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ROTOR INSTABILITY DUE TO A GEAR COUPLING CONNECTED TO A BEARINGLESS SUN WHEEL OF A PLANETARY GEAR*

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A 21 MW electric power generating unit comprises a gas turbine, a planetary gear, and a generator connected together by gear couplings. For simplicity of the design and high performance the pinion of the gear has no bearing. It is centered by the planet wheels only. The original design showed a strong instability and a natural frequency increasing with the load between 2 and 6.5 MW. In this operating range the natural frequency was below the operating speed of the gas turbine, n_{PT} = 7729 RPM. By shortening the pinion shaft and reduction of its moment of inertia the unstable natural frequency was shifted well above the operating speed. With that measure the unit now operates with stability in the entire load range.

INTRODUCTION

Any new machine design is characterized by the manufacturing cost and, for thermal machines, also by the overall efficiency. Therefore, the machines become more compact, with a tendency to smaller shaft size for the same power than in the past and a reduced number of machine elements.

These rules were considered in the design of an electric generating unit comprising a gas turbine, a planetary reduction gear and a generator (Fig. 1). The shafts are connected by gear couplings. The rated power of the set is 21 MW, the operating speed of the gas turbine n_{PT} = 7729.4 RPM, and of the generator n_{GEN} = 1800 RPM. The planetary gear has a gear ratio of 1:4.294. For simplicity of the design (fewer parts) and improved efficiency (fewer bearings) the sunwheel of the gear is centered by four planet wheels only, and not as is usual by a bearing. The pinion shaft is connected to the power turbine shaft by a gear coupling to accommodate transient and residual misalignment between both shafts (Fig. 2). The vibrations of the gear were monitored and recorded as well as the actual electrical power and the oil temperature of the axial bearing, etc.

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During the commissioning period the unit was started and run up to full power at synchronous speed. A typical recording of the generated power, the gas temperature (T7) and the vibrations of the gear are shown in Fig. 3. In all these run-ups a steep increase of the vibration level on the gear was observed at a unit load range of 2 to 6.5 MW. From 2 to 4 MW the vibrations increased gradually, from as low as 2 mm/s (RMS) to a prohibitively high level of 11 mm/s. They remained at that high level in the range 4 - 6.5 MW and "fell" to a low level at approximately 6.5 MW load. During run-down similar recordings were obtained; the vibrations of the gear were mainly load dependent (Fig. 4).

MEASUREMENT OF THE VIBRATIONS

From the overall recordings (e.g., Figs. 3 and 4) a sound analysis of the problem was not possible. However, the gear with the coupling was expected to be the source of the excitation, as the maximum vibrations occurred at the gear box. Therefore additional probes (eddy current transducers) were installed at the gear and the gas turbine in order to observe the motion of the spacer of the coupling. The relative radial and axial displacements were recorded on tape and later analyzed by a 2 channel FFT analyzer. Figs. 5, 6 and 7 were generated from the recorded start Nr. 29. Fig. 5 shows the frequency spectrum of the "peak hold" amplitudes of the displacement of the sunwheel in an axial direction, while the power was increased from 2 to 7 MW. Several peaks at the rotational frequencies and their multiples were observed. Excitations at 1800, 3600, 5400, 7200 RPM are caused by the generator, at 7730 RPM caused by the gas turbine and the sunwheel and at 4680 RPM by the planets. These excitations remain constant during the increase of the load. As the speeds of these shafts remained unchanged during operation the rest of the peaks could not be explained by excitations of rotational speeds. Fig. 6 shows the frequency spectrum of the radial displacement of the coupling. It is essentially similar to Fig. 5; dominant peaks occurred at the same frequencies as explained above. In the frequency band 5100 RPM $<$ f $<$ 7300 RPM the peaks were high but not dependent on operating speeds of the unit. In Fig. 7 the predominant frequency is given as a function of the load. In start-up Nr. 30 the radial deflection of the coupling on the gear side was analyzed at constant load. The result is plotted as frequency spectra in Fig. 8. The dominant frequency and its amplitude increased with the load of the unit from 2 to 6.4 MW. At 6.4 MW it coincided with the operating speed of the turbine and at the same point the amplitude reduced to the value of that speed.

During start-up Nr. 30 the oil temperature of the axial bearings in the gear were also recorded. Fig. 9 shows the inlet temperature and the outlet temperatures of the axial bearings on the turbine and the generator side. The temperature on the generator side remained almost constant while on the turbine side the temperature remained constant up to 6.4 MW, above this load it increased gradually with the load. The friction forces became dominant and prevented spontaneous relative movements in the coupling. However, it decreased gradually over a long period of time at full load. It seems the gear coupling does not allow for relative movements above 6.4 MW as the spacer is blocked. This fact is also demonstrated by the measurement of the axial movement of the pinion (see Fig. 9). The relaxation of the pinion took a long period of time.

From these diagrams one may conclude:

- In the load range 2 to 6.4 MW the unit shows unstable behaviour
- The natural frequency of the bending mode depends on the load of the shaft
- If the natural frequency is above the operating speed of the turbine the unit behaves in a stable manner
- The instability may be due to the indefined position of the spacer of the gear coupling. As there is no bearing for the pinion the angular deflection of the pinion is not prescribed and the spacer is free to move in both hubs.

THEORETICAL MODEL

Modelling

The measurements indicated an unstable lateral frequency on the pinion side of the gear. From the strong dependence on the load of such a natural frequency on the high vibrations of the gear a nonlinear spring stiffness in the gear was expected. Two different models were set-up:

a) Model 1 is a single degree-of-freedom system shown in Fig. 10. The spring couple consists of the teeth of the pinion and planets. The stiffness of the gear depends on the Hertzian stress which is highly nonlinear, so that the stiffness increases with increasing load. The mass may be given by the pinion and the spacer so that

$$m = \frac{J_p}{r^2} + \alpha m_s$$

J_p = moment of inertia of the pinion
r = distance between center of spring couple and the center of the mass
m_s = mass of the spacer
α = coupling factor ≈ 0.5

Then the natural frequency may be estimated by

$$f_e = \frac{1}{2\pi}\sqrt{\frac{2a^2 k}{r^2 m}}$$

k = spring stiffness (constant at constant load)
a = half the length of the pinion gear

The natural frequencies are known by the measurement and the calculation. The spring stiffness is composed of the stiffnesses of, (a) gear contact between sunwheel-planets, (b) oil film between sunwheel-planets, (c) oil film between planet and planet shafts, (d) planet shaft, and (4) planet shaft bearings. The resultant flexibility (reciprocal value of stiffness) is given in Fig. 11.

b) Model 2 is a finite element model comprising the complete rotor of the gas turbine, gear coupling and pinion shaft. The following assumptions were made:

- the elastic rotor is undamped
- the radial bearings are damped
- the spacer of the gear coupling is hinged in its hubs. These assumptions may be valid at very low load, when the static friction force in the coupling is smaller than the restoring force in the hub.
- the pinion is fixed by a spring couple. The magnitude of the springs is estimated by model 1.
- the rotor behaves linearly at a set load and speed

The calculation of the natural frequencies and the mode shapes were calculated by the computer program MADYN designed for rotor dynamics.

RESULTS

The calculation of the natural frequencies and mode shapes were executed for loads P = 2, 4 and 6 MW. The natural frequencies are tabulated in Table 1. Most of them are hardly influenced by the stiffness of the gear. However, the natural frequency which is sensitive to changes at the gear compares well with the measured ones. The mode shapes of that frequency show maximum amplitudes at the coupling hub of pinion-coupling (Fig. 12). For completeness the reduced model 1 was also compared with the measurements taking into consideration the natural frequency and the spring stiffness of the spring couple. The effective mass may be estimated by

$$m = \frac{2a^2k}{(2\pi f_e)^2 r^2}$$

The values of the effective mass for different loads are also tabulated in Table 1. For each load case this mass remains almost constant.

DISCUSSION OF THE RESULTS

The measured and the calculated natural frequencies and mode shapes compare very well. The assumptions made are confirmed, such as

- the lateral natural frequency depends on the stiffness of the meshing gears and

- the connection spacer-pinion is hinged

Both of the models may be used for any further discussions and improvements of the design. The calculation revealed several natural frequencies below the rotating frequency of the turbine. However, apparently only one mode behaves unstably. The unstable mode shape shows a maximum deflection at the hub between pinion and spacer.

IMPROVEMENTS OF THE SET IN SITU

Method

The high vibrations measured at the gear box and the coupling were due to an instability in the rotor system. The unstable mode shape was excited subsynchronously in the gear coupling. In order to improve the system the natural frequency of that mode had to be shifted above the rotating frequency of the gas turbine, f_{PT} = 128.8 Hz. The operating load of the generating unit ranges from 2 to 21 MW. Therefore, the measured frequency at 2 MW, f_e = 86 Hz, had to be lifted above 129 Hz. Several measures may be considered in a new design, e.g., at the gear box, a bearing for the pinion, which fixes the pinion and suppresses rotations around any axis perpendicular to the axis of the shaft, or a reduced distance between the gear box and gas turbine. However, such remedies were out of the question. To discuss other measures one may observe model 1 which estimates the unstable natural frequency

$$f_e = \frac{1}{2\pi} \frac{a}{r} \sqrt{\frac{2k}{m}}$$

This formula indicates the effect of the main parameters on the frequency. An increase of a and k and a reduction of r and m will increase the natural frequency. Changes on the gears, pinion and planets, would have necessitated a new gear box, thus a change of a and k was out of the question; therefore r and m had to be reduced substantially. This measure was then further investigated. Finally m was reduced by 2.5 % to m = 86.5 kg and r by 40 % to 179.5 mm.

The new design is shown in Fig. 13. A detailed calculation (model 2) estimates the natural frequency f = 146 Hz at 2 MW which is well above the operating speed of the rotor (see Table 1). Fig. 14 shows the sensitive mode shape of the improved system. The maximum displacement is still at the gear coupling.

RESULT OF THE IMPROVEMENT

The pinion of the gear was changed in situ based on the drawing shown in Fig. 13. In all of the later start-ups and run-downs the vibrations on the gear and the bearings of the gas turbine remained well below the limits (Fig. 15). The frequencies of excitation were all rotating speeds or their harmonics and constant in the entire load range of the unit from 2 MW to 21 MW. The unit now has stable behaviour. The unstable natural frequency is above the operating speed of the turbine.

CONTI-BARBARAN-NUMBER

Gear couplings are inherently unstable at low loads (Ref. 1) This is due to the radial clearances between the teeth. At low load the spacer is free to move as the contact (circumferential) force is smaller than the inertia (centrifugal) force. An indicator of the behaviour of the coupling is given by the Conti-Barbara-Number E (or Newtons number).

$$E \sim \frac{F_u}{F_c}$$

$$E = \frac{1.36\ P}{M\ D^2\ N^3}$$

where P = transmitted power in MW
M = mass of the spacer in kg = 105 kg
D = pitch circle diameter in m = 0.28 m
N = operating speed in RPM = 7729 RPM
F_u = circumferential force in N
F_c = centrifugal force in N

Experience shows if

$E > 10$	no problems may be expected
$5 < E < 10$	problems may be expected
$E < 5$	problems should be expected

However, these limits are very rough and do neither consider design and manufacturing quality of the teeth nor the alignment of the units. For the gear coupling of the above unit

$$E_o = .358\ P$$

applies for the original design and

$$E_p = .417\ P$$

applies for the improved design.

In the entire load range 2 to 21 MW, E is < 10 at, e.g. 2 MW E_o = 0.7 (E_p = 0.8). The instability occurs only if there exists a feedback mechanism as in the original design, which was the instable natural frequency below the operating speed of the turbine. The mode shape of that natural frequency has its maximum displacement at the gear coupling and therefore is easily excited by the gear coupling.

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Table 1: Calculated and measured natural frequencies (model 2) and the estimated spring stiffness and effective mass (model 1)

Number of Mode	Natural Frequency at Load P 2 MW	4 MW	6 MW	2 MW**
1	34.4 Hz	34 Hz	34 Hz	34 Hz
2	50 Hz	50 Hz	50 Hz	50 Hz
3	73 Hz	73 Hz	73 Hz	74 Hz
4	87* Hz	93 Hz	94 Hz	141 Hz
5	94 Hz	112 Hz*	119 Hz*	146 Hz**
6	104 Hz	139 Hz	136 Hz	160 Hz
Measured Dominant Frequency	86.2	112.5	120	
Spring Stiffness of the Couple N/m	$2.068 \cdot 10^8$	$3.47 \cdot 10^8$	$3.967 \cdot 10^8$	
Effective Mass	86.5	88.4	89.0	

* Natural frequency with a large deflection in the gear coupling

** Lowest natural frequency with large deflection in the gear coupling

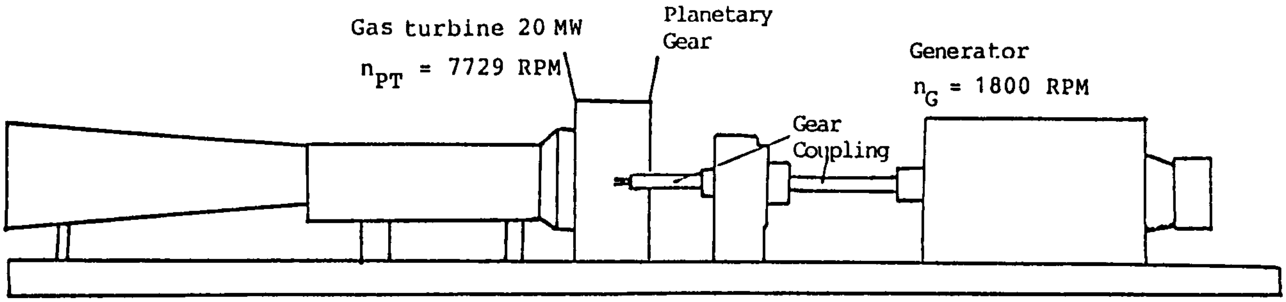

Fig. 1: Schematic of the 21 MW Electric Power Generating Unit.

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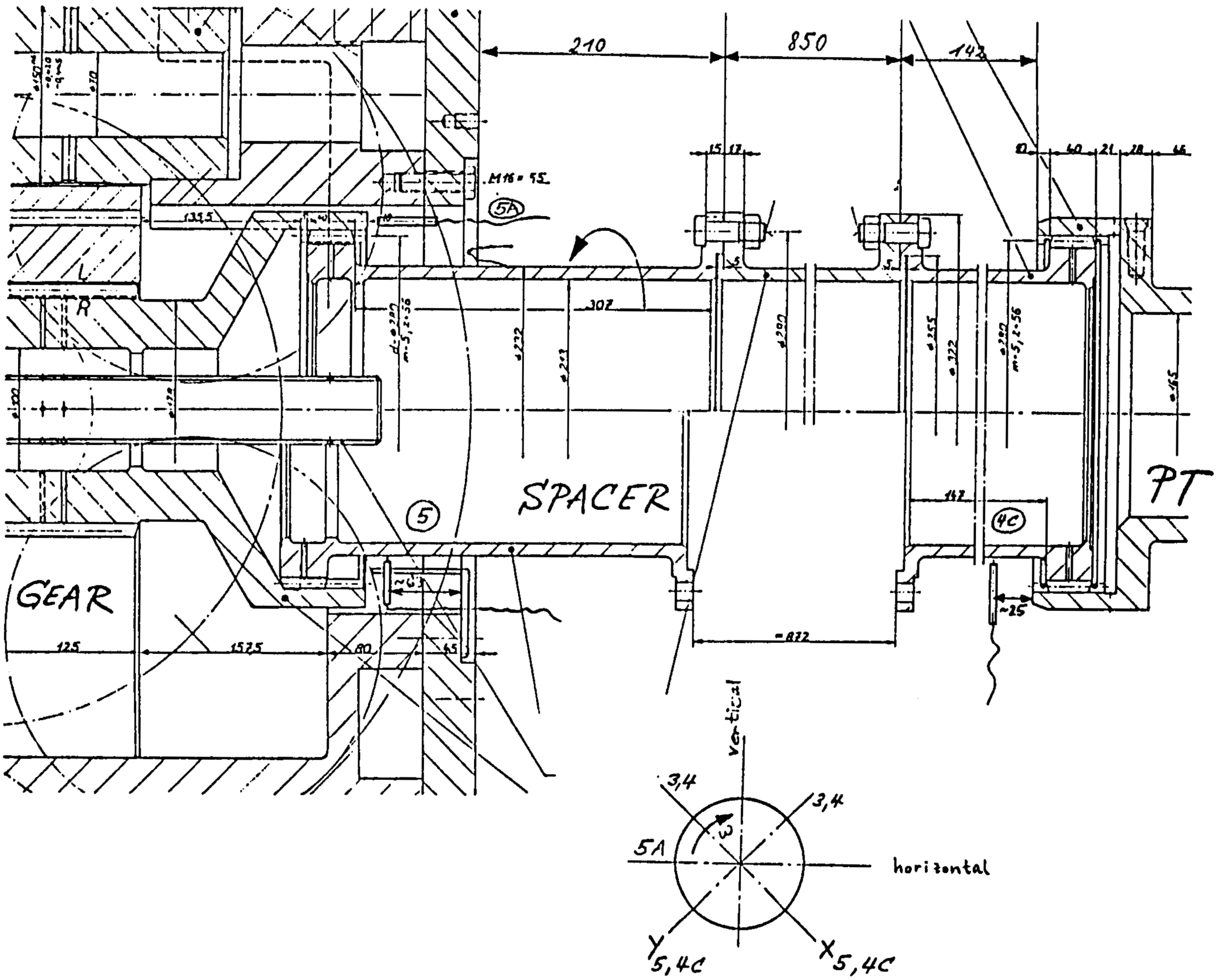

Fig. 2: Drawing of the Pinion and Gear Coupling and the Locations of the Additional Probes.

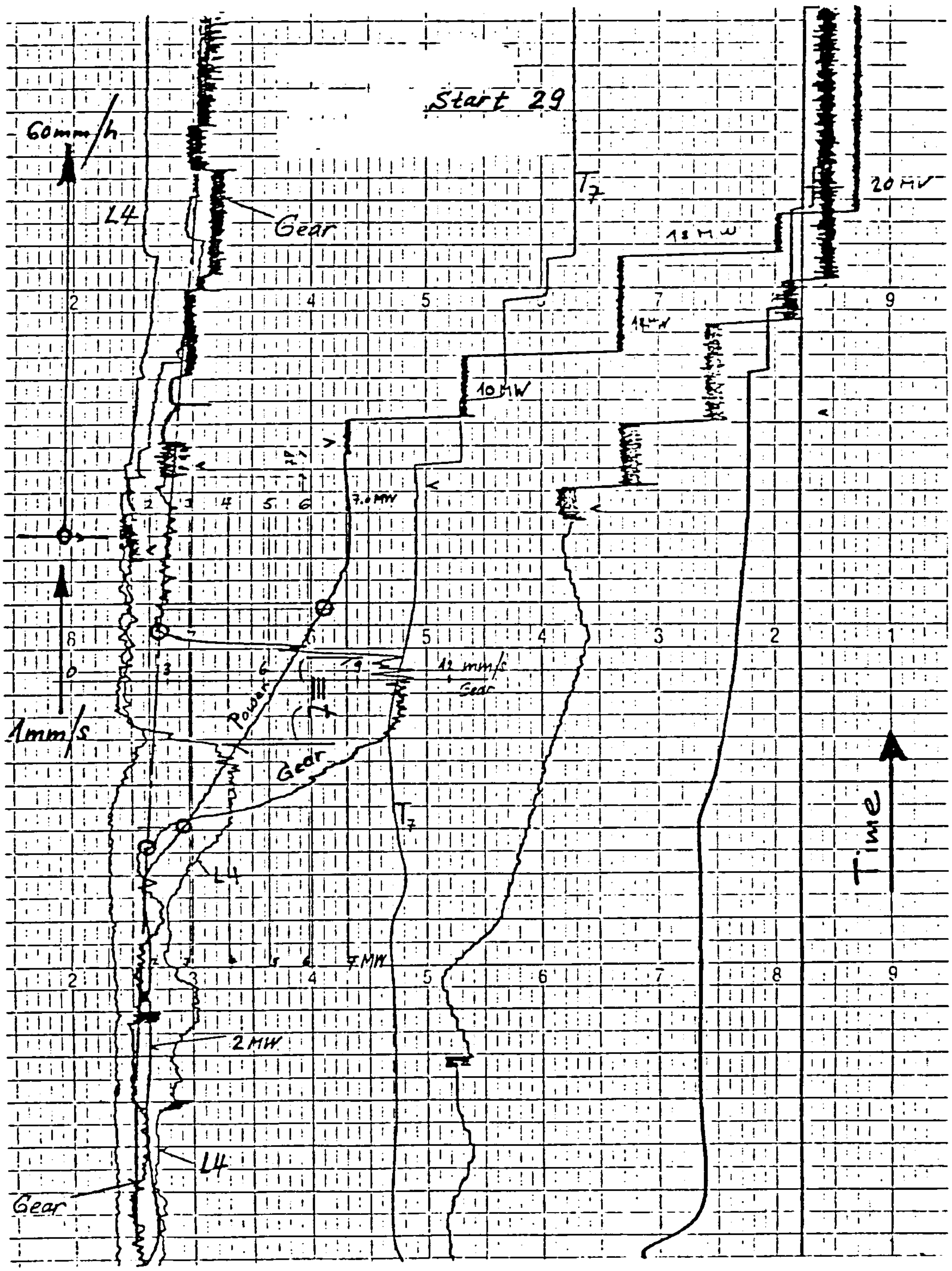

Fig. 3: Record of Start-up Nr. 29.

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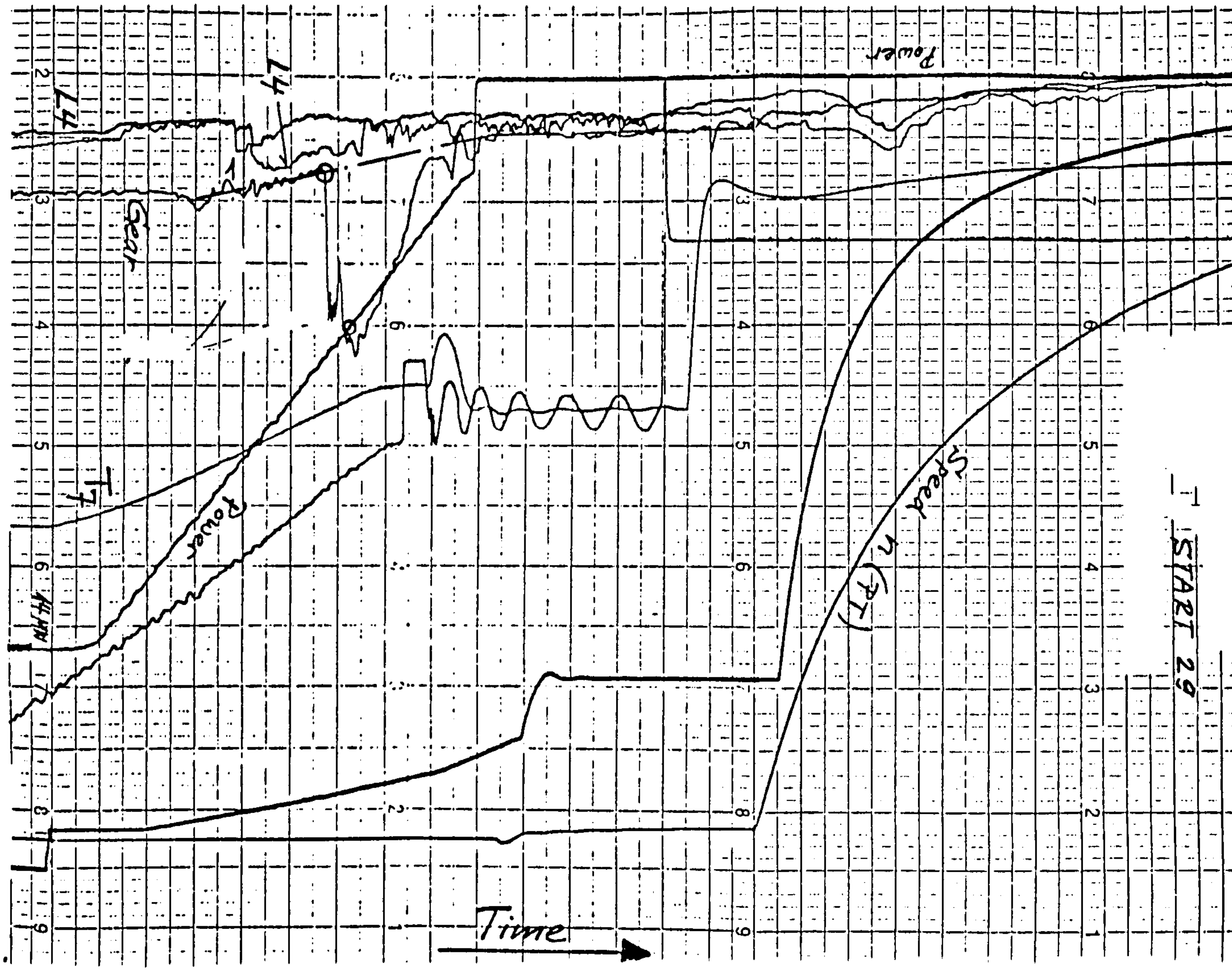

Fig. 4: Record of the Run-down Nr. 29.

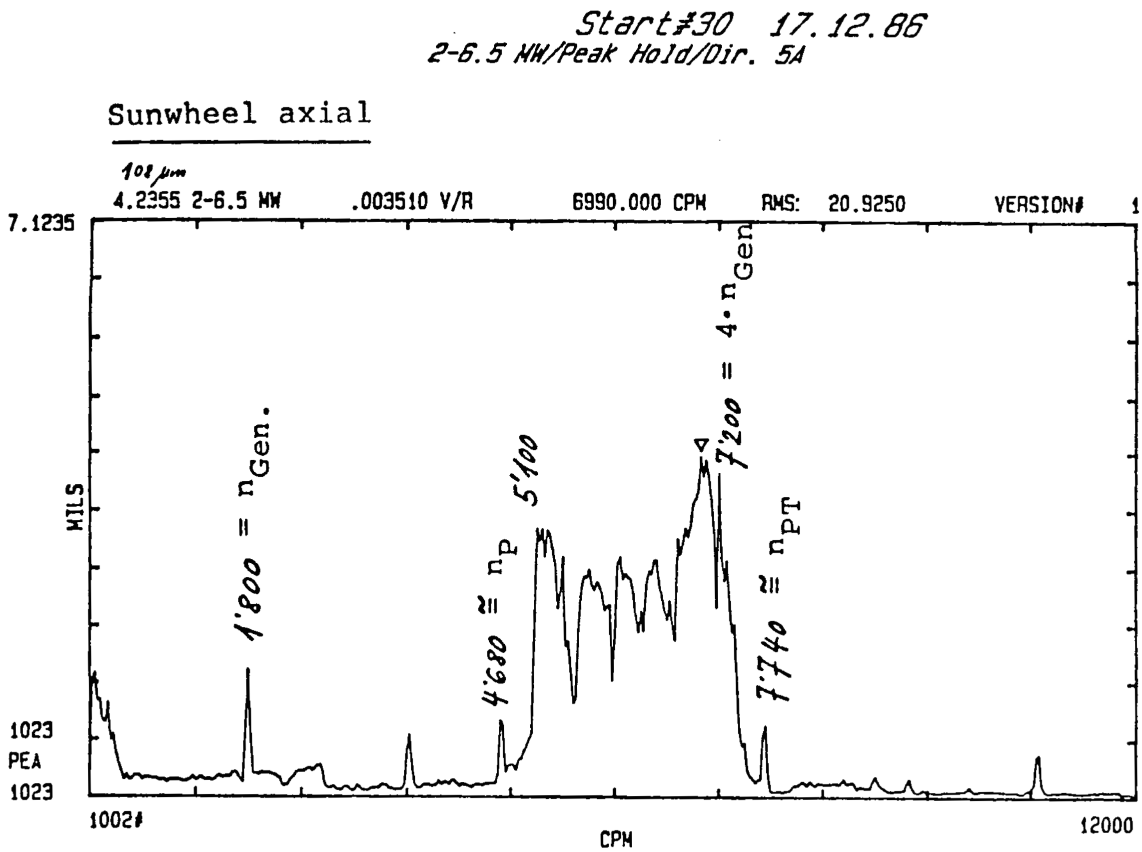

Fig. 5: Frequency Spectrum of "Peak Hold" Axial Displacement of the Pinion; Start-up Nr. 30; 2 ÷ 6.5 MW.

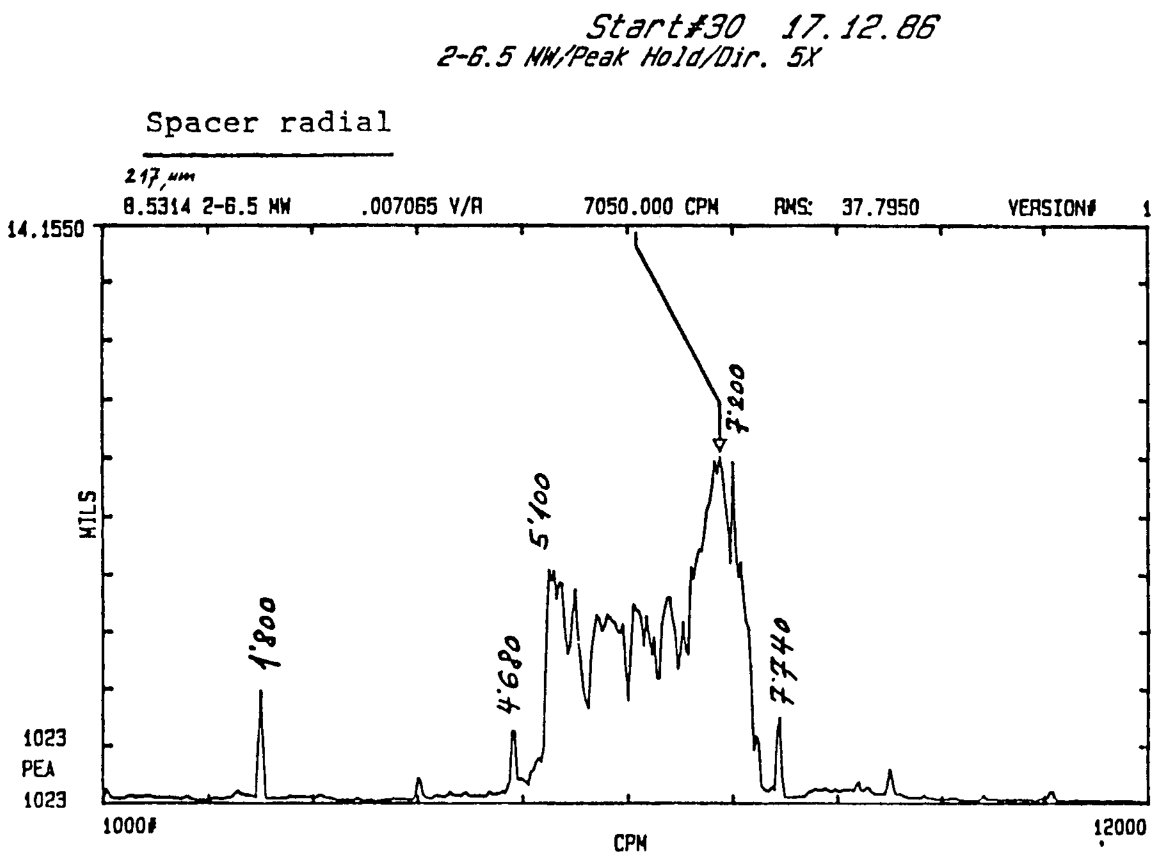

Fig. 6: Frequency Spectrum of "Peak Hold" Radial Displacement of the Coupling Spacer; Start-up Nr. 30; 2 ÷ 6.5 MW.

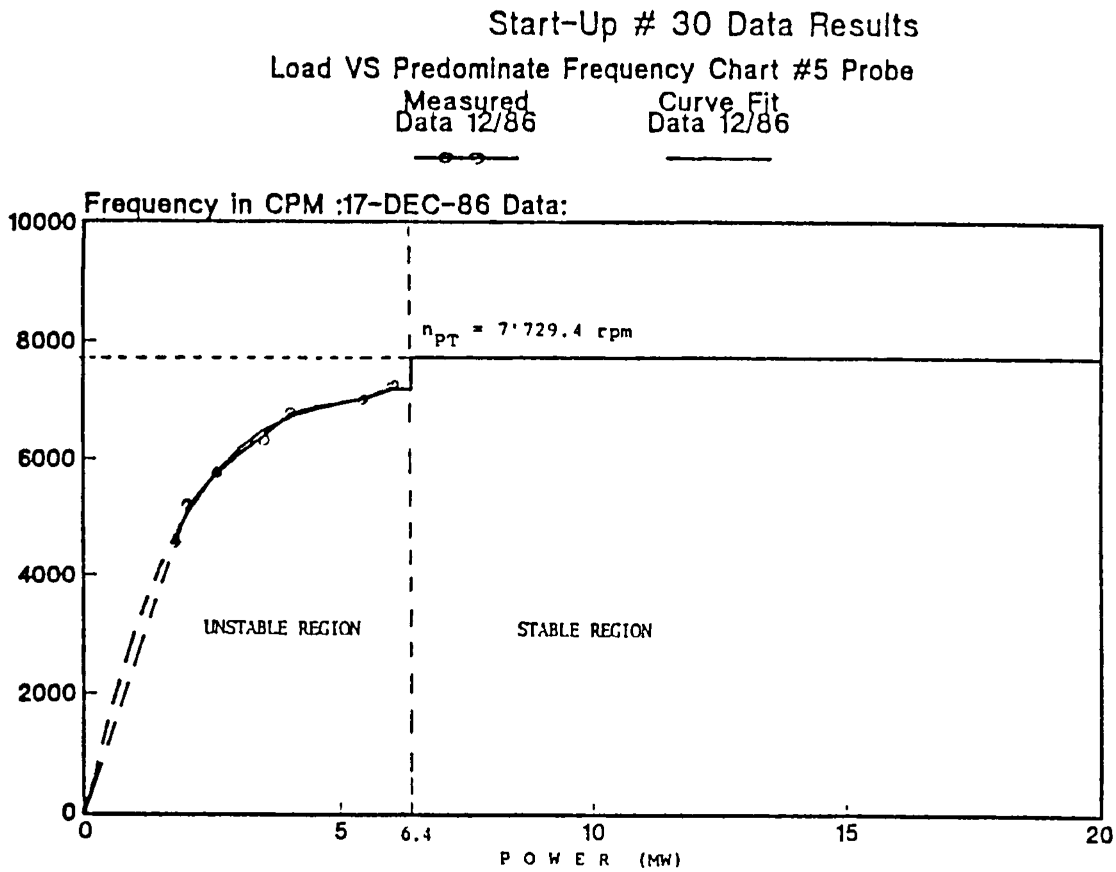

Fig. 7: Dominant Frequency as a Function of Load (Power)

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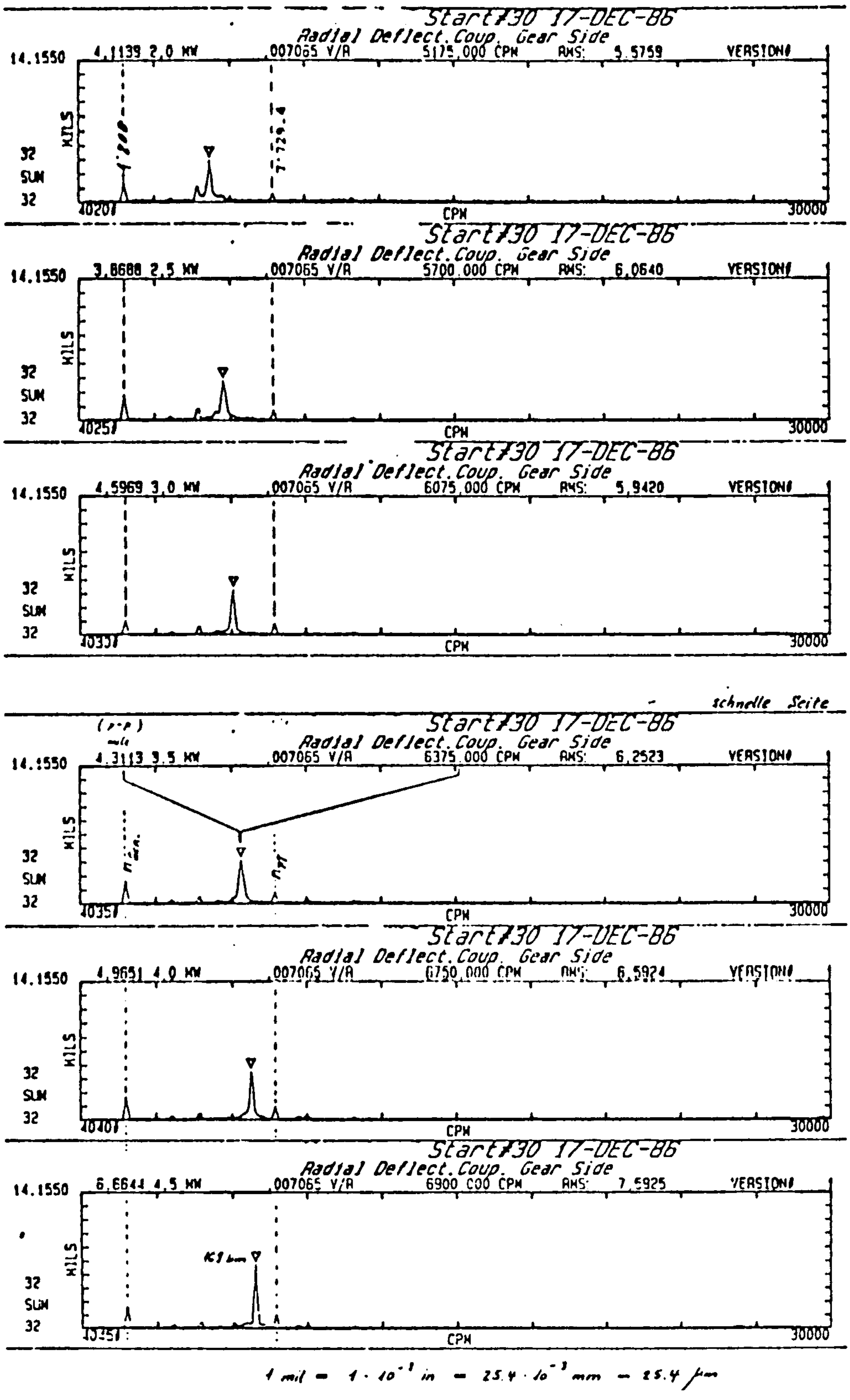


Fig. 8: Frequency Spectra of the Radial Displacement of the Coupling at Constant Speed and Load (Start Nr. 30)

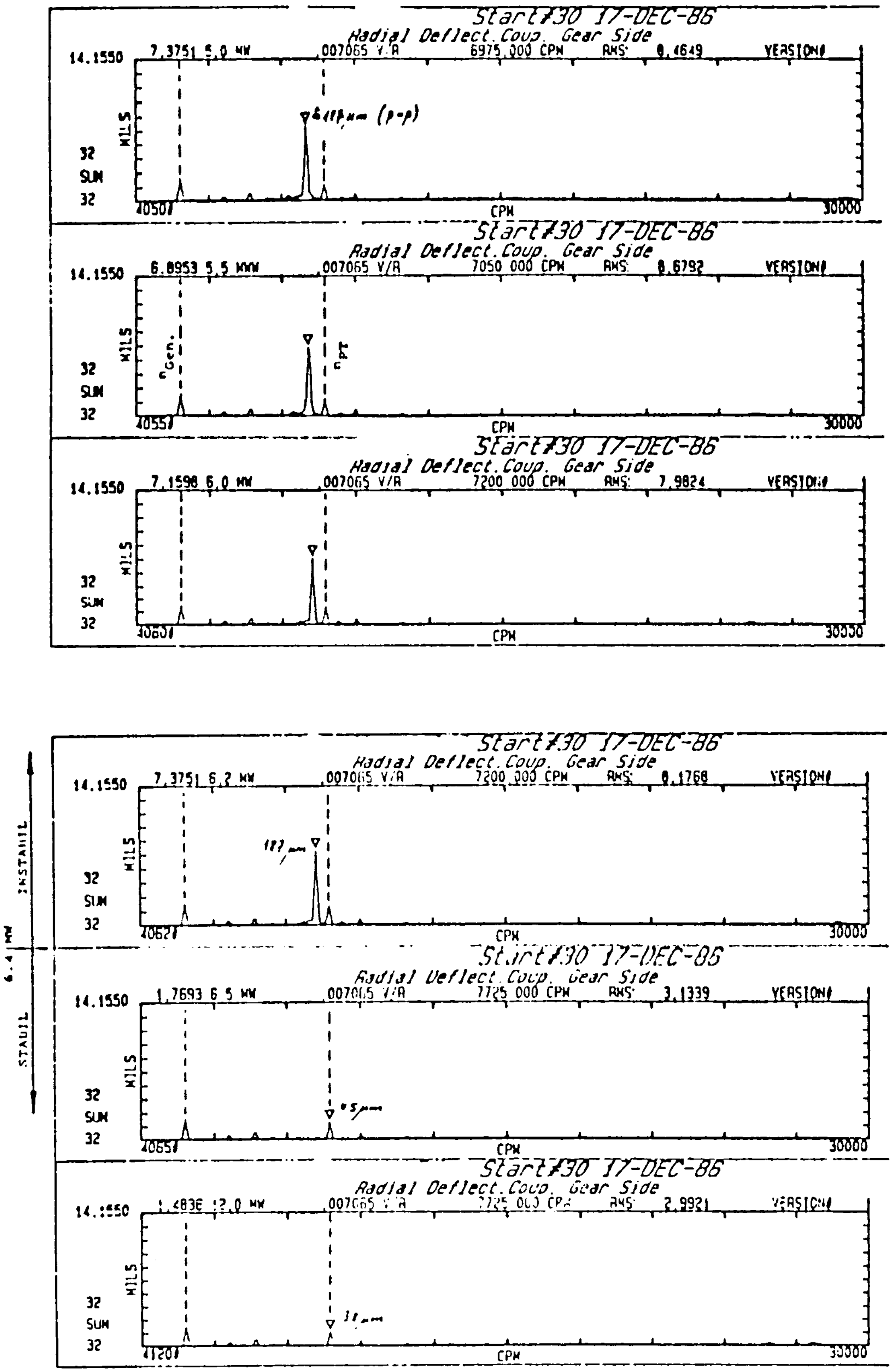


Fig. 8: Continued

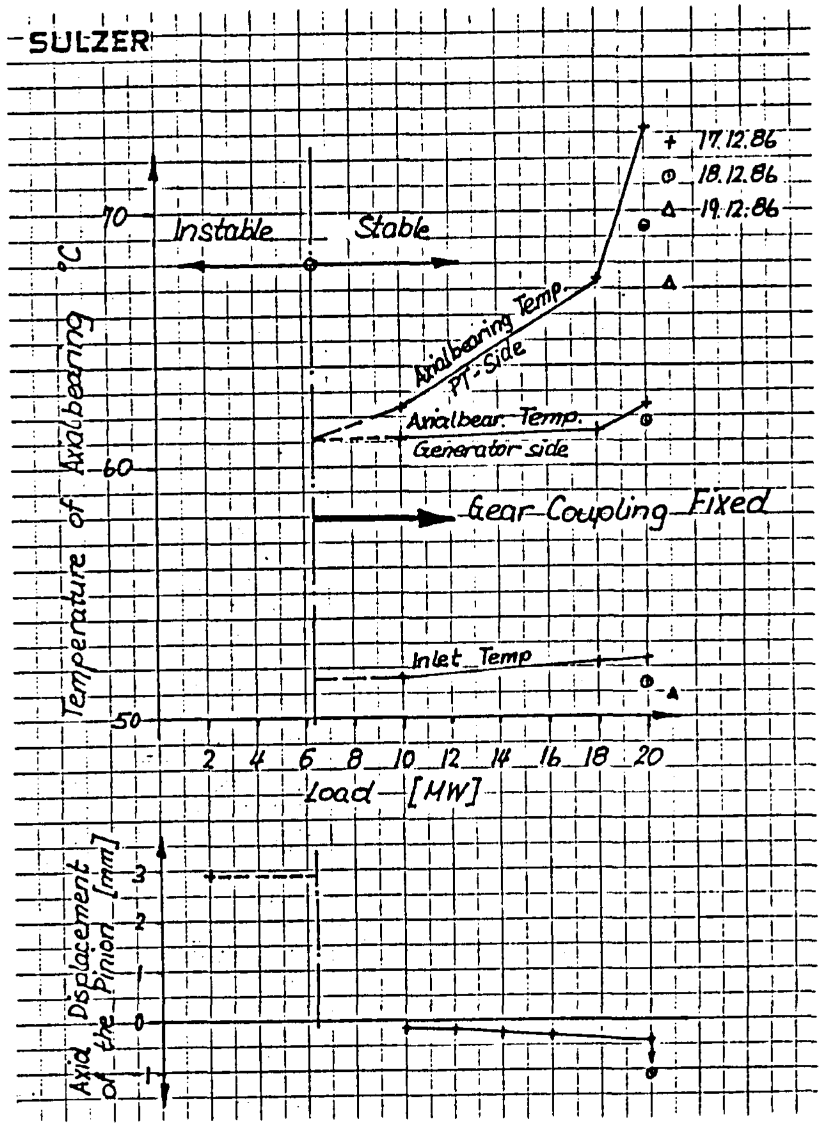
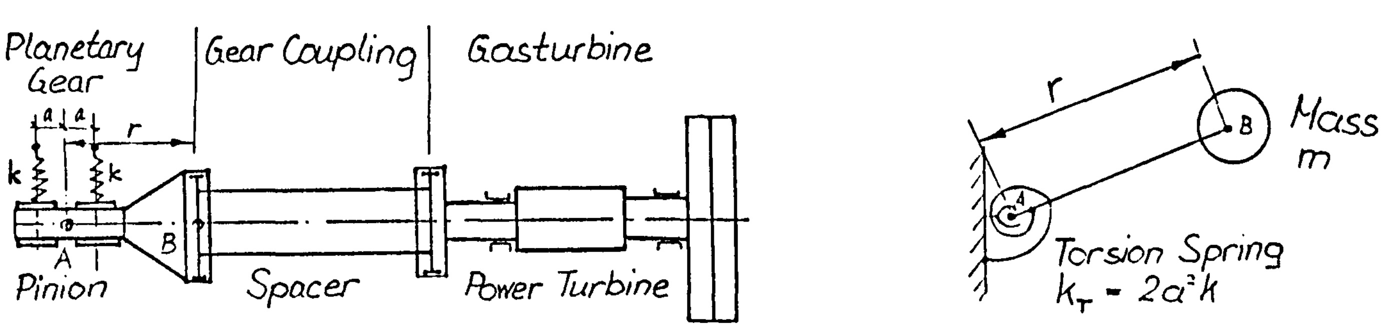


Fig. 9: Axial Bearing Temperature as a Function of Load and Time



a) Physical Model

b) Mathematical Model 1

Fig. 10.: Physical and Mathematical Models

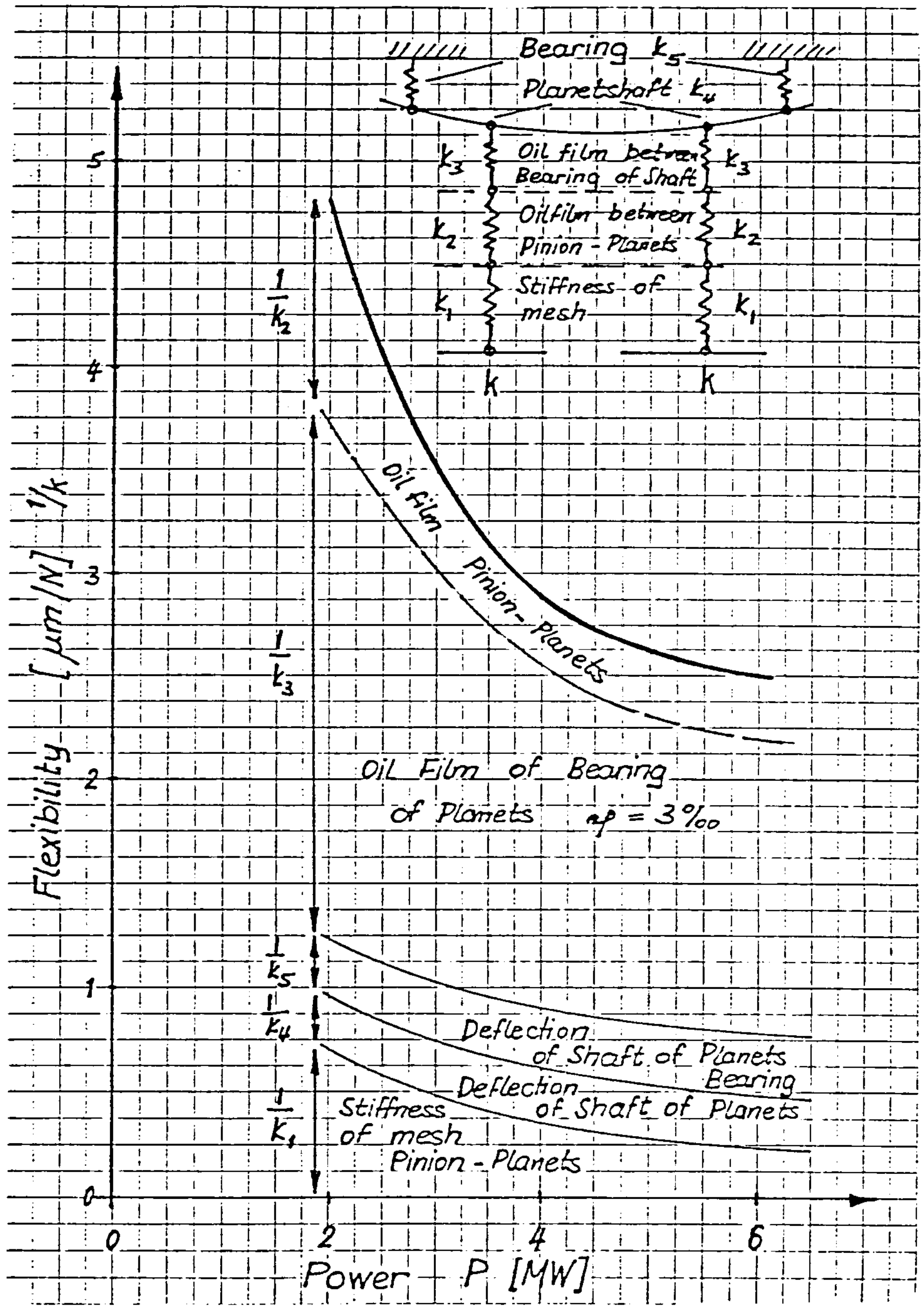


Fig. 11: Combined Spring Stiffness of the Pinion-Planet-Mesh

- Mathematical Model
- Dependence on the Load

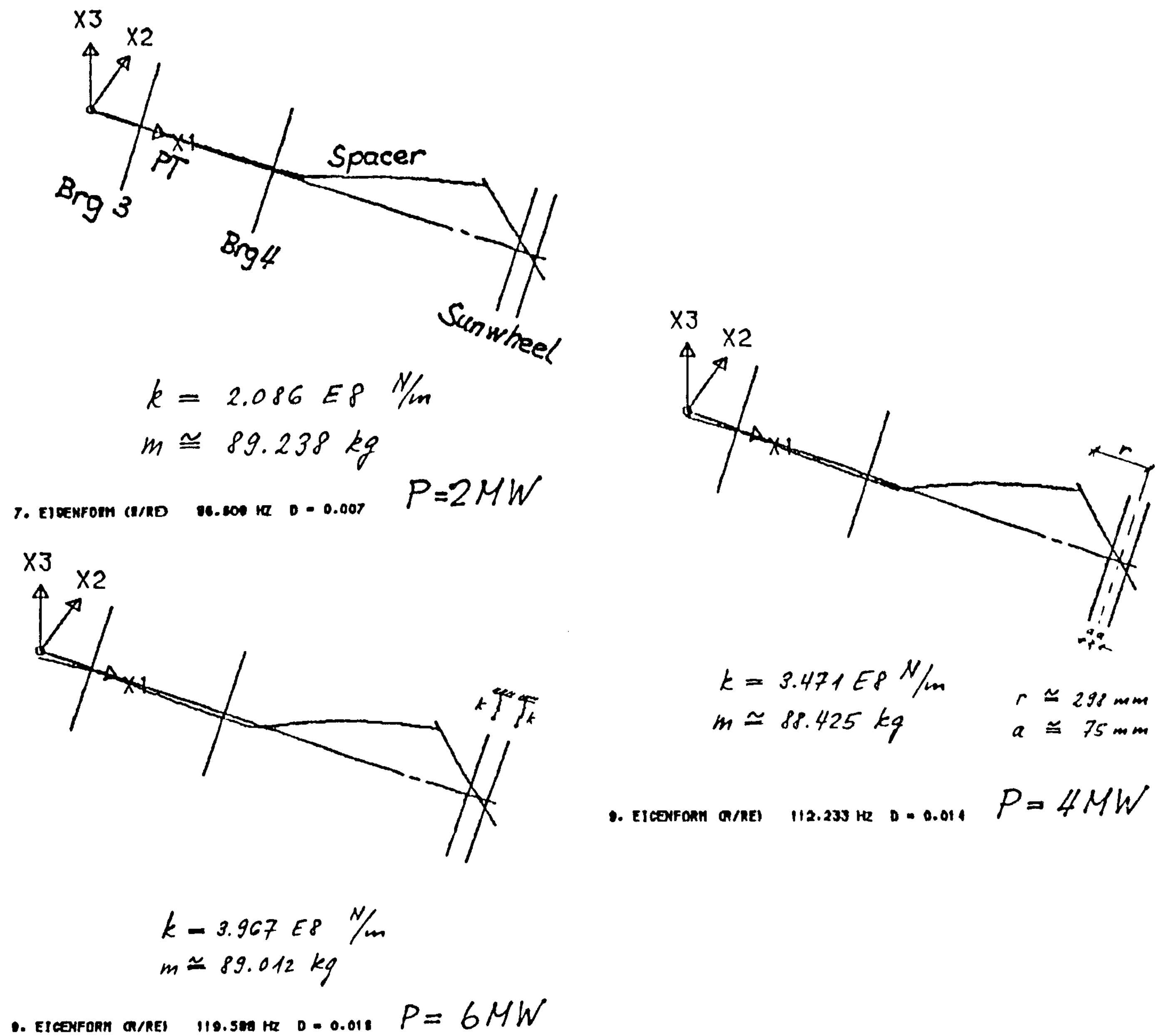


Fig. 12: Modeshape of the Unstable Natural Frequency having its Maximum Displacement at the Pinion-Spacer Hub for P = 2, 4 and 6 MW.

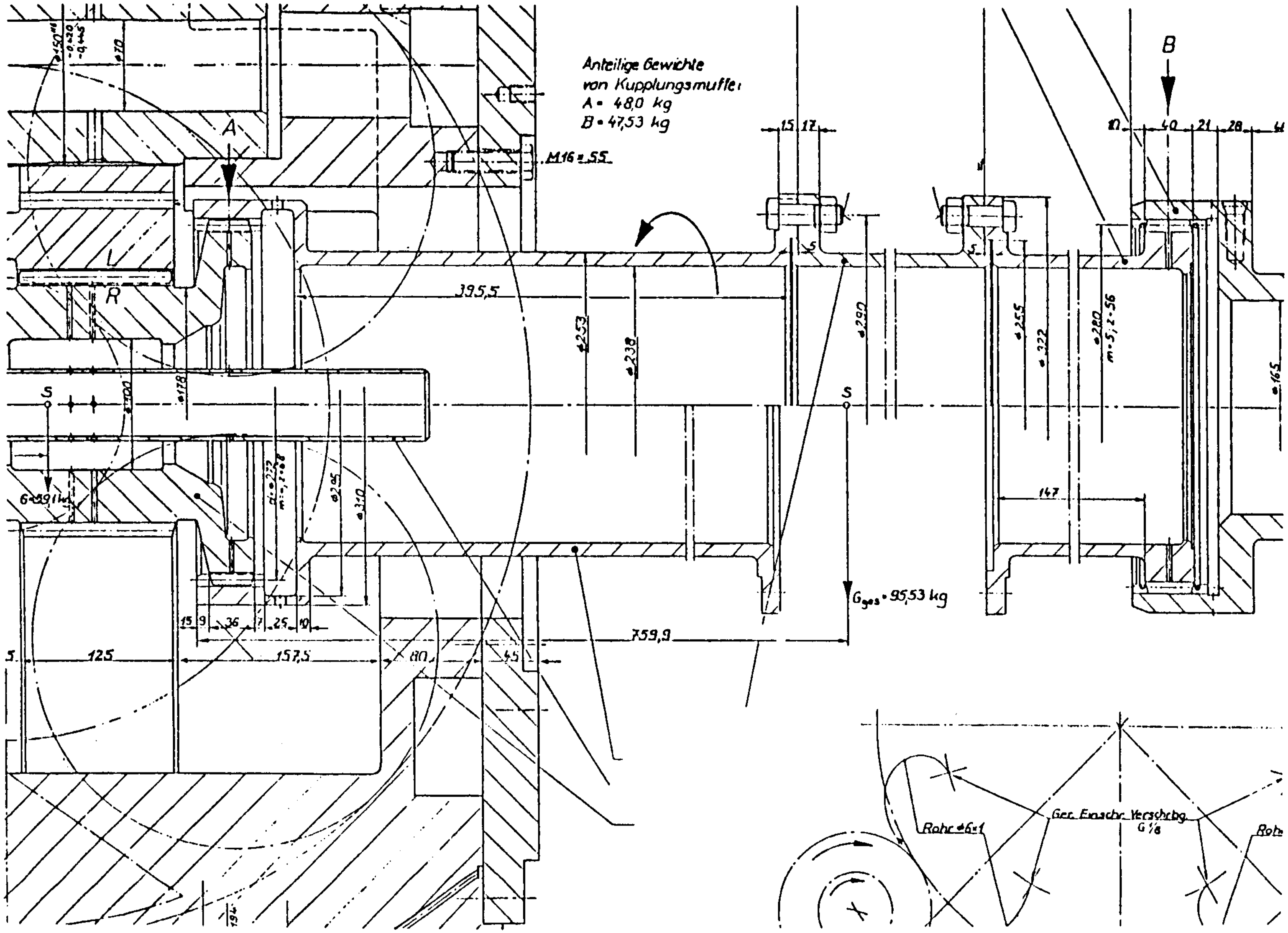


Fig. 13: Improved and Final Design of Pinion and Gear Coupling with Reduced Length of Pinion Shaft

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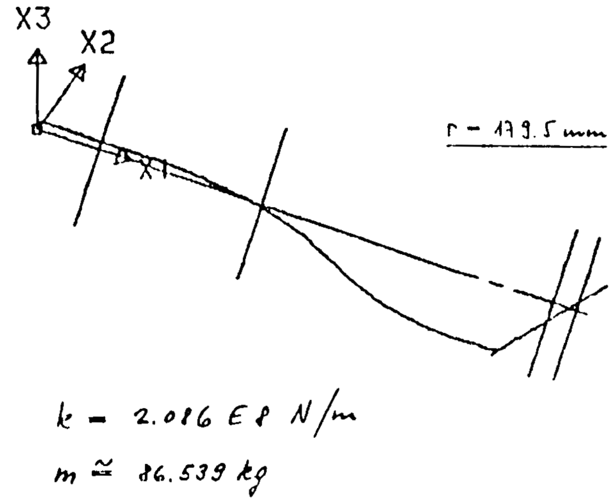

11. EIGENFORM (R/RE) 148.010 Hz D = 0.044

Fig. 14: Mode Shape of the Improved Design for Load P = 2 MW having its Natural Frequency at f_e = 146 Hz, and well above the Speed of Operation f_o = 128.8 Hz

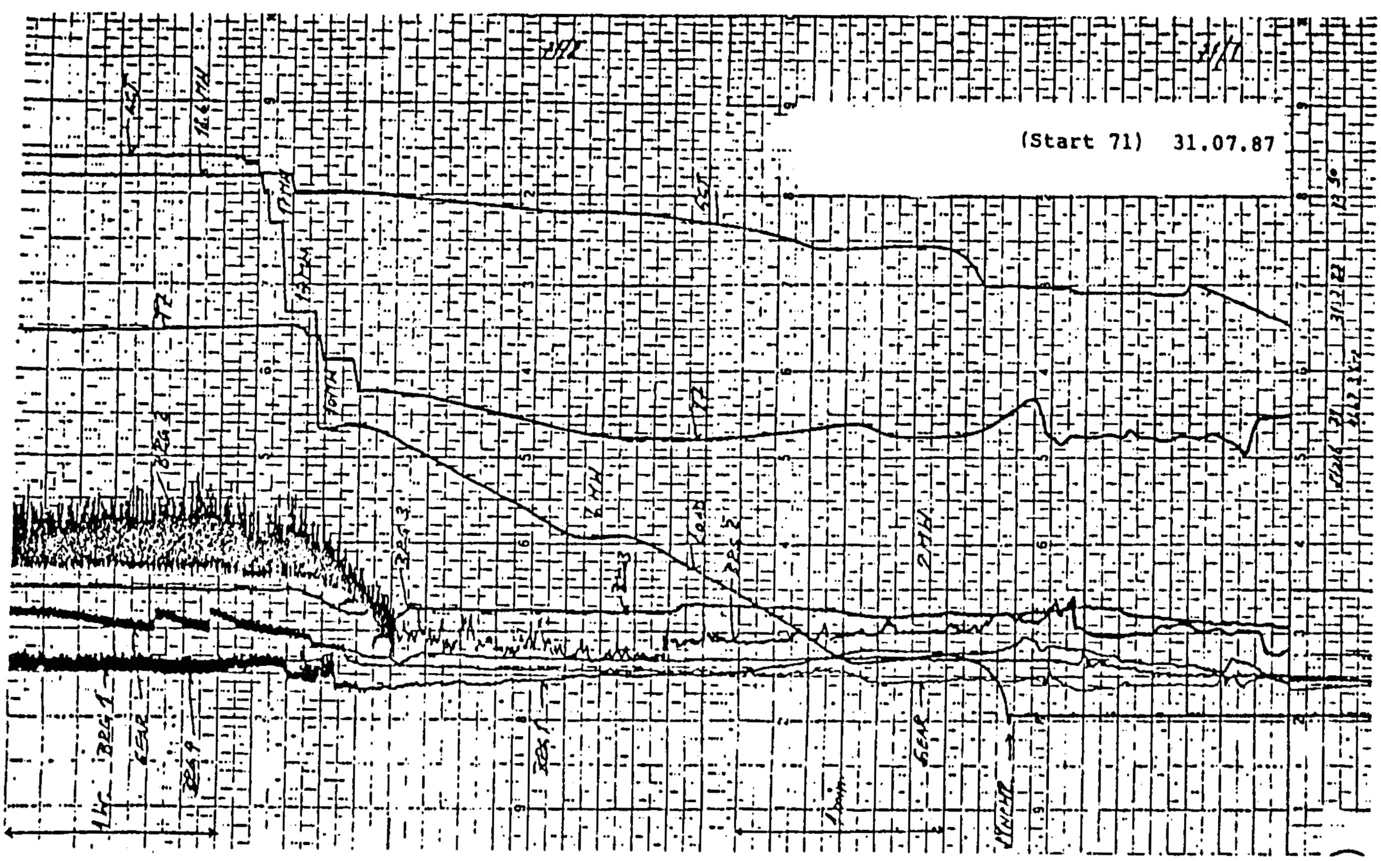

Fig. 15: Record of Start-up Nr. 71 with the Improved Design