Vibration Tolerance

Attempts have been made in the past to establish tolerances for machinery vibration, none of which have been entirely satisfactory or practical for it is not practical to set up arbitrary criteria to serve as a rigid rule for passing or rejecting a given amplitude of vibration for all cases. Vibration may be classified as smooth, rough, destructive and annoying from practical observations, and the conclusions presented in the form of charts. Many factors and reservations which enter into the problem, however, must be made before any such tolerance chart can be used with reasonable assurance. There are two aspects to the general vibration problem:—first, the mechanical, as it affects the safety or maintenance of the machine, and second, the physiological, as it pertains to the comfort of an observer.

By
T. C. RATHBONE
Chief Engineer
Turbine and Machinery Div.
The Fidelity & Casualty Co. of N. Y.
Philadelphia, Pa.

LAST EFFORTS to establish a relationship between allowable vibration amplitudes on machines based on their operating speeds have generally been founded on faulty premises. For example, it has been assumed that the intensity or seriousness of vibration depends on the acceleration of the motion rather than directly on the amplitude. Thus, the force with which a rider is "thrown backward" in a car depends on how quickly the car is accelerated from say, zero to ten miles an hour, rather than on the distance traversed.

With simple harmonic motion, like that of the crosshead of an engine with an infinitely long connecting rod, the acceleration varies directly with the amplitude, but with the square of the frequency or the speed causing the vibratory disturbance. Thus, for two equal vibration amplitudes, one at twice the frequency of the other, the acceleration of one is the square of two, or four times the other.

This fact, together with the questioned assumption that the deleterious effect is in direct proportion to the acceleration, has given rise to the so-called "inverse square" rule in attempting to establish vibration tolerances for various speeds. By this rule, if an amplitude of 0.002 in. is permissible for an 1800 r.p.m. machine, then 0.008 in. would be allowed at 900 r.p.m. but only 0.0005 in. at 3600 r.p.m. Such a rule is inconsistent with experience and cannot be applied generally for there are also other factors which prevent rigid comparative values.

Vibration classification may be affected according to the purpose of the criteria. For example, the manufacturer and the purchaser of new equipment are interested in whether the operation comes within what has been vaguely termed "commercial balance," while the insurance underwriter is more vitally concerned with the extent of vibration allowable before the operation becomes hazardous. In this paper, vibration in its relation to possible damage will be the major consideration in attempting to establish tolerances.

Often the problem is concerned more particularly with higher speed machines, such as steam turbine generators and their running balance, but there is no reason why it should be so confined. Professor G. B. Karelitz of Columbia University, while associated with the author on vibration studies at the Westinghouse South Philadelphia Works, tabulated the opinions of several practical engineers and inspectors as to the smoothness or roughness of a number of turbines while on test, operating mostly at 1800 r.p.m., and correlated this data with simultaneous vibration records made with the vibrometer. To this has been added considerable data obtained by the author and his present associates in their machinery inspection activities on all classes of apparatus. The whole has been digested and averaged to arrive at a tabulation of amplitudes in the various categories of smooth, fair, rough, and dangerous, etc., throughout a wide speed range. The resulting data is used by the turbine engineers of the Fidelity & Casualty Co. merely as a guide in their inspection activities.

Measuring Instruments

Instruments for measuring machinery vibration are now fairly common. They are generally of the seismometer type, and customarily are applied at the bearings. The majority of vibrometers measure one component of the motion, such as the vertical, lateral (transverse) or axial (longitudinal). The Davey photographic vibrometer indicates and records the

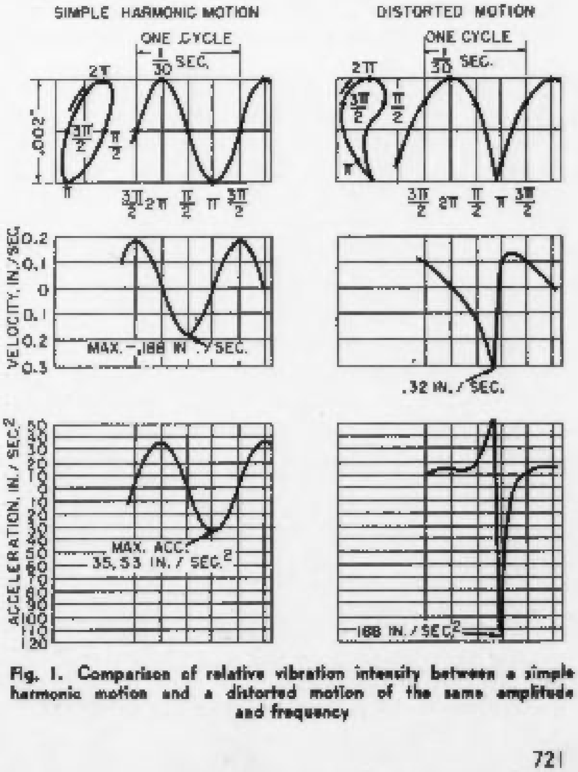

Fig. 1. Comparison of relative vibration intensity between a simple harmonic motion and a distorted motion of the same amplitude and frequency

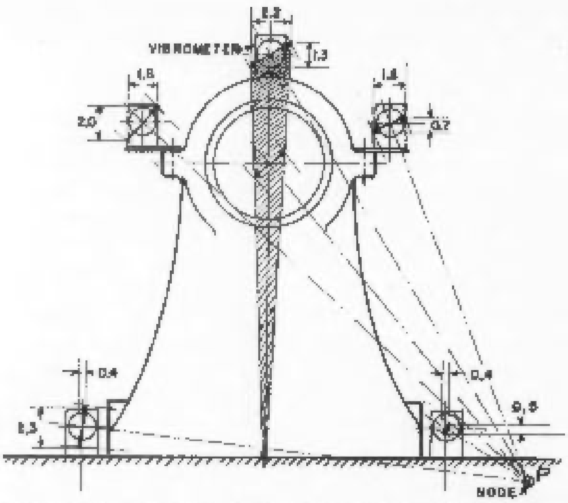

Fig. 2. Turbine pedestal bearing showing why the vibration on the two sides may not be equal

actual path of vibratory motion in a vertical plane, either transverse or axial, so that both vertical and horizontal components are directly observed as the shape or path of the resultant motion. Electric type vibrometers indicate vibration velocity, calibrated in terms of amplitude for given speeds.

If the vibration is simple harmonic at a common frequency, the figure is some form of the ellipse. The figures are often distorted, however, by the presence of two or more frequencies, or by impactive forces such as pounding at the bearing keys, loose wheels, coupling or gear disturances, etc. A common figure due usually to impact resembles a mutton chop. Acceleration is no longer proportional to displacement with such distorted motion. This is why two vibrations with the same amplitude and frequency may "feel" quite different, one smooth, the other hard or sharp.

At the left of Fig. 1 is shown a simple harmonic vibration, of true elliptic form, with the displacement, velocity and acceleration graphs for the 2 mils vertical amplitude at 1800 r.p.m. The right hand side of Fig. 1 represents a vibration having a 2 mil vertical component, at 1800 r.p.m., but this motion, as exhibited by the vibration figure, is distorted by impact or other causes. The reference points on the figure represent the 4 phase points, 90 deg. apart in rotation. The velocity and especially the acceleration graph for this motion have values far greater than for the simple motion, the maximum acceleration actually corresponding to the normal maximum acceleration for a simple harmonic vibration, of over three times the amplitude, or 6 mils.

In estimating the severity of a vibration it is thus important to know its character, to make due allowance for any abnormal condition or deviation from simple harmonic motion. The tolerance for one component may not agree with that permissible for the component in another direction. For example, 2 mils of vertical amplitude may create a much greater general disturbance to the unit as a whole than double that amount in the lateral direction. In fact, such a condition is the rule rather than the exception.

Individual bearings often require separate consideration. On the larger condensing turbine-generator sets with massive and rigid supporting of the No. 2 bearing, an amplitude of say 3 mils vertical would generally represent a far more severe disturbance to the unit as a whole than an equivalent amplitude at the thrust or outboard bearing. Thus a more stringent tolerance would be in order at this location.

Outboard bearing pedestals are usually independent members and their vibration is more or less localized. It is not uncommon to find amplitudes as great as 3 or 4 mils at 1800 r.p.m. in the lateral and particularly the axial direction, with little disturbance to the main unit. Thus, a greater tolerance may often be permissible for this bearing.

Amplitude Alone Meaningless

The statement that the vibration at a bearing has a certain amplitude may be meaningless, unless the location of the vibrometer is definitely specified and comparative readings always referred to that location. Almost invariably the amplitudes observed simultaneously at the two sides of the bearing show some difference, and sometimes the amplitude at one side may be double or more the other. The amplitude at the top center of the bearing pedestal cap is generally greater than at the horizontal joint flanges, particularly in the lateral and axial direction. This condition is due to a common rotary or conical mode of vibration about a node located at one side of the base of the pedestal, as illustrated in Fig. 2. Such a mode of vibration can readily be explored by means of the small hand type vibrometers, and should always be ascertained if the vibration is appreciable. It is recommended that all bearing vibration be observed at a definite location, such as the top of the pedestal cap.

Resonance characteristics of the structure as a whole may operate to allow increased tolerances, particularly if the bodily movement of the rotating and stationary parts are in unison. It is now understood that the amplitude of vibration for a given unit of unbalance at a given speed is largely a matter of accident, as it depends on the amplification factor, which is fixed by the relation between the operating speed and the resonant speeds of the several modes of vibration.

It is not yet possible to control or predetermine these resonance characteristics with any degree of certainty. Although turbine generator units may be

Fig. 3. Resonance curves of the vertical vibration of two turbines made from actual test data. Curves A and B are for the No. 1 bearings of two different units

identically constructed, the foundations, which play such an important role in determining the resonance characteristics, vary greatly. A pound of unbalance at the balance hole radius may cause a vibration of 10 mils on one unit, but only 2 mils on another unit at the same speed. Thus, to reduce the amplitude on the first unit, extreme refinement in balancing is necessary, due to the much greater sensitivity of response. Obviously the destructiveness cannot be in proportion, as the centrifugal forces responsible for the disturbances are roughly the same for relatively rigid rotors.

Resonance Curves

Allowance may properly be made in applying tolerances with such a situation, if a large amplification factor at the operating speed is demonstrated by actual resonance tests. Such a test merely involves the determination of amplitudes corresponding to various speeds, preferably up to 10 per cent overspeed. The plotted results are called resonance curves, and represent the inherent characteristics of the installation. Such an actual test is shown by curve A in Fig. 3. This unit had very little residual unbalance, but a relatively large resonance peak existed slightly above the operating speed. Slight changes in speed around 1800 r.p.m. or slight shifting of weight reactions due to temperature distortions in the structure resulted in large differences in vibration amplitudes, therefore the particular amplitude observed at 1800 r.p.m. has less significance. Curve B, on the other hand, although having a smaller value at 1800 r.p.m., represents a much more serious disturbance.

These facts indicate that a better criterion might be based on the amount of residual unbalance rather than on the resulting amplitude. With the more modern field balancing apparatus now available which determines the phase as well as the amplitude of vibration, the residual unbalance is readily determined by the unit vector process. When unusual sensitivity is encountered, the residual unbalance information actually should be recognized in establishing a tolerance.

Resonance, or great sensitivity for small unbalance cannot, of course, serve to condone a vibration which is palpably hazardous to the unit. Resonance of individual parts and appendages, such as the steam lines, governor dome, valve chest, etc., unless corrected by structural alterations, may require more stringent tolerances in the bearing vibration.

The general mode of vibration of the installation at the operating speed may result in a node or dead point occurring near one or more bearings. In such a case, the general disturbance of the unit as a whole may be out of all proportion to the amplitudes observed at the bearings, and individual treatment again becomes necessary. The vibration or jumping of the shaft itself, apart from the bearing vibration, may aid in estimating the condition of the unit, if the reference location on the shaft is known to be smooth and concentric with the journals. Where the general vibration is great but the bearing vibration is small, the shaft motion itself may disclose an unsatisfactory

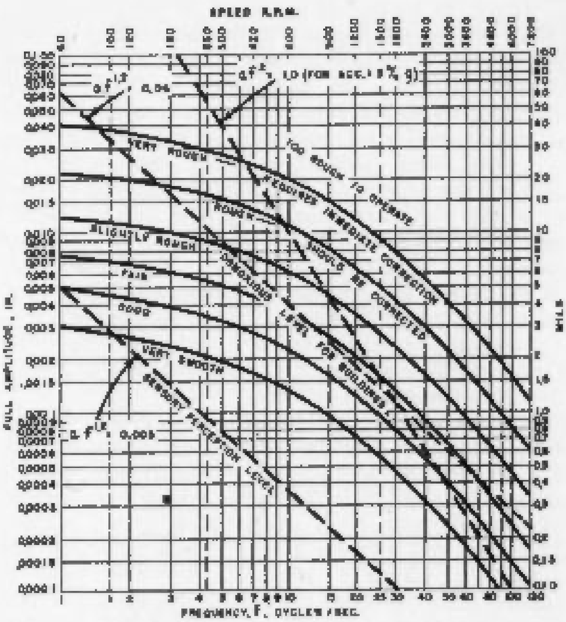

Fig. 4. Machinery vibration tolerance chart

condition, but large bearing vibration may exist with little relative motion between the bearing and shaft.

In estimating the general disturbance to the unit as a whole, the experienced observer will not be deceived by the clatter and chatter sometimes set up by loose floor plates, hand rails and the like, which may occur even with a smooth running unit. He is concerned however, with the vibration of the piping, linkages, governor and other appendages, the general vibration throughout the entire structure and surroundings, and the noises heard in the bearings and cylinder parts with the listening stick.

Danger Point

The magnitude of vibration that is almost certain to cause damage is of course problematical. One important factor is the closeness of the clearance between the stationary and rotating parts, and whether the rotor vibration is in or out of phase with the movement of the stationary parts. The possibility of great shaft deformations as a result of rubbing contact is important. The author once witnessed a vibration at the No. 1 bearing of a large 1800 r.p.m. turbine which reached a momentary value of 30 mils, with no ensuing damage. This is exceptional, as blading and packing rubs causing damage have occurred with vibration less than 8 or 10 mils, at this speed. Continued large vibration is of course, more serious than momentary peaks such as occur with sudden load changes, as fatigue of parts or bearing damage may ensue.

It is evident from what has been said, that each installation requires individual consideration, based on experienced judgment. But with the understanding that many factors prevent the assigning of hard and fast tolerances to cover all classes of apparatus, the chart Fig. 4 is offered to serve as a guide in estimating the comparative severity of the vibration of machinery.

It will be noticed at once that the graphs do not follow the inverse square of frequency law, or any other fixed index, particularly at the lower ranges. If we held to this law and extrapolated to low frequencies, we would soon reach a situation where an allowable amplitude would be great enough to disrupt the parts. For example, assuming the inverse frequency squared rule, if 2 mils can be allowed at 1800 r.p.m., then the tolerance would be 8 mils at 900, 32 mils at 450, 128 mils at 225, and 512 mils—over half an inch, on a 112 r.p.m. engine. This is obviously unthinkable. For low speed reciprocating machinery, with its unavoidable residual external reactions, it is common engineering practice to design the base with sufficient mass to limit the maximum rocking to a definite amount, such as 5 mils.

It will also be noticed that the graphs in the frequency range from 600 to 3600 r.p.m. roughly follow the slope of the proposed threshold and nuisance graphs for comfort in buildings, which were arrived at independently from physiological considerations. But below 600 r.p.m. or 10 per second frequency, the machinery tolerance graphs curve downward to smaller values, crossing the physiological graphs. This is reasonable, as the sensory perception of low frequency vibration bears no relation to the stresses set up by the movement. Experience beyond 5000 r.p.m. is somewhat meager, but the slopes extrapolated with the index $af^{1.7}$ appear to be reasonable and consistent.

Summarizing, the chart values characterizing the severity of machinery vibration represent the results of a great number of observations on many types of machines, and are based on practical rather than on theoretical data conforming to some assumed relationship. Variations of 25 per cent or more either way are to be expected in individual cases. With proper recognition of the various influencing factors described, the chart should furnish a reasonable guide for estimating the severity of vibration.

How Much Gas?

By James O. G. Gibbons

ANYBODY who may have to design, or make changes, in boiler settings, breechings, etc., will find himself in a more satisfactory position, if he has some definite idea as to the amount of gases which will have to be handled.

It is not necessary that this information should be very exact, indeed; extreme accuracy is impossible, as there are too many variables, such as maximum load, exact quality of the fuel, excess air, etc., which can only be roughly approximated. Nevertheless, we want to be in a position to make our estimates in such a way that we may have confidence in the reasonableness of the results. For this purpose, the following will be more than sufficiently accurate. The figures are based upon bituminous coal having a heat content of 14,000 B.t.u. per pound, with a combined efficiency of approximately 75 per cent, but, as will be shown later, for purposes of design, the data can be used for other grades of coal or even oil.

Air (at 70 deg. F.) required per minute, per boiler hp.
0% Excess air, per minute.............. 7.5 cu. ft.
For every 10% excess air, add........ 0.90 cu. ft.
100% Excess air, 7.5 + 9.0..............16.50 cu. ft.
Cubic feet of flue gas at 550 deg. F. per minute, per boiler hp.
0% Excess air, per minute..............14.5 cu. ft.
For every 10% excess air, add....... 1.75 cu. ft.
100% Excess air, 14.5 + 17.5............32.00 cu. ft.
Weight of flue gas per minute, per boiler hp.
0% Excess air .... 5.8 lb.
For every 10% excess air, add........... 0.7 lb.
100% Excess air, 5.8 + 7.....................12.8 lb.

It is conservative to assume that there will be 20 lb. of flue gas per pound of coal, and 18 lb. per pound of oil.

Cubic feet of gas per minute, per boiler hp. at 2000 deg. F.
0% Excess air ....35.5 cu. ft.
For every 10% excess air, add........ 4.5 cu. ft.
100% Excess air 35.5 + 45..............80.5 cu. ft.

The air per minute per boiler horsepower will be required when estimating the capacity of air intakes, forced draft fans, etc.

The data relating to flue gases will be useful in designing breechings, estimating the capacity of induced draft fans, and stack draft losses.

The volume of gas at 2000 deg. F. will, in connection with that of the exit gases, be useful in estimating the volume of gas to be carried in the different boiler passes.

Of course, in the case of a new boiler, it may be assumed that the design is such that there will be no excessive draft losses, but very often stokers or oil burners are installed on old boilers, with the intention of increasing the rating, in which case it is well to keep in mind that this increases the amount of gas to be handled, but does not increase the area of the spaces through which it has to pass, so it is well to see that the draft will be adequate under the new conditions.

A mistake is often made in assuming that the installation of oil burners will decrease the amount of draft required, and enable the present stack to carry a much greater load. This is true, in some cases, especially when the air is taken into the furnace through a checker floor but, with register type burners, the resistence is sometimes so great that they are equipped with forced draft fans.

As the theoretical number of cubic feet of air required for any fuel is nearly one per cent of the B.t.u. value of the fuel, and the number of B.t.u. in a boiler horsepower hour (or minute) is a constant; when operating at the same efficiency and with the same per cent of excess air, the number of cubic feet of air required is the same for any class of fuel, so if the efficiency is around 75 per cent, the data given will apply to any fuel, and all we shall have to estimate is the excess air, which will probably be about as follows:

Oil burning, 20 to 25 per cent.
Stoker fired coal, 40 to 60 per cent.
Hand fired coal, 40 to 100 per cent.

If the expected efficiency is other than 75 per cent, a correction can readily be made.