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IDENTIFICATION OF MECHANICAL ALTERATIONS FROM THEIR EFFECT ON PERFORMANCE OF A RADIAL COMPRESSOR

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ABSTRACT

An experimental study of the influence of mechanical alterations in a stage of a radial compressor with a vaned diffuser is presented. The mechanical alterations considered correspond to changes which can be produced by the occurrence of faults or deterioration of the compressor. They include the insertion of an inlet obstruction, an obstruction in a diffuser passage, increase of impeller tip clearance, and impeller fouling.

The change in the compressor performance parameters, from the reference condition, is established from the experimental results. These changes are referred to the overall stage performance but also to its components, impeller, and diffuser. In order to establish diagnostic abilities, appropriate indices are introduced. The behaviour of these indices is related to the altered stage conditions and the possibility of using them for identifying the stage condition is demonstrated.

1. Introduction.

The concept of Condition Monitoring has been first advanced in the field of Jet Engines, where reliability is crucial, while engine performance degradation has a direct financial impact on operating costs in a very competitive market. Condition Monitoring of Industrial Gas Turbines has also advanced and finds a wide application today, because these engines are producing large amounts of power and are widely used for this purpose in electricity generation and in many industrial applications. For both types of application, the interest has been focused in engines producing large output (whether in the form of thrust or mechanical power), since in such cases even comparatively small improvements correspond to large absolute savings.

The axial compressor is a component commonly encountered in these applications, because of its ability to handle efficiently large mass flow rates. Therefore, the investigation of the behaviour of axial compressors in different types of malfunctions has received much attention by various investigators (see for example reviews by Meher-Homji et al (1992) and Lakshminarasimha et al (1992)). Radial compressors, on the other hand, have received less attention, since they are used by relatively smaller engines. Also, the fact that flow in radial compressors, being more complex, has not been understood in the same extent as in the axial ones, did not allow deductions for their behaviour in faulty conditions by simple physical reasoning.

Even though radial compressors have not been used in the field of power generation, as the axial ones, they nevertheless find wide applications. They are used in the field of aeronautics for small engines as well as in helicopter engines. They are used in small gas turbines, such as Auxiliary Power units or other small shaft power gas turbines. They are widely used in the process industries and in turbochargers of internal combustion engines. This wide range of applications together with present day demands for efficiency and reliable operation has made condition monitoring of systems including radial compressors an area of interest for development.

The present paper deals with the problem of radial compressor monitoring and in particular, performance monitoring. The purpose is to derive information as to how performance of the compressor is influenced by the occurrence of faults during its operation. Such information is useful for setting up monitoring procedures, in the following ways: (a) it can be used as a basis for formulation of the monitoring methodology, (b) it contributes to the constitution of a data base or knowledge base supporting the monitoring (e.g. data base of "fault signatures").

The main point of interest is to examine if the faults influence the performance of the compressor and, therefore, if their identification is possible from performance measurements.

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In order to establish the correspondence between the presence of a fault and performance alterations, a series of experiments is performed. The compressor is first run at its intact condition and the baseline performance is established. Mechanical alterations are then introduced and the performance is measured again. Comparison with baseline gives the information on the influence of the alterations to the performance.

Ways of exploiting such information for assessment of the condition of a radial compressor are then presented and discussed.To the author's knowledge, such an investigation has not been conducted before.

2. Test Facility, Measuring Equipment.

The experimental investigation has been performed on a single stage radial compressor fitted on the Compressor Test Rig of the Lab of Thermal Turbomachines, National Technical University of Athens. The compressor is operated in an open circuit and consists of an unshrouded impeller with 14 blades with radial exit and a tip diameter of .35 m, followed by a vaned diffuser with 11 vanes. The inlet and outlet ducting of the compressor is equipped with throttling valves, which allow both inlet and outlet throttling. The compressor is driven by a DC motor with a thyristor drive, which allows rotational speed adjustment. This arrangement allows the setting of operating conditions covering the entire performance map of the compressor.

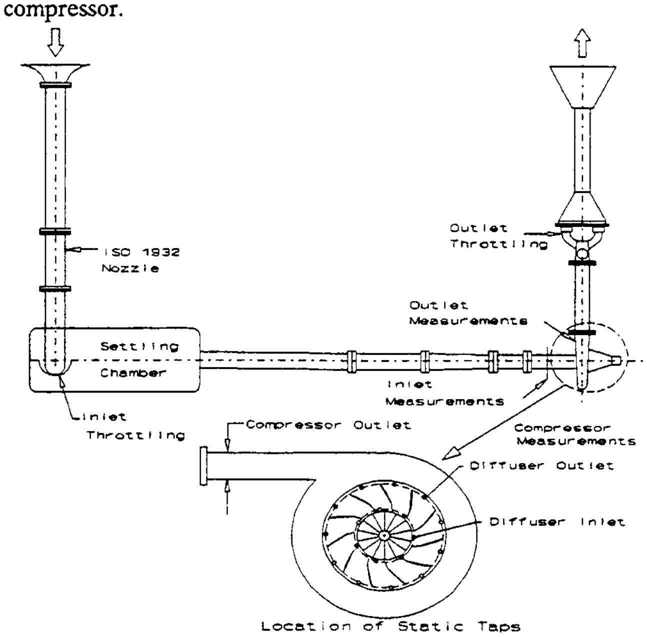

Figure 1. The test compressor and the measuring positions.

The compressor is fitted with instrumentation allowing measurement of the quantities necessary to derive the overall performance, as well as the performance of individual components (impeller and diffuser). The layout of the compressor rig and the measuring locations are shown in figure 1. Mass flow rate is measured with an ISO-1932 nozzle fitted at the inlet ducting. At the compressor inlet, total Temperature and Pressure are measured with a three-hole probe, equipped with a thermocouple, while static pressure from wall taps at the same axial location. Rotor outlet and diffuser outlet static pressure are measured as the average from wall taps. Compressor Outlet total pressure is measured with a pressure rake, connected in a manifolded arrangement, while static pressure is measured with wall taps at the same location. Compressor Outlet total temperature is measured by averaging the readings of eight Type K thermocouples placed on two circular rings in the outlet duct.

All measurements were recorded with a data acquisition system based on a personal computer. The PC is fitted with a data acquisition card with a 12 bit A/D converter, maximum aggregate sampling frequency of 100 KHZ and adjustable input ranges. The measurements for the present investigation were obtained as time averages of 300 samples over one second for each measuring channel.

3. Experimental Procedure.

As mentioned in the introduction, the purpose of the present work is to identify the effect of mechanical alterations to the performance of the compressor. For this purpose an experimental study has been organized as described below.

The compressor in its initial condition, namely before any alteration is introduced, is tested. The performance data for the intact compressor are thus established, and are used as a reference for subsequent testing. These tests and the corresponding conditions are termed as "datum".

Subsequently, mechanical alterations are introduced, one at a time. The performance is measured after each alteration is realized, while once the alteration is removed, it is verified that behaviour is identical to the datum condition. The alterations considered have been chosen in order to reproduce or simulate faults in the compressor and are the following:

(a) Obstruction at the inlet: An obstruction is introduced in the compressor inlet, figure 2a.

(b) Obstruction in a Diffuser Passage: One passage of the vaned diffuser was modified by inserting a screw, figure 2b.

(c) Tip Clearance Variation: The clearance between the compressor impeller and its casing was varied, by moving the casing axially upstream. Two tip clearance increases have been installed, figure 2c: Tip Clearance-1 with $\Delta\lambda$=0.136 and Tip Clearance-2 with $\Delta\lambda$=0.091, where λ=(axial clearance gap)/(impeller outlet width).

(d) Impeller Fouling: The impeller surface was covered with a paint, simulating fouling, figure 2d.

We will use the terms "faults" or "mechanical alteration", to refer to these different compressor conditions in the rest of the paper.

We note here that from a practical point of view alterations a,b do are not actually operational faults, but rather simulate such faults. For example the inlet obstruction, would correspond to mistuned individual vanes, if such vanes exist at the compressor inlet. Screw insertion corresponds to effects which would lead to blockage of individual diffuser passage.

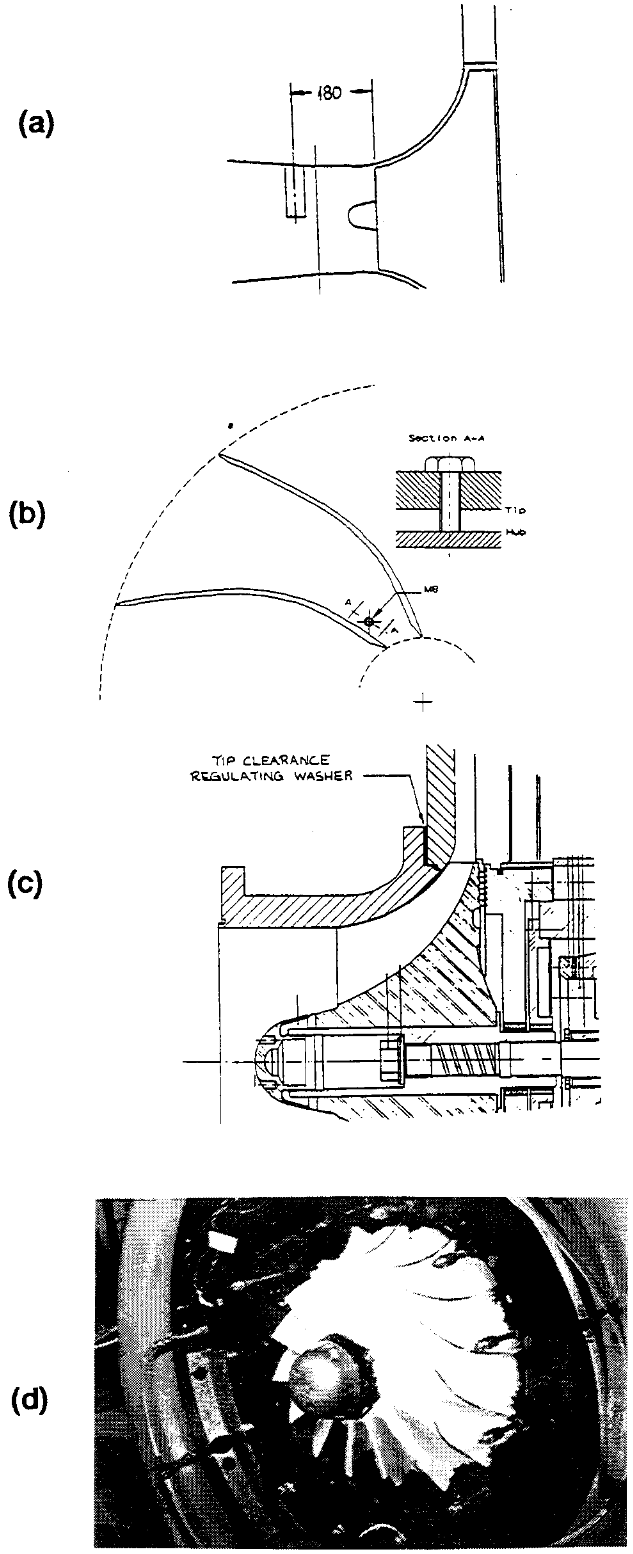

Figure 2. Alterations to the test compressor (a) Inlet Obstruction (b) Obstruction in a Diffuser Passage (c) Tip Clearance Variation and (d) Impeller Fouling.

The experimental program consists of performing measurements at various operating conditions of the compressor, for every mechanical condition. Three rotational speeds of operation were used: 12000 rpm, 14000 rpm and 16000 rpm. At each speed, operating points covering the entire operating range, from maximum mass flow rate to the surge limit, were acquired.

Having set the rotational speed, each operating point is set by applying throttling through the inlet and outlet throttle valves of the compressor. In order to make sure that sufficient data points are taken and that data are obtained at corresponding conditions, the operating point is defined by means of an <u>on-line data acquisition and reduction</u> procedure. The data reduction provides a graphics display of the compressor characteristic curves and the operating points are visualized on the screen. Apart from the characteristic curves, the measurement values themselves are displayed for continuous monitoring of instrument outputs. A typical screen display when the experiments are performed is shown in figure 3.

This set-up contains also some self-monitoring capabilities, i.e. the instrument readings are checked by the software and an alarm (audio-visual) is activated when the useful ranges of the instruments are exceeded. These possibilities allow a fine determination of the operating point, so that data are taken at operating conditions characterizing particular operating regimes.

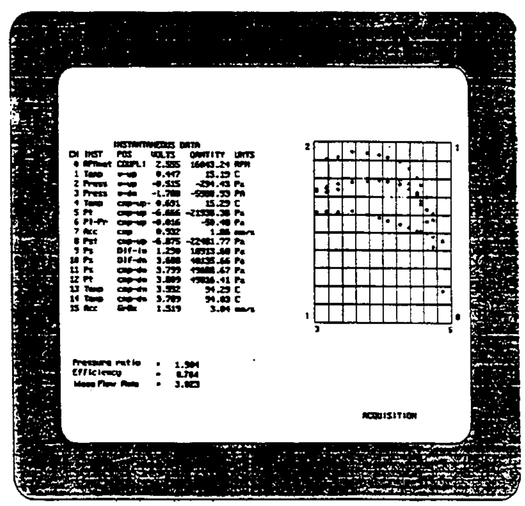

Figure 3. Typical display of on-line data acquisition and reduction system.

4. Measurements and Evaluation of Effects on Compressor Performance.

In order to establish the effect of the different alterations on compressor performance, the performance maps of the compressor and its components are compared for the different mechanical conditions. It is recalled here that the interest of the present study is focused on the examination of the significance of the observed effects, in relation to possibilities for practical applications for compressor condition assessment. We are seeking ways to identify the condition of the compressor from performance measurements. We start by presenting some sample performance results and then how observed modifications can be quantified in a form useful for condition assessment.

4.1 Modification of Performance by Alterations.

The occurrence of an alteration from the ones considered here results, in general, to a reduction of the performances of the compressor. Examples of the modification of the overall performance map of the compressor due to a fault is shown in figures 4 and 5. In these figures the (mass flow - pressure ratio) and (mass flow - efficiency) characteristics of the compressor are shown for three conditions: (a) healthy compressor, (b) compressor with rotor tip clearance of $\Delta\lambda=0.136$ and (c) with tip clearance of $\Delta\lambda=0.091$. The observations which can be made can be summarized as follows:
- The occurrence of these faults results in reduction of the swallowing capacity of the compressor, namely for the same pressure ratio, the faulty compressor handles smaller mass flow. Pressure rise capacity is also reduced, namely for a certain mass flow rate the faulty compressor produces a smaller pressure ratio than the healthy one. Efficiency is also reduced in the faulty compressor.
- While the above differences can be observed at all rotational speeds, they are more pronounced at the higher speeds.
- The effects of the smaller clearance are similar to the ones of the larger one, but of smaller magnitude, as it is expected.

Before discussing results any further a remark should be made as to the way of reducing the experimental data in order to compare performances. In order to get meaningful results and have a true comparison of performances, corresponding points should be chosen for certain values of corrected performance parameters. For example, a drop in pressure ratio is manifested by observing corresponding points, namely points at the same corrected mass flow rate and corrected rotational speed. This implies that data from the experiments should be processed in such a way that the maps are derived in terms of corrected quantities and dependence of particular testing conditions (as for example ambient conditions or rotational speed variations) is taken out.

Further to the observations of overall performance changes, it is useful to examine the performance of individual components. Examining performance characteristics of individual components may be helpful for identification of the kind of fault, which has caused a certain overall performance shift. An example of performance modification of the impeller alone is given through the results presented in figures 6 to 9. In figures 6 and 7 the impeller performance modification by tip clearance increase is shown, while in figures 8 and 9 the effect of the diffuser fault is shown.

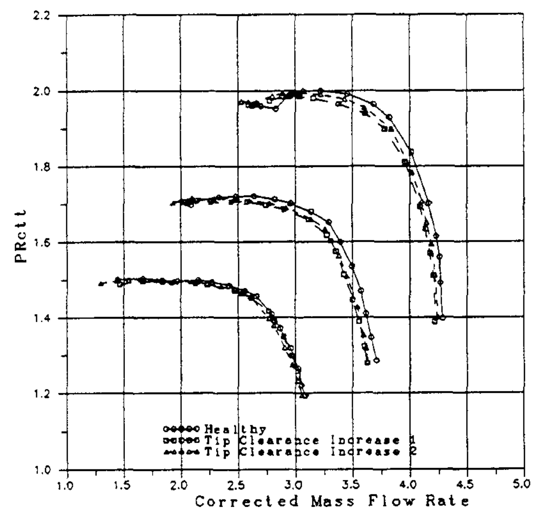

Figure 4 Overall map modification for tip clearance faults.

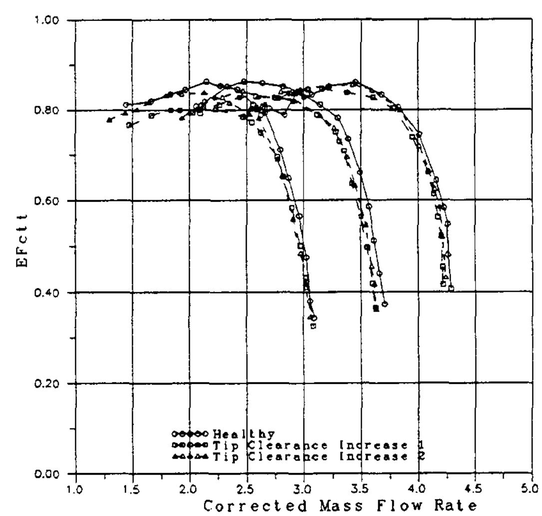

Figure 5 Overall compressor efficiency for tip clearance faults.

It is observed that the tip clearance increase has a direct and quite important effect on impeller performance, as should be expected. On the other hand, the diffuser fault, seems to have a minor influence on impeller performance. This influence, though very small, can exist even though the impeller itself has not been touched. It is nevertheless influenced because it actually operates coupled with the diffuser, so changes in the diffuser may also reflect on impeller performance (for example change in circumferential uniformity at impeller exit is a factor which is known to affect impeller performance).

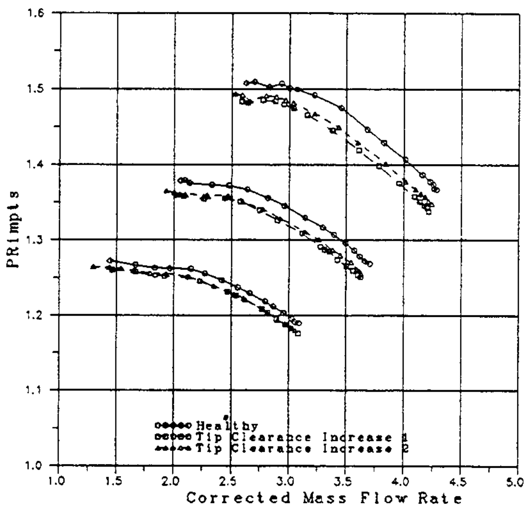

Figure 6 Impeller pressure ratio for tip clearance faults.

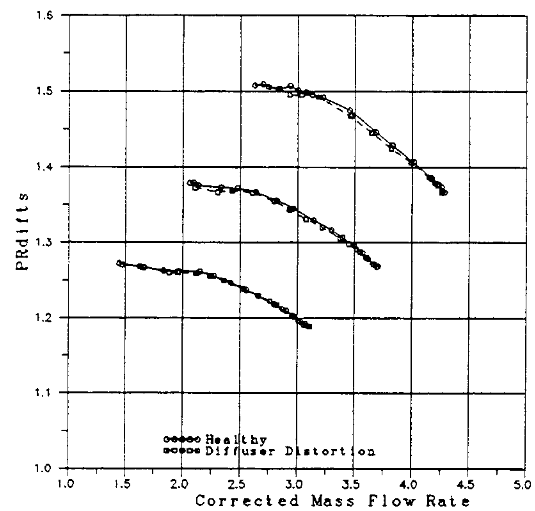

Figure 8 Impeller pressure ratio for diffuser fault.

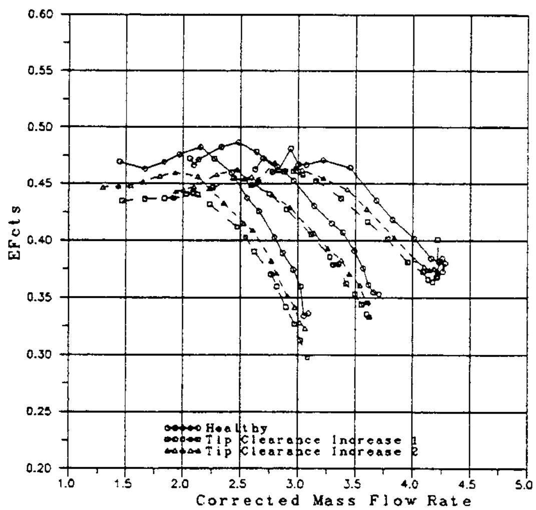

Figure 7 Impeller Efficiency for tip clearance faults.

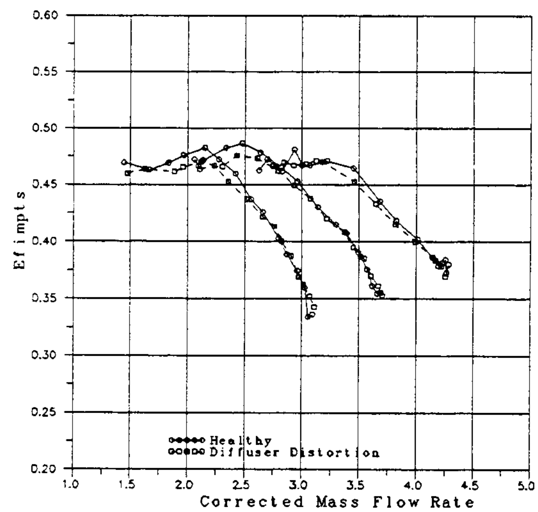

Figure 9 Impeller efficiency for diffuser fault.

4.2 The Use of Generalized Performance Parameters.

When the performance of a compressor is monitored, care must be taken in order to correctly reference a data set to the corresponding healthy compressor performance. For example, when operating data are available at a corrected rotational speed of 14000 rpm, the performance of the compressor at this speed should be known, in order to compare with the data in hand. Otherwise, the performance data should be referenced to another corrected speed.

In order to reduce the free parameters when such questions are posed, it is useful to have a representation of the performance with as few parameters as possible. It is known from the theory of turbomachinery operation that this can be achieved with the use of generalized performance parameters. For example, it is known that with suitably defined mass flow parameter Φ and pressure rise parameter K_{is}, it is possible that the curve Φ-K_{is} can be unique for a wide range of rotational speeds. In this way the dependence of performance on rotational speed is eliminated. We are going to see now how we can use this general knowledge in observing effects of faults on compressor performance. The definition of the parameters we employ is the following:

$$K_{is} = \frac{\eta_{is} \cdot (h_{t2} - h_{t1})}{U_2^2} \tag{1}$$

$$\phi = \frac{\dot{m}}{\left(\frac{\rho_1+\rho_5}{2}\right)\cdot A_2\cdot U_2} \tag{2}$$

The symbols of these equations are: η_{tt} compressor isentropic efficiency, h_t total enthalpy, ρ density, A area and U circumferential velocity. The subscript 1 corresponds to the compressor inlet, 2 to the impeller outlet and 5 to the compressor outlet.

Before we examine the influence of faults, we shall see how the Φ-K_{is} curve depends on rotational speed. The Φ-K_{is} curves for the three rotational speeds of our experiments are shown in figure 10.

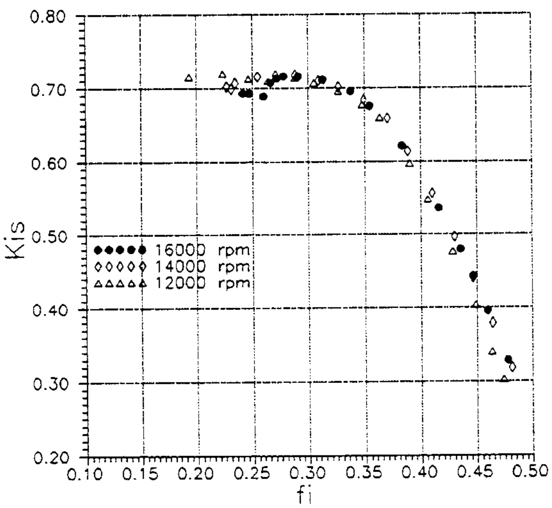

Figure 10 Generalized characteristic for healthy compressor at different rpm.

It is observed on this figure the curves for 14000 and 16000 rpm coincide, while the curve for 12000 rpm shows a slight difference at the higher Φ values. What is interesting for our case is the fact that for the range 14000 to 16000 rpm we have a unique Φ-K_{is} characteristic. This means that we can reference performance modifications to this characteristics, eliminating the need of requiring data at one particular rotational speed. It must be pointed out here that when the parameter Φ is based just on the inlet volume flow rate (as is customary in axial flow compressors), then the characteristic is not unique for different speeds. We shall now examine how the generalized characteristic is influenced by the different alterations we have examined.

The compressor characteristics for the different cases examined are shown in figures 11 and 12. It can be seen that the rotor faults have much stronger impact on performance than the stationary ones, the latter being of very small extent anyway. In all cases a drop in performance is observed. In these figures, apart from the experimental points, best fit curves are also presented. We shall now discuss the use of these curves for characterizing the different cases.

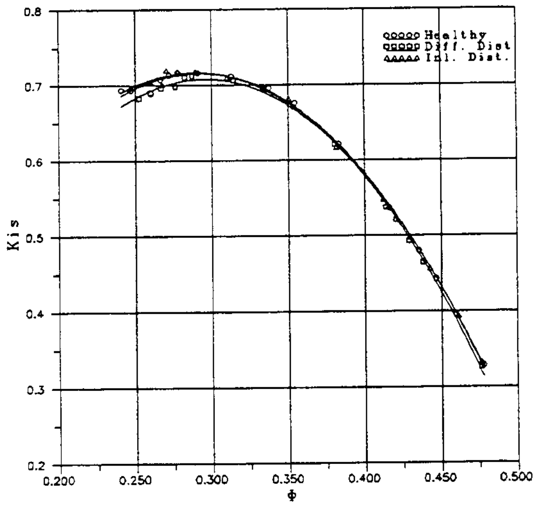

Figure 11 Generalized characteristics for stationary alterations.

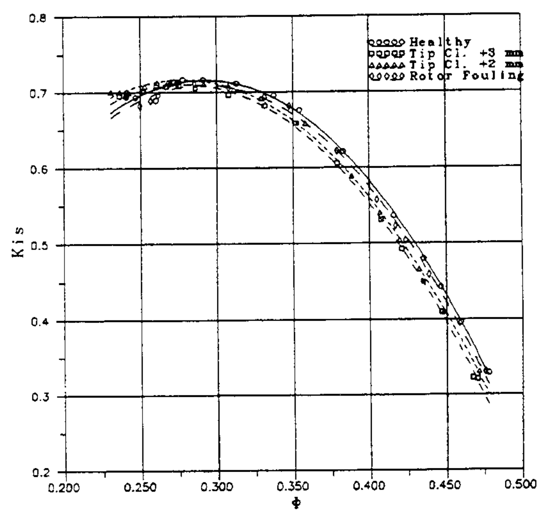

Figure 12 Generalized characteristic for alterations related to the rotor.

5. Identification of Compressor Condition.

Identification of faults is based on the fact that the modifications of performance for each fault is of a different form. For example one fault may give large shifts in swallowing capacity, while another one in pressure rise. This actually means that the form of the corresponding performance curves is different.

A way to quantify the differentiation of performance curves is to produce an analytic representation for them and then compare the corresponding analytic expressions. If, for example, each one from a set of curves is represented by a polynomial expression of a certain degree, then observing changes in the polynomial coefficients can be used to characterize the change in the shape of the characteristics.

For our case, we have fitted 2nd degree polynomials to the data of figures 11 and 12. The differentiation of the curves corresponding to the faults from the healthy one are shown in figure 13.

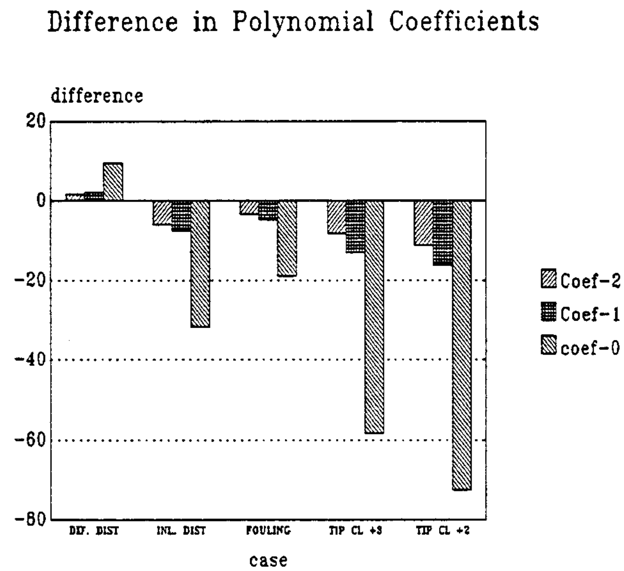

Figure 13 Differences in polynomial coefficients of ϕ-Kis curves for different faults (data from 16000 rpm).

It is shown from this figure that different faults cause a different pattern in the change of the characteristics. The patterns change in magnitude for the different faults. The form of the pattern of Diffuser Distortion is distinctly different from the other ones, the other four patterns however, do not show such a distinct differentiation in their form.

At this point the following remark can be made: since we visually observe different curves for the different faults and we have represented them by analytic expressions, it should be possible to produce a "clear" quantitative criterion for their distinction. A second way of distinguishing these curves is presented in figure 14. In this figure the deviations in K_{is} and Φ at two points of the healthy characteristic are used to describe the deviation of the characteristic of each fault. This figure gives picture which can be used for a clearer distinction between the faults. We call it the "Diagnostic Plane". It must be noted that this figure also gives information with a direct physical meaning: it gives an idea of reduction of swallowing and pressure rise capacities of the stage. It is observed that, the tip clearance fault gives a reduction in performance in different proportions from the fouling. This is expressed by lines of different slope on the diagnostic plane.

Finally, a similar procedure has been applied to the performance curves of the impeller alone, as shown in figures 15 and 16. Performance is now expressed in terms of pressure ratio and mass flow rate, at constant rotational speed. The Rotor Diagnostic Plane shows again a possibility to distinguish the different faults. In this case the effect is more pronounced while the differentiation is more distinguishable for the different faults.

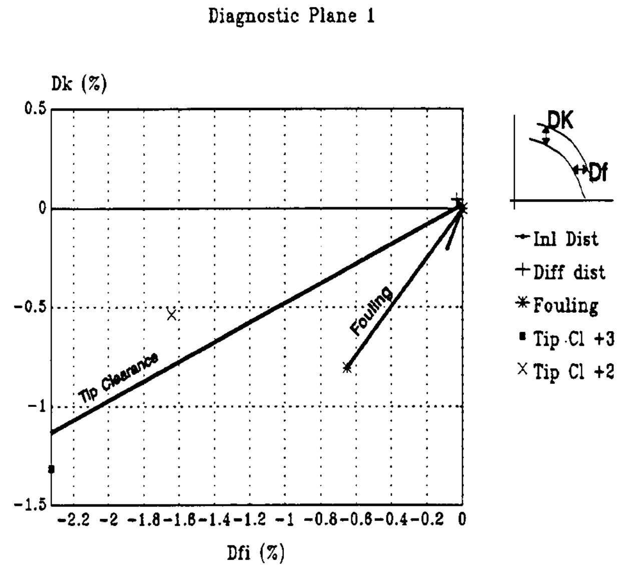

Figure 14 Differences in Kis for ϕ=0.3 versus differences in ϕ for Kis=0.45, of fault cases from healthy condition.

6. Discussion.

The results of the experimental study presented above show that it is possible to identify the alteration which has caused a certain drop in performance of a radial compressor. This can be concluded from the fact that different alterations are presented by different points on the compressor or rotor diagnostic plane, figures 14, 16. It can be claimed that each kind of alteration corresponds to a particular direction on this plane as indicated on these figures. Such an effect should not be surprising, since the overall performance is made up by the flow field, which is modified when geometry of the flow path changes. This type of effect of faults on performance is known to happen in axial compressors In this respect the experimental evidence provided here can serve for the deviation of roules similar to the ones used for axial compressor (e.g Stamatis et al, 1990, Lakshminarashimha et al, 1992).

A question that arises is whether the findings of the present experimental investigation have some more general validity. For this purpose the literature has been searched, in order to trace experimental data obtained by other investigators. Such data were found for the case of tip clearance increase, not seen in the context of Condition Monitoring, in Senoo (1984). We use two different parameters, to match the format in which the data are presented by Senoo. These parameters are the impellers static-to-static load coefficient, K_{s-s} and the flow parameter Φ based on inlet volume flow rate, defined by the following equations:

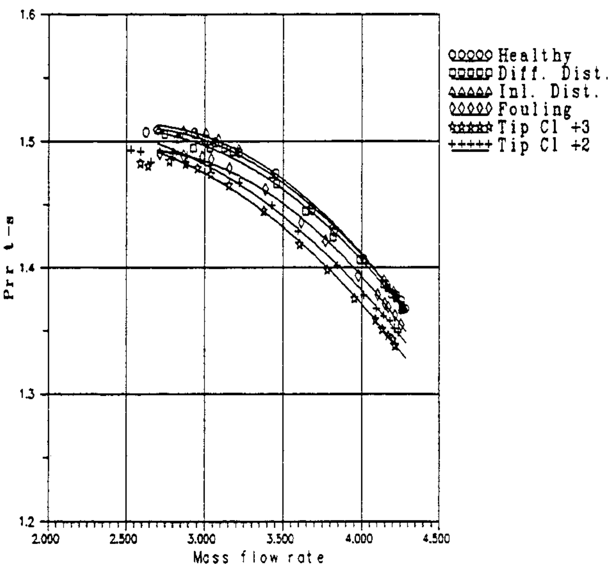

Figure 15 Rotor characteristics and 2nd order polynomial fits. Corrected speed 16000 rpm.

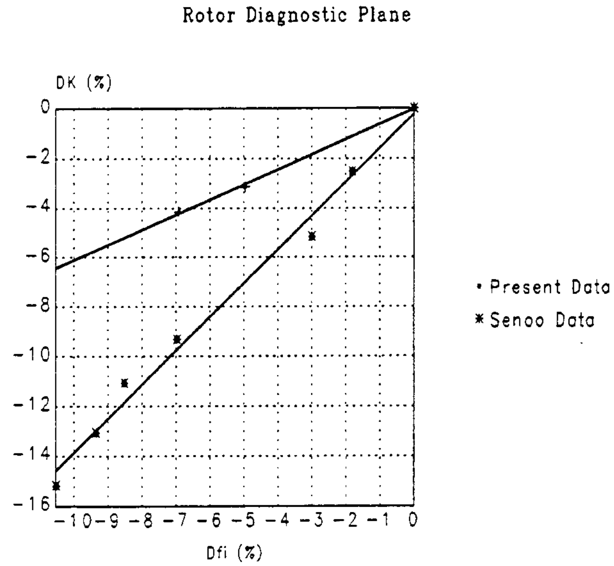

Figure 17. Differences in Ψ_{ss} versus differences in Φ of tip clearance from present data and from Senoo data.

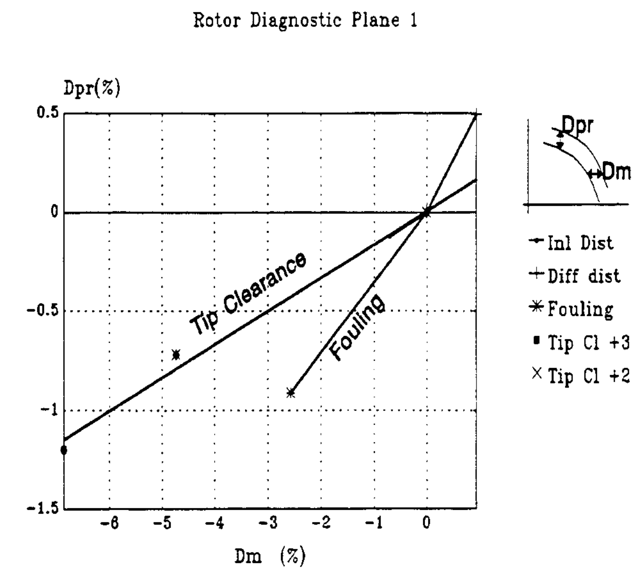

Figure 16 Differences in pressure ratio for m=2.8 versus differences in m for pr=1.4 for fault cases from healthy condition.

$$K_{s-s}=\frac{C_p T_1\left(\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}-1\right)}{U_2^2} \tag{3}$$

$$\Phi=\frac{\dot{m}}{\rho_1 A_2 U_2} \tag{4}$$

T is the static temperature and P the static pressure, while the rest of the symbols and the subscripts are the same as those of equations (1) and (2).

Application of the fitting and comparison procedure to such curves gives the results of figure 17. From this figure we can conclude that our observations for characteristic directions representing faults on the diagnostic plane, are also verified by using data of other investigators. We note, however, that the slope for data from another compressor is different from the slope for the compressor used in the present study, for the same type of fault. It is not possible to explain this difference, since we do not possess enough details to compare the two compressors and the way of application of the fault. We cannot, therefore, draw a conclusion whether the slopes of a fault are invariant for certain type of compressors, or what factors would make them different, for different compressors. This is a matter which needs further investigation, either by using experimental data from different compressors or even by employing (reliable) computer simulations.

From the data we present here, figures 14,16,17, it can be concluded that the slope corresponding to a fault does not depend on its magnitude. Increase of the magnitude of a fault, in general, increases the distance from the origin on the diagnostic plane, along a constant slope line. The same figures demonstrate the applicability of a certain procedure based on different parameters each time. This implies, that the success of its application will depend on the particular parameters chosen (by "success" we mean how clearly the presence of alterations is reflected).

A further comment on the procedure presented above, is the fact that the measurements employed are rather easy to be done, in the sense that we employ simple instruments which don't require any difficult mechanic alterations of the compressor. From this point of view, they can be easily realized in an industrial environment. The only measurement which is more difficult to materialize is the mass flow measurement. For this measurement a part of the circuit should be appropriately calibrated or information from static and total pressure measurements should be exploited. The later possibility is a matter of pottential further investigation.

Another fact relevant to practical application of the present methodology is the fact that measurement data points should cover a part of a speedline of the compressor. A typical example of a practical situation where such operation is encountered is the single shaft gas turbine with a speed governor. When rotational speed is kept constant, load variations lead to different points along a compressor speedline. If speed variations occur, then applicability depends on how wide a region of the generalized characteristic is covered. This means that, in general, applicability of the method should be examined according to the way of operation of a particular radial compressor.

In this respect, the influence of the width of the covered range of the characteristic on the diagnostic information has been examined. It was found that maximum clarity is obtained if the entire operating range is covered, excluding the two end regions. If the range shrinks around the nominal operating point, the presence of faults is still clearly indicated, but distinction of the type of fault may be less clear, depending also on the particular parameters employed for the diagnostic plane.

Our investigation has shown that faults of minor extent, such as inlet and diffuser obstructions, do not produce a clearly distinguishable signature on performance. Such alterations can be traced by other kinds of measurements, such as fast response measurements, in analogy to similar cases of axial compressors (e.g. Loukis et al, 1991). The authors have actually verified that casing accelerometers can be used to identify them (Mathioudakis et al, 1994).

The methodology presented above follows an approach which can be coupled with computerized techniques for Turbomachinery Health Monitoring. All information is appropriately quantified, so that decisions can be taken by a computer, on the basis of algebraic data rather than visual inspection. The method is therefore suitable for integration in a computerized system, employng techniques similar to techniques developed for axial compressors (e.g. Stamatis et al, 1990, Loukis et al 1994).

As a last comment, it should be said that this work has shown how a monitoring procedure for radial compressors can be established and has provided fault signatures for particular faults. Further investigations should be necessary, in order to examine dependence of the findings on compressor design and type of application, as well as to extend the repertory of faults which can be covered. It is therefore hoped that a methodology is introduced, which will be helpful for organizing information related to the condition of radial compressors for the purpose of setting up computerized monitoring procedure.

7. Conclusions.

The effect of different alterations, corresponding to operational faults, on the performance of a radial compressor has been experimentally investigated. It was shown that performance of the stage as well as of the rotor alone degrades in the presence of these alterations. While rotor alterations, namely tip clearance increase and fouling, produce clearly distinguishable effects, the smaller alterations of inlet and diffuser passage obstruction give only small differentiations.

Ways of expressing the performance degradations in a quantitative manner have been presented and it was discussed how they can be used for diagnostic purposes. The concept of diagnostic plane is introduced and it is shown that different faults correspond to points characterized by different slopes on this plane.

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