ANALYSIS OF 1ST STAGE COMPRESSOR ROTOR BLADE STRESS AND VIBRATION AMPLITUDES IN ONE-PASS JET ENGINE

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Abstract

This paper considers 1st stage compressor blade dynamics in the one-pass jet engines of trainer aircraft. Research was carried out on an engine test bench using the SAD system and its results were compared with those obtained using the tensometric approach. In this paper presented basic dynamic properties of rotor blades, bench test of rotor blade dynamics, Bench tests of the dynamic behaviour of blades subjected to external impacts and then Comparison of strain gauge and SAD tip-timing results. Then discusses the results of tests assessing the accuracy of the 1987 ITWL device by comparing strain gauge signals with those recorded by SAD apparatus in a running engine. It also presented simultaneous vibration readings of all the rotor blades at selected rotation speeds. Also shows that increased stress in these blades may be due to repeated engine surges, normal and hot engine surges, entry into the engine of a foreign object.

Among others selection and layout of rotor blades in the 1st stage of a one-pass engine compressor, stress amplitudes for rotor blade, stress in rotor blade, stress amplitudes, free inlet flow and smooth engine acceleration, vibration amplitudes, asynchronous vibrations are presented in the paper.

Keywords: *diagnostics of aircraft engines, compressor blade dynamics*

1. Introduction

In the years 1985-1987 one-pass turbojet engine failures occurred as a result of compressor 1st stage blade breakage. The nature of these failures (two cases of a number of blades breaking off simultaneously) with no discernible material or manufacturing defects led to investigations conducted by the ITWL. Two main lines of investigation were considered. The first involved tensometric studies of 1st stage of compressor rotor blade vibration amplitudes and stress taking into account operation factors that could lead to momentary or periodical growth of stress [7] [6]. The second approach involved developing apparatus that would be able to measure vibration amplitudes in-flight using the tip-timing method [6]. In 1986 ITWL also researched the rate of crack propagation in blades. Research was carried out on an engine test bench using the SAD system (presented in chapter 4) and its results were compared with those obtained using the tensometric approach.

2. Basic dynamic properties of rotor blades

Experimental studies established that the average fatigue strength for the oscillatory cycle Z_{-1} [2], p. 13] of 15 new randomly selected 1st stage compressor rotor blades was Z_{-1} = 693 MPa. In the case of four blades from the same batch used in an engine where the rotor blades broke off during operation the fatigue strength was Z_{-1}= 700 MPa, whilst in the case of rotor blades after 1,200 hours of operation it was Z_{-1}= 636 MPa.

On this basis it possible to state that the difference between the average fatigue strength Z_{-1} of new blades and those that had operated for 1,200 hours is 9%.

Extensive experimental analyses of titanium and steel rotor blade fatigue strength in various operating conditions are presented in [3]. These analyses show that under various operating conditions, over various periods of time and depending on the blade material machining procedure, deviations in fatigue strength may reach up to 30%.

Experimental studies of 1,350 1st compressor stage rotor blades revealed that natural frequencies ranged from 318 to 385 Hz [7].

It should be stressed that 1st and 2nd mode shapes predominate in the 1st stage of compressor blades of a one-pass engine. The 2nd mode shape only predominates in a narrow range of rotation speeds (10,500-11,500 rpm).

In order to study rotor behaviour, the blades were arranged so as to obtain maximal mistuning.

The layout of the compressor 1st stage blades with their specific free vibration frequencies is shown in 0. Here one can see that the frequencies were within the 318 to 385 Hz range. An engine that had had confirmed 1st compressor stage blade break-offs was selected for the strain gauging and further tests. Out of a total of 28 blades, 21 had strain gauges. These were installed on the suction base near the base, where blade vibration causes maximum bending stresses of the 1st and 2nd mode Fig. 2. Three blades, 11, 21 and 23, had additional strain gauges installed near their trailing edge bases Fig. 1. Moreover, two gauges were installed on the 1st compressor stage disc surface to confirm the occurrence of any coupling between the disc and the rotor blades.

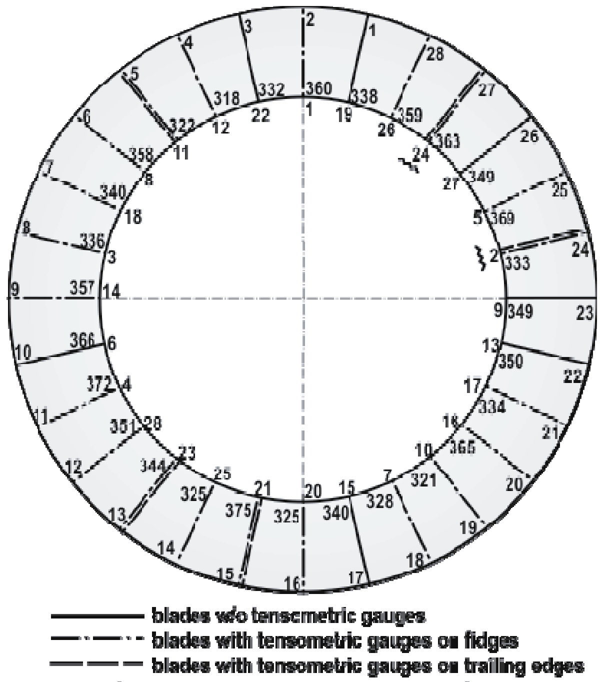

Fig. 1. Selection and layout of rotor blades in the 1st stage of a one-pass engine compressor

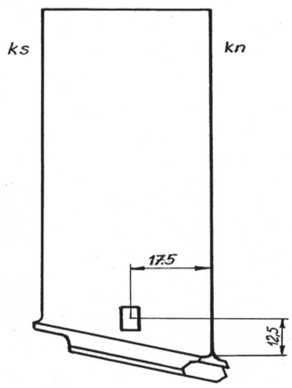

Fig. 2. Location of tensometric gauge on a 1st compressor stage rotor blade

3. Bench test of rotor blade dynamics

The highest levels of rotor blade stress occur at resonance frequencies and during unsteady compressor operating conditions. There are several kinds of unsteady aircraft engine operation conditions: ones associated with combustion, aero-elastic unsteadiness such as rotor blade flutter [4] as well as aerodynamic instability caused by rotating stall zones and compressor surges [1]. Rotating separation zones are a disruption of flow through the compressor involving its detachment from rotor or stator blades. This process expands, causing the detachment of flow from individual blades to merge into separation zones partially or fully encompassing the compressor flow channel. Their speed is dictated by and slightly lower than the rotation speed of the blades.

Compressor stalls involve periodic axial pressure pulsations in the compressor's flow duct. They are associated with higher amplitudes of pressure oscillation than in the case of rotating separation zones but with a much lower frequency. Engine stalls are almost always preceded by the occurrence of separation zones. Unlike the rotating separation zone, stalling occurs axis-symmetrically and uniformly around the compressor perimeter. Literature [1] distinguishes between the ordinary stall with large, low frequency pressure fluctuations and the modified stall, which combines a deep, expanded ordinary stall with a reverse flow through the compressor.

Deep surges are most dangerous as far as stress on blades is concerned.

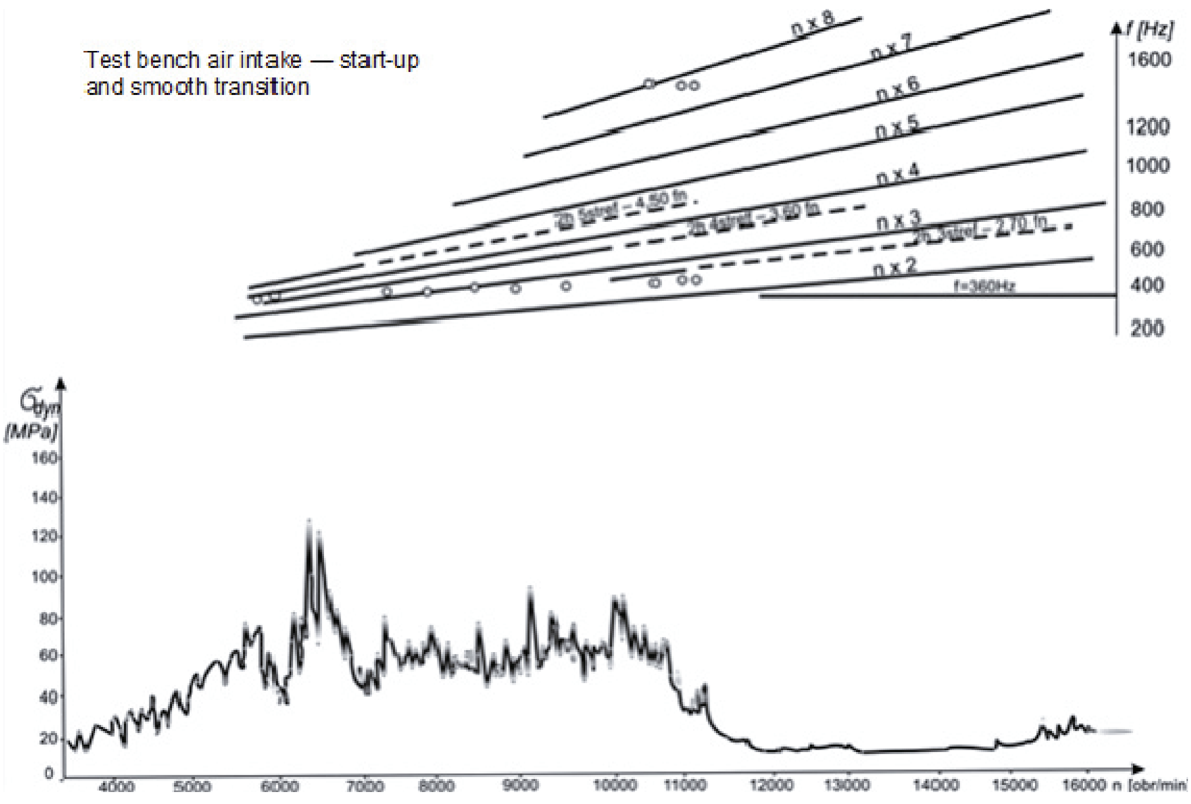

Fig. 3. Stress amplitudes for rotor blade 1 installed in the 1st stage of a one-pass engine compressor

Figure 3 shows the time-varying stress amplitudes in blade 1 (360 Hz, Fig. 1, blade numbering on the inner circle), in the test bench inlet, with rotation speed smoothly increased from 6,000 rpm to 16,070 rpm. The Campbell graph 'circles' denote the blades' measured free vibration frequencies of the 1st and 2nd mode. This shows that in the 1st stage three stall zones occur at 9,500 rpm and 400 Hz and four zones at 6,600 rpm and 410 Hz. Fig. 4 presents time-related variations of stress in blade 2 (333 Hz, Fig. 1). The presented test bench measurements concern a single engine inlet (continuous line) and a two-channel/aeroplane engine inlet (dashed line). The latter causes asymmetric circumferential pressure on the guide vanes, forcing low-frequency excitations [5], thus maximum stress in the blade occurs at different and slightly higher rotation speeds than at the airframe inlet. Fig. 5.3 also presents the measured stress amplitude values in the case of a deep stall and repeated stalls at certain rotation speeds.

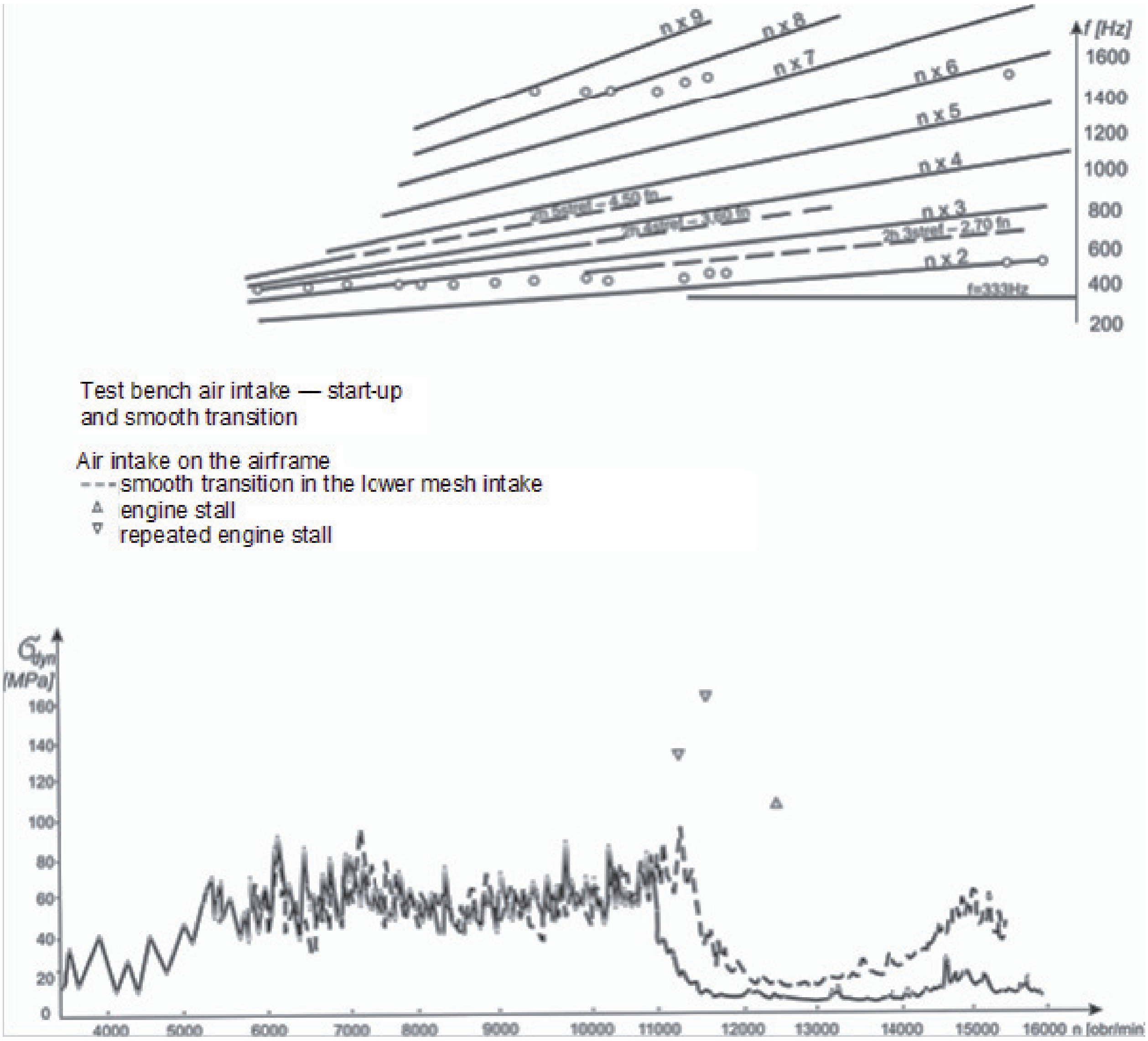

Fig. 4. Stress in rotor blade 2 of 1st stage of a one-pass engine compressor

4. Bench tests of the dynamic behaviour of blades subjected to external impacts

These studies showed the largest stress amplitudes occur during repeated stalls in blade 4 at 372 Hz ($\sigma_\delta \cong 280$ MPa at 11300 rpm) (Fig. 5), during ordinary, high-temperature engine stalls with 1/3 of inlet covered in blade 4 ($\sigma_\delta \cong 300$ MPa at 11300 rpm) as well as after the intake of epoxy resin fragments (inlet failure) in blade 14 at 325 Hz (461 MPa at 14900 rpm). It should be remembered that cold stalls cause aircraft engine unsteadiness without a temperature rise downstream of T_4 turbine, whilst hot (high temperature) engine stalls cause unsteadiness with substantial temperature growth downstream of T_4 turbine. In the case of inlet failure the highest stress amplitude, 461 MPa, occurred in a blade with a free vibration frequency of 325 Hz. When the inlet was partly covered, the highest stress amplitude, 378 MPa, was recorded for a blade with a recorded vibration frequency of 351 Hz.

It should be noted that with 1/3 of one of the inlets covered stress increases by as much as 50-60% when experiencing a conventional surge (from n_{mg} to n_{max}) in comparison with conditions when there is no surge (Fig. 5-7).

Assuming a similar rate of stress increase occurs during repeated stalling, its level would reach ca. 420-450 MPa. This could be very hazardous to blades, all the more so because on account of their low temperature, repeated stalling is not usually recorded.

Stress amplitude variations resulting from rotation speed are different in the case of each blade. Figure 8 presents the stress amplitude variations of all the blades in their consecutive order (as numbered on the outer circle in Fig. 1).

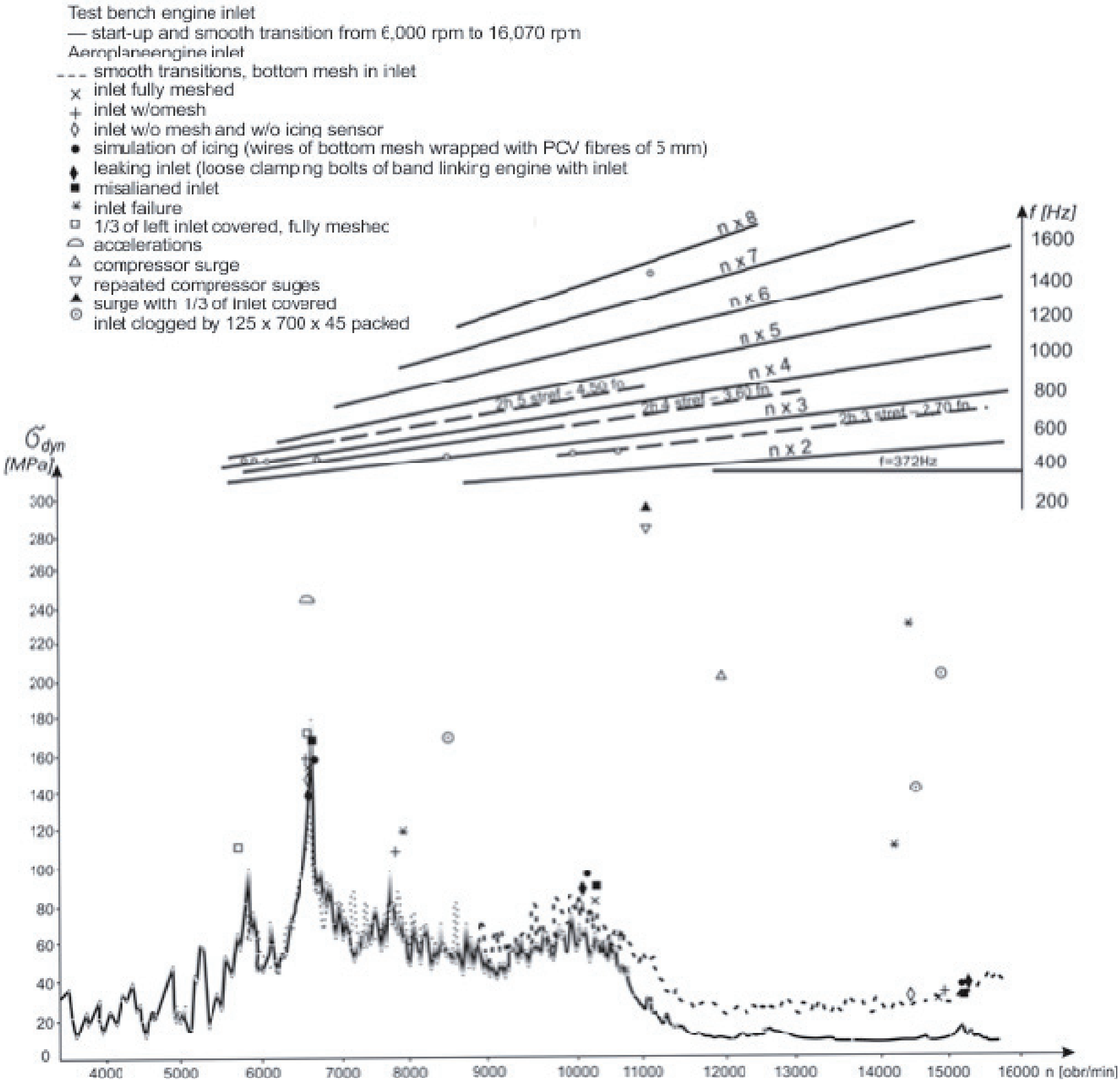

Fig. 5. Stress amplitudes in 1st stage rotor blade 4 of one-pass engine compressor

The maximum stress amplitude for rotor blade 1 (on the inner circle and 2 on the outer circle in Fig. 1) is 128 MPa at 6,500 rpm (Fig. 3), for rotor blade 2 (24 on outer circle) it is 96 MPa at 6,300 rpm (Fig. 4), and so on. These deviations result from differences in the geometrical dimensions of the blades and the nature of the flow, which is associated with rotating stall zones.

5. Comparison of strain gauge and sad tip-timing results

The first series of tests confirmed good correspondence between strain gauge and SAD signals as well as the nature of the fluctuations as seen on an oscilloscope display (Fig. 9).

Figure 10 shows SAD registered vibration amplitudes of all the rotor blade tips in the 1st stage of a prior to modernisation one-pass engine compressor at various rotation speeds.

At n = 10 300 rpm rotating stall zones occur, thus the blade vibration amplitudes are higher than n = 7100 rpm. At 12,000 rpm vibration amplitudes diminish as there are no major excitations in the flow.

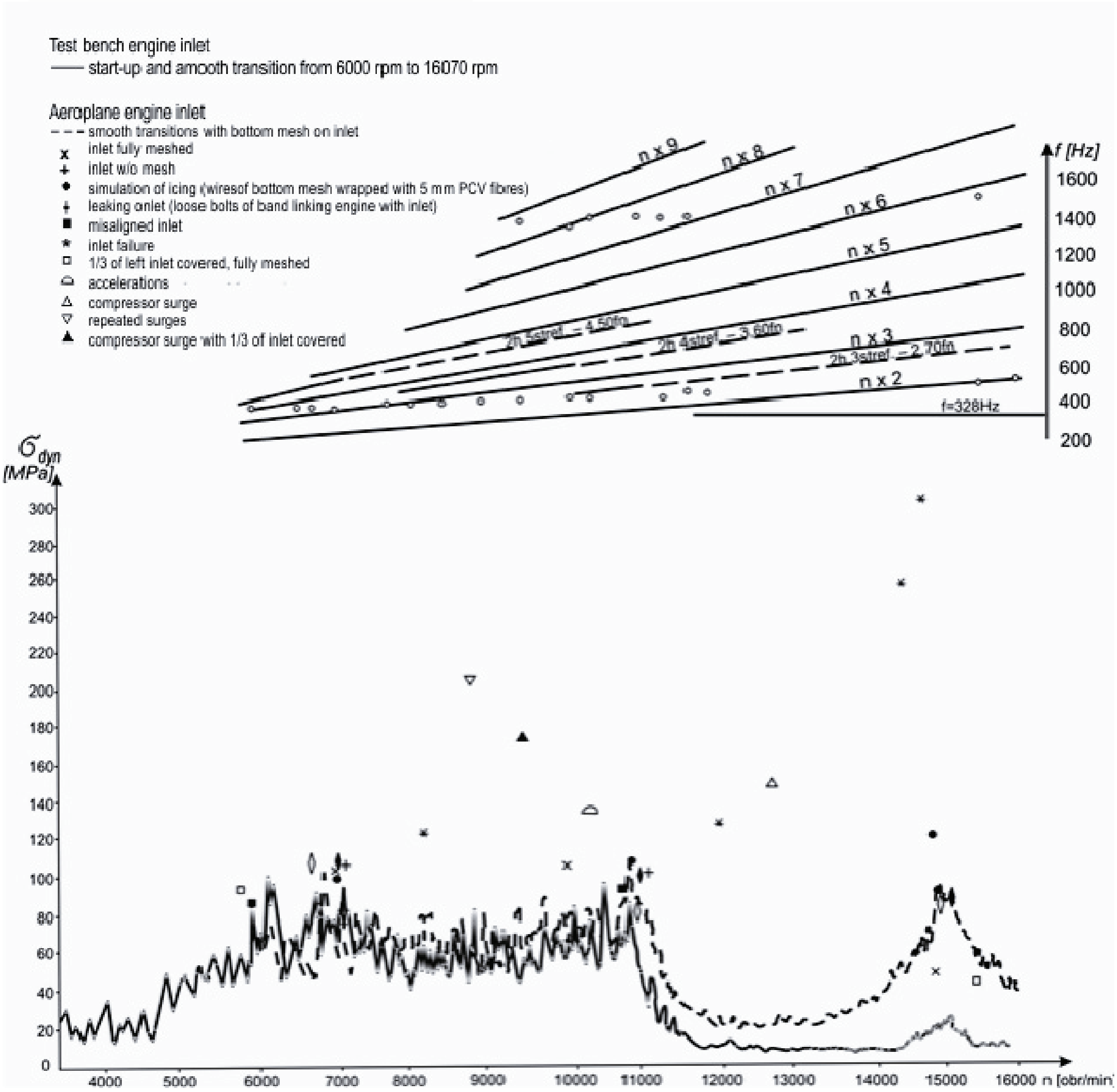

Fig. 6. Stress amplitudes in 1st stage rotor blade 7 of one-pass engine compressor

Figure 9 demonstrates that each rotor blade vibrates with different amplitude around a different equilibrium. This results from differences in the geometrical dimensions of blades, the precision of how blades are mounted onto the disc and the way in the bladed disc vibrates. The figures reveal the complexity of individual blade movement in a real turbine and the advantages of SAD measurements over those of strain gauges.

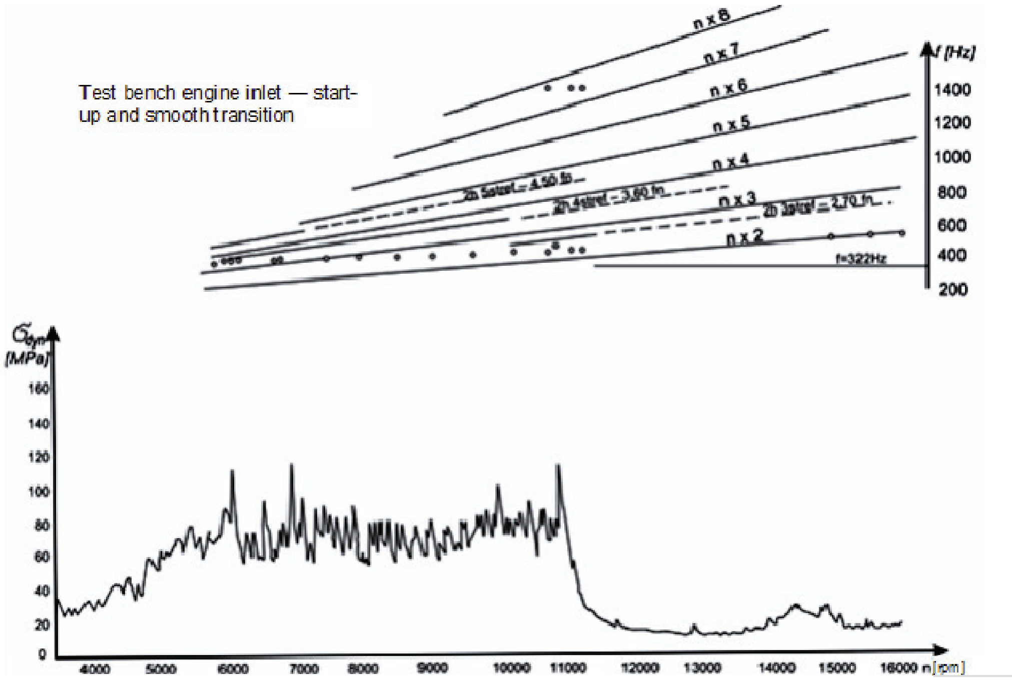

Fig. 7. Stress amplitudes in 1st stage rotor blade 28 of one-pass engine compressor

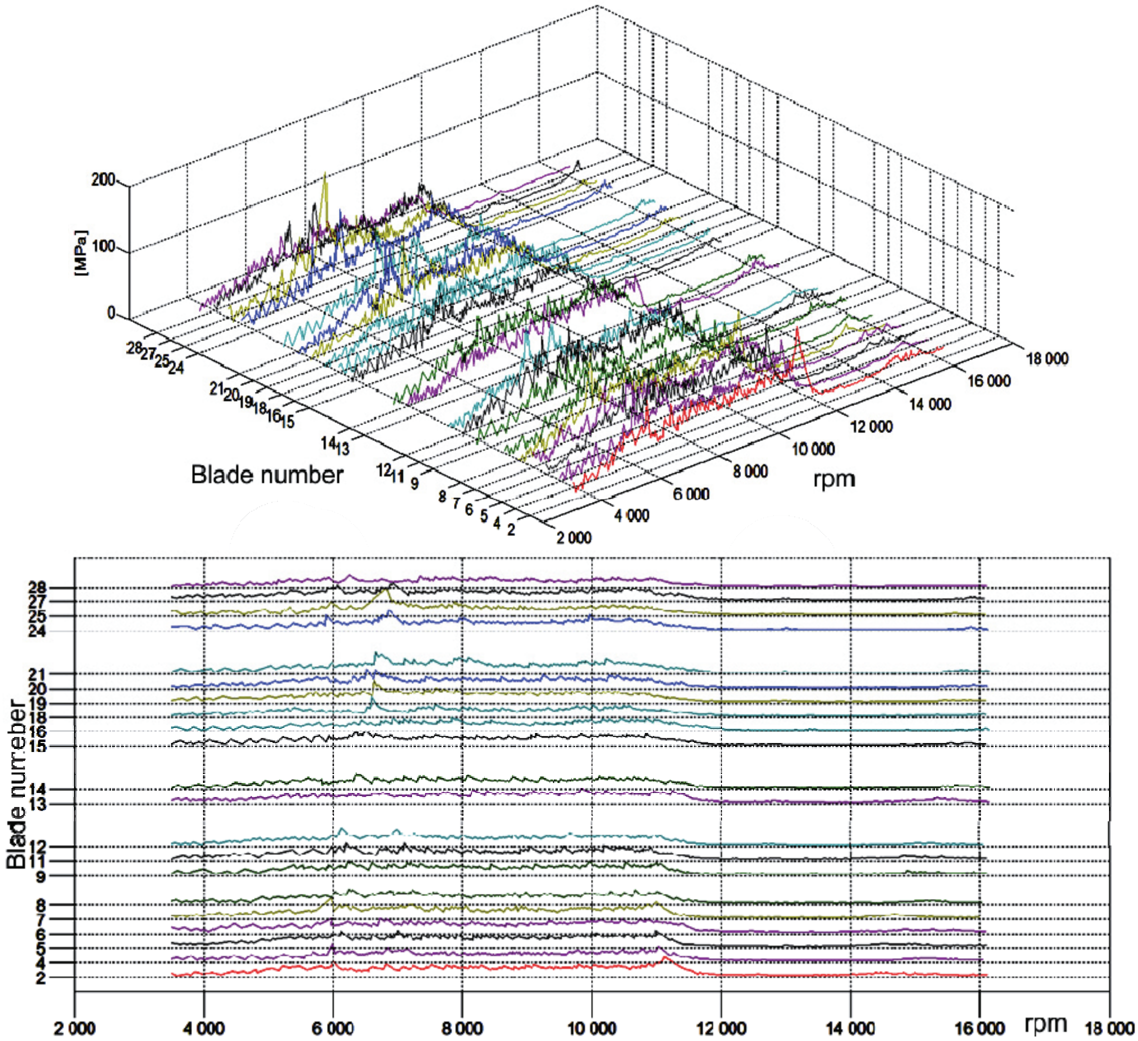

Fig. 8. Stress amplitudes of 1st stage rotor blade in one-pass engine compressor with varying rotation speeds, free inlet flow and smooth engine acceleration

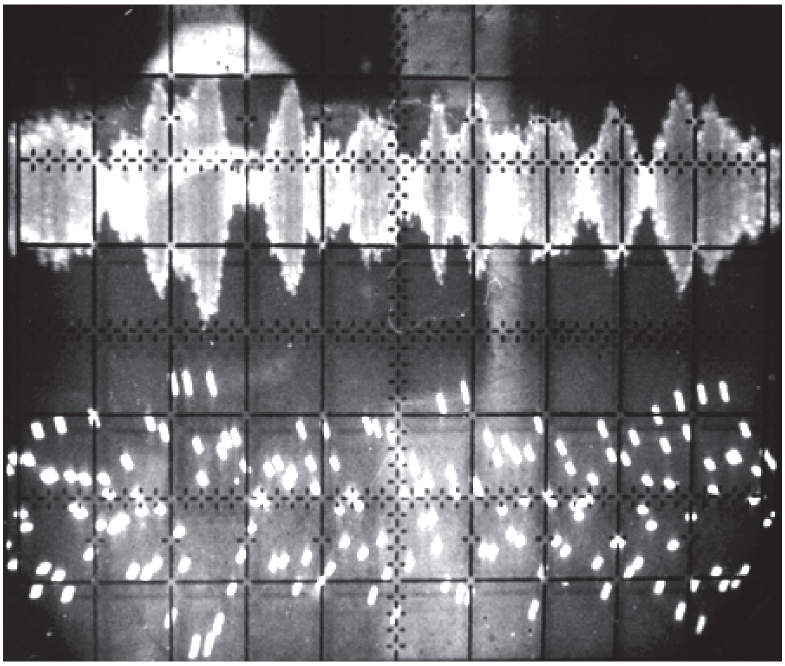

Fig. 9. Comparison of strain gauge analogue signals with digital SAD signals regarding the 1st stage rotor blades of a one-pass engine

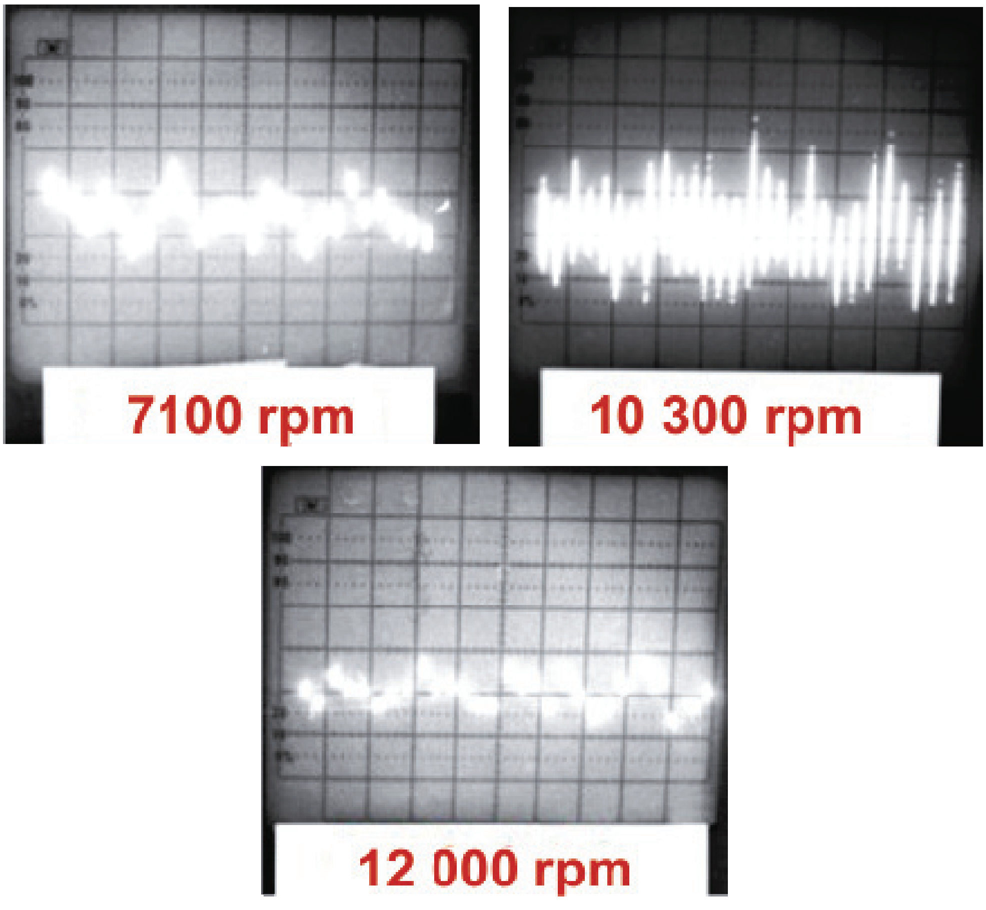

Fig. 10. Vibration amplitudes of 1st stage blade tips in a one-pass engine compressor prior to modernisation and at various rotation speeds. Asynchronous vibrations appear at n = 7,100 rpm; 10 300 rpm and blade vibration amplitudes are substantially reduced at n = 12,000 rpm

6. Conclusions

The conducted research into the dynamic behaviour 1st stage rotor blades in a one-pass engine compressor, before and after modernisation, shows that increased stress in these blades may be due to the following:

- repeated engine surges with no temperature increase downstream of turbine T_4, considered harmless and not even recorded in aircraft operating documents. Stress (bending) amplitudes up to $\sigma_d = 280$ MPa,
- normal (hot) engine surges with rotation acceleration stopped and a rapid increase of temperature in T_4, occurring when 1/3 of one of the two inlets is covered. The (bending) stress amplitudes up to $\sigma_d = 300$ MPa,
- entry into the engine of a foreign object (e.g. rag, piece of paper, bird or glove) partly covering the compressor guide vanes and therefore causing strong forced vibrations with stress amplitudes of up to 700 MPa.

Other operating conditions including meshes in aeroplane engine inlets, with full meshes, without meshes or an icing sensor, imitation of icing (wires in bottom meshes wrapped with plasticized PVC fibres up to $g = 5$ mm thick), leaking inlet (loosened band in front of engine inlet), misaligned inlet, left inlet partly covered (1/3 of inlet + full meshes), steady operating conditions with 1/3 of left inlet covered area, did not cause a significant increases of stress. One should note that the selected free vibration frequencies of the blades where within the free vibration range ($f = 318$-385 Hz) and did not result in a significant increase of stress. Approximately 180 MPa was recorded for a blade with a free vibration frequency of $f = 372$ Hz (Fig. 5). This blade had resonance vibrations forced by the first harmonic component of the four-zoned configuration with rotating stalls.

It is probable that stress levels resulting from conditions prepared in the bench tests may be considerably higher in certain phases of flight, especially in cases where the intake of air is substantially disturbed. This is suggested by the examined effect of hot surges on level of stress in the blades. The maximum stress recorded during such surges did not actually exceed $\sigma_d \approx 200$ MPa. However, when 1/3 of the inlet was covered, the same surges increased stress by about 50-60% of the initial value to reach $\sigma_d \approx 300$ MPa. Assuming that overlapping a wind tunnel on the ground corresponds to diminished inlet size during certain phases of flight, and supposing that blade stress increase in repeated surges is proportional to blade stress increases in hot surges, its amplitudes may even reach 420-450 MPa. Such stress may seriously affect blade durability, especially engines with a tendency to have surges (e.g. on account of badly adjusted or faulty automatic engine acceleration (APS).

It was therefore decided that further tests needed to be carry out, both on the ground and in flight, during regular operation and under surge conditions with a lateral engine inlet (ground tests) and an air intake reduction during typical aircraft manoeuvres, high angles of attack and sideslip.

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