DESIGN AGAINST FLOW INDUCED VIBRATION AND FRETTING WEAR IN FAST BREEDER REACTOR

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ABSTRACT

Most of the tubular structures in heat exchangers and steam generators are subjected to cross flow of fluid and hence they are checked for vortex shedding, fluid elastic instability and turbulence. In pool type fast breeder reactors a new type of fluid elastic instability of shell structures due to weir flow has been observed during commissioning of fast reactor Super Phenix. A similar type of instability has been checked for PFBR. The fretting wear of tubular structures at loose supports is very common phenomenon. The excitation forces of various mechanisms like vortex shedding, fluid elastic instability and turbulence are discussed.

INTRODUCTION

The majority of components in Fast Breeder Reactors are thin walled structures in order to increase in heat transfer and to reduce the thermal stress. Because of this, many components are subjected to Flow Induced Vibration (FIV). FIV can be viewed in two major divisions. One is cross flow and the other is parallel flow induced vibrations. Further there are two divisions as per the components are concerned viz. one is tube structures and the other is shell structure.

In cross flow induced vibration of tube structures, there are three mechanisms viz vibration due to vortex shedding, second is Fluid Elastic Instability and the third is turbulence buffeting. Out of these three mechanisms the vortex shedding and fluid elastic instability are more damaging. Fluid elastic instability can occur in array of tubes but not in single tube. As per ASME Appendix N there are criteria to avoid the vortex shedding and fluid elastic instability. The turbulent buffeting can not be avoided completely but can be reduced. There is a peculiar type of instability in the shell structures due to weir flow over thermal baffle which was observed in the case of SuperPhenix fast breeder reactor in France. In Proto type Fast Breeder Reactor(PFBR) which is in construction stage at Kalpakkam, India there is similar type of weir flow over thermal baffle. For this Weir flow induced Instability, Aita has derived analytically two criteria to avoid such instability. First Criteria is for low frequency sloshing type of instability which involve two coupled sloshing modes and the second criteria is for higher frequency of single mode of Fluid Structure Interaction(FSI).

FIV on its own causing failure and also it causes failure due to fretting wear in the tube structures. Fretting occurs at the loose supports of the tubular structures. The fretting wear rate can be minimized by reducing the work rate. The work rate can be reduced by way of reducing the clearance between the tube and the support and/or by suitable material for the support..

CROSS FLOW INDUCED VIBRATION OF TUBULAR STRUCTURES [1]

Vortex Shedding:

For the tubes under the cross flow when the fluid reaches a tube, its kinetic energy is converted in to pressure energy and this added with the fluid pressure, gives the maximum pressure at the leading edge. Then fluid flows around the tube in a boundary layer flow. At the widest section the boundary layer splits into two free –shear layers which form the wake at the rear of the tube. The innermost part of the shear layer moves at a slower velocity than the outer most part resulting in the layers converting in to the discrete vortices. This is called vortex shedding This occur alternately from each side of the tube. Vortices exert force, which changes in direction, on the tube. These oscillating forces vibrate the bundle.

To avoid lock-in synchronization, the values of reduced damping Cn and reduced velocity (V/f_nD) are calculated as per the equations 1 and 2 given below . And Lock-in synchronization is avoided if any one of the following conditions is satisfied

$V/f_lD < 1$ OR $C_n > 64$ OR $V/f_nD < 3.3$ and $C_n > 1.2$

Reduced damping $C_n = 4\pi\xi M_n / [\rho\, D^2\ \{\int_{Le}\varphi_n^2(x)\, dx\}]$ (1)

Where M_n = effective modal mass/length for n^{th} vibration mode

$= \int_0^L m_t\varphi_n^2(x)dx$ (2)

Reduced Velocity V/f_nD

f_n = Natural frequency of n^{th} vibration mode
ρ = fluid density
D = tube diameter

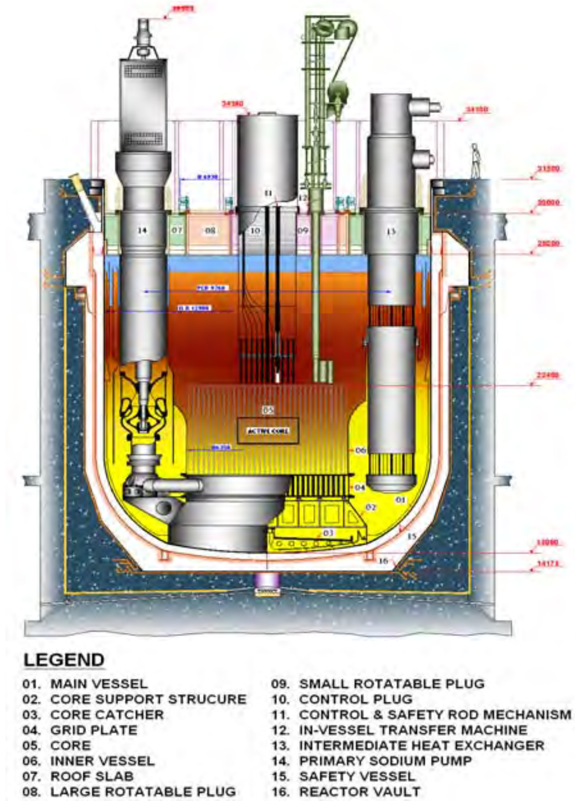

Fig. 1 Reactor Assembly of PFBR

$\varphi_n(x)$ = n^{th} vibration mode shape
m_t = Total mass per unit length of the tube
ξ = Damping

Fluid elastic instability

When the fluid in cross flow over an array of tubes displaces some of them , the force field exerted by the fluid over the tube bundle also changes. The tube experiences lift and drag forces and this can result in the vibration of the tubes. If the cross flow velocity is above the critical velocity V_{crit}., i.e. the energy fed to the tubes exceeds that which can be dissipated by damping. This type of vibratory motion of the tube is called Fluid elastic instability.

The critical velocity and the modal weighted velocity are calculated as per equation 3 & 4 as given below.

Critical velocity $V_c/F_nD = C\,(m_t/\rho D^2)^a(2\pi\xi_n)^b$ (3)

Modal weighted velocity $V_e^2 = [\int_0^{Le} V^2(x)\,\varphi_n^2(x) * dx\,] / [\int_0^{L} \varphi_n^2(x) * dx]$ (4)

Tubes are stable if $Ve < Vc$ for all modes.

Turbulence

A fluid in turbulent flow has random fluctuating velocity components in all directions with a broad range of frequencies. These turbulent velocity components impart energy to the tubes over which the fluid is flowing in cross flow. This results in random tube vibrations and is called turbulent buffeting . This can cause fretting of the tubes and fatigue them to failure over a period of time. The tube extracts maximum energy from that component of turbulence whose frequency is closer to the fundamental frequency of the tubes. This can set in resonance and lead to the failure.

Mean square response to the homogenous turbulence excitation was determined considering the cross flow velocity distribution over the entire portion, as per the equation 5.

$$y^2(x) = \sum_j \sum_i \frac{L_i G_f^{\,i}(f_j)\phi_j^{\,2}(x)}{64\pi^3 M_j^{\,2} f_j^{\,3}\xi_j}\left(j_{jj}^{\,j}\right)^2 \qquad (5)$$

here $G^i_f(f_j) = G_f(f_j) \int_0^{Li} \phi^2_j(x)\, dx$ and

$G_f(f_j)$ = Mode shape weighted single sided power spectral density.

$= (D/2)^2 C^2_R(f_j) \int_0^L [\rho U^2(x)]^2 \phi^2_j(x)\, dx$

All above equations and conditions can be referred in [1].

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FLUID ELASTIC INSTABILITY DUE TO WEIR FLOW [2]

A new type of fluidelastic instability was observed during commissioning of Superphenix LMFBR. This instability is due to fluid discharge over a flexible weir flow shell which separates two thin fluid sheets(feeding and restitution collectors).

An analytical nonlinear model was realized [2]. The flow and impact force sources at the free surface of collectors were described and projected on the modal basis of the system formed by the feeding and restitution collectors and the weir shell. Simplified criteria were extracted based on simplified assumptions.

Weir flow induced instability in PFBR

In Prototype Fast Breeder Reactor (PFBR) the main vessel (MV) is being cooled by the cold sodium sent through a passage between the MV and the thermal baffle provided on the inner side of MV. This sodium after cooling the MV is flown over weir shell and it is falling through a height H before it is hitting the free surface of sodium on the other side. This fall of sodium becomes a source of energy which makes this baffle unstable under certain fall height and flow rate. This particular type of instability has been noticed in SPX1 during its commissioning stage. This problem has been solved by conducting several mock-ups and parallely some theoretical studies. In Indira Gandhi Centre for Atomic Research also this phenomenon has been studied for PFBR and found two types of mechanisms involved. Type-1 mechanism is of sloshing type of instability in which there are two closely located sloshing modes of low frequencies which are coupled. Type 2 mechanism is of fluid structure interaction (FSI) type in which single mode of high frequency is involved.

From each of above mechanisms, it is possible to set a stability chart in the domain of fall height and flow rate. From these stability chart it is able to solve this problem by selecting appropriate fall height and flow rate which lies in the stable zone.

Flow over weir shell

The flow over a weir shell is defined as :

$$qd = K\sqrt{g}h_1^{3/2} \qquad h1>0 \qquad (6)$$

$$= \quad 0 \qquad h_1 \le 0 \qquad (7)$$

Where K = weir constant,

h_1 = over flow thickness,

q_d = over flow rate over weir shell

Flow from feeding collector

The plenum between MV and outer thermal baffle is called feeding collector.The flow from the feeding collector is defined as :

$$\Delta q_d = q_d(t,\theta) = -K\sqrt{g}h_1^{3/2}(t,\theta) \qquad (8)$$

Where (-) = sign indicates that the flow is going out,

h_1 = is the over flow height,

t = time,

θ = circumferential angular position.

Force on the restitution collector

The plenum between outer baffle and the inner plenum is called restitution collector. The falling sodium on the free surface of the restitution collector impacts some force and it is represented by :

$$F(t,\theta) = -\rho_f \; q_d(t-\tau,\theta) V_f \tag{9}$$

where ρ_f = density of fluid
τ = delay time,
V_f = impact velocity of fluid.

Friction of the falling fluid film

The impact velocity (V_f) and the delay time (τ) due to fall of the fluid are computed by taking into account of the friction in the wall of the baffle. The friction fall of the sodium is considered as a open channel. Then it can be correlated using a Moody type diagram:

$$\frac{1}{\sqrt{f}} = -2.0 \; Log_{10} \frac{\eta / D_h}{3.7} + \frac{2.51}{\mathrm{Re}_{D_h} \sqrt{f}} \tag{10}$$

Aita's simplified criteria

For this problem AITA and al. has arrived at 2 simplified criteria one for couple sloshing modes and the other for fluid structure interaction mode.

For sloshing modes

$$Sin\beta - U \cos \beta < \frac{\varepsilon}{\Delta\Omega_\circ} \tag{11}$$

For FSI mode:

$$Cos \; \beta \; \Omega_{nm} + U \; \Omega_{nm} \; Sin \; \beta \; \Omega_{nm} < \frac{Z_{nm}^{(1)}}{Z_{nm}^{(2)}} \tag{12}$$

Where

$$\Omega = \frac{\omega}{\omega_0} \quad , \beta = \omega 0 \tau ,$$

$$\lambda = \frac{3}{2} \frac{U^{1/3}}{M_{nm}} \left(\frac{e}{L} \right)^{-2/3}$$

$$U = \frac{q_0}{eV_0}, \quad V_0 = \sqrt{gL}, \; q_0 = g^{1/2} \left(Z_0^{(1)} \right)^{3/2}, \; e = e_1 = e_2, \; M_{nm} = \frac{M_{nm}}{M_0}$$

$$M_0 = \Pi \rho L \,\mathrm{Re}, \quad L = \frac{R}{n},$$ n = Circumferential wave number

FRETTING WEAR OF TUBULAR STRUCTURES

The vortex shedding, turbulence and fluid elastic instability are those among the few important mechanisms of flow induced vibrations of tubular structures, which can produce excessive damage at the support locations through fretting wear and fatigue. So it becomes very much necessary to have some predictive tools to estimate those damages. For this purpose, here is a code called "TURB-GEN" developed under the framework of CEA-IGCAR CASTEM collaboration.

The code which is mainly used for this study is "TURB-GEN". In addition to the instability models viz negative damping and coupling force which are already available in this code.

Vortex Shedding.

The exciting forces due to vortex shedding can be obtained by oscillating lift and drag force produced on the tube which are as follows:

$F_l = 1/2\ \rho V^2 DL(\sin(2\Pi f_s t))$

$F_d = 0.1*(1/2\ \rho V^2 DL(\sin(4\Pi f_s t)))$

Where

ρ= density of fluid
V=velocity of flow
D=diameter of tube
L=length of the span.
f_s= vortex shedding frequency = 0.2V/D.

Turbulence
Theoretical background
Tube dynamics

The dynamic equation of the tube with loose support under turbulent excitation can be written on the modal basis as:

$$m_n q_n(t) + A_n q_n(t) + k_n q_n(t) = f_e^t(t) + f_i^n(t) \quad (20)$$

Where $q_n(t)$, $q_n(t)$, $q_n(t)$: modal displacement, velocity and acceleration,

$f_e^t(t)$: generalized force due to turbulence,

$f_i^n(t)$: generalized force due to impact on the support.

<u>**Random force:**</u>

In the absence of support $f_e^n(t)$ is nothing but a banded white noise of spectrale density:

$$Sn = \left[\frac{1}{2}\overline{\rho}\ \overline{V}^2 D\right]^2 \frac{D}{V} L^2 \frac{D}{D_0} \frac{L_0}{L} a_n \phi_f^e(F_r) \quad (21)$$

Where

D	:	tube diameter
D0	:	tube reference diameter
L	:	tube length,
L0	:	tube reference length,
$\overline{\rho}$	:	density of secondary fluid,
$\overline{V}$ V	:	average velocity
a_n	:	modal correlation factor which is defined as:

$$an = \frac{1}{\lambda_e L} \int_0^L \int_0^L \Phi_n(s_1)\Phi_n(s_2) r(s_1) r(s_2) U^2(s_1) U^2(s_2) e^{\frac{|S_1 - S_2|}{\lambda_e}} ds_1 s_2 \quad (22)$$

$$\phi_F^e(f_r) \rightarrow 4\,x\,10^{-4}\,f_r^{-0.5} \qquad 0.01 < f_r < 0.2$$

$$\rightarrow 3\,x\,10-6\,f_r^{-3.5} \qquad 0.2 < f_r < 3 \qquad (23)$$

with L_0 = 1m, D_0 = 0.02 m

also $\Phi_F^e(f_r)$ can be obtained from SRM spectrum as follows:

$$\phi_F^e(f_r) \rightarrow 4\,x\,10^{-0.7}\,f_r^{-2.2} \qquad 0.1 < f_r < 2 \qquad (24)$$

with L_0 = 1 m, D_0 = 0.019 m

Taking first modal spectrale density as reference we can define the generalized force for n[th] mode as:

$$f_e^n(t) = \sqrt{\frac{s_n}{s_1}}\left[\gamma_n^x B_x(t) + \gamma_n^\gamma B_y(t) + \gamma_n^z B_z(t)\right] \qquad (25)$$

Where $B_x(t)$, $B_y(t)$,$B_z(t)$: white noise with spectral density for first mode,

$\gamma_n^x, \quad \gamma_n^y, \quad \gamma_n^z$: modal directional weighting factors defined as:

$$\gamma_n^{x^2} = \frac{\int_0^L (\phi(s).\vec{i}_x)^2\ ds}{\int_0^L (\vec{\phi}_n(s)\vec{\phi}_n(s)\ ds} \qquad (26)$$

Impact forces:

In the presence of support, the impact forces in the normal as well as in the tangential directions are defined as follows:

$$\vec{F}_N = -K_N < \vec{x}.\vec{n} - g > \qquad (27)$$

where $\vec{F}n$: normal impact force,

$\vec{x}$: tube displacement,

$\vec{n}$: unit vector in the normal direction,

g : gap size,

K_N : normal impact stiffness,

< > : indicates the positive value.

$$
\begin{aligned}
&\vec{\mathbf{F}}_{\mathbf{T}} = -\mu(\mathbf{F_N})\frac{\vec{\dot{\mathbf{X}}}\mathbf{T}}{\left|\vec{\dot{\mathbf{X}}}_{\mathbf{T}}\right|} \quad \textbf{if} \quad \dot{\mathbf{X}}_{\mathbf{T}} \neq 0 \quad (\textbf{sliding}) \\
&\left|\vec{\mathbf{F}}\mathbf{T}\right| < \mu\left|\vec{\mathbf{F}}_{\mathbf{N}}\right| \quad \textbf{for} \quad \dot{\mathbf{X}}_{\mathbf{T}} = 0 \quad (\textbf{adherence}) \\
&\quad -K_T\left(X_T - X_T^0\right) - C_T \quad \dot{X}_T
\end{aligned} \tag{28}
$$

Where $\vec{FT}$: tangential impact force,

μ : coefficient of friction,

$\dot{X}_T$: velocity in the tangential direction,

K_T, C_T : adherence stiffness and damping,

X_T^0 : zero velocity tangential displacement

Support stiffness and damping [3]:

The support normal as well as tangential stiffnesses KN and KT are calculated as follows:

$$\frac{1}{KN} = \frac{1}{Kc} + \left[\frac{1}{K_{SN}} - \sum_{n=1}^{N}\frac{\phi_{iN}^2}{K_{iN}}\right] \tag{29}$$

$$\frac{1}{K_T} = \frac{1}{K_{ST}} + \frac{1}{K_{rot}} - \sum_{n=1}^{N}\frac{\phi_{iT}^2}{K_{iT}} \tag{30}$$

Where Kc : ovalisation stiffness of the tube, $1{,}9\frac{E\,e^2}{D}\sqrt{\frac{e}{D}}$,

K_{SN} : bending stiffness in normal direction,

K_{ST} : bending stiffness in tangential direction,

K_{rot} : rotational stiffness of the tube,

ϕ_{iN} : modal displacement in normal direction at the support,

ϕ_{iT} : mode displacement in tangential direction at the support,

K_{iN}, K_{iT} : generalized stiffness in normal and tangential directions.

Fluid Elastic Instability

Theoretical background on instability:

Among many other models of instability, two important models which are incorporated in this code "TURB-GEN" are as follows:

Negative damping model
Coupling force model

A brief description of the Negative damping model can be found below.

Negative damping:

The modes which re unstable are provided with negative damping as defined

$$\varepsilon = \varepsilon_0 \left[1 - \left(\frac{V}{V_c} \right)^2 \right] \tag{31}$$

where ε : damping of unstable mode,
ε_0 : damping of stable mode,
$\overline{V}_c$: critical velocity of the mode,
V : actual flow velocity.

The critical velocity of the unstable mode can be calculated using Connors model as follows:

$$Vc = fi \quad DK \left[\frac{2\Pi \varepsilon_i \; m_i}{\overline{\rho} D^2 \int_0^L r(s)\, u^2(s)\, \phi_i^2(s)\, ds} \right]^{1/2} \quad (Connors \quad \text{mod}\,el)$$

CONCLUSION

Various flow induced vibration mechanisms and their preventions have been discussed. Due to cross flow the tubular structures are subjected to vortex shedding, fluid elastic instability, turbulence and acoustic vibrations. The prevention of above mechanisms are mentioned. The fluid elastic instabilities of shell structures due to weir flow and their prevention are investigated. The fretting wear of tubular structures at their loose supports and their estimation are discussed in this paper.

REFERENCE

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