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COMBUSTION INDUCED VIBRATION IN INDUSTRIAL BOILERS

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1. BACKGROUND

Low frequency combustion induced vibration has occasionally been encountered on both oil and gas fired boilers in recent years. Figure 1a shows a typical large boiler and Figure 1b the main components of the burners installed in such an unit.

The vibration is caused by the furnace acoustic modes (in one or two dimensions) being amplified by disturbances in the combustion process. The controlling feature in this phenomenon is the well established Rayleigh Criterion, as specified below.

The onset of vibration is usually very rapid and, if sustained, can lead to the failure of major boiler and burner components.

The frequency of vibration ranges from 5 to 100 Hz depending on the combustion chamber acoustic natural frequencies. These are dependent on the furnace geometry as shown by equation 1:-

$$f_a = \frac{c}{2}\sqrt{\frac{m^2}{l^2} + \frac{n^2}{w^2} + \frac{p^2}{h^2}} \quad (m,\ n,\ p = 0,\ 1,\ 2,\ 3\ \ldots..) \qquad - (1)$$

where l = length, w = width, h = height, c = speed of sound and a denotes the ath acoustic mode. A schematic drawing of possible acoustic modes in the furnace is shown in Figure 2.

Although it is possible for more than one mode of the boiler to show large amplitudes of vibration at one time, in general, once the Rayleigh Criterion has been satisfied, excessive vibration tends to occur at one frequency only. Because it is the boiler acoustics which are amplified, the onset of vibration can best be detected by measuring the furnace dynamic pressure using microphones or pressure transducers. Figure 3a shows a typical spectrum of dynamic furnace pressure amplitude vs frequency. In this case, the main mode of vibration has a frequency of 32 Hz which corresponds to a half standing wave from furnace front-to-back. This mode is shown schematically in Figure 3b. Other acoustic modes are also in evidence in the spectrum at higher frequencies.

2. THE RAYLEIGH CRITERION

The phenomenon of coupling between the heat release rate and furnace pressure was first described by JWS Rayleigh (Ref 1) as follows:-

"If heat be periodically communicated to, and abstracted from, a mass of air vibrating (for example) in a cylinder bounded by a piston, the effect produced will depend on the phase of vibration at which the transfer of heat takes place. If heat be given to air at the moment of greatest condensation or be taken from it at the moment of greatest rarefaction, the vibration is encouraged. On the other hand, if heat be given at the moment of greatest rarefaction, or abstracted at the moment of greatest condensation, the vibration is discouraged".

Expressed mathematically, vibration is likely to be sustained if the condition

$$\oint h(t)\ p(t)\ d(\omega t) > 0 \qquad (2)$$

is met, where h(t) is the heat release rate and p(t) is the furnace pressure oscillations both with respect to time, t.

The phase relationship between the fluctuations in heat release rate and the furnace pressure pulsations is controlled by the time delay between a disturbance in the flow and the resultant variation in heat release. This time delay, τ, is dependant on a large number of factors associated with the overall combustion process. These factors include fuel composition, burning velocity, fuel flowrate, fuel port configuration, atomisation (of oil), air flow characteristics, amount of excess air and burner geometry. When a boiler encounters vibration problems, the solution invariably involves a change in the value of τ. This can be achieved either by a deliberate attempt to change τ, or indirectly as a result of modifications to the flame geometry, arising from burner alterations made to reduce the excitation mechanism. This adjusts the phase relationship and reduces the amplification of the furnace pulsations.

For oil firing, the determination of τ is made difficult by the combustion characteristics. The atomisation of the oil by air, steam or by high pressure, forms a spray consisting of various sized droplets. The subsequent vapourisation and combustion of the droplets does not allow "single droplet" theory (Ref 2) to be applied because of the close proximity of the droplets to each other. This means that oxygen does not become available to the same degree or at the same rate as when a single droplet is burned. Because of this complicated combustion process, interpretation as to the effect of modifications to the oil tip on the accurate value of τ is currently impossible to deduce. The variation in τ for cases where atomisation is achieved by steam, air or the use of pressure jets is also indeterminate.

For gas firing, the value of τ can be adjusted by modification of the gas port configuration and rotation of gas spuds, although the actual numerical value of τ may be open to interpretation because a factor is required to define the flame shape. Using the equation defined by Putnam in Reference 3 for a simple flame:-

$$\tau = \frac{\delta}{V_p} + \frac{a}{nF} \tag{3}$$

where δ = dark space between exit plane and flame front

V_p = gas velocity

a = effective port radius

F = fuel flame speed

n = flame shape factor (between 2 and 3 for a simple burner)

$$\text{If } \frac{a}{nF} \gg \frac{\delta}{V_p} \text{ then } \tau = \frac{a}{nF} \tag{4}$$

If this is applied to an industrial burner, a mean effective radius has to be specified for gas spuds with more than one port hole. This takes into account the proportion of gas flowing through holes of different sizes.

Surprisingly, data on boiler conditions at vibration onset obtained from actual plant has regularly given values of n in the range 1 to 3 which, considering the assumption that $2 < n < 3$ is based on a very simple single port flame, shows that the principles used may still be applied to multiburner, multiport units.

3. EXCITATION SOURCES

It has been Babcock Power's experience that the initial disturbances which cause the fluctuations in heat release rate are predominantly controlled by either fuel or air flow characteristics, generated by the burner components.

A few situations whereby the mixing process between fuel and air in the combustion zone causes pertubations at the burner exit have also been observed. It has also been found that certain types of excitation have more influence on boiler vibration characteristics than others. Examples are given in Section 3.1 of those having a greater effect, these being termed primary sources. Amplification of boiler acoustic modes becomes more likely when these phenomena occur than when secondary sources are detected ie. those which do not have such a profound effect or have seldom been encountered (Section 3.2).

3.1 Primary Sources of Excitation

3.1.1 Fuel Controlled Inherent Instabilities

For gas firing Babcock have developed an empirical curve of vibration onset conditions shown in Figure 4. At low loads, the problem is one of mixing between the gas and air, but as throughputs increase the curve levels out indicating that a critical gas velocity is reached, and air flow characteristics have no effect.

Recent work has shown that this curve is useful as a general indication of limit on vibration zones. However, for fuels ranging from ethane/propane mixtures to hydrogen, experimental results have shown that the gas constituents can have an effect on the critical fuel velocity.

A tentative relationship increasing the critical gas velocity with decreasing gas density has provided a better fit over the above wide range of gases.

3.1.2 Aerodynamic Disturbances

The most probable source of aerodynamic excitation is that caused by disturbances generated by burner components such as swirlers, gas spuds or the quarl exit. These disturbances can result in vibration onset at certain critical air velocities which are nearly independent of the fuel velocity. This effect has been observed to be scaleable by an effective Strouhal number, defined as:

$$S_i = \frac{f_a d_i}{V_{ia}} \tag{5}$$

where f_a = furnace acoustic natural frequency

d_i = diameter of appropiate burner components

V_{ia} = critical air velocity over burner component i for acoustic mode a, at a frequency f_a

Apart from f_a, these values can only be estimated to a certain degree. There is a choice of burner component diameters; the v is dependent on air velocity profiles; and various S values have been proposed similar to those employed for flow over bluff bodies. Figure 5 shows how air velocities, which potentially may lead to resonances, can be determined by relating furnace acoustic modes with estimated effective Strouhal numbers. Burner components can be designed to allow for the possibility of this phenomenon taking place. If, as is often the case, this is impractical, the fuel port geometry is changed to give a Rayleigh Criterion which does not sustain pulsations. However, the excitation processes still exist and monitoring of the boiler should take place when load is raised through the critical ranges for the first time.

Another type of aerodynamic instability encountered is the precessing vortex, which has been studied, mainly by FDO (Ref 4). It involves the flame actually rotating as a whole round the axis of the burner and coupling with a furnace acoustic mode. Typical Strouhal numbers associated with this type of instability are 5 to 10 times that encountered with bluff body disturbances.

On site observations agree with the work carried out by FDO and suggests that this type of excitation can be decoupled if the circumferential component of fuel injection is reduced and the flow is directed towards the burner centre line.

3.1.3 Gas/Air Interaction

Disturbances can also be caused due to mixing of fuel and air prior to the combustion zone. These can be overcome by modifying the gas or air flow characteristics at the burner exit. One of the more common examples is the interaction between the air flow over gas spuds with a large number of small gas jets directed perpendicular to the air flow. This has usually been solved by limiting the number and size of these holes. However, care has to be taken that combustion is not affected detrimentally, as the initial purpose of these small holes was to improve flame stabilisation.

3.2 Secondary Sources of Excitation

The causes and solutions described in the previous section were all associated with conditions at the furnace inlet/ burner exit. Some investigators have claimed that disturbances in fuel supply lines and windbox have a significant effect. However, this has not been Babcock's experience. There have been occasions when certain conditions in the ductwork, eg. air heater bypass dampers or gas recirculation damper position, have had an effect on the exact condition for vibration, but these phenomena have not been seen to be a source of excitation. A similar situation exists regarding standing waves in the fuel supply line and air ducts. Although these have been detected and have influenced the exact frequency at which vibration occurs, they are not thought to have a direct bearing on whether or not vibration will take place.

4. INVESTIGATIONS ON BOILER PLANT

4.1 Oil Firing

Three cases of vibration on large scale oil fired plant have been investigated to date. On two different 660 MW units, it was thought that the excitation mechanism was due to aerodynamic effects. On one unit, this excitation coupled with a quarter vertical standing wave in the furnace, whereas on the other, the coupling was with a half standing wave across the boiler (front to back).

The final solution to the problems on both these units was to modify the swirlers to allow more air through the centre of the swirler, the philosopy being to reduce the intensity of the recirculation zone downstream of the swirler. On one unit, an open swirler with curved blades was installed with blades cut back at the root. On the other unit, every alternate blade in the original swirler was removed, the result of this being shown in Figure 6. It is thought that both these modifications also caused the flame front to be established further downstream from the burner, ie. the time delay τ was increased. This weakened the coupling mechanism between the heat release and the furnace acoustic mode.

In the third case, the boiler suffered from vibration only with all nine burners in operation at loads less than 30% MCR. The solution was to operate on 8 burners or less at these loads hence reducing burner interaction. Since this imposed no operational restriction, further investigation of the mechanism was not required by the client.

A small two burner oil fired package boiler, on which vibration was encountered at a frequency corresponding to a coupled Helmholtz resonator mode, also required burner modifications. The solution employed was to change the effective blockage of the swirler and to use a wider angled oil tip. This allowed the flame to act as an acoustic barrier to the pulsations in the neck of the resonator (the burner throat) and the resonator volume. This was one of the few times that the acoustic mode could be altered to achieve a solution.

4.2 Gas Firing

The most common source of combustion induced pulsation on gas fired plant has been due to high gas velocities, as discussed in 3.1.1. This has been observed on plant ranging from 120 MW (9 burners) to 660 MW (36 burners - Figure 7). The solution is nearly always obtained by increasing the net port area but the significance of the hole pattern in the spud has been highlighted on a number of occasions. This has a major effect on the flame characteristics (and the time delay) and can cause significant differences in boiler vibration characteristics.

Aerodynamically controlled combustion pulsations as discussed in 3.1.2 have been observed on a number of occasions. These have usually been overcome by increasing the free flow area and thus reducing the air velocity. Typical ways of altering the velocity are to increase the area of the throat (which is very expensive and is generally avoided whenever possible), remove the gas spud stabilising discs (after modifying the gas jets to make the discs redundant), or in the case of combined gas/oil fired burners, change the axial position of the swirler. The effect of this latter modification on a 660 MW boiler is shown in Figure 8.

A number of cases have been encountered when the point of vibration onset is not dependant on one particular gas or air velocity eg. Figure 9. These cases have been attributed to the mixing characteristics of the gas and air. Solutions to such problems have been obtained by varying the direction of the gas flow into the combustion chamber.

5. OBSERVATION OF BOILER PLANT BEHAVIOUR

Babcocks consider that, at present, rig tests of a single burner give results which are significantly different from those observed in multi-burner furnaces. In particular the acoustic mode of the pulsations and the interaction between burners is poorly simulated. The definitive solution is therefore usually obtained sooner if direct tests can be carried out on the actual plant.

This policy has been successfully applied for various clients throughout the world during the last few years producing a significant data-bank of the dependence of vibration on operational parameters. A few examples of these observations are given below:-

a) Burner interaction has been observed to depend on the macroscopic combustion zone density, ie. the number of burners firing, rather than the detailed positioning of the burners.

The interaction mechanism has taken the form, in aerodynamically controlled situations, of increasing the effective Strouhal number as defined in Section 3.1.2. This change in Strouhal only becomes evident when more than half the full complement of burners are being fired for apparent bluff body disturbances (ie. $Sn \approx 0.15 - 0.3$).

However in the limited situations where the mechanism appears to be controlled by a precessing vortex core (large effective S) the interaction has been very rapid and intense, making it virtually impossible to operate with the last burner in service.

b) The direction of fuel gas injection from multiport gas spuds has been observed to cause significant changes in boiler vibration characteristics as well as affecting the flame shape. It has been found that when firing some of the gas outside the burner periphery, it is often beneficial to alternate the direction of the spuds as shown in Figure 10a. This results in shorter delay times and short intense flames. The setting of all the spuds at the mean angle in such a situation usually has a major effect (usually detrimental) on vibration characteristics and flame profiles. However, in situations when firing gas towards the burner centre is necessary, alternating the spud directions has the same effect as all spuds being at the mean angle (Figure 10b). Larger time delays and longer luminous flames usually result, for this situation of firing gas towards the centre.

c) In practice, excessive vibration does not necessarily occur even if the Rayleigh Criterion is satisfied. The original criterion was developed by Putnam into a plot of expected vibration regions. Equation 6 was developed for the simple ideal situation with no damping and is the basis of the checkboard plot shown in Figure 11a;

$$\sin 2\pi f \tau . \quad \sin 2\pi f/f_b > 0 \qquad (6)$$

where f = frequency of excitation
f_b = burner acoustic natural frequency
τ = time delay

The presence of damping reduces the possibility of vibration and this is indicated by the reduction in the size of the pulsation squares as shown in Figure 11b. At present it is impossible to define quantitatively the extent of the above reduction without testwork on the plant, and the full pulsation area should be considered for design calculations.

Testwork has shown that the amount of excess air affects the boiler vibration characteristics as shown in Figure 12. When the instability is well damped, or has a low amount of excitation energy, it appears that low O_2 levels are beneficial (Figure 12a). If there is a small amount of damping, or high excitation energy, ie. the resonant range cannot be passed through because of excessive boiler vibrations, it appears that low O_2 levels cause a faster increase in pulsation levels (Figure 12b). Figure 13 shows an example of two instabilities encountered on a 9 burner gas fired boiler which confirmed this phenomenon.

Even when no modifications have been carried out, variations in acoustic damping can occur. On one 660 MW unit, testwork on consectutive days with similar fuels, flowrates and operating conditions resulted in different vibration characteristics as shown in Figure 14. The actual causes of these differences were not obvious. Factors such as ambient conditions and acoustic conditions at the boiler exit would require close examination to determine if there was any connection between vibration levels and acoustic damping of the furnace characteristics.

d) Structural damping characteristics of the boiler can also diminish the load range of the unacceptable vibration region.In some cases, it eliminates vibration altogether, ie. high peak pressure levels can result in low dynamic stresses. For instance, on one boiler a very high amplitude half standing wave in the furnace was being generated, but the structural vibration that resulted, was not excessive because of the rigidity of the unit. In general, it is found that small scale plant, eg. package boilers, can often withstand substantially higher furnace dynamic pressure levels than large multiburner units. In these latter more flexible units, aeroelastic feedback effects have been considered, but not observed up to the point at which the boiler vibration has become uncontrollable.

e) Once the Rayleigh Criterion has been satisfied for a particular set of boiler conditions, the dynamic furnace pressure becomes unacceptably high at one particular frequency. It was found that on one boiler, any one of three modes (56 Hz, 65 Hz, 76 Hz) could be excited depending on the fuel being fired (gas or oil) and the direction of the fuel jets at burner exit. For a particular configuration, the two dimensional front-to-back and side-to-side mode at 76 Hz indicated high pressure levels throughout the load range (Figure 15a), but once the air reached a certain velocity, it was the one dimensional mode from front to back at 56 Hz that caused excessive boiler vibration (Figure 15b). Other burner configurations led to the 65 Hz or 76 Hz modes becoming unstable.

f) Changing the burner design to improve the pulsation levels at loads far removed from the instability region does not guarantee an improvement at the vibration onset point. Conversely, it has also been found that higher levels at these low loads may be a result of the resonance being highly damped. This is shown schematically in Figure 16.

6. FUTURE REQUIREMENTS

To date, several mathematical models have been constructed to describe the phenomenon of combustion induced pulsations in industrial plant. These models have either considered a specific excitation mechanism or functions which, in Babcock experience, are of secondary importance. It is felt that in order to construct a mathematical model of the combustion induced vibration characteristics of boiler plant, which can be used at the design stage, the conditions at the burner exit must be represented.

Experience has shown that the terms in this model must be connected to parameters which can be measured on operational plant and related to practical aspects of burner design. Experimental work carried out to date will be used to validate the model.

It is probable that no single equation can cover every situation; only certain aspects of the combustion process will dominate in each case. The mathematical representation of this situation must be able to include the parameters which are of importance for a certain burner/furnace configuration and fuel type, and ignore those which will not contribute to the coupling between the heat release rate and furnace pressure pulsations.

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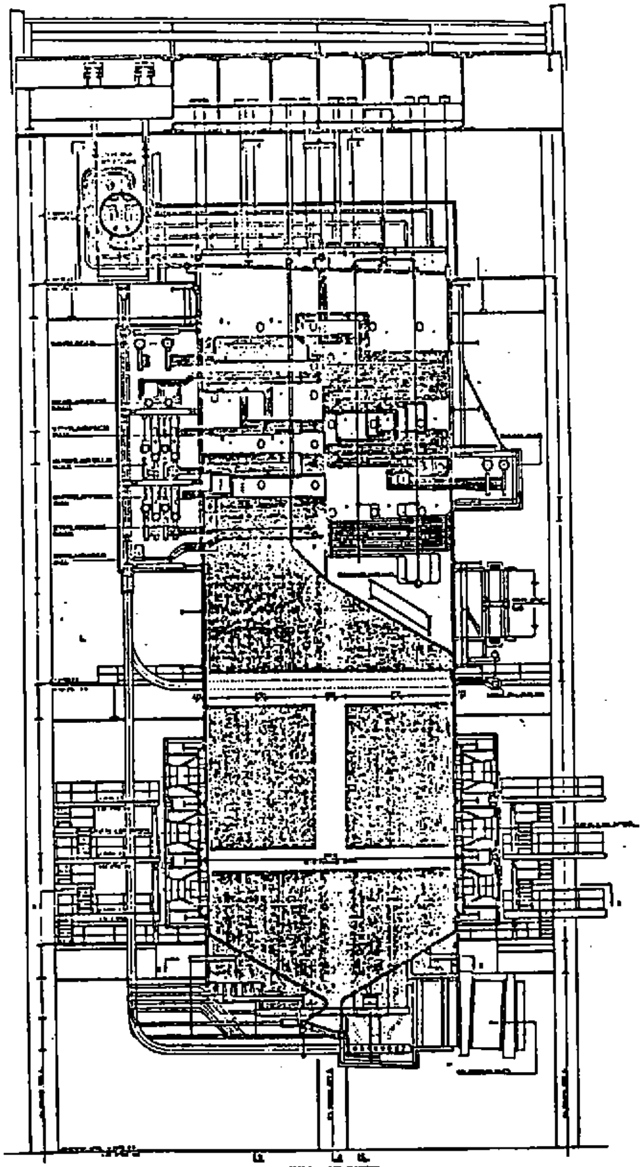
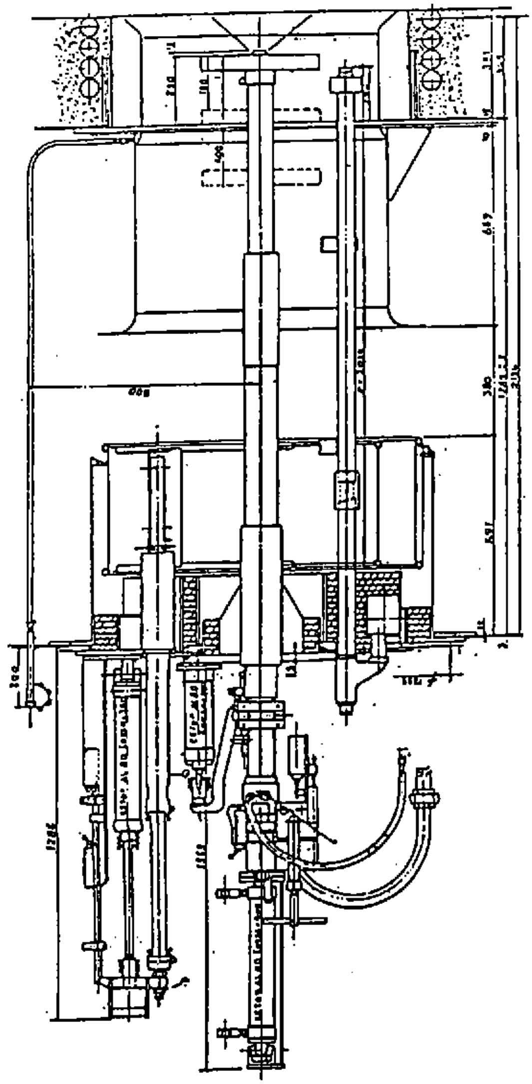
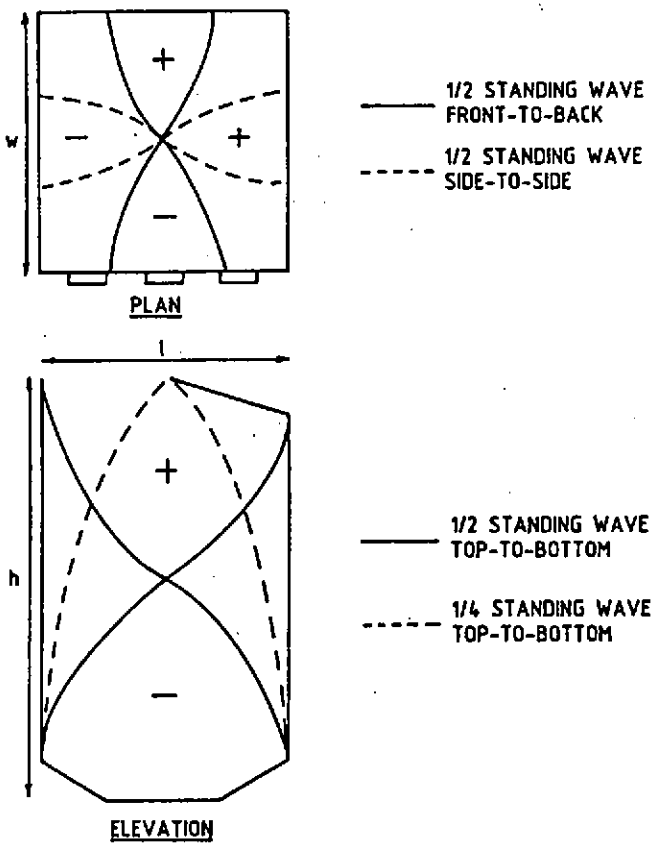


SECTIONAL SIDE ELEVATION OF A TYPICAL BOILER. FIG. 1A

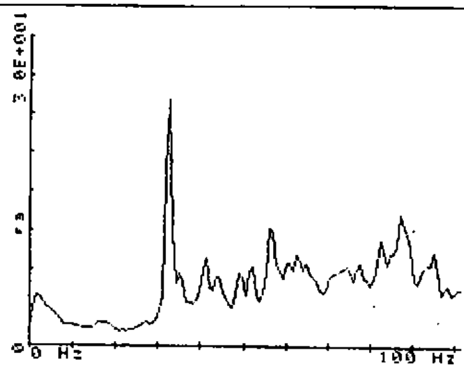


TYPICAL BURNER CONFIGURATION FIG. 1B

$$f_a = \frac{c}{2}\sqrt{\left(\frac{m}{l}\right)^2+\left(\frac{n}{w}\right)^2+\left(\frac{p}{h}\right)^2} \quad (m,n,p=0,1,2,3, \ldots\ldots) \qquad -(1)$$

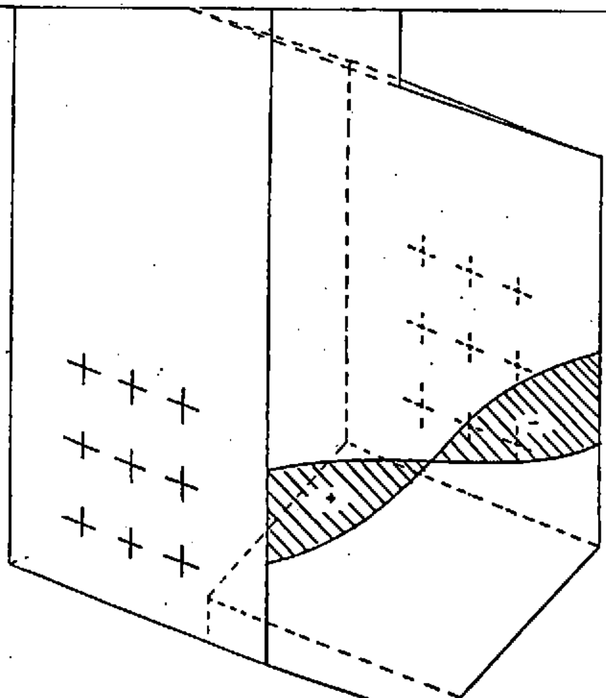


TYPICAL FURNACE ACOUSTIC MODES OF VIBRATION FIG. 2



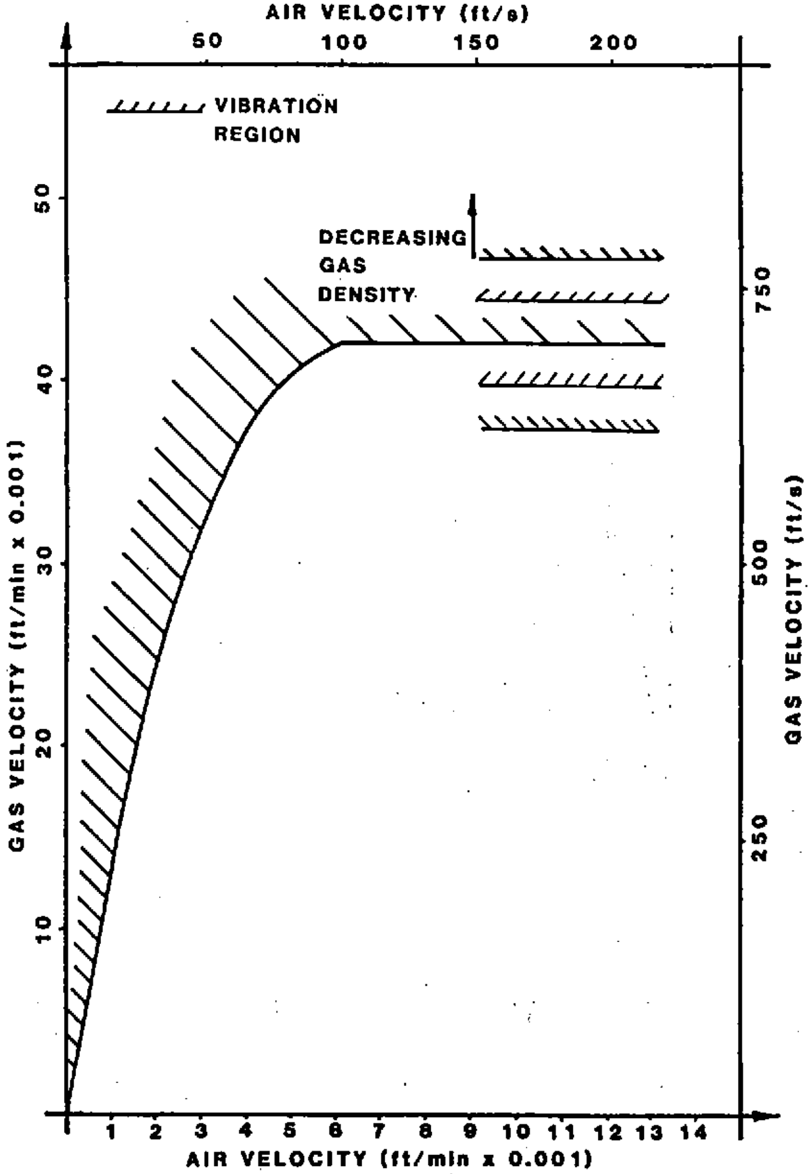
FURNACE PRESSURE FREQUENCY SPECTRUM

FIG 3a

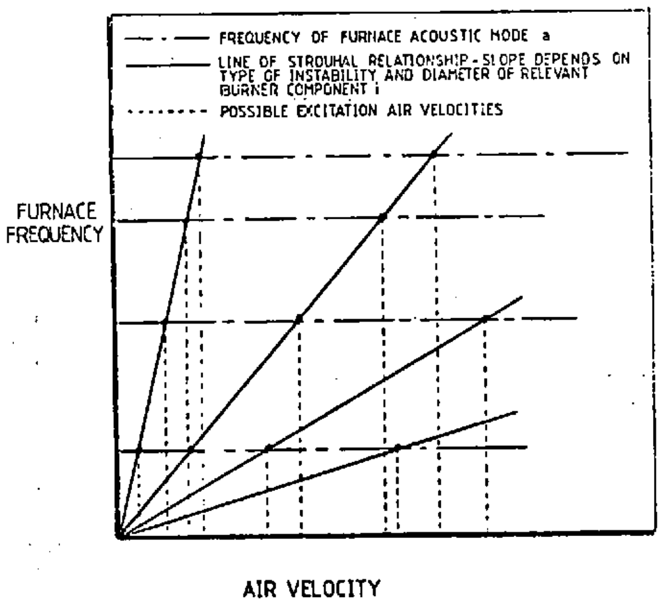


SCHEMATIC DIAGRAM SHOWING MODE OF VIBRATION - 1/2 STANDING WAVE BETWEEN BURNER WALLS

FIG 3b

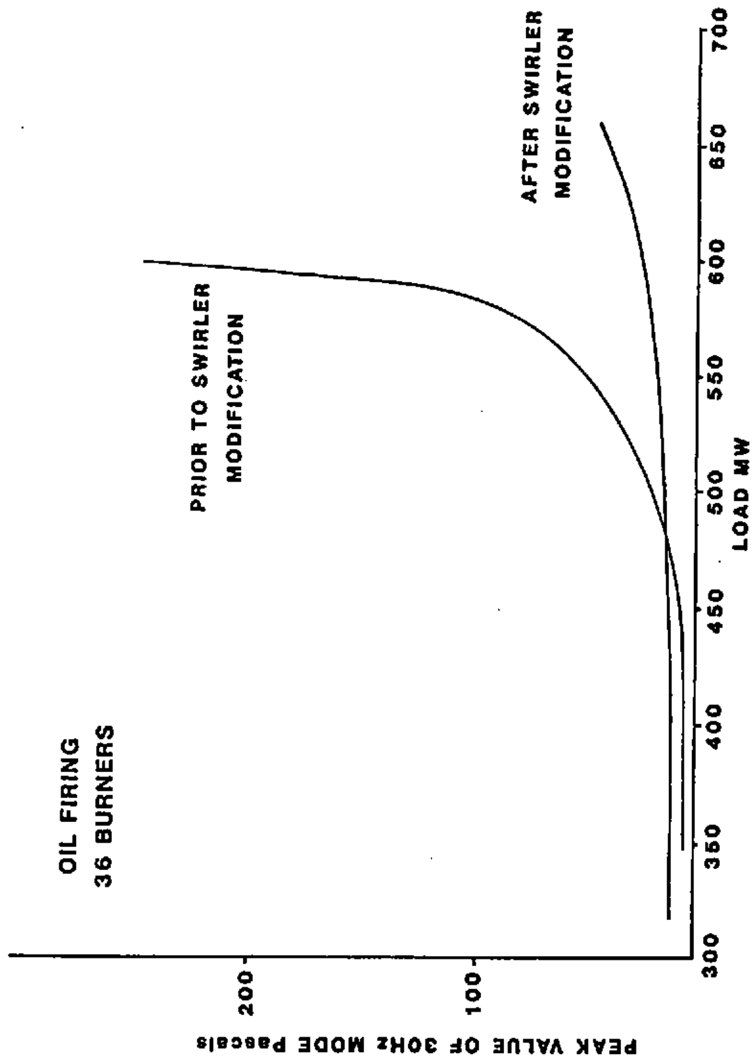


PULSATION CURVE (Gas Firing) FIG. 4



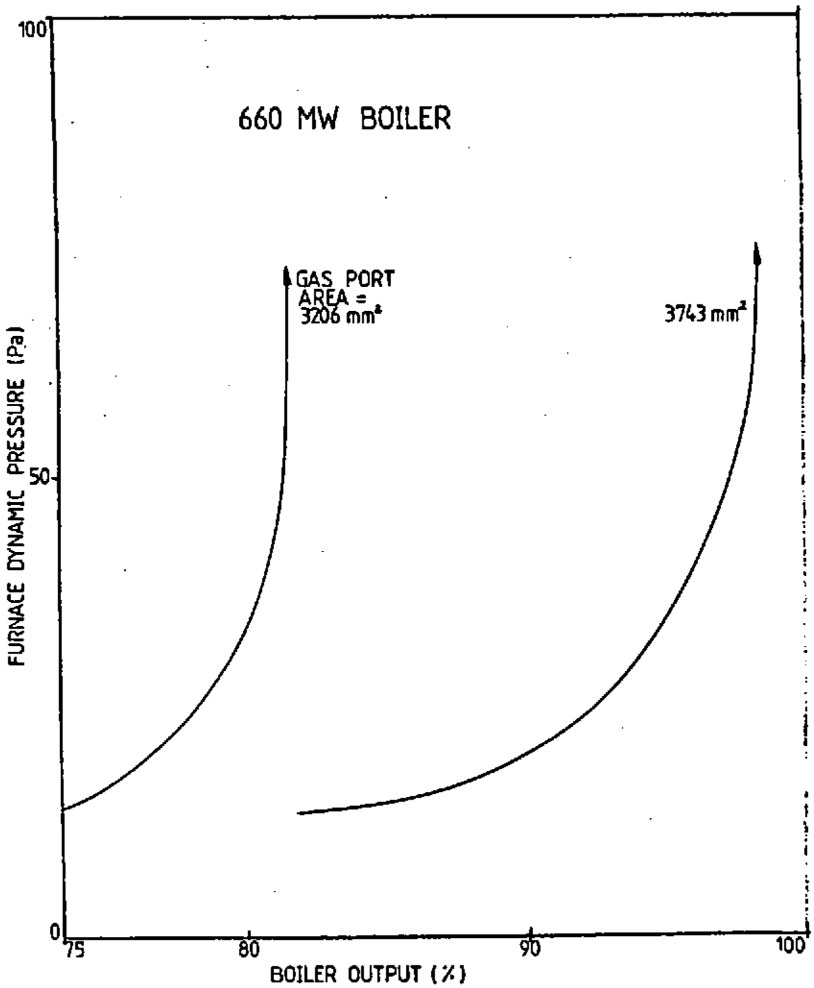
STROUHAL RELATED INSTABILITIES - FURNACE FREQUENCY v AIR VELOCITIES

FIG. 5



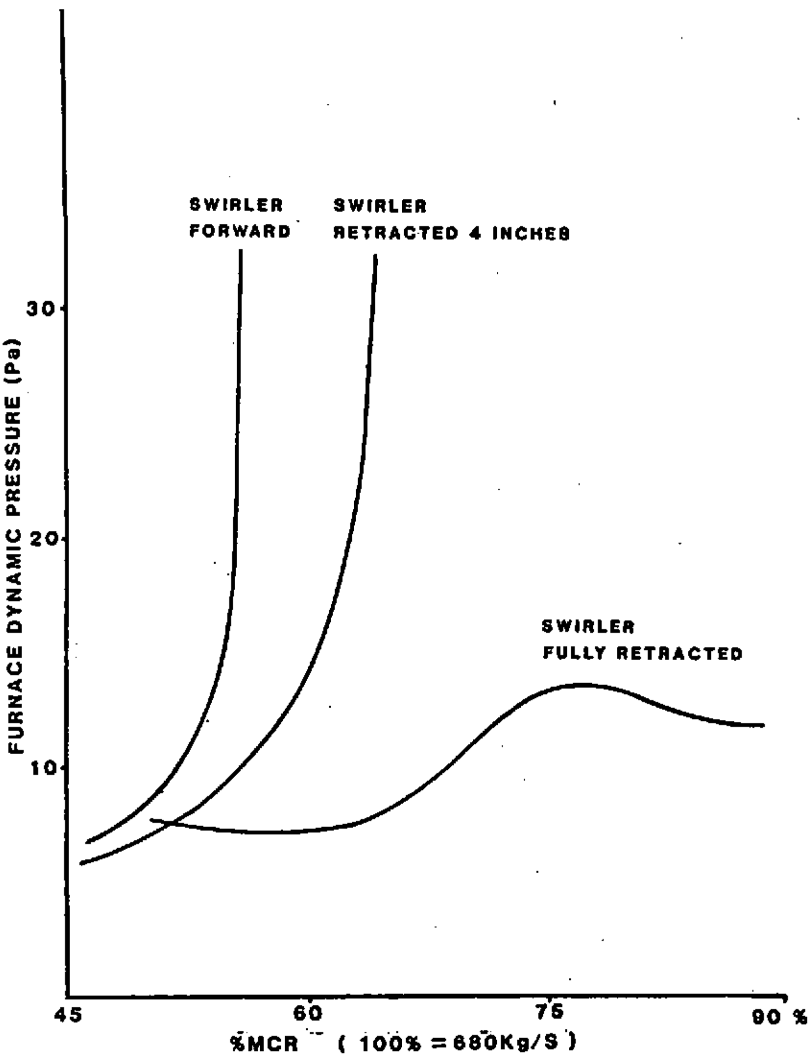
EFFECT OF SWIRLER MODIFICATIONS —
I.E. REMOVAL OF EVERY SECOND BLADE

FIG. 6

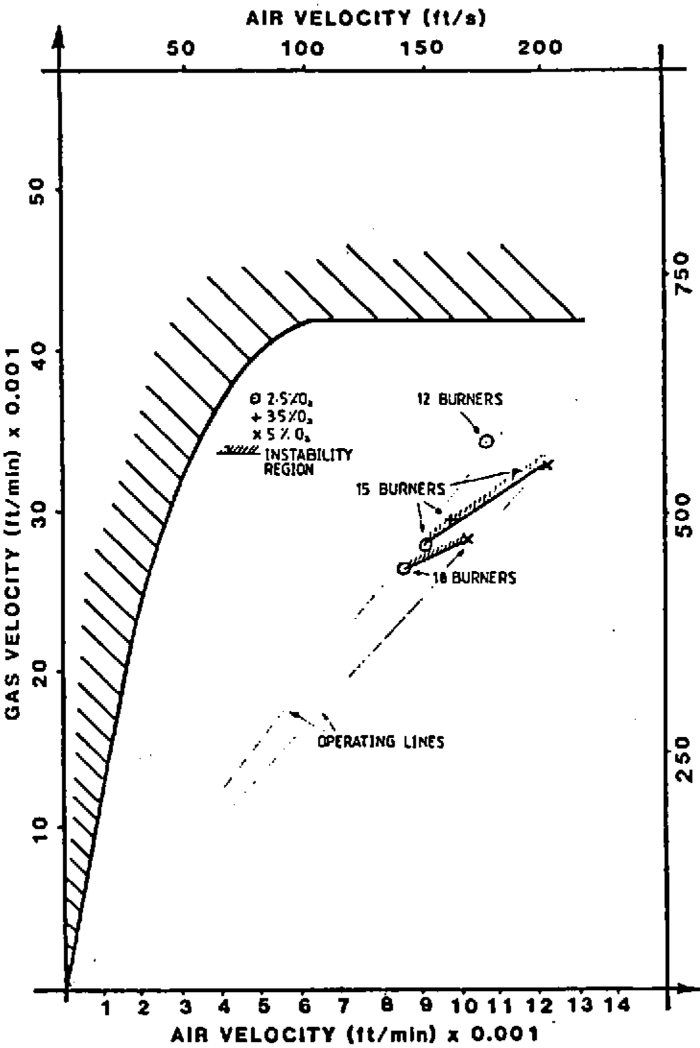


EFFECT OF INCREASING GAS PORT AREA FIG. 7

EFFECT OF SWIRLER RETRACTION

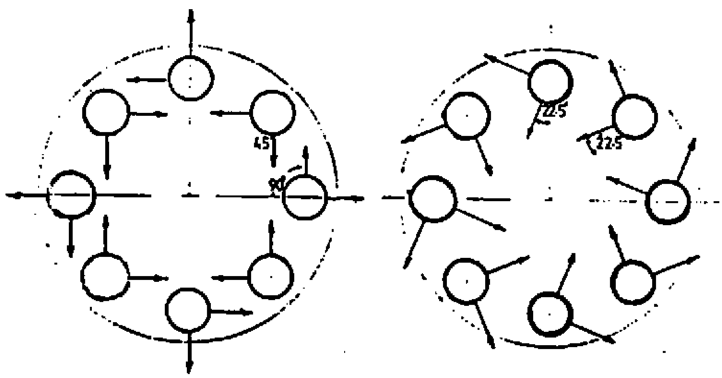


AIR FLOW INSTABILITY FIG. 8



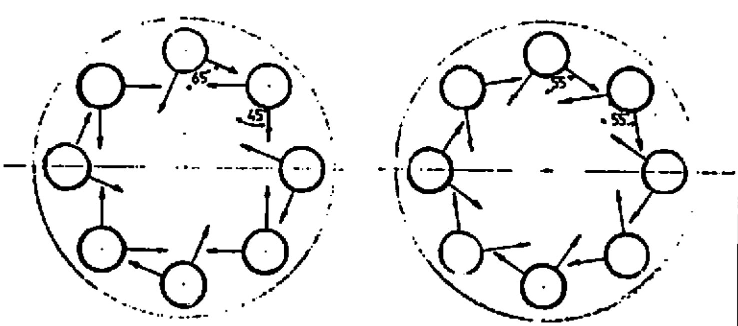
INSTABILITY DUE TO MIXING CHARACTERISTICS

FIG. 9



VIBRATION AND FLAME CHARACTERISTICS DIFFERENT

FIG 10a

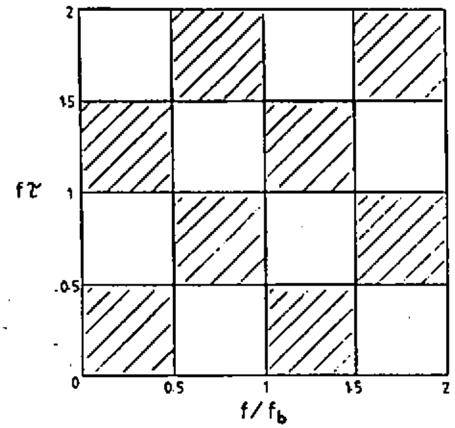


VIBRATION AND FLAME CHARACTERISTICS SIMILAR

FIG 10b

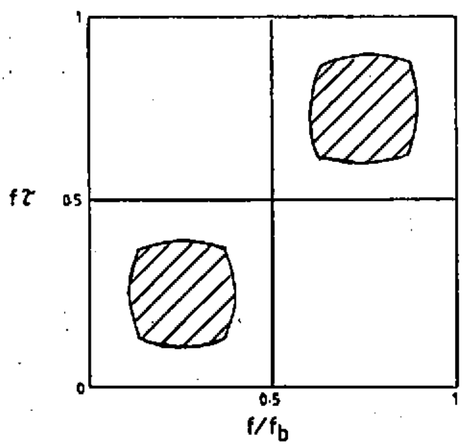
EFFECT OF ALTERNATE GAS SPUD ORIENTATION

FIG. 10'



PULSATION REGIONS WITHOUT DAMPING

FIG 11a



PULSATION REGIONS WITH DAMPING FIG 11b

FIG. 11

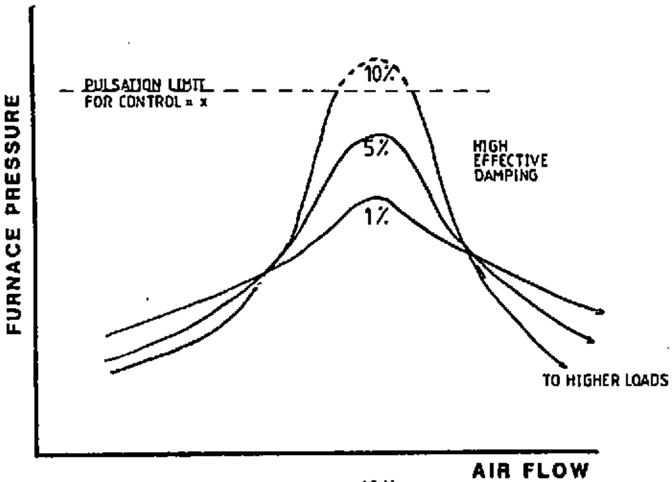


FIG 12a

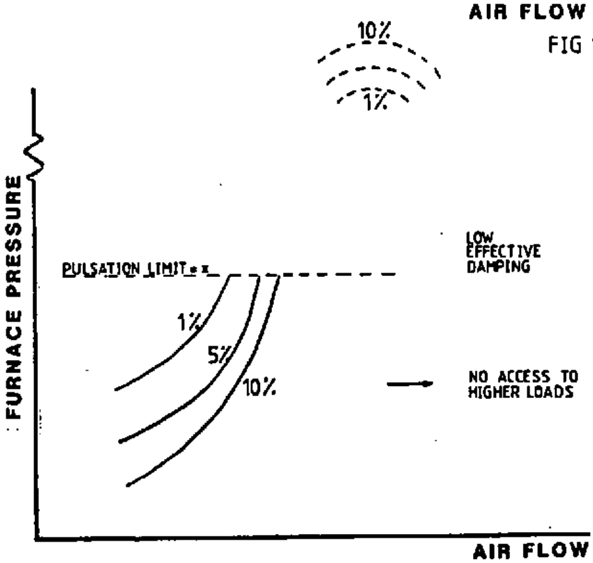
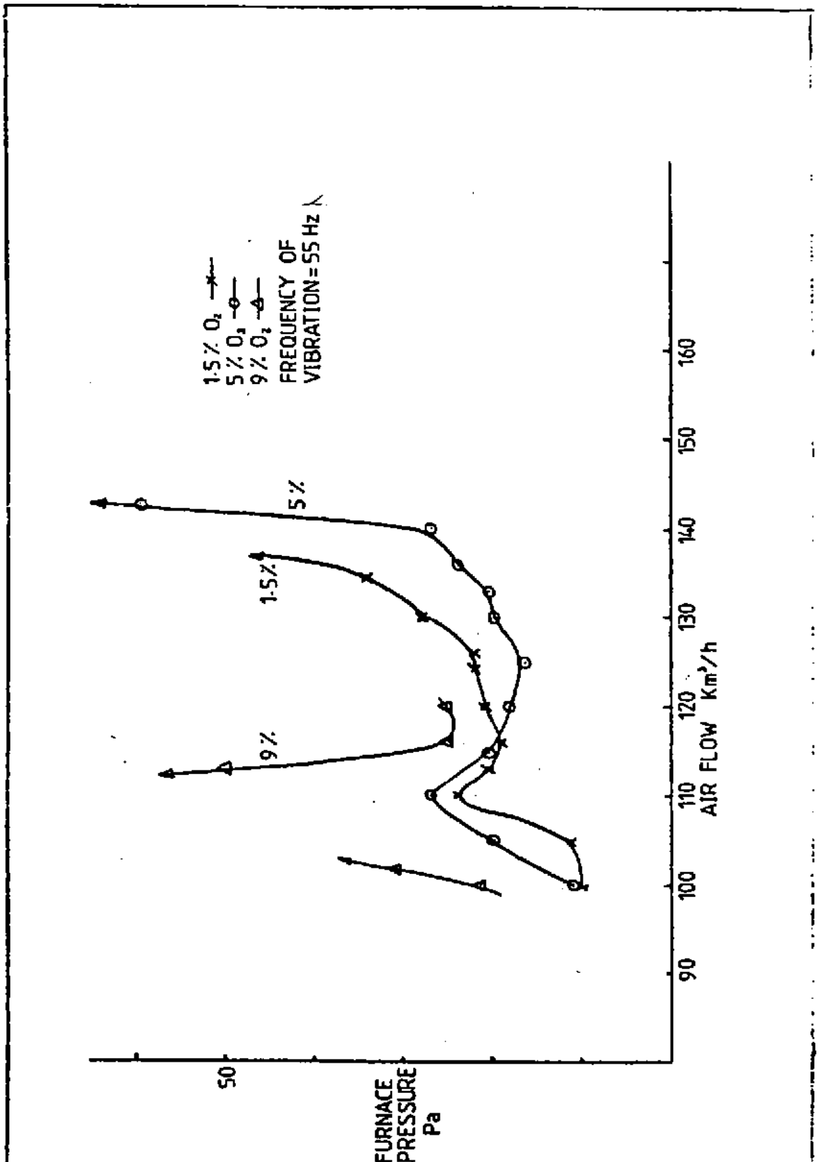


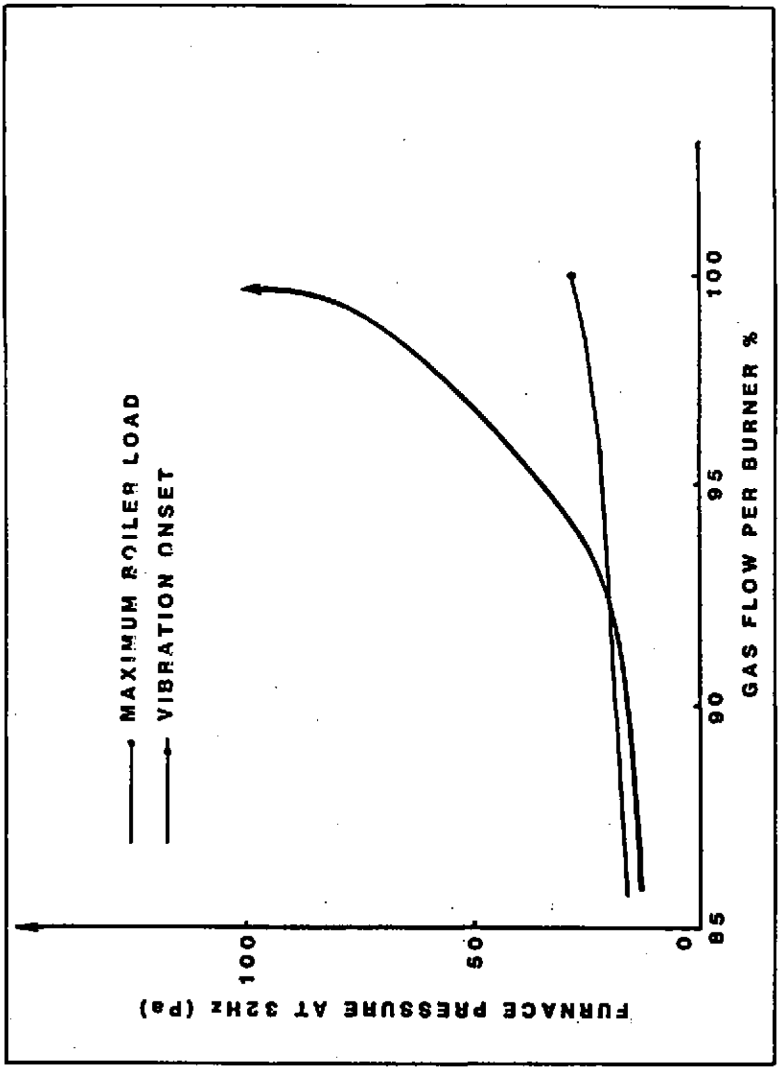
FIG 12b

GENERALISED FURNACE RESPONSE -

VARIATION OF PULSATION LEVEL WITH %O_2



EFFECT OF % O_2 ON VIBRATION LEVELS FIG 13



VARIATION IN EFFECTIVE DAMPING

FIG 14

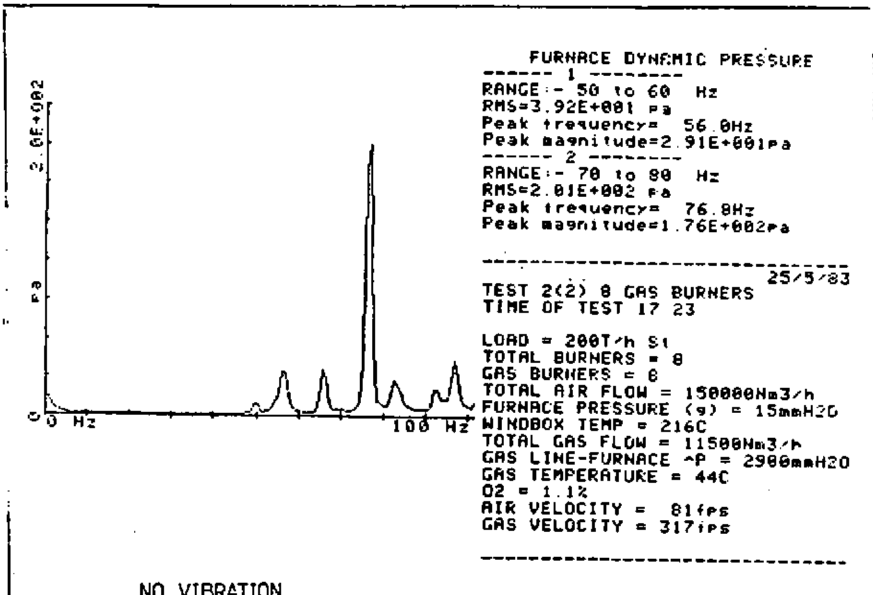


FIG 15a

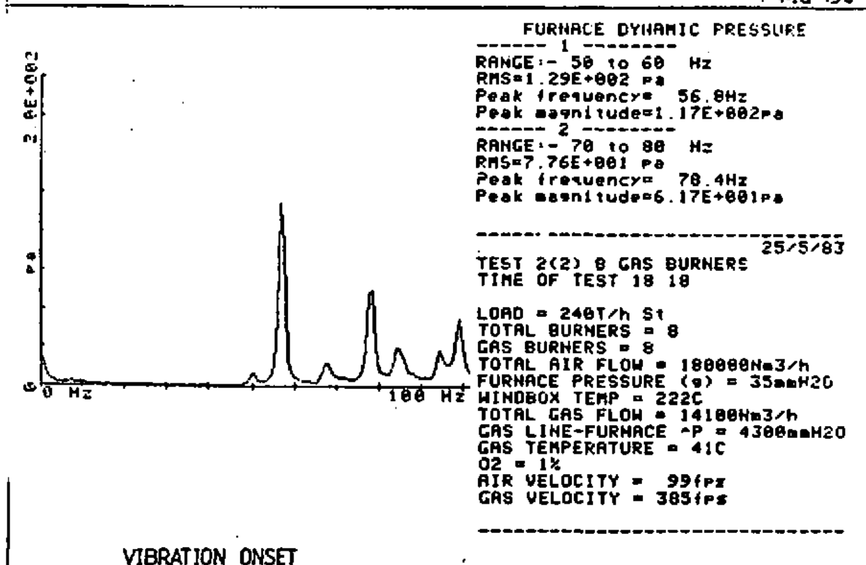
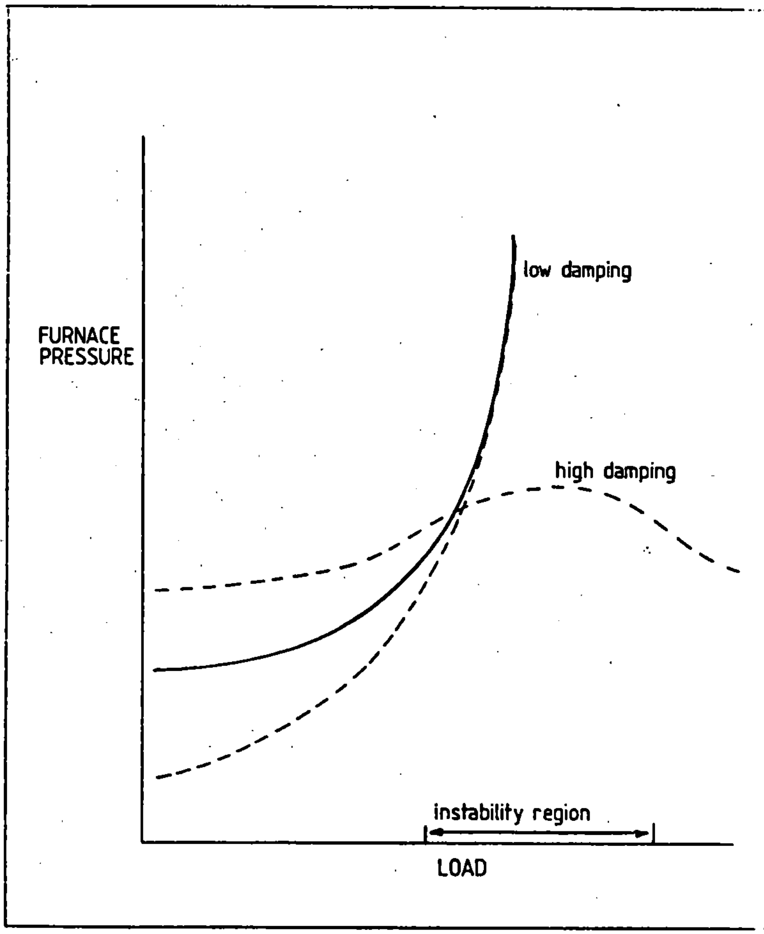


FIG 15b

FURNACE PRESSURE SPECTRA



LOW LOAD PULSATION LEVELS -
POSSIBLE EFFECT OF BURNER MODIFICATIONS

FIG 16