LOW FREQUENCY BUCKETS FOR INDUSTRIAL STEAM TURBINES

by

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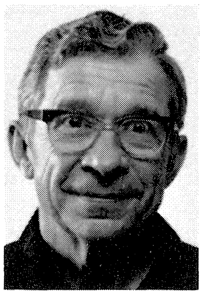

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ABSTRACT

Bucket failures have been known to occur on the later stages of industrial type steam turbines after prolonged operational life of as much as five or six years. All of these failures are characteristically fatigue in nature. It is the purpose of this paper to discuss the reasons for these failures as related to the operating load and speed of the turbine and to the design of the buckets. Bucket banding can be used to minimize the possibility of failure.

INTRODUCTION

Bucket failures have been known to occur in the later stages of industrial type turbines after prolonged operational life of as much as five years or more. All of these failures are characteristically fatigue in nature. It is typical of these applications that both load and speed may vary over the life of the equipment.

It is the purpose of this paper to consider the special case in which the bucket is tall and its first natural frequency is relatively low. This bucket may have its first natural frequency excited by a stimulus having a frequency equal to some low multiple of turbine running speed.

The various sources of this low frequency stimulus are discussed, and the effects of turbine speed and loading on bucket life are considered. A relationship is shown in which the shroud band can be used to reduce the vibratory response of the banded group of buckets and improve their life.

NOZZLE PASSING FREQUENCY

Nonuniform bucket loading around the 360° arc that the bucket travels is the source of vibratory stimulus on the bucket. One obvious source of nonuniform loading is the stationary nozzles that direct the steam flow to the bucket. These nozzles, being of finite thickness, cause perturbations in the steam flow pattern at a frequency corresponding to the number of nozzles per 360° times the operating speed. This stimulus is commonly called the *nozzle passing frequency* stimulus. Since this stimulus is not sinusoidal in nature, it contains components not only at the nozzle passing frequency, but also at higher multiples of that frequency. These stimuli have frequencies that are generally 50 times running speed or more. These higher frequency stimuli, although important in turbine design practice, are not being considered in the text of this article.

LOW FREQUENCY STIMULUS— NONUNIFORM BUCKET LOADING

The same nozzles that were discussed above are also a source of nonuniform bucket loading that generates low frequency stimulus to the buckets. In the manufacture of the stationary blading, or diaphragm, there is always some variation in stationary, or nozzle, blade spacing, and therefore some variation in nozzle area per unit of circumferential distance around the diaphragm. These variations in area per unit of circumference are usually at a maximum at each side of the horizontal joint of the diaphragm, or casing. Therefore, it is not uncommon to find that the diaphragm is a significant source of "two per revolution" (2 Per-Rev) stimulus, due to the variations in nozzle area at the horizontal joint. Since the area variations are not sinusoidal in nature, it can be found that most diaphragms are sources not only of 2 Per-Rev stimuli, but also of all the lower Per-Rev multiples, such as 1×, 2×, 3×, 4×, 5×, etc. Most of the time, the majority of these stimulus components are relatively small, being less than one or two percent of the normal driving force on the bucket. Their size is, of course, directly dependent on the accuracy of manufacture of the stationary bladed parts.

Another source of low frequency stimulus is found in the exhaust configuration of steam turbines. The exhaust casing acts to control the direction of steam flow and to turn it perpendicular to the axis of the turbine. This action results in a nonuniform exhaust pressure around the 360° circumference of the last stage. This nonuniform loading generally produces primarily a 2 Per-Rev stimulus, but it also may produce other significant low multiple stimuli.

The exhaust area distribution also is not always consistent with that needed for uniform loading of the bucket. It is common practice to strengthen and support large condensing exhaust casings with stay bracing and stiffening ribs. All of these features are potential sources of nonuniform loading on the 360° circumference of the stages of buckets that are subject to these variations. This not only includes the last stage buckets, but it also may include the next to last stage and other rows of buckets that are involved in industrial type turbine double flow condensing exhaust designs.

Obviously, none of the factors discussed above are of a purely sinusoidal nature. Therefore, it can be expected that they will generate stimuli at all the lower Per-Rev multiples. There are no extensive data on the size of these stimuli, but bucket failures that have been associated with them indicate that they can be of the order of magnitude of 5% to 15% of the normal driving force on the bucket.

The same types of physical features that are described for the turbine exhaust may also be found in other areas of the turbine, such as at extraction openings or other discontinuous features in the design of the turbine casing and other internal stationary parts.

The low frequency stimuli will cause vibratory stresses in the bucket that are small as long as the bucket does not operate on or near its natural frequency. The bucket vibratory stress, off resonance, will be in approximately direct proportion to the low frequency stimulating force. For example, a stimulus that is 5% of the normal driving force will produce a vibratory stress that is equal to approximately 5% of the normal steady state steam bending stress, provided that the bucket is not on resonance.

From the above discussion it can be seen that low frequency stimuli will be prevalent throughout the turbine to a larger or smaller degree, depending on the physical construction of the stationary parts in the steam path of the turbine. However, the effect of these stimuli on bucket stresses will be small as long as the buckets are not on resonance.

LOW FREQUENCY STIMULUS—ROTOR VIBRATION

A source of low frequency stimulus to the bucket is found in the vibratory spectrum of the turbine rotor. It is well known that the vibration signature of the turbine rotor always shows some vibration at several of the lower multiples of running speed. Figure 1 shows one such typical low frequency vibration spectrum.

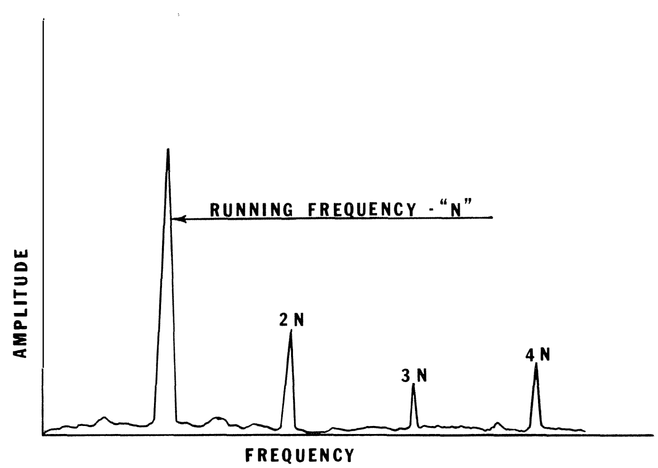

Figure 1. Typical Low Frequency Vibration Spectrum.

The available vibration standards [1] are usually interpreted only in terms of running speed. That is, the machinery overall vibration in mils, or velocity in inches per second, is judged to be adequate or not based on comparison to a standard limit that is determined at the turbine operating speed. There are today no specific rules for determining vibration limits for the vibration at multiples of running speed. For example, API Standard 612 for Special Purpose Steam Turbines for Refinery Services has the following shaft vibration limit:

$$\text{Unfiltered double amplitude including runout (mils)} = 1.25\sqrt{\frac{12{,}000}{\text{rpm}}}$$

The specification also requires that, at slow roll, the shaft runout should not exceed one-fifth the above value or 0.25 mils, whichever is less. This could be interpreted to say that a turbine operating at any speed of 12,000 rpm or less could have journals out of round by 0.25 mils. This out-of-round could be in any shape of two, three or more lobes to stimulate the turbine at 2×, 3×, 4×, etc.

The rotor vibration is a complex function depending on the rotor and bearing designs, as well as the runout of the journals and the rotor unbalance. Although the journal runout is not a direct measure of the resultant rotor vibration, it is some indication of the probable level of vibration at the various multiples of running speed.

When there is a geared drive, the vibration signature of the gear and the driven equipment may also show up on the turbine rotor vibration signature. Figure 2 shows an example of a bearing cap reading taken on a turbine that drives a compressor through a step up gear. The step up gear ratio was such that the 3× compressor vibration occurred at 4.87× turbine speed. That is, the 5× turbine vibration nearly coincided with the 3× compressor vibration to make this a major component of the vibration spectrum of the turbine. The vibration level was 0.044 inches per second at 547 Hz on the turbine bearing cap. This is a relatively low velocity. Casing velocities of 0.15 to 0.20 inches per second are generally considered as acceptable levels of vibration. However, this turbine had experienced bucket failures that were attributed to excitation of the bucket first tangential frequency by 5 Per-Rev stimulus from the turbine.

BLADING RESPONSE TO ROTOR VIBRATION

It has been stated earlier that bucket failures have been thought to have been caused by stimuli of the order of 5% to 15% of the normal driving force on the bucket. This would correspond to about 100 psi to 300 psi off-resonant vibratory stress in a bucket having 2000 psi steady state bending stress.

There are so many variations in bucket design that it is impossible to generalize to cover all buckets. However, for a 10 inch long by 2 inch wide uniform bucket, having a cross section area of 1.11 square inches and a section modulus of 0.108 cubic inches, the rotor vibration necessary to produce 100 psi alternating stress at the base of the bucket is found to be as follows:

$$a = \frac{1.38 \times 10^4}{N^2} \tag{1}$$

where

a = peak to peak rotor vibration, mils
N = frequency, Hz

This equation has been used to calculate the information in

Table 1, which shows the peak to peak vibration amplitude and the velocity required to produce 100 psi alternating stress in the 10 inch long uniform bucket for various forcing frequencies. Therefore, if the 10 inch long bucket were operated at 6000 rpm (100 Hz), it could be expected to have fatigue failures if the bucket resonance were excited by any of the rotor vibrations in Table 1.

Table 1. Rotor Vibration Necessary to Produce 100 psi Alternating Stress in a 10 Inch Bucket.

FREQUENCY	AMPLITUDE	VELOCITY
100 Hz.	P-P 1.38 MILS	.44 IN/SEC
200	.34	.21
400	.09	.11
600	.04	.07
800	.02	.05

However, there will be few 10 inch long straight buckets operating at 6000 rpm. Instead, they will most likely be tapered. A bucket has a relatively large taper when the cross section area at the bucket tip is equal to only half the area at the bucket root. For a bucket having this amount of taper, the rotor vibration required to produce a given stress at the root will be approximately three times the amount required for a straight bucket of the same length.

The rotor vibration amplitude required to produce a given forced vibration stress in the bucket will vary approximately directly with the bucket width and inversely with the square of the bucket length. So, longer or narrower buckets will be easier to excite with rotor vibration, and shorter and wider buckets will be more difficult.

Today's vibration instrumentation gives an indication of the rotor vibration, but does not give a direct measurement comparable to the values discussed above. The casing or bearing cap measurement obviously does not give rotor vibration. The so-called "shaft reading" instrumentation really measures the difference between the bearing bracket vibration and the shaft vibration. Both of these measurements are taken near the ends of the rotor, while the rotor vibration calculated by Equation 1 and shown in Table 1 is the rotor vibration at the bucketed stage. The rotor vibration at the bucketed stage will, in most cases, be substantially larger than the vibration at the bearings. The vibration levels in Table 1 are not directly comparable to usual vibration readings, but they are indicative that it is possible that bucket failure can be caused by levels of rotor vibration not considered objectionable by current vibration standards.

STIMULUS RELATIVE IMPORTANCE

The low frequency stimulus has little or no effect on the majority of buckets in the usual industrial steam turbine. The reason for this is that most buckets have a first (or lowest) natural frequency that is well above the low frequency stimulus that we have been discussing. For example, the first natural frequency (first tangential) of a typical straight, uniform section bucket that is 1.125 inches wide and 4 inches tall will be about 1,000 Hz. It would require a 10 Per-Rev stimulus to excite this frequency bucket for a turbine operating at 6,000 rpm. While it

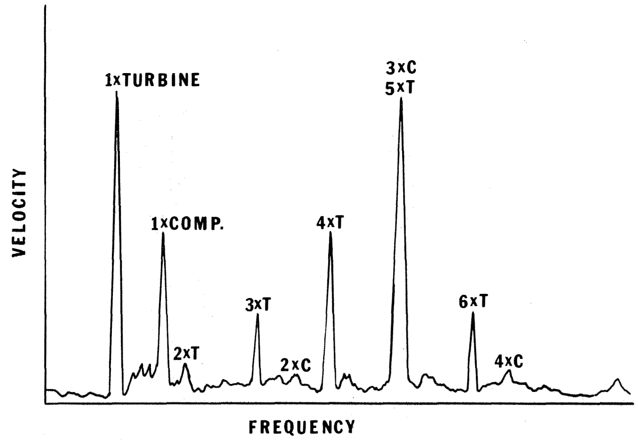

Figure 2. Turbine Bearing Cap Vibration Velocity Showing Excitation from Gear Driven Compressor.

is not impossible for this stimulus to exist, it is highly unlikely that it will be of sufficient magnitude to cause failure of the bucket.

There have been a few isolated cases in which bucket failure of these smaller, high frequency buckets has been attributed to rotor vibration. In these cases, the rotor was vibrating at higher frequencies, corresponding to a stimulus from the driven equipment that was usually geared to the turbine.

The problem of low frequency bucket stimulus becomes more critical as the bucket becomes taller and its first tangential frequency becomes lower. There is no absolute limit of increased first tangential frequency beyond which you can conclude that there is no probability of a problem. There is experience of bucket failures due to operation with the bucket first tangential natural frequency stimulated by multiples of running speed from 2× to 7×. There seem to be few, if any, failures due to this cause when the bucket natural frequency is above 7× running speed. Also, there have been no recent failures at 1×, because all manufacturers successfully design to avoid operation with the bucket natural frequency on running speed. For industrial drive turbines the bucket frequency-speed relation shown in Figure 3 is somewhat typical for the last stage condensing bucket.

The relationship of Per-Rev stimulus and bucket failure as presented above is a great oversimplification of this problem. Obviously, a bucket operating with very low steady state steam bending stress will be more likely to operate successfully on any given Per-Rev stimulus than a bucket with very high steam bending stresses.

TURBINE SPEED RANGE

Figure 3 shows an example of a relationship of the bucket first tangential frequency to the lower multiples of running speed. In this example, the bucket first tangential frequency can be excited by either 4× or 5× running speed within the normal operating speed range of the turbine. You can see that a minor increase in bucket natural frequency will not eliminate the problem of potential resonant excitation. As long as the operating speed range remains as shown, raising the bucket frequency by some moderate amount will only tend to shift the excitation sources to 5× and 6× or some other higher multiple of the running speed.

To change the bucket design to increase its frequency will usually reduce the risk of failure. The higher frequency bucket will be stronger and stiffer than the lower frequency bucket, and therefore better able to withstand any given vibratory

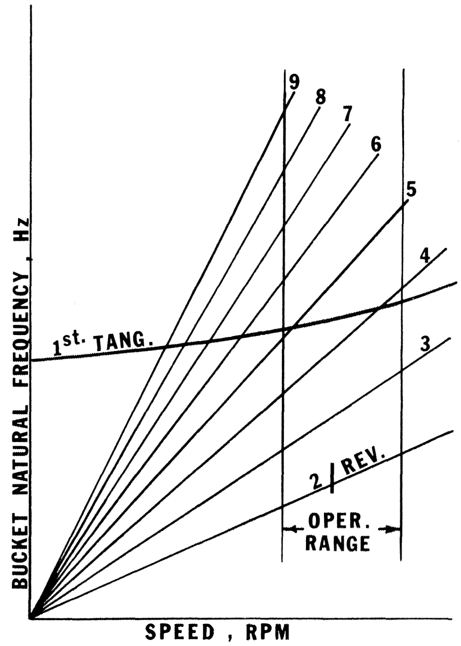

Figure 3. Bucket Resonant Frequency and Per-Rev Stimulus Versus Turbine Speed.

stimulus. Also, it is likely that the stimulus at a higher Per-Rev resonance will be smaller than at the lower Per-Rev unless there is some specific identified source of the higher Per-Rev stimulus, such as the number of casing internal stay-bars per 360°, or significant rotor vibration response at the correct frequency.

If the operating speed range in Figure 3 were broadened a small amount, it could be possible for any one of three Per-Rev stimuli to excite the bucket natural frequency, provided that the appropriate speed were run. Usually, this possibility is complicated by the facts that the bucket natural frequency is not precisely known and that the natural frequency is a range of values, due to manufacturing tolerances, rather than a single specific number. So, taking these factors into account, it is not unusual for a broad operating range of speed to have the possibility of two or three different low Per-Rev stimuli that could excite the bucket somewhere within the speed range.

EFFECT OF OPERATING SPEED ON VIBRATION STRESSES

The relative vibration magnitude for a simple spring mass system that is excited by a vibratory force is given by Equation 2.

$$A = \frac{1}{\sqrt{[1-(N/N_c)^2]^2 + [(\delta/\pi)\ (N/N_c)]^2}} \tag{2}$$

where

- A = magnification factor or relative vibration
- N = frequency of exciting force, which is proportional to turbine operating speed, Hz
- N_c = natural frequency of the system, or bucket natural frequency, Hz
- δ = logarithmic decrement damping of the system, per unit
- π = 3.1416

From Equation 2 it is seen that the vibration amplitude, A, and, therefore, the bucket vibratory stress amplitude are primarily controlled by damping, δ, when the system is operating on or near resonance ($N = N_c$). Measurements that have been made indicate that the logarithmic damping for banded turbine buckets is probably between 1.0% and 4.0% [2]. A value of 2.0% (or 0.020) was selected as representative, and Equation 2 was solved for "A" for various ratios of N/N_c. This information is plotted in Figure 4, which illustrates the major effect of turbine operating speed on bucket vibratory stress. From Figure 4 it can be seen that, if the turbine were operating only 1% off the bucket resonance (that is, $N/N_c = 0.99$), the vibratory stress would be only approximately a third of what it would be if the turbine were operating at a speed that placed the bucket exactly on resonance ($N/N_c = 1.00$). This indicates that, for a turbine operating at about 6000 rpm, it is possible that a 60 rpm change in speed could change the vibratory stress in a low frequency bucket by a factor of as much as three to one. This is the most likely explanation for the experience of those industrial turbines that have operated successfully for several years, but then have had repeated bucket failures after a relatively small change in normal operating conditions that included a small change in speed.

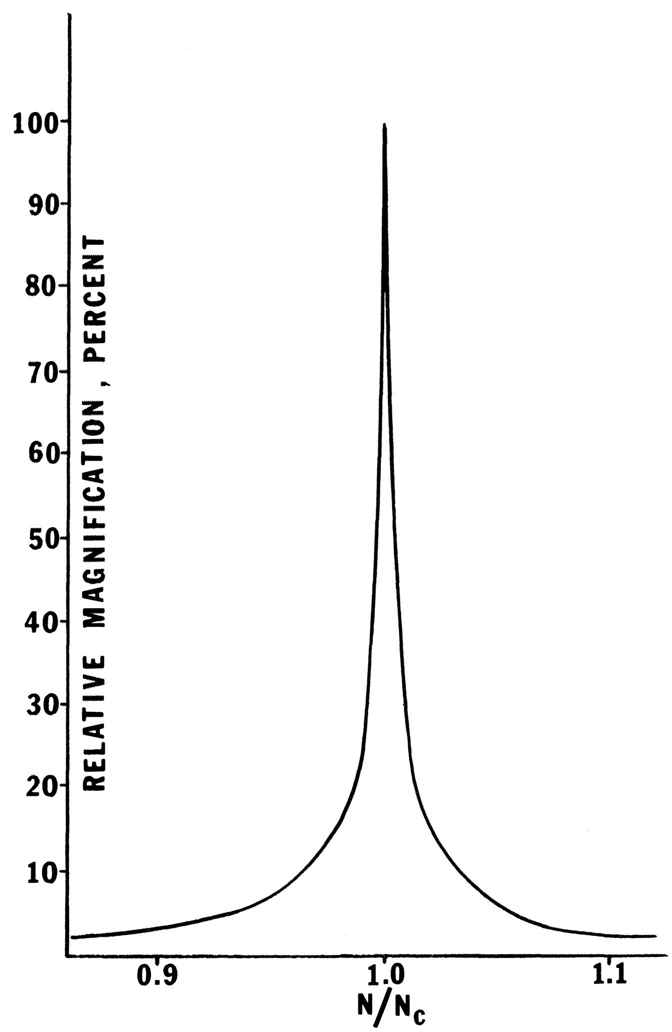

Figure 4. Bucket Relative Vibratory Stress for 2% Damping Versus the Ratio of Turbine Speed to Bucket Resonant Speed.

BUCKET LOADING

So far, the discussion has been on bucket natural frequencies and potential exciting frequencies. An equally significant factor is the normal steady state bucket bending stress due to the average tangential driving force on the buckets. In general, the amount of the stimuli discussed above will be in approximately direct relation to the bucket bending stress. That is, if the same bucket were applied in two different turbines, and the normal bucket loading, and therefore the bucket bending stress, were double in one turbine what it was in the other turbine, it would be expected that, with all else equal, the higher loaded bucket would have significantly higher vibratory stimulus, approaching double that of the lower loaded bucket.

Steady state bucket bending stresses are low. Practices vary within the industry, but usual experience is that buckets are designed to have steady state bending stresses less than about 3000 psi. Operating experience indicates that the turbine can be expected to run successfully with the bucket first tangential natural frequency excited by 4 Per-Rev and higher multiples of running speed, if the normal bucket bending stress due to the average driving force on the bucket does not exceed approximately 1,000 psi. Even such a low design stress does not assure safe operation, since there has been at least one failure from bucket resonance with these lightly loaded buckets.

BUCKET LIFE

When operating on or through a harmful resonant range, a bucket incurs fatigue damage. This damage is cumulative in nature and may, in time, result in the failure of the bucket. The length of time that it takes for failure to occur is dependent on the vibratory stress that exists at any operating speed and the length of time the turbine operates at that particular speed. Therefore, for any given design of turbine having a bucket that can operate on harmful resonance, the operating time of the turbine for failure may vary from only a few hours to a few years, or perhaps never, depending on the speeds at which the turbine is operated.

For example, let us assume that the overall design is such that the bucket, when operating exactly on resonance, will have a vibratory stress just equal to the fatigue strength of the material. Conventional engineering would indicate that, under these conditions, the bucket life would be about 10 million cycles. It does not take very long to operate for this number of cycles, even with a so called "low frequency" bucket. The usual range of natural frequencies for the "low frequency" bucket is about 100 Hz to 500 Hz. Using these natural frequencies, it can be seen that it will take only about 6 to 28 hours of operation to fail these buckets, provided that they are operated precisely at resonance with a vibratory stress equal to the fatigue strength of the material.

These same buckets may have a vibratory stress of only 10% of the fatigue strength when operating slightly off resonance. At this stress level, the blading can operate indefinitely without measurable fatigue effect. Therefore, the total operating time to failure for these buckets depends on how long it takes in the normal operating cycle of the turbine, perhaps changing from season to season, to accumulate the 6 to 28 hours total life at or very near to resonant operating speed.

The major factor in turbine bucket life is the operating speed at which the turbine is run. Relatively small variations in speed can result in large changes in vibratory stress as the bucket moves into and out of resonance. There are also other turbine operating variables, such as changes in turbine load and steam conditions, that may cause changes in the bucket operating vibratory stress. The effect of these factors is generally small, compared to the effect of changes in speed.

Corrosive steam environment can also be a factor acting to reduce the fatigue strength of the material. The short time effect of a wet steam atmosphere is to reduce the bucket fatigue strength to about half of what it would have been in air, or in superheated steam. If there are corrosive elements in the wet steam, they will tend to cause a continuing deterioration of the material fatigue strength with time, such that it may affect the long time life of the buckets. There may be pitting of the bucket surface with time. Pits increase the stress concentration factor in the blading and therefore increase the effective vibratory stress. In these cases, when blades have cracked, it is typical to find that the crack started at a corrosion pit. The effect of corrosive elements in the steam environment is not normally evaluated unless evidence of stress corrosion or corrosion fatigue is detected by metallurgical examination of the fracture surface of the failed bucket.

BUCKET BANDING EFFECT ON VIBRATORY STRESS

Bucket banding on low frequency buckets generally has been of the nature that it had little effect on either the natural frequency of the bucket or the resonant response of the buckets to some stimulus at low multiples of running speed. That is, the number of buckets banded together under a single band has been so few that the banded group responded essentially like a single bucket would at its first tangential natural frequency. This will no longer be true if the buckets are banded in large groups [3]. For example, if it is assumed that the bucket first tangential natural frequency is equal to C times the rotating speed of the rotor, and that there are P bucket pitches in the 360° arc of the wheel, then it is apparent that the total vibratory energy input to a banded group of buckets will be zero when the number of buckets per banded group is equal to P/C, or to 2P/C, or to 3P/C, or to any multiple of the ratio of P/C.

By solving for the average energy input to a banded group of buckets, it can be shown that the relative energy input, or stimulus per bucket, when there are B number of buckets per banded group, is as shown in Figure 5. It is obvious that in no case can B be greater than P, since this represents a single band 360° around the wheel.

From inspection of Figure 5, it is apparent that if the number of buckets per band, B, is selected on the basis of the smallest Per-Rev, C, that is likely to be encountered in the operating speed range, this number of buckets per band will also be very effective in reducing the relative stimulus of the other higher multiples of running speed that may be in the operating range.

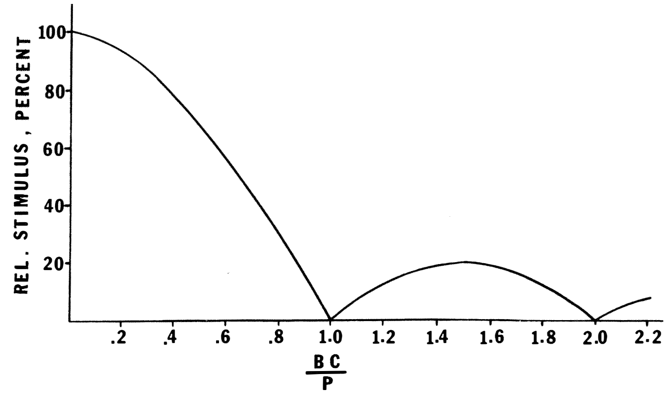

Figure 5. Relative Bucket Stimulus for Various Numbers of Buckets Per Band.

Table 2 shows the effective vibratory response of a banded group of buckets, relative to a single bucket response, for various Per-Rev resonances and for various amounts of continuous band arc. For example, if a bucket operating on 4 Per-Rev is continuously banded in 60° arcs, it will have only 39% of the vibratory stress that it would have had as a free standing bucket.

Table 2. Relative Bucket Stimulus for Various Per-Rev Excitations and Various Lengths of Band.

	RELATIVE STIMULUS, PERCENT				
PER REV	BAND ARC 30°	60°	90°	120°	180°
2	93	83	64	39	Zero
3	88	64	29	Zero	21
4	83	39	Zero	20	Zero
5	73	19	18	17	12

THERMAL EFFECTS

Changes in temperature in the turbine stage can be expected to cause transient stresses in the banded group. The band is thinner than either the turbine bucket or wheel and will therefore respond to a temperature change at a faster rate. This will cause bending stresses in both the band and the buckets. The longer the band is, the higher these stresses will be.

The stage temperature changes are primarily associated with load changes on the turbine. How much this may be depends on individual turbine design. However, in the superheated steam region of the turbine, a 30% change in load will cause approximately 50°F change in stage temperature. The temperature change for the same load change will tend to be less in the saturated steam sections of the turbine. There are very small temperature changes in the condensing exhaust under normal operating conditions. However, there may be a temperature change as large as 200°F on start-up when going from prolonged operation at no load to a loaded condition. These thermal considerations indicate that the band should not be made any longer than is necessary to give safe operation of the buckets.

SUMMARY

Stimuli at low multiples of running speed exist throughout the turbine. These stimuli have caused failure of buckets operating on resonance with the bucket first tangential natural frequency. Stimuli may come from rotor vibration, as well as from the steam path of the turbine.

Bucket failure and length of life are largely dependent on operating speeds. Banding can be used to reduce or eliminate the risk of bucket failure due to low multiple stimuli. However, the length of band should be as short as possible to reduce stimuli caused by thermal stresses.

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