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Half-Speed Bearing Whirl Excited by a Single Propagating Stall Cell in a Multi-Stage Axial-Flow Compressor

By J. RINGROSE

National Gas Turbine Establishment

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Summary.

A plain journal bearing is spontaneously unstable above a critical speed and loses much of its load carrying capacity when subjected to a load rotating at about half the speed of the shaft. The almost axially symmetric flow in a compressor can break down at low flow into a rotating pattern of stall cells but only the special case of a single propagating stall cell gives rise to a mechanical couple about an axis normal to and rotating about the axis of the machine.

It was observed in a multi-stage axial-flow compressor that the propagational speed of a single stall cell coincided with the dynamic weakness of the two plain bearings supporting the rotor and resulted in severe vibration at about half the shaft speed. This was interpreted as an excited half-speed bearing whirl and is believed to be a unique observation.

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*Replaces NGTE R.298 (A.R.C. 30 293).

LIST OF ILLUSTRATIONS

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1. *Introduction.*

It has been known for many years that a plain journal bearing becomes spontaneously unstable if the shaft speed is increased sufficiently far. A good summary of leading references is given by Marsh[1] who develops a general theory of the stability of self-acting gas-lubricated bearings. Half-speed whirl is the common instability, at the onset of which the centre of the shaft describes an approximately elliptical orbit about its previous steady location in the bearing. The orbital frequency is often nearly half that of the shaft rotation and is in the same direction, but frequencies as low as a seventh and also above a half have been reported so that the term used to describe this form of instability does not imply a precise relationship; rather it arises from an approximate theoretical treatment for single-phase compressible lubricants.

Many observations have been made of *spontaneous* half-speed whirl but it is also a characteristic of a plain journal bearing that it loses much of its load carrying capacity when subjected to a load rotating at a frequency near to half the shaft speed. The bearing becomes soft. It is however unusual for accidental mechanical excitation at the critical frequency to be present in a rotating assembly.

It is customary to portray the overall characteristics of an axial-flow compressor in the mass-flow rate, pressure-ratio plane by drawing lines representing the performance at a number of constant rotational speeds (Figure 1). At any one speed, if the throttle is slowly closed so that the pressure ratio rises, there comes a point at which no further increase in pressure ratio can be achieved and in many systems violent oscillations (surging) in the airflow occur. The locus of these points is the surge line which divides the mass-flow rate, pressure-ratio plane into two regions: one, below the line, in which each point represents a physically attainable combination of mass-flow rate and pressure ratio: the other, above the line, in which points do not correspond to physically attainable states.

Propagating stall is a familiar phenomenon, occuring somewhere in the operating range of most axial compressors. In multi-stage axial compressors steadily propagating stall is often observed at speeds lower than that corresponding to a kink in the surge line as shown in Figure 1. The number of cells depends on the speed and pressure ratio but as a rule the higher the values taken by these two variables the lower is likely to be the number of cells, except that a *single* cell appears to be fairly rare. The rate of propagation has about the same magnitude but opposite sign, relative to the stator or the rotor so that the cells move at about half the shaft speed. If there are several stall cells then they tend to be distributed with maximum symmetry and the consequence that the only arrangement which lacks more than one diametral axis of symmetry is the single cell.

A 13-stage axial compressor (Figures 2 and 3) of conservative design with fixed geometry, except for the inlet guide vanes, is used for multi-stage investigations at N.G.T.E. The rotor is carried in two oil-lubricated journal bearings (Figures 3 and 4) and is driven by a steam turbine through a fairly long shaft with no intermediate bearings. Part of the drive shaft is sometimes replaced by an optical torquemeter and this also has no intermediate bearings. There is a flexible diaphragm coupling between the drive shaft and the compressor (Figure 5).

It has been observed that the compressor gives rise to severe vibration when its operating state (pressure ratio, speed, mass flow) lies in a small zone (Figure 6) near to surge at speeds just less than that corresponding to the sharp kink which the surge line exhibits in the mass-flow rate, pressure-ratio plane*. The phenomenon is metastable, for the boundaries of the critical zone show considerable hysteresis and the zone may not even be found at all when the locus of the operating state moves in some directions. A hypothesis explaining the severe vibration, the planning of an experiment around the hypothesis and the achieved results will be described.

2. *A Hypothesis Explaining the Source of Severe Vibration.*

It was readily established by mounting seismic transducers at numerous places on the test rig that the vibration was felt as an elliptical orbit of the rig as a whole at about 40 Hz which corresponded almost exactly to half the shaft speed. This low frequency did not correspond to any natural period of the mechanical system: stiffnesses were too high in relation to the suspended inertias to respond at 40 Hz. The dependence of the presence of vibration on the pressure ratio even at constant rotational speed suggested that an aerodynamic process was either responsible for or closely connected with the behaviour.

The locus of the operating state lies in a surface in pressure ratio, speed, mass flow space if all changes take place sufficiently slowly and the severe vibration is experienced in a critical zone in which, comparing Figures 1 and 6, it may be expected that the compressor is stalled. If a stall *cell* is present the axial component of aerodynamic force can be less on a blade immersed in the cell than on one in attached flow (Figure 7). An axial view of a rotor row containing a single stall cell is shown in Figure 8, wherein the size of each cross is indicative of the magnitude of the axial component of the aerodynamic force. It is clear that the stall cell gives rise to a couple about a diameter of the row and that the axis of the couple rotates with the stall cell at its propagating speed. The couple will be felt as rotating loads on the two bearings of the rotor shaft. It is clear that only a *single stall cell* can give rise to such a couple and that the presence of more than one diametral axis of symmetry in multiple-cell patterns rules out a resultant couple. The magnitude of the rotating couple can be quite large. If, for example, a stall cell extends through several stages of the compressor under investigation, it has been found that the pressure decrement through the stall cell is of the order of one atmosphere acting on the stalled area of one rotor row. If in addition it is supposed that this area is of the order of a tenth of the annulus area then, relating these figures to the dimensions of the compressor, a couple of 500 lbf ft is generated. This is about a tenth of the shaft torque.

The hypothesis is that all points in the critical zone of the characteristic surface represent a state in which there is a single stall cell propagating at about half the shaft speed, that at least one of the two plain bearings is dynamically soft to excitement at the stall propagating frequency and allows the rotor to move in a relatively large orbit giving the effect of an unbalanced force rotating at about half the shaft speed.

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*The shape of the surge line is typical of an undeveloped multi-stage axial-flow compressor with fixed geometry.

The accessible region of the characteristic surface not including the critical zone represents states in which either there is no stall, the stall is disordered or the stall pattern has greater symmetry than the single cell. Each of these alternatives fails to provide a suitable excitation and the bearings remain stiff and locate the centre of inertia of the rotating assembly.

3. *Experimental Design.*

It is recognised that this hypothesis cannot exclusively be proved to be true. It is possible only to design critical experiments and show that the observations do not contradict the hypothesis.

The orbit of the centre of a shaft supported in a plain bearing is not difficult to display. Two capacitative non-contacting probes were fixed rigidly to the bearing housing and connected to Wayne Kerr vibration meters so that the separation between the shaft surface and each probe was translated into a linearly proportional electrical analogue. Probes were chosen so that the full range of expected shaft movement would be accommodated in their linear range; one probe observed vertical displacement and the other horizontal (Figure 9). If the shaft were a circle concentric with the inertial axis during *steady* rotation, then the separation between the surface and either of the probes would be constant; any out-of-roundness or eccentricity would be reflected in a periodic variation in separation at the same frequency as the shaft rotational speed. Two consequences arise. One is that out-of-roundness and accentricity will always be present to give rise to an obscuring interference with any more interesting signal and therefore that the portion of the shaft viewed should be chosen or speedily designed to minimise the effects. The other is that the primary interest in this experiment is in cyclic phenomena which are likely to take place at a frequency different from, in fact about half the value of, the shaft rotational speed and that, therefore, the interference due to any residual out-of-roundness and eccentricity should be easily distinguished from the relevant signal.

In the absence of out-of-roundness and eccentricity or after the effect of them has been subtracted, the variation in separation between each probe and the shaft surface, provided this is small compared with the diameter of the shaft, is the same as the movement of the centre of the shaft or axis of steady rotation relative to the probe support which is in this case the bearing housing.

Excursion of the shaft from its average position, translated into electrical analogues representative of an abscissa and an ordinate in a rectangular Cartesian frame may be displayed in two useful ways. One is to use each to deflect the beam of a cathode ray oscilloscope in one of two orthogonal directions so that the motion of the spot on the screen replicates, except for residual out-of-roundness and eccentricity, the motion of the centre of the shaft relative to the bearing housing. The scale of the picture is easily controlled by the amount of electrical amplification and can represent a magnification of several hundred or even thousand times the original mechanical motion. Alternatively each electric analogue may be displayed or recorded independently of the other against an abscissa representing time. Both these means, each having advantages, were used in the experiment.

The presence of a propagating stall cell in an axial compressor is easily detected. A pressure transducer mounted either with its diaphragm flush with the inside surface of the casing or connected by a sufficiently short pipe to avoid distortion by Helmolz or organ-pipe effects will register the variation in static pressure due to the passage of a stall cell. A hot-wire anemometer immersed in the fluid will respond to the velocity changes associated with a passing stall cell. The latter method was employed in the experiment and two hot-wires separated by about $5\pi/6$ in the circumferential direction were mounted in the inter-blade-row space downstream of the second stage of the compressor and projected a short distance into the fluid. A similar arrangement was provided downstream of the third stage. D.I.S.A. constant temperature anemometers were used and the electric analogue signals from the anemometers were displayed against a time base for cathode ray oscilloscopic observation.

All electric analogues, of shaft displacement and stall cell paddage, were recorded throughout the experiment on an Ampex 14-channel magnetic tape recorder. Simultaneous oscilloscopic display was provided for experimental handling. Post-experimental processing of the recordings consisted mainly of replaying and recording in visual form using a multi-channel ultra-violet oscillograph providing a trace on photo-sensitive paper.

4. *Experimental Results.*

It is often observed with multi-stage axial compressors, and was so with this unit (Figure 10), that the surge line exhibits a pronounced kink at about 0·7 of the design speed. The sequence of events leading to surge of the system is different according to whether the locus of the operating point intersects the surge line above (higher speed, flow rate, pressure ratio) the kink or below the kink. If in the former part the blades in the compressor remain unstalled until immediately prior to surge when a small patch of stall (a single cell) grows rapidly to cause a collapse of the pressure raising ability of the compressor resulting in surging of the system. In the latter part there exists a region of the mass-flow rate, pressure-ratio plane adjacent to the surge line representing operating states in which some or all of the blades in the compressor are stalled (Figure 1). Stable progression through the stalled region to the surge line can be accomplished and it is only immediately prior to surge that the stalled state grows unstably and rapidly to result in a collapse of pressure raising ability similar to that initiated from an unstalled state*.

In the stalled region it would be expected from previous experience that if the stall were of the propagating cellular form then the number of cells observed would be smaller the more to the upper right corner of the region the operating state existed.

It was observed that there existed a region in the mass-flow rate, pressure-ratio plane where the compressor exhibited a single stall cell, this region is shown in Figure 10 and was found to coincide with the region of severe vibration. (It was found that if one stall cell was present the compressor was always vibrating and conversely if the compressor was vibrating then always there was one stall cell.) It was also observed that the frequency of vibration of the single stall cell coincided precisely with the frequency of vibration of the compressor and that during vibration and only during vibration the shaft executed a large (0·007 inches diameter) orbit within the bearing at the same frequency. A casual connexion or common cause was indicated and it is suggested that the hypothesis advanced in Section 2.0 provided the mechanism. Detailed evidence for these statements follows.

The critical state (in which the single cell and the vibration were observed) was *almost always* encountered when a locus of varying speeds of rotation and a constant throttle setting was followed, such as that shown as AA in Figure 10. However, sometimes no critical state was observed and the locus apparently passed up a narrow channel between the surge line and the critical region when never less than two stall cells and no vibration were recorded. This channel was so thin that it could not be distinguished from the surge line, for this reason it has not been shown in Figure 10. It was also found that if a locus of constant speed of rotation and a varying throttle setting were followed, such as that shown as BB in Figure 10, then sometimes, if the throttle were closed very gently, the system would surge from the critical (vibrating) state, recover after the first loss of pressure and then remain indefinitely in an unsurging, uncritical state with two stall cells. Further *slight* closing of the throttle would cause irreversible surging, that is, the throttle would have to be opened for recovery to take place**.

The sequence of events in traversing a locus such as AA from high speed to low is depicted in Figures 11 to 14. The first of these shows the patterns produced when the analogues from the two shaft-displacement probes are used as ordinate and abscissa on a cathode ray oscilloscope display. The sequence of three pictures is related to operating states represented on the adjacent characteristic diagram. Figure 12 shows the two stall analogues from the third stage of the compressor and the two displacement analogues on the same oscilloscope with a common time base. As in Figure 11 the right hand picture corresponds to an unstalled state so that the stall-cell analogue gives a small indication; there was no vibration and the shaft displacement relative to the bearing housing results only from residual out-of-roundness and eccentricity and has a cyclic period equal to one revolution of the shaft. The middle picture corresponds to the critical state of a single stall cell propagating at about half the shaft rotational speed which is accompanied by large excursions of the shaft in an orbit traversed at about half its rotational speed and synchronously with the stall propagation. The phase relationship between the two stall analogues,

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*Multiple cells decay into a single cell before a surge.

**It should not be concluded that the single stall cell accompanied by vibration was a difficult state to produce; rather was the two-cell state in the immediate neighbourhood of the surge line rare.

recalling the $5\pi/6$ angular separation of the anemometers, is almost unambiguously indicative of a single cell propagating at about half the shaft rotational speed. The left hand picture shows that the shaft displacement has returned to the residual indication but that there are two stall cells which, in the light of the proposed hypothesis would not be able to excite the plain bearings even though the propagational speed of the cells is almost the same as that of the preceding single cell, half the shaft rotational speed.

The timing of the passage along the line AA was as follows. A stable unstalled operating state was established above the critical region and the speed reduction was initiated. After a short time the critical state was abruptly encountered. This is indicated in Figure 13 which is a reproduction of a tracing of the part of the ultra-violet oscillographic record of events near to onset of the critical state. The almost instantaneous growth of the excursion of the shaft in its bearing accompanying the establishing of a single cell is witness to the very strong forcing which the latter has been shown to be capable of providing.

During the subsequent 24 seconds exact synchronism of the shaft orbit and the stall cell was observed for nearly 900 cycles. There were slight changes in the ratio of the cell propagation speed to the shaft rotational speed during this period but the ratio did not differ much from 1/2 at any time.

After 24 seconds the vibration ceased as abruptly as it had started, accompanying a sudden change from one to two stall cells. Figure 14 shows a reproduction of a tracing of the relevant part of the U.V. recording.

5. *Conclusion and Recommendation.*

It would be very difficult to escape the conclusion that the bearing whirl and hence the vibration of the compressor had a common property with the existence of a single cell of propagating stall. The hypothesis presented was not disproved and does provide a plausible mechanism whereby the asymmetric aerodynamic load on a partially stalled rotor may excite the bearings in precisely that mode to which they are most vulnerable. It has not been necessary to suppose the existence of any other mechanism or property and the causative connexion seems to be the most tenable.

Further evidence in favour of the causative connexion could have been sought but the experiment was terminated by the higher priority of other commitments.

If it were thought necessary to cure the compressor of its vibration it is suggested that a replacement of the plain bearings by roller bearings or even tilting pad bearings which do not respond so readily would alleviate the problem but would not remove the aerodynamic cause. The elimination of the cause has been shown to be possible by adjustment of the stagger of the inlet guide vanes when it appeared that an increase in stagger reduced the area of the critical region and it was possible to remove it altogether by a sufficiently severe adjustment.

REFERENCE

No.	*Author*	*Title, etc.*
1	Marsh, H.	The stability of aerodynamic gas bearings. Mechanical Engineering Science Monograph *2*. The Institution of Mechanical Engineers (1965).

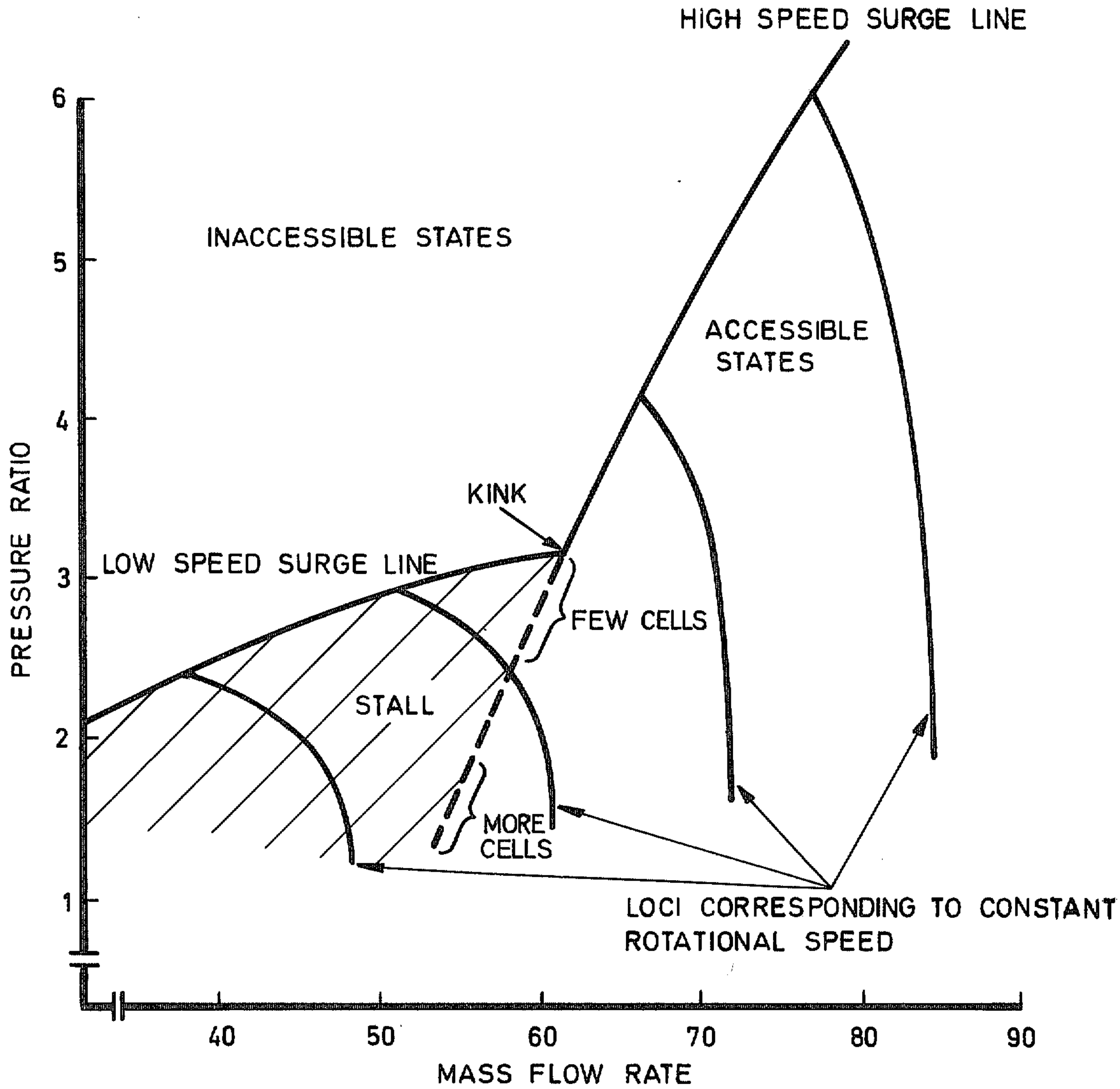

FIG. 1. Surge line and stalled states of a multi-stage axial-flow compressor.

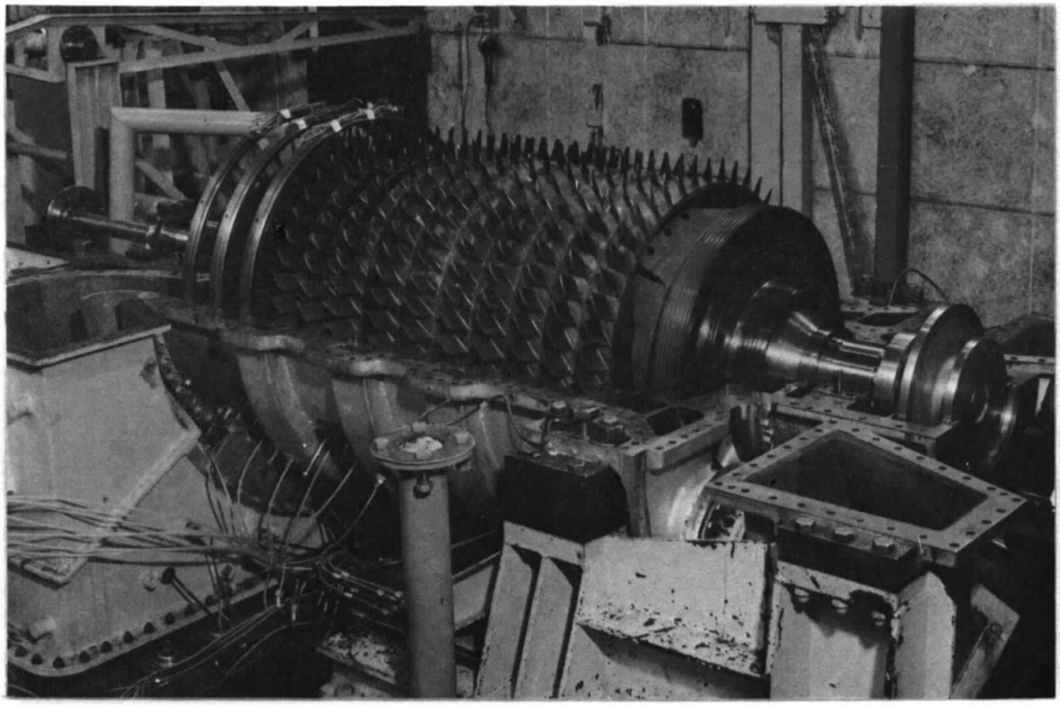

FIG. 2. Compressor rotor.

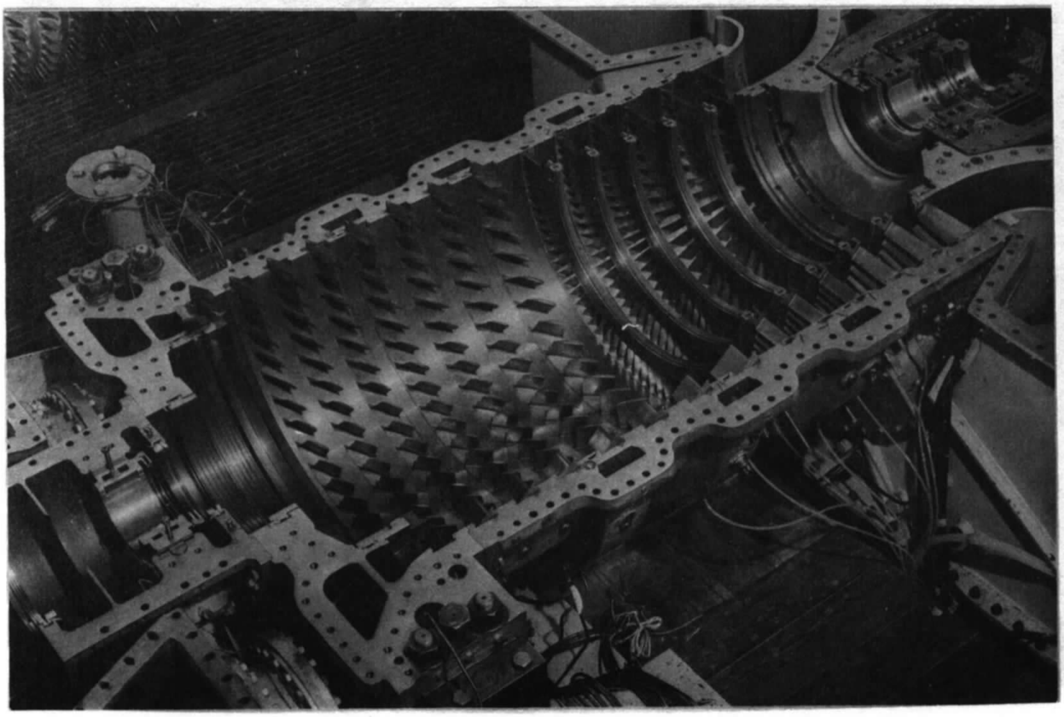

FIG. 3. Compressor stators.

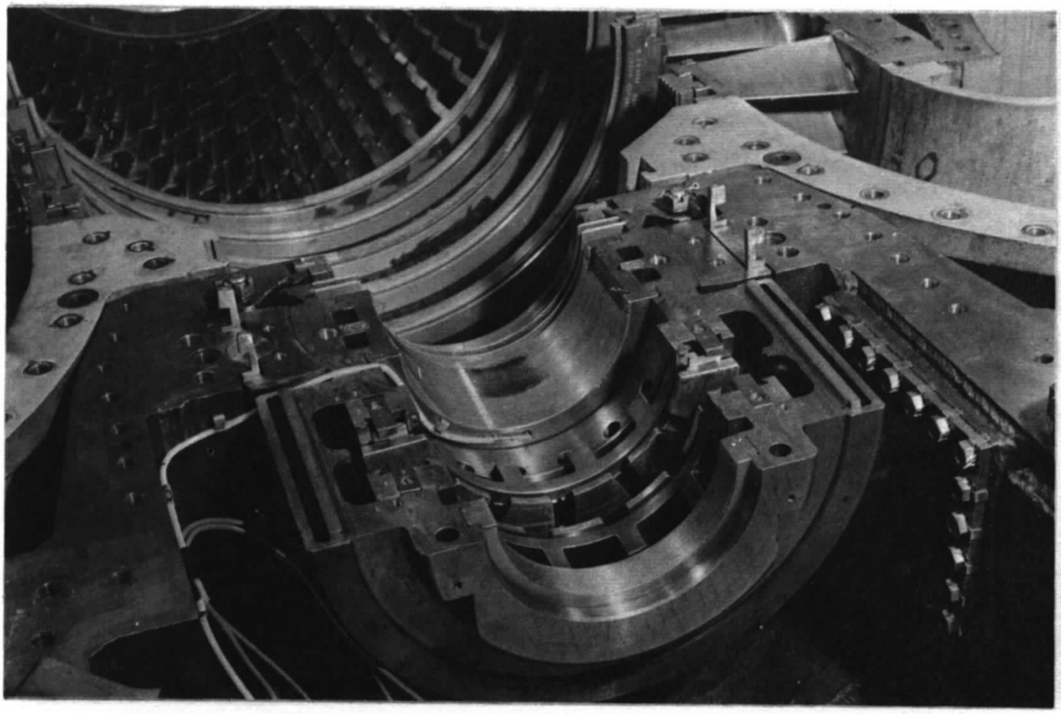

FIG. 4. Journal bearing.

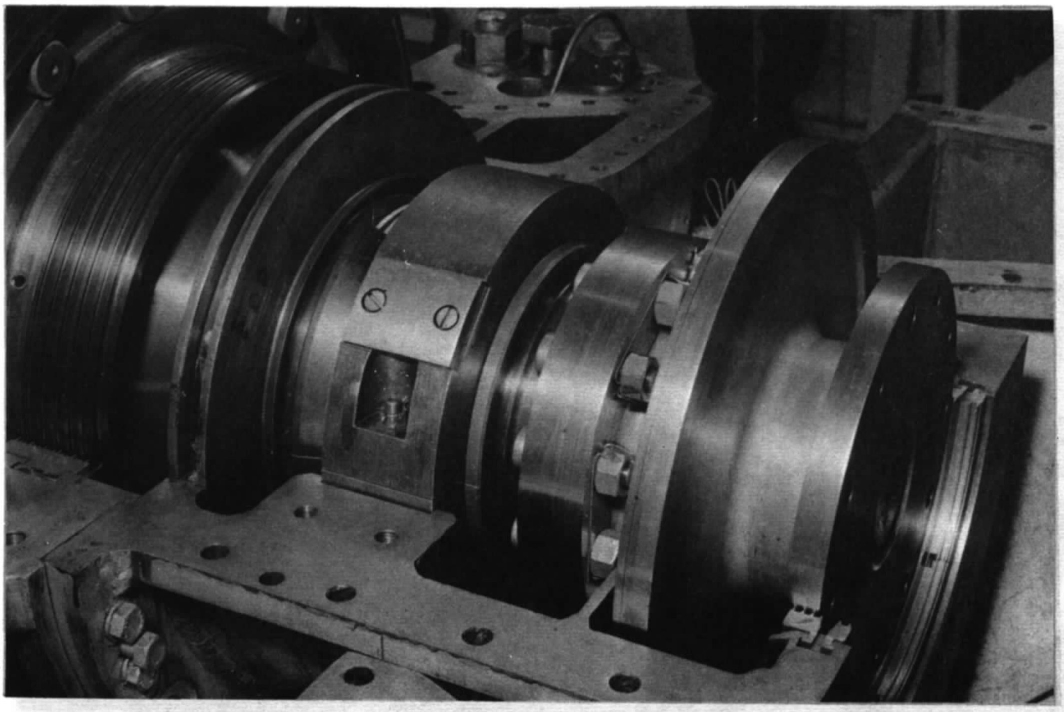

FIG. 5. Flexible coupling.

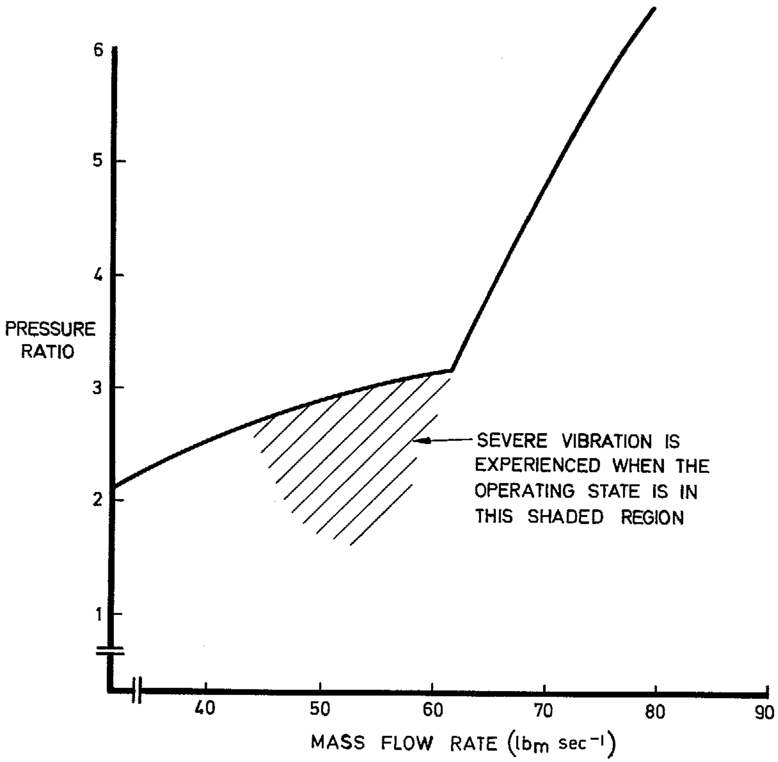

FIG. 6. Part of the mass-flow rate/pressure-ratio plane in which severe vibration is experienced.

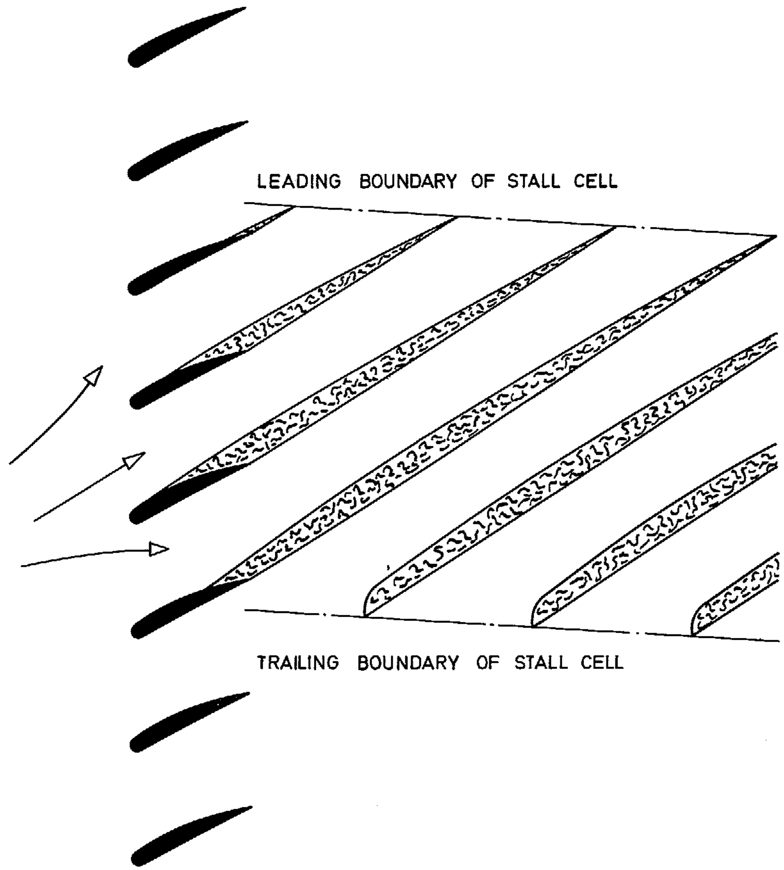

FIG. 7. Stall cell flow pattern.

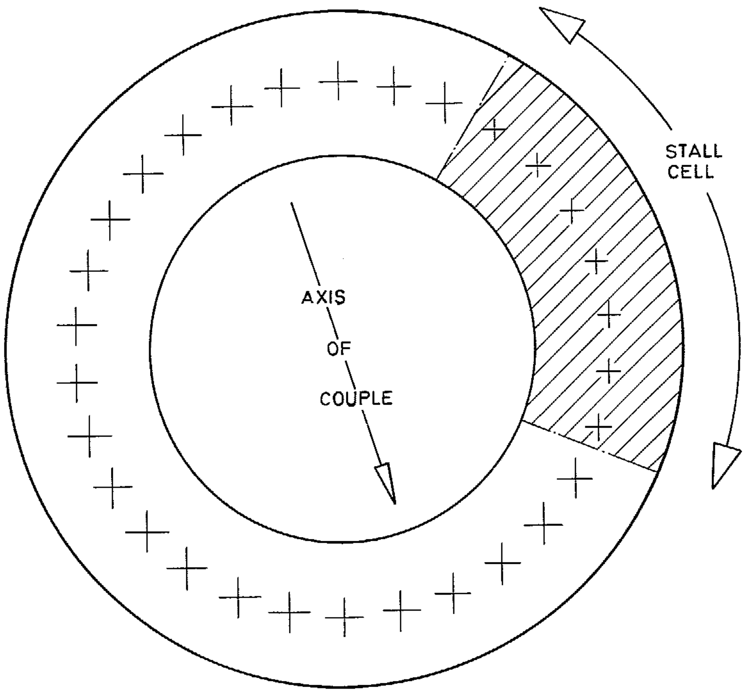

FIG. 8. Mechanical couple caused by single stall cell.

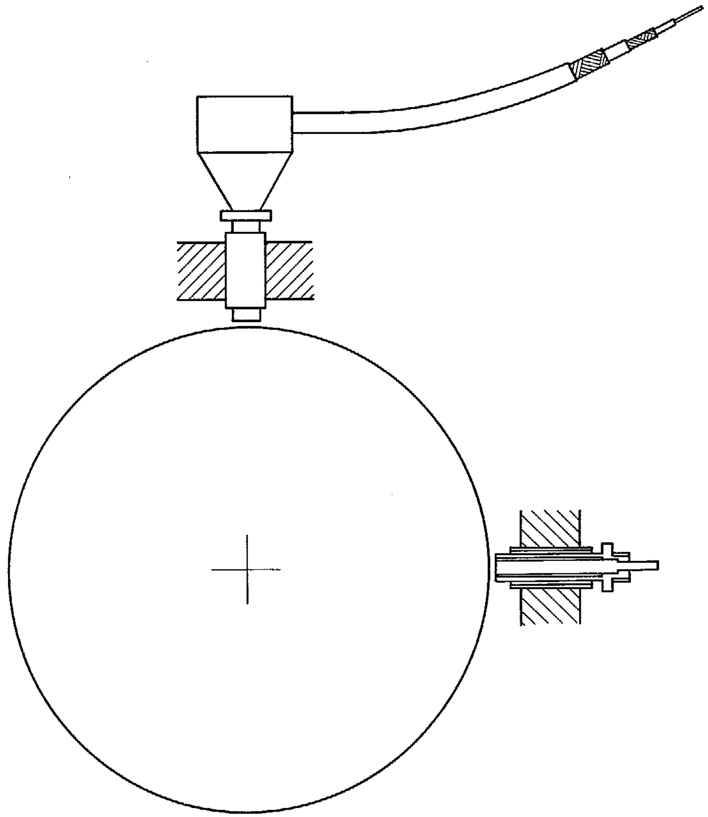

FIG. 9. Locations of vibration probes.

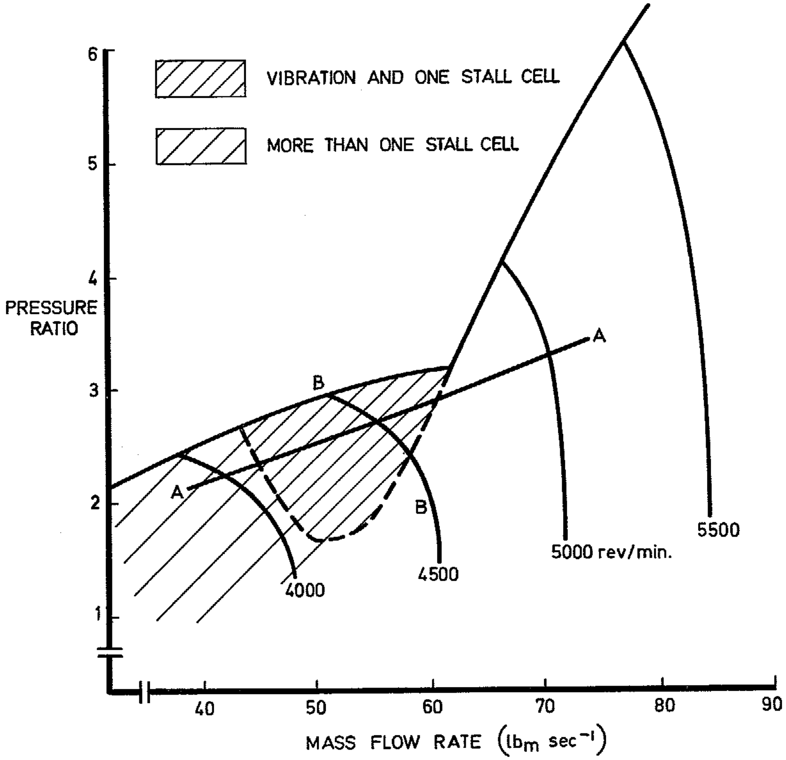

FIG. 10. Surge line and stall-cell behaviour of test compressor.

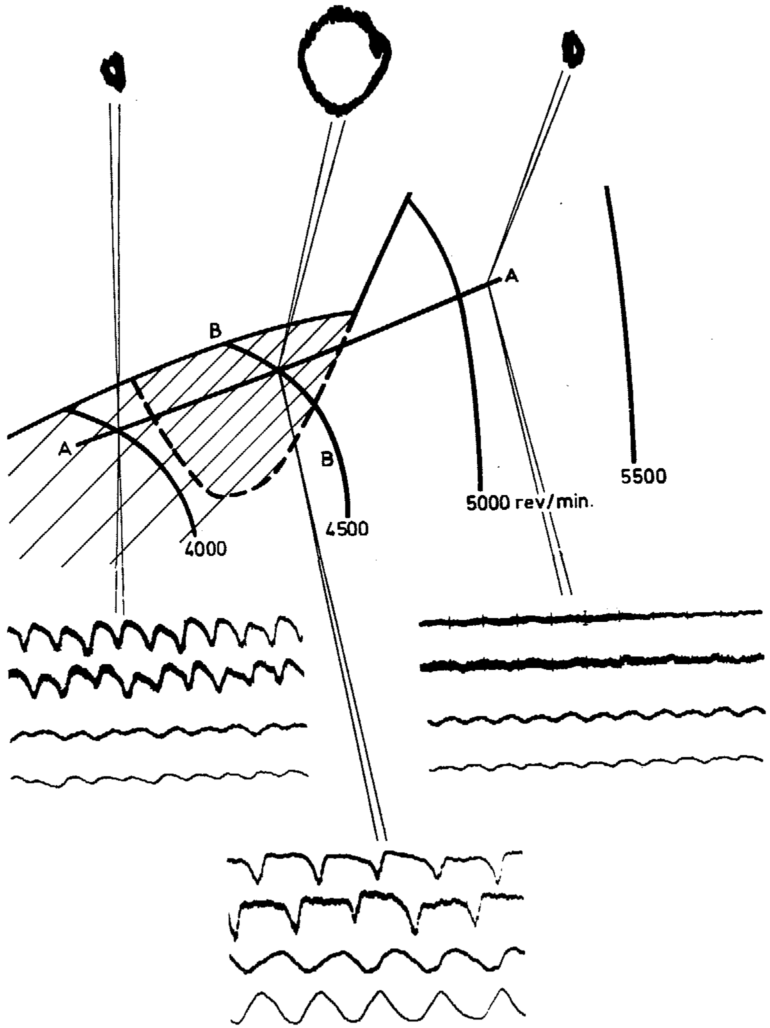

FIG. 11. Orbits of shaft axis.
FIG. 12. Stall and shaft-displacement analogues.

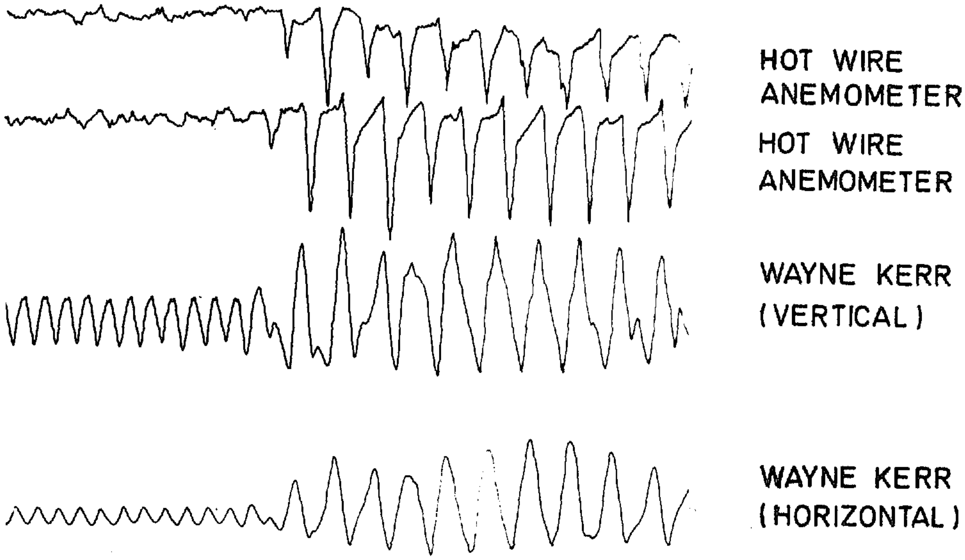

FIG. 13. Onset of stall and vibration during speed reduction.

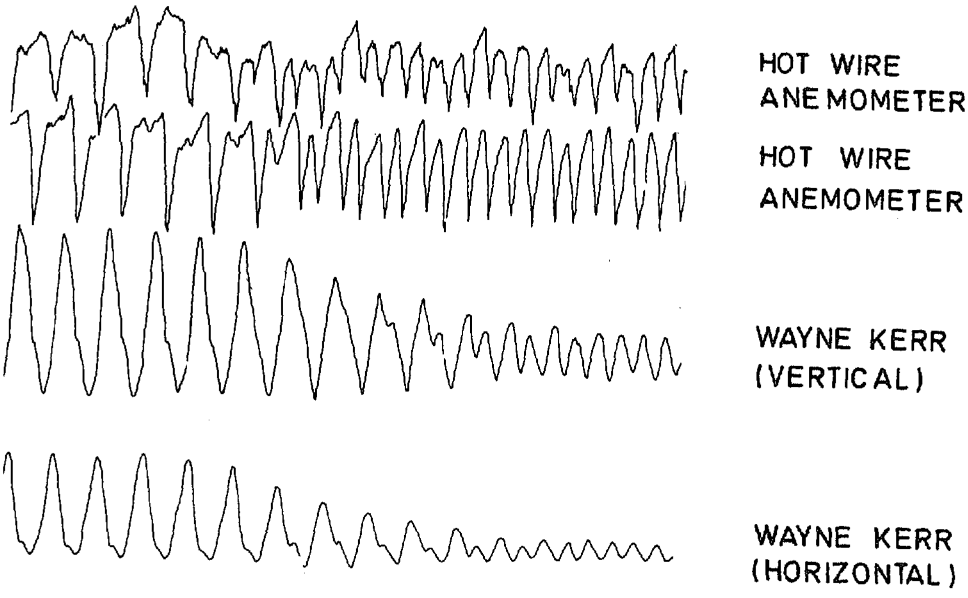

FIG. 14. Change from one to two stall cells.

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