Journal of Fluids and Structures (2002) **16**(3), 361–373
doi:10.1006/jfls.2001.0421, available online at http://www.idealibrary.com on IDEAL®

ACOUSTIC RESONANCE IN THE INLET SCROLL OF A TURBO-COMPRESSOR

S. ZIADA

Department of Mechanical Enginnering, McMaster University
Hamilton, Ont. L8S 4L7, Canada

A. OENGÖREN

Sulzer Innotec Limited, Winterthur, Switzerland

AND

A. VOGEL

Sulzer Turbo Limited, Zurich, Switzerland

(Received 16 August 2000, and in final form 7 June 2001)

During the commissioning period of a 35 MW turbo-compressor in a natural gas storage station, the vibration level of the compressor rotor increased sharply when the volume flow rate exceeded a critical value. The test results indicated that the acoustic standing waves in the ring chamber formed by the inlet scroll are excited by vortex shedding from struts in a downstream radial flow chamber. To alleviate vortex shedding from the struts, it was decided to mount small airfoils with a thin trailing edge in the wake of the struts. However, due to design constraints, streamwise gaps and transverse offsets between the struts and the airfoils could not be avoided. To investigate the effect of these gaps and offsets on the resonance mechanism, wind tunnel tests of a simple but conservative model were performed. Subsequent implementing of the airfoils into the wakes of the struts suppressed the acoustic resonance mechanism and thereby the rotor vibration at the acoustic resonance frequency was eliminated.

1. INTRODUCTION

VORTEX SHEDDING FROM BLUFF BODIES IN ENCLOSED VOLUMES OR DUCTS often excites transverse acoustic modes whose frequencies and mode shapes are dependent upon the dimensions of the enclosed volume and the bluff bodies. When these resonances are excited, the resulting acoustic pressure may exceed the dynamic head of the mean flow. Wake-excited acoustic resonances can cause acute noise and/or vibration problems in many industrial equipment such as turbo-compressors (Parker 1967), containers of tube bundles (Oengören & Ziada 1992; Weaver 1993) and piping systems (Hourigan *et al.* 1990). Excellent reviews of wake-excited acoustic resonances have been published by Parker and Stoneman (1989) and Welsh *et al.* (1990).

The simplest and most investigated flow geometry comprising wake excitation of acoustic resonance is that of uniform flow past a flat plate positioned in a rectangular duct (Parker 1966, 1967; Cumpsty & Whitehead 1971; Archibald 1975). In general, the resonance occurs when the frequency of vortex shedding from the bluff body approaches that of an acoustic mode. Mechanical resonance of the plate, if separated from the acoustic resonance frequencies, has been found to have no effect on the acoustic resonance mechanism. The frequencies

0889–9746/02/030361 + 13 $35.00/0

and mode shapes of the excited modes have been investigated by Parker (1967) and are referred to in the literature as the β- and α-modes. Parker (1967) has shown that the frequency of the lowest mode, the β-mode, is lower than the cut-off frequency of the empty duct, i.e., without the plate. The pressure field of this mode is "attached" to the plate and the duct wall such that the pressure amplitude decays rapidly with streamwise distance from the plate. When the plate cord is increased, the resonance frequency decreases and the rate of decay of the acoustic pressure with distance from the plate increases. This diminishes acoustic transmission into the main duct and therefore the resonances become more intense as the plate cord is increased.

Similar phenomena occur in the case of cascade of plates in rectangular or annular ducts. However, in addition to the β- and α-modes, higher order modes, the so-called δ- and γ-modes, become plausible. Parker & Pryce (1974) have shown that the resonance frequencies of *annular* cascades can be estimated from similar *two-dimensional rectangular* cascades with a height equal to the equivalent circumference of the annulus housing the cascade. In other words, the addition of plate-like bodies, such as struts, in rectangular or annular ducts reduces the resonance frequencies from those of the empty duct by an amount depending upon the plate dimensions.

This paper describes a case history of wake-excited acoustic resonance in a 35 MW turbo-compressor. In this case, however, *the resonant mode is found to be different from those described above*. Acoustic standing waves in the inlet ring chamber (the inlet scroll) were excited by vortex shedding from radial struts located in a downstream radial flow channel which leads to the first impeller of the compressor.

After a brief description of the problem, attention is focussed on how the excitation mechanism was diagnosed by means of *indirect measurements*. Thereafter, feasible design modifications to alleviate the excitation mechanism are discussed. Air tests of a conservative simple model representing the most practical, economical but still promising design modification are then addressed. Upon implementing this design modification in the compressor, the acoustic resonance phenomenon was totally suppressed and the rotor vibration was reduced to the expected normal level.

2. DESCRIPTION OF THE PROBLEM

During the commissioning period of a new 35 MW compressor in a natural gas storage station, unexpected excessive vibration of the compressor rotor was observed. The vibration frequency was constant at about 135 Hz and was not influenced by the speed of the rotor, which could be varied between 7200 and 10 700 r.p.m. Since the compressor was scheduled to be handed over to the customer within 6 weeks, the cause of vibration as well as a suitable solution had to be investigated within a very short time to avoid any delay in the delivery date. The investigation, however, was hampered by the difficulty of carrying out any pressure measurements, due to the very high pressure of the system (up to 320 bar) and the danger of explosion when the measuring equipment is used within the compressor room. A test program was therefore developed which could reveal the cause of the problem from measurements of the rotor vibration only.

2.1. Compressor Geometry

The geometry of the compressor is shown in Figure 1. It consists of two groups of centrifugal impellers, referred to hereafter as sections. The first section has four impellers and the second has three. Each section has separate inlet and outlet nozzles and the piping system connecting them has a cooler and re-circulating pipes such that the inlet conditions

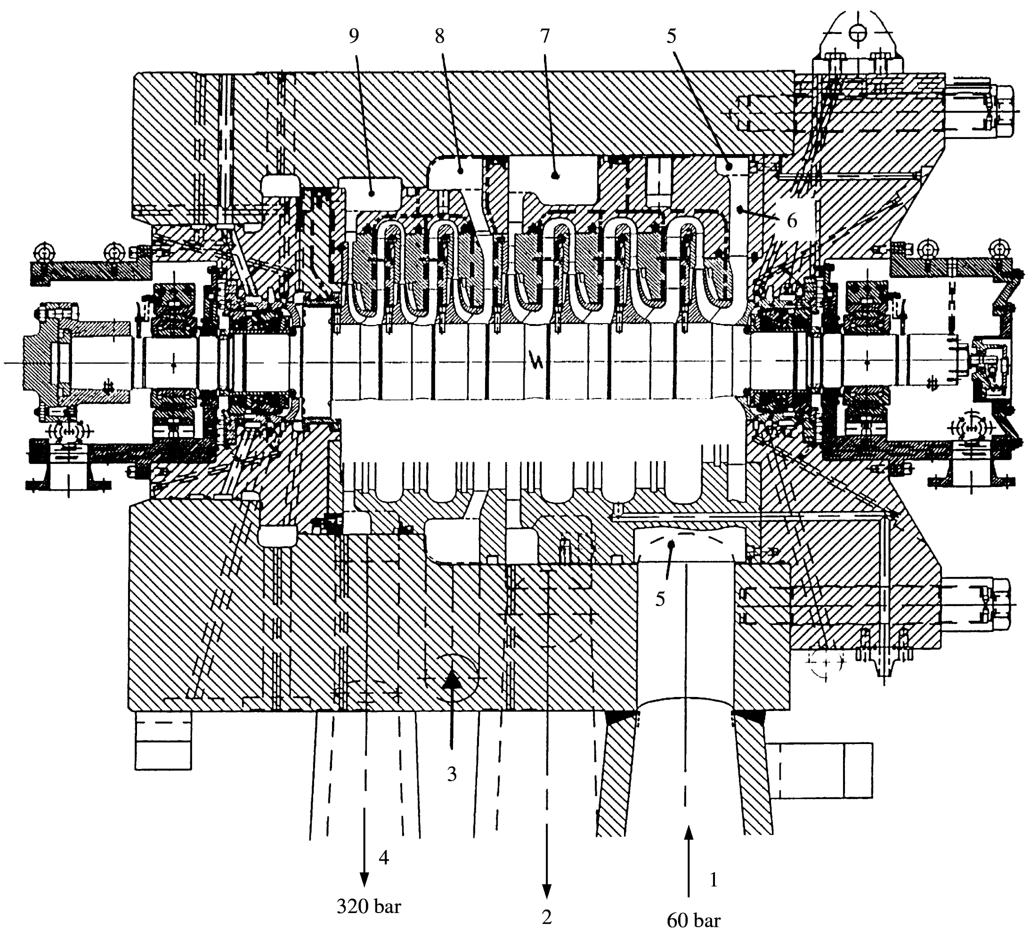

Figure 1. Geometry of the compressor showing its two sections and their inlet and outlet scrolls. The first section of the compressor consists of four stages (i.e., four impellers) and the second section has three stages: 1, inlet to first section; 2, discharge from first section; 3, inlet to second section; 4, discharge from second section; 5, inlet ring chamber (scroll) of first section; 6, supporting struts (see Figures 2 and 3 for details); 7, exit scroll of first section; 8, inlet scroll of second section; 9, exit scroll of second section.

of each section (pressure, temperature and mass flow rate) can be changed independently from those of the other section. Typical conditions at full load are 60 bar at the inlet of the compressor, and 320 bar at its outlet.

At the inlet and exit of each section, the gas is distributed or collected, respectively, by means of ring chambers referred to hereafter as the inlet and outlet scrolls (see items 5 and 7–9 in Figure 1). A schematic of the inlet scroll of the first section is given in Figure 2. The flow area decreases with distance from the inlet nozzle. The scroll is followed by a narrow radial flow chamber, which leads to the inlet, or the “eye”, of the first impeller. This chamber houses 14 struts as shown in Figure 3. The main function of these struts is to support the impeller casing. However, since they are also optimized to produce uniform flow at the inlet of the first impeller, they have different geometries as can be seen in Figure 3.

As mentioned earlier, the desired compressor load is obtained by varying the rotor speed. Figure 4(a) shows the performance curves of the compressor first section for rotor speeds ranging from 70 to 105% of the rated speed.

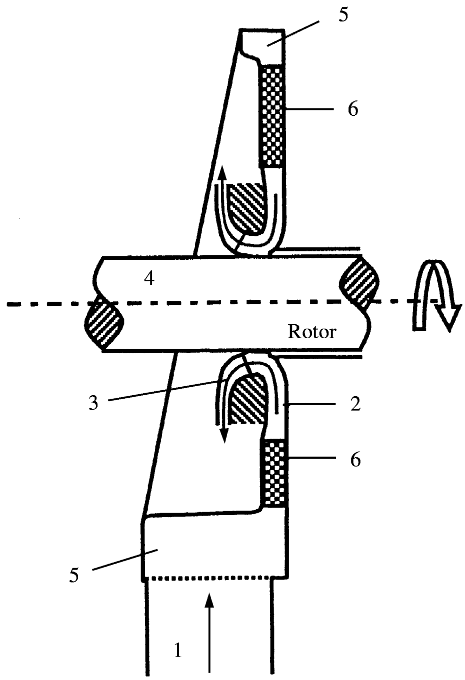

Figure 2. Schematic illustrating the geometry of the inlet scroll, the struts and the impeller of the first stage: 1, inlet pipe; 2, radial flow channel; 3, impeller; 4, rotor; 5, inlet scroll; 6, struts.

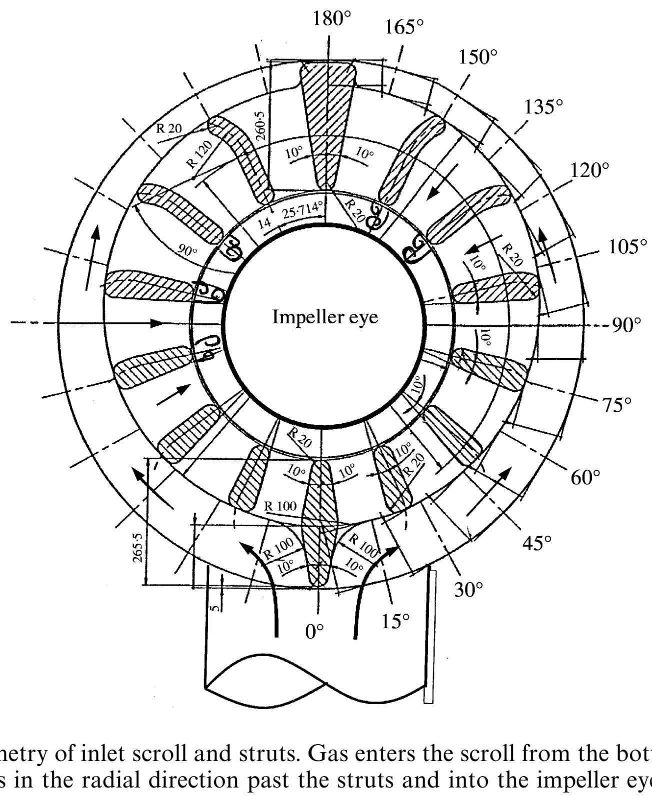

Figure 3. Geometry of inlet scroll and struts. Gas enters the scroll from the bottom into the scroll and then flows in the radial direction past the struts and into the impeller eye at the centre.

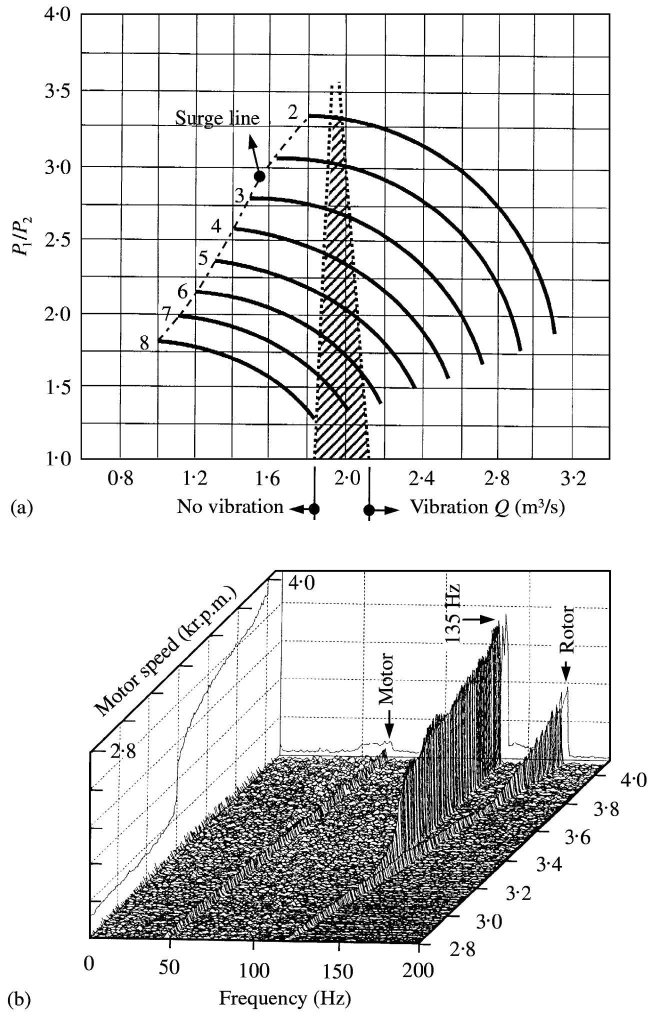

Figure 4(a) Performance curves of the first section, showing the range of volume flow at which the vibration starts, and (b) spectra of rotor vibration, showing the rotor response as the motor speed is increased from 2800 to 4000 r.p.m. In (a) curve 1, 100% of rated speed; curve 2, 105%; curve 8, 70%.

2.2. Vibration of Compressor Rotor

The rotor vibration was measured by means of eddy-current probes mounted near the bearings. When the inlet flow rate exceeded about $2\,m^3/s$, the rotor vibration increased abruptly to a level near the alarm set-value. Figure 4(b) shows typical spectra of the rotor vibration as the *motor* speed is increased from 2800 to 4000 r.p.m. As can be seen, the vibration starts at about 3400 r.p.m. and occurs at a constant frequency near 135 Hz. This frequency is different from the motor and the compressor speeds, as can be seen from Figure 4(b).

Additional tests were then conducted to assess the flow conditions at the onset of vibration. The results are summarized in the following.

(a) Regardless of the compressor speed, the vibration started at a constant suction volume flow of approximately $2\,m^3/s$, and persisted for higher flow rates. The vibration subsided when the flow rate was decreased below this limit. The phenomenon was reproducible. The range of vibration *onset* is illustrated in Figure 4(a).

(b) There was a threshold of suction pressure (about 30 bar) below which vibration did not occur.

(c) The rotor vibration level was substantially higher at the discharge side than at the suction side. On the gearbox, no excessive values were observed.

(d) The vibration was predominantly in the radial direction.

(e) Changes in the intake flow conditions of the first section influenced the vibration, whereas variations in the second section inlet conditions (volume flow and temperature) had no influence on the vibration.

(f) The piping, the compressor casing and the bearing casings did not show excessive vibration levels.

3. EXCITATION MECHANISM

3.1. Preliminary Diagnoses

Initially, several excitation mechanisms, which may be causing the observed vibration, were examined. These included rotor instability, rotating stall, instability of bearings and acoustic resonance in process piping. None of these was found to be the cause of the problem for the following reasons: (a) the vibration frequency was quite removed from any resonance frequency of the rotor; (b) rotating stall occurs near the surge line and at low frequencies. As can be seen from Figure 4, the vibration range was separated from the surge line and the frequency was relatively high; (c) bearing instabilities occur at low frequencies and can be influenced by the rotor speed, which, again, was not the case for the observed vibration; (d) the piping did not show any abnormal level of vibration.

Numerical simulation of the rotor vibration has shown that the level of pressure pulsation acting on the rotor must be substantial in order to produce the observed vibration level. The simulation showed also that the rotor response to an excitation at 135 Hz would be most pronounced at the discharge side if this excitation were applied at the suction side. Based on the measured vibration amplitude, the excitation level at the suction side was estimated to be 3 bar, which amounts to about 5% of the inlet pressure.

3.2. Acoustic Resonance in the Inlet Scroll

The fact that the vibration was initiated at a constant suction volume flow, *i.e.*, *at a constant flow velocity*, together with the observation that the vibration frequency remained constant when the flow rate was increased indicated that the vibration is most likely caused by a flow-excited acoustic resonance mechanism. Mechanical resonance was judged as less likely because, as mentioned earlier, the observed frequency did not approximate any resonance frequency. In cases of flow-excited acoustic resonance, an acoustic resonator and a flow excitation source must exist. These are identified in the following.

The ring chamber of the inlet scroll, item 5 in Figures 1 and 2, has a mean diameter of $\Phi = 1{\cdot}01$ m, and the speed of sound at the inlet conditions is $c = 420$ m/s. For acoustic standing waves in this chamber, the lowest mode corresponds to one wavelength along the scroll and its frequency, f_1, can be approximated by [note that the accurate value is dependent upon the ratio between the inner and outer diameters of the scroll (Blevins 1979)]:

$$f_1 = c/\pi\,\Phi = 131\,\mathrm{Hz}. \tag{1}$$

This value is in good agreement with the measured frequency of 135 Hz.

The flow excitation is generated by vortex shedding from the struts in the downstream radial flow channel, item 6 in Figures 1 and 2. For similar profiles with chord to thickness ratios from 2 to 10, Nguyen and Naudascher (1991) reported a Strouhal number of vortex shedding of approximately 0·27. The Strouhal number is defined as

$$S = fD/V, \tag{2}$$

where f is the vortex shedding frequency, D the strut thickness and V the flow velocity. In the present case, the flow velocity at the trailing edge of the struts and at the onset of vibration is estimated to be 30 m/s, and the strut thickness at its trailing edge varies between 55 and 60 mm. The Strouhal number based on the measured frequency (135 Hz) is therefore 0·25–0·27, which is almost identical to the value given by Nguyen and Naudascher ($S = 0{\cdot}27$). Thus, at the onset of vibration, the frequency of vortex shedding from the struts coincides with that of the scroll lowest acoustic mode. As the flow velocity is increased beyond 30 m/s, the phenomenon of "lock-in" occurs where the resonant acoustic mode controls the vortex shedding frequency, resulting in the sustenance of acoustic resonance with increasing flow velocity.

3.3. Confirmation of the Excitation Mechanism

Additional tests were conducted to confirm the postulated excitation mechanism. In the first test series, the volume flow of the two sections was varied independently and the flow conditions at the onset of vibration were recorded. The results were in agreement with the postulated excitation mechanism, as discussed already in Section 2.2. The second test series focused on varying the gas temperature at the inlet of the first section. As shown in Figure 5, the resonance frequency increases with temperature due to the increase in the speed of

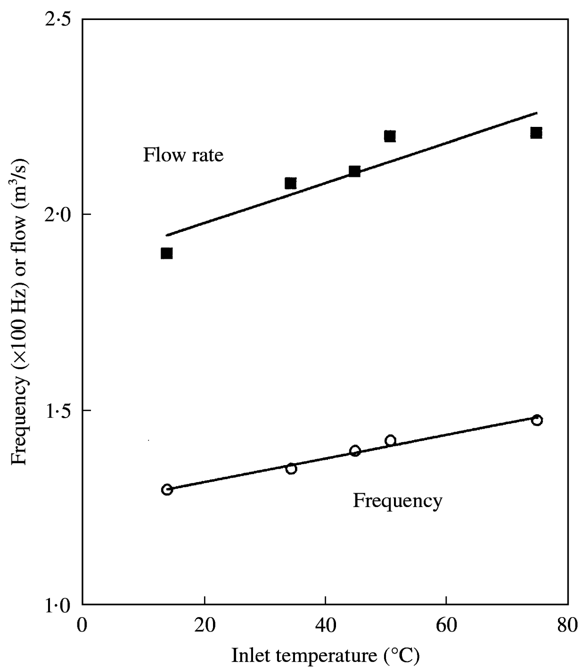

Figure 5. Effect of temperature on the critical volume flow rate and the vibration frequency.

sound. This results in an increase in the critical volume flow, i.e., in the critical flow velocity, which is in agreement with the postulated excitation mechanism.

Since the inlet scrolls of the two sections are virtually similar in size, compare items 5 and 8 in Figure 1, they must have approximately equal resonance frequencies. However, acoustic resonance did not occur in the second section of the compressor. This is because at the inlet of the second section, the gas density is 1·5–3·3 times higher than that of the first section. The outcome is a lower flow velocity, and consequently a lower frequency of vortex shedding from the struts of the second section. Another feature which may be of importance here is that the radial flow channel housing the struts in the second section is substantially narrower than that of the first section. Three-dimensional effects caused by the boundary layers in this narrow channel may also contribute to weakening the process of vortex shedding from the struts.

The last observation that still needed clarification was: why "similar" machines already operating in other plants do not exhibit similar vibration? A survey of these machines indicated two major differences: they were either smaller in size or their inlet pressure was lower. Due to the first difference, the resonance frequency was found to be always higher than that of vortex shedding from the struts. In the cases when a frequency coincidence was predicted for any other compressor, the inlet pressure was found to be less than 30 bar, agreeing well with the present observation that vibration did not occur when the inlet pressure was reduced below about 30 bar. This latter feature can be explained by the fact that at lower pressures, the sound power generated by vortex shedding is lower (Blevins 1984) and sound absorption by the flowing gas is higher (Kinsler *et al.* 1999).

4. DESIGN MODIFICATIONS

4.1. Evaluation of Feasible Design Modifications

In order to alleviate the mechanism of acoustic resonance, three different design modifications were formulated and evaluated. The first was to modify the trailing edges of the struts

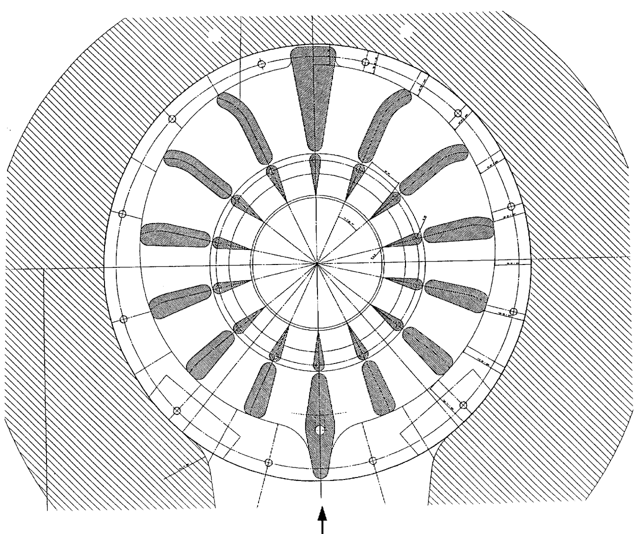

Figure 6. Geometry of the inlet scroll, the struts and the suggested airfoils to alleviate vortex shedding from the struts.

such that one side is bevelled at 45° and thereby reduce the intensity of vortex shedding. Although this modification seemed promising, as illustrated by Heskestad & Olberts (1960), it was rejected because the time required to machine the struts in the plant was too long to be acceptable.

The second idea that was considered entailed extending the strut facing the inlet nozzle to form a splitter plate within the inlet pipe. The objective was to disturb the lowest acoustic mode of the scroll. However, design and assembly constraints necessitated that a small gap must be left between the strut and the splitter plate. Acoustic "leakage" through this gap would have diminished the effect of the splitter plate on sound waves in the scroll. For this reason, this approach was also rejected.

The purpose of the third modification was to alleviate vortex shedding excitation by installing splitter plates, or airfoils, in the wakes of the struts as shown in Figure 6. This idea was acceptable because the airfoils could be prepared in advance and then installed into the compressor during a planned shutdown of the unit. However, due to design constraints and tolerances, radial gaps and lateral offsets up to 9 mm may exist between the struts and the airfoils. The effect of these gaps and offsets on the effectiveness of the airfoils against acoustic resonance needed to be studied before the airfoils are incorporated in the compressor.

4.2. Model Tests of Design Modification

Model tests were performed to investigate the effect of streamwise gaps and transverse offsets between the struts and the airfoils on the mechanism of wake-excited acoustic resonance. Since it was not possible to test a geometrically similar model within an

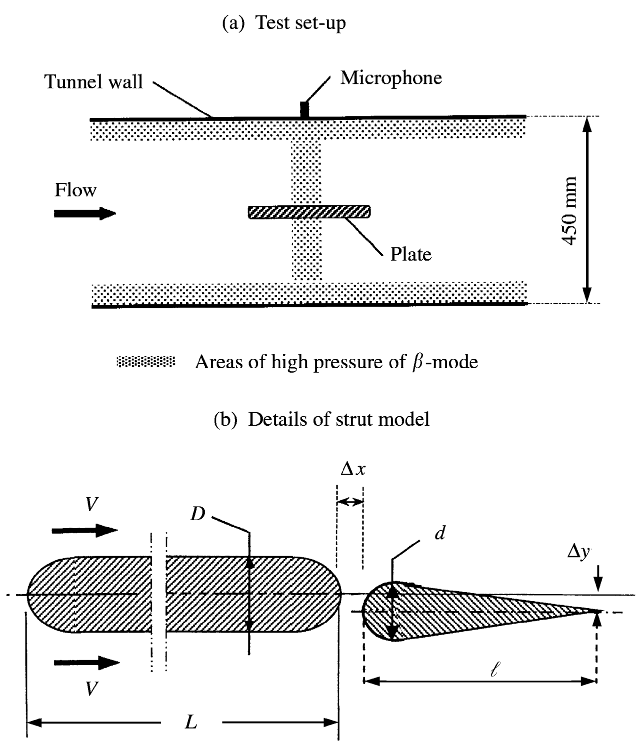

Figure 7. Experimental set-up used to perform the model tests and details of the strut and airfoil models. $L = 195$ mm, $D = 20$ mm, $l = 50$ mm, $d = 15$ mm.

acceptable time period, a much simpler model exhibiting wake-excitation mechanism was sought. As shown in Figure 7, a flat plate with cord to thickness ratio $L/D = 9{\cdot}75$ was placed at the mid-height of a wind tunnel. The plate had rounded leading and trailing edges to simulate the geometry of the struts. A relatively long L/D ratio was chosen in order to generate a strong case of β-mode resonance.

The system response as the flow velocity was increased to 50 m/s was studied by means of a microphone located on the top wall of the tunnel and at the centre of the plate, Figure 7. The frequency and amplitude of pressure fluctuations, measured by this microphone, are given in Figure 8 as functions of flow velocity. Strong resonance of the β-mode is seen to occur with a wide range of lock-in between vortex shedding and the β-mode. Outside the lock-in range, vortex shedding occurs at a Strouhal number of 0·26, agreeing well with the value reported by Nguyen and Naudascher (1991), and with that observed for the compressor. At resonance, the sound pressure amplitude exceeds 1000 Pa, which amounts to a sound pressure level of 154 dB re 20 μPa. This level was considered to be sufficiently strong to provide a rigorous test of the effect of downstream airfoils with offsets.

It should be noted that the nature of the acoustic mode in the model tests is different from that observed in the compressor. As mentioned earlier, the resonance occurs in the inlet scroll, which is upstream of the struts, whereas in the model tests, the β-mode is attached to the plate and therefore the trailing edge is positioned near the location of the maximum particle velocity of the resonant mode. Thus, the coupling mechanism between the wake and the resonant mode is stronger in the model. This means that the excitation mechanism of

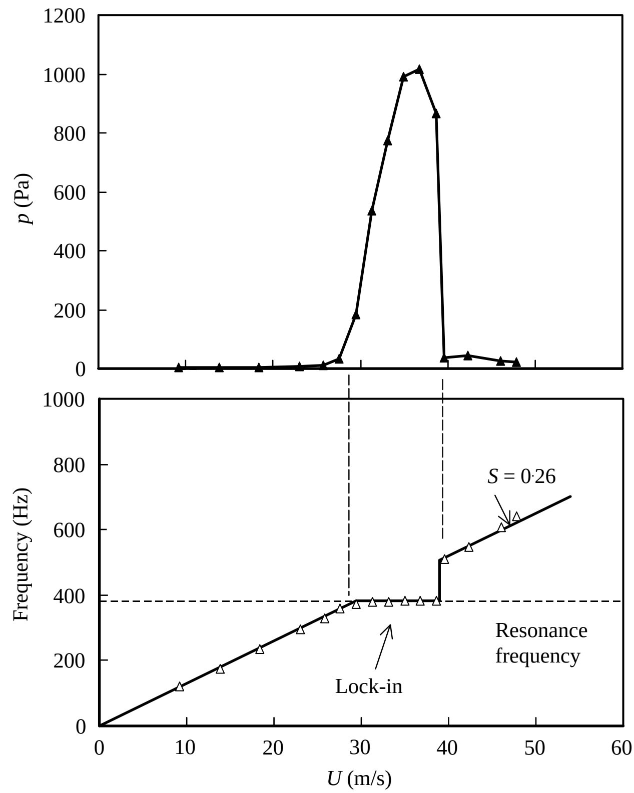

Figure 8. Amplitude and frequency of pressure fluctuation on the tunnel wall above the centre of the plate, $L/D = 9{\cdot}8$. Model tests without downstream airfoil.

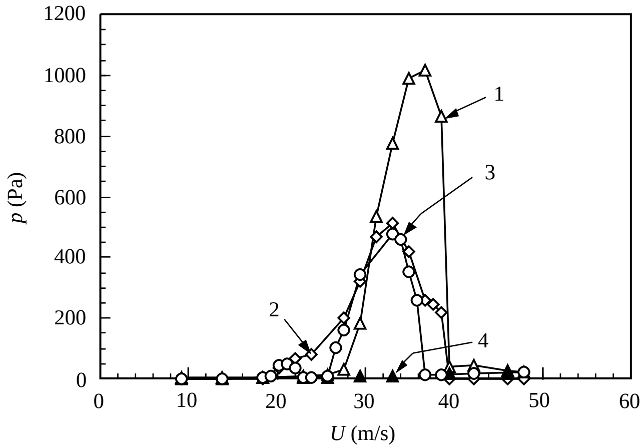

Figure 9. Effect of the streamwise gap (between the plate and the airfoil) on pressure fluctuation. Curve 1, $\Delta x/D = \infty$, Curve 2, 0·5; Curve 3, 0·4; Curve 4, 0·15; transverse offset $\Delta y/D = 0$ in all cases.

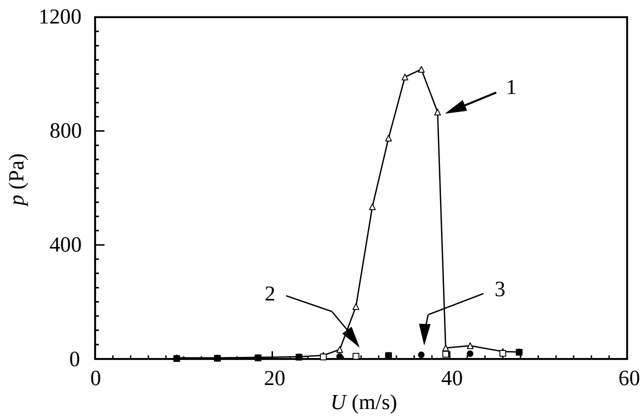

Figure 10. Effect of streamwise gap and transverse offset on pressure fluctuation. Curve 1, without airfoil; Curve 2, with airfoil at $\Delta x/D = 0{\cdot}15$ and $\Delta y/D = 0$; Curve 3: $\Delta x/D = 0{\cdot}15$ and $\Delta y/D = 0{\cdot}25$.

the tested geometry can be considered as a conservative model of that occurring in the compressor.

The system response with the airfoil in the wake of the plate is compared with that without the airfoil in Figure 9. The results correspond to three cases of streamwise gap and *no* transverse offset (i.e. $\Delta y/D = 0$ and $\Delta x/D = 0{\cdot}15$, 0·4 and 0·5). It is clear from these results that the resonance is totally suppressed when the gap $\Delta x/D \leq 0{\cdot}15$. As the gap is gradually increased, the resonance of the β-mode recovers, however, the maximum amplitude for the tested cases does not exceed about 50% of that without the airfoil. The results show also that, for large gaps, the resonance starts at lower velocities, presumably because of the turbulence generated by flow separation between the plate and the airfoil.

The airfoil was then positioned such that the streamwise gap and the transverse offset corresponded to the expected maximum values in the compressor ($\Delta x/D = 0{\cdot}15$ and $\Delta y/D = 0{\cdot}25$). The results are compared with the case without airfoil and also with the case without transverse offset in Figure 10. The airfoil is seen to be very effective in suppressing the acoustic resonance. As mentioned earlier, since the flow acoustic coupling

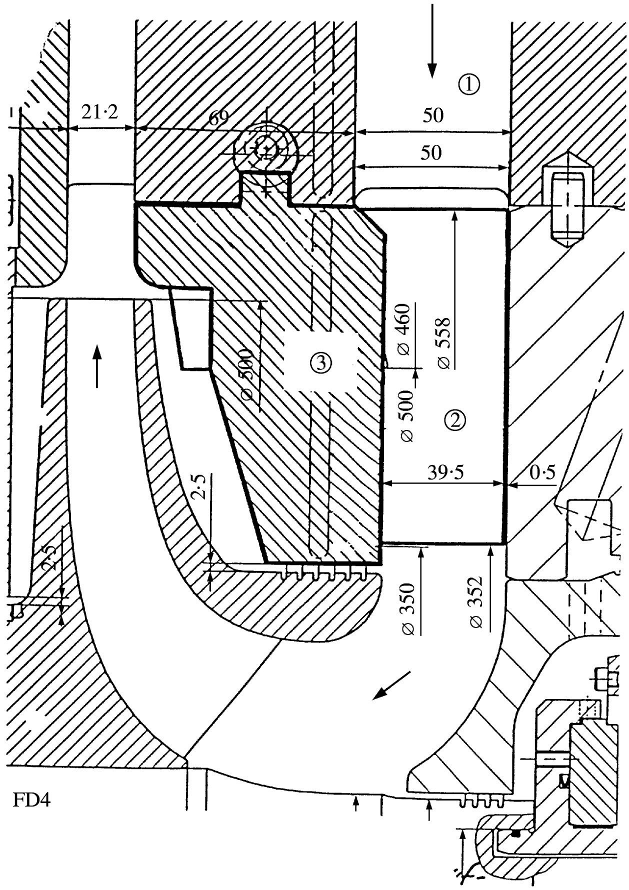

Figure 11. Design modification which was implemented in the compressor: Part 3 was replaced by another with appended airfoils, Part 2, in the wakes of the struts, Part 1.

mechanism is more effective (or stronger) for the tested model, a similar effect was expected when the airfoils are mounted into the compressor.

4.3. In-Plant Implementation of Design Modification

As illustrated in Figure 11, a part of the compressor at the inlet of the first impeller was redesigned to incorporate airfoils aligned with the trailing edges of the struts, see also Figure 6. This modified part was then installed into the compressor during the first shutdown of the unit. Subsequent measurements of rotor vibration showed that the spectral peak at the acoustic resonance frequency (135 Hz) was completely eliminated.

5. SUMMARY

The rotor of a high-pressure turbo-compressor exhibited alarming vibration levels when the intake volume flow and the pressure exceeded critical values. Since it was not possible to conduct detailed pressure measurements, a test program was performed to identify the excitation mechanism from the rotor vibration measurements alone. The results showed that the lowest acoustic mode of the inlet scroll is excited by vortex shedding from struts in a downstream radial flow chamber. The problem was solved by alleviating the vortex shedding excitation by means of small airfoils, which were mounted in the wakes of the struts. However, before incorporating the airfoils in the compressor, the effect of unavoidable small radial and transverse gaps between the struts and the airfoils was investigated by means of wind tunnel tests of a conservative model. The tests showed that gaps within the expected range in the compressor do not degenerate the effectiveness of the airfoils against acoustic resonance. For larger gaps, however, the tests showed that the acoustic resonance recovers, albeit with less intensity than in the case without the airfoil.

References

ARCHIBALD, F. S. 1975 Self-excitation of an acoustic resonance by vortex shedding. *Journal of Sound and Vibration* **38,** 81–103.

BLEVINS, R. D. 1979 *Formulas for Natural Frequency and Mode Shape*. New York: Van Nostrand Reinhold.

BLEVINS, R. D. 1984 Review of sound induced by vortex shedding from cylinders. *Journal of Sound and Vibration* **92,** 455–470.

CUMPSTY, N. A. & WHITEHEAD, D. S. 1971 The excitation of acoustic resonance by wake shedding. *Journal of Sound and Vibration* **18,** 353–369.

HESKESTAD, G. & OLBERTS, D. R. 1960 Influence of the trailing edge geometry on hydraulic-turbine-blade vibration resulting from vortex excitation. *ASME Journal of Engineering for Power* **82,** 103–110.

HOURIGAN, K., WELSH, M. C., THOMPSON, M. C. & STOKES, A. N. 1990 Aerodynamic sources of acoustic resonance in a duct with baffles. *Journal of Fluids and Structures* **4,** 345–370.

KINSLER, L. E., FREY, A. R., COPPENS, A. B. & SANDERS, J. V. 1999 *Fundamentals of Acoustics*, 4th edition. New York: John Wiley & Sons.

OENGÖREN, A. & ZIADA, S. 1992 Vorticity shedding and acoustic resonance in an in-line tube bundle: acoustic resonance. *Journal of Fluids and Structures* **6,** 293–309.

PARKER, R. 1966 Resonance effects on wake shedding from parallel plates: some experimental observations. *Journal of Sound and Vibration* **4,** 62–72.

PARKER, R. 1967 Resonance effects on wake shedding from parallel plates: calculation of resonant frequencies. *Journal of Sound and Vibration* **5,** 330–343.

PARKER, R. & PRYCE, D. C. 1974 Wake excited resonances in an annular cascade: an experimental investigation. *Journal of Sound and Vibration* **37,** 247–261.

PARKER, R. & STONEMAN, S. A. T. 1989 The excitation and consequences of acoustic resonances in enclosed fluid flow around solid bodies. *Proceedings of the Institution of Mechanical Engineers* **203,** 9–19.

NGUYEN, D. T. & NAUDASCHER, E. 1991 Vibration of beams and trashracks in parallel and inclined flows. *ASCE Journal of Hydraulic Engineering* **117,** 1056–1076.

WEAVER, D. S. 1993. Vortex shedding and acoustic resonance in heat exchanger tube arrays. In *Technology for the* 90'*s* (ed. M. K. AU YANG), Part III, Chap. 6, pp. 775–810. New York: ASME.

WELSH, M. C., HOURIGAN, K., WELSH, L. W., DOWNIE, R. J., THOMPSON, M. C. & STOKES, A. N. 1990 Acoustics and experimental methods: the influence of sound on flow and heat transfer. *Experimental Thermal and Fluid Science* **3,** 138–152.