Proceedings of The Institute of Acoustics

ROUTINE VIBRATION TESTING IN THE QUALITY ASSURANCE OF INDUSTRIAL FANS

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1.0 INTRODUCTION

1.1 Sources of Vibration

In any roto-dynamic machine vibrationn is manifest. Its magnitude may, or may not, be acceptable. Just as with airborne noise, there will be an interaction between the wishes of the user and the practical possibilities of the manufacturer. Inevitably the 'lubricant' is price. Vibration levels acceptable to a Citröen 2CV driver would undoubtedly be unacceptable to the purchaser of a Rolls Royce. So with the humble industrial ventilation fan. Provided, therefore, that the unit meets any present, or intended legislation, and that it is safe to operate, further reduction in levels may be equated to design margins and manufacturing quality.

It is impossible to completely eliminate vibration from a fan as this arises from the dynamic effects of residual out-of-balance, misalignment, clearances and looseness of adjoining parts, rubbing or rolling contacts, the accumulation of individual part tolerances and so on. In themselves, the vibration from these sources may be small, but they can often excite the resonant frequencies of stationary parts such as casings or bearing pedestals. Where the fan is directly driven by an electric motor, electro-magnetic disturbances will also exist, these too producing further vibration.

1.2 Vibration Measuring Parameters

Vibration is the periodic motion in alternately opposite directions about a reference equilibrium position. The number of complete motion cycles in unit time is the frequency. In the S.I. System, the unit is the hertz (Hz) equivalent to cylces per second.

The motion could consist of a single frequency, as with a tuning fork. With a fan, however, there will inevitably be a number of different motions taking place simultaneously, and each at a different frequency.

There are three properties of a vibration which can be measured. According to the application, each may be of value. Where the motion bears a fixed relationship to the fan rotational speed, as with out-of-balance of the impeller, misalignment of the rotating assembly, gross defects of the bearings etc. the motion is simple harmonic or sinusoidal and the properties are then mathematically inter-related. A description of each is given below together with an equation describing the motion and its relationship to the others:-

a) Displacement, or the size of the movement, is of importance where running clearances have to be maintained for efficient performance, or where contact between stationary and rotating surfaces could take place. Most weight is given to low frequency components.

$$\text{displacement} \quad e = e_{peak} \sin \omega t$$

b) Velocity, or the speed of the movement, is directly proportional to a given energy level. All frequencies are therefore equally weighted. The disturbing effects on people and other equipment are by experience proportional to velocity over a wide range of frequencies.

$$\text{velocity} \quad v = e_{peak} \sin\left(\omega t + \frac{\pi}{2}\right)$$

c) Acceleration, or the rate of change of velocity, is a measure of the forces, and therefore stresses, set up within the fan and motor, or between these and the foundations. Weighted towards the higher frequencies and therefore should be used where such components exist, eg electro-magnetic forces within the motor or bearing surface roughness.

$$\text{acceleration } a = \omega^2 e_{peak} \sin(\omega t + \pi)$$

2.0 THE TECHNICAL PROBLEM

2.1 Which Vibration Level to Measure

As the vibrational displacement, velocity and acceleration all vary with time ie through each cycle or revolution, it is necessary to reduce them to single figures for analysis. The peak-to-peak value indicates the total excursion of the wave and is useful in calculating maximum stress values, or determining mechanical clearances. Probably the most important measure, however, is the root-mean-square (rms) value, as this takes into account the cycle time and gives an amplitude measure proportional to the energy content, and therefore, the destructive capabilities of the vibration. For all vibrations having a frequency directly related to rotational speed, and therefore simple harmonic or sine-wave in character $e_{rms} \times \sqrt{2} = e_{peak}$. The same relationships also hold for velocity and acceleration, and so $v_{rms} \times \sqrt{2} = v_{peak}$. The rms velocity is important as it is used in BS4675 as the measure of vibration severity in the range 600 to 12000 rev/min.

It must be emphasized that the relationship connecting rms and peak values only applies to a sine-wave. Vibrations arising from certain sources, such as rough rolling element bearings or air turbulence, may not follow this form. Acceleration values, especially, may be much higher.

Where sine-wave conditions do exist, by taking time-average measurements, the effects of phase may be ignored and

$$a = e\omega^2 = e \times 4\pi^2 f^2$$
$$v = e\omega = e \times 2\pi f$$

The values of e, v or a are normally given as rms values, but may be peak where applicable.

2.2 Units of Measurement

Where absolute measurements are used, the Systéme Internationale (SI) conventions dictate that the units shown in the table below are appropriate:

Property	Displacement	Velocity	Acceleration	Frequency
Units	m or mm	m/s or mm/s	m/s^2	Hz

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As the range of values of velocity and acceleration can vary enormously, and as modified sound level metering is often used for measuring, it is convenient for these quantities also to be compared on a decibel scale. It should be remembered that the decibel (dB) is the ratio of a measured level to a reference level, and of itself has no dimensions. To convert to absolute levels the reference level must be used. Those recommended in ISO1683 are given in the table -

Property	Definition	Reference Level
Velocity	$L_v = 20\log\frac{v}{v_o}$ VdB	$V_o = 10^{-9}$ m/s
Acceleration	$L_a = 20\log\frac{a}{a_o}$ AdB	$a_o = 10^{-6}$ m/s²

3.0 FAN RESPONSE

The fan and its parts may be likened to a spring-mass system, and this fact is of importance in revealing the causes of resonance. Every fan will have three basic properties: (1) mass 'm', measured in kg - the force due to the mass of the system is an inertial force or a measure of the tendency of the body to remain at rest; (2) damping 'c' is the damping force per unit velocity of a system: it is a measure of the slowing down of the vibrations and is given in N s/mm; and (3) stiffness 'k' is a measure of the force required to deflect a part of the fan unit distance and has the units N/mm.

The combined effects of these restraining forces determine how a fan will respond to a given vibratory force (eg imbalance). Thus, we may state that

$$\frac{md^2e_p}{dt} + \frac{cde_p}{dt} + ke_p = M_u\omega^2 r \sin(\omega t-\phi)$$

$$= M\omega^2 e \sin(\omega t-\phi)$$

or

$$me_p\omega^2 \sin\omega t + ce_p\omega \sin(\omega t + \tfrac{\pi}{2}) + ke_p$$

$$= M_u\omega^2 r \sin(\omega t-\phi) = M\omega^2 e \sin(\omega t-\phi)$$

where e is displacement of centre of gravity from centre of rotation, e_p is displacement of part due to vibratory force, M is mass of rotating parts, M is mass of residual unbalance, r is distance of unbalance from rotating centre and ϕ is phase angle between exciting force and actual vibration, or

Inertial force + damping force + stiffness force = Vibratory force

It will be seen that the three restraining forces are not working together and that the inertial and stiffness forces are 180° out of phase and tending to cancel each other out. At the frequency at which they are equal 'resonance' occurs, and there is only the damping (which is 90° out of phase) to keep the system vibrations down.

All fans together with their supporting bases consist of a number of different spring-mass systems, each with its own natural frequency possible with various degrees of freedom and a different resonant frequency for each. So far we have only considered imbalance as the exciting force, but there will be numerous other sources such that resonance can be a common problem.

4.0 MEASURING THE VIBRATION

4.1 Transducers

How the vibration is measured and what equipment is used is of prime importance. We invariably use piezoelectric accelerometers which consist of a mass rigidly attached to a ceramic element which when compressed or extended produces an electrical charge. The voltage generated is proportional to the force applied and since the mass of the accelerometer is constant, is therefore proportional to acceleration. As acceleration is a function of frequency squared, it is most sensitive to high frequency vibration. For slow speed fans, with significant amounts of low frequency vibration, it is sometimes necessary to use a heavy seismic velocity pick-up.

4.2 Accelerometer Mounting

Bad mounting can drastically reduce the measurable frequency range. Whilst a threaded stud onto a flat machined surface is deal, it is seldom possible and an intermediate block for adhesive fixing is used. It is stuck into position using Loctite. The block is drilled and tapped on three sides such that the accelerometer can be screwed in to obtain vertical, axial and transverse vibration.

4.3 Vibration Metering

There are available today sound level meters which can easily be adapted to the measurement of vibration. An integrator adapter is fitted to the instrument head in place of the microphone. The accelerometer is attached to the integrator via a suitable cable. A switch within the integrator enables velocity or displacement (single or double integration) to be additionally measured.

4.4 Measurement Points

The selection of measuring points is determined by the use to which the information gained will be put. Vibration levels at the terminal flanges of the fan will be of value to the designer of the associated ductwork system, whilst those at the fan feet may be used by the foundation or supporting steelwork installer. Readings taken adjacent to the fan and/or motor bearings, however, may be expected to give the most consistent figures as they will be neither amplified or attenuated by the fan casing with its stiffness or damping effects. They are, therefore, essential in any machine defects analysis. The forthcoming BS848 Part 6 - Method of Testing for Fan Vibration recognises the value of all these readings. It is recommended that as base data for any Machinery Health Monitoring programme all should be obtained.

4.5 The Fan Mounting for Test

By bolting the fan to any ductwork or supporting structure, it has to be recognised that this may materially increase the resultant fan stiffness and/or mass, and therefore reduce the measured vibration levels. As the absolute levels may be low, the fan should ideally be isolated from its surroundings. With a directly driven unit, this is best achieved by suspending on rubber cords with a high static deflection such that the natural frequency of the system is well below any forcing frequencies produced by the fan. Fig 1 shows the general arrangement and the whole is suspended from an 'A' frame.

Where the unit is directly driven through vee ropes from a separate motor, or where the total mass is beyond the rope limits, it is necessary to mount both on a combination baseframe. Provided that the weight of this frame is 10% or less of the total mass, it may be anticipated that results will not be materially affected. Support beneath the mass is then by soft steel springs chosen to give a static deflection of 75 to 125mm according to size. Fig 2 shows a typical test arrangement.

5.0 THE QUALITY STRATEGY

At the instigation of the programme, the first 5 fans of a given type were tested, and acceleration decibel readings taken in the usual octave bands in the three directions at the prescribed accelerometer positions. All these fans were assessed as satisfactory according to the normal subjective inspection then in use. Taking the highest reading in any direction at any measurement position, it was possible to set a Preliminary Level. A tolerance of +2AdB in any two bands was given for the acceptance/rejection of all succeeding units. After 20 units of a given type had been manufactured, the figures were re-assessed and an Acceptance Level set to give an 85% pass rate ie the level was set at the fourth highest reading obtained for all the units in all positions. It should be appreciated that these levels are unique to a particular fan design and speed. Values are given in the table for four typical but diverse types -

Usage	Fan Size/Type	Speed rev/min	Power kW	AdB re $10^{-6}m/s^2$ in each octave band, Hz								
				<45	63	125	250	500	1k	2k	4k	>5.6k
Aux. Mine Ventilation	610mm Inline Radial	2940	18.5	114	115	111	119	115	122	126	135	127
Offshore Oil Rig	800mm Mixed Flow	1180	22.5	102	111	118	129	133	131	130	128	118
Marine Ventilation	180mm Axial Flow	3500	0.3	99	106	110	122	122	137	128	127	124
Co_2 Scrubber	450mm Centrifugal	3500	1.5	100	112	114	128	130	130	128	126	124

Such acceptance levels are constantly under review. Each fan is logged and trends noted. With constant scrutiny and attention to the faults indicated by Condition Diagnosis, it has proved possible to gradually lower the acceptance levels. Certainly, with imiproved balancing machines, the levels in the appropriate band have reduced considerably over the years.

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6.0 CONDITION DIAGNOSIS

6.1 General Comments

For the purposes of maintaining quality, it is necessary to be able to identify the causes of vibration and their likely effects on the fan - which could be catastrophic in the event of a total breakdown. Where the acceptance levels in a particular band are exceeded, remedial work is called for.

The keys to the identification are frequency and velocity over most of the frequency range. Below about 10 Hz displacement will be of primary importance, whereas above about 1 kHz acceleration is paramount. Different causes of vibration occur at different frequencies. For example, a faulty ball-bearing would cause high-frequency vibration at many times the fan rotational frequency, and imbalance or misalignment produces vibration at the rotating-speed frequency. Nevertheless, in all frequency bands, it is possible to use acceleration measurements, and with experience, identify likely causes in the particular octave band. If more than one possibility exists, it is sometimes necessary to carry out a discrete frequency analysis. The various sources of vibration are considered in the succeeding paragraphs. In all cases f_1 is defined as the rotational frequency Hz and equals rev/min ÷ 60.

6.2 Mechanical Problems

Balancing: is the process of improving the distribution of mass in an impeller so that it can rotate in its bearings without producing unbalanced centrifugal forces. Perfection is impossible and even after balancing there will be residual imbalance, its magnitude being dependent on the machinery available and the quality necessary for the application.

The relevant Standard is BS5265. Recommendations are made for various rotor groups to a avoid gross deficiencies or unattainable requirements. If the recommended limits are followed satisfactory running can be expected.

An unbalanced impeller will create forces at its bearings and foundations and the complete fan will vibrate. At any given speed the effects depend on the proportions and mass distribution of the impeller as well as the stiffness of the bearing supports.

In general, the greater the impeller mass, the greater the permissible imbalance. It is therefore possible to relate the residual imbalance U to the impeller mass m. The specific imbalance $e = \frac{U}{m}$ is equivalent to the displacement of the centre of gravity where this coincides with the plane of the static imbalance. Practical experience shows that e varies inversely with the speed N over the range 100 to 30000 rev/min for a given balance quality. It has also been found experimentally that eN = constant.

Quality grades have therefore been established which permit a classification of requirements. For commercial grade fan impellers G6.3 is recommended where $e\omega = 6.3$ mm/s, but for critical fans a balance grade G2.5 is often necessary. For defence applications G1.0 may be called for. It should be noted that the quality grade is essentially a 'peak' velocity.

In a completed fan, the heavy spot would give a 'pulse' to the transducer once every revolution and imbalance would be identified by high readings in the horizontal and vertical directions at the rotational frequency, ie f_1 Hz. This is a common cause of vibration. High readings at commissioning can indicate residual imbalance in manufacture, or 'sag' if the fan has not been rotated regularly during storage.

Reductions of 6 to 20 AdB in the appropriate band have been achieved after rebalancing.

Misalignment: This is almost as common as imbalance, and even with the so-called 'self-aligning' type of bearing it is still necessary to line up as well as possible. A bent shaft produces angular misalignment. Radial and axial forces are always produced; the magnitude of such forces, and therefore, the resulting vibration, is proportional to this misalignment. Axial readings are usually 50% or more of the radial readings and again the frequency is normally f_1Hz. When the misalignment is severe, however, vibration at $2f_1$Hz and even $3f_1$Hz may be experienced. MIsalignment can also occur where a fan has been distorted by tightening down the bearing onto a pedestal which itself is not level. With sleeve bearings this will produce vibration according to the amount of residual imbalance, but with ball- or roller-bearings an axial vibration would be produced even if the impeller were 'perfectly' balanced, which is physically impossible.

Eccentricity: A typical example might be where an impeller centre with excessive bore is pushed over by a taper key. The centre of rotation does not then coincide with the geometric centre. As far as a fan impeller is concerned, this leads to a greater mass on one side of the rotational centre than on the other - that is, imbalance. It can be corrected by rebalancing, provided that the rebalancing takes place in its own shaft and bearings and that with ball/roller-bearings the position of the inner race on the shaft also does not change. The predominant frequency is, of course, f_1Hz.

Looseness: Common forms are excessive bearing clearances or inadequately tightened bolts. Vibration from this source will not occur unless there is some other exciting force, such as imbalance or misalignment, to encourage it; however, only small forces are necessary to excite the looseness and produce large vibrations. Although rebalancing or realignment may help, great accuracy would be necessary, which may be impossible to achieve. To determine the characteristic frequency of looseness, consider an unbalanced impeller fitted to a shaft running in a bearing with loose holding-down bolts. When the heavy spot is downward the bearing will be forced against its pedestal, and when the heavy spot is upward, it will lift the bearing, whereas at positions 90° away the force will neither lift nor hold down, and the bearing will drop against the pedestal because of weight alone. Thus, there are two applied forces each revolution of the shaft, and the vibration frequency is $2f_1$Hz. This is the characteristic frequency of looseness.

Aerodynamic Forces: can excite some part of the fan to vibrate at the resonant frequency. An investigation of the vibration spectrum will usually indicate possible troublesome frequencies. The following have been identified:

a) blade passing frequency = number of blades x f_1Hz
present when the impeller of an axial fan is eccentric to its casing with consequent varying tip gap. By halving the eccentricity, reductions of 3AdB have been made.

b) guide vane frequency = number of guide vanes x f_1Hz.

c) secondary frequencies at harmonics of the above may be present where the disturbance is excessive.

d) support frequency = number of supports × f_1Hz.

e) interactive frequencies will occur at (a + b)Hz (a-b)Hz (b + c)Hz etc. An example has been noted where the gap between an impeller and downstream guide vanes varied by 3mm and led to an excess of 6AdB in the octave band containing (a + b)Hz.

f) where the blade and guide vane numbers are equal, both even integers, both odd integers or differ by an even integer, there is a possibility of beat frequencies.

Vee Belt Drives: Often the balancing of pulleys has been overlooked and must be specified when ordering. Misalignment of the drive can produce severe vibration.

Such drives have good resistance to shock and vibration but may be blamed for causing trouble as they can be readily seen to whip and flutter especially when the belts are unmatched. Belts are often changed unnecessarily when the fault is really that of imbalance, misalignment etc. Nevertheless, the importance of using matched sets of belts cannot be emphasized enough.

Vibration from faults in the belts themselves occur at multiples of belt speed. The relevant frequencies are:

$$1,2\ 3 \text{ or } 4 \times \frac{\text{Pulley diameter}}{\text{Belt length}} \times f_p\text{Hz}$$

$$\text{where } f_p = \frac{\text{Pulley rev/min}}{60}$$

Likely faults are pieces broken off, hard or soft spots etc.

Faults in pulleys, such as chipped grooves etc. will be identified at the speed of the relevant pulley f_pHz.

6.3 Vibration from Electric Motors

Squirrel-cage induction-type electric motors are normally used to drive ventilation fans. Most vibration will be mechanical in origin, and imbalance, misalignment and faulty bearings are just as prevalent in the motor as in the rest of the fan.

The rotor must be concentric with the stator bore, and this requires that the bearing and end-shield location and stator pack tolerances all be closely controlled during manufacture. Bearing housings and end-shields need to be sufficiently rigid to avoid distortion during assembly. If the motor casing is of fabricated construction, stress-relieving is desirable before the final machining operation. The method of attaching the motor to its mounting is of importance. If this is not flat, or if the bolts are not properly tightened, vibrations will be produced.

Major sources of vibration are the various magnetic forces present within the motor. These can never be totally eliminated but can be reduced by careful electrical design. In general terms the greater the size of iron core per kilowatt of output at a given speed, the lower will be the level of magnetic vibration. Other features that have an effect are core material, size and geometry; natural frequency of the core, core to frame fit and core-pack axial pressure; lamination insulation and burr height, number of stator slots, type and fit of stator coils; type and fit of slot wedges, pitch of coils, connexion of coils and coil groups; and impregnation, number of rotor slots, air-gap length and frame stiffness.

Power supplied to a three-phase stator winding sets up a rotating magnetic field. This induces an opposing current in the rotor winding and thus another magnetic field. Interaction of these two fields produces a tangential force. As the rotor shaft is only restrained by its bearings, it has to rotate.

Viewed from a fixed point on the rotor, the air-gap permeance around a rotor with R slots will have R cycles of variation. Similarly, a stator with S slots will produce S cycles of variation. As the power to the stator has a frequency f_1Hz, and as the winding is distributed around the stator in slots, the stator will produce vibrations proportional to field strength squared, related to the supply frequency, winding pitch and number of slots per pole-pitch. Harmonics will also be present and, together with all the interactive frequencies, a very complex situation results. The rotating magnetic field of the stator produces low-frequency vibration, whereas rotor slot permeance variation and its reactions with supply frequency lead to higher frequencies. These may be calculated from

$$(R \times f_1) - 2f_L, \text{Hz} \qquad \text{(a)}$$
$$R \times f_1, \text{Hz} \qquad \text{(b)}$$
$$(R \times f_1) + 2f_L \qquad \text{(c)}$$

where f_L is the line frequency.

When $R > S$, (a) is usually of more importance. If $S > R$, (c) predominates. Again, many harmonics will be present.

At the design stage the stator-rotor slot combination can be chosen to minimize vibration. To achieve this the number of vibration nodes should be as high as possible: number of nodes = $(2R-2S) \pm 2P$, where P is the number of poles.

Forces in the air-gap between rotor and stator tend to pull these together and produce vibration at double the line frequency. Normally, this vibration is small, except in two-pole motors, and if the air-gap varies, or if the tightness of stator laminations or winding in the stator varies. The second and third harmonics may also be important.

In general, slip frequency = $f_1 - f_L$ Hz will not in itself be important, as it will be of very low frequency. Its interaction with higher frequencies can, however, produce pulsations.

If the rotor is severely unbalanced, the high spot will come closer to the stator than other points. As it passes the stator poles a greater pull is exerted and the vibration occurs at double the slip frequency on a two-pole motor. The magnitude of the readings in this frequency, (increases of 10 AdB have been noted), can indicate whether the problem is simply due to the lack of balance, a change in the air-gap, worn journals, broken rotor bars, etc.

If a resonance condition exists within the motor at the line frequency, large vibrations can be produced. More often this is a result of an unbalanced magnetic pull and can be overcome by changing stator connexions.

With suspected electrical sources of vibration a simple check is to switch off the motor, when they should 'die'. This is the opposite to mechanical sources, which will gradually decay with decreasing fan spee.

6.4 Specific Problems of Ball and Roller Bearings

Flaws or incorrect installation: if there are flaws on the balls, rollers or raceways, this will usually cause a high-frequency vibration.

Flaw in outer raceway or vibration in stiffness around housing

$$f_2 = f_1 \times \frac{n}{2}\left(1 - \frac{d}{D}\cos A\right), \text{ Hz}$$

Flaw in inner raceway

$$f_3 = f_1 \times \frac{n}{2}\left(1 + \frac{d}{D}\cos A\right), \text{ Hz}$$

Irregularity in cage or rough spot on ball/roller

$$f_5 = f_1 \times \frac{1}{2}\left(1 - \frac{d}{D}\cos A\right), \text{ Hz}$$

Flaw in ball or roller

$$f_4 = f_1 \times \frac{D}{d}\left(1 - \frac{d^2}{D^2}\cos^2 A\right), \text{ Hz}$$

where n is the number of balls or rollers, d is diameter of balls or rollers, D is pitch circle diameter of race and A is angle of contact of ball/rollers. Such vibrations are not easily transmitted to the rest of the fan and will therefore be recognised by readings on the bearing housing. Severe misalignment of a race will sometimes result in a frequency at $n \times f_1$ Hz, even when the bearing itself is satisfactory.

Quality: Ball and roller bearings from reputable manufacturers are manufactured to a high standard and with correct installation/lubrication are unlikely to cause trouble. Nevertheless, vibrations at frequencies above 2k Hz can vary by as much as 10 AdB. It is therefore advisable to choose the specially selected qualities such as the RHP designation Q9 or the SKF QE6. It should be noted that certain imported bearings have been shown to have considerable vibration in the >5.6k Hz due to roughness of the cages. Bearing problems are more often the fault of imbalance, misalignment or use at speeds/loads/temperatures in excess of those recommended by the manufacturers. With major faults readings can be 15 AdB greater than the reading of a good bearing. Great care should be taken in the selection of shaft and housing limits. An interference fit of the bearing to the shaft and a small clearance between the outer raceway and the bearing housing are preferable.

Axial End Thrust: Electric motors often incorporate 'waved' washers to obtain the best results from the ball bearings used to support the rotor shaft. When inserted in the bearing housing, they exert pressure on the outer race and create a diagonal pressure through the balls to the inner race for a true rolling effect. In recent years higher working temperatures have been accepted and the motor size has been reduced for a given output. This has caused expansion of the motor components, including lineal expansion of the shaft, and extra pressure on the fixed bearing assembly. To relieve this stress the waved washer is not fully compressed at normal temperatures. The bearing end cap is securely clamped to the housing, leaving a machined spigot facing the outer ring of the bearing. The gap size between the spigot face and the outer ring to accommodate the partially-compressed waved washer is critical for optimum results. Inadequate axial pre-loads have increased vibration by up to 6 AdB. The fan impeller tries to pull the rotor out of the stator field and this is resisted by the electromagnetic force. Axial pre-load from a waved washer reduces the oscillation otherwise present.

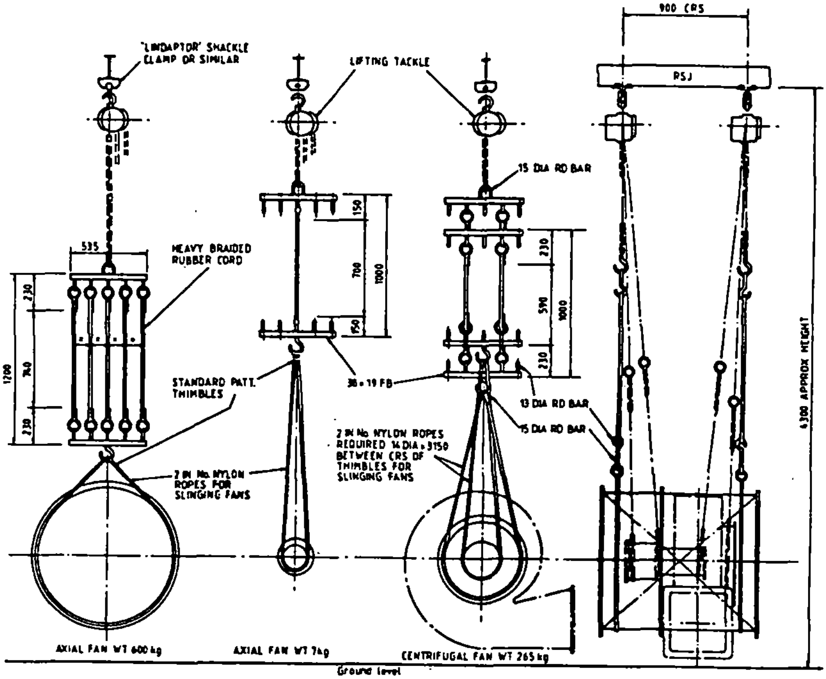

Fig 1 Examples of Resilient Elastic Rope mounted fans

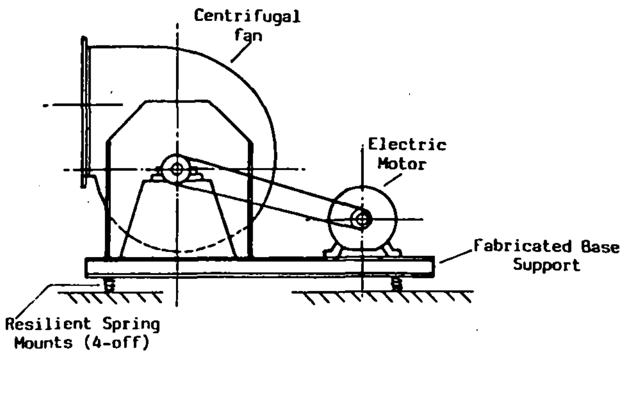

Fig 2 Centrifugal fan assembly on resilient mounts