Vibration Instabilities on a Steam Turbine Generator Line

Kurzfassung

Schwingungsinstabilitäten am Dampfturbinengeneratorstrang

Bei der Inbetriebnahme eines 8,5 MW-Generatorstrangs in Österreich, bestehend aus Dampfturbine, einstufigem Stirnradgetriebe und Generator, wurden im Teillastbereich stark erhöhte Wellenschwingungen festgestellt. Die Endabnahme wurde vom Betreiber verweigert. Da die Ursache für die Schwingungserhöhung unbekannt war, wurde KÖTTER Consulting Engineers mit einer umfangreichen messtechnischen Untersuchung beauftragt. Es konnten zwei Problembereiche hinsichtlich erhöhter Schwingungen festgestellt werden. Unabhängig vom Lastbereich traten lokal begrenzt am ausgangsseitigen Lager des Niederdruckteils erhöhte Lagerbockschwingungen auf. Die Schwingungsanregung erfolgte durch die langsamlaufende Getriebewelle über den gemeinsamen Rahmenaufbau. Verantwortlich hierfür war die konstruktive Ausführung des Lagers, die unter anderem bei 25 Hz eine mechanische Eigenfrequenz aufwies. Im Teillastbereich bis 3 MW konnten am Getriebe erhöhte Lagerbockschwingungen sowie erhöhte Wellenschwingungen an der schnellaufenden Ritzelwelle bis zu s_{max} *= 95 µm festgestellt werden. Die erhöhten Schwingungen traten in erster Linie als Einzelfrequenzen bei dem 0,3-fachen der Drehfrequenz (*f_{Dreh} *= 201 Hz) auf. Diese subsynchrone Frequenz wies in Abhängigkeit von der aufgegebenen Last leichte Frequenzänderungen auf. Zur weiteren Analyse der erhöhten Wellenschwingungen wurde die Orbitdarstellung für unterschiedliche Lasten verglichen. Es zeigte sich, dass insbesondere an den Ritzelwellenlagern im Teillastbereich von 2 MW stark erhöhte Amplituden festzustellen waren. Die Laufrichtung, mit der die Orbits durchlaufen werden, war gleichlaufend mit der Wellendrehrichtung. Ab einer Last von ca. 4 MW stellte sich wiederum ein stabiles dynamisches Verhalten der Ritzelwelle mit Schwingungswerten unter 20 µm (p-p) ein. Die festgestellten Phänomene deuteten auf selbsterregte Schwingungen hin. Mögliche Quellen für diese Instabilitäten können Gleitlager, Schaufelspiele oder Dichtungsspalte sein. Das Vorstadium z. B. einer Gleitlagerinstabilität tritt meist mit mäßigen Amplituden und einer Instabilitätsfrequenz unterhalb der halben Drehfrequenz auf („oil-whirl"). Bei Übereinstimmung dieser Frequenz mit einer mechanischen Welleneigenfrequenz kommt es zu gefährlichen lastabhängigen Schwingungen („oil-whip"). Dies bestätigte sich beim Hochlauf der Turbine. Es zeigte sich, dass bei ca. 60 Hz eine Schwingungsüberhöhung auftrat, die auch aufgrund der vorgefundenen Phasenlage und der Schwingungsform auf die erste Biegeeigenfrequenz der Turbinenwelle zurückgeführt werden konnte. Weiterhin wurde festgestellt, dass die eingesetzten Gleitlager der Ritzelwelle hinsichtlich des Lagerspiels und der Umfangsgeschwindigkeit im grenzlastigen Bereich der Ritzelwelle gefahren wurden. Als Gegenmaßnahme wurde eine Modifizierung des Gleitlagersystems vorgeschlagen. Ferner wurde zur Minderung der aufgeführten Lagerbockschwingungen eine zusätzliche Versteifung empfohlen. Die Lagerbockschwingungen konnten nach Realisierung der Maßnahmen auf Werte von maximal v = 1,8 mm/s eff (vorher v = 7,1 mm/s eff) reduziert werden. Darüber hinaus waren keine Instabilitäten bzw. erhöhte Schwingungen der schnellaufenden Welle mehr festzustellen. Im bemängelten unteren Lastbereich wurden maximal 14 µm p-p radiale Wellenschwingungen festgestellt, sodass die Anlage dem Betreiber „schwingungsfrei" übergeben werden konnte.*

Author

Dr.-Ing. J. Lenz
Head of department structure dynamic, KÖTTER Consulting Engineers, Rheine/Germany.

Status

Depending on the resources available, gas as well as steam admission turbine plants are used for power conversion. During the start-up of such a generator line in Austria (Figure 1), considerably increased shaft vibrations were detected in the partial load range. As a result, the plant operator refused acceptance. The reason for the increase in vibrations was unknown, therefore, KÖTTER Consulting Engineers was commissioned to perform a comprehensive measurement-technical analytical examination.

Measurement

Figure 2 shows the principle arrangement of the generator line as well as the locations of the adopted measuring points. Proceeding from the steam turbine (rated output 8.5 MW) with a high-pressure part and a low-pressure part as well as a fixed operating speed of 12,065 1/min (f_N = 201 Hz), the torque is transferred to the slow-running generator shaft with 1,500 1/min (f_G = 25 Hz) by way of a single-stage gear with pinion and wheel shaft.

For the purpose of performing the measurement operation, the existing shaft vibration sensors as well as additionally installed vibration speed sensors on the bearing housings were included simultaneously with the speed at varying operating conditions (32 channels).

Based on the measurements performed, two problem zones were detected with reference to increased vibrations:

— Independent of the load range, increased bearing housing vibrations occurred locally confined at the outlet-side bearing of the low pressure part (DMP2_Turb-HP_Rear_h). The effective vibration speed of $v_{eff.}$ = 7 mm/s measured in the horizontal direction was primarily determined by a 25 Hz-component.

Figure 1. Photo of a steam turbine generator line (not the problem case).

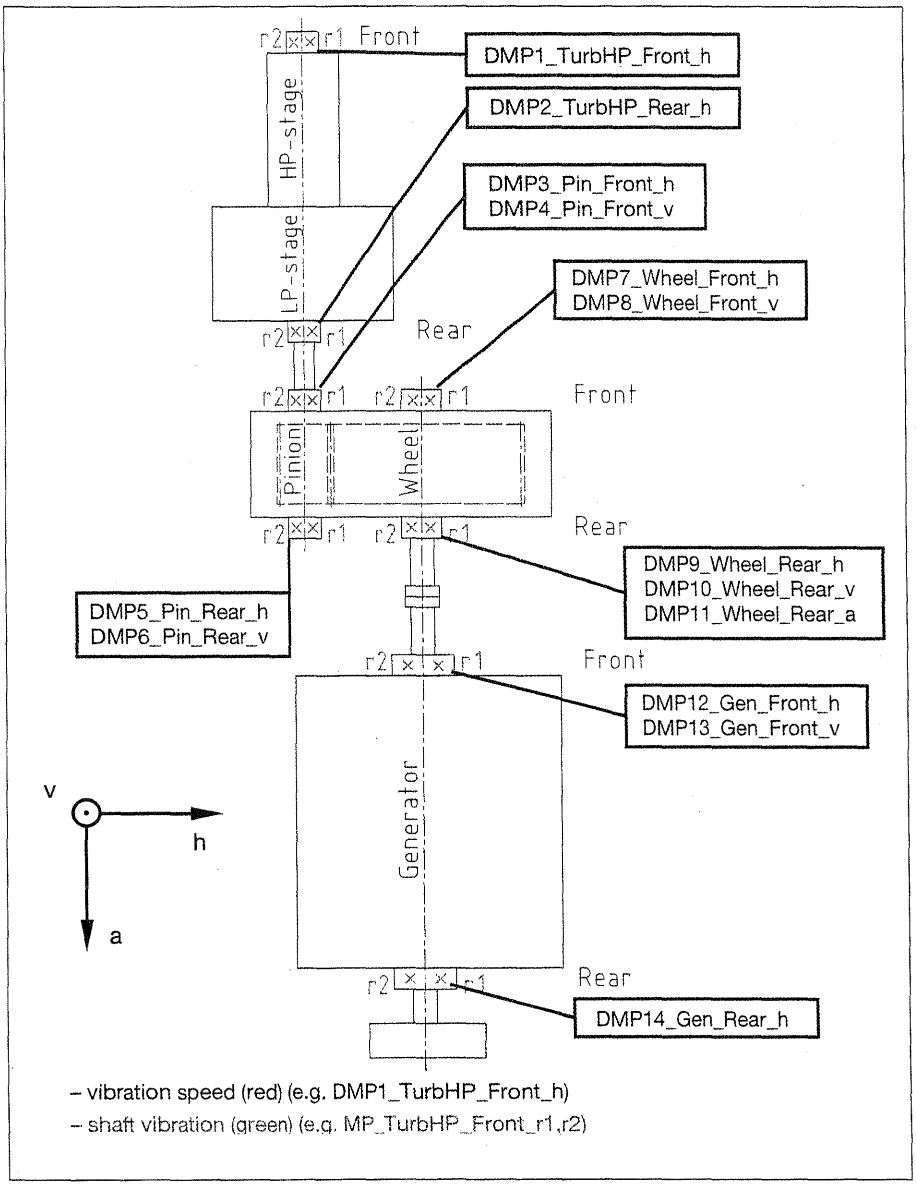


Figure 2. Principle sketch of the generator line as well as locations of the measuring points.

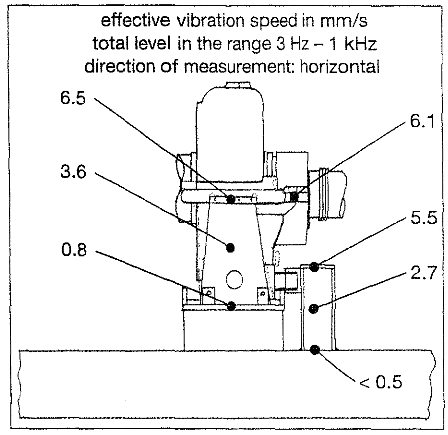


Figure 3. Measured effective vibration speeds in the zone of the turbine bearing on the low-pressure side, in the horizontal direction.

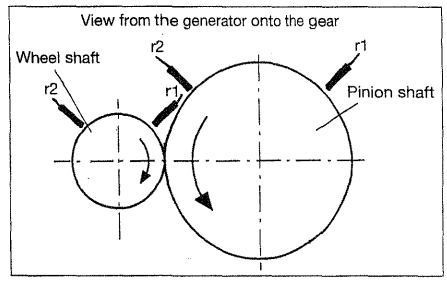


Figure 4. Position of the shaft vibration sensors r1 and r2 on the pinion and wheel shaft.

— In the partial load range (0.3 MW to 2 MW) increased bearing housing vibrations $v_{eff.}$ = 5.2 mm/s were detected at the gear and increased shaft vibrations were detected at the pinion shaft S_{max} = 95 μm. The increased vibrations primarily occurred as single frequencies at approximately 60 Hz. As a factor of the load, this subsynchronous frequency indicated minor changes.

For the purpose of a closer analysis of the increased bearing housing vibrations, the vibration level (effective vibration speed) was observed at various positions in the horizontal direction in the zone of the turbine bearing on the low-pressure side.

It was evident that the horizontal vibration level, proceeding from the foundation frame up to the level of the shaft line, had increased considerably (Figure 3). In order to record this amplification mechanism, impact tests were carried out at the bearing housing, with the plant shut down, for the purpose of determining the horizontal dynamic transmission behaviour. The impact test was carried out in the zone of the actual bearing housing (level of the shaft line) with a modal hammer tuned to a frequency range to be excitated up to 300 Hz.

In the range of approximately 26 Hz, a natural frequency was clearly detected in the horizontal direction at the bearing housing of the low-pressure side turbine.

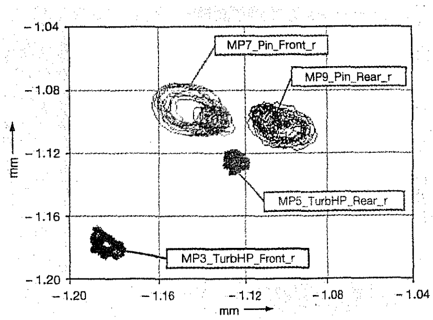


Figure 5. Orbit illustration of the shaft vibrations of turbine and pinion shaft at 0.3 MW (r1 - axis horizontal, r2 - axis vertical).

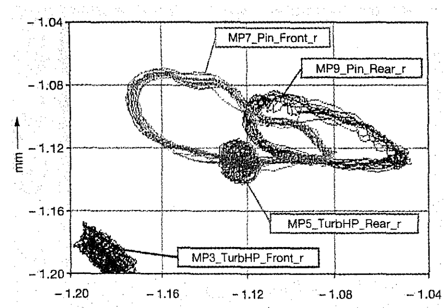


Figure 6. Orbit illustration of the shaft vibrations of turbine and pinion shaft at 2 MW (r1 - axis horizontal, r2 - axis vertical).

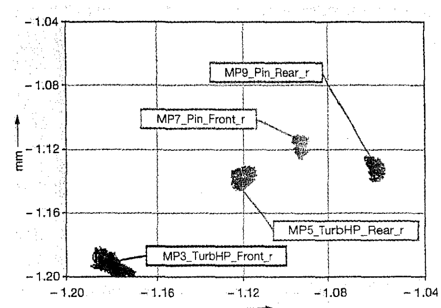


Figure 7. Orbit illustration of the shaft vibrations of turbine and pinion shaft at 4 MW (r1 - axis horizontal, r2 - axis vertical).

For the purpose of further analysis of the increased pinion shaft vibration in the partial load range, the orbit illustrations of the fast-running shaft line were compared for various loads. In order to serve orientation, the positions of the shaft vibration sensors (r_1, r_2) are shown in Figure 4 in an exemplary manner for pinion and wheel shaft.

Figures 5 to 7 show the orbits of the fast-running shaft for a stationary load of 0.3, 2 and 4 MW.

Considerably increased amplitudes are detectable particularly at the pinion shaft bearing in the partial load range of 2 MW (Figure 6). Higher shaft vibrations occur here also at the turbine shaft. However, judging by the size of the occurring disturbance amplitudes, no conclusions can be drawn as to the location of origin of the vibrations. The comparison of Figures 5 to 7 also shows how the shaft location, particularly at the pinion shaft bearing, changes considerably under minor load changes. The running direction, with which the orbits are passed through, is concurrent with the shaft direction of rotation. As from a load of approximately 4 MW, a very stable dynamic behaviour of the pinion shaft settles in again, with vibration levels below 20 µm p-p (previously 95 µm p-p).

The described observations of this sub-synchronous shaft vibration are summarised as follows:

- In the partial load range between 0.3 to 3.0 MW, increased shaft vibrations occur at the pinion shaft in particular.
- The frequency dominating the vibration lies at approximately 1/3 of the rotation frequency (sub-synchronous shaft vibrations, refer also to Table 1). The vibration frequency undergoes a minor change depending on the load condition.
- The direction of rotation of the increased sub-synchronous vibration is "concurrent" to the direction of rotation of the shaft.

Table 1. Measured vibration frequencies and shaft rotation frequencies in dependence of the load condition of the turbine line.

Load [MW]	Main vibration frequency f_H [Hz]	Rotation frequency of the shaft f_N [Hz]	Frequency ratio f_H/f_N
0.3	57.5	201.1	0.28
1	58.3	201.1	0.29
2	60.6	201.0	0.30
3	63.8	201.1	0.31

The occurrence of these determined vibration phenomena indicates a general instability problem of the fast-running shaft.

Introduction to the Theory of Vibration Instability

With most of the occurring vibration appearances, the forced resonance amplitude is reduced by the occurring damping, or the shock-excitated vibration decays, respectively.

Insofar as self-excitated vibrations occur, increased amplitudes result, if forces are formed which counteract the damping forces. If these excitating forces neutralise the damping forces, an instability (upswing) of the vibrations occurs which can lead to major amplitudes. The sources for such self-excitation are journal bearings, blade clearances or seal clearances. Of the types of instabilities mentioned here, the journal bearing instability is the most frequent. The following points are to be observed when journal bearing instability occurs:

- The preliminary phase of a journal bearing instability usually occurs with moderate amplitudes and a frequency below half of the rotating frequency (oil-whirl).
- The vibration instability is always concurrent with the direction of rotation of the shaft and is usually load-dependent.
- With a conformity of the oil-whirl-frequency and a mechanical natural frequency, the result is conspicuous and considerably increased load-dependent vibrations (oil-whip).

In the following Figure 8, these basic phenomena are shown in principle. The vibration amplitudes are plotted as a two-side Campbell diagram during the capacity-increase of a machine. As a diagonal traced, the single rotating frequency (+ 1 x) is shown as a dashed line and a further diagonal below the half rotating frequency (+ 0.47 x). On observing the single rotating frequency component during the recorded capacity-increase of the machine, the typical vibration camber occurs briefly when passing over the first natural frequency. If we compare to this the component of 0.47-times the speed, typically irregular vibration increases can be recognised (whirl) up to that time where the first natural frequency is reached here also. Increased vibrations then occur at this fixed frequency. This is also described as a "lock-in" (whip).

Examination of the Vibration Behaviour as Found

The decisive factor is that, as a rule, critical vibrations occur only when the oil-whirl-frequency and a natural frequency of the shaft coincide. In order to examine the location of the first bending natural frequency of the fast-running shaft, the capacity-increase of the turbine shaft in this case was recorded

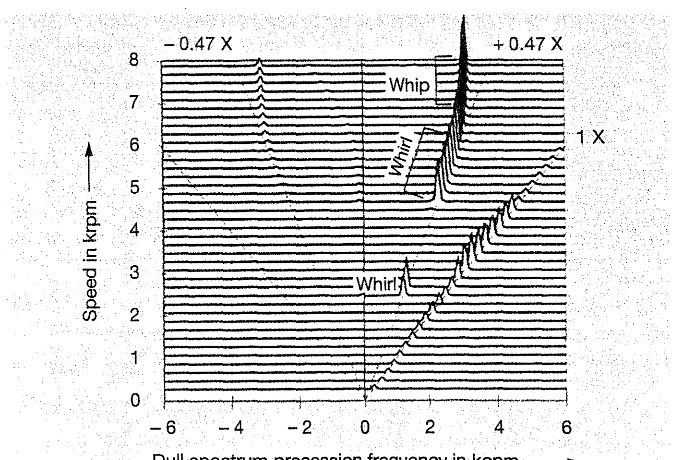


Figure 8. Two-side Campbell diagram with illustration of the oil-whirl and oil-whip phenomenon during the capacity-increase of a machine [1].

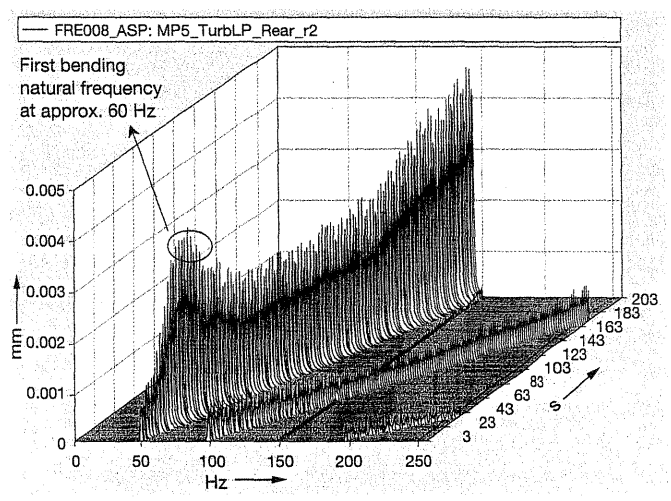


Figure 9. Waterfall diagram as amplitude spectrum of the radial shaft vibrations during capacity increase of the turbine shaft.

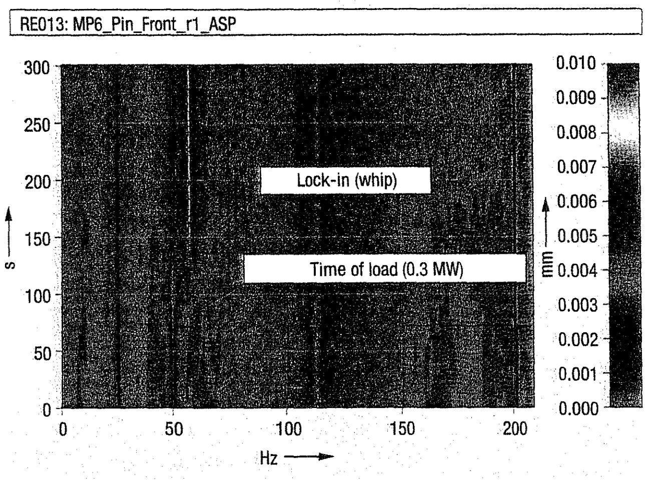


Figure 10. Colour map of the pinion shaft vibration at the beginning of load application.

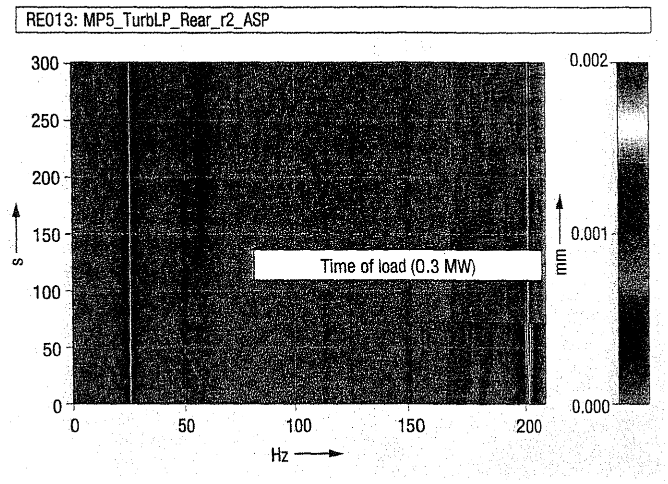


Figure 11. Colour map of the turbine shaft vibrations at the beginning of load.

from approx. 3,000 to 9,000 1/min. This is illustrated in Figure 9 as a waterfall diagram of the radial turbine shaft vibrations.

It can be seen that, at approx. 60 Hz, a first vibration camber occurs which can be attributed to the first flexural natural frequency. This is confirmed based on the existing phase location of the shaft vibration. During the capacity-increase, however, the instability phenomena did not occur as shown in Figure 9. In fact the sub-synchronous vibrations were detected only after reaching the nominal speed and after the synchronisation of the generator, meaning, at the time of load.

Based on the simultaneously recorded signals, the following illustration shows how the sub-synchronous vibrations originate. For this purpose, the amplitude spectrums of the shaft vibrations at the pinion shaft and at the turbine shaft upon transition of the synchronisation to load of 0.3 MW are illustrated in the colour maps of the Figures 10 and 11.

It can be recognised in Figure 10 that, at the time of load and proceeding from a broad-band spread vibration amplitude at approximately 115 Hz, amplitude and frequency fluctuations in the range of 50 Hz (T = 90 to 120 s) occur and then, as from the time T = 125 s, an amplitude at a fixed frequency of approximately 57 Hz settles in. It is noticeable that, at first, vibrations at approximately 0.6-times the rotating frequencies are observed which then, after load has been given up, diminish and increased vibrations settle in at approximately 0.3-times the rotating frequencies.

Figure 11 shows this sequence at the turbine shaft with a slight delay. However, it is not possible to clarify exactly where the phenomenon first occurred.

For the purpose of a further comparison, Figure 12 shows the load case with 3 MW (constant) as a colour map. It is obvious that this is a "lock-in" of the determined phenomenon on the flexural natural frequency (oil whip). During the course of events, load-dependent fluctuations with regard to frequency are no longer recognisable.

The furthermore determined sub-synchronous frequency change from approximately 57 Hz at min-load to approximately 64 Hz at a load of 3 MW was attributed to the influence of the slightly changing overall bearing stiffness on the flexural natural frequency of the turbine shaft.

Conclusion, Realisation and Examination of the Recommended Measures

From the preceding statements it can be concluded that the design execution of the bearing unit which has, among other things, a mechanical natural frequency at 25 Hz and a rather minor stiffness in the horizontal direction, is responsible for increased bearing housing vibrations occurring at the outlet-side bearing of the low pressure part of the turbine.

The cause of the increased sub-synchronous shaft vibration is the occurrence of an instability in conjunction with the bending natural frequency of the fast-running line (turbine shaft). The results of the examination indicate a journal bearing instability of the pinion shaft (oil-whip). The detected sub-synchronous vibration frequency lies in the range between 0.28-times and 0.31-times the rotating frequency. The measurements verify that the instability occurs concurrent with the direction of rotation of the shaft and on a load-dependent basis.

The criterion for general instability is a limit value of the logarithmic decrement. The calculated value of the turbine lies at $\vartheta = 0.3$ and can therefore be evaluated as being non-critical. The applied journal bearings of the pinion shaft, by contrast, were operated in the limit range of the pinion shaft with regard to bearing clearance and the rotational speed. From this and from the previously explanatory statements on the findings, the cause was attributed to the instability of the pinion shaft journal bearings.

A modification of the pinion shaft journal bearing system was recommended as a remedial measure for the purpose of avoiding instability. In this case, influence can be effected by way of an alteration of the bearing geometry (bearing type, width ratio, clearance) and also by way of the bearing loading and/or the viscosity of the lubricant. In addition to this, a stiffening of the low-pressure side bearing

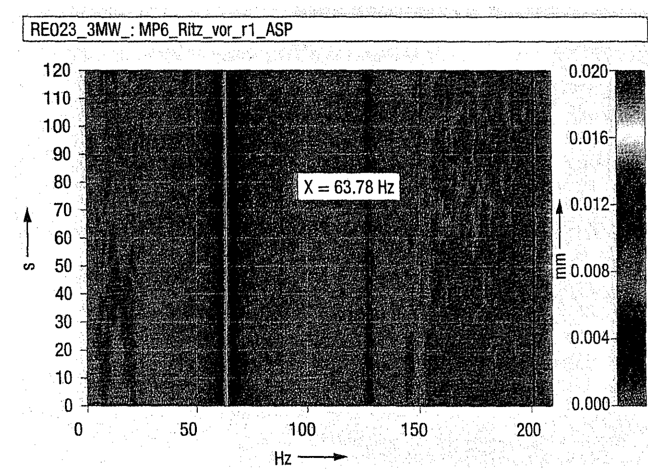


Figure 12. Colour map of the sub-synchronous vibration of the pinion shaft "locked-in" on the flexural natural frequency of the turbine shaft.

housing in the horizontal direction was proposed for the purpose of reducing the bearing housing vibration as mentioned above.

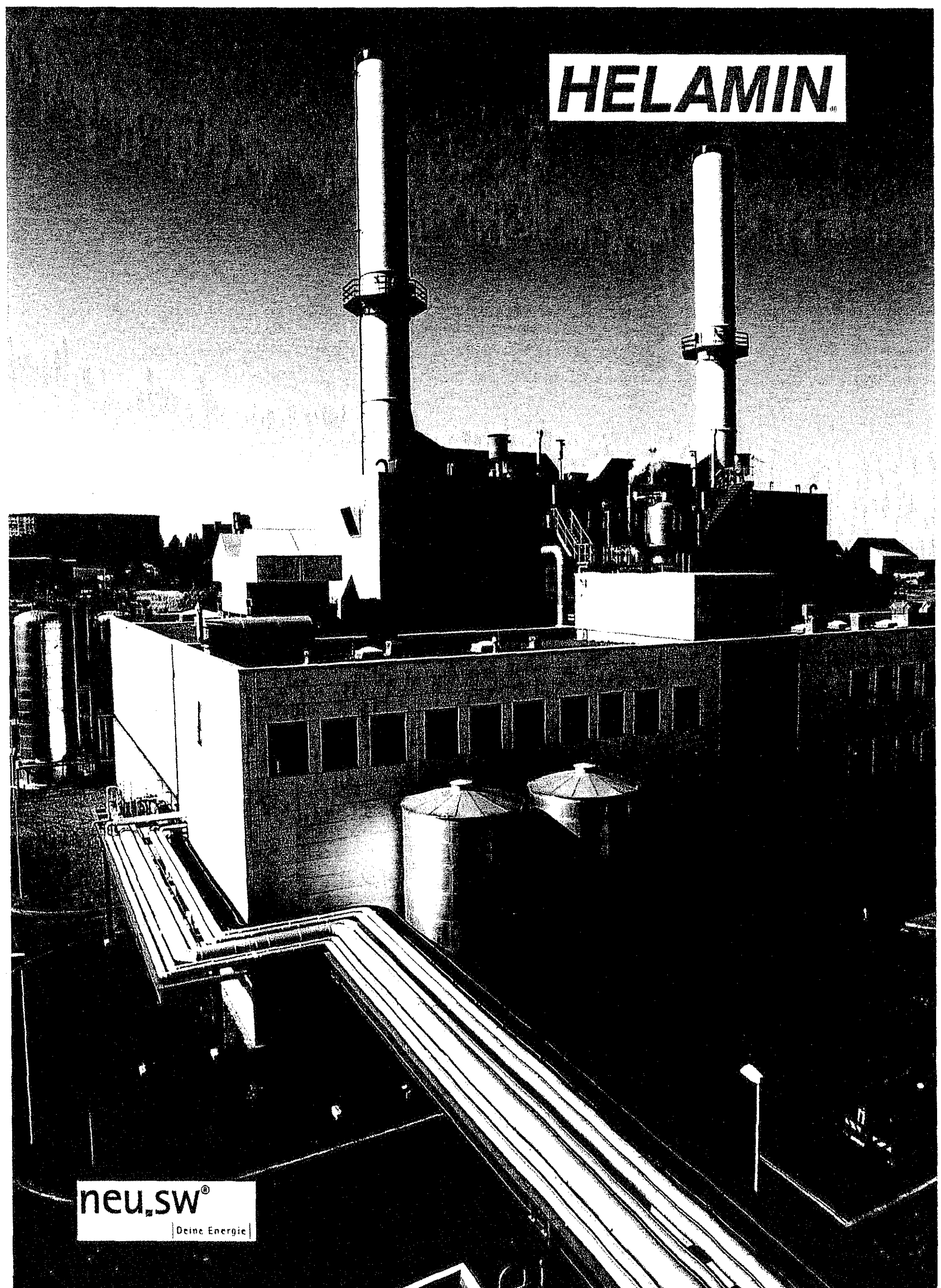
The recommended remedial measures were implemented immediately. The radial bearing clearance of the pinion shaft was reduced. After realisation, vibration measurements were performed by the manufacturer. The bearing housing vibrations were reduced to levels of maximum 1.8 mm/s eff (previously 7.1 mm/s eff). In addition, no instabilities whatsoever could be detected on the fast-running shaft. On the pinion shaft and in the objectionable lower load range, radial shaft vibrations with a maximum of only 14 µm p-p (previously 95 µm p-p) were detected so that the plant was then handed over to the operator after implementation of the remedial measures, also with regard to the vibration behaviour in the "green zone". □

References

[1] *Bently, Ch., D., and Hatch, E. T.:* Fundamentals of Rotating Machinery Diagnostics. Bently Pressurized Bearing Company, Minden (USA), 2002.

[2] *Lenz, J.:* Vibrationen auf der Spur. Firmeneigene Projektbroschüre, available under: www.koetter-consulting.com

7/2004



International Journal
for Electricity and Heat Generation

Publication of
VGB PowerTech e.V.
www.vgb.org

International Edition