C508/013/96

Torsional instability of a geared compressor shaft train

F VIGGIANO and **J SCHMIED**
Sulzer Turbo Limited, Zurich, Switzerland

SYNOPSIS

For turborotor trains with gear, the torsional and lateral motion of the gear wheels are coupled. Due to this coupling lateral forces in the gear bearings can excite and even destabilize the torsional vibrations of the complete rotor train. In this paper such a case is presented. It deals with a 30 MW compressor plant where strong gear vibrations occurred, when running at low loads. The vibration problem was solved by the exchange of the pinion gear bearings. The theoretical and practical steps to the solution are shown.

1 INTRODUCTION

Turbocompressors are often coupled with the drive, e.g. an electric motor or a turbine, by means of a gear. In the gear box the torsional and the lateral motion of the gear wheels are coupled due to the offset of the shaft centerlines. An energy exchange between these motions is therefore possible by the gear bearings. In that way the radial gear bearings can considerable increase the torsional system damping [1]. However, the gear bearings can also excite the torsional vibrations [2] (for lateral vibrations this effect is refereed to as "Oil Whip").
In this paper such a case is presented. It deals with a 30 MW axial compressor which is driven through a gear by means of a synchronous motor (see figure 2). During the compressor run up the guide vanes are closed, i.e. the load is reduced. At a certain speed the gear vibrations strongly rise. These vibrations disappear when the compressor guide vanes are opened, respectively when the compressor load, hence the load on the gear bearings, is increased. In the following the explanation and solution of this vibration phenomena is presented.

2 DESCRIPTION OF THE VIBRATION PHENOMENON

Figure 1 shows a waterfall diagram of the lateral pinion vibration during run up with closed guide vanes. The compressor running speed is clearly visible. This specific vibration is the lateral unbalance response. At a compressor speed of about 2600 rpm a speed independent vibration of large amplitude with a frequency of 17 Hz appears. After the run up, respectively when the compressor guide vanes are opened and therefore the compressor load and the force on the gear bearings is increased, this vibration disappears.

As the first torsional system eigenmode is situated at 17 Hz, a connection between these large gear vibrations and the first torsional eigenmode had to exist. Because of the coupling of the lateral and torsional motion in the gear the torsional vibrations can be detected with the radial vibration probes at the pinion. Indeed it was not clear, whereby the torsional eigenmode was excited so strongly. First it was assumed, that transient, sporadic torque pulsations of the electric motor were exciting the vibrations. The following study however clearly shows, that the described vibration is caused by the first torsional eigenmode, which is excited by the low loaded gear bearings.

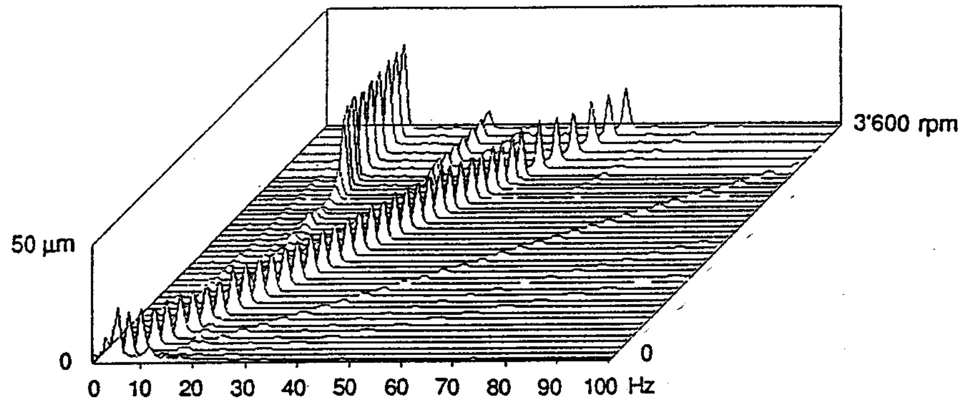

Fig 1 Waterfall diagram of the lateral pinion vibration during run up

3 SYSTEM MODELING

Since there seemed to be a connection between the large lateral pinion vibrations and the torsional eigenmode at 17 Hz, the coupled torsional-lateral vibration behaviour of the complete rotor system was studied. Figure 2 shows the finite element model of the rotor system. In contrast to the usual rotor dynamic analysis, where torsional and lateral vibrations are regarded separately, in this case every node of the model can effect torsional as well as lateral movements. Because of the helical gear toothing a certain coupling of the axial motion with the other vibrations also excists. This coupling however is substantially weaker than the coupling between the torsional and lateral vibration and will be neglected.

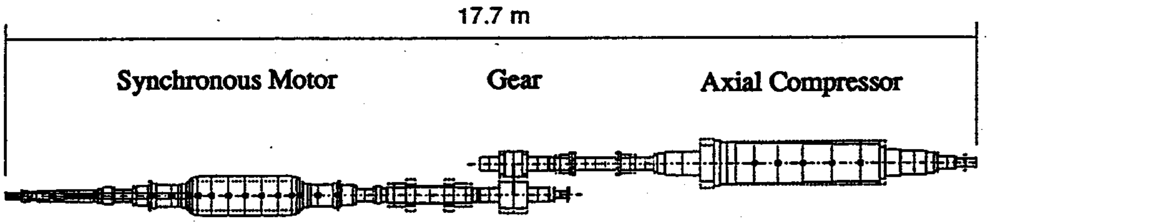

Fig 2 Finite element model of the rotor system

For the kinematic coupling in the gear it is assumed, that the gear wheel and pinion are rigid and stay in contact in the vertical y-direction, while they can move freely in the horizontal x-direction (see figure 3). For small movements the kinematic relationship can be expressed as follows

$$r_1\varphi_1 - y_1 = r_2\varphi_2 - y_2 \qquad (1)$$

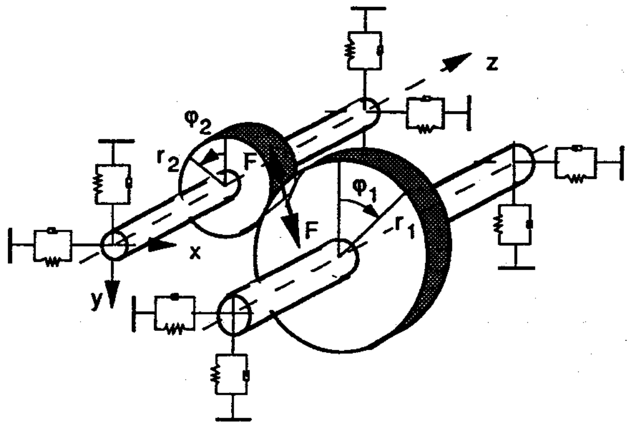

Fig 3 Coupling of the torsional and lateral motion in the gear

An additional boundary condition results from the bearing support of the rotor system. At the motor- and compressor bearings the torsional and lateral motion are not coupled. These bearings have no influence on the above described vibration phenomenon and are assumed as rigid (infinite stiffness). However the journal bearings in the gear can influence the torsional vibrations considerably because of the above described kinematic coupling. Therefore the properties of the gear bearings have to be considered. For small lateral displacements x, y the bearing forces are given by

$$\begin{bmatrix} F_x \\ F_y \end{bmatrix} = \begin{bmatrix} d_{xx} & d_{xy} \\ d_{yx} & d_{yy} \end{bmatrix} \begin{bmatrix} \dot{x} \\ \dot{y} \end{bmatrix} + \begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix} \qquad (2)$$

The damping and the stiffness coefficients in the above equation depend on the rotor speed, the bearing load, respectively the transmitted gear load, and the bearing type. The bearing coefficients and the coupled torsional-lateral eigenmodes can be calculated if these parameters are known.

4 SIMULATION RESULTS

For the simulations the finite element program MADYN was used. As a simulation result figure 4 shows the first coupled system eigenmode at rated speed (motor: 3000 rpm, compressor: 4360 rpm) and a gear load of 9 MW. The bearing coefficients in equation (2)

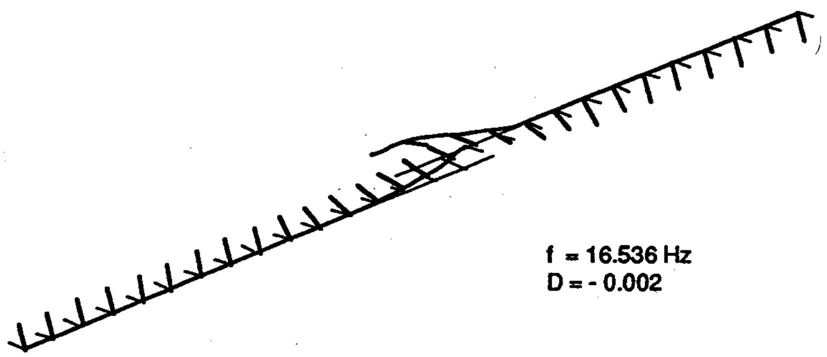

Fig 4 Shape of the first coupled system eigenmode (P = 9 MW, n = 4360 rpm)

were calculated for the lemon type bearings originally mounted in the gear. One can see that in the gear area there is a visible lateral vibration. The torsional vibration is made visible by means of massless beams perpendicular to the shaft axes. The torsional deflection as well as the frequency of this mode correspond to the first torsional eigenmode calculated for the rotor system without consideration of the coupling effect. The damping ratio of the coupled eigenmode is slightly negative, i.e. it is just below the stability threshold.
In figure 5 the calculated damping ratio of the first coupled system eigenmode at rated speed is shown as a function of the transmitted gear load and for different pinion gear bearings. One can see, that with the originally mounted lemon type bearings the system becomes unstable at loads below 10 MW. On site a severe increase of gear vibrations was observed at loads below 9 MW. As structural damping effects have been neglected for the simulation, the calculated and the measured stability threshold coincide quite well.
In case the lemon type bearings in the pinion are replaced by four lobe bearings the stability behaviour is inverted, i.e. with decreasing gear power the stability is increased. Now the coupled system eigenmode can only be excited by an external force, e.g. by pulsating motor torques.

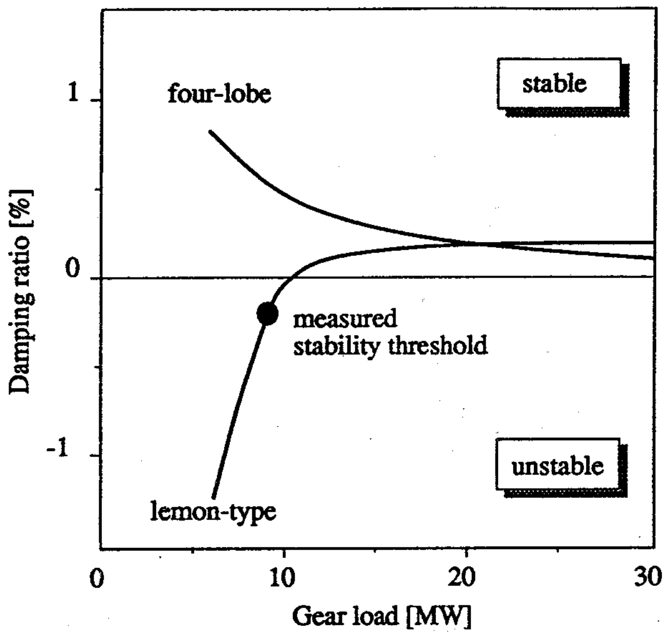

Fig 5 Damping ratio of the first coupled eigenmode in function of the transmitted gear load and for different pinion gear bearings at rated speed.

According to these results the observed vibration phenomenon is an instability of the first coupled eigenmode caused by self-excitation in the lemon type bearings of the pinion gear. The following consideration supports this result. We observe the orbit of the unstable motion in the pinion gear bearing and calculate the energy, which is conveyed to the rotor (excitation energy E_A) respectively dissipated (damping energy E_D) during one vibration period. The calculation is shown in detail in the appendix. Figure 6 shows the calculated orbit of the pinion shaft at a load of 9 MW and a compressor speed of 4360 rpm (not considering the vibration growth). The shape of the orbit results from the eigenmode calculation. It is an ellipse in the case of the lemon type bearings. The magnitude of the ellipse axes can not be determined by calculation and was chosen in a manner, that it corresponds to the measured orbit (see paragraph 5). For the stable motion in the four lobe bearings the shaft is just moving on the vertical axis (assumed maximum amplitude: 50 µm), because in the simulation

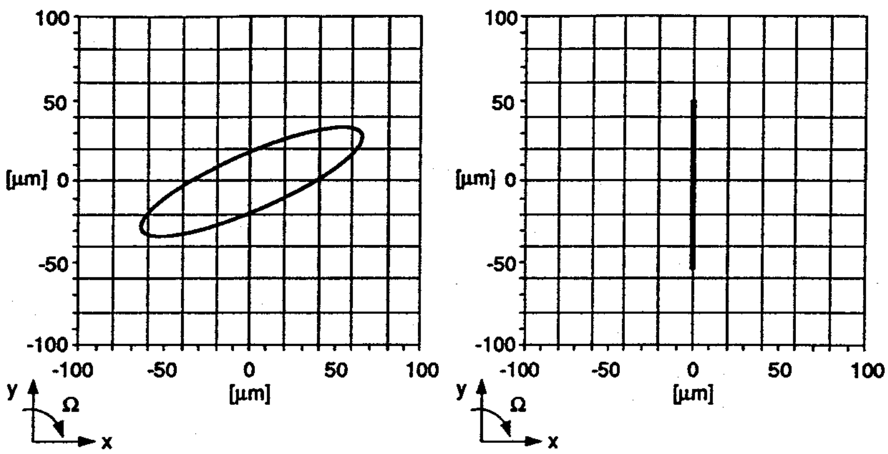

Fig 6 Pinion shaft motion in the lemon type bearing (left) and in the four lobe bearing (right)

the lateral forces on the gear bearings caused by a torsional motion are only in vertical direction. In the case of the lemon type bearings the orbit is also pushed in the softer horizontal direction by the whirling oil.
The energies (for one bearing) of these orbits are given in Table 1. In the case of the lemon type bearing the excitation energy is larger than the damping energy, i.e. the system vibrations will be excited by the whirling oil. On the other hand for the four lobe bearings the excitation energy is practically zero.

Table 1 Excitation and damping energy of the 17 Hz Vibration at 9 MW and 4360 rpm

	lemon type bearing	four lobe bearing
excitation energy	1.81 Nm	≈ 0 Nm
damping energy	1.27 Nm	6.55 Nm

5 MEASUREMENTS

Because of the simulation results the two lobe bearings in the pinion gear were replaced by four lobe bearings. Before as well as after the bearing exchange measurements were made. In addition to the lateral gear vibrations the torsional vibration was also measured. For this purpose a tachometer was mounted at the compressor free end.
Figures 7 and 8 show the measured frequency response of the lateral pinion gear vibration and the torsional vibration before and after the gear modification. For the measurements the gear load was reduced by closing the compressor guide vanes. One can see, that before the modification for both the lateral as well as the torsional vibration a dominant part at 17 Hz appears below 9 MW load. Additionally multiples of the 17 Hz vibration appear. After the gear modification these vibrations completely disappear. For the pinion gear just the synchronous unbalance response at 72.6 Hz remains.
Figure 9 shows the influence of the bearing exchange in the pinion on the run up behaviour of the compressor. Before the modification a large gear vibration with the constant frequency of 17 Hz appears when surpassing a speed of 2600 rpm. After the modification only the synchronous unbalance response is visible.

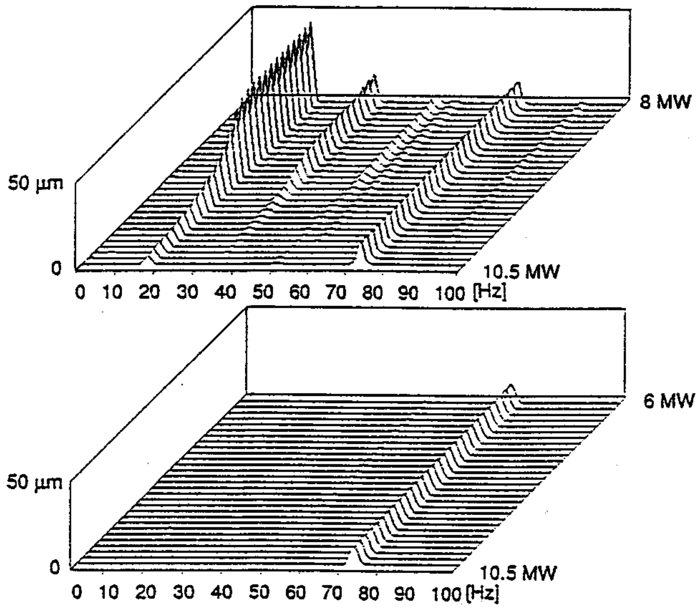

Fig 7 Measured frequency response of the lateral pinion gear vibration before and after the bearing exchange. The vibration behavior is shown for decreasing gear load (z-axis).

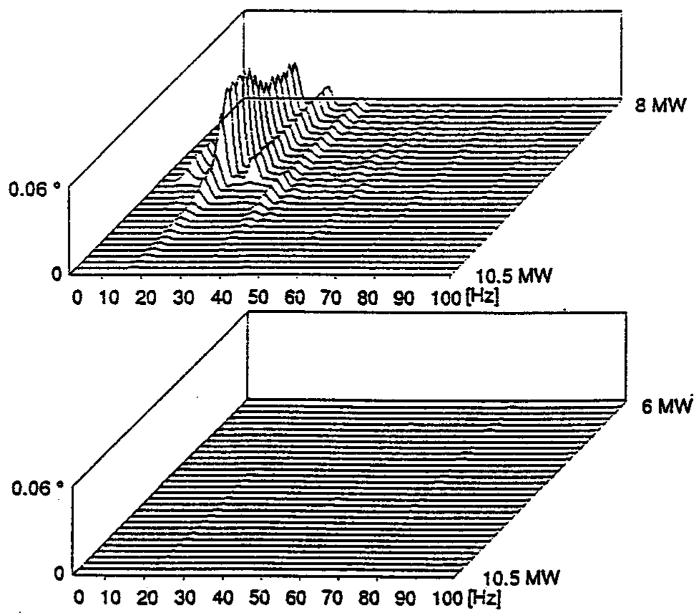

Fig 8 Measured frequency response of the torsional vibration before and after the bearing exchange. The vibration behavior is shown for decreasing gear load (z-axis).

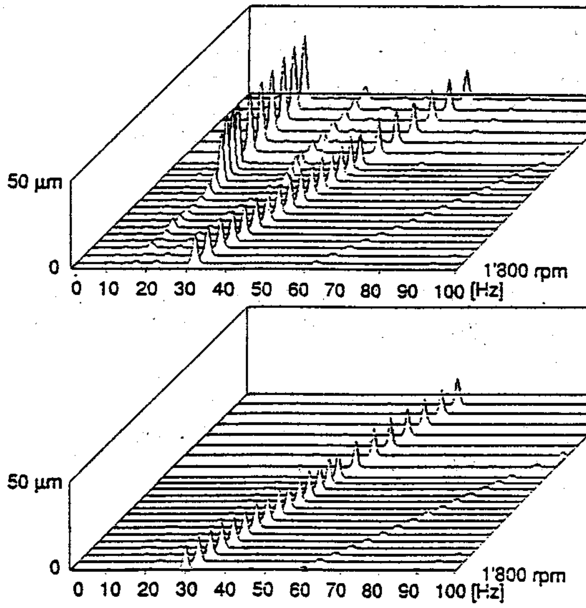

Fig 9 Measured frequency response of the lateral pinion vibration during run up before and after the bearing exchange.

Figure 10 finally shows the measured orbits of the pinion gear in the lemon type, respectively four lobe bearing, for a rotor speed of 4360 rpm and a load of 9 MW. In the case of the lemon type bearing higher frequency vibrations are superimposed on the elliptical orbit, which represents the motion of the unstable coupled eigenmode at 17 Hz. The rotating direction and inclination of the measured ellipse coincide well with the simulated orbit shown in figure 6. After the exchange of the lemon type bearings with four lobe bearings just the synchronous unbalance response is visible. For this measurement the coupled eigenmode is stable and is only excitable by external forces. As there is no motor excitation at these operating conditions the simulated vertical motion presented in figure 6 cannot appear.

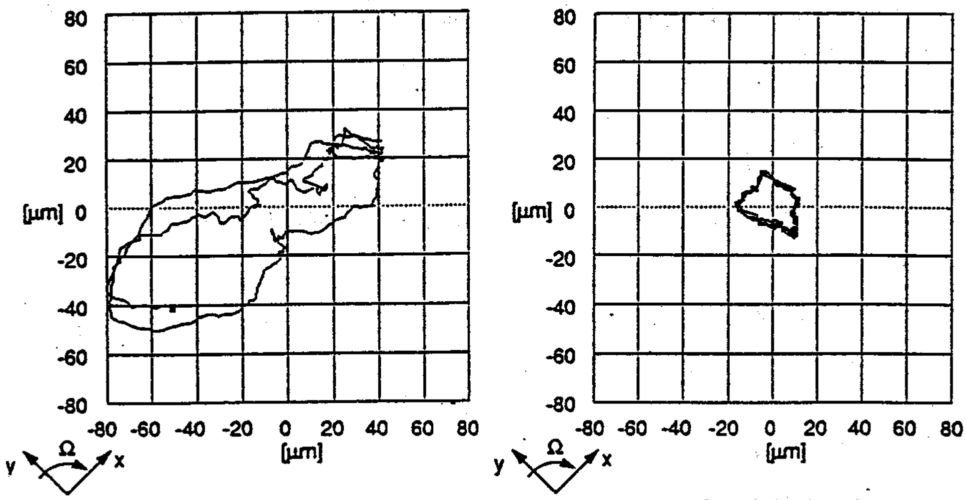

Fig 10 Measured pinion shaft motion in the lemon type and in the four lobe bearings.

6 SUMMARY

The first torsional eigenmode of a rotor system consisting of a motor, a gear and a compressor was destabilized by the low loaded radial bearings in the pinion gear. By the coupling of the torsional and the lateral motion in the gear the radial bearing force excited the torsional vibrations. This could clearly be verified by simulations as well as by measurements. The problem was solved by the exchange of the originally mounted lemon type bearings by four lobe bearings .

REFERENCES

[1] VIGGIANO, F. and WATTINGER, W. A Torsional Damping Model for Geared Compressor Shaft Trains, Vibration in Fluid Machinery, 1994 (Mechanical Engineering Publications, London), pp. 1-6.

[2] SCHWIBINGER, P. and NORDMANN, R. The Influence of Torsional-Lateral Coupling on theStability Behaviour of Geared Rotor Systems, Proc. of 4th Workshop on Rotor Dynamic Instability Problems in High-Performance Turbomachinery, Texas A&M University, 1986.

APPENDIX

The energy which is conveyed to respectively away from the rotor during one vibration period is

$$E = \int_0^T F_x \dot{x}\, dt + F_y \dot{y}\, dt \tag{1}$$

whereby F_X and F_y are the linear bearing forces in the lateral directions x and y due to a motion x(t) and y(t). The non-conservative parts of F_X respectively F_y are

$$F_x = k_{xy}\, y + d_{xx}\, \dot{x} \tag{2}$$

$$F_y = k_{yx}\, x + d_{yy}\, \dot{y} \tag{3}$$

with the bearing coefficients k_{xy}, k_{yx}, d_{xx} and d_{yy}. For x and y a harmonic motion with the frequency ω is assumed

$$x = \hat{x} \cos(\omega t + \phi_x) \tag{4}$$

$$y = \hat{y} \cos(\omega t + \phi_y) \tag{5}$$

By combining the equations (2) to (5) with equation (1) the energy results as

$$E = E_D - E_A \tag{6}$$

with the excitation energy

$$E_A = -\pi \hat{x} \hat{y} \left[(k_{xy} - k_{yx}) \left(\sin\phi_x \cos\phi_y - \cos\phi_x \sin\phi_y \right) \right] \tag{7}$$

and the damping energy

$$E_D = \pi\, \omega \left(d_{xx} \hat{x}^2 + d_{yy} \hat{y}^2 \right) \tag{8}$$