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LOFT DTT RAKE ASSEMBLY DESIGN FLUID LOADS

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LOFT DTT RAKE ASSEMBLY DESIGN FLUID LOADS

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IDAHO NATIONAL ENGINEERING LABORATORY
LOFT TECHNICAL REPORT
LOFT PROGRAM

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SUMMARY

The maximum fluid forces acting on the LOFT DTT Rake Assembly during steady state and LOCE modes of operation, were estimated and presented for the 5-inch equivalent break flow area.

A total of six loading mechanisms for each of the two modes of operation were identified. The most significant mechanisms were determined and the resulting loads and frequencies were estimated.

The significant loading mechanisms were the Karman vortex-induced alternating lift forces, the steady lift forces due to pipe elbow secondary flows, and the steady drag forces.

The fluid forces calculated herein should be used in further analyses, as specified by the task manager, to determine the suitability of the LOFT DTT rake assembly design.

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1.0 INTRODUCTION

The five LOFT DTT Rake Assemblies must be designed to withstand the most severe expected fluid-induced forces that will be encountered in service. Combined stresses on the structure must not exceed allowable levels, and for cyclic loading the structure must have the capability of withstanding a sufficient number of loading cycles at maximum expected stress levels in order to avoid long-term fatigue failure. Six possible fluid loading mechanisms were identified for each of the two flow conditions.

1.1 Description of System

A cross section of the 14-inch Schedule 160 LOFT piping with the DTT Rake Assembly in place, is shown in Figure 1⁽¹⁾. The DTT Rake Assembly that was analyzed consists of two mounting sleeves, two blades (made up of two parts each, held together at the bottom with machine screws and welded to the mounting sleeves) as shown at the top of Figure 1, and three DTT sub-assemblies mounted on the blades with machine screws. The locations of the five DTT Rake Assemblies in the LOFT Primary Coolant System, are given in Figure 2⁽²⁾. Two DTT Rake Assemblies are located in the Blowdown Loop, at locations FE-BL-1 (cold leg vessel nozzle) and FE-BL-2 (hot leg vessel nozzle), with the blades oriented vertically. The other three DTT Rake Assemblies are mounted in the Intact Loop, at locations FE-PC-1 (cold leg), FE-PC-2 (hot leg), and FE-PC-3 (SG-pump crossover leg).

1.2 Purpose of the Analysis

The analysis must define the most severe expected loading conditions that the DTT Rake Assemblies will be subjected to in service, as well as their magnitudes, frequencies, and points of application.

on the structure. The results are used in a structural model to determine the adequacy of the DTT Rake Assemblies in withstanding the expected vibration environment.

1.3 Method of Analysis

The six possible fluid loading mechanisms, as well as the two different LOFT flow conditions under which they may occur, are summarized in the matrix of Table I.

TABLE I
LOFT DTT ASSEMBLY FLUID LOADING MECHANISMS

Fluid-Induced Loading Mechanisms	Flow Condition	
	A. Steady-State Operating Conditions	B. LOCE
I. Flutter due to Karman Vortex Shedding	Lift and Drag-Fatigue (DTT's in Operating Loop)	Lift and Drag - Flutter Divergence (DTT in Blow-down Loop)
II. Secondary Flows from Pipe Elbows	Lift and Drag - Fatigue (DTT's in Operating Loop)	Lift and Drag
III. Turbulent Buffeting	Lift and Drag	Lift and Drag
IV. "Gallop" Mechanism - Additional Lift Due to Blade Twisting	Lift and Drag	Lift and Drag
V. LOCE Wave Load	N/A	Drag
VI. Acoustic Resonance Transverse to Pipe Centerline	Lift	Lift

The non-negligible combinations of fluid-induced loading mechanisms and flow conditions given in Table I were investigated to find the most severe combinations. The worst-case load combinations resulted from the (I.B) and (II.B) mechanisms, LOCE flutter divergence and LOCE secondary flows from pipe elbows which were assumed to act simultaneously on the same DTT Rake Assembly. The assumption also was made that the cyclic combination of loads (I.B) and (II.B) exists for an infinite period of time and that the DTT Rake Assembly must withstand it for an infinite number of cycles in order to meet the fatigue limitation. This is a very conservative requirement since the maximum loads will exist only for a short time during each LOCE. The detailed description of each loading mechanism follows:

1.3.1 Flutter Divergence During LOCE (Mechanism (I.B) in Table I)

A. Comparison of Vortex Shedding Frequencies with Natural Frequencies

The maximum lift forces due to flutter, as well as the highest expected vortex shedding frequencies, will occur at the maximum value of momentum flux (ρV^2) that occurs during a LOCE. The maximum momentum flux is:

$$(\rho V^2)_{\max} = 188,000 \frac{\text{lbm}}{\text{ft}^2 \text{sec}^2} \quad (3)$$

The corresponding velocity in the LOFT piping, is

$$V_{\max} = 62.7 \text{ ft/sec} \quad (3)$$

The blade thickness Reynolds Number is

$$Re_b = \frac{\rho V t}{\mu} = \frac{(47.8)(62.7)\left(\frac{0.375}{12}\right)}{(7.3 \times 10^{-5})} = 1.3 \times 10^6$$

The Strouhal Number range for the blade, ranges from $0.33 < St < 0.45^{(4)}$ at this Reynolds Number.

The resulting blade-only vortex shedding frequency is then

$$\omega_{s_b} = \frac{VSt}{t}, \text{ or } 660 < \omega_{s_b} < 740 \text{ Hz.}$$

The DTT cylinder Reynolds Number is

$$Re_c = \frac{\rho Vd}{\mu} = \frac{1.5}{.375} (1.3 \times 10^6) = 5.2 \times 10^6,$$

and the Strouhal Number is $St_c = 0.27$.

The resulting cylinder-only vortex shedding frequency is

$$\omega_{s_c} = \frac{(1.27)(62.7)(12)}{(1.5)} = 135 \text{ Hz.}$$

For the assembly as a whole, the Strouhal Number is not known, although Hoerner⁽⁸⁾ reports a dependence of the Strouhal Number on overall drag coefficient as (Figure 3)

$$St = \frac{0.21}{C_D^{3/4}}$$

The drag coefficient of the structure is estimated to be on the order of $C_D = 1.5$. The Strouhal estimate is then

$$St = \frac{0.21}{(1.5)^{3/4}} = 0.155,$$

with a shedding frequency of

$$\omega_s = \frac{(0.155)(62.7)(12)}{(2.375)} = 49 \text{ Hz.}$$

Whether the blades and cylinders shed vortices separately at their particular frequencies or whether the structure sheds a vortex as a whole, is not known. The DTT Rake Assembly therefore is assumed to shed vortices over the range

$$50 < \omega_s < 740 \text{ Hz};$$

however, the actual vortex shedding frequency is not known because of the complicated geometry and scatter in the Strouhal data.

Vortex shedding has been observed⁽⁶⁾ at structural resonant frequencies and has been denoted as "lock-in". The lock-in phenomenon reportedly⁽⁶⁾ occurs when the vortex shedding frequency is within 40% of the resonant frequency. During lock-in the structure then vibrates at its natural frequency, independently of the vortex-shedding frequency.

The uncertainty of the actual vortex shedding frequency coupled with the lock-in phenomenon strongly suggests that resonant excitation be assumed so that

$$\omega_s = \omega_n = 313 \text{ Hz}$$

and

$$L = L_o \sin (2\pi(313 t)).$$

B. Fluid-Induced Lift Force Due to the Vortex Shedding Process

The lift force due to the vortex-shedding process can be expressed by⁽⁶⁾

$$L_o = C_e A_L \frac{\rho V^2}{2gc},$$

where L_o is the peak lift force, C_e is the peak lift coefficient, and A_L is the projected plan form area over which the force acts.

The RMS value $\bar{C}_e$, reported by Gerlach⁽⁶⁾ and Chen⁽⁷⁾, is about 0.15 at the high Reynolds Numbers of interest. (This contrasts with values predicted by theory of about 0.3 to 0.6. Apparently the discrepancy is due to the effect of "correlation length", or the lack of spanwise coherence of the vortex sheet. (Figure 5 and 6)). The resulting peak value, C_e , is then

$$C_e = \sqrt{2} \bar{C}_e = 0.21.$$

The effective projected lift area (plan form) for the two DTT blades, is estimated by assuming that the separation line is located at the after shoulder of each blade. The unsymmetric lift force is due to the unsymmetric vortex-shedding process; therefore, the effective projected area for lift, A_L , is only that part of the body downstream of the separation line. This area is

$$A_L = \frac{(2)(0.5)(9.7)}{144} = .067 \text{ ft}^2.$$

The peak lift force due to vortex-shedding, therefore, is

$$L_o = \frac{(0.21)(.067)(188,000)}{(2)(32.17)} = 41 \text{ lb}_f$$

or, $L = L_o \sin \omega t = 41 \sin (2\pi(313t))$.

Half this lift force is applied at the trailing edge of each blade, in a direction 90 degrees to the pipe centerline.

C. Drag Forces

The vortex-shedding process results in an alternating component of drag also, which operates at twice the shedding frequency; however, the drag force component is an order of magnitude less than the lift force and is then ignored.

The drag coefficients of the DTT Rake Assembly are estimated from Hoerner⁽⁸⁾ as 0.5 for the blades and 0.8 for the DTT's. The interference effect is estimated to increase the total drag by 30%⁽⁸⁾. This increases the blade C_D value to $(1.3)(.5) = 0.65$. The projected blade area is $S_b = (10.5)(0.375)(2) = 7.88 \text{ in}^2 = .0547 \text{ ft}^2$, and the total projected area of blades and DTT's is 14% of the pipe flow area⁽⁹⁾, or $S_T = (0.14)(0.6827) = .0956 \text{ ft}^2$.

The total drag is then:

$$D = \left[C_{D_b} \left(\frac{S_b}{S_T} \right) + C_{D_{DTT}} \left(1 - \frac{S_b}{S_T} \right) \right] (S_T) \left(\frac{\rho V^2}{2gc} \right)$$

$$D = \frac{[(0.65)(0.57) + (0.8)(0.43)] (.0956)(188,000)}{(2)(32.17)}$$

or, $D = 200 \text{ lb}_f$.

The drag forces are apportioned according to their respective projected areas on the DTT Rake Assembly, in a direction parallel to the pipe centerline. This results in a drag force of 52 lb_f on each blade and 32 lb_f on each DTT.

1.3.2 LOCE Secondary Flows from Elbows (Mechanisms (II.A) and (II.B) in Table I)

Secondary flow in a pipe elbow is caused by interaction of the turbulent velocity profile across the pipe with the centrifugal force caused by the change in flow direction. The higher-velocity fluid at the centerline undergoes greater acceleration and tends to move to the outside of the turn. This results in two secondary vortices, as shown in the sketch:

This results in a steady angle of attack and a spanwise lift force distribution on each blade, as shown in the sketch. The flat areas of each lift profile are caused by stalling of the blade, i.e., angle of attack exceeds the stall value and flow separates from the blade. A conservative estimate was made for the lift distribution due to secondary flow, by assuming that the entire work dissipated in the elbow turning loss is converted to rotational fluid kinetic energy.

For the 45-degree elbows upstream of the DTT's, the radius-to-diameter ratio is $(\frac{r}{D}) = 2.0^{(10)}$.

The Reynolds Number based on pipe inside diameter is

$$Re_D = \frac{\rho DV}{\mu} = \frac{(47.8)(11.188/12)(62.7)}{(7.3 \times 10^{-5})}$$

$$Re_D = 38.3 \times 10^6 .$$

For a 90-degree elbow, $K_{T90} = .085^{(11)}$.

For a 45-degree elbow, $\frac{K}{K_{T90}} = 0.63^{(11)}$,

$$\text{or, } K_{450} = (.63)(.085) = .054 .$$

The fluid between the elbow and the DTT is assumed to rotate as a solid body, or $\ell = 3 \text{ ft}^{(12)}$, where ℓ is the length of the rotating fluid body.

$$I = \int_0^R r^2 dm = \int_0^R r^2 (\rho \ell) 2\pi r dr = \frac{\pi}{2} \rho \ell R^4 .$$

$$I = \left(\frac{\pi}{2}\right) (3 \text{ ft}) \left(47.8 \frac{\text{lbm}}{\text{ft}^3}\right) \left(\frac{11.188}{4 \times 12}\right)^4 \text{ ft}^4$$

$$I = 0.67 \text{ lbm} \cdot \text{ft}^2 \text{ (each vortex) .}$$

The kinetic energy in each vortex is

$$KE = \frac{I \omega^2}{2g_c} ,$$

and also is equal to the turning loss, or

$$KE = K_{450} \frac{\rho V^2}{2g_c}$$

$$KE = \frac{(.054)(188,000)}{(2)(32.17)} = 158 \text{ ft} \cdot \text{lb}_f .$$

$$\frac{I \omega^2}{2g_c} = \frac{158}{2} \text{ ft} \cdot \text{lb}_f \text{ for each vortex.}$$

Then the rotational frequency ω is

$$\omega = \sqrt{\frac{(158)(2g_c)}{(2)(I)}} = \sqrt{\frac{(158)(32.17)}{(0.67)}}$$

$$\omega = 87 \text{ radians/sec} = 13.9 \text{ revolution/sec} = 832 \text{ rpm.}$$

The maximum tangential velocity is

$$U_{\max} = R\omega = \left(\frac{11.188}{4}\right)\left(\frac{87}{12}\right) = 20 \text{ ft/sec.}$$

The maximum angle of attack then is

$$\alpha_{\max} = \tan^{-1} \left(\frac{20}{62.7} \right) = 17.7^\circ$$

The blades will stall before reaching $\alpha_{\max}$, since they are not designed as airfoils. Hoerner⁽¹³⁾ shows a maximum lift coefficient for this type of profile of approximately

$$C_{L_{\max}} = \sqrt{0.5} = 0.7$$

for a biconvex foil section with zero camber.

The inverse lift-curve slope at small angles of attack is given in terms of aspect ratio (AR), as

$$\frac{d\alpha}{dC_L} = \frac{1}{2\pi} + \frac{1}{\pi AR},$$

where AR is related to the span (b) and plan form area (s) and chord (c) by

$$AR = \frac{b^2}{s} = \frac{b^2}{bc} = \frac{b}{c}$$

or, $AR = \frac{9.7}{3.3} = 3$. Due to the "end-plate" effects of the pipe walls, however, the effective AR is large.

$$\therefore \frac{d\alpha}{dC_L} = \frac{1}{2\pi} = 0.1592 \text{ radians}$$

or,

$$\frac{dC_L}{d\alpha} = \frac{1}{0.1592} = 6.28 \text{ per radian.}$$

In degrees,

$$\frac{dC_L}{d\alpha} = \frac{6.28 * 2\pi}{360} = 0.1 \text{ per degree.}$$

Since $C_{L_{\max}}$ is only 0.7, this predicts a stall angle of 7° .

In reality, the lift-curve slope decreases as the stall angle is approached⁽¹³⁾; a more realistic stall angle is 10 to 15 degrees.

The secondary-flow lift calculations are given below, where y is the distance from the pipe wall along the span of the two blades, r is the distance from the center of each vortex, u is the transverse velocity component, α is angle of attack, C_L is lift coefficient, and ΔL is incremental lift for that section of span for both blades.

TABLE II

LIFT DUE TO SECONDARY FLOWS IN ELBOWS (LOCE)

<u>y(in.)</u>	<u>($\frac{y}{\ell}$)</u>	<u>r(in.)</u>	<u>u($\frac{\text{ft}}{\text{sec}}$)</u>	<u>α</u>	<u>C_L</u>	<u>$\Delta L (1b_f)$</u>
0.35	.0313	2.45	17.8	16°	0.7	66
1.049	.0938	1.75	12.7	11°	0.7	66
1.748	.1562	1.05	7.6	7°	0.7	66
2.447	.2187	0.35	2.5	2°	0.2	20
3.1466	.2812	0.35	2.5	2°	0.2	20
3.846	.3438	1.05	7.6	7°	0.7	66
4.545	.4062	1.75	12.7	11°	0.7	66
5.244	.4687	2.45	17.8	16°	0.7	66
5.943	.5312	2.45	17.8	16°	0.7	66
6.643	.5938	1.75	12.7	11°	0.7	66
7.342	.6562	1.05	7.6	7°	0.7	66
8.041	.7187	0.35	2.5	2°	0.2	20
8.740	.7812	0.35	2.5	2°	0.2	20
9.440	.8438	1.05	7.6	7°	0.7	66
10.14	.9063	1.75	12.7	11°	0.7	66
10.84	.9689	2.45	17.8	16°	0.7	66

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1.3.3 Turbulent Buffeting During LOCE (Mechanism (III.B) in Table I)

Turbulence intensity is expected to be significant in LOFT, due to geometric effects such as bends, area changes, wall roughness, sharp-edged vessel internals, and pump impellers, as well as subcooled LOCE unsteady-flow effects on changing fluid boundary layers.

Turbulence intensity, T , defined in terms of RMS velocity fluctuations, $\overline{u'^2}$, and time-averaged velocity, u , by

$$T = \frac{\sqrt{\overline{u'^2}}}{u}$$

is expected to exceed 10 percent. Measurements of this parameter in LOFT have not been made; however the practice of filtering the LOFT test data suggests the turbulence "noise" is significant.

A few measurements taken in a wind tunnel by Reichardt and reported by Schlichting⁽¹⁴⁾, indicate T values from 13 percent near the test section wall to 4 percent at the centerline, with an average of 8 percent over the cross section. These data were taken in a wind tunnel which supposedly was designed to reduce turbulence effects. Therefore LOFT is expected to have considerably greater turbulence levels.

A few test data points taken by Messrs. Tai, Crowe, and Roberson⁽¹⁵⁾ for an H-section beam having the flanges oriented parallel to the flow (similar to the DTT Rake Assembly but with a solid web), indicate the following behavior:

- (a) the transient behavior of the system is sensitive to turbulence intensity T ;
- (b) at $T = 1$ percent, organized vortex shedding predominates;

(c) at $T = 5$ percent the dominant mechanism is a combination of vortex shedding and "gallop" (the increase of lift due to increased angle of attack as an airfoil twists about its neutral axis);

(d) at $T \geq 10$ percent turbulent buffeting interferes with the vortex-shedding and gallop mechanisms. (The results of these tests must be interpreted with caution because of the presence of the solid web in the H-beam configuration.)

Turbulent-buffeting loads are not added to the vortex-induced lift loads on the DTT Rake Assembly, since turbulence will interfere with and oppose the organized formation of vortices. Transient forces due to turbulent buffeting are expected to increase the drag loads somewhat; however, this mechanism was ignored for the following reasons.

(a) accurate information is unavailable on turbulence intensity values in LOFT,

(b) drag load increase is expected to be the main effect of this mechanism, but the DTT Assembly is strongest in that direction, and

(c) the turbulent buffeting mechanism interferes with the organized vortex shedding and gallop mechanisms, thereby acting to reduce net time-averaged lift forces on the structure.

1.3.4 "Gallop" Mechanism: Additional Lift and Drag Due to Blade Twisting During LOCE (Mechanism (IV.B) in Table I)

The "gallop" mechanism depends on the net effect of other forces in twisting the blade, and on transverse turbulent velocity fluctuations, so that an effective angle of attack results. Applying the other loads (Mechanisms (I.B) and III.B)) to the DTT Rake Assembly, resulted in a very small twist angle, so that the resulting lift and drag forces due to "gallop" were

negligible; i.e., the DTT Rake Assembly is relatively stiff in torsion, so that this mechanism has negligible effect.

1.3.5 LOCE Wave Load (Mechanism (V.B.) in Table I)

Unsteady decompression waves are generated initially by opening the LOFT QOBV's during LOCE. Assuming a minimum QOBV opening time of 8 msec (milliseconds)⁽¹²⁾, the wave strength at the break is

$$F_1 = \Delta P = 2(P_o - p_{sat_{CL}}) ,$$

as reflected from an obstacle, or

$$F_1 = \Delta P = 2(2250 - 1000) = 2500 \text{ psid} .$$

$$\text{Valve Break Area} = .09 \text{ ft}^2 \quad (12)$$

Pipe Flow Area at DTT ,

$$A_2 = A_f = (.6827)(1.0 - 0.14)^{(9)} = 0.59 \text{ ft}^2 .$$

$$F_2 = s F_1$$

$$s = \frac{2A_1/c_1}{\frac{A_1}{c_1} + \frac{A_2}{c_2}} \quad (16)$$

$$c_1 = c_2$$

$$s = \frac{2A_1}{A_1 + A_2} = \frac{2(.09)}{(.09) + 0.59} = 0.26$$

$$F_2 = (0.26)(2500) = 650 \text{ psid}.$$

The wave amplitude at the DTT is 650 psid; however, it acts as a ramp over a time period of at least 8 msec (actually it must be longer than this because decompression waves flatten with time). When the leading edge of the wave passes over the blade and is located at the trailing edge of the blade, then the leading edge of the blade sees a drag force that is greater by

the amount

$$\Delta P_w = \frac{(650)(3.3)}{(12)(3000)(8 \times 10^{-3})} = 7.4 \text{ psid.}$$

The net drag force due to the LOCE wave is

$$F_{wD} = A_p \Delta P_w = (7.4)(0.14)(0.6827)(144) = 110 \text{ lb}_f.$$

The net DTT Assembly impulse due to this force is

$$I_w = F_{wD} \Delta t = \frac{(110)(3.3)}{(12)(3000)} = .01 \text{ lb}_f \cdot \text{sec.}$$

The LOCE wave load is negligible in comparison to the vortex-induced forces.

1.3.5 Acoustic Resonance Transverse to Pipe Centerline (Mechanism (VI.B) in Table I)

If the vortex-shedding frequency (ω_s) is close to the transverse acoustic frequency for the inside diameter of the 14-inch Schedule 160 pipe where the DTT Rake Assemblies are mounted, then a coupled resonance can occur.

The transverse frequency (ω_x) is

$$\omega_x = \frac{nc}{2ID}, \quad n = 1, 2, 3, \dots$$

$$c = 3000 \text{ ft/sec}$$

$$ID = 11.188 \text{ in}/12 = 0.932 \text{ ft.}$$

$$\omega_x = \frac{(1)(3000)}{(2)(.932)} = 1610 \text{ cps.}$$

The higher modes are integral multiples of this value.

The transverse acoustic resonance frequency is more than an octave higher than the vortex-shedding range, therefore this mechanism is not a factor in the design of the DTT Rake Assembly.

1.3.6 Fatigue Flutter Loads (Steady State Operating Conditions) on Intact-Loop DTT Rake Assembly

A. The maximum steady state operating-loop flowrate is

$$\dot{m} = 3.9 \times 10^6 \text{ lbm/hr}^{(17)}$$

$$\rho = 47.8 \text{ lbm/ft}^3$$

$$V = \frac{(3.9 \times 10^6)}{(0.6827)(47.7)(3600)} = 33.2 \text{ ft/sec},$$

which results in the Reynolds Number based on thickness of the blade,

$$Re_x = \frac{\rho x V}{\mu} = \frac{(47.8) \left(\frac{0.375}{12} \right) (33.2)}{(7.3 \times 10^{-5})} = 0.7 \times 10^6$$

The Strouhal Number,

$$St = \frac{\omega_s X}{V} = 0.3$$

The Vortex Shedding Frequency,

$$\omega_s = \frac{St V}{x} = \frac{(0.3)(33.2)(12)}{(0.375)} = 320 \text{ Hz}$$

$$\rho V^2 = (47.8)(33.2)^2 = 52,700 \frac{\text{lbm}}{\text{ft sec}^2}$$

The Drag Force,

$$D = \frac{[(0.65)(0.57) + (0.8)(0.43)] (.0956)(52,700)}{(2)(32.17)} = 56 \text{ lb}_f,$$

and the Peak Lift Force,

$$L_o = \frac{(0.21)(2)(0.5)(9.7)(52,700)}{(144)(2)(32.17)} = 12 \text{ lb}_f.$$

$$L = L_o \sin \omega_s t = 12 \sin 2\pi(320t).$$

B. The minimum steady state operating flowrate is⁽¹⁷⁾

$$\dot{m} = 2 \times 10^6 \text{ lbm/hr}.$$

$$\rho = 47.8 \text{ lbm/ft}^3$$

$$V = \frac{2 \times 10^6}{(0.6827)(47.8)(3600)} = 17 \text{ ft/sec},$$

which results in:

the Reynolds Number based on blade thickness

$$Re_x = \frac{\rho X V}{\mu} = \frac{(47.8)(.375)(17)}{(12)(7.3 \times 10^{-5})} = 0.35 \times 10^6 ,$$

the Strouhal Number,

$$St = 0.22 ,$$

the Vortex Shedding Frequency,

$$\omega_s = \frac{St V}{x} = \frac{(0.22)(17)(12)}{(0.375)} = 120 \text{ Hz} .$$

$$\rho V^2 = (47.8)(17)^2 = 13,800 \text{ lbm/ft sec}^2$$

the Drag Force,

$$D = \frac{[(0.65)(0.57) + (0.8)(0.43)] (.0956)(13,800)}{(2)(32.17)} = 151 b_f$$

and the Peak lift force,

$$L_o = \frac{(0.21)(2)(0.5)(9.7)(13,800)}{(144)(2)(32.17)} = 31 b_f .$$

$$L = L_o \sin \omega_s t = 3 \sin 2\pi(120 t) .$$

1.3.7 Steady-State Operating Conditions (III.A), (IV.A), and VI.A)

All steady-state conditions except (I.A) and (II.A) were ignored, since the loads they induce are much less than the LOCE forces.

2.0 RESULTS

Results of the DTT Rake Assembly fluid loads analysis are presented below in Tables III and IV, which give the vortex-shedding frequencies, structure natural frequencies calculated by R. G. Rahl⁽⁵⁾ given for convenience, and lift drag forces.

TABLE III
KARMAN FLUID SHEDDING VORTEX FREQUENCIES (ω_s)
vs
DTT RAKE ASSEMBLY NATURAL FREQUENCIES** (ω_n)

<u>LOCE[@]</u> <u>ω_s</u>	<u>Steady-State[@]</u> <u>ω_s</u>	<u>+</u> <u>ω_x</u>	<u>ω_{n1}</u> <u>(LIFT)</u>	<u>ω_{n2}</u> <u>(DRAG)</u>	<u>ω_{n3}</u> <u>(TWIST)</u>
50 to 740	320 Hz	1610 Hz	313 Hz	626 Hz	1130 Hz

**Calculated by R. G. Rahl with SAP Code.⁽⁵⁾

@ Mid-range values at $\rho V^2 = 188,000$ lbm/ft-sec² (LOCE) and $\rho V^2 = 52,700$ lbm/ft-sec² (ST.ST.)

+ Transverse acoustic frequency

TABLE IV
RESULTS OF FLUID LOADS ANALYSIS

<u>Fluid Load Condition</u>	<u>Lift Force</u>	<u>Drag Force</u>
*I.B. - Karman Vortex (LOCE)	$(41 \text{ lb}_f) * \sin [2\pi(313t)]^{**}$	200 lb_f
*II.B. - Secondary Flows from Elbow (LOCE)	$218 - 218 \text{ lb}_f$ (zero net)	--
III.B. - Turbulent Buf- feting	Negligible - Opposes I.B. above	Negligible
IV.B. - "Gallop" Mechanism	Negligible	Negligible
V.B. - LOCE Wave Load	N/A	110 lb_f
VI.B. - Acoustic Resonance Transverse to Pipe Centerline	Negligible	N/A

*Loads from conditions I.B and II.B were applied to structure simultaneously.
LOCE loads were larger than steady state loads, and were used to determine
steady state fatigue requirements as well.

**Lift force was applied at structure first-mode natural frequency.

3.0 CONCLUSIONS

a) The Karman vortex-shedding mechanism results in fluid forcing frequencies which cover a wide range of values, due to the effects of the complicated geometry of the DTT Rake Assembly, turbulence, data scatter, and the lock-in phenomenon.

b) The alternating force components must be applied to the structure at the appropriate (first-mode) natural frequency.

c) The fluid lift and drag loads due to vortex-shedding, secondary flow from pipe elbows, and LOCE drag loads, are to be combined and applied simultaneously to the structure.

4.0 RECOMMENDATIONS

It is recommended that the fluid forces calculated in this analysis be used to determine the suitability of the LOFT DTT Assembly in withstanding the fluid forces expected during LOCE and steady-state operations with a 5-inch equivalent break.

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FIGURE 2
LOCATIONS OF LOFT DTT ASSEMBLIES

FIGURE 3
VORTEX - SHEDDING FREQUENCY
AS A FUNCTION OF GEOMETRY

FIGURE 4
 "LOCK-IN" EFFECT OF STRUCTURE MOTION
 ON VORTEX-SHEDDING FREQUENCY

Fig. 4(a) Fluctuating lift coefficient C_L plotted over Reynolds number R ; curves 1 and 2 from theory, the other curves and points from a number of experimental results [14-32]

(b) Ratio β between the minimum value of C_L , measured and the theoretical value of C_L (curve 1), as well as vortex correlation length versus Reynolds number R [33-35].

have been proved to be reasonably accurate by the experimental results in a previous investigation [13]. However, the direct measured values of C_L given by Kovasznay (1949) [26] and Philips (1956) [27] are higher than the theoretical curve (see Fig. 4(a)). How reliable these measurements are can only be examined by further research work.

The transcritical Reynolds number branch of the theoretical curve 1 lies much higher than the experimental results obtained by Schmidt (1965/1966) [23, 24] and Jones (1968) [25] (see Fig. 4(a)). The ratio of the measured value to the theoretical one is shown as curve 11' in Fig. 4(b). These ratios are as small as that in the critical Reynolds number range (see curve 11 at $R = 5 \times 10^5$). This reveals that the vortex correlation length in the transcritical Reynolds number range will be in the same order of magnitude as in the critical Reynolds number range (see curve 13).

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RMS LIFT COEFFICIENTS -- ROUND CYLINDERS

Fig. 6 Compilation of RMS lift coefficients (see Fig. 5 for legend)

vibration amplitude to the tube diameter x_o/D and the vibration frequency to the Strouhal frequency f/f_s ; thus,

$$C_L = C_L(R_e, \lambda/D, x_o/D, f/f_s) \quad (6)$$

The phase angle θ of the fluid force relative to the cylinder motion is also a function of these parameters, so

$$\theta = \theta(R_e, \lambda/D, x_o/D, f/f_s) \quad (7)$$

Figure 7 shows typical correlations of C_L and θ with f/f_s and x_o/D . It is clear from the drastic changes in C_L and θ for small changes in f/f_s and x_o/D that the dynamics of vortex shedding is quite sensitive to the tube motion.

The designer is still somewhat dubious about these plots since they were obtained under forced oscillation conditions, although he is interested in self-excited conditions. At any rate, he decides to make some stress calculations. He has already determined the modes of vibration of interest, for the purposes of discussion, consider that these modes are the two shown in Fig. 8. But now he discovers that no procedure is presently available to allow him to predict the non-uniform distribution of vortex force on a tube vibrating in an arbitrary bending mode; all of the compiled data are for a tube vibrating as a rigid body normal to its axis. However, the designer is willing to idealize his problem and assume a constant vortex force distribution over the tube, then, the problem actually is resolved into the investigation of the lumped mass model shown in Fig. 8c. The equations of motion for this model are

$$m\ddot{x} + b\dot{x} + kx = F_o \sin(2\pi ft + \theta) \quad (8)$$

Fig. 7 Example of available experimental force coefficient amplitude and phase angle data for oscillating circular cylinders--taken from Ref. (1)