FIELD MEASUREMENTS OF BLADE STRESSES ON INDUSTRIAL TURBOMACHINES

by

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Symbols listed at the end of the paper.

SUMMARY

High blade stresses can be induced in axial compressors by various asymmetrical flow conditions or adverse working conditions. A series of measurements were carried out on industrial and gas turbine axial compressors. After a brief review of the various measuring methods, test results are presented and commented. The information gained refers to various working conditions, such as rotating stall, surging, partial injection of the turbine of superchargers for Diesel engines as well as to the influence of the wake of blading, especially for compressors with movable guide vanes. The influence of lacing wires on axial compressor bladings is presented and commented with respect to efficiency loss and increase of noise level.

INTRODUCTION

Blade vibrations in axial turbomachines are the cause of costly operating failures. Due to the complexity of the phenomena involved, it is difficult to find the reasons for the blade excitation and thereby the solution eliminating a blade failure. Due to the lack of the precise knowledge of the mechanism leading to blade failure, it is often impossible to predict the alternating stress level prior to the commissioning of a unit.

The diversity of the causes and parameters leading to dangerous blade excitations imposes direct field testing. Laboratory tests are too limited in scope to be able to furnish the needed statistical evaluation of the various parameters involved. Laboratory tests are, however, useful to study the fundamental causes inducing blade vibrations and to develop reliable measuring methods. Information corresponding to blade stresses for example cannot be transposed directly to smaller or larger similar machines.

After a review of the causes responsible for blade excitation, of the measuring methods and of measurements carried out on various machines, recommendations are made to protect axial compressors against blade failures.

With the exception of one case referring to the turbine blade of a Diesel supercharger, the paper is devoted to axial compressors.

CAUSES FOR BLADE VIBRATIONS

The most common causes creating blade excitation in axial compressors are:

— Wake of the blade rows placed before and after the considered blading
— Variable guide vanes setting
— Rotating stall
— Surging
— Back flow
— Chocking
— Inlet or diffusor supporting vanes
— Asymmetrical inlet flow distribution
— Blade flutter
— Karman vortices from the piping
— Inlet or diffusor flow separation
— Uneven inlet flow density due to side streams
— Variable tip clearance
— Mechanical rotor vibration
— Partial injection (turbines)

Figure 1 represents the general characteristics of an axial compressor. Within the first quadrant the working points correspond to a positive pressure ratio π and flow V. The torque M as well as the sense of rotation n of the compressor are also positive or in the normal direction. The normal working field of the compressor is limited by the surge line S, the chocking line K and the rotating stall area R. Beyond the surge line S the working condition of the compressor is unstable, and the pressure flow characteristics are not well defined.

Within the second quadrant, the pressure ratio is still positive, which means that the pressure at the discharge of the machine is still higher than the pressure at the suction. The flow is, however, negative or reversed compared with the normal flow direction. Within the area extending to the curve n = O, the compressor is working as a brake. The speed and the applied torque are acting in the normal direction, but the flow is reversed, the compressor absorbs in this case a certain quantity of power without communicating a positive energy to the gas stream. Beyond the line n = O, which corresponds to the resistance of the blading to a reverse flow when the machine is not rotating, the speed starts being negative, corresponding to a reverse speed operation, and the compressor works as a turbine. The torque is still positive and is generated by the energy lost by the reverse flow passing through the machine. The speed increases with the flow up to the zero torque line, which is a characteristic of axial machines. This working point corresponds to the runaway speed of the machine. In fact, in industrial installations this extreme working condition cannot be reached since a given torque is always delivered by the compressor to drive backwards the driving turbine or the motor. However, this speed can be fairly high and can endanger the machine. The runaway speed is a function of the applied pressure ratio and is given by the crossing point of the speed curves n with the torque curve M = O. In the case of axial Kaplan water turbines, it is known that the runaway speed is close to twice the normal speed [1]. Beyond the line M = O the compressor works again as a brake.

Within the third quadrant, the flow is negative, and the pressure ratio is also negative. The torque as well as the speed are negative. This working condition corresponds to the operation of the machine as a compressor. The general characteristics are similar to the ones of the normal operation, but the efficiency is low due to the fact that the blading is working in the reverse sense. For semi-axial machines [2] and of course in the case of radial impellers [3], a negative operation can produce a positive head and a positive flow. This is due to the marked influence of the centrifugal effect induced by the impeller itself.

Within the fourth quadrant one can distinguish three zones. For negative speeds and up to n = O the machine is working as a brake. The line n = O corresponds to the resistance of the machine at rest, when the gas stream is forced through the compressor from the inlet to the discharge. Beyond this line, the speed is positive as well as the flow, but the torque is negative. This operating condition corresponds to the operation of the compressor as a turbine. It is submitted to an inlet pressure which is greater than the discharge pressure (negative pressure ratio). Beyond the line M = O corresponding to the runaway speeds of the machine, the compressor is again working as a brake.

The working conditions most likely to occur are confined in the first quadrant and up to the runaway curve M = O in the second quadrant. The operating points within the first quadrant will be the object of the following chapters. Concerning the second quadrant, let us mention two typical operating cases.

The first one refers to single stage gas circulators for atomic power stations [2]. The four circulators are driven by steam turbines and are working in parallel. No back-flow valves are present in the layout of the installation. In case of a shut down of one unit, the other ones being still in operation, its speed decreases rapidly to zero. A non-return speed device maintains the machine at zero speed and back-flow takes place [4], [5]. Due to the drop of the pressure ratio and of the corresponding shift of the flow towards higher values of the other three circulators, the total mass flow through the reactor diminishes only from 100% to 83%. The circulators are designed to cope with this abnormal working condition indefinitely. They are even built for full runaway speed, the lubrication and the bearings being capable of working at full reverse speed.

The second case corresponds to an axial multistage compressor used in catalytic cracking units. A non-return valve is fitted by the customer in the discharge line to avoid back-flow in case of a trip down of the driving turbine. In one occasion the non assisted non-return valve did not close, the unit was running backwards up to its maximum runaway speed. Since the bearings and the lubricating system were conceived for this case, no damage of the bearings took place. However, due to the hot return flow (450 °C) and consequently to the rapid radial thermal expansion of the blading, the blade tips came in contact with the casing and the rotor, causing considerable damage. In principle, only assisted and remote actuated non-return flow valves should be used in such installations.

A further third operating case refers to the fourth quadrant. It is usual to drive slowly a gas turbine rotor after shut down in order to insure a uniform cooling. Generally a motor driven bearing gear is used. In the case of the SULZER 10 MW gas turbine unit [6], the machine is maintained in slow rotation by a fan pressurizing air in the gas turbine compressor. The compressor is working as a turbine with negative pressure ratio and positive flow. This case corresponds to a working point in the general characteristics (figure 1) situated between the curves n = O and T = O. The unit still rotates in the positive direction.

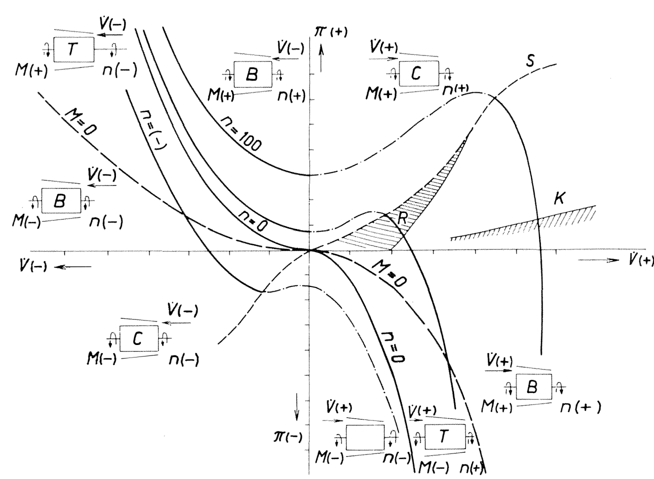

Figure 1. General characteristics of an axial compressor.

C = compressor operating.
T = turbine operation.
B = brake.

It is clear that working conditions other than normal may induce high alternating stresses in the blading of compressors or turbines. This is also true under normal working conditions, especially for radial or axial compressors heavily loaded, handling gases of high density. In this respect research work has been carried out on radial boiler feed pumps submitted to high pressure per stage (up to 100 bar, 1,400 psi) [7], [8]. A particular case was investigated for high lift radial pumps, for which high pressure fluctuations were taking place due to the superposition of pressure waves leaving each diffusor. The pressure fluctuations could be such as to endanger the adjacent piping and the impellers [9]. Similar phenomena could probably take place in axial compressors handling high density gases.

As stated by the authors in their paper 'Case Histories of specialized Turbomachinery Problems' [10], vibration problems in rotating equipment have the many aspects of the classic murder mystery 'Who did it and why?' To answer these questions, measurements of blade vibrations together with the detection of the flow pattern responsible for it are necessary, but it remains often the important question to be answered: What should be done to avoid the recurrence of difficulties? At this stage of our knowledge, this paper will rather draw the attention of engineers on particular vibration problems than give an answer toward their solution.

BLADE VIBRATIONS—MEASURING METHODS

Since the stationary or rotating blades will exhibit a different damping (logarithmic damping decrement) which is equal to the logarithm of the ratio of two consecutive amplitudes, it is clear that when the blades are submitted to the same intensity of excitation, the amplitude response, and therefore the alternating stresses will be different from one blade to the other. This is due to the inherent variations in blade fixing, related to tolerances or blade root surface finish for example. It would therefore be of interest to develop a measuring method to detect blade vibrations such that all the blades of one stage could be tested. Today's techniques, however, are confined to methods with which only a limited number of blades can be measured. Measurements of the blade deflection by laser or induction may, however, lead to the solution of this problem.

The various measuring methods will be rapidly reviewed.

a) *Strain Gages*

In this case one or several strain gages are stuck on the blade surface. For the first bending, for example, one gage is placed on the convex side of the profile.

The alternating bending stress σ_w is proportional to the measured strain ϵ

$$\sigma_w = \epsilon \cdot E = E\,k\,\frac{\Delta R}{R}$$

in which ϵ is the strain at the measured point. $\epsilon = \frac{\Delta l}{l}$.

R is the strain gage resistance and ΔR is its variation. The factor $k = \epsilon / \frac{\Delta R}{R}$ is usually close to ½ as can be easily shown.

For stresses τ due to torsion, the gage is placed at 45° to the axis of the torsional moment. The following relation is used,

$$\tau = \gamma \cdot G = \epsilon \cdot \frac{E}{2\,(1+\upsilon)} = E\,k\,\frac{\Delta R}{R}\;\frac{1}{2\,(1+\upsilon)}$$

since $\gamma = 2\epsilon$ and $G = \frac{E}{2\,(1+\upsilon)}$ ·

The signal transmission was realized long ago by the use of brush transmitters. Some high speed versions used up to 20,000 rpm (Vibrometer Ltd.) were based on mercury contacts. However, these apparatuses are delicate and limited in their use. Today, one prefers the contactless system by micro radio senders placed directly on the rotor of the machine. A fixed antenna is placed around the shaft at a suitable place. The power supply is provided by a battery placed on the rotor and having a limited life of about 24 hrs. For this reason, new power transmission systems are being developed by using contactless induction power transmission to the rotor.

The complete recording chain with antenna amplicators is unfortunately not free from electronic noise which can be as great as an equivalent of 200 to 1,000 bar stresses. Therefore, zero stress checking is necessary to estimate the noise level. The relation between stresses and the measured value $\Delta R/R$ can be determined by a proper static calibration.

b) *Frequency modulated Grid*

This method of measurement, called also the meander method, was first developed by Bristol Siddley around 1960 [11]. The components of the system are a tiny cylindrical magnet inserted in the tip of the blade (or the blade could also be magnetized) and a meander pick-up loop imbedded in an insulating mass placed around the inside of the compressor casing in the plane of the rotating blade to be measured. Figure 2 shows in a simplified form the essentials of this method. The voltage pulse S induced by the magnet M in the meander can be fairly sinusoidal if the grid spacing is chosen correctly. The wave length of these waves varies porportionally with the velocity with which the magnet is moving past the grid, so that such disposition is suitable for determining directly the value a • f (amplitude times frequency = vibration velocity) being proportional to the stress in the blade, as will be shown later. The pulse S is amplified and filtered to eliminate frequency noise and main interferences. The filter output is fed to a limiting amplifier forming a frequency modulated square waves signal. The frequency spectrum of this pulse train includes a constant level signal proportional to the rotor speed. Finally, the vibration signal is isolated by a filter which rejects the

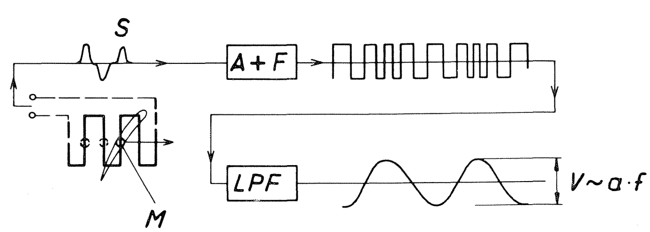

Figure 2. Principle of the meander method.

constant level signal. The vibration signal thus obtained is converted in a low pass filter to a signal of varying voltage, the amplitude of which is proportional to the product a • f. Torsional vibrations can be measured by using two meanders placed side by side and two magnets on the blade tip. The phase difference between the two informations gives the blade angular deflection. This method was successfully applied by us on long thin compressor blades to evaluate the unwinding due to centrifugal forces. A value of about 2° was found.

c) *Laser*

This method of measurement is based on the reflexion of a laser beam by the blade tip itself. The reflected beam is sent to a photo diode giving a pulse at the blade passage. The maximum variation of the blade passage with time is equal to the amplitude of vibration of the blade in the peripheral direction. In order to eliminate an uneven rotation of the machine which could be interpretated as a blade vibration, the signal is compared to a reference signal given by the rotation of the rotor itself. By using two adjacent laser beams, the maximum speed of the vibration can be obtained and therefore the frequency, since the amplitude is also known. Tests were conducted to find out the amplitude and the frequency by the laser method and compared to the classical measurement by strain gages. A dispersion of ± 15% was found, indicating that further work must be done to improve the method. This method has the advantage of leading to minor modifications of the machine and to the possibility of selecting various blades in a given stage. This is important if one remembers that the safety margin of a blading against failure is given by the most loaded blade.

d) *Induction Pick up*

This method, which is also in the development stage, is identical to the laser method. The phase shift of the pulse induced by a given preselected blade is recorded. The total dispersion corresponds to the amplitude. By using two stations close to each other, the speed of the vibration can also be measured.

ALTERNATING STRESSES

Let us first find the relation between alternating stresses and the characteristic values of the vibrations for a simple cantilever. The frequency is proportional to

$$f \simeq \frac{1}{l^2} \cdot \sqrt{\frac{E\ I}{\rho\ F}}$$

in which f is the frequency, l the length, I the moment of inertia, F the cross-section, ρ the specific mass of the material and E the modulus of elasticity.

The bending stress σ_b is equal to the moment M divided by the moment of resistance

$$\sigma_b = \frac{M}{W}$$

The deflection a is proportional to

$$a \simeq M \cdot \frac{l^2}{E\ I}$$

Therefore

$$\sigma_b \simeq \frac{a}{l^2} \cdot \frac{E\ I}{W}$$

introducing (3) into (6)

$$\sigma_b \simeq a \cdot f \cdot \frac{E}{\sqrt{\frac{E}{\rho}}}\ c$$

in which $c = \frac{I\ F}{W^2}$ is a constant, function of the shape of the cross-section. It is interesting to note that the length of the cantilever does not appear in this relation.

Relation (7) shows that the bending stress is proportional to the maximum speed of the vibration a • f.

Writing (7) in the following form

$$\frac{\sigma_b}{E} = \epsilon \simeq \frac{a \cdot f}{\sqrt{\frac{E}{\rho}}}$$

shows that the strain in the blade is proportional to the ratio of the speed of the vibration to the speed of sound in the material.

Experience has shown that these relations are approximately true of blades, regardless of their aerodynamic form and applies to torsional as well as flexural modes of vibrations. Knowing the fatigue limit of various material in bending, a corresponding a • f values can be defined

Aluminum	1.7 m/s
Steel (13% Cr)	2.0 m/s
Titanium	3.4 m/s

A very important parameter in blade vibration is the logarithmic damping decrement δ [12], defined by the relation

$$\delta = \ln \frac{A_1}{A_2} = \vartheta \cdot T$$

called the damping coefficient and describes the envelope in which A_1 and A_2 are two consecutive amplitudes, ϑ is $e^{-\vartheta t}$ of the vibration. It is related to the damping factor β entering into the vibration differential equation by the relation $\vartheta = \frac{\beta}{2m}$, in which m is the reduced mass of the blade. T is the period of vibration.

The damping decrement δ is directly responsible for the amplitude of the vibration of the blade in or out of resonance. In resonance, the amplification factor, which is the ratio of the amplitude to the static deflection of the blade under the forces of excitation, is given by

$$\text{amplification factor} = \frac{\pi}{\delta} \qquad (10)$$

Let us now consider a blade in resonance submitted to an aerodynamic excitation. The static bending stress σ_b due to the aerodynamic excitation is equal to

$$\sigma_b = H \cdot S \cdot \sigma_g$$

in which H is a function of the blade form and of the type of vibration, S is the stimulus proportional to a fraction of the static gas bending stress σ_g on the blade, responsible for the excitation.

The alternating stress σ_w in resonance is therefore equal to the static stress σ_b multiplied by the amplification factor $\frac{\pi}{\delta}$ of relation (10)

$$\sigma_w = \frac{\pi}{\delta} H \cdot S \cdot \sigma_g \tag{12}$$

Various values for H and S can be found in the published literature [13]. Out of resonance the amplification factor is given by the resonance curve of the blade.

As an example, for an average value of δ of 1.8% for a rotary blade of compressors and 10% off resonance, an amplification factor of about 5 is found. With a value of H = 0.9 corresponding to the first bending and a stimulus S = 0.1, the alternating stress in the blade is:

$$\sigma_w/\sigma_g = 5 \cdot 0.9 \cdot 0.1 = \pm 0.45.$$

The stimulus S can vary usually from 0.1 to 0.2 according to the intensity of the wake and is also a function of the distance between the blade rows.

Relation (12) can in fact be used for other disturbances exciting blade vibrations in resonance.

DETERMINATION OF THE DAMPING DECREMENT δ

Two measuring methods can be used. The first one is based on the decay of vibrations when the blade is suddenly set free from a given static deflection. The slope of the envelope of the vibrations is related to the decrement δ.

A better method consists in measuring the resonance frequency f_o as well as the frequency gap ΔF for amplitudes $\frac{a}{\sqrt{2}}$ taken on each side of the resonance value f_o. It can be shown that the damping decrement σ is, with sufficient accuracy, equal to

$$\delta = \pi \frac{\Delta F}{f_o} \tag{13}$$

This expression is easily derived from the general relation giving the amplification factor as a function of δ [12]. This last method was used to determine the logarithmic damping decrement δ of a large quantity of long compressor blades, having an aspect ratio of about 2.5. Figure 3 shows the recorded values of three blades. The damping decrement is function of the blade strain ϵ or stress at the root. The damping is increasing markedly with the alternating stress level. A checking of the damping decrement from the decay of vibration amplitude of rotary blades after surging has given values of the order of 1.8%. This value is in good agreement with those of figure 3. The smaller the aspect ratio of the blades, the greater is the dispersion of the damping values and of the blade frequency. For aspect ratios of about 1, a dispersion of about ± 30% for the frequency can take place. The dispersion is only ± 3% for aspect ratios of about 2. When measured on the rotor without rotation, the dispersion is too great for blades of small aspect ratios. Therefore measurements in the laboratory should be carried out by casting the root in low melting alloys to diminish the dispersion. For stationary blades, which are not submitted to the centrifugal forces, the dispersion is very large. For instance, the damping decrement may vary from 1 to 10%. This shows that stress measurements on a limited number of blades must be considered with great reserve.

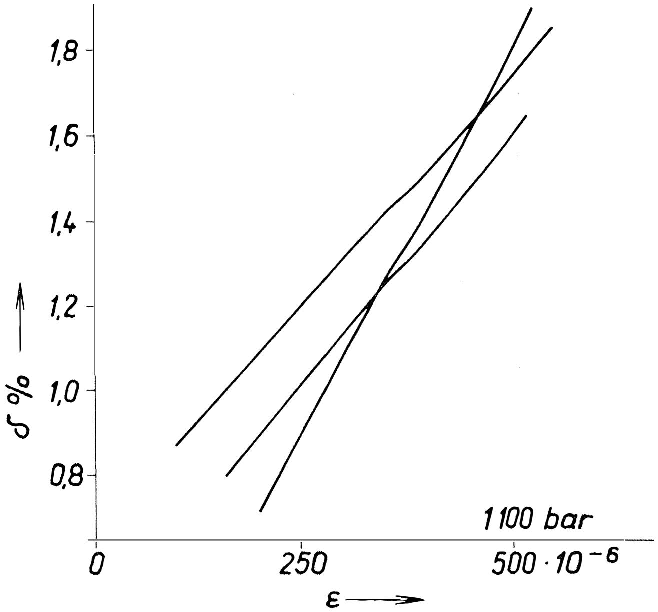

Figure 3. Logarithmic damping decrement of 3 blades.

For flexural modes it is usual to compare the alternating stresses σ_w to the static gas bending stresses σ_g at the normal operating point, since they are directly related as shown by relation (12).

$$\chi_b = \sigma_w/\sigma_g \tag{14}$$

For torsional modes it is convenient to form the following ratio

$$\chi_t = \frac{M_t/s}{M_b/l} \tag{15}$$

in which M_t/s is the equivalent torsional moment inducing the measured shearing stress τ divided by the chord s of the profile, and M_b/l is the bending moment corresponding to the gas bending stress σ_g, divided by the blade length l. In fact, χ_t is nothing but the ratio of two fictive forces acting on the blade. The relation (15) can be written.

$$\chi_t = \frac{W_t}{W_b} \cdot \frac{l}{s} \cdot \frac{\tau}{\sigma_g} \tag{16}$$

in which W_t and W_b are the torsional and axial moments of resistance of the profile. For irregular cross-sections such as blade profiles the value of W_t is usually different and smaller than the polar moment of resistance of the profile [14]. Its value can best be defined experimentally.

BLADE VIBRATIONS IN ROTATING MACHINES

A few typical examples of blade vibration induced by various causes will be discussed in this chapter.

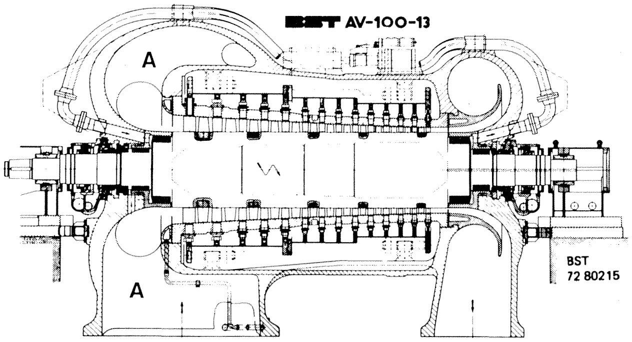

Figure 4. Cross-section through an axial compressor with movable guide vanes.

A = inlet casing vanes.

a) *Supporting Vanes at Compressor Inlet*

A blade failure occured shortly after the commissioning of an 11 MW gas turbine compressor due to rotary blade excitation caused by two supporting walls A placed vertically in the inlet of the casing of the fix blade axial compressor as shown on figure 4, representing an industrial compressor with movable guide vanes.

The speed of rotation was 3,000 rpm ($50\ s^{-1}$). The blade frequency of the first row of rotary blades was situated between 195 and 205 s^{-1}. Although the separation walls of 30 mm thickness were 500 mm from the rotary blade, a failure occurred by fatigue. The rupture was due to the second harmonic of the fundamental disturbance having a frequency of $100\ s^{-1}$. This example clearly shows that harmonics of a disturbance are to be taken seriously into consideration during the layout of a compressor. Due to the degree of reaction of 0.8, the inlet to the rotary blades is axial, and no inlet guide vanes were present.

b) *Partial Injection*

This case refers to the turbine of a SULZER supercharger for a four stroke SULZER locomotive Diesel engine. The 12 cylinders were connected to four groups of nozzles. Two types of partial injection patterns were prevailing. The first one was corresponding to a 180° injection and the second one to two 90° injections separated by two inactive nozzle groups of each 90°.

No difficulty had been encountered as long as the engine was running at its original rating. After uprating by 10%, however, with a corresponding increase in the supercharger speed, fatigue failures occur at the first serration of the axial fir tree root of the blade.

It was soon recognized that the wakes of the nozzle blades could not be the cause of the rotary blade failure, due to the fact that the level of excitation was too low, and resonance for the first bending occurs at a too low speed. Torsional failure was also ruled out, since the speed of the supercharger lies here above the operating range.

A further possible cause was the excitation due to the harmonics of the partial injection. The two types of partial injections idealized by step functions lead to the following Fourier coefficients

$$A(\lambda) = \frac{2A}{\pi\lambda} \quad \text{for } \lambda = 2p - 1 \tag{17}$$

for the 180° partial injection and

$$A(\lambda) = \frac{4A}{\pi\lambda} \quad \text{for } \lambda = 4p - 2 \tag{18}$$

for the 90° partial injections.

In these relations λ is the order of the harmonic, p is any whole number and A a proportionality factor related to the blade load intensity. Resonance between the blade frequency f and one harmonic λ is given by the obvious relation

$$n = \frac{60 \cdot f}{\lambda} \text{ or } \frac{f}{f_n} = \lambda \tag{19}$$

in which n is the supercharger speed in rpm and f_n its speed in rps.

The blade frequency f was measured and was close in operation to $f = 2{,}485\ s^{-1}$. As can be seen, the relation (18) gives the greatest Fourrier coefficient for a given λ (twice as large as relation (17). The corresponding harmonics are 2, 6, 10, 14. . . . The harmonics 2 and 6 correspond to charger speeds outside the operating range. The harmonics 10 and 14 could be the cause of the damage. According to (18), the Fourrier coefficient for the harmonic 10 is of course greater by a factor 14/10 than the one of harmonic 14. Therefore, the harmonic 10 is very likely responsible for the blade failure. A logarithmic damping coefficient of 1.8% was found by preliminary tests on blades fitted in the turbine disc, leading to an amplification factor in resonance of 17.5, given by the relation (10). It is, however, quite possible that in operation, due to the centrifugal force, the damping factor is smaller, and if it approaches 1% as a minimum, an amplification factor of 31 would be reached.

According to relation (19), the supercharger speed would be 14,900 rpm, for resonance with the harmonic 10.

A test was carried out by sticking one strain gage on four different blades. The stresses at the fir tree first serration were calculated, taking into account the stress concentration factor of 1.7 [15]. The results calculated from the strain gage measurements are given on figure 5. As can be seen, a maximum stress of about ± 1,800 bar is found. The fatigue limit of the material under the working temperature of 540°C is about ± 2,000 bar. If one considers that some blades can exhibit smaller damping

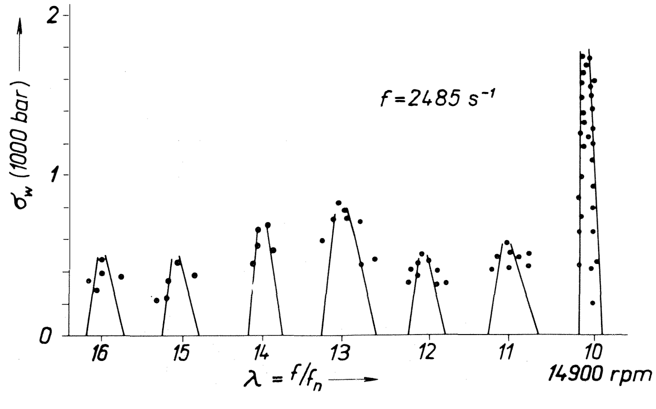

Figure 5. Resonance spectrum of a supercharger turbine blading.

coefficients and therefore higher stresses, it is obvious that the 10th harmonic of the 90° injections was responsible for the blade failure.

The problem was solved by the installation of a lacing wire. New tests after this modification have shown that all the resonances had disappeared. It would have also been possible to increase the fatigue strength by 30% by rolling the blade root groove, but it was questionable whether this strengthening would have been preserved at the high working temperature.

c) *Asymmetrical Inlet Flow Distribution*

This case refers to the compressor blading of a 3 MW gas turbine with separate power turbine [16]. The speed variation of the compressor turbine shaft could vary from 60% to 110% of the normal 7.200 rpm. The static gas bending stress on the rotary blades of the first stage was of the order of 400 bar.

The reason for the stress measurements was that the speed range was extended from 80% to 60%, region in which some blade resonance of the first torsional and second flexural modes were expected. Tests were carried out on three gas turbines by means of strain gages placed on one of the first and second stage rotary blade and on the stationary blades of the stages 1, 3, 4, 6, 7 and 9.

Tests have shown that over the whole operating range, the alternating stresses, due to the expected blade resonances, were acceptable. However, strong peak stress resonances appeared at given engine speeds. Figure 6 shows the alternating stress spectrum σ_w of the first rotary blades. Due to the selected degree of reaction of 0.8, the inlet flow direction to the first rotary stage is axial, and no inlet guide vanes are necessary. After several unsuccessful attempts to improve the situation, it was decided to place 34 axial inlet guide vanes of 44 mm chord length before the first stage rotary blading. A marked improvement in flow distribution was obtained as indicated on figure 6, showing that inlet guide vanes tend to erase the past history of the flow. This influence would still be more pronounced by inlet guide vanes with flow acceleration, as is the case for compressors with a degree of reaction of 0.5.

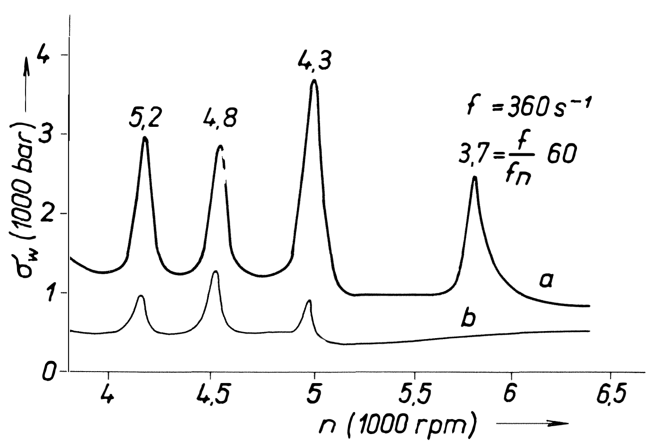

Figure 6. Resonance spectrum of the first stage rotary blade of a gas turbine compressor.

a = without inlet guide vanes.
b = with inlet guide vanes.

The fact that blade resonance occurs at values which are not a whole multiple of the speed is rather unusual and leads to think that the disturbance is varying with time either in amplitude or in position. The excitation with varying amplitude is more likely to occur.

Let us first assume that the disturbance is distributed periodically around the periphery of the inlet channel, with a periodicity k. The rotating blade of angular velocity ω would be submitted to a disturbance of the form

$$A = C \sin(k\omega t) \tag{20}$$

If the amplitude C is also varying with time,

$$C = a \sin(m\omega t) \tag{21}$$

in which m can be any number, even fractional, which expresses how many times the pulsation takes place during one revolution.

The final relation is therefore

$$A = a \sin(m\omega t) \cdot \sin(k\omega t) \tag{22}$$

which can be written

$$A = \frac{a}{2} \cos [(k\text{-}m) \omega t] - \frac{a}{2} \cos [(k+m) \omega t] \tag{23}$$

showing that the disturbances of frequency $\frac{1}{2\pi}$ (k-m)ω and $\frac{1}{2\pi}$ (k+m)ω are present. Referring this fraquency to the frequency of rotation, one obtains the frequency ratios (k−m) and k + m) which can be different from a whole number, as indicated by the measurements. Such excitation could well be due to unstable flow pattern induced by the suction casing.

Measurements on the second stage have shown that the intensity of the disturbance was about 4 times smaller.

Uneven flow distribution at the inlet of axial compressors could have a marked influence on the running stability of the rotor, especially for high density mediums. The disturbance could be due to the unstable bad mixing of two gas streams of different density at the inlet of the compressor.

d) *Wake of Inlet and Diffusor Blading*

Tests have shown that a blade row can be markedly excited by a down stream blade row. A measurement of blade excitation was made on a large axial fan having aluminum rotary blades of 3.7 aspect ratio. The number of rotary blades was 9, and the length was 670 mm. The number of inlet guide vanes was 14 and of diffusor guide vanes 10. Marked amplitudes were observed at 270, 325, 390, 500 and 1,000 rpm.

The resonance at 270 and 390 rpm can be explained by the excitation due to the inlet and the diffusor blades. At these speeds the blade frequency was 62 and 65 s^{-1} respectively. The other resonances could be explained by the superposition of the two excitation frequencies due to the inlet and discharge guide vanes. This combination leads to frequencies being induced by the difference or the

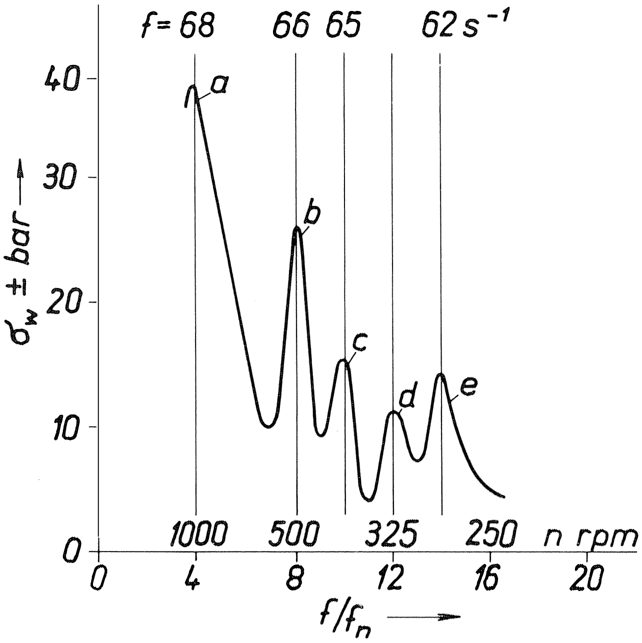

Figure 7. Resonance spectrum of the rotary blades of an axial fan, due to:

a = inlet discharge guide vanes,
b = second harmonic of (inlet–discharge) guide vanes,
c = two discharge guide vanes,
d = third harmonic of (inlet–discharge) guide vanes,
e = inlet guide vanes.

summation of the number of guide vanes. Therefore, the excitation would be 4 and 24 times the speed of rotation. The value 24 is to be rejected as being too high to produce resonance, but the value 4 times the speed explains the excitation at 1,000 rpm (blade frequency 68 s^{-1}) and at 500 rpm (blade frequency 66 s^{-1}). At this speed of 500 rpm the second harmonic of the disturbance is probably involved. Figure 7 shows the recorded values.

e) *Influence of Rotating Stall, Surging and Guide Vane Setting on Blade Stresses in Industrial Axial Compressors*

Figure 8 gives a dimensionless pressure flow diagram of an industrial axial compressor with movable guide vanes for flow regulation. The machine corresponds to a normal compressor used in blast furnace installations for example. The working range is limited by the surge line S, the rotating stall area R and the maximum pressure limitation P.

It is of interest to estimate the operating safety marging of axial compressors by measuring the order of magnitude of the alternating stresses induced in the rotary and in the stationary blading by normal working conditions.

For this reason, a research program on industrial compressors has been initiated. Since differences were observed in the stress level between the different machines, the presented information gives average measured stress values. Measurements were made either by the strain gage or by the meander methods.

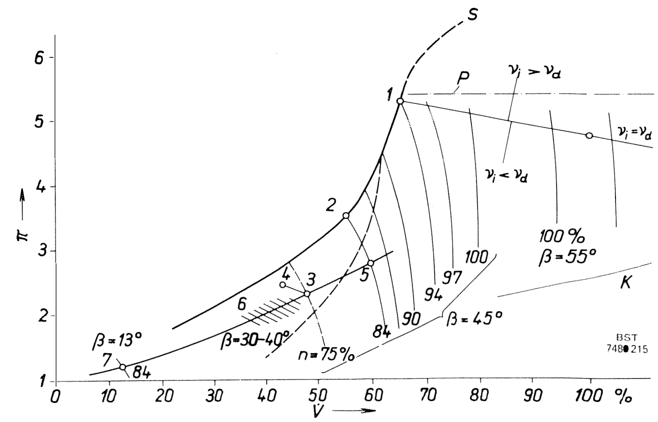

Figure 8. Pressure flow characteristics of an axial compressor with movable guide vanes.

In this presentation β is the guide vanes' setting angle to the tangential direction, ν is the flow coefficient defined as the average axial flow speed, divided by the peripheral speed at the rotor surface, and n is the speed of the machine in rpm. Mu is the dimensionless speed coefficient, expressed as the rotation of the peripheral speed of the rotor hub, divided by the speed of sound of the gas at this location.

The gas bending stress level of the rotary blades at the normal operating point was of the order of magnitude of 300 bar for the first stage being the object of this presentation.

1. Rotating Stall

Measurements of the pressure fluctuation after the first stage of an axial industrial compressor were carried out at two locations around the periphery of the channel. For a working point ③ in figure 8, a periodical change in amplitude was observed as shown on figure 9. The pressure measurements at the two locations A and B around the periphery of the channel reproduce the time lag due to the angular distance of the two pressure gages

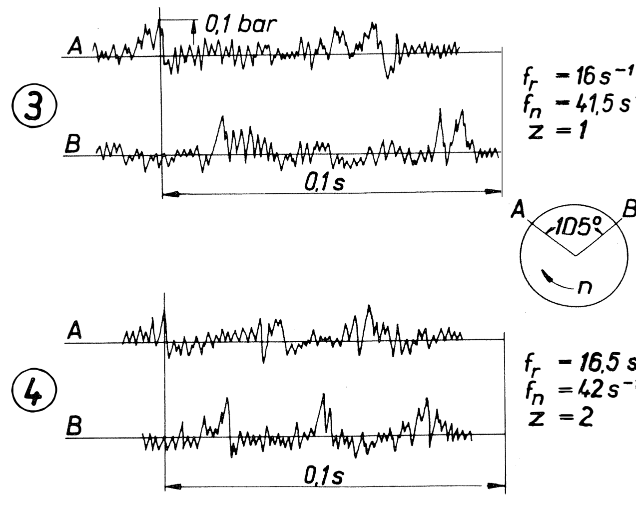

Figure 9. Rotating stall pressure measurements.
③ and ④ working points in figure 8.

(105°). This information clearly shows that one rotating stall cell is moving at the frequency f_r of 16 s^{-1}. the frequency of rotation f_n was equal to 41.5 s^{-1}.

As the working point is approaching the surge line, point ④, the number of cells changes suddenly to two as shown on the same figure. Here again, the time lag between the measurements at A and B is found to correspond to 105°.

It is known [17] that the number of cells increases with the degree of penetration in the rotating stall area. At the surge line the cells invade the complete channel width and are almost spread uniformly around the periphery.

The induced stresses in this area were of the order of ± 800 bar, and the frequency corresponds to the first bending of the blade $f_s = 272\ s^{-1}$.

The following relation exists at resonance with the blade frequency f_s and the number of rotating stall cells z, the rotating frequency of the cells f_r and the frequency of rotation f_n,

$$f_s = \frac{1}{2\pi}(\omega_n - \omega_r)\, z = (f_n - f_r) z \qquad (24)$$

As can be seen, this relation is not satisfied and resonance does not take place here.

2. Surging

Figure 10 shows the first stage rotating blade stress when approaching the working point ② of figure 8. A great turbulence, due probably to flow separation on the blades or rotating stall, is announcing surging. The pressure fluctuations are, however, small and rather uniformly distributed. The blade stresses then suddenly diminish after reaching very high values. This phenomenon is accompanied by a rapid pressure drop behind the rotating blades. A few seconds later, another phase of high stress amplitudes is recorded, which could correspond to backflow or to a new surging process. However, it is interesting to note that this high stress level is not accompanied by a corresponding large pressure fluctuation, as was the case first. This fact leads us to think that the flow pattern is not similar.

At point ①, figure 11, surging was induced by a small speed variation. Here again a rather high stress level is recorded. The frequency of the alternating stresses is, for all cases, always corresponding to the first bending of the blade.

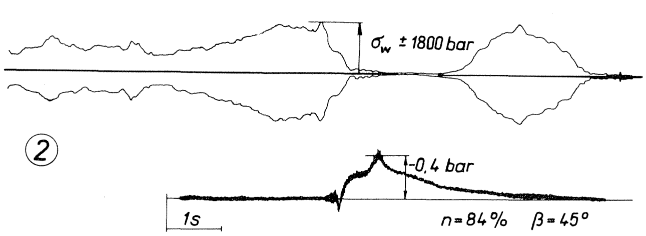

Figure 10. Alternating bending stresses and pressure fluctuations during surging of the first stage rotary blade of an axial compressor.

② *working points in figure 8.*

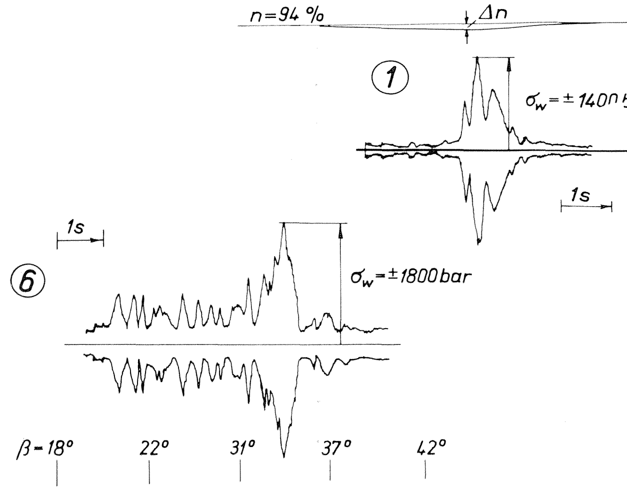

Figure 11. Alternating bending stresses first stage rotary blades of an axial compressor.

① *working point in figure 8, surging.*

⑥ *working region in figure 8. opening the guide vanes.*

Along the surge line, between the points ① and ② of figure 8, the calculated flow coefficients ν_i at the inlet as well as ν_d at the discharge of the compressor were plotted on figure 12 for a given guide vane setting corresponding to three different machines. As can be seen, the ν_i values are rather constant when going from point ② to point ① by varying the dimensionless speed $M_u = U_n / \sqrt{kRT_i}$. The discharge flow coefficient ν_d is in the contrary rather variable. The relative values of ν_i and ν_d show that surging takes place in the first stages, the other stages having higher flow coefficients. For cases corresponding to working points for which $\nu_i > \nu_d$ and for high pressure ratios, surging may take place in the last stages.

It is very difficult to find out a valid criterion for defining surging. In some publication [18] it is proposed for certain types of compressors to use the average value $\frac{\nu_i + \nu_d}{2}$ = constant as a criterion.

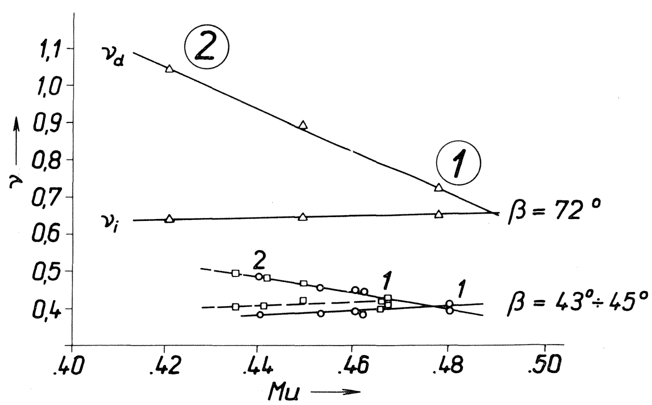

Figure 12. Dimensionless flow coefficients ν at surging between points 2 and 1 in figure 8, as a function of the speed Mu, for 3 guide vane setting β.

Due to the high stress level, prolonged surging will cause blade failure. Since the excitation is very complex, all the blades of one stage could be excited, regardless of the frequency dispersion. This is not the case or excitations due to pure resonance, for which only the few blades having the right frequency can be excited in resonance. Failures due to prolonged surging, where the first five stages were involved in fatigue failure, are reported in the technical literature [19]. Surging will also cause great temperature increases in gas turbines' combustion chambers due to rapid flow reductions [20].

Below the chocking line K of diagram 8, the ν_a values are much higher than at the normal point, since the pressure ratio is low. The ν_i coefficients, however, correspond almost to the layout value, since the inlet volume flow remains practically unchanged, due to the steep head-flow characteristics. In this region, the last stages could therefore be submitted to alternating stresses induced by flow separation in the blading. Tests carried out on a laboratory compressor have shown that the stress level is still acceptable.

3. Blade Vibrations due to variable Guide Vane Angle

The tested machine corresponds to the design shown on figure 4. Movable guide vanes are used for flow regulation. The degree of reaction of the machine is close to 0.5. Therefore, inlet guide vanes before the first rotary blades are necessary to deviate the inlet axial flow to the suitable angle. During flow regulation and at extreme angular positions, the incidence of the flow to the leading edge of the guide vanes departs considerably from the ideal value. This could lead to flow separation and to rotary blades' excitations. Tests were carried out along the line 5-7 of figure 8 at 84% speed. For small guide vane angles, point 7, the alternating stresses are relatively small. In the region 6, for blade setting angles of $\beta = 30° - 40°$, a sharp increase in stresses was noticed, as shown on figure 11 (working area 6). Such excitations, due to the guide vane setting, are mentioned in the technical literature [21] related to aircraft engines. It was believed in this particular case that blade flutter, due to detrimental operating flow incidence, was responsible for the observed marked increase in blade stresses. However, for the industrial compressors tested here, it is very unlikely that this might be the cause of the observed sharp stress increase. It is believed that a brutal appearance of flow separation in the guide vanes might be the reason.

It was found that the stress level diminishes with the rapidity at which the opening or closing of the guide vanes is made. The peak values diminish by about 30% if the total blade angle variation from 13° to 45° is made within 4 sec instead of 12 sec for example.

POSSIBLE MEASURES AGAINST BLADE EXCITATION

Based on the previous study, it seems advisable to try to diminish the alternating stress level due to surging, rotating stall and guide vane setting during starting. For compressors having pressure ratios greater than 5, it is recommended to include in the design an intermediate

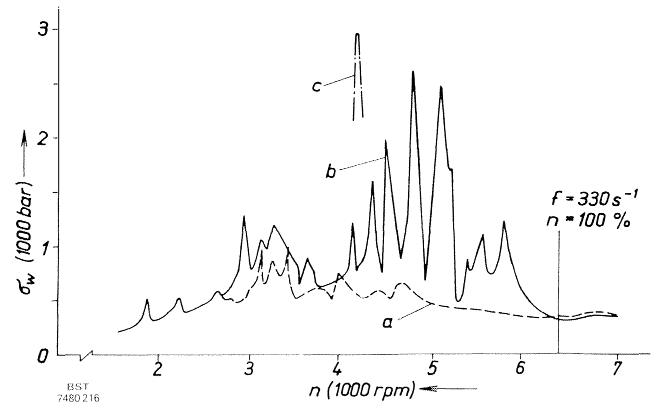

Figure 13. Influence on the rotating stall and surging of an axial compressor of a gas turbine at shut down.

b = intermediate discharge valve closed (b), open (a).
c = approaching surging slowly.

discharge, in order to diminish or even eliminate rotating stall. Figure 13 shows the alternating stresses due to rotating stall and surging during the normal shut down of a 10 MW gas turbine having a pressure ratio of 7.5. Curve b corresponds to the case with the intermediate discharge valve closed. When the valve is open, the alternating stresses are given by the curve a. Curve c gives higher values when the speed of the machine is reduced very slowly towards surging.

A lacing wire on the rotary blades could of course drastically reduce the alternating stresses due to rotating stall or surging. This method was used in older designs, and tests have shown that the alternating stresses are disappearing completely. The loss in efficiency for a stage of radius ratio of about 1.5 is of the order of 5%, introducing a total efficiency loss of about 0.5 to 1% for compressors having a compression ratio of about 5. The lacing wire also contributes to increase the noise level in the machine by about 5 db.

Some interesting information is published regarding the noise intensity as a function of the ratio blade chord to blade spacing [22]. In most of industrial compressors this ratio is situated between 0.3 and 0.5. The difference in noise intensity is of the order of 1 to 2 db.

PROTECTION OF AXIAL COMPRESSORS

The alternating stress measurements presented in this paper have shown that an axial compressor can be submitted to adverse working conditions inducing high alternative stress levels during short periods of time. The cumulative effect of fatigue can lead to blade failure after many years of operation. The following measures or protections are recommended.

a) Since the fatigue strength of blades is diminished drastically with corrosion, periodical checking must be made, and the corroded blades must be removed.

b) About 80% of the blade failures by fatigue are concentrated on the first stage in industrial axial compressors. Since the cumulative effect of fatigue can take many years to cause failure, it is recommended to inspect during routine maintenance the blading of the first stages by Magnaflux in order to detect incipient cracking.

c) A surge counter is recommended. An estimation of the cumulative effect of high stress levels is then possible.

d) If the anti-surge by-pass valve does not open, surging will occur. An alarm should attract attention, and the machine should be shut down if surging persists after 5 sec for example.

e) A detection of the blade amplitude by laser or other means, capable of changing the working condition of the machine or of causing its shut down if excessive amplitudes are present, should be developed.

f) For compressors with movable guide vanes the minimum setting angle for flow regulation should be limited, and the opening of the blading during starting should be made as fast as possible.

CONCLUSIONS

This presentation has shown that axial compressors are submitted to adverse working conditions which may endanger the machine by causing fatigue of the blading. The stress values given in this paper represent average values of tests carried out on about six large industrial compressors of various sizes. Some of the recorded values are higher, some are smaller than those published. However, the stress level is high enough to require further investigations to better understand the mechanism responsible for the measured stresses. Tests should be carried out, both in the laboratory and on industrial machines in order to be able to find design modifications or operating sequences diminishing the stress level and thereby increasing the operating safety factor.

The compressor of figure 4, for example, corresponds to an 80 MW gas compressor for LNG liquefaction plants. It is fitted with meander to measure blade vibrations of the first and the sixth stage situated after the anti-rotating stall discharge valve, during commissioning.

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LIST OF SYMBOLS

π	=	pressure ratio
V	=	volume flow in %
M	=	torque
n	=	speed rpm
a	=	amplitude of blade vibration
f	=	rotating stall frequency
f_n	=	blade frequency
f_r	=	frequency of rotation
z	=	number of rotating stall cells
s	=	second (time)
β	=	guide vane angle to the tangential direction
σ_w	=	alternating blade stress ($\pm$) in bar
ν	=	flow coefficient c/u_n (i = inlet,
c	=	d = discharge)
u_n	=	average axial velocity m/s
Mu	=	rotor speed m/s
1 bar	=	speed coefficient = $u_n/\sqrt{\kappa R T_i}$
1 inch	=	14.5 psi
		24.4 mm