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A PARAMETRIC STUDY OF FLUIDELASTIC INSTABILITY AND WEAR DAMAGE IN FAST REACTOR STEAM GENERATORS

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SUMMARY

Fluidelastic instability of steam generator tubes results in vibrations and wear damage of the tubes. A predictive analysis of this phenomenon is presented for Liquid Metal Fast Reactor Straight tubes Steam Generator.

The computer code GERBOISE is used to analyse the sensitivity of the instability ratio and of the wear work-rate of the tubes against loose supports as a function of supporting grids arrangement, tube-to-support clearance, eccentricity of the tube, ...

1 - INTRODUCTION

This study deals with fluidelastic instability and wear damage of straight tubes Steam Generator for Liquid Metal Fast Reactor (LMFR). We consider a vertically mounted Steam Generator (SG) with feedwater/steam mixture on the tube side and secondary sodium out of the tubes. The tubes are supported at intermediate points. The supports are provided by restricted clearance through a number of grid plates (figure 1a). For manufacturing considerations, there is a clearance between tube and tube supports (figure 1b).

The secondary sodium enters the annular inlet window situated at the top of the SG. It passes through the first tubes rows perpendicularly and then flows down axially to the bundle. The sodium emerges at the outlet window where the flow again is normal to the bundle axis.

The fluidelastic instability is a severe self-excited vibration which occurs when the cross-flow velocity exceeds a critical value. It can cause the tubes to impact and to rubb against their supports, with wear damages.

We present an analysis of this non linear dynamic behaviour. The objective is to establish the influence of various supporting conditions (number, position and spacing of the grids, value of the clearance) on the instability and the wear. We make use of the computer code GERBOISE developped by FRAMATOME and CEA [1].

2 - COMPUTATIONNAL METHOD

The method implemented in the code GERBOISE was presented in previous papers, see [1, 2], where more detailed informations will be found.

GERBOISE computes the response of a tube submitted to the following excitations :

a/ Turbulent forces due to the external cross flow. They are modelized as a banded white noise whose spectral density is scaled versus various experiments.

b/ Fluidelastic forces : fluidelastic instability is modelled according to the CONNORS semi-empirical model [4].

The fluidelastic instability growth rate is simply given by a negative damping ζ_i^* :

$$\zeta_i^* = \zeta_i \left(1 - R_i^2\right)$$

where : ζ_i is the damping of the i^{th} mode,
R_i is the instability ratio of the i^{th} mode,
R_i = (effective sodium velocity)/(critical velocity).

The tube is represented by a classical beam finite element technique.

The non linear dynamic response is computed in two steps :

a/ The modal basis of the tube is calculated for linear Boundary Conditions (simple supports, clamped conditions). The tube is free to move at loose supports.

b/ The equations of motion are solved in the time domain by projection on the modal basis. The loose supports are considered at this step :

When the tube is touching its support, an impact dynamic is modelled with an elastic impact force and a frictionnal force (Coulomb's equations). The following impact parameters are used :

K_C impact stiffness
μ coefficient of friction.

To modelize the adherence process, a specific stiffness coefficient K_T and damping coefficient C_T are used [6].

An instantaneous wear work-rate is defined using Archard's equation :

$$\dot{W}(t) = F_N(t) \ \dot{X}_T(t)$$

with : $F_N(t)$ is the time dependant normal impact force
$\dot{X}_T(t)$ is the tangential tube velocity against the support.

For wear prediction, time averages of displacements, forces, wear, ... are performed. The limits of the averaging interval have to be choosen carefully in order to provide a sufficient duration in the steady state regime.

3 - MODELS DESCRIPTION

Figure 2 presents the Reference geometry. It is designed in order to avoid instability when all supports are assumed effective. Despite of the tube-to-support clearance, this is realistic for many supports thanks to static bend of the tubes, misalignment of the grids, of the tubular plates, ... However among the 1 000 and more tubes of the bundle, it is highly probable that one or more grid supports will became ineffective (that is, the tube can move freely in all directions within the circular hole). And the worst situation corresponds to ineffective supports in the vicinity of the inlet or outlet window were the cross flow and the fluidelastic excitation can occur.

The tube is modelled as clamped at the tubular plates and simply supported at each effective grid support. Three models will be considered here, see table 1. The tube outer diameter is 16.4 mm ; its wall thickness is 2.2 mm.

Table 1. Models heights with number of grids for reference geometry.

Tube model	Lower boundary	Upper boundary	Number of grids
Whole tube	X = 0, Lower Tubular Plate	X = 33.3 m, Upper Tubular Plate	35
"7 spans"	X = 0, Lower Tubular Plate	X = 6.55 m, Grid 7	7
"3 spans"	X = 0, Lower Tubular Plate	X = 2.75 m, Grid 3	3

The paper will be focused on the effect of various changes in the outlet window region (the tendancies are the same for the inlet window and the vibrations at the two windows are uncoupled) :

a/ Number of ineffective supports and for these :

- eccentricity of the tube within the grid hole,
- clearance of the tube within the grid hole.

b/ Window height.

c/ Additional grid supports.

The sodium cross flow distribution within the windows has a triangular shape with a mean velocity : $\bar{V} = 1.42$ m/s (outlet, reference geometry). When the height of the window varies its value is adjusted in order to keep constant the flowrate.

For each configuration of effective supports, the modal basis of the tube is established with the bending modes of vibration in two orthogonal horizontal directions (Y and Z). The number of modes required is physically limited by the impact stiffness K_C [2].

The impact characteristics of the tube against the loose supports are assumed to be :

$K_C = 38.10^6$ N/m
$K_T = 12.10^6$ N/m
$C_T = 6\ 600$ Ns/m
$\mu = 0.4$

The tube-to-support clearance, J, will vary : $0.3 \le J \le 0.7$ mm.
The tube's eccentricity, e, will vary from e = 0 (tube centered) to e = J (tube in contact at rest with the support).

The modal damping value is assumed to be :

ζ_i = 0.5 % for configuration with only one ineffective support.

ζ_i = 1 % for configuration with 2 or more loose supports, except modes which are unstable (regarding fluidelastic instability) where ζ is estimated from squeeze-film damping in the clearance between tube and support [3].

The random forces due to the cross-flow turbulence are taken in account. They initiate the motion of the tube. When this is done, the fluidelastic forces became predominent.

The Threshold instability constant K of the CONNORS instability model is fixed to the value :

$K = 2$

The tube response is computed for 10 seconds.

4 - RESULTS AND DISCUSSION

4.1 - Effect of the number of ineffective supports

The table 2 presents the GERBOISE instability and wear results for the whole tube model with 0, 1 and 2 loose supports. The geometry is the reference geometry (figure 2). The clearance is 0.7 mm with off-centering of the tube at rest.

$\overline{W}$ is the normalized mean wear work-rate of the tube against its loose support (at the support were it is maximum when they are many). f_i is the frequency of the most critical mode.

One can observe that the wear damage is a highly non linear function of the instability ratio : here an increase of 45 % of R_i leads to an increase of 700 % of the wear work-rate.

Table 2. Sensitivity to the number of loose supports

	Most critical mode		Wear results
Ineffective supports	f_i (Hz)	R_i	$\overline{W}$
N°	37	0.56	0
Grid 2	17	4.2	1
Grids 1 and 2	8.7	6.1	6.8

4.2 - Effect of the number of spans

The simplified models (7 spans, 3 spans) allow easier sensitivity studies than the whole tube model. However, their predictions have to be considered **for qualitative purpose only**.

The table 3 emphasizes this effect of model. In the present case, the "3 spans" model underestimates the wear work-rate by a factor 9 although the instability ratio value is correct.

Table 3. Effect of models
Calculus with 2 loose supports (grids 1 and 2), clearance 0.7 mm, tube eccentered

Model	Most critical mode		Wear results
	f_i (Hz)	R_i	$\overline{W}$
Whole tube	8.7	6.1	6.8
7 spans	8.7	6.2	4.4
3 spans	6.9	6.25	0.75

4.3 - Effect of the tube-to-support eccentricity

The figure 3 presents the variation in mean wear work-rate as a function of the initial (at rest) position of the tube axis within the second grid hole. The calculation is performed with the whole tube model, one support is ineffective (grid 2), the clearance is 0.7 mm.

One can observe that the wear work-rate is maximum when the tube initially is half-eccentered and minimal when it was in contact with the support. An analysis of the impacts characteristics shows that :

- The half-eccentered position makes easier the impact with slidding motion against the support. Then friction and wear are increased.
- The centered position provides rather normal impacts.
- The in-contact position acts like some quasi-effective support ; the tube displacement is strongly reduced at the support.

This result is strongly dependant of the fluidelastic forces level :

With two ineffective supports (grids 1 and 2), the other parameter being fixed, the **wear work-rate became independant of the initial position of the tube within the grid's hole** : the impact forces are much more important. They tend to center the tube within the holes whatever its initial position is.

4.4 - Effect of supports spacing

The three-span model is used to investigate the sensitivity of wear results to the distance between grids 1 and 2. These grids give the lower and upper boundaries of the outlet window. They are assumed to act as ineffective supports with a clearance 0.7 mm.

The figure 4 presents the instability and wear results : it exists an optimal value of the grids spacing which minimizes the wear damage. In the present configuration **it does not correspond to the value which minimizes the instability ratio.**

4.5 - Effect of additional supports

An intuitive solution, when fluidelastic instability occurs, consists to reinforce the supporting system of the tubes, with in the present case, more grids and correlatively a reduction of the span height.

However, when ineffective support conditions occur, fluidelastic instability can arise again. The figure 5 presents the consequences on the wear results for various reinforced solutions in comparison with the reference geometry results. The clearance is 0.7 mm.

One can see that :

- The wear work-rate does not vary similarly to the instability ratio : reinforced solutions with **higher frequencies for critical modes give important increases in $\overline{W}$ although R_I can be diminished.**

- The supporting systems with more grids do not provide an amelioration of the wear damage relatively to the reference geometry (for the present data and with the assumption of loose supports).

4.6 - Effect of tube-to-support clearance

The figure 6 emphasizes the sensitivity of the wear work-rate $\overline{W}$ to variations in the tube-to-support clearance.
Reducing the clearance decreases strongly the wear. $\overline{W}$ looks roughly proportionnal to the square of the clearance.

Such an effect of gap was previously noted in [2] in a similar case of fluidelastic loading. It must be noticed that this trend is opposite to that observed in the case of forced excitation by turbulence, see [1, 5].

5 - CONCLUSION

This analysis presents a first predictive vibrations and wear study of LMFR Steam Generator tubes in case of fluidelastic instability.

The computer code GERBOISE is used to analyse the sensitivity of the fluidelastic instability and of the wear work-rate of tubes against loose supports to various supporting conditions.

The results indicate that there exist complex and highly non linear relations between the various parameters of the problem : the wear damage is not a growing function of the instability ratio. Some supports arrangements which minimize the fluidelastic instability risk when all supports are effective can provide highly damaging wear results when some supports became ineffective.

The GERBOISE code is a fruitfull tool to explore this domain. This work is on-going with objectives of full validation of the code for LMFR applications, of improvements of the model (fluidelastic forces for instance [7]), and of pursuit of our engineer approach in order to understand these complicate non linear and multi-parameter phenomena.

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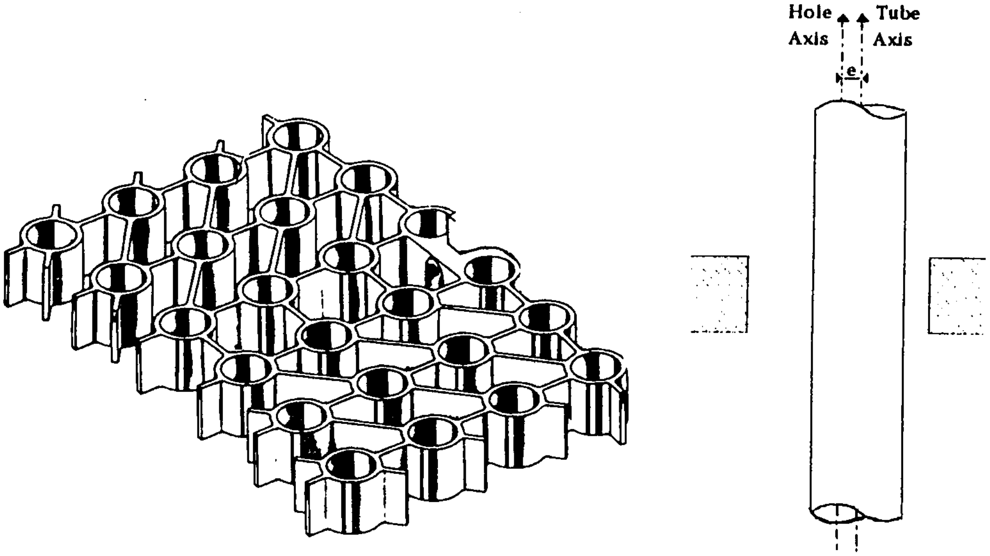


1a, grid plate 1b, tube support

Figure 1 : GRID SUPPORT

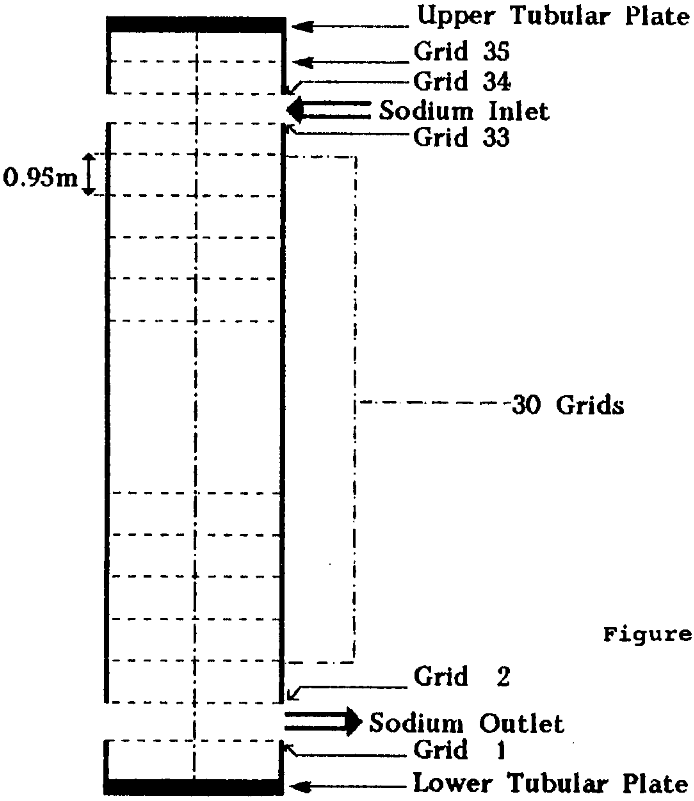


Figure 2 : REFERENCE GEOMETRY OF THE TUBE BUNDLE

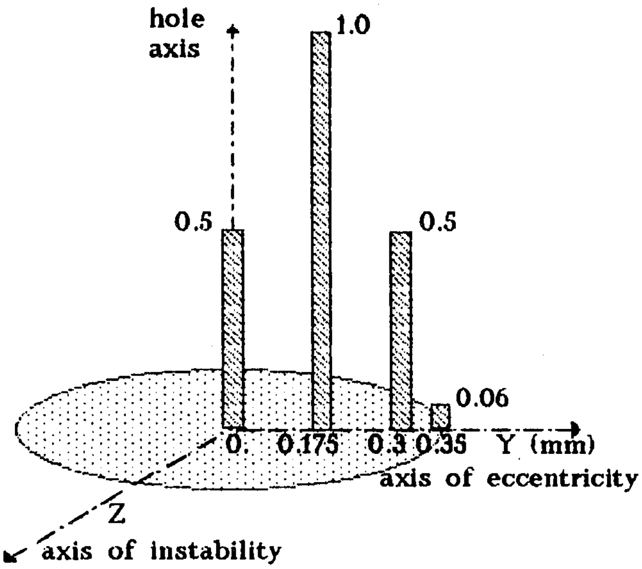


Figure 3 : EFFECT OF ECCENTRICITY ON WEAR WORK-RATE $\overline{W}$

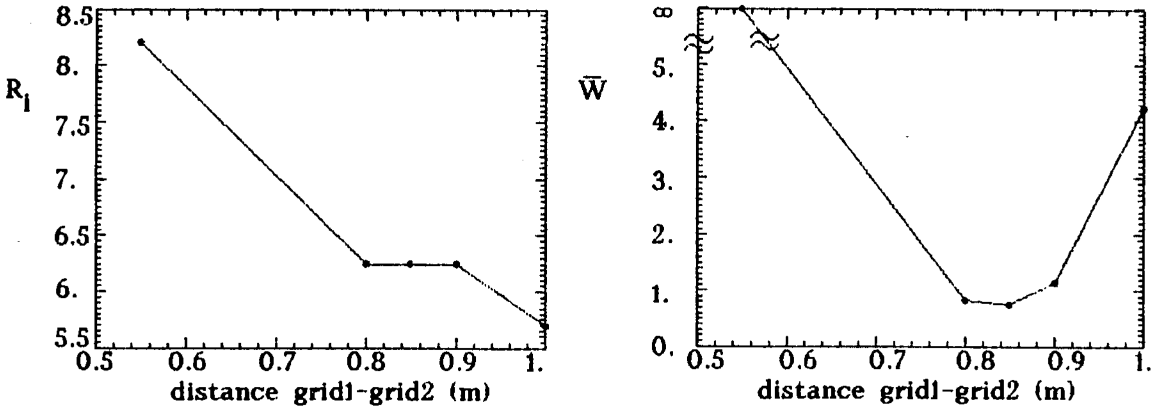
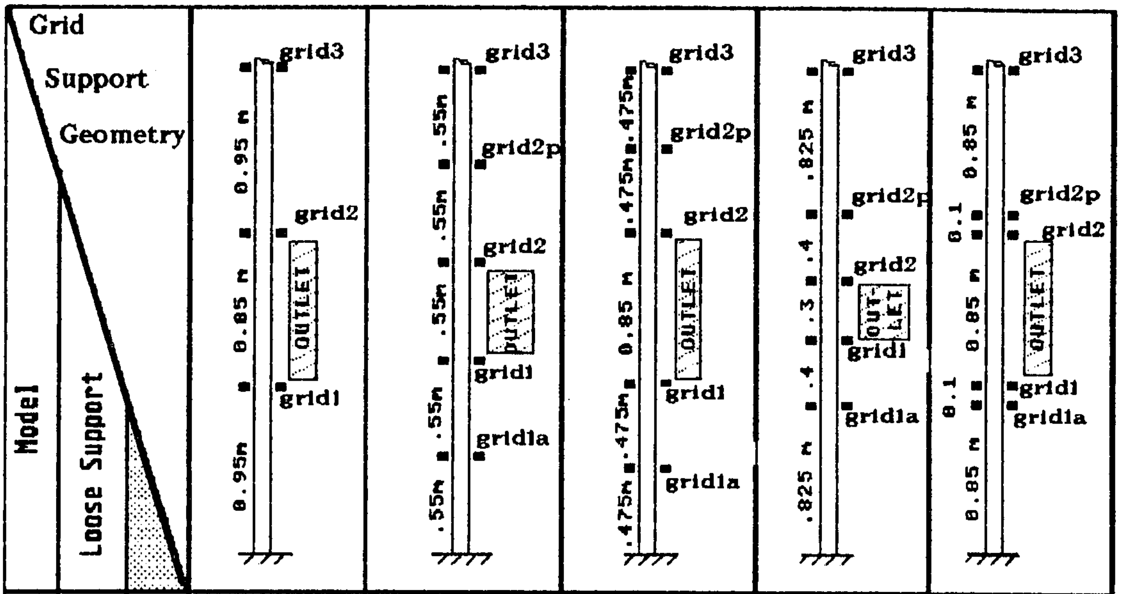


Figure 4 : EFFECT OF SUPPORT SPACING (GRIDS 1 and 2)



3 Spans	Grid 2	R_i	5.			1.47	
		f_i	13.5 Hz			111. Hz	
		$\bar{W}$	*0.54*			*1.9*	
	grids 1 & 2	R_i	6.25			3.79	
		f_i	6.9 Hz			38. Hz	
		$\bar{W}$	*0.75*			*	
	grids 1a, 1, 2, 2p	R_i			6.25		6.25
		f_i			6.9 Hz		6.9 Hz
		$\bar{W}$			*1.14*		*1.11*
7 Spans	grids 1 & 2	R_i	6.2	4.28			1.8
		f_i	8.7 Hz	21.6 Hz			41.1 Hz
		$\bar{W}$	*4.4*	*8.3*			*
Whole Tube	grids 1 & 2	R_i	6.1	4.3			
		f_i	8.7 Hz	21.6 Hz			
		$\bar{W}$	*6.8*	*33.8*			

*** Wear growing exponentially with time**

Figure 5 : EFFECT OF MORE SUPPORTS, WITH VARIOUS ARRANGEMENTS
The first column is the reference geometry.
Grids 1a and 2p are additional grids.

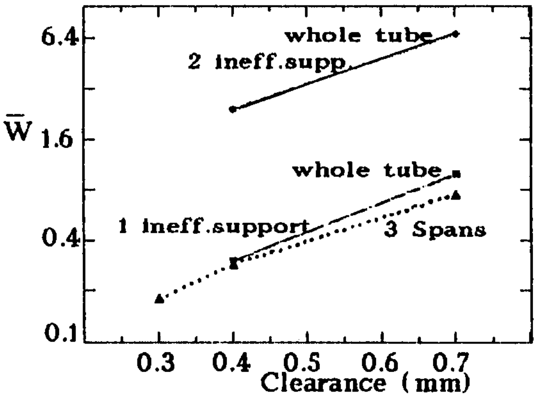


Figure 6 : EFFECT OF TUBE-TO-SUPPORT CLEARANCE (reference geometry)