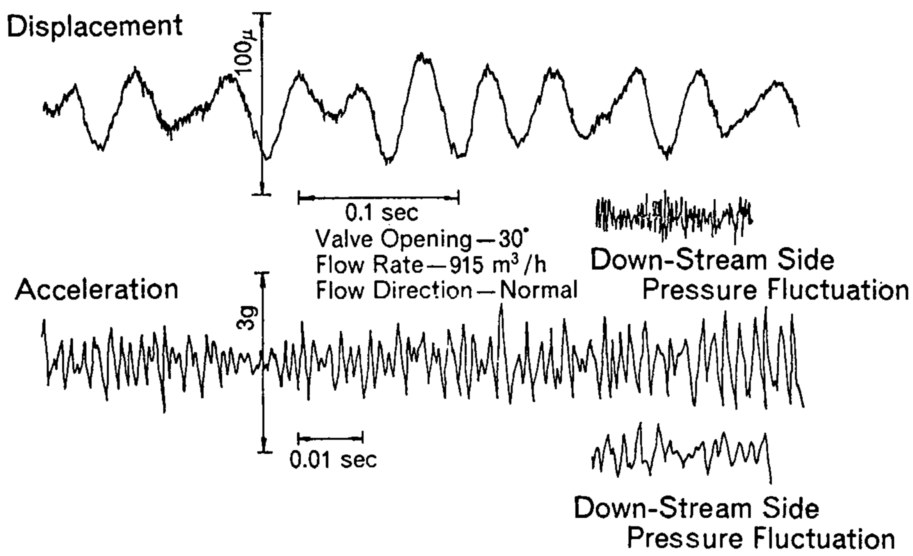

Fig. 21 Vibration Waveform Recorded on an Electro-Magnetic Oscilograph

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Heat Exchanger Vibration by D. J. W. Richards, United Kingdom.

1. Introduction

The heat exchangers of various types are common items of plant in the generation and transmission of electricity. The amount of attention given to the flow-induced vibrations of heat exchangers by designers is usually related to the operational history of similar items of plant. Consequently, if a particular design procedure yields items of plant which behave in a satisfactory manner during their operational life, there is little incentive to improve or refine the design procedure. On the other hand, failures of heat exchangers clearly indicate deficiencies in the design procedures or in the data available to the designer. When such failures are attributable to flow-induced vibrations, the identification of the mechanisms involved is a prime importance. Ideally, basic research work provides the background understanding and the techniques necessary to be able to identify the important mechanisms. In practice, the investigation of a flow-induced vibration problem may identify the presence of mechanisms but may not be able to quantify their effects adequately. In these circumstances the need for additional research work is established and the objectives of the research programme emerge.

The purpose of this paper is to outline the background to the current research programme at C.E.R.L. on heat exchanger vibration.

2. Background

As an operator of heat exchangers the availability of the plant for the life time of a station is a factor of considerable importance. If the availability can be reduced significantly by failures, then the extent of the consideration of the mechanisms that may cause such failures should be related to the scope for repairs or replacement of the heat exchanger. Nuclear heat exchangers are often large compact items of plant with very limited scope for repairs and their replacement is usually a lengthy and costly operation. Consequently, it is considered that research work aimed at improving the design procedures for nuclear heat exchangers is justified. However, it should be appreciated that many of the heat exchangers, associated with operational gas cooled reactors, have given satisfactory service; failures of nuclear heat exchangers are relatively rare and only a proportion of these failures can be attributed to flow-induced vibrations.

3. Early work on flow-induced vibrations of heat exchangers

The early work on flow-induced vibrations of heat exchangers often tacitly assumed that the excitation experienced by a stationary tube could be used to determine the vibration levels of tubes given the structural dynamics of the tube and its support system. Clearly, this assumption neglects the aeroelastic effects relating the enhancement of the excitation by the tube motion. Consequently, the initial investigations of the fluid dynamics of heat exchanger vibration attempted to allow for the aeroelastic effects. The essence of the approach follows from a consideration of the equation of motion of a single degree of freedom system, namely:-

$$\ddot{X} + 2\zeta_s \omega_o \dot{X} + \omega_o^2 X = \frac{\rho d^2 V^2}{2Md} F(t) \qquad \ldots(1)$$

where X = cross-flow displacement

ζ_s = ratio of structural damping to the critical structural damping

ω_o = undamped natural frequency

M = generalised mass per unit length

d = diameter of tube

V = reference velocity

$F(t)$ = aerodynamic generalised force coefficient

ρ = density of the gas

Introducing the non-dimensional displacement $\eta = X/d$ and the non-dimensional time $\bar{t} = Vt/d$ reduces equation (1) to:-

$$\ddot{\eta} + 2\zeta_s \left[\frac{\omega_o d}{V}\right] \dot{\eta} + \left[\frac{\omega_o d}{V}\right]^2 \eta = \left[\frac{\rho d^2}{2M}\right] F(\bar{t}) \qquad \ldots(2)$$

In general, $F(\bar{t})$ is a function of the fluid dynamic properties of the flow and the geometry of the tube array. It is convenient to write:-

$$F(\bar{t}) = F_o(\bar{t}) - K_a\dot{\eta} - H_a\eta - M_a\ddot{\eta} + \text{etc.} \qquad \ldots(3)$$

Following the practice adopted previously during the investigation of the wind loading on chimneys, equation (3) is simplified to:-

$$F(\bar{t}) = F_o(\bar{t}) - K_a\dot{\eta} \qquad \ldots(4)$$

where $F_o(\bar{t})$ = aerodynamic generalised force coefficient on a stationary tube

Then substituting equation (4) in equation (2) gives:-

$$\ddot{\eta} + 2\left[\zeta_s + \zeta_a\right]\left[\frac{\omega_o d}{V}\right] \cdot \dot{\eta} + \left[\frac{\omega_o d}{V}\right]^2 \eta = \left[\frac{\rho d^2}{2M}\right] F_o(\bar{t}) \qquad \ldots(5)$$

where $\zeta_a = \left[\frac{\rho d^2}{2M}\right] \cdot \left[\frac{K_a}{2}\right]\left[\frac{V}{\omega_o d}\right]$ = aerodynamic damping ratio

The solution of equation (5) can be obtained from:-

$$\bar{\eta}^2 = \left[\frac{\rho d^2}{2M}\right]^2 \int_o^\infty \frac{P(f)}{\left\{1 - \left[\frac{f}{f_o}\right]^2\right\}^2 + \left\{2\left[\zeta_s + \zeta_a\right]\frac{f}{f_o}\right\}^2} \qquad \ldots(6)$$

where $f = \frac{\omega d}{V}$

$f_o = \frac{\omega_o d}{V}$

$P(f)$ is the power spectral density of the function $F_o(\bar{t})$

Since in heat exchanger tube arrays $\left[\zeta_s + \zeta_a\right] \ll 1$ and as it was anticipated that the power spectral bandwidth of $p(f)$ at $f = f_o$ would be wide compared with $f_o\left[\zeta_s + \zeta_a\right]$, equation (6) reduces to:-

$$\eta' = S_o\left(\frac{\omega_o d}{V}\right) \cdot \left[\frac{\rho d^2}{2M}\right]\left\{\zeta_s + \frac{\rho d^2}{2M} D\left(\frac{\omega_o d}{V}\right)\right\}^{-\frac{1}{2}} \qquad \ldots(7)$$

where $D = \left[\frac{K_a}{2}\right]\left[\frac{V}{\omega_o d}\right]$ = aerodynamic damping coefficient

S_o = amplitude spectral function

η' = r.m.s. value of η

$S_o(\omega_o d/V)$ and $D(\omega_o d/V)$ are functions of the fluid dynamic properties of the flow and the geometry of the tube array. $S_o(\omega_o d/V)$ can be derived from a knowledge of the fluctuating pressure distributions on a stationary tube. However, $S_o(\omega_o d/V)$ and $D(\omega_o d/V)$ can be determined directly from the response of an aeroelastic tube installed in a tube array. Essentially, the response of the aeroelastic tube is measured, the aeroelastic tube is replaced by another aeroelastic tube with a different $M/\rho d^2$ and different ζ_s but the same external geometry and its response measured. If the suffices 1 and 2 identify the two aeroelastic tubes and their responses, manipulation of equation (7) yields:-

$$D\left(\frac{\omega_o d}{V}\right) = \frac{2m_1\left\{H^2\left(\frac{\omega_o d}{V}\right) \cdot \left[\frac{m_2}{m_1}\right]^2 \zeta_{s2} - \zeta_{s1}\right\}}{\left\{1 - H^2\left(\frac{\omega_o d}{V}\right) \cdot \left[\frac{m_2}{m_1}\right]\right\}} \qquad \ldots(8)$$

where $H\left(\frac{\omega_o d}{V}\right) = \eta_2'/\eta_1'$

$m = \frac{M}{\rho d^2}$

$$S_o\left(\frac{\omega_o d}{V}\right) = \eta_1'\left(\frac{\omega_o d}{V}\right)\left\{2m_1\left[2m_1\zeta_{s1} + D\left(\frac{\omega_o d}{V}\right)\right]\right\}^{\frac{1}{2}} \qquad \ldots(9)$$

4. Application of the theory

The above theory has applied successfully to the case of a model test bank with one aeroelastic tube mounted in an array of nominally rigidly fixed tubes. The results were found to fit the form of equation (7) and the variations of $D(\omega_o d/V)$ and $S_o(\omega_o d/V)$ were determined using equations (8) and (9).

It was found that as $V/\omega_o d$ increased $S_o(\omega_o d/V)$ increased to a maximum at the critical value of $V/\omega_o d$ corresponding to the Strouhal Number for the bank while $D(\omega_o d/V)$ decreased to a minimum and negative value. For this particular bank the minimum value of $D(\omega_o d/V)$ was approximately -0.4. Since $M/\rho d^2$ is of the order 100 in nuclear exchangers, it follows that ζ_a is of order -0.002. As ζ_s is usually greater than 0.2%, it appeared that equation (7) was a valid representation of the behaviour of the excitation and response of boiler tubes.

5. Subsequent applications

In the above theory $D(\omega_o d/V)$ decreasing represents an increase in the correlation length of the excitation due to coherent vortex shedding. However, for some value of $2M/\rho d^2 \cdot \zeta_s$ greater than $-D(\omega_o d/V)$ the response will become sufficiently great to correlate the coherent vortex shedding over the full length of the tube. When this happens there will be no further increase in $F(\overline{t})$ due to this effect. Equation (7) is no longer applicable in this situation $F(\overline{t})$ must be expanded differently:-

$$F(\overline{t}) = F_2(\overline{t}) - K_{2a}\,\dot{\eta} \qquad \ldots(10)$$

In equation (10) $F_2(\overline{t})$ is a much larger force that $F_o(\overline{t})$ in equation (4) and K_{2a} is the aerodynamic damping appropriate to the fully correlated situation for large values of η. Whereas $D(\omega_o d/V)$ is negative for values of $V/\omega_o d$ near the critical values, $D_2 = \left[K_{2a}/2\right]\left[V/\omega_o d\right]$ is positive. Under these circumstances the bandwidth of the power spectral density function of $F_2(\overline{t})$ at $f = f_o$ is no longer wide compared with $f_o\left[\zeta_s + \zeta_{2a}\right]$ so that equation (6) reduces to:-

$$\eta' = S_2\left(\frac{\omega_o d}{V}\right) \cdot \left[\frac{\rho d^2}{2M}\right]\left\{\zeta_s + \frac{\rho d^2}{2M} D_2\left(\frac{\omega_o d}{V}\right)\right\}^{-1} \qquad \ldots(11)$$

and S_2 = new amplitude spectral function.

In practice, it has been found that in another test bank, again using a single aeroelastic tube in an array of nominally rigid tubes, the measurements of response at low amplitudes fitted equation (7) while at high amplitudes the measurements tended towards the form of equation (11). For this test bank the approximate minimum value of $D(\omega_o d/V)$ appeared to be of the order -2.6. As there appear to be very few measurements of $D(\omega_o d/V)$ available, it is not possible to confirm the validity of this minimum value. However, it should be appreciated that for $M/\rho d^2$ of the order 100 ζ_a follows as being of order -0.013. Consequently, for ζ_s of order 1.3% both equations (7) and (11) are invalid.

6. Present investigations

Recently an opportunity arose to make some measurements of the vibration amplitudes of a few tubes in a nuclear heat exchanger. Consequently, the likelihood of large amplitude vibrations was assessed. It was concluded that the gas velocity under operating conditions was too low for fluid-elastic whirl to occur. On the other hand, laboratory tests using a two-dimensional bank of rigid tubes indicated the presence of narrow band excitation associated with vortex shedding.

In this particular nuclear heat exchanger the boiler tubes are arranged in plattens which are assembled in packets in the factory and the packets installed in the heat exchanger at site. The longitudinal and lateral pitches of the tubes were nominally uniform within a packet. However, there were no attachments between the packets in the heat exchanger. Consequently, small gaps or increases in the lateral pitch of the tubes were present locally.

The effects of local gaps were examined during the aforementioned laboratory tests. It was found that the centre frequency of the narrow band excitation experienced by the end-of-packet plattens was approximately 50% greater than that associated with mid-packet tubes for an increase of lateral pitch of ¼ inch.

The likelihood of a resonant situation occurring was then examined using a theoretical analysis of the dynamics of the boiler tubes. This analysis employed a finite element representation of the serpentine arrangement of boiler tubes. In order to provide an adequate theoretical analysis it proved necessary to incorporate the tube support system in detail. The results of this analysis indicated that coincidence of a natural frequency and the centre frequency of the narrow band excitation was unlikely to occur for cross-wind or horizontal modes of vibration. However, for end-of-packet plattens there was the possibility that coincidence of a natural frequency and twice the centre frequency of the narrow band excitation would be present for along-wind or vertical modes of vibration. Consequently, a few resonant situations were anticipated.

It was expected that the measurements of vibration amplitudes would be in accordance with the above. In particular, at some gas velocity a resonance would occur and a small peak in the response in the vertical direction would result. This expectation was based on the understanding the fluctuating drag forces at twice the coherent vortex shedding frequency would be much less than the fluctuating lift forces at the coherent vortex shedding frequency.

As mentioned above measurements of vibration amplitudes could only be made for a few tubes as both access and time were restricted. In the event, the actual situation prevaling in the heat exchanger was more complicated than the relatively simple picture presented above. As the analysis of the measurements is not complete, an illustration of the situation only is presented in the attached figure.

The graphs presented show the variations of the energy of vibration on a log scale with inlet guide vane angle of the gas circulators; an inlet guide vane angle of zero degrees corresponds to maximum gas velocity while one of 43 degrees corresponds to approximately 76% of maximum gas velocity. The dotted line is applicable to all boilers operational and consequently, the tubes in this platten are filled with water/steam, while the solid line is applicable to two boilers operational and consequently, the tubes in this platten are empty.

This particular platten is at the end of a packet and therefore a small gap was probably present locally. The graphs show the vertical vibration of a tube in this platten.

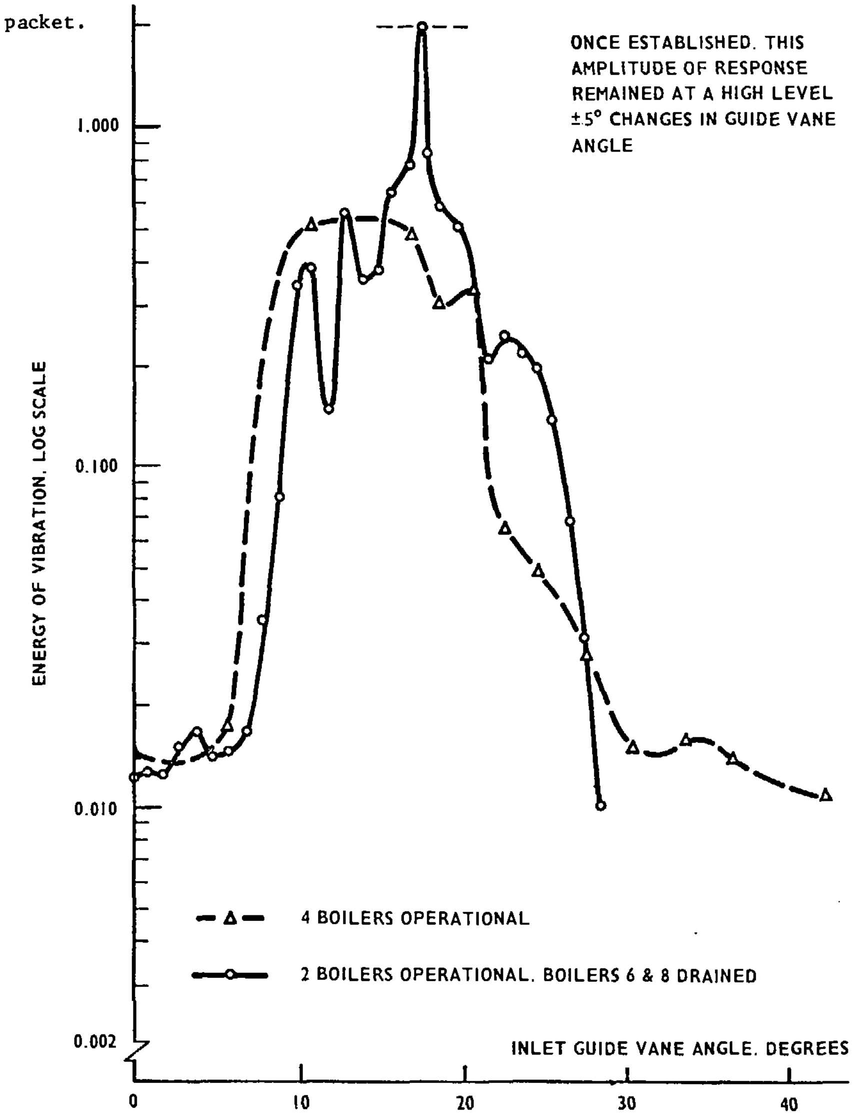
The first effect demonstrated in these graphs is that of a change in $M/\rho d^2$, the mass parameter. As the mass parameter is reduced by draining the platten, the response increases dramatically at an IGV angle of 18^o.

The second effect is the presence of a peak energy of vibration in the drained platten case which once reached is maintained for small changes in IGV angle or gas velocity. This effect is similar to the vortex locking phenomenon experienced by moving cylinders. A power spectral density analysis of the energy of vibration indicates that the motion of the tube is virtually at a single frequency, namely a natural frequency of the tube. Furthermore, this frequency is approximately twice the frequency of coherent vortex shedding.

The third effect is the presence of smaller peaks in the energy of vibration. These appear to be a forced vibration of the tube due to the fluctuating pressure field induced by other tubes experiencing peak responses at their resonances.

As stated previously, the analysis of these measurements is not complete. However, it is evident that interaction effects due to fluid dynamic coupling are present and that the vortex locking phenomenon precludes the use of the above theory.

From the practical point of view it is evident that small departures in tube geometry can produce large changes in excitation and response as only low energies of vibration were measured on tubes within a packet.



VERTICAL VIBRATION OF PLATTEN 544, 4 CIRCULATORS IN OPERATION.