^2(\dot{\beta} + \dot{\phi})^2/12 + (1/2)I_R\dot{\beta}^2 \end{aligned} \qquad (1\text{-B})$$

The LaGrange equation of motion takes the form

$$\frac{d}{dt}\left[\frac{\partial(KE)}{\partial\,\dot{q}_i}\right] - \frac{\partial(KE)}{\partial\,q_i} = Q_i \quad .$$

The generalized force with respect to β is

$$Q_\beta = L_0\cos(pt - \psi_L) - K_L\,\beta \quad . \qquad (2\text{-B})$$

The generalized force with respect to ϕ is composed of a centrifugal restoring torque and a Coriolis torque due to the conservation of angular momentum, both torques acting about the hinge axis. From figure 2, page 9

$$\text{Centrifugal torque} = M\Omega^2\overline{OA}\,d\,\sin\gamma_1 + md\,\Omega^2\overline{OB}(d/2)\sin\gamma_2 \quad .$$

However

$$\sin \gamma_1 = \frac{c \sin \phi}{OA} \qquad \text{and} \qquad \sin \gamma_2 = \frac{c \sin \phi}{OB} \quad .$$

Then

$$\text{Centrifugal torque} = M\Omega^2 cd \sin \phi + (1/2) md\Omega^2 cd \sin \phi \quad .$$

and the

$$\text{Coriolis Torque} = 2M\Omega\, d^2 \dot{\phi} \sin \phi + 2md\Omega\,(d/2)^2 \dot{\phi} \sin \phi \quad .$$

The generalized force with respect to ϕ is then

$$Q_\phi = -\left[M + (1/2)md\right]\left[\Omega^2 cd - 2\Omega d^2 \dot{\phi}\right] \sin \phi \quad . \tag{3-B}$$

Assuming small oscillations, i.e. $\cos \phi = 1$; $\sin \phi = \phi$, and that the pendulum bob approximates a point mass, equations (1-B) and (3-B) become

$$\begin{aligned} KE = \ & (1/2)M\left[c^2\dot{\beta}^2 + d^2(\dot{\beta} + \dot{\phi})^2 + 2cd\dot{\beta}(\dot{\beta} + \dot{\phi})\right] \\ & + (1/2)md\left[c^2\dot{\beta}^2 + (d/2)^2(\dot{\beta} + \dot{\phi})^2 + cd\dot{\beta}(\dot{\beta} + \dot{\phi})\right] \\ & + (1/24)mdd^2(\dot{\beta} + \dot{\phi})^2 + (1/2)I_R\dot{\beta}^2 \end{aligned} \tag{4-B}$$

$$Q_\phi = -\left[M + (1/2)md\right]\left[\Omega^2 cd - 2\Omega d^2\dot{\phi}\right]\phi \tag{5-B}$$

Applying LaGrange's equation for the coordinates β and ϕ gives respectively

$$\begin{aligned} &\left\{I_R + M\left[c^2 + d^2 + 2cd\right] + md\left[c^2 + (1/3)d^2 + cd\right]\right\}\ddot{\beta} + \left\{M\left[d^2 + cd\right] + md\left[(1/3)d^2 + (1/2)cd\right]\right\}\ddot{\phi} \\ &\qquad = L_0 \cos(pt - \psi_L) - K_L\,\beta \end{aligned} \tag{6-B}$$

$$\begin{aligned} &\left\{M\left[d + c\right] + md\left[(1/3)d + (1/2)c\right]\right\}\ddot{\beta} + \left[Md + (1/3)md^2\right]\ddot{\phi} \\ &\qquad = -\left[M + (1/2)md\right]\left[\Omega^2 c\phi - 2\Omega d\dot{\phi}\,\phi\right] \end{aligned} \tag{7-B}$$

Assuming no damping, the variation of β and ϕ will be of the form:

$$\beta = \beta_o \cos(pt - \psi_1)$$

$$\phi = \phi_o \cos(pt - \psi_1)$$

Substituting in equations (6-B) and (7-B) gives:

$$-\{I_R + M[c^2+d^2+2cd] + md[c^2+(1/3)d^2+cd]\}\beta_o p^2 \cos(pt-\psi_1)$$
$$-\{M[d^2+cd] + md[(1/3)d^2+(1/2)cd]\}\phi_o p^2 \cos(pt-\psi_1)$$
$$= L_o \cos(pt-\psi_1) - K_L \beta_o \cos(pt-\psi_1) \tag{8-B}$$

$$-\{M[d+c] + md[(1/3)d+(1/2)c]\}\beta_o p^2 \cos(pt-\psi_1)$$
$$-\{Md + (1/3)md^2\}\phi_o p^2 \cos(pt-\psi_1)$$
$$= -[M+(1/2)md][\Omega^2 c\, \phi_o \cos(pt-\psi_1) + 2\Omega p \phi_o^2 \sin(pt-\psi_1)\cos(pt-\psi_1)d] \tag{9-B}$$

Neglecting the coriolis torque for small arcs, equations (8-B) and (9-B) may be solved for β_o and ϕ_o. Then

$$\phi_o = \frac{\{M[d+c] + md[(1/3)d+(1/2)c]\}p^2 L_o}{[K_L - \{I_R + M[c^2+d^2+2cd] + md[c^2+(1/3)d^2+cd]\}p^2][[M+(1/2)md]\Omega^2 c - [Md+(1/3)md^2]p^2] - \{M[d+c] + md[(1/3)d+(1/2)c]\}^2 dp^4} \tag{10-B}$$

$$\beta_o = \frac{\{[M+(1/2)md]\Omega^2 c - [Md+(1/3)md^2]p^2\}L_o}{[K_L - \{I_R + M[c^2+d^2+2cd] + md[c^2+(1/3)d^2+cd]\}p^2][[M+(1/2)md]\Omega^2 c - [Md+(1/3)md^2]p^2] - \{M[d+c] + md[(1/3)d+(1/2)c]\}^2 dp^4} \tag{11-B}$$

If there is no torsional vibration, β_o must equal zero. Therefore

$$[M + (1/2)md]\Omega^2 c - [Md + (1/3)md^2]p^2 = 0$$

The tuning and amplitude equations then become

$$p = \Omega\left\{\frac{c[M + (1/2)md]}{d[M + (1/3)md]}\right\}^{\frac{1}{2}} \tag{12-B}$$

$$\phi_o = \frac{-L_o}{\{Md[d+c] + md^2[(1/3)d + (1/2)c]\}p^2} \tag{13-B}$$

PART II. Alternating Forces to be Absorbed.

The variations in forces are calculated by means of the formulae derived in reference (1) for rotary wing aircraft with counter rotating, single bladed rotors so controlled that the planes of rotation remain horizontal and the ship is propelled by a pusher propeller. The blades are so mounted on the drive shaft and the shafts so phased by gearing, that they cross in a plane through the drive shaft perpendicular to the longitudinal axis of the ship, which is assumed parallel to the flight path. This configuration results in a force-phase angle relationship of 180 degrees between the two blades. For the first harmonic thrust and torque components, the advancing blade approaches its maximum value and the receding blade approaches its minimum as the two blades cross. The maximum and minimum values do not occur at the crossing point due to a phase angle with respect to the zero reference line. For the second harmonic thrust component, the two blades approach a maximum and a minimum together. That is, as the blades cross, the second harmonic thrust component for both blades reaches its maximum value. Higher torque harmonics than of the first order and thrust harmonics of the second order are neglected.

The procedure is as follows. The necessary thrust coefficient is calculated from the design specifications. Having this value, the induced velocity, V_i, and the ultimate angle of slipstream downwash, χ, may be found. The blade angle of incidence, θ, is found from the constant portion of the C_T equation, and the different values of the harmonic forces are calculated by separating the thrust and torque equations into their component harmonic parts.

Design Specifications:

Gross weight of ship .... 1800 lbs.

Design velocity, V .... 100 MPH

Angular rotation of blades............ 25 radians/second

Blade taper ratio, R_t 3

Blade radius .... 15 feet

$z = R_t - 1 = 2$

$\rho = 0.002378$

Airfoil Section:

N.A.C.A. 23015

Slope of lift curve, $a_o = 2\pi$

Profile drag coefficient, $C_{Do} = C_{Doo} + \epsilon(C_L/a_o)^2$

$C_{Doo} = 0.012$

$\epsilon = 0.894$

Rotor weight:

• Blade weight = 80 lbs. at 40% span (radius)

From reference (1) the following equations are used.

$$C_T = T/\rho\pi\Omega^2 R^4 \tag{1}$$

$$\mu = V/\Omega R \tag{2}$$

$$\lambda = V_i/\Omega R \quad \text{(for level flight with pusher propeller)} \tag{3}$$

$$V_i = C_T \Omega R/(\mu^2 + \lambda^2)^{\frac{1}{2}} 2 = C_T \Omega^2 R^2/(V^2 + V_i^2)^{\frac{1}{2}} 2 \tag{4}$$

$$\chi = \text{arc cot}\left\{[\lambda/\mu] + [C_T/2\mu(\mu^2+\lambda^2)^{1/2}]\right\} \tag{5}$$

$$w = [V_i/\Omega R]\tan(\chi/2) \tag{6}$$

= longitudinal induced velocity variation parameter.

$$C_T = (a_o \sigma_t/2)\Big[\Big\{\theta R_t[(1/3)+(\mu^2/2)]-(1/2)R_t\lambda - \theta z[(1/4)+(\mu^2/4)]$$
$$+(1/3)z\lambda +(1/4)R_t\lambda\mu^2 -(1/32)z\theta\mu^4\Big\}$$
$$+\Big\{R_t\theta\mu - R_t\lambda\mu -(2/3)z\theta\mu +(1/2)z\lambda\mu -(1/4)R_t\theta\mu^3+(1/8)z\lambda\mu^3\Big\}\sin\psi$$
$$+\Big\{(-1/3)R_t w + (1/4)zw - (1/96)zw\mu^4\Big\}\cos\psi$$
$$+\Big\{(-1/2)R_t w\mu +(1/6)zw\mu -(1/24)R_t w\mu^3\Big\}\sin 2\psi$$
$$+\Big\{(-1/2)R_t\theta\mu^2+(1/4)z\theta\mu^2-(1/4)R_t\lambda\mu^2+(1/24)z\theta\mu^4\Big\}\cos 2\psi$$
$$+ \text{higher harmonics}\Big] \tag{7}$$

$$C_Q = (\sigma_t/2)\Big[\Big[C_{Doo}+\epsilon\theta^2\Big]\Big[(1/4)R_t+(1/4)R_t\mu^2-(1/5)z -(1/6)z\mu^2$$
$$+(1/32)R_t\mu^4\Big] + \Big[a_o\theta -2\epsilon\theta\Big]\Big[(1/3)R_t\lambda -(1/4)z\lambda +(1/32)z\lambda\mu^4\Big]$$
$$+ \Big[\epsilon - a_o\Big]\Big[(1/2)R_t\lambda^2+(1/8)R_t w^2-(1/3)z\lambda^2-(1/10)zw^2$$
$$+(1/4)R_t\mu^2\left(\lambda^2+(11/16)w^2\mu^2\right)\Big]$$
$$+ \Big\{\Big[C_{Doo}+\epsilon\theta^2\Big]\Big[(2/3)R_t\mu -(1/2)z\mu +(1/48)z\mu^5\Big]$$
$$+ \Big[a_o\theta -2\epsilon\theta\Big]\Big[(1/2)\lambda\mu R_t-(1/3)z\lambda\mu +(1/8)R_t\mu^3\lambda\Big]$$
$$+ \Big[\epsilon - a_o\Big]\Big[(1/4)z\lambda^2\mu^3\Big]\Big\}\sin\psi$$
$$+\Big\{\Big[a_o\theta -2\epsilon\theta\Big]\Big[(1/4)wR_t+(1/3)wR_t\mu -(1/5)zw -(1/4)zw\mu$$
$$-(1/32)zw\mu^5 +(1/32)R_t w\mu^4\Big]$$
$$+ \Big[\epsilon - a_o\Big]\Big[(2/3)R_t w\lambda -(1/2)zw\lambda -(1/16)zw\lambda\mu^4\Big]\Big\}\cos\psi$$
$$+ \text{higher harmonics}\Big] \tag{8}$$

Substituting the proper values in equations (1) through (6) give:

$$C_T = 0.00763$$
$$\mu = 0.391$$
$$V_i = 3.66 \text{ ft/sec}$$
$$\lambda = 0.00976$$
$$\chi = 87^\circ\ 8'$$
$$w = 0.00927$$

Equating the constant part of equation (7) to the value of equation (1), since both of these values represent the average over the rotor disk, and solving for the blade angle of incidence, θ , gives for both blades:

$$C_T = 2(a_0 \sigma_t/2) \Big\{ \theta R_t \left[(1/3)+(\mu^2/2)\right] -(1/2) R_t \lambda - \theta z \left[(1/4)+(\mu^2/4)\right] \\ +(1/3) z \lambda +(1/4) R_t \lambda \mu^2 -(1/32) z \theta \mu^4 \Big\}$$

from which

$$\theta = 0.0984 \text{ radians or } 5.64^\circ \; .$$

A. First harmonic thrust component.

The first harmonic thrust component for one blade may be found by solving terms 2 and 3 in equation (7). For two blades, substitute for $\sin\psi$ $\left[\sin\psi + \sin(180 + \psi)\right]$ and for $\cos\psi$, $\left[\cos\psi + \cos(180 + \psi)\right]$. These two substitutions are zero, so the result is that the first harmonic thrust variation cancels between the two blades. This result could have been seen immediately since the first harmonic thrust phase angle relationship between the two blades is 180 degrees.

B. Second harmonic thrust component.

The second harmonic thrust component may be found similarly from equation (7), using the fourth and fifth terms. For two blades this becomes

$$N_2 = (a_0 \sigma_t/2) \Big[\Big\{ w \mu \left[(-1/2) R_t + (1/6) z\right] - (1/24) R_t w \mu^3 \Big\} \left[\sin 2\psi + \sin 2(180+\psi)\right] \\ + \Big\{ \theta \mu^2 \left[(-1/2) R_t + (1/4) z\right] - (1/4) R_t \lambda \mu^2 \\ + (1/24) z \theta \mu^4 \Big\} \left[\cos 2\psi + \cos 2(180+\psi)\right] \Big] \rho \pi \Omega^2 R^4$$

which upon substituting values gives

$$N_2 = -50 \sin 2\psi - 505 \cos 2\psi$$

$$N_2 = -509 \cos\left[2 \Omega t - (2/63)\pi\right] \qquad \text{(pounds)}$$

C. First harmonic torque component.

The first harmonic torque component is found in a like manner except that the component is found for each blade separately.

$$
\begin{aligned}
L_1 = (\sigma_t/2) \Big[& \Big\{ \big[C_{Doo} + \epsilon\,\theta^2\big] \big[(2/3) R_t \mu - (1/2) z \mu + (1/48) z \mu^5\big] \\
& + \theta \big[a_0 - 2\epsilon\big] \big[(1/2) \lambda \mu R_t - (1/3) z \lambda \mu + (1/8) R_t \lambda \mu^3\big] \\
& + \big[\epsilon - a_0\big] \big[(1/4) z \lambda^2 \mu^3\big] \Big\} \sin\psi \\
& + \Big\{ \theta \big[a_0 - 2\epsilon\big] \big[(1/4) R_t + (1/3) R_t \mu - (1/5) z - (1/4) z \mu - (1/32) z \mu^5 \\
& + (1/32) R_t \mu^4\big] w + \big[\epsilon - a_0\big] \big[(2/3) R_t - (1/2) z - (1/16) z \mu^4\big] \lambda w \Big\} \cos\psi \Big] \rho \pi \Omega^2 R^5
\end{aligned}
$$

which upon substituting values gives

$$L_1 = 358 \sin\psi + 66 \cos\psi$$

$$L = 364 \cos\left[\Omega t - (4/9)\pi\right] , \qquad \textbf{(foot-pounds)}$$

the first harmonic torque component.of one rotor.

PART III. Design of Pendulum Absorbers.

Three tuned pendulum type vibration absorbers are to be designed. One tuned to the first harmonic thrust component, one tuned to the second harmonic thrust component and the last, tuned to the first harmonic torque component.

The thrust vibration absorbers are designed to act not only as vibration absorbers, but also as counterweights for the single bladed rotors. This will necessitate mounting the absorbers on the drive shaft at some distance below the blade, so that the average centrifugal moment of the absorber about the intersection of the blade axis and drive shaft axis will be equal and opposite to the average thrust moment about the same point. The first harmonic thrust absorber is to be mounted between the rotor planes of rotation, acting as a first harmonic vibration absorber for both blades, and as a counterweight for the top blade. The second harmonic thrust absorber is to be mounted on the drive shaft below the lower blade, acting as a second harmonic vibration absorber for both blades, and as a counterweight for the lower blade. By so designing the thrust absorbers, no additional weight need be added which is an important consideration in the design of rotary wing aircraft.

It was shown in PART II, that the first harmonic thrust component cancels out between the two blades due to a phase angle difference of 180 degrees, thereby making a first harmonic thrust absorber unnecessary. However, since it does not add any additional weight to the ship, the counterweight of the top rotor will be designed as a first harmonic thrust absorber. This will allow for any first harmonic transient vibrations, and the possibility that the forces between the two blades do not cancel. Such a condition would occur in a yawed flight condition, which is not treated in this paper.

The first harmonic torque component, however, must be absorbed by two pairs of tuned pendulums, one pair mounted on the drive shaft at each blade. This is necessary because the manner in which the engine is geared to the counterrotating drive shafts, causes the torque unbalances between the two blades to add absolutely, instead of algebraically. The advantage gained by using two pairs of absorbers, being that of absorbing the vibration before it is transmitted into the drive shaft.

The pendulum arms are designed from stainless steel streamline tubing. A load factor of four in tension and two in bending is used to allow for possible overspeeding of the rotors and design is made to the ultimate strength of the material. The design is checked for fatigue at normal operating conditions with a load factor of one, in tension and in bending.

The weight of each blade is eighty pounds and its center of gravity is at the 40% radius which for this particular ship is six feet. Arbitrarily setting seven feet as the distance from shaft axis to pendulum center of gravity, it is found that the necessary mass of each counterweight is 2.13 lbs-sec^2/ft.

A. Design of first harmonic thrust pendulum.

From PART I, Section A, the tuning equation for the thrust absorber is found to be

$$p = \Omega \left\{ \frac{[M + (1/2)mb]a + [M + (1/3)mb]b}{[M + (1/3)mb]b} \right\}^{\frac{1}{2}}$$

Which for a first harmonic variation, p = , becomes

$$1 = \frac{[M + (1/2)mb]a + [M + (1/3)mb]b}{[M + (1/3)mb]b} .$$

Assuming mb = .150 M; this equation then becomes

$$M(1 + .075)a + M(1 + .050)b = M(1 + .050)b$$

or

$$\frac{a}{b} = 0 .$$

Setting the limitation that $a + b = 7.5'$, gives

$$0 + b = 7.5$$

$$b = 7.5'$$

$$a = 0 .$$

The mass of the bob and arm are

$$M + mb = 2.13 \text{ lbs.sec}^2/\text{ft}$$

$$M(1 + .15) = 2.13 \text{ lbs.sec}^2/\text{ft}$$

$$M = 1.85 \text{ lbs.sec}^2/\text{ft}$$

$$mb = .28 \text{ lbs.sec}^2/\text{ft}$$

Check on assumed pendulum center of gravity of 7.0'

Moment about hinge = $68.6r = 7.5(59.6) + (7.5/2)(9.0)$

$$r = 7.0'$$

Distance to c.g. = a + r = 0 + 7.00 = 7.00'

Use stainless steel streamline tubing (reference 6)

Equivalent round = 2''X.058''

Major axis = 2.70''; Minor axis = 1.14''

Area = 0.354 in^2 ; wt/ft = 1.203 lbs.; Z(minor) = .1732

Fatigue strength = 75,000psi

Ultimate strength = 185,000 psi

Critical station will be at hinge pin.for the overspeed condition.

$$\text{Tensile stress, } f_t = 4\left[M\Omega^2(a+b) + mb\,\Omega^2(a+b/2)\right]/A$$

$$f_t = 4\left[(1.85)(25)^2(7.5)+(.28)(25)^2(3.75)\right]/.354$$

$$f = 104,200 \text{ psi}$$

Fatigue is not considered since first harmonic thrust components cancel, and pendulum does not oscillate constantly.

$$\text{Margin of safety} = \frac{185,000}{104,200} - 1 = .77$$

B. Design of second harmonic thrust absorber.

From PART I, Section A, The tuning equation for the thrust absorber is found to be

$$p = \Omega\left\{\frac{[M+(1/2)mb]\,a + [M+(1/3)mb]\,b}{[M+(1/3)mb]\,b}\right\}^{\frac{1}{2}}$$

which for a second harmonic variation, p = 2 Ω , becomes

$$4 = \frac{[M+(1/2)mb]\,a + [M+(1/3)mb]\,b}{[M+(1/3)mb]\,b}$$

Assuming mb = .0375 M, this equation in turn becomes

$$M(1 + .01875)a + M(1 + .0125)b = 4M(1 + .0125)b$$

or

$$\frac{a}{b} = \frac{3.0375}{1.0187}$$

Setting the limitation that $a + b = 7.5'$, gives

$$1.0187(7.5 - b) = 3.0375\ b$$

$$b = 1.88'$$

$$a = 5.62'$$

From Section A, above, the mass of the bob is found to be 1.85 lbs.sec^2/ft. Therefore

$$mb = .0375(1.85) = .0694 \text{ lbs.sec}^2/\text{ft.}$$

The amplitude equation from PART I, Section A is

$$\alpha_o = \frac{-N_o}{[M + (1/2)mb]bp^2} \quad .$$

The value of N_o is 509 lbs. from PART II, Section B. The amplitude of oscillations is then

$$\alpha_o = \frac{-509}{[1.85 + (1/2)(.0694)](1.88)(25^2)(2)^2}$$

$$\alpha_o = -.0569 \text{ radians or} -3.16^\circ$$

which is in the allowable range of small angles.

Use the same size streamline tubing for the second as for the first harmonic absorber. The critical station is at the shaft. Since both absorbers are identical as to mass distribution, the tensile stress in the tube at the shaft will be the same as for the first plus an additional term due to $\dot{\alpha}$.

$$f_t = 26{,}300 + \left[Mb\dot{\alpha}^2 + mb(b/2)\dot{\alpha}^2\right]/A$$

$$\alpha = \frac{-N_0\cos(pt-\psi_\nu)}{[M+(1/2)mb]\,bp^2}$$

$$\dot{\alpha} = \frac{N_0 p\sin(pt-\psi_\nu)}{[M+(1/2)mb]\,bp^2}$$

$$\dot{\alpha} = 509/(1.88)(1.88)(2)(25)$$

$$\dot{\alpha} = 2.86 \text{ radians/sec.}$$

$$f_t = 26{,}300 + \left[(1.88)(1.85)+(.0694)(1.88/2)\right](2.86)^2/.354$$

$$f_t = 26{,}400 \text{ psi}$$

The moment at the shaft due to the conservation of angular momentum when the pendulum moves through an angle is found as follows:

$$\text{Moment at shaft} = 2M\Omega(a+b)^2\dot{\alpha}\alpha + 2mb\,\Omega\,(a+b/2)^2\dot{\alpha}\alpha$$

where α and $\dot{\alpha}$ are found as above, and the moment is a maximum when $(pt-\psi_\nu) = \pi/4$ at which $\alpha = .0402$ radians and $\dot{\alpha} = 2.02$ radians/sec.

$$\text{Moment at shaft} = \left[2(1.85)(25)(7.5)^2+2(.0694)(25)(6.56)^2\right](.0402)(2.02)$$

$$= 435 \text{ ft-lbs.}$$

$$\text{Bending stress, } f_b = \text{Moment}/Z = 435(12)/(.1732)$$

$$f_b = 30{,}200 \text{ psi}$$

Then the maximum tensile stress = 26,400 + 30,200

= 56,600 psi

Margin of safety = 75,000/56,600 - 1 = .322

Margin of safety in overspeed condition:

$$f_T = 4(26{,}400) + 2(30{,}200) = 165{,}800 \text{ psi}$$

Margin of safety = 185,000/165,800 - 1 = .117

C. Design of first harmonic torque pendulum.

From PART I, Section B, the tuning equation for the torque absorber is found to be

$$p = \Omega \left\{ \frac{c[M + (1/2)md]}{d[M + (1/3)md]} \right\}^{\frac{1}{2}}$$

which for a first harmonic variation, $p = \Omega$, becomes

$$\frac{c[M + (1/2)md]}{d[M + (1/3)md]} = 1 \quad .$$

From PART I, Section B, the amplitude equation is found to be

$$\phi_o = \frac{-L_o}{[Md(d+c) + md^2(d/3 + c/2)]\, p^2}$$

Setting $|\phi_o| < 12^o$ for which $\sin \phi_o \approx \phi_o$ and $\cos \phi_o \approx 1$, the amplitude equation becomes, for a first harmonic variation, and from PART II, Section C, for $L_o = (364/2)$ (since a pair of pendulums is to be used in each absorber)

$$-.208 = \frac{-182}{\{[M + (1/3)md]d + [M + (1/2)md]c\} d\Omega^2}$$

or

$$\frac{[M + (1/2)md]c}{[M + (1/3)md]d} = \frac{35}{[M + (1/3)md]d^2} - 1 \quad .$$

As a trial solution, take $md = .10\ M$. Then substituting into

$$\frac{c[M + (1/2)md]}{d[M + (1/3)md]} = 1$$

$$\frac{cM(1 + .050)}{dM(1 + .033)} = 1$$

$$\frac{c}{d} = \frac{1.033}{1.050} \quad .$$

Setting the limitation that $c + d = 7.5'$

$$c = 7.5 - d = (1.037/1.050)\ d$$

$$d = 3.78'$$

$$c = 3.72'$$

Then combining the tuning and amplitude equations gives

$$\frac{[M + (1/2)md]c}{[M + (1/3)md]d} = 1 = \frac{35}{[M + (1/3)md]d^2} - 1$$

or

$$\frac{35}{M(1 + .033)d^2} = 2$$

and

$$M = 1.194 \text{ lbs-sec}^2/\text{ft}$$

$$md = .1194 \text{ Lbs-sec}^2/\text{ft}$$

For a design of the pendulum arm, try stainless steel streamline tubing (reference 6)

Equivalent Round = 2''X .049''

Major axis = 2.70''; Minor axis = 1.14''

Area = .3003 in^2; Wt/ft =1.021 lbs; Z(major)=.0893

Fatigue strength = 75,000 psi

Ultimate strength = 125,000 psi

Critical station will be at hinge pin. The tensile stress,

$$f_t = \left[M\Omega^2(c+d) + md\Omega^2(c+d/2) + (Md + (1/2)md^2)\dot{\phi}^2\right] \quad /\text{Area}$$

$$\phi = \frac{-L_o\cos(pt - \psi_L)}{\{[M + (1/3)md]d + [M + (1/2)md]c\}d\Omega^2}$$

$$\dot{\phi} = \frac{L_o\sin(pt - \psi_L)}{\{[M + (1/3)md]d + [M + (1/2)md]c\}d\ \Omega}$$

$$\dot{\phi} = \frac{L_o}{\{[M + (1/3)md]d + [M + (1/2)md]c\}d\Omega} = .254 \text{ rad/sec.}$$

$$f_t = \left\{\left[(1.194)(7.5)+(.1194)(5.61)\right](25)^2 + \left[(1.194)(3.78)+(1/2)(.1194)(3.78)\right](.254)^2\right\} /.3003$$

$$f_t = 19{,}800 \text{ psi}$$

Bending stress, f_b = Moment about hinge/ Z

$$\text{Moment} = g(M + md/2)d$$

$$= 125 \text{ ft-lbs.}$$

$$f_b = (125)(12)/(.0893)$$

$$f_b = 16{,}800 \text{ psi}$$

Total stress (tensile) = 19,800 + 16,800

= 36,600 psi

Margin of Safety = (75,000/36,600) - 1

= 1.05

Margin of Safety in overspeed condition:

Tensile stress (total) = 4(19,800) + 2(16,800)

= 112,800 psi

Margin of Safety = (125,000/112,800) - 1 = .11

CONCLUSIONS

Thus, for the particular ship under consideration, the theory developed in this paper indicates that the use of pendulum type thrust and torque variation absorbers, in lieu of cyclic pitch control, is practical insofar as forward flight at a constant speed is concerned. Due to the incorporation of the counterweights as thrust variation absorbers, the only additional weight is from the first harmonic torque absorbers. This additional weight amounts to two-hundred and sixty-nine pounds and is probably prohibitive in magnitude.

Since no experimental verification has, as yet, been made on the practicability of such absorbers, it appears that such experimentatation would be well worth while before application of the theory to an actual design.

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APPENDIX

DESIGN CHARTS

Included in this appendix are design charts for the tuning and for the determination of the amplitude for both the thrust and torque absorbers. Only the thrust absorber will be illustrated for the design of the torque absorber is exactly similar. The method is as follows.

Enter the tuning chart at the order of the undesirable force to be absorbed. Choose a suitable M/mb ratio and read the corresponding a/b ratio on the ordinate. Both a and b may then be determined from the practical limitations placed on the length of the pendulum. The value of $N_0/amb(M/mb + 1/2)p^2$ may then be calculated for the given problem. Having this value, enter the Amplitude Chart at the previously determined a/b ratio. The amplitude is then read on the ordinate.

Should the value of the amplitude read be undesirable due to the limitations of the problem, a second choice of M/mb should be made and the process repeated. For most cases, only one trace through the charts need be made.

The included charts, therefore, offer a quick, simple method for designing unbalanced force absorbers of the type herein treated.

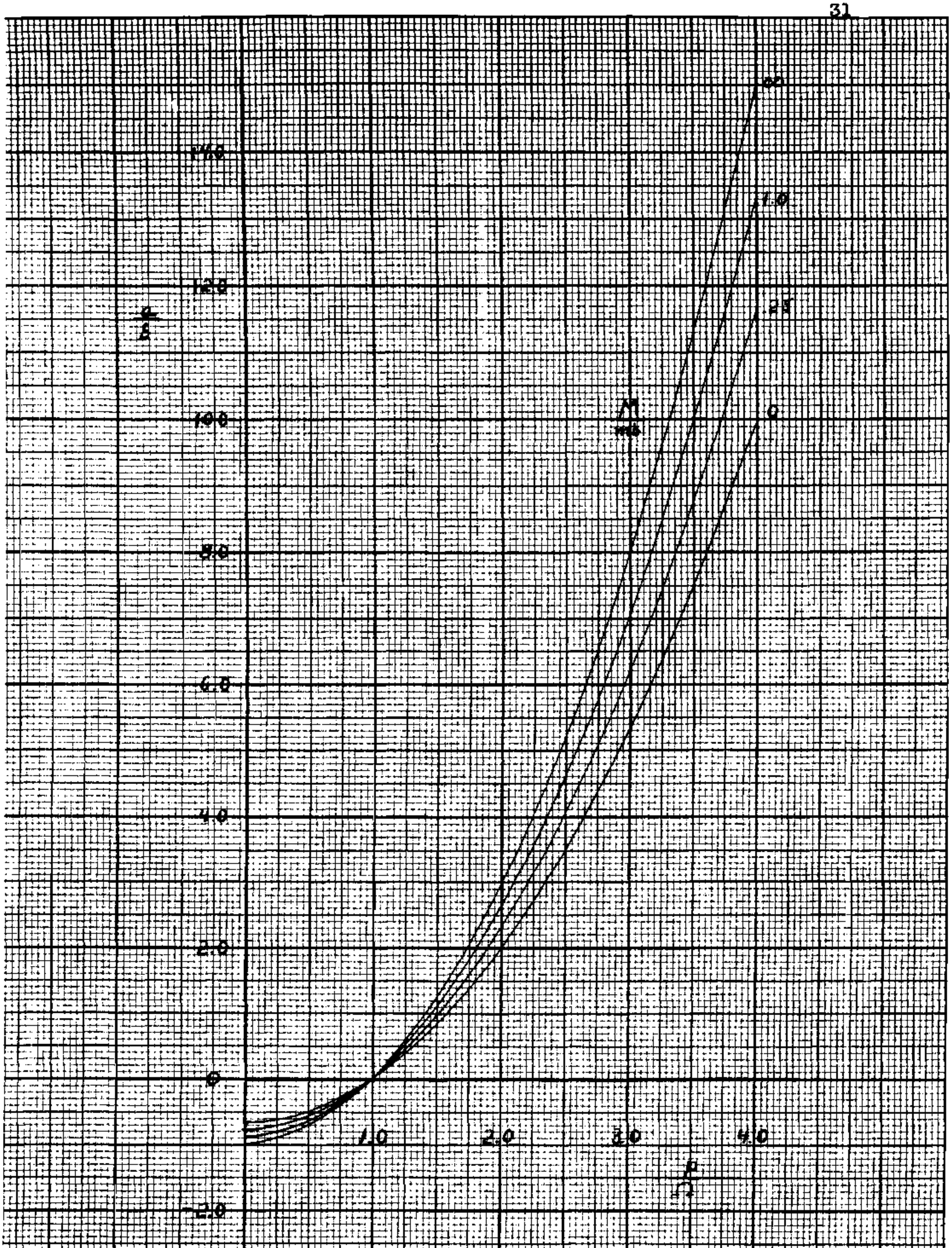

FIGURE 3. CHART FOR TUNING THRUST ABSORBER.

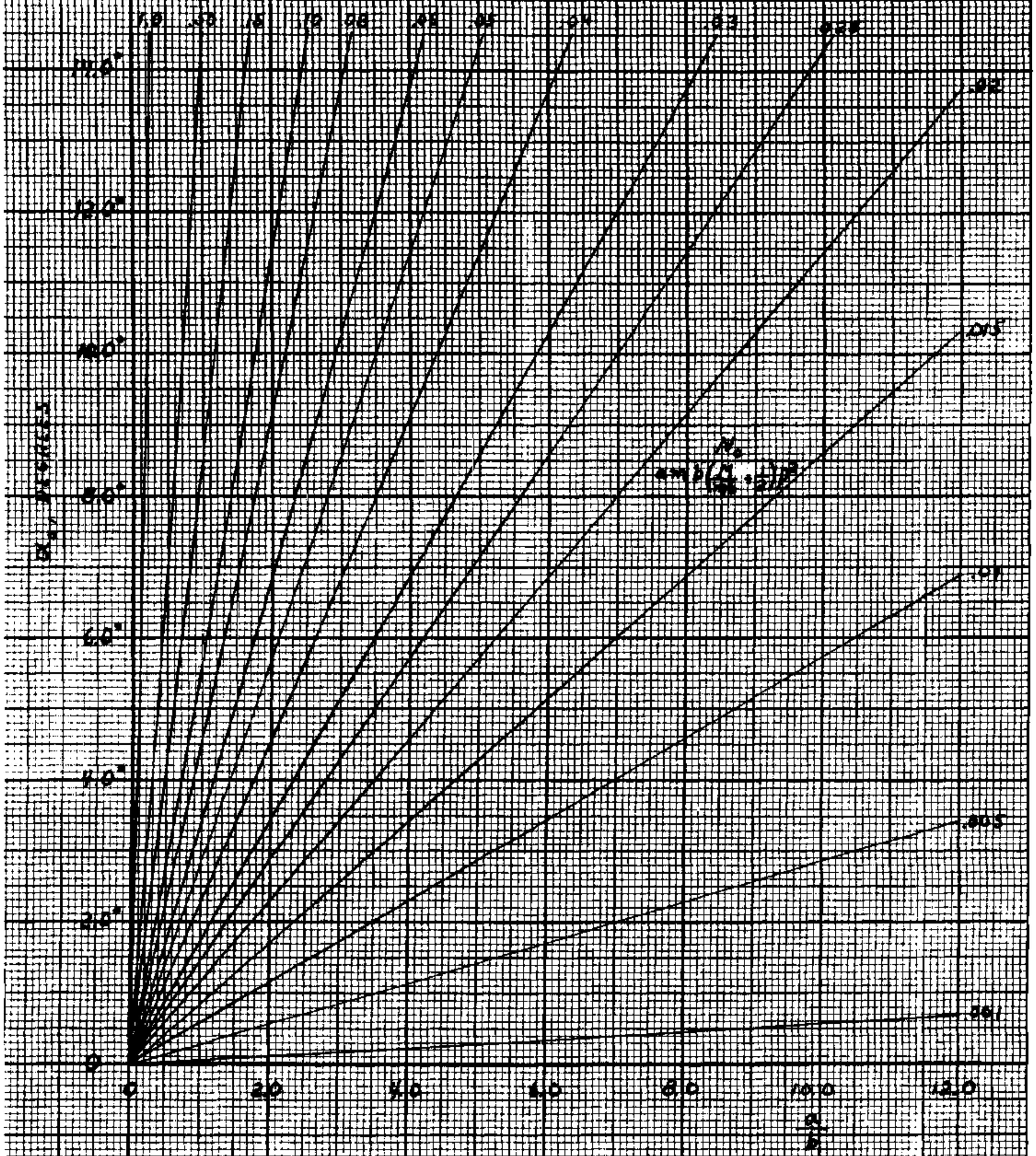

FIGURE 4. CHART FOR DETERMINATION OF AMPLITUDE OF THRUST ABSORBER.

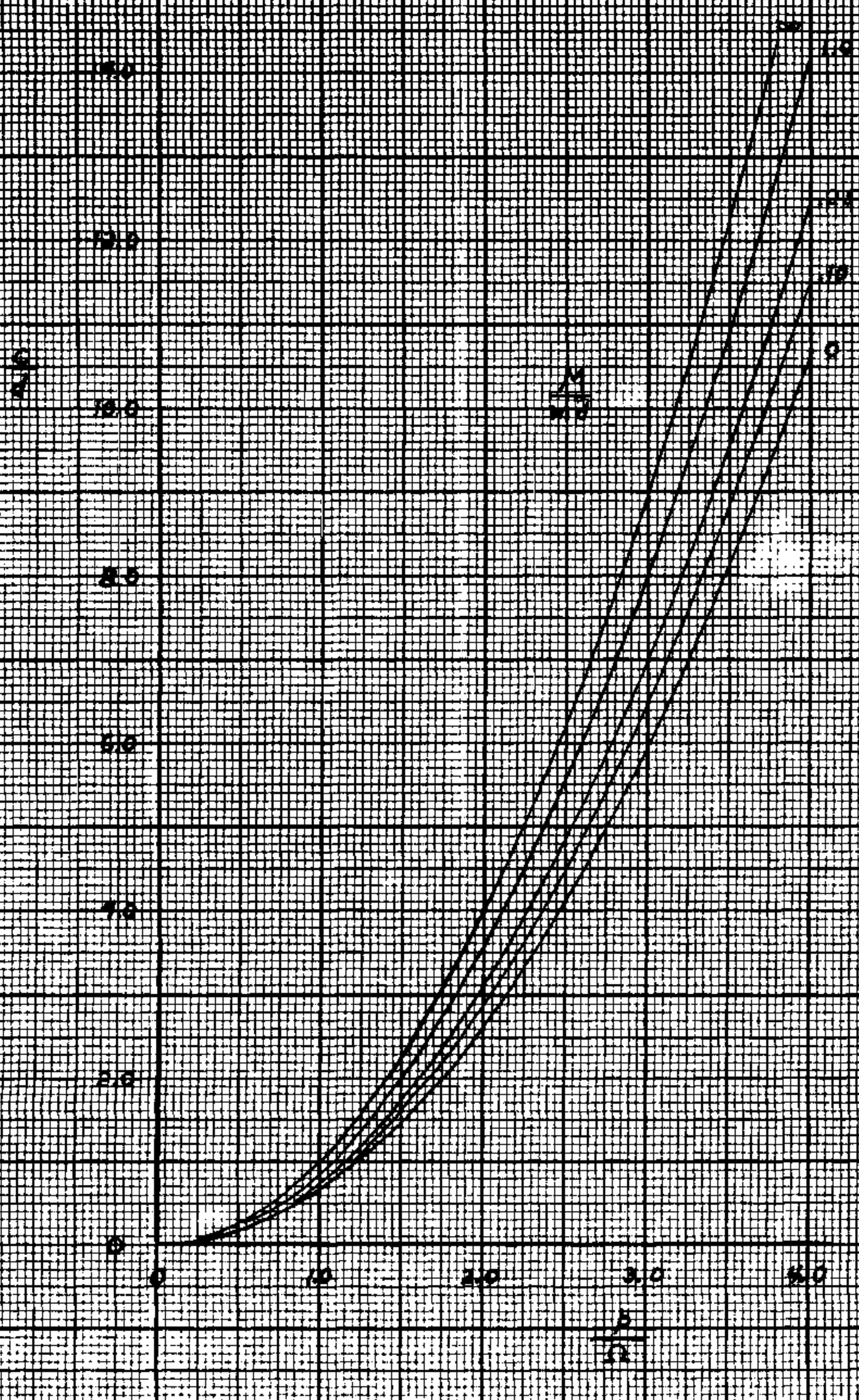

FIGURE 5. CHART FOR TUNING TORQUE ABSORBER

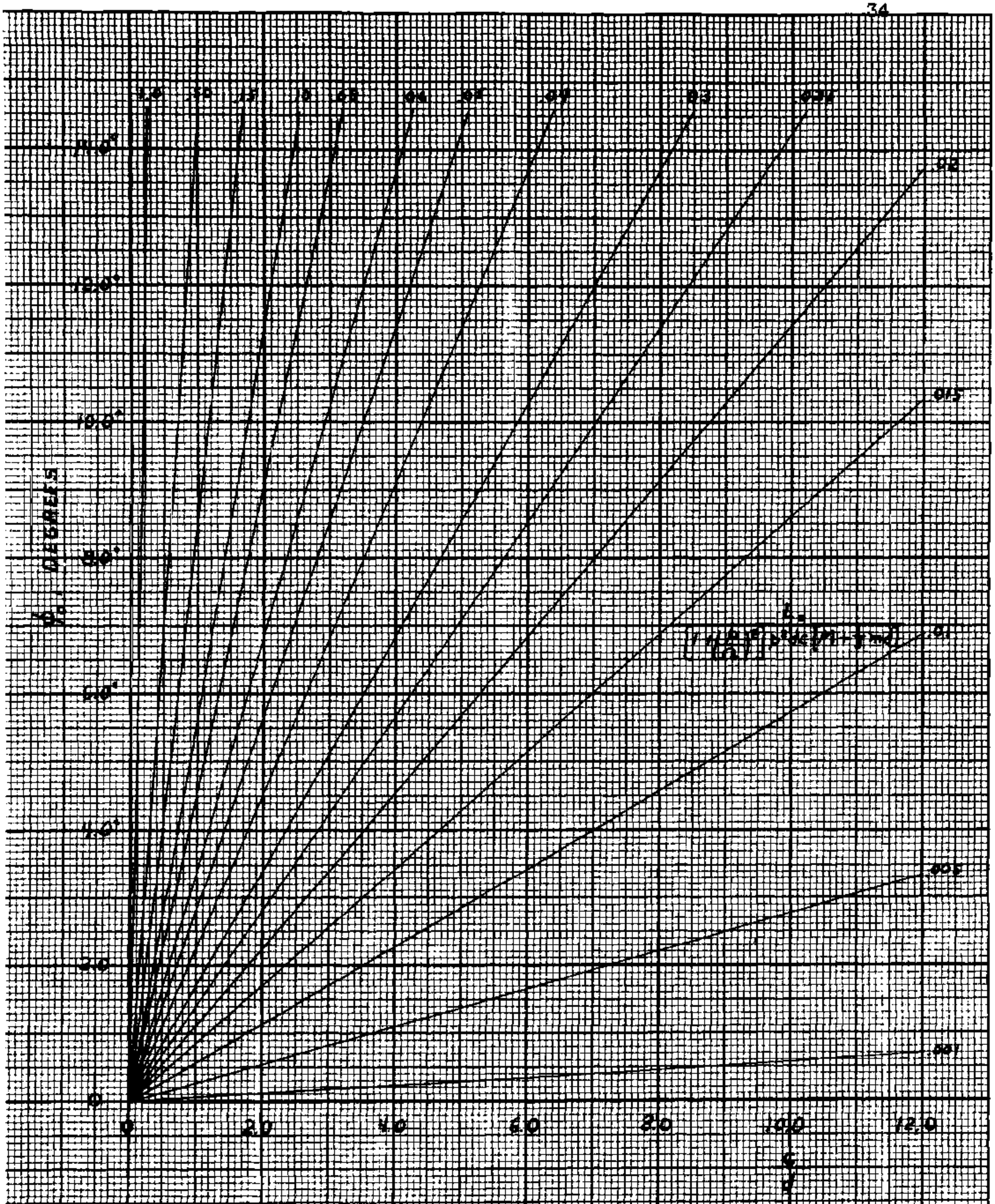

FIGURE 6. CHART FOR DETERMINATION OF AMPLITUDE OF TORQUE ABSORBER