

SILENT SHIP PROPULSION GEAR SYSTEMS

by

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1. INTRODUCTION

Modern propulsion machinery for Naval crafts must be highly reliable and must have low structure - and airborne noise emission.

MAAG looks back on 50 years of experience with surface hardened and ground gearing for marine applications (see ref. list).

Our reputation is high safety and continuous reliability over the whole time of operation. The know-how in the design and calculation of these propulsion gears is well established and appreciated by many Navies all over the world. Modern gear calculating methods, as well as a manufacturing equipment of highest accuracy, are being used.

The prediction of transduced gear noise is not an easy task. However, MAAG has gathered much experience on their own from test bed results and sea trials, but also in close collaboration with a number of reputable Navies. We are working continuously on some most important projects where the design and manufacturing of extremely silent gear units is demanded.

The following chapters discuss the major factors which are considered having an influence on the noise level.

2. DESIGN PRINCIPLES

2.1) GEAR CONCEPT

Gear meshing noise is dependent on the specific tooth load (N/mm).

The size of a propulsion gear system must cope with the specially adapted engine room lay-out requirements. Lowest specific tooth loads can only be achieved by a gear concept with split power train.

Fig. 1 shows the principle of a double reduction gear of this kind.

The input power is divided into two primary gears which drive two secondary pinions via quill shafts acc. to Fig. 1a). Thus, the power is transmitted to the bull gear by two tooth meshes. (instead of only one as in a standard double reduction gear).

This type of this gear concept, known as the "locked train" configuration, is accepted by all modern Navies not only as the most reliable solution but also as that with the lowest vibration levels.

These design principles are also obtained with the cross-connect system where the GT power is split into two gear trains, see Fig 1b). A split on the diesel side is not necessary as the power is smaller and the specific tooth load accordingly lower.

As an additional main feature the location of all gear meshes related to each other are to be calculated properly.

2.2) GEAR TOOTH GEOMETRY

For low gear meshing noise excitation the tooth geometry must be designed with proper

- a) Contact ratio,
- b) Overlap ratio,
- c) Tooth corrections,
- d) Torsional vibration analysis.

- a) MAAG applies the so called high tooth form with optimized contact-ratio values (see Fig. 2.1)
- b) With surface hardened split power train gears the specific tooth loads are relatively low. Thus, the module is able to be minimized accordingly. Therefore, the overlap-ratio for a given helix angle becomes large and can be adjusted to values due to lowest noise excitations (see Fig. 2.2).
- c) Under load the pinions and gears deflect elastically. These deflections are compensated by tooth profile and lead corrections. The corrections are carefully designed for the power where lowest noise emission is required. Thus, the effect of the specific tooth load on noise (see section 1) is compensated within a certain speed range (see Fig. 2.3).
- d) By means of a torsional vibration analysis the masses and stiffnesses of rotating parts are optimized due to natural frequencies. Thus, an unfavourable interference with gear mesh frequencies can be avoided. This aspect also is considered as a main design principle for the calculation of the basic tooth geometry datas of all gear meshes.

In general, according to our extended experience in manufacturing of silent marine gears, the lowest noise excitation only is able to be realized with single helical gears. Double helical gears are not suitable with this respect, as in principal the tooth accuracy of one gear stage concerning noise influencing deviation values is lower. Additionally, axial vibrations generated by double helical gears increase the noise level.

2.3) ACCURACY IN MANUFACTURING

For low noise levels at all frequencies all rotating gear elements must be manufactured with the highest possible accuracy.

Special attention must be given to:

- a) The toothings,
- b) The journals,
- c) The dynamic balancing

- a) All gear teeth are ground on specially equipped grinding machines installed in air conditioned halls to an accuracy which corresponds at least to the quality ISO/DIN 2 .

In order to ensure and to maintain this high grade of accuracy, it is necessary to check all tooth errors continuously. By means of the highly developed MAAG measuring equipment, the single and accumulated pitch deviations, the profile, lead and undulation deviations are measured and recorded (see Fig. 3).

The rotational harmonic frequencies and their even numbers significantly appear dependent on the accuracy of the gear teeth, s. Fig. 4.

- b) The journal accuracies influence a silent operation as well. The journal surfaces are ground to a high degree of roundness and diameter accuracy.

In order to obtain a proper tooth contact at any load the two journals of one gear shaft are manufactured with equal diameters and selected to each other to a very tight tolerance.

- c) All rotating components and subassemblies are dynamically balanced. Different sizes of calibrated balancing machines are used due to the weight of the work pieces.

2.4) RELIABILITY

Silent operation of the gears must last for the life time of the ship.

This requirement can only be achieved if wear is reduced only to tooth surface polishing during the running-in period. Experience over decades has shown that properly designed marine gears with hardened and ground teeth do not change shape once the surface roughness of the flank is minimized. Often the bull gear is made of a high tensile steel, meshing with carburized and hardened pinions. The tooth flanks of these pinions polish much faster than those of the through-hardened bull gear.

Experience shows that the noise level decreases with the time of operation, but only applying properly designed gears. This is proved by different noise measurements aboard.

As a further main experience, the influence of load on the noise level above a certain stage is negligible, s. Fig. 5. Thus, an optimized gear design is required especially concerning the slow and smooth low cruising

The above, of course, applies to properly maintained machinery. (e.g. Corrosion during standstill periods must be avoided!) Lube oil system must stay clean, must be checked periodically and changed when necessary.

2.5) SHAFT BALANCING

Shaft vibration are caused by imbalances of rotating parts or by dynamic instabilities of the journals in their bearings.

Shaft noise is often a dominating factor in the whole vibration spectrum. Special attention must be given to the following elements:

a) Balancing of shaft arrangements

All shafts with mounted gear train components are to be dynamically balanced. In order to ensure a high degree of accuracy, the balancing machines are selected with regard to size and weight of the work piece. With the MAAG balancing equipment, the quality stage G1 normally is achieved.

b) Bearings. By careful selection of the clearances and shape of oil grooves , instabilities are successfully avoided throughout the speed range. Lay-shafts (shafts without radial load) are supported by multi-lobe bearings.

c) Clutches. A MAAG synchronizing clutch is composed of a number of ring-shaped sleeves which must be allowed to move freely relative to each other. In order to prevent high unbalancing forces, the following design principles have been realized:

- o small clearances,
- o low weight of moavable parts,
- o high manufacturing accuracy,
- o surface hardened sliding faces.

With the design principle of synchronizing clutches fretting corrosion is impossible. This guarantees continuously low clutch vibrations and infinite durability.

2.6) GEAR CASING DESIGN

Tooth misalignment not only reduces the reliability of a gear unit but also increases noise and vibrations.

The following requirements are to be considered:

- a) The gear casing must be rigid enough as the tooth meshing conditions according to the design are maintained throughout the whole operation power range.
- b) A gear box with adequate rigidity can be hard or soft mounted on the ship foundation.

The hard mounting should come as close as possible to a "three-point" support, s. Fig. 6. This prevents superimposed loads caused by ship hull deflections.

- c) The design of the casing is influenced by the weight limits imposed by the specification. By proper design of casing walls and ribs the noise transfer down to the foot plates is able to be minimized.

2.7) SPECIAL DESIGN MEASURES

Noise radiation reduction at its very origin combined with limitation of noise transfer leads to optimized silent marine gear operation.

With the design features as described in above paragraphs normally satisfying silent operating marine gears are obtained. If special low noise requirements are to be additionally fulfilled, the following design measures have to be considered:

a) Elastic mounting of the gear box system.

Due to allowable shaft deflections under load and required natural frequency of the mounting system soft-elastic or hard-elastic resilient mounts between gear box footplate and ship foundation are applicable. In cooperation with highly competent mount manufacturers, the calculation and determination of those mounts can be provided. Fig. 6 shows an example of noise specification which would be satisfied by this means.

b) Gear box design

The connection between footplate and main walls and further parts of the gear casing are to be optimized by means of calculation and design concerning limitation of noise transfer.

c) Gear teeth manufacturing

A newly developed MAAG machine grinding process which has proved its reliability during recent years is able to be applied in order to achieve the highest accuracy of the gear teeth. The manufacturing deviations are even lower than those obtained with the well known conventional MAAG grinding process.

With these main special design measures and further minor adaptations, highly restrictive noise limit specifications are able to be fulfilled.

2.8) GEAR ASSEMBLY AND SHOP TESTING

Reliable assembly work is most important to assure proper load distribution at each gear mesh.

Every gear unit is run on the test bed within the total operating speed range up to overspeed, (as an example of a testing arrangement see photo, fig. 7). The operation mode change-overs are tested as well. After the shop test, complete records are available on the following items:

- o Gear losses,
- o Bearing temperatures,
- o Oil consumption, pressure, temperature,
- o Function of synchronizing clutches,
- o Characteristics of lube oil pumps,
- o Vibration (structure borne noise) at gear foot plate at various points and speeds,
- o Tooth load patterns,
- o Visual test after dismantling.

After this thorough testing in the shop only minor adjustments may be necessary on board ship to get ready for sea trials.

2.9) QUALITY ASSURANCE

Stringent testing and controls are necessary during design and the entire manufacturing in order to achieve a gear box which operates at the lowest possible noise level.

The MAAG Quality Assurance System comprises not only the extensive measurements as described in section 2.3 but all important parts during the entire design and manufacturing process. Fig. 8 shows as an overview the combination of all departments which are necessary to ensure the high quality of MAAG gears.

Included in this system is the regular calibration of the tooling, as well as the measuring instruments and machines, the balancing machines, the controls for the heat treatment facilities and the air conditioning system for the tooth grinding areas, etc.

2.10) INSTALLATION ON BOARD SHIP

Careful installation of the gear box and alignment of the input and output shafting is important for reliable and silent operation.

The gear box units are installed on board ship and lined up to the propeller shafting as complete assemblies. After placing all chocks and bolts, the tooth contacts are checked to make sure that the tooth alignment has not suffered due to gear box distortion under weight.

The instrumentation (controls and interlocks) are wired to terminal boxes fitted to the gear casing. The wiring to the control rooms is simply plugged into these terminals.

Oil tank and lube oil system must be thoroughly cleaned before filling with lube oil. Flushing of the system prior to sea trials prevents scratched bearings and journals.

The above work is supervised by a MAAG gear instructor. For later maintenance and inspection work, a complete instruction and installation manual with the description of all main gear parts, as well as supplied auxiliary equipment, is issued.

3. CALCULATION AND MEASURING TECHNIQUE

3.1) Torsional vibration calculation

For silent ship propulsion, the gear box system must not be driven in torsional natural frequencies. As an example, Fig. 9 shows that the noise level generated by a reduction gear and the number of the appearing harmonics increase if it is operated at the natural frequency.

Therefore, prior to all manufacturing work of a marine gear box, a torsional vibration analysis of the rotor system is essential. It consists mainly of the gear rotors which are connected to each other by gear tooth stiffnesses (s. Fig. 10) as well as shaft sections. With the mass distribution in the special application of marine gear boxes it is normally possible to reduce the entire system back to two or three mass systems, which represent the single gear stages with added shafts and/or couplings. Due to the low stiffness of the intermediate or quill shaft, the gear stages are able to be calculated separately, but including the locked train configuration, of course.

For example, the natural frequency of a two-mass system is given by:

$$\nu_{res} = 1/(2\pi) \sqrt{c_{\varphi} / J_{red}} \quad (1)$$

where J_{red} represent the inertias of both masses:

$$J_{red} = J_1 J_2 i^2 / (J_1 i^2 + J_2) \text{ in } \text{kgm}^2 \quad (2)$$

reduced to the centre line of mass No. 2.

In terms of tooth contact the referring stiffness value is to be calculated with:

$$c_{\varphi} = 0.25 d^2 \cos \alpha_{wt} b c_{\chi} \text{ in } \text{Nm/rad} \quad (3)$$

d : pitch diameter of the gear wheel,

α_{wt} : transverse working pressure angle,

b : face width,

c_{χ} : mean value of tooth stiffness (s. Fig. 11),
20...24 N/(mm μ m).

An analytical solution for a three-mass system (i.e. two natural frequencies) is available as well. Numerical solutions are to be used for more complex mass systems. In practice, the tooth stiffness is not constant at c_y , but changes periodically acc. to Fig. 11, as the height of the contact path above the tooth root varies as well as the load sharing on the teeth during one tooth contact sequence.

The tooth stiffness amplitude is significantly different between spur and helical gears. Fig. 12 shows as an example the referring tooth load curves under various operation speeds. These effects during load transfer in tooth mesh are the main reasons for noise excitement in gears.

The purpose of the analysis is to examine, whether the dominating natural frequencies of the rotating system cope with rotational frequencies ν_r or tooth mesh frequencies generated during operation within the required speed range. Under normal operating conditions and with highly accurate balanced rotors, the tooth mesh frequencies are regarded as critical items:

$$\nu_z = n z / 60 \quad \text{in Hz} \quad (4)$$

with z as the number of gear teeth and n as the rotational speed in rpm.

The evaluation of the results can be shown in diagrams acc. to Fig. 13. The tooth mesh frequency lines at ν_{zx} and ν_{zx} cross the horizontal lines which refer to the natural frequencies of the 1st and 2nd gear stage. With the applied example, a subcritical operation within the speed range up to $n_{cr} = 4000$ rpm is proven, unless at the very nominal speed a resonance in the 1st gear stage is to be expected. This is able to be avoided by adapting inertia moment corrections such that the ν_{zx} -line inclination being lowered.

3.2 Noise level definitions

3.2.1 Air borne noise

The airborne noise level depends on the amplitude of sound pressure variations. The pressure range leads from 10^{-4} μbar up to 10^{+3} μbar . As the sensitivity of the human ear is not proportional to this range, a logarithmic functional connection is defined as:

$$L_p = 10 \log(p^2/p_0^2) \text{ in dB} \quad (5)$$

with L_p as sound pressure level, p as actual sound pressure, $p_0 = 2 \cdot 10^{-4}$ μbar as sound pressure base. Thus L_p varies from zero (not audible) to 130 dB (for example noise from an aeroplane propulsion turbine, causes ear pain).

The audible frequencies normally lead from 20 Hz up to 18000 Hz. But frequencies lower than 500 Hz at constant sound pressure are perceived to be softer the lower they are. Therefore, noise measurements can be filtered according to different filter types as they are shown in fig. 14. Type (A) filter normally corresponds best to human sensitivity. It is, therefore, used as a standard for noise measurement evaluation.

3.2.2 Structure borne noise level

The structure borne noise level in principal is evaluated the same way as the airborne noise level. With the acceleration picked up at discrete structure locations, it is given by:

$$L_a = 10 \log(a^2/a_0^2) \text{ in dB} \quad (6)$$

Herein mean:

L_a : acceleration level,

a : actual acceleration,

a_0 : acceleration base = 10^{-5} m/s^2 .

A further calculation method often being used in European standard organizations is based on velocity values:

$$L_v = 10 \log (v^2/v_0^2) \text{ in dB} \quad (7)$$

with:

L_v : velocity level,

v : actual velocity at measuring point,

v_0 : velocity base = $5 \cdot 10^{-8}$ m/s.

The structure borne noise level at the footplate of gearboxes is influenced by the foundation structure and the mounting system. The frequency absorbing behaviour of the foundation system can be described with the impedance level:

$$L_I = 10 \log (J/J_0^2) \text{ in dB} \quad (8)$$

where the base $J_0 = 1$ Ns/m is normally used.

The transformation between equation (6) and (7) is given as follows:

$$L_v = L_a + 20 \log (a_0/v_0) - 20 \log (2\pi f) \quad (9)$$

dependant on the frequency value f , respectively.

3.3 Measuring Methods

Fig. 15 shows the arrangement for partial load tests with a frigate gear box, in order to measure the structure borne noise at the foot plate. At four previously selected locations, the installed acceleration transducers are able to measure in three orthogonal directions.

By means of a FFT-(fast Fourier transformation)-analyzer the results are obtained as octave or terz band evaluation. In Fig. 16 the noise level appears as spread band curve where the measurement values of all locations are included. According to most naval specifications, the power average value may not be exceeded which is a logarithmic addition:

$$\sum L_v = 10 \log \left(\sqrt{10^{L_{v1}/10} + 10^{L_{v2}/10} + \dots + 10^{L_{vn}/10}} \right) / n \quad (10)$$

For proper design of low noise gear boxes, it is useful to know the impedance level of the corresponding ship

An extensive amount of noise level measurements of various MAAG marine gear boxes were analyzed. According to the type of excited frequencies the total range could be separated roughly into three sections:

- a) 0 ... 100 Hz: rotational frequencies,
- b) 100 ... 1500 Hz: tooth mesh frequencies,
- c) > 1500 Hz: harmonics

Due to the intensity reduction of harmonics, the curves decline with increasing frequencies. Noise reduction achieved with resilient mount application arises to values up to 20 dB. In all cases the referring specification requirements were fulfilled.

The frequency damping characteristics of an elastic mount is represented by its own natural frequency:

$$\nu_E = 1/(2\pi) \cdot \sqrt{g/f_F} \quad (11)$$

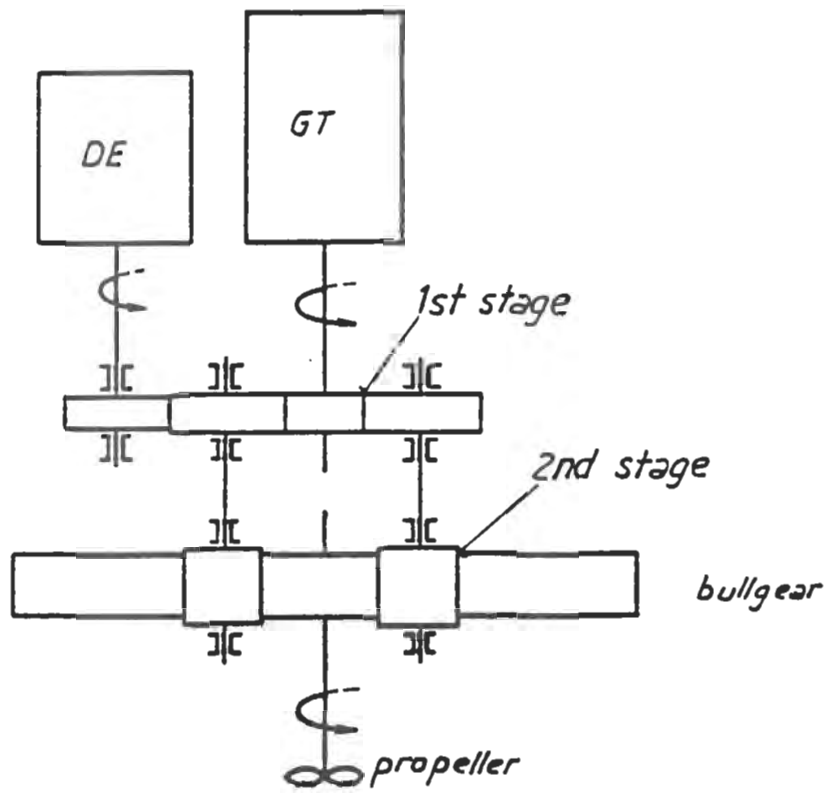
Herein mean:

g : nominal acceleration = 9.81 m/s²,

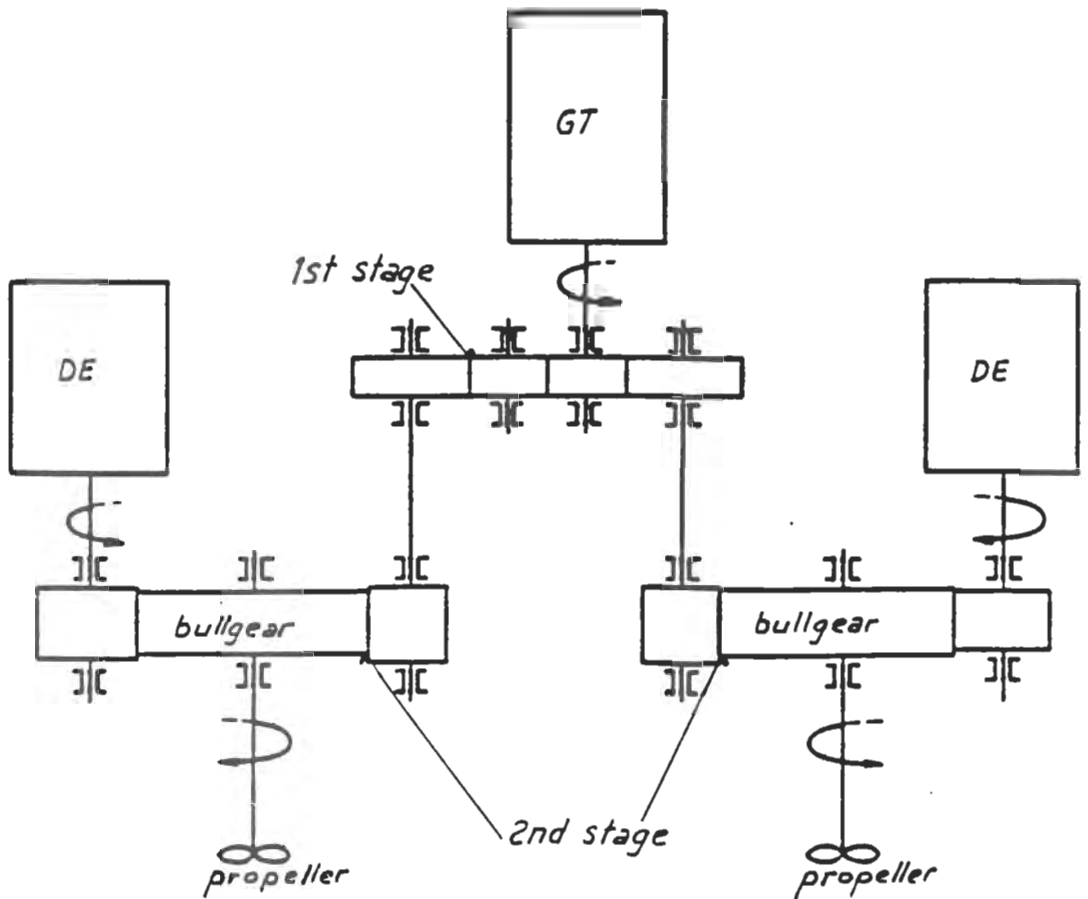
f_F : mount deflection under gearbox weight.

Thus, a favourable low natural frequency is limited by allowable deflection values given by input and output shaft connections. In general, one has to find a compromise between both aspects and optimize the design of the mounting system.

a)



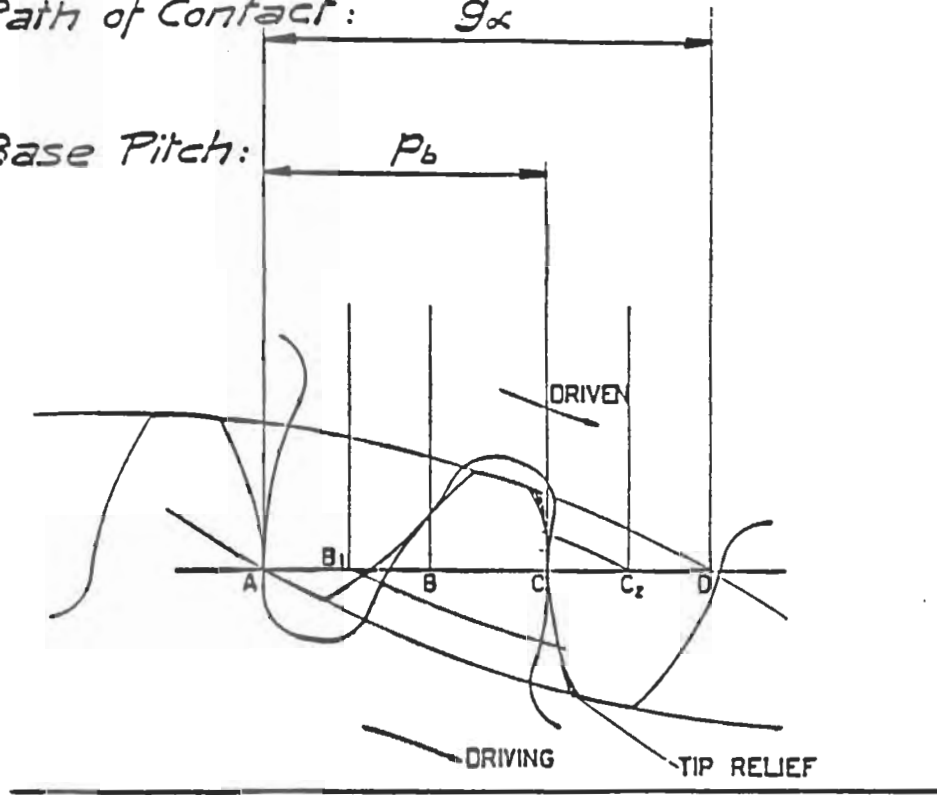
b)



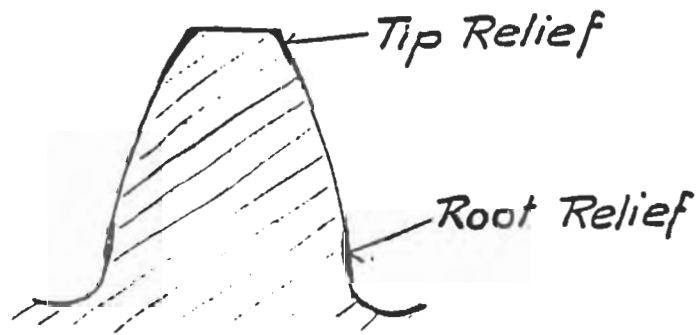
Contact Ratio $\epsilon_{\alpha} = \frac{g_{\alpha}}{p_b}$

Path of Contact: g_{α}

Base Pitch: p_b

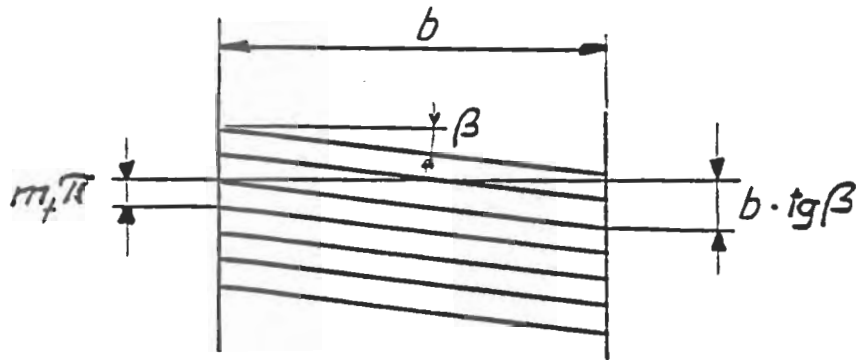


Pinion Tooth with Profile Corrections



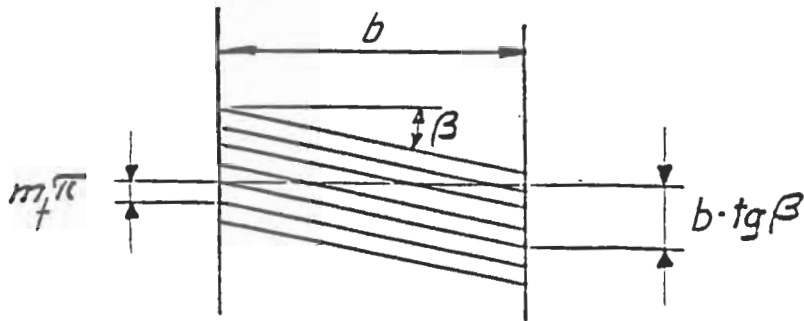
TOOTH OVERLAP RATIO.

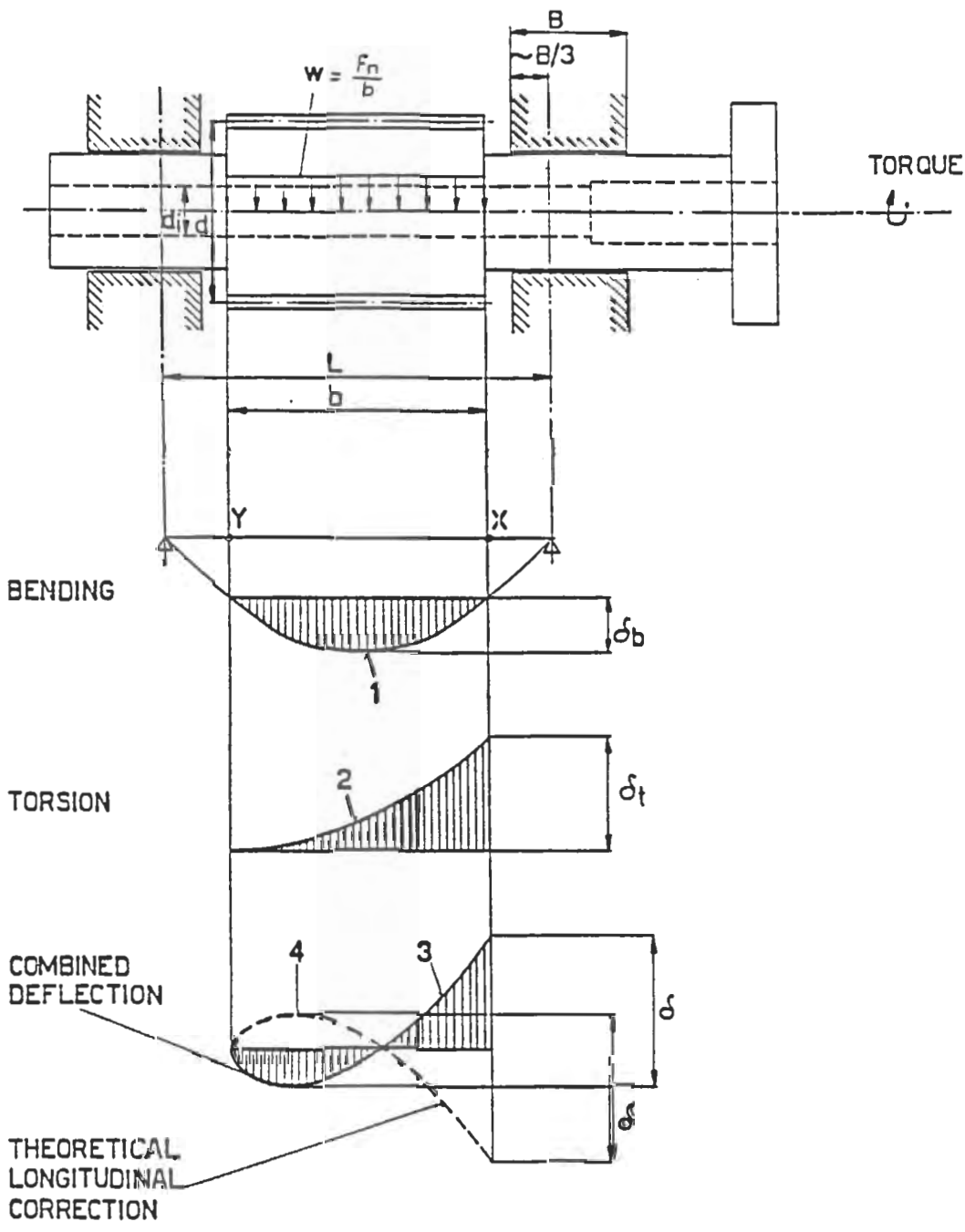
TANDEM



$$\varepsilon_{\beta} = \frac{b \cdot \operatorname{tg} \beta}{m_f \pi}$$

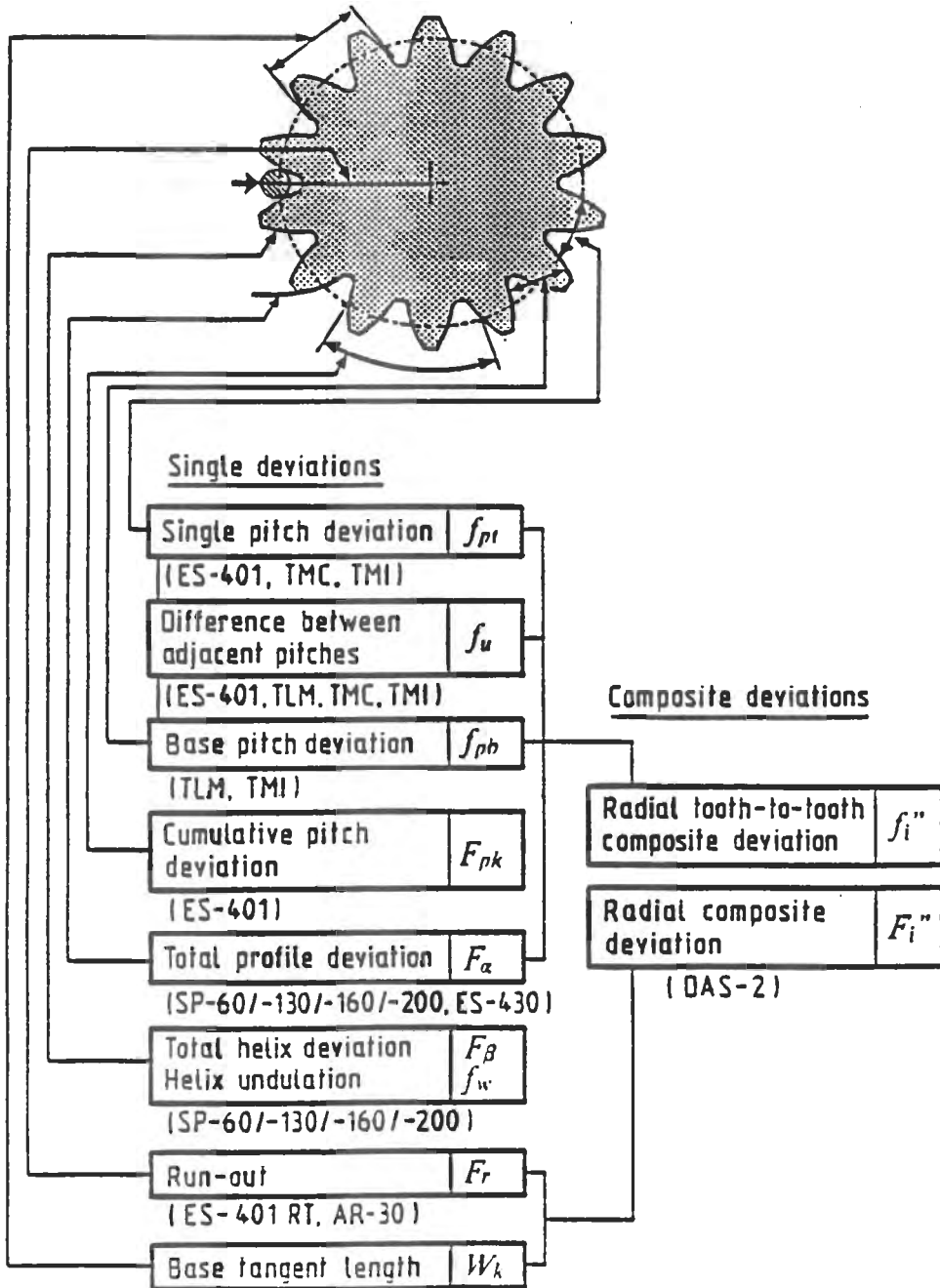
LOCKED TRAIN



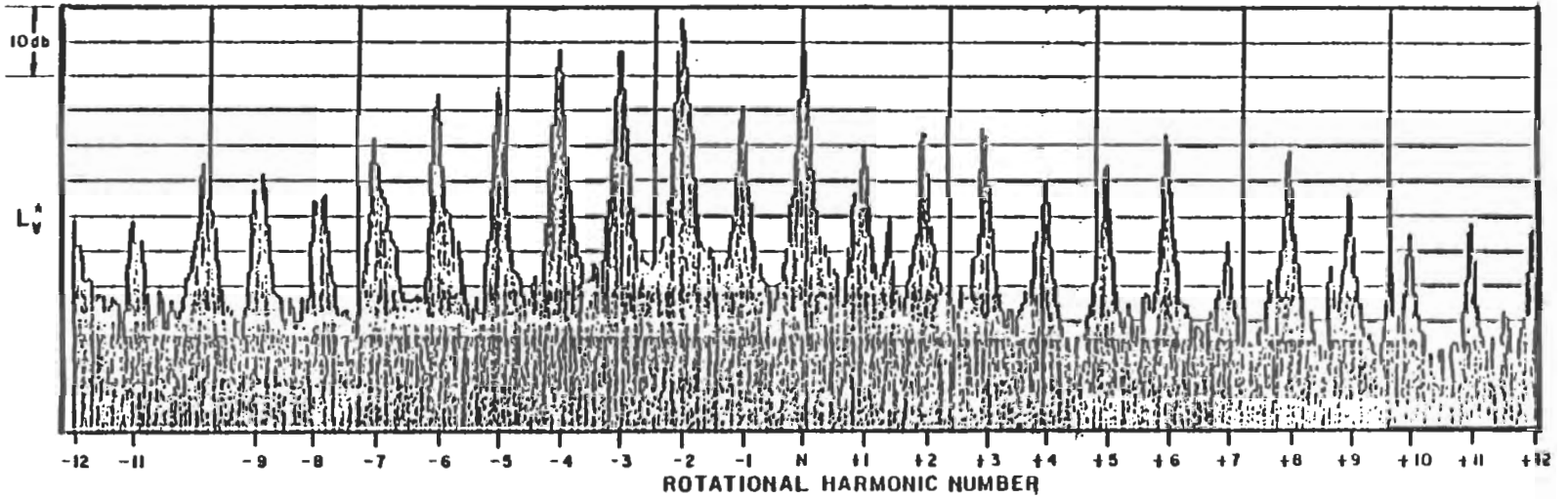


PINION DEFLECTIONS AND LONGITUDINAL CORRECTIONS

Single and composite gear deviations of cylindrical gears and appropriate MAAG checking equipment



LOW QUALITY GROUND GEAR - ROTATIONAL HARMONIC STRUCTURE



HIGH QUALITY GROUND GEAR - ROTATIONAL HARMONIC STRUCTURE

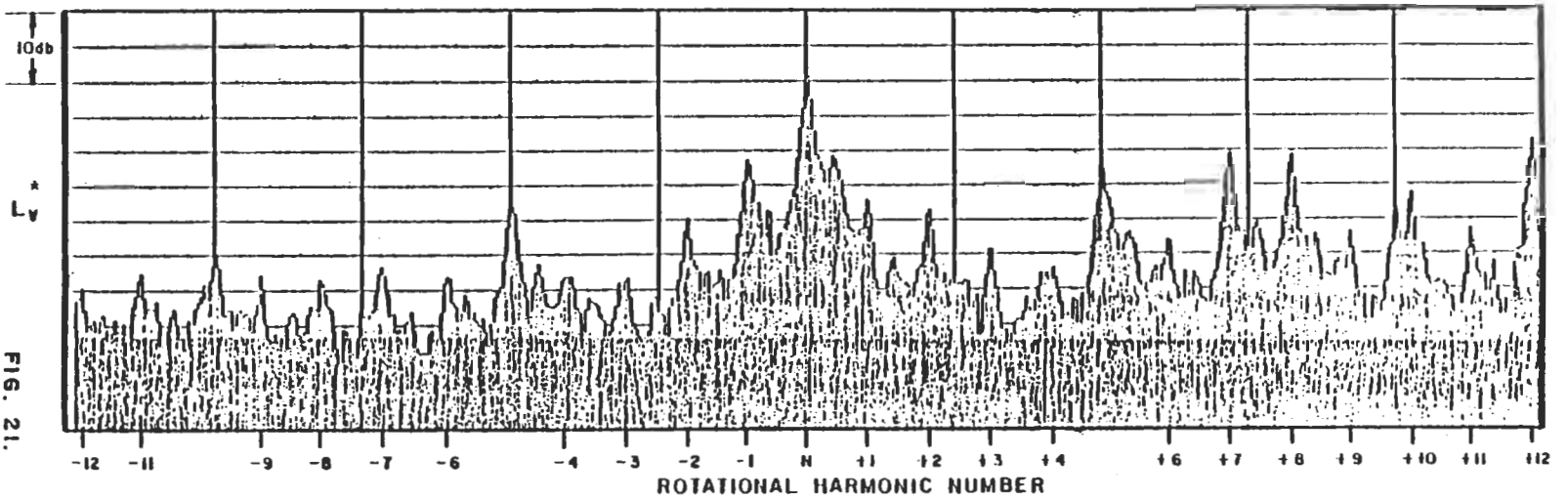
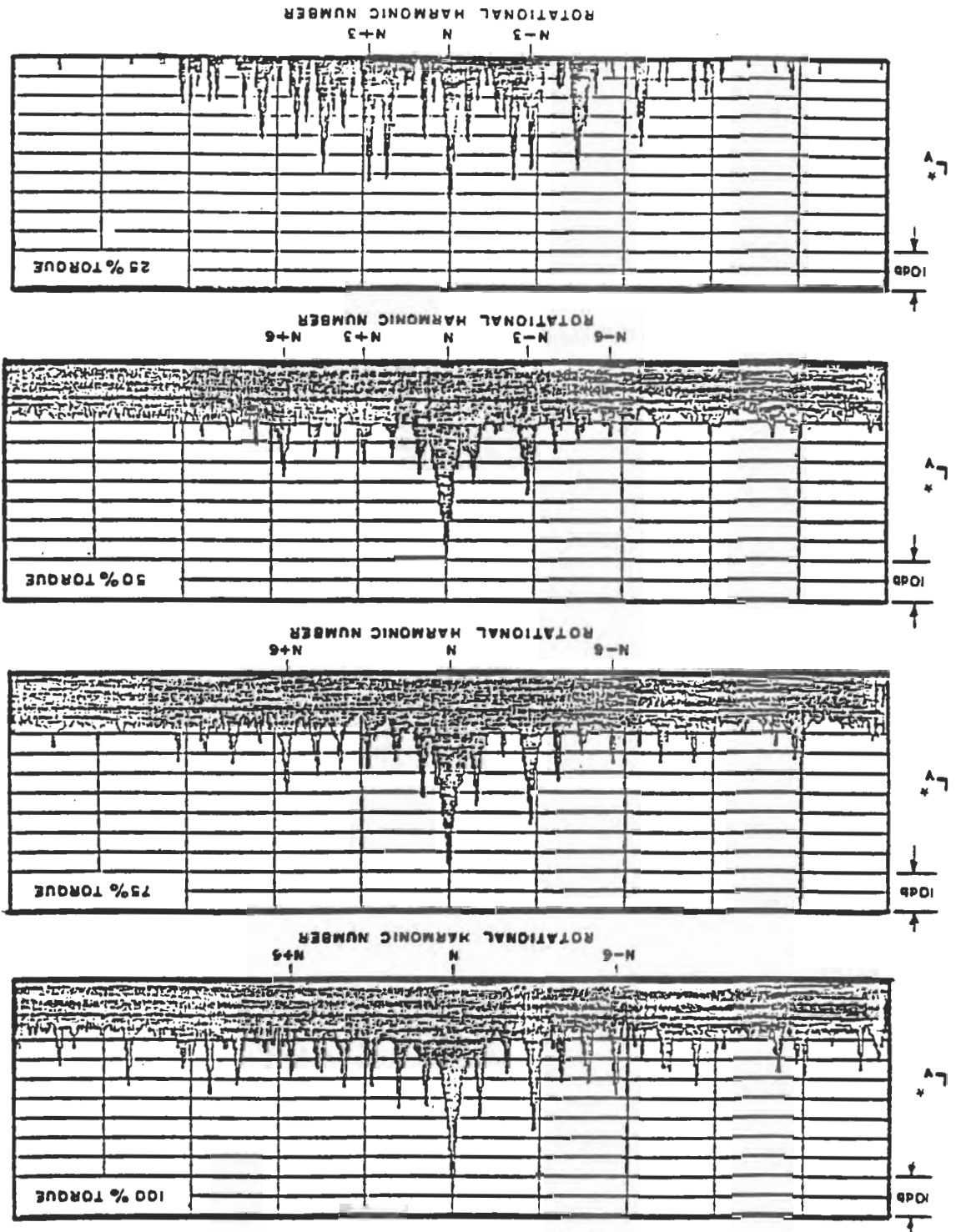


FIG. 21.

**COMPARISON OF ROTATIONAL HARMONIC STRUCTURE
WITH DIFFERENT LOADS**



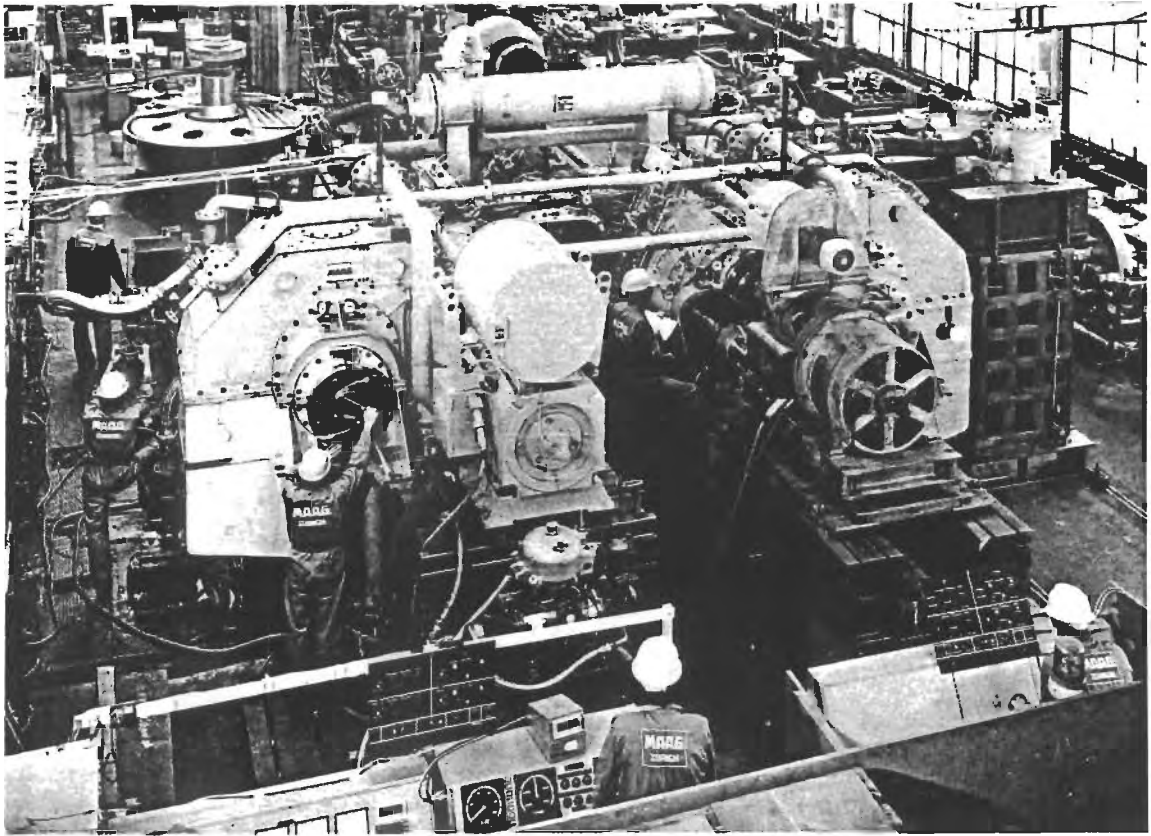


Fig. 7: Marine gear set for corvette on test bed.

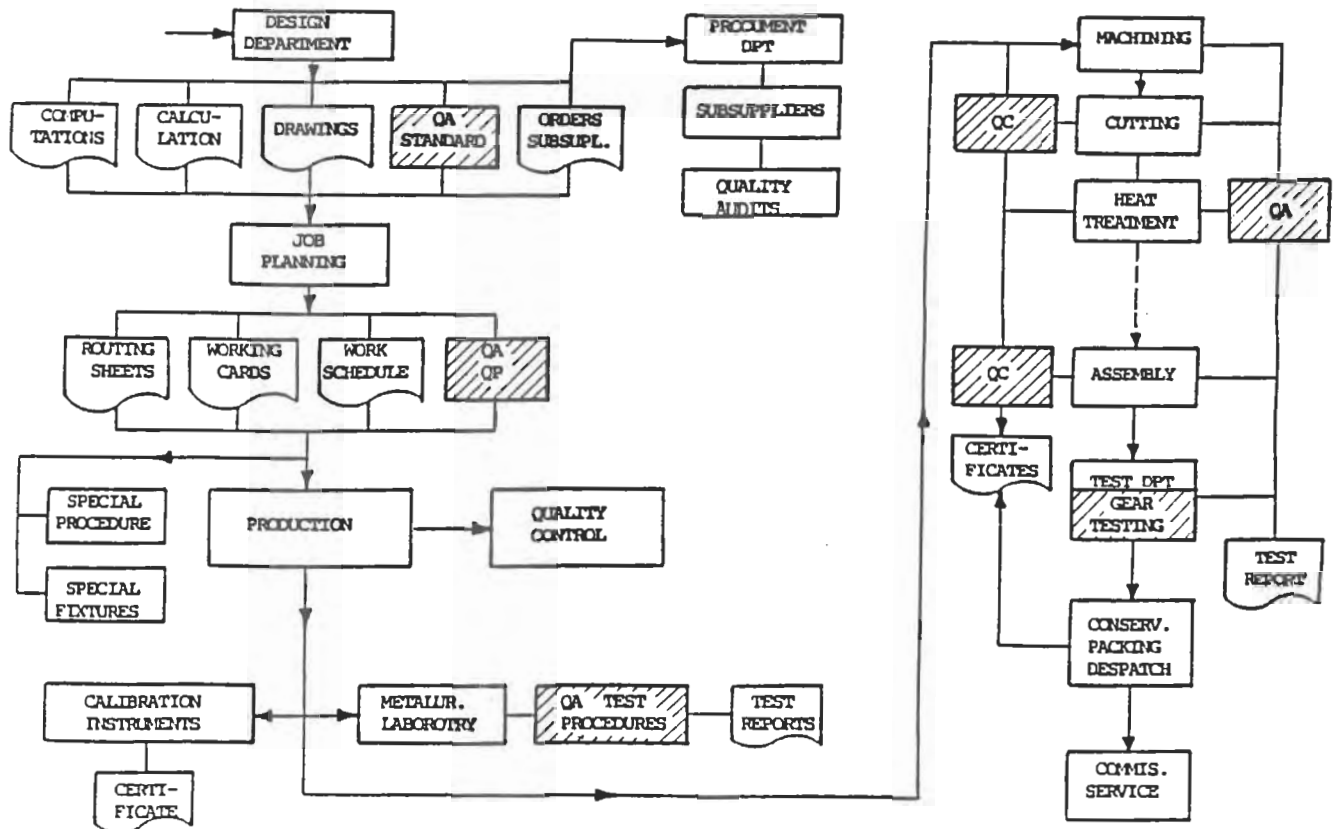


Fig.8: Quality assurance system.

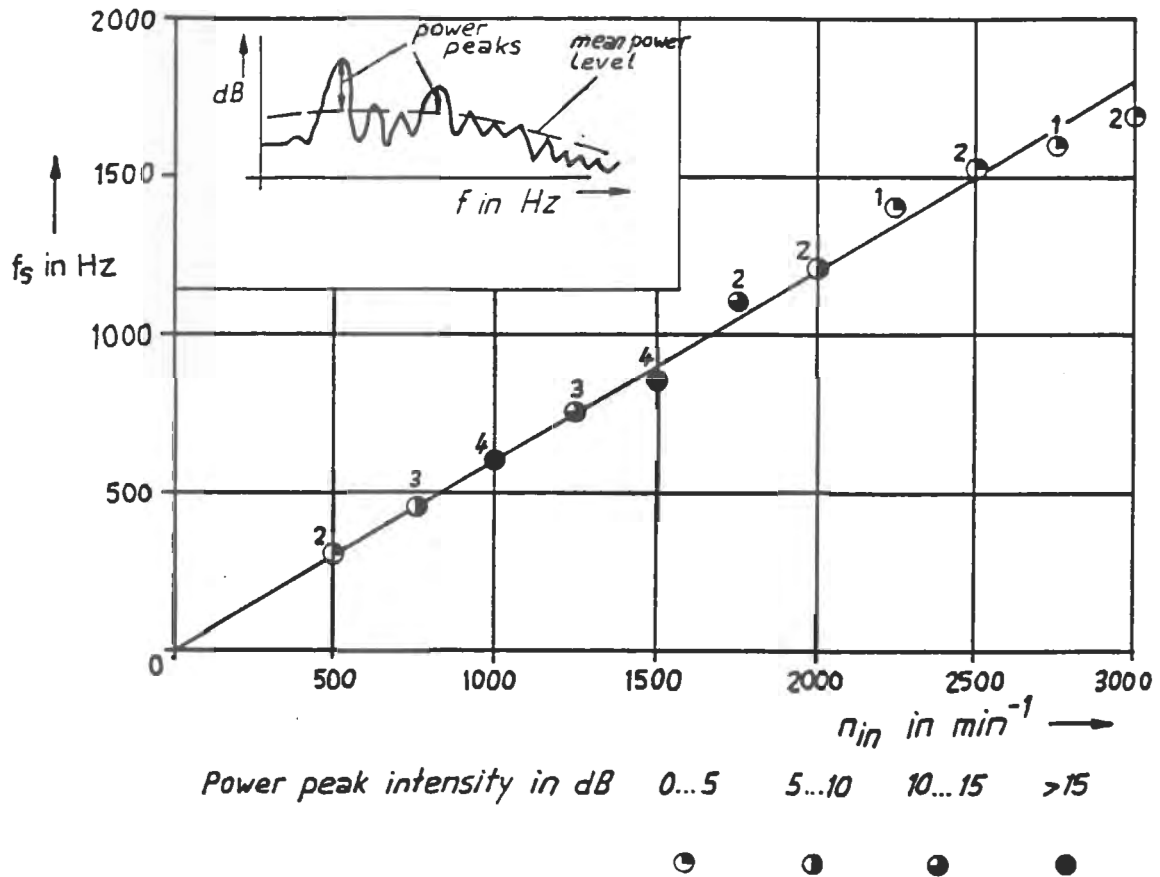


Fig. 9: Significant frequencies f_s of a gear system operated at various input speeds n_{in} . The intensity is marked by the darkness of the symbols, number means harmonic until grade n . Line acc. to calculated tooth mesh frequency.

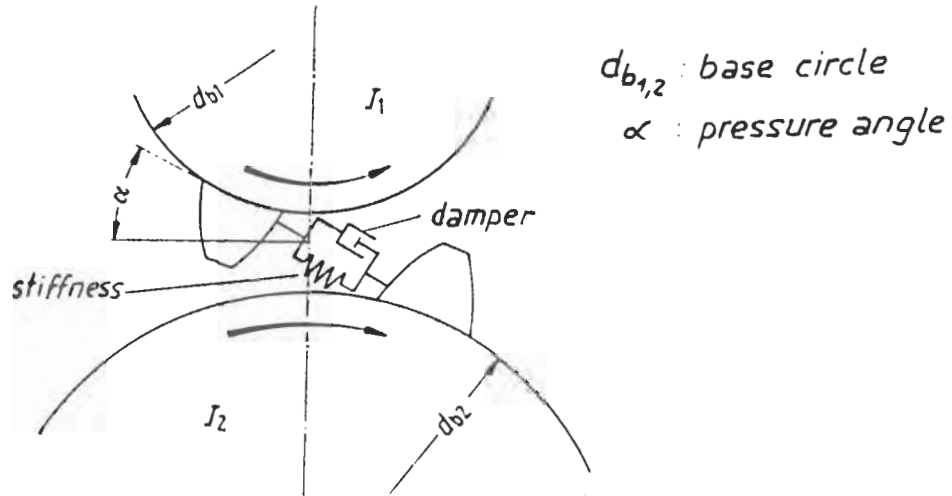


Fig. 10: Two-mass torsional vibration system, consisting of pinion J_1 , gear wheel J_2 , and tooth mesh stiffness and damper.

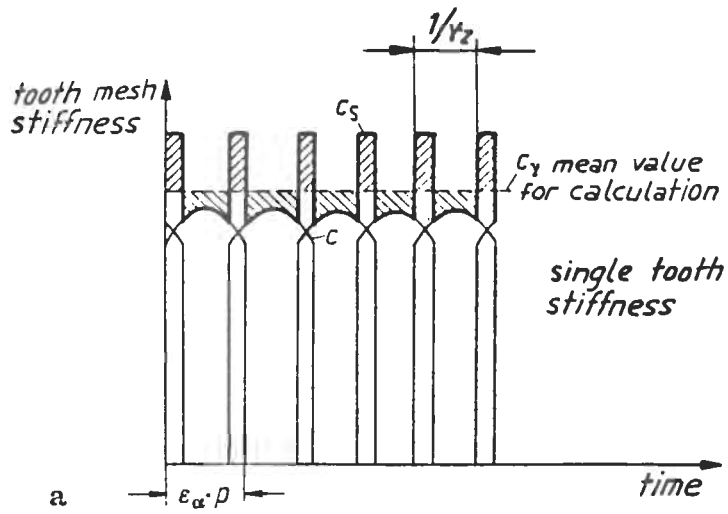
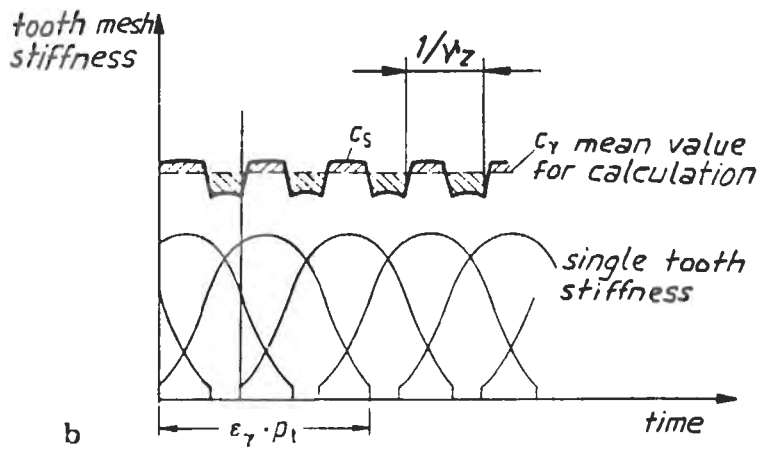


Fig. 11: Periodical change of tooth mesh stiffness with
 a) spur gears, b) helical gears.

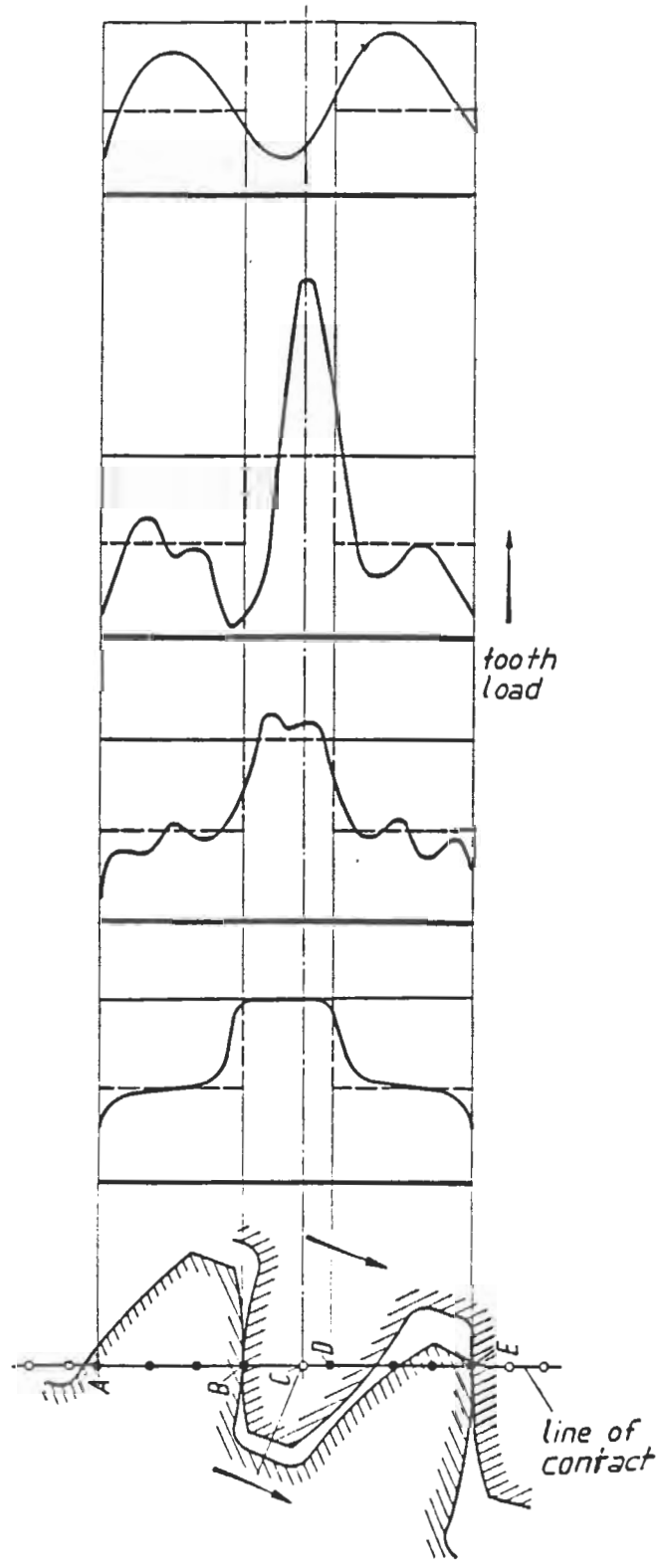


d)

c)

b)

a)



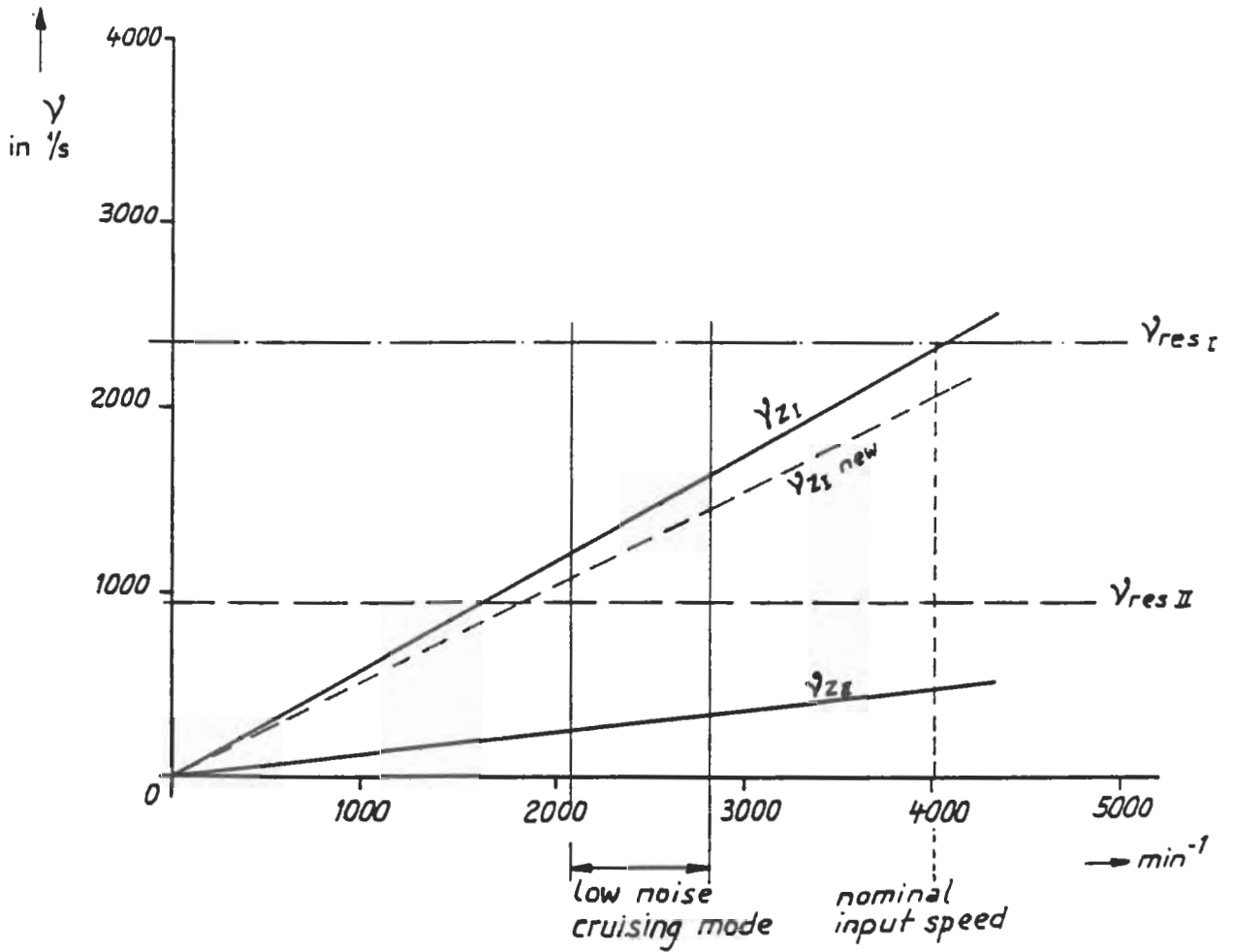


Fig. 13: Comparison of tooth mesh and natural frequencies of a two-stage gear box system .