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ELIMINATING AIR-FLOW-INDUCED ACOUSTIC VIBRATION IN COAL PULVERIZERS

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A novel method is presented here for eliminating acoustic vibration within coal pulverizers. The method emerged from the study of an acoustic vibration which developed within the casing of a coal pulverizer at low to medium loads. The vibration was characterized by an acoustic quarterwave driven by swirling air-flow of parallel high velocity air-jets issuing from a rotating multi-air port arrangement. By converting the single frequency excitation energy of the air-jets into turbulent energy, utilizing a new air port design with strongly interacting (colliding) air-jets, the vibration was eliminated. The theoretical background and results of the pre-and post-modification operational tests are presented.

1. INTRODUCTION

In this paper we report a case study of vibration in a coal mill, which was experienced during the initial period of operation. The vibration was caused by acoustic resonance occurring within the casing of the mill. The contributing factor to the acoustic resonance condition was the fact that the mill was operating at a higher speed than other mills in its class, therefore maintaining a lower coal inventory within the grinding section and allowing easier development of acoustic waves (this phenomenon has not been experienced in any standard speed mills). The lower coal bed and the higher speed, in conjunction with other parameter improvements, had made the mill more efficient with regard to performance and power consumption. It turned out, however, that the somewhat lower coal bed could not adequately suppress or dampen the development of the acoustic waves. As will be shown later, the acoustic waves have a tendency to develop due to strong resonant conditions which typically exist within the mill casing from the air-flow needed for coal grinding, transport and combustion. If not sufficiently suppressed, the acoustic waves develop, leading to acoustic resonance and vibration, conditions which cannot be tolerated. Following the analysis of the problem, a novel design improvement eliminated the vibration.

1.1. General Background

Coal pulverizers or mills grind coal typically from 10 to 50 mm size pieces to provide fine coal dust particles usually less than a micron up to several microns in size (with at least 70% by weight not exceeding 75 μm) for feed to combustion furnaces. The grinding is accomplished by multiple grinding or pulverizing rollers rotating about their own axes and crushing the coal against a rotating table driven by a motor through a speed reducer. The

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rotation of the table induces rotation of the rollers which are pressed downwards either by springs or by hydraulic or pneumatic means towards the rotating table to enhance the coal crushing and pulverizing action. The raw coal feed enters the mill vertically by gravity and the ground pulverized coal is carried from the mill by air entrainment upwards through a classifier section to external burners from combustion. The mill classifier allows the fine enough particles to pass on to the external burners, while the coarser size particles are returned to the mill rotating table for further grinding and size reduction. Figure 1 shows a typical design configuration of the pulverizer.

A substantial air-flow rate is needed to carry the pulverized coal from the mill table upwards through the classifier and to the burners. The air/coal weight ratio needed

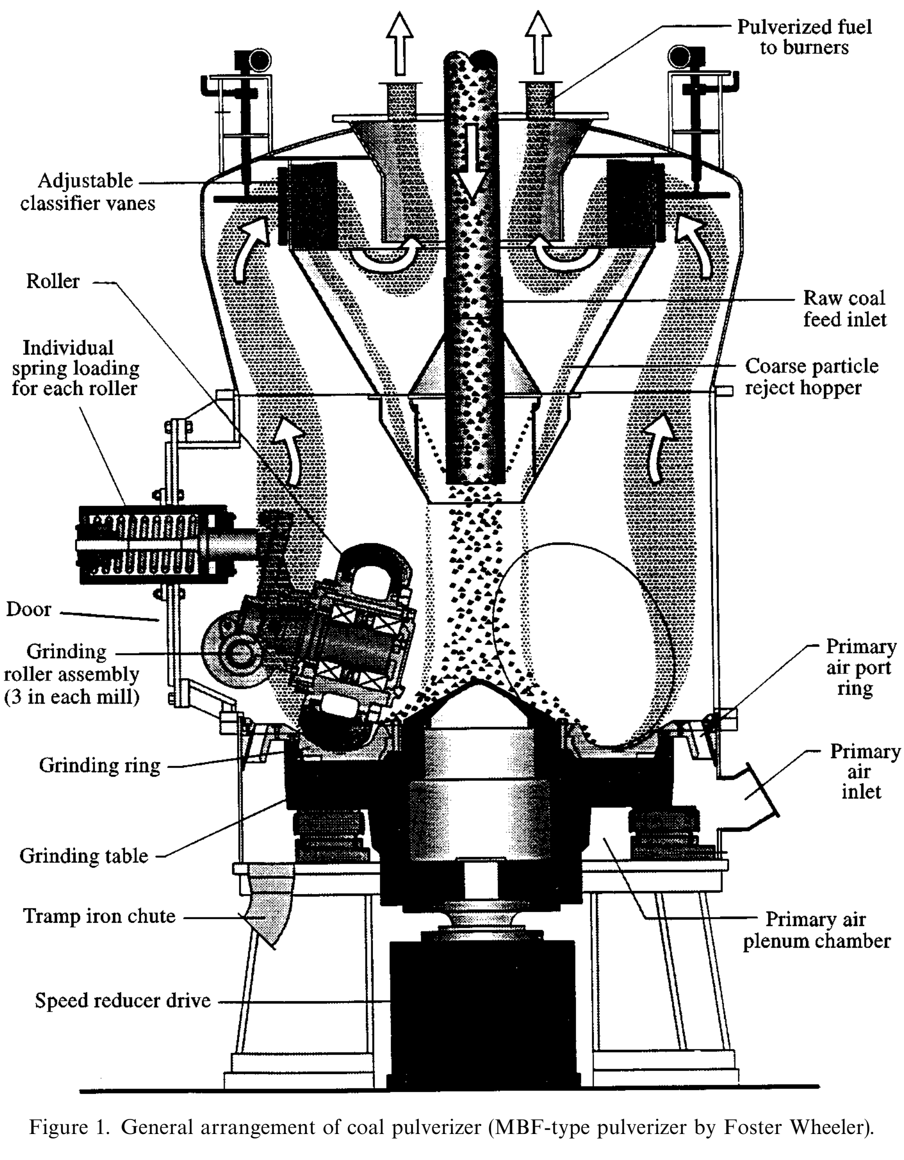

Figure 1. General arrangement of coal pulverizer (MBF-type pulverizer by Foster Wheeler).

for pulverization, coal particle transport and combustion is in the range of 1·7/l to 3·5/l, depending on coal type and flow rate. The air enters the mill plenum located beneath the grinding table, and enters the grinding section through multiple air ports which are typically evenly spaced around the grinding table circumference. Many air ports are used, typically in the range of 16–40, depending on the mill size and the type of coal being ground. The air ports are usually of the rotary type, attached to the rotatable grinding table (as was the case in the mill described here), but can be of stationary (non-rotating) type, attached to the mill casing.

The air exits from the air ports as high-velocity air-jet streams, which are typically provided parallel to each other and have a forward angle relative to the plane of the table which is typically at a 30–45° angle, but other angles may be used. The air-jets generate a swirling action for the entrained coal particles, the orientation of which is preferably in the same direction as the table rotation, for reasons of good performance of the pulverizer.

Pulverizer mill operation experience has shown (see description of the problem later) that the issuing air-jet action can cause undesirable acoustic resonance inside the pulverizer mill housing and structural vibration of the mill. The excitation of the vibration is generated by the air-jet swirling action, and is accompanied by a corresponding pressure pulsation representing a forcing function which excites the acoustic vibration. The acoustic resonance occurs when the excitation frequency generated by the air-jet streams coincides with one of the acoustic (natural) frequencies of the air or air/coal mixture inside the mill. Coincidence of the air jet excitation frequency with the fundamental acoustic or natural (first mode) frequency typically generates the most severe resonance, leading to large acoustic pressures inside the mill housing and resulting in severe structural vibrations of the mill. Coincidence of air-jet excitation with higher natural frequency modes (second, third, etc.) results typically in lower acoustic pressures inside the mill. Such vibration interferes with the normal operation of the pulverizer and may also produce structural damage, and it cannot be tolerated.

The required air-jet velocity in a pulverizer has lower limits, because a minimum air velocity is needed to entrain the coal upwardly from the rotary table and prevent it from falling back through the air port openings into the air plenum. This minimum air-jet velocity is a function of the coal particle size and weight. For a coal particle size of about 50 mm and air temperature of about 230°C, the minimum required jet velocity is approximately 46 m/s, which prevents the coal particles from falling back down through the air ports. For the reasons explained above, lowering the air-jet velocities in order to avoid acoustic resonance vibrations becomes impractical. To avoid acoustic resonance conditions, increasing the air-jet velocity in conjunction with a reduction in the jet stream angle remains the only viable option. However, there are two problems with increasing the air-jet velocity to avoid resonance within the entire operating range of the mill. The air-jet velocities must be quite high (in the 90–120 m/s range) for avoiding acoustic resonance within the entire range of air velocities and coal particle flows, and such high velocities may detrimentally affect mill performance and increase mill erosion. Also, such high air-jet velocities would generate an undersirable pressure drop across a pulverizer, thereby reducing mill and fan efficiency.

Even if frequency separation between the excitation and acoustic frequencies is achieved by changing mill coal flow and thereby changing air-flow for optimum mill performance, the frequency separation may be reduced to the point that the pulverizer would become sensitive to acoustic resonance. If the frequency separation becomes insufficient, the mill may commence vibrating. Once a mill starts vibrating, it will continue to vibrate through a large range of air-flow velocities due to the well-known lock-in phenomenon. Only a significant change in air-flow and/or coal flow will interrupt the mill vibratory condition. Thus, it can be seen that the solution to the acoustic resonance vibrations by way of separation of frequencies is not a desirable or viable solution in most cases; hence, other remedies have been sought as described later.

1.2. Brief Review of Published Work on Swirling Flow

The swirling air-flow which is needed for good mill performance is capable of becoming harmful at certain conditions when its frequency composition coincides with the frequencies of the mill acoustic spaces. There is ample evidence in the literature that swirling air-flows are generally a source of pressure pulsations. The swirling air effects were studied primarily in connection with burners where air swirl is used for flame stabilization. Based on experiments on swirling flows in tubes (Vonnegut 1954), it was found that as the fluid approaches the exit plane it becomes unstable, and periodic fluctuations of motion are produced, generating sound whose frequency is proportional to the flow rate in a manner reminiscent of vortex shedding. There is experimental evidence (Chanaud 1965), that, in certain swirling flows, a spiral vortex-induced precession velocity is superimposed. The frequency of this vortex (called the precessing vortex core-PVC) increases with flow rate. This precessing velocity may lead to longitudinal oscillations, which may cause resonance instabilities in swirl-stabilized burners and excite furnaces, due to the periodic fluctuations of velocity and pressure emanating from the burner (Syred *et al.* 1973).

Additional evidence of oscillations generated by swirling flow can be found for example in publication by Hosoi (1973), Chen (1979), Escudier (1979) and Hayama *et al.* (1990). [See also Eisinger (1991) for further references.]

2. DESCRIPTION OF VIBRATION PROBLEM

The mill operated very smoothly at high loads, in excess of about 70% of its rated full load coal flow capacity of 64 500 kg/h. At lower loads, vibration started suddenly and reached very high levels and typically became stronger. By raising the load to the previous non-vibratory condition, the vibration disappeared. There was a transition region in the load range of about 50–70% within which vibration would either start or cease to exist, depending upon the type of coal being ground. Here the coal moisture content played a significant role. The higher the moisture content, the higher the load at which the mill would start vibrating, and *vice versa*. Higher moisture coal needed higher air temperatures for drying, which at the same air/coal ratio resulted in higher air-flow velocities. This was the first indication that the vibration might be air-flow or air-flow-velocity-induced.

Figure 2 shows typical structural vibration acceleration spectra taken on the mill casing in the horizontal (lateral) direction at the elevation of the spring plungers (see Figure 1). Results are shown for three conditions: (i) nonvibratory at high load ($\sim$ 70% load); (ii) vibratory condition ($\sim$ 40% load); and (iii) severe vibratory condition ($\sim$ 30% load).

As can be seen from the data, there are single frequency peaks in the frequency range of 19–25 Hz present in the “nonvibratory” and also in the “vibratory” spectra. The amplitude of these peaks grows from a value of 3×10^{-3} g (r.m.s.) in the nonvibratory condition, to 76×10^{-3} g (r.m.s.) and 132×10^{-3} g (r.m.s.) in the vibratory and the severe vibratory condition, respectively; here g is used as a unit and is equal to the acceleration due to

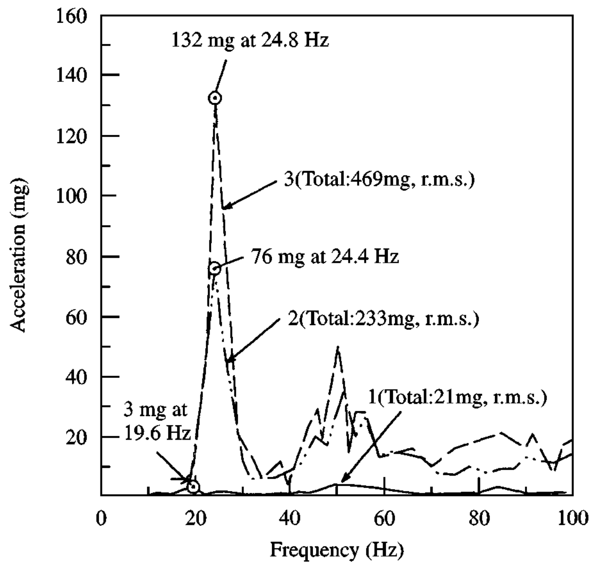

Figure 2. Vibration acceleration spectra taken at mill casing at door of spring plunger: 1, no vibration at ~ 70% load; 2, vibration at ~ 40% load; 3, severe vibration at ~ 30% load; (mill original configuration).

gravity. The total r.m.s. values (the square root of the sum of the squares of individual r.m.s. values) measured in the frequency range of up to 100 Hz were 21×10^{-3} g (r.m.s.) for the nonvibratory with 233×10^{-3} g (r.m.s.) at the vibratory and 469×10^{-3} g (r.m.s.) at the severe vibratory condition, respectively.

Figure 3 shows a typical vibration amplitude versus load relationship showing the nonvibratory and the vibratory load ranges. It can be seen that the vibration resembles an instability-like condition in which a sudden and steep increase in vibration levels occurs, once the vibration threshold has been reached. It is clear that the vibration problem is dominated by the presence of the 24–25 Hz peaks, related to the internal mill loading and resulting in significant structural vibration of the mill casing. We will further study this problem in the next section.

3. ANALYSIS

3.1. Acoustic Frequency of Mill Enclosure

The mill internal space within the mill grinding section between the table and the classifier has been modeled by a finite element acoustic model. The results for the first two modes are shown in Figure 4. The results show that the internal space acoustic properties can be represented by an acoustic quarterwave with the maximum acoustic pressure at the table and zero pressure at the upper elevation of the classifier vane opening. The frequency of this fundamental wave was computed to be 24·7 Hz for a typical air temperature distribution. This frequency may be reduced, due to the reduction of sound speed by the suspended coal particles inside the mill. It can be seen that the computed acoustic frequency correlates well with the measured frequencies of 24–25 Hz associated with the narrow-band peaks in Figure 2. The effect of the second-mode frequency of 52·2 Hz

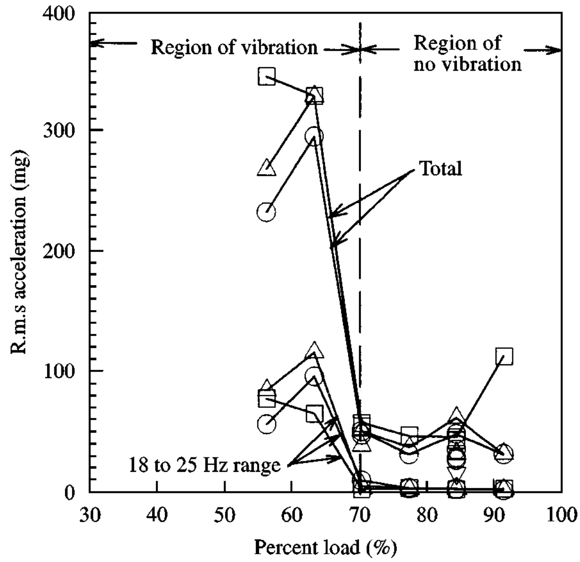

Figure 3. Typical vibration versus load relationship showing regions of no vibration and vibration. Acceleration readings taken at mill casing at three locations at door of spring plunger; (mill original configuration).

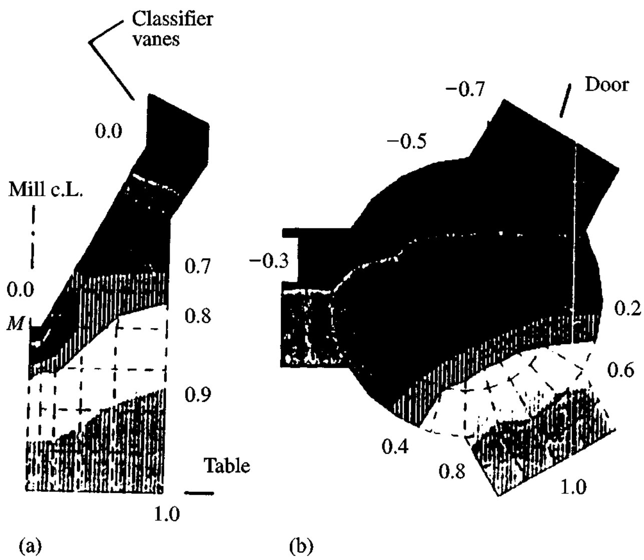

Figure 4. Acoustic pressure modes inside mill casing in mill grinding zone: (a) fundamental quarterwave vertical mode at frequency of 24·7 Hz; (b) second horizontal (transverse) mode at frequency of 52·2 Hz.

can also be seen in the vibration spectra in Figure 2, where a wide-band peak develops in this frequency range.

3.2. Effects of Swirling Air-Flow

The air enters the grinding section of the mill through 24 circumferentially equi-spaced rotating air ports which are directly connected to the table. The mean diameter of the air

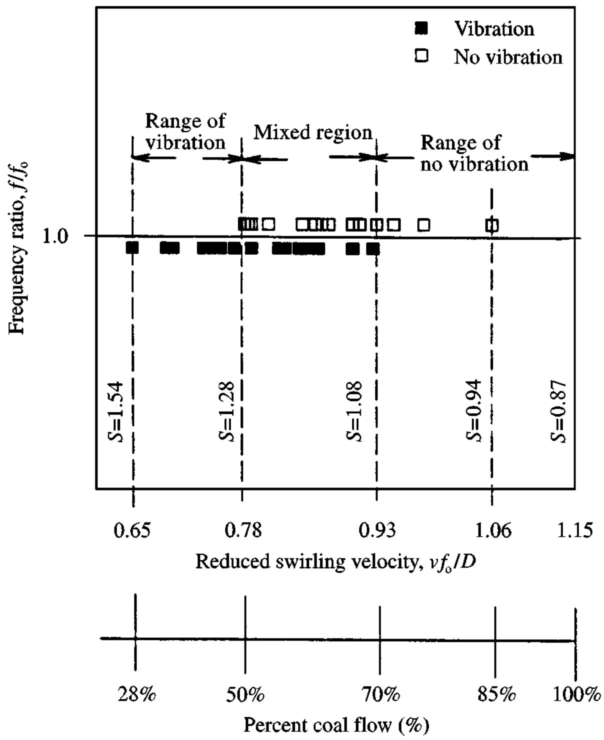

Figure 5. Map showing cases of vibration and no vibration versus coal flow and reduced swirling velocity. Interpretation of test results based on coincidence of swirl-induced excitation frequency f with fundamental quarterwave acoustic frequency f_0; (mill original configuration).

port openings is 2·79 m. The air-jets exiting the air ports are parallel to each other, are oriented tangentially and have a forward angle of 45° relative to the plane of the table. The orientation of the swirling action of the air jets is in the direction of the table rotation for reasons of good performance.

Figure 5 shows a map of vibration test results obtained during mill operation by grinding three different coals with coal moisture in the range of 10–30%. The data in Figure 5 are presented on the basis that resonance exists between the swirling flow excitation frequency f and the standing quarterware frequency f_0. The reduced velocity of the swirling flow is given by

$$v_r = vf_0/D, \tag{1}$$

where v_r is the reduced (dimensionless) flow velocity of swirling flow, f_0 is the fundamental frequency of the quarterwave standing wave in Hz, D is the mean diameter of air-jets exiting the air ports with $D = 2{\cdot}79$ m and v is the tangential air-jet velocity in m/s.

The tangential air-jet velocity representing the swirling (rotating) portion of the air-flow is given by the expression

$$v = v_{\text{jet}} \cos \alpha + \tfrac{1}{2}\Omega D, \tag{2}$$

where v_{jet} is the average air-jet velocity leaving the air ports, α is the forward angle between air-jet and table ($\alpha = 45°$), and Ω is the angular velocity of table rotation ($\Omega = 3{\cdot}27$ rad/s).

The excitation frequency of the swirling flow can be expressed by the linear relationship

$$f = S\frac{v}{D}, \tag{3}$$

where f is the excitation frequency of the swirling flow in Hz and S the Strouhal number.

From Figure 5 it can be seen that the mill did not vibrate at high reduced velocities ($v_r > 0{\cdot}93$). There was a mixed region ($0{\cdot}78 < v_r < 0{\cdot}93$) with vibration occurring or not occurring, depending upon the coal type and load. A clearly vibrating region in the low load and low reduced velocity range can be seen ($v_r < 0{\cdot}78$).

The experimental results obtained are consistent with published results on swirling flow, where it is known that there is a maximum reduced velocity (or Rossby number) above which no periodic motion will occur (Chanaud 1965). On the other hand, at low Rossby numbers or higher Strouhal numbers ($S > 1$), acoustic vibration is likely.

From these results it can be seen that, in order to avoid acoustic resonance conditions, high swirling air flow velocities are needed. These can be partially obtained by reducing the air-jet angle α; however, only by increasing the air-jet velocities v_{jet} substantially could one avoid the vibratory region completely. This solution is impractical, however, as it would lead to high air-jet velocities in the low-load and medium-load regions, and to very high velocities at high loads. Such high velocities are not practically possible for reasons of erosion, and also high power consumption due to increased flow and pressure drop through the system. We shall examine alternative solutions in the next sections.

4. PROBLEM SOLUTION

As discussed in the preceding sections, the mill performance requires a substantial quantity of air to carry the pulverized coal from the mill table upwards though the classifier and to the burners.

Considering the air/coal weight ratios in the range of about 3·5/l and 1·7/l at low and high loads, respectively, quite high air-flows are needed at low loads and proportionately lower at high loads. There is a lower limit jet velocity needed at the air port exit to prevent coal particles escaping from the grinding section through the air ports to the air plenum. As discussed earlier, air velocity of about 46 m/s is needed to keep the coal in the grinding section. At increased loads the air velocities increase according to the air/coal ratio required for coal transport and combustion.

Once the minimum air-flow jet velocity is met, based on experimental evidence, good mill performance does not require high air jet velocities inside the mill grinding section. What is needed is sufficient air flow and only a relatively mild swirling action for good mixing. The harmful effect of the high air swirl generating acoustic resonance and vibration can thus be eliminated without affecting mill performance.

4.1. Mutually Interacting Air-Flow Jets

An innovative solution of utilizing interactive air ports (Eisinger 1997) is described here, which suppresses the vibratory excitation generated by the air-jets and thus eliminates acoustic resonance in the entire air flow and coal flow range of the mill.

The air ports and the issuing air-jet directions (angles) are designed such that interaction between the air-jets is encouraged to the extreme condition of direct collision of the jets.

Interaction or collision of a pair of jets, or interaction and collision of a number or of all of the jets can be achieved by a special design of the air ports. The interaction and collision of the air jets leads to a partial or a full break-up of some or all of the air jets. As a result of this interaction, the following beneficial effects occur.

1. The total energy of the swirling flow is reduced, and thus less energy is available to drive the acoustic vibration.
2. The interaction (collision) of the air jets is accompanied by a significant amount of turbulence which has a frequency spectrum resembling that of "white noise". In other words, a "flat" frequency-independent turbulence spectrum is generated by the jet interaction. The single frequency peaks or the single frequency pressure amplitudes generated by the jet swirling action, which originally were driving the acoustic resonance, are now surrounded by or submerged in the "sea" of the turbulent spectrum. The superimposed turbulence thus reduces substantially or eliminates fully the effectiveness of the single frequency peaks, suppressing or damping the excitation.
3. The interaction points or areas of air-jet interactions are selected in reasonably close vicinity to the air-jet from the air ports and sufficiently far away from the mill internals and mill casing. This will minimize erosion of mill parts from the air-jets and entrained coal particles.
4. The interaction between the air-jets does not affect the air velocity within the air port openings. As explained before, a certain air velocity within the air port openings is required for preventing the unground coal particles from falling through.

4.2. Experimental Test with Colliding Air-jets

An experimental study of the acoustic emissions of two parallel air-jets and of two strongly interacting colliding air-jets with an included angle of 22·5° was performed to evaluate the effects qualitatively. Figure 6 shows the acoustic spectrum for two air-jets in parallel, and

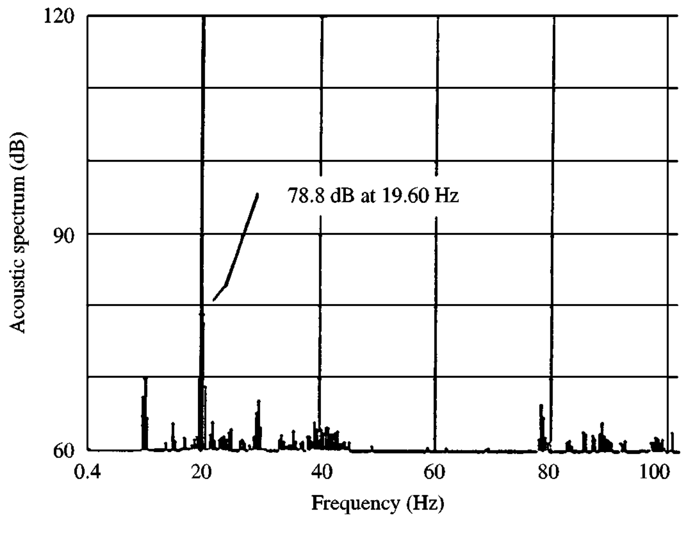

Figure 6. Acoustic spectrum of two parallel non-interacting air-jets issuing from 25 mm diameter tubes; obtained with 12 averages. Measured at 1 m distance in free space. Reynolds number Re = 100 000.

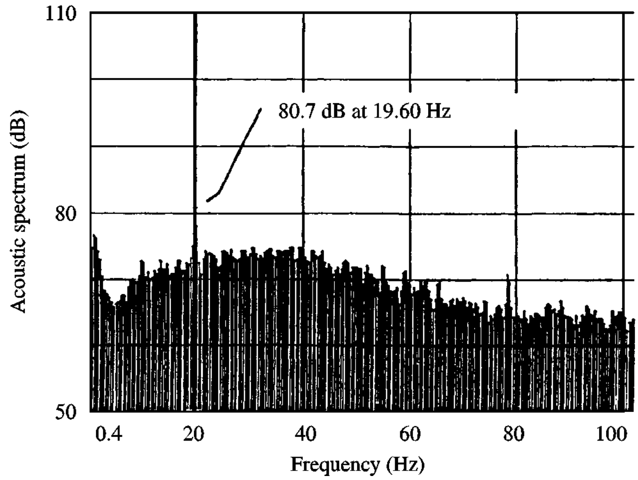

Figure 7. Acoustic spectrum of two intersecting (colliding) air-jets issuing from 25 mm diameter tubes; obtained with 30 averages. Measured at 1 m distance in free space. Reynolds number Re = 100 000.

Figure 7 shows the spectrum obtained when the two air-jets collide. When the two jets are parallel to each other, a single frequency peak at 19·6 Hz is seen to dominate the spectrum (Figure 6). This peak was generated by the air compressor used in the test set-up and its magnitude reached 78·8 dB (r.m.s.) with a total in the 100 Hz range of 84·7 dB.

When the air-jets collide, a significant portion of the jet flow energy changes to turbulence, such that the single frequency peak is virtually totally submerged within the "sea" of the turbulent spectrum with the peak rising slightly to 80·7 dB (r.m.s.) and the total increasing to 93·5 dB (r.m.s.). This experiment confirms that the strong swirling action of the air-jets issuing from the air ports can be substantially reduced, and thereby acoustic resonance of the mill suppressed by utilizing the mutually interacting air-flow jets.

Figure 8 shows several, noninteracting, mildly interacting and strongly interacting air port air-jets arrangements schematically. All of these are shown to have the general direction of the reduced swirling flow in the direction of table rotation.

4.3. Intersecting Air-Flow Jets in Full-Scale Coal Mill

At first, the air ports of every other of the air port openings were equipped with a nozzle attachment to redirect the air-flow and cause it to collide with the neighboring air-jet (Figure 9). The mill was tested throughout the entire load range with various coals, and it was found that the vibration was fully eliminated and the mill operated smoothly with no detrimental effects on performance. Following this successful experiment, the air ports were fully redesigned and the air-jets collision features implemented. Figure 10 shows a plan view and a cross-section of the design configuration of the new interactive air ports installed. Utilizing the new air ports, the acoustic vibration of the coal mill was fully eliminated within the entire range of mill operation.

Figure 11 shows samples of the vibration spectra of the smoothly operating coal mill with the interactive air ports in place. For comparison, the vibratory spectra of the previously vibrating mill are also superimposed. It can be seen that the vibration levels are very small,

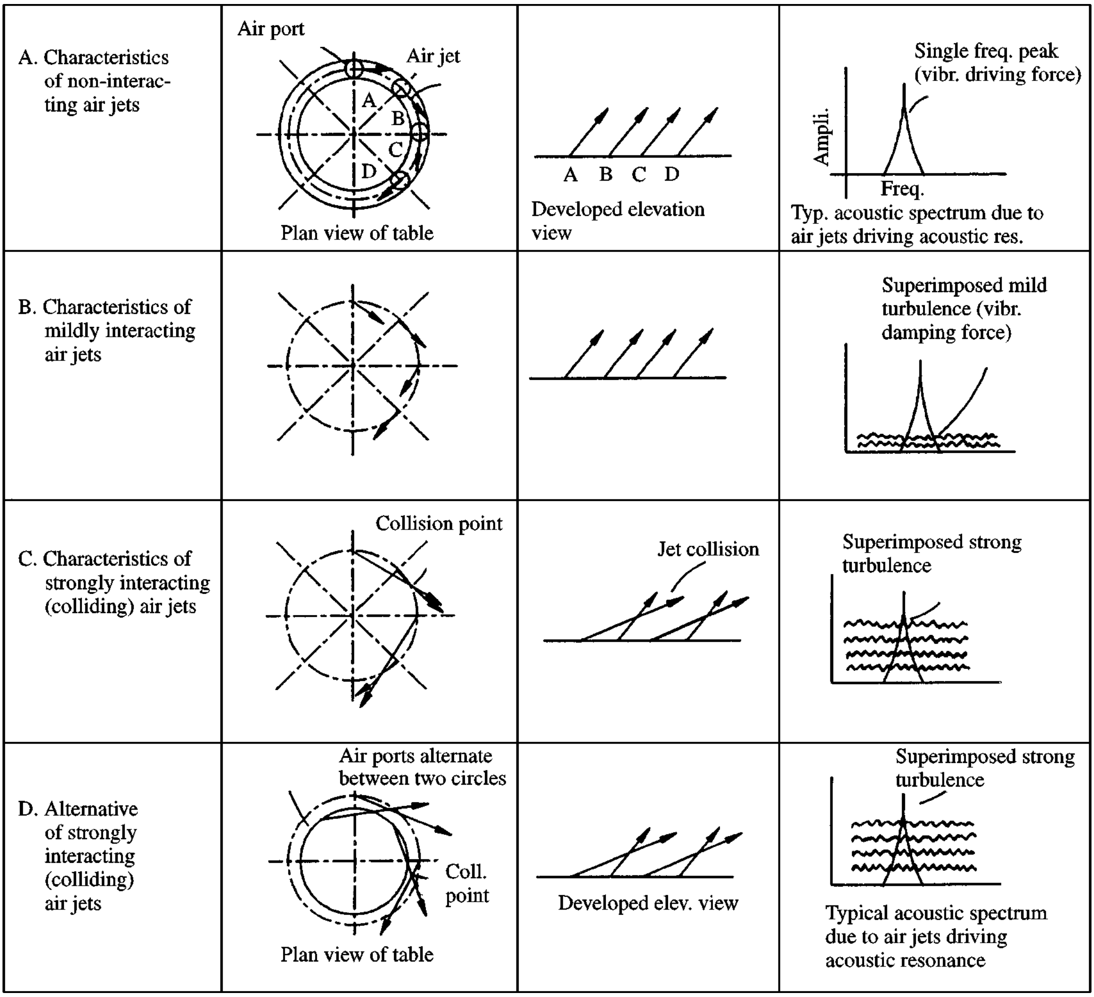

Figure 8. Schematic arrangement and acoustic vibration potential of non-interacting, mildly interacting and strongly interacting air-jets issuing from air ports (sample configurations).

with corresponding maximum vibratory displacements of 8 μm r.m.s. i.e., levels representative of smoothly operating high-speed rotating machinary.

5. SUMMARY AND CONCLUSIONS

Air-flow-induced acoustic vibration of the coal mill occurred in the low-load and medium-load range when the higher-speed mill was equipped with air ports producing high-velocity parallel air-jets oriented tangenitally, with a forward angle of 45° relative to the table. The vibration was characterized by a high amplitude narrow peak at a frequency of 24–25 Hz, depending on the type of coal and load. The analysis showed that the vibration was due to an acoustic resonance condition, in which the excitation frequency generated by the swirling air-flow coincided with the standing (quarterwave) wave natural frequency

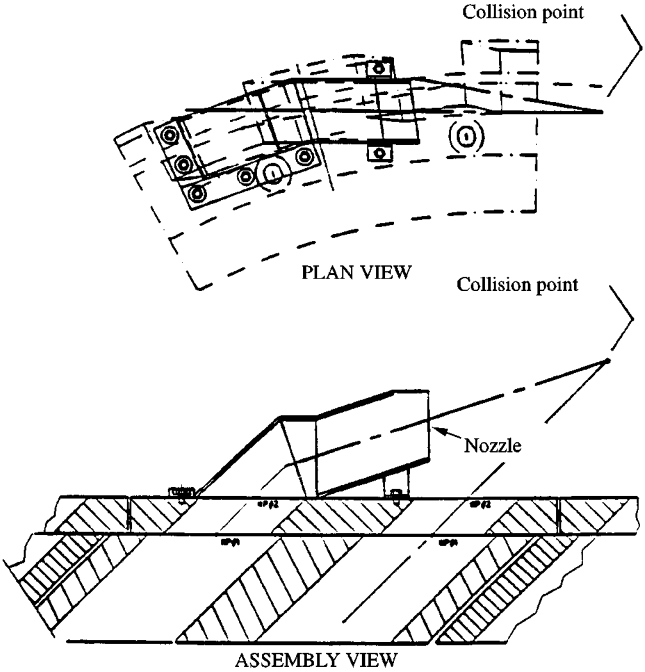

Figure 9. Nozzles attached to air ports of mill redirecting air-flow. Nozzles attached to every other air port to produce collision of air-jets; (test arrangement).

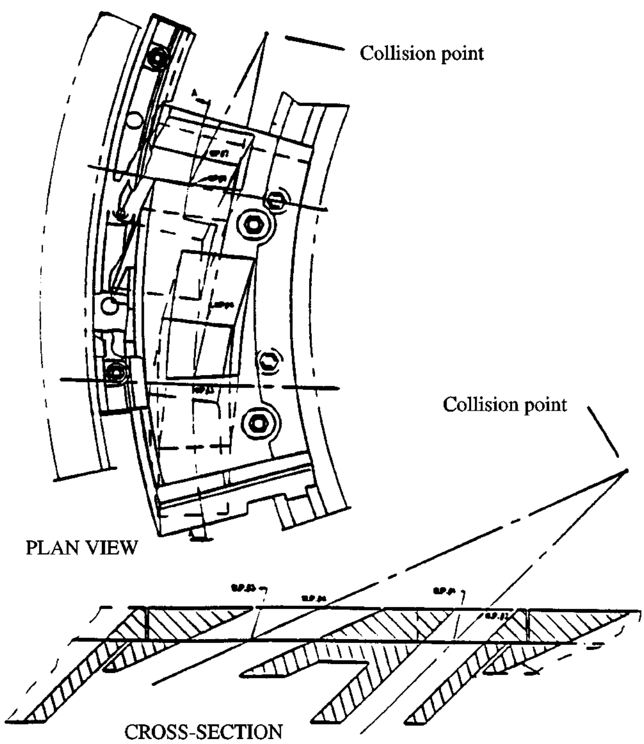

Figure 10. New design of mill air ports with air-jet interaction (collision) features incorporated.

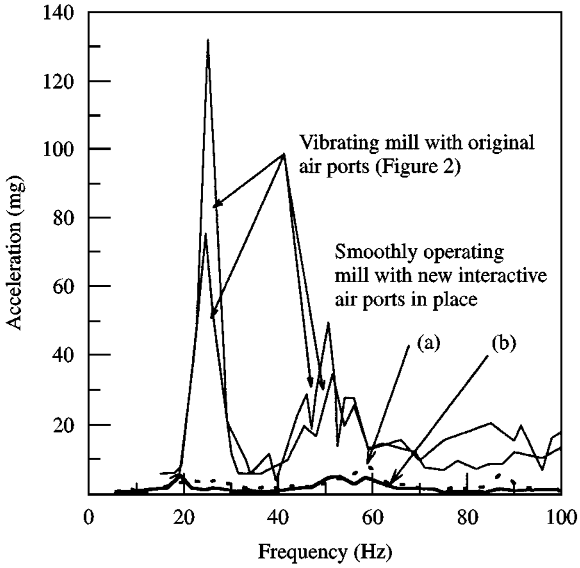

Figure 11. Comparison of mill vibratory and nonvibratory conditions with the original and new interactive air ports in place, respectively. Vibration acceleration spectra taken at door of spring plunger. *Non-vibratory condition*: (a) at 35% load, (b) at 92% load. Maximum vibratory acceleration: 5 mg r.m.s.; maximum total vibratory displacement: 8 μm, r.m.s.

of the internal spaces of the mill. At high loads and thus high reduced swirling velocities (high Rossby numbers), the mill did not vibrate and operated smoothly. At lower reduced velocities, and lower loads, the mill vibrated severely. This vibration behavior of the mill is consistent with published results in the literature on swirling flow-induced oscillations.

The results show that it is generally not feasible to eliminate the resonant condition, as it would require a substantial increase in air-flow velocities which would affect mill performance negatively.

A novel solution of mutually fully interesting (colliding) alternating air-flow jets, having forward angles relative to the table of 22·5° and 45° respectively, was tested successfully. The collision led to a reduction and dampening of the swirling action due to the transformation of the excessive kinetic energy of the jets into turbulence. The remaining swirl following the collision was found to be adequate and, overall, proved to be beneficial for mill performance. The mill equipped with the new air ports operated smoothly without vibration in the entire load range.

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APPENDIX: NOMENCLATURE

D	mean diameter of air-jets exiting air ports (m)
f	excitation frequency of swirling flow (Hz)
f_0	fundamental frequency of the quarterwave standing wave (Hz)
Re	Reynolds number $= vD/\nu$
S	Strouhal number
v	tangential air-jet velocity (m/s)
$v_{\rm jet}$	average air-jet velocity leaving air ports (m/s)
v_r	reduced flow velocity of swirling flow (dimensionless)
α	forward angle between air-jet and table
ν	kinematic viscosity (m^2/s)
Ω	angular velocity of table rotation (rad/s)