HELICOPTER INTERNAL NOISE REDUCTION RESEARCH AND DEVELOPMENT

APPLICATION TO THE SA 360 AND SA 365 DAUPHIN

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INTRODUCTION

With the extension of the civil commercial market, the noise inside helicopter cabins is becoming one of the foremost problems for the comfort of passengers.

As shown in the statistical study of figure 1, the helicopter is the most noisy vehicle in comparison with other ground or flying vehicles.

QUALIFICATION AND IDENTIFICATION OF ANNOYANCE

We have been concerned for several years by the internal noise problem, and, as a first step, the sources of noise inside cabins have been investigated. As shown in figure 2, there are many possible sources os noise (main rotor, tail rotor, engine(s), gear boxes, or accessories). As each of them has peculiar acoustic noise characteristics (rotational noise, principally), they can be identified by narrow-band analysis of the sound signal.

Analysis of noise recordings taken inside bare cabin helicopters in flight has shown that the most significant source of noise is the main gear box (fig. 3) which gives rise to a large number of pure tones emerging 10 to 30 dB from the broad-band spectrum.

Knowing the source of noise, we have quantified the annoyance by applying the present conventional units (fig. 4):

1) A and D weighted dB

2) Speech interference levels by evaluating the mean level in the voice frequencies

3) Perceived noise level from Noys curves and correction of this level for spectral irregularities

Very soon, we saw from the passengers evaluation that the usual units could be inconsistent with the subjective answers collected and that they were accordingly only imperfectly representative of the annoyance actually felt in helicopter cabins. (See fig. 5.) Such disagreement is bound to the existence of pure tones within the 500 to 5000 Hz range; these frequencies exceed the 20 dB broad-band noise and even sometimes can exceed 30 dB.

In order to quantify the additional annoyance, we have carried out psychoacoustic studies on the effect of the pure tone emergence. (See fig. 6). Wide band type noises actually measured on the helicopter were submitted to a large number of juries for comparison with the same noise to which a pure frequency of 1000, 2000 Hz or 1000 and 2000 Hz had been electrically added. The results of this study have demonstrated that the conventional units,dBA, dBD, PNdB, and even TPNdB, underrated the effect of the emergence of pure tones within the 3 to 9 dB range for dBD and PNdB and within the 1 to 5 dB range for dBA and TPNdB (emergence of 20 dB and 30 dB in narrow-band analysis). This nonexhaustive study of the emergence effect has shown that the significant parameters of the internal noise of helicopters are the pure tones at the meshing frequencies of the main gear box.

MESHING NOISE REDUCTION STUDY OF MAIN GEAR BOX

The purpose of the study undertaken is the acquisition of knowledge on (fig. 7):

(a) mechanisms of vibration generation inside the main gear box - study of meshing

(b) mechanisms of transmission between source and cabin, through the study of dynamic behaviour of main gear box components (pinions, casings, etc.) or those linking the main gear box to the cabin structure

Noise Sources

Gear meshing is a noise generator due to its design and realization; angular meshing errors are generating vibrations which will excite the structure. (See fig. 8). Until the last few years, the compromise made at the design stage between the various gear parameters had for its main objective a minimum weight while ensuring a satisfactory service life. For that purpose, gear toothing was designed to work as closely as possible to the maximum permissible stresses and specific pressures but also to limit axial, radial, and tangential loads on bearings.

This choice is the contrary of the continuous meshing concept; as the tooth bending increases with the load, the low driving and overlap ratios achieved with low spiral angles and diametral pitch or high pressure angle induce sudden load variations during meshing and do not ensure the compensation of machining errors which would require the simultaneous meshing of several teeth.

In a first stage, we have measured the angular meshing error on a pair of pinions under no load by using a "GOULDER MIKRON" type checking machine (fig. 9). The results, recorded on paper in analog form, clearly show the existence of tooth profile errors superimposed on an offset error or distortion of the basic circle. The spectral analysis of these analog signals allows the separation of these phenomena and the quantification of the effect of

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additional parameters such as backlash.

An example of toothing geometry modification is as follows. On the SA 365 main gear box, the input spiral bevel gear toothing has been redesigned, taking the acoustic aspect into account; the tooth bearing pattern has been optimized to ensure a better meshing continuity. The gain achieved over the original meshing is approximately 15 dB.

Dynamic Behaviour of Detail Parts

If a meshing concept taking the acoustic aspect into account is a necessary condition to achieve a low noise level, it is not sufficient. In fact, in the transfer of vibration energy to the structure, the dynamic behaviour of each of the components constituting the transfer path (pinions, shafts, bearings, casing, and main gear box attachment fittings) has to be considered.

Axisymmetric Part Modes (Pinions, Shafts)

In a first stage, an experimental and theoretical mode determination has been made for the parts constituting the geat train.

Refer to figure 10 for the results of a mode determination made by using a laser holography method and a finite-element mathematical model on a SA 365 spiral bevel and planet gear assembly. The mathematical model established allows the determination of the axisymmetric part modes under load and in rotation. The agreement between modes calculated and those measured in the laboratory using laser holography is excellent up to 7 to 8 kHz.

The search for agreement between the SA 365 main gear box natural and excitation frequencies (fig. 11) shows that it is difficult to design a complete main gear box in which no component natural frequency would be in accordance with a meshing frequency. This difficulty of mastering the full gear train dynamic behaviour has been checked on an actual SA 360 main gear box in which the spiral bevel ring gear rigidity had been modified.

Figure 12 shows the changes in noise levels, measured on the acceptance test bench, for one of the spiral bevel gear meshing frequencies versus rotational speed and in two different configurations, initial ring gear and reinforced ring gear. According to the rotational speed, the modification may be beneficial or not, and for nonnegligible gains achieved at nominal r.p.m. at this frequency, there were appreciable losses at other meshing frequencies.

The introduction of some damping in all the gear train seems to be a useful line to follow in view of reducing the gear train dynamic responses. A calculation model of the forced response for damped axisymmetric parts should be established to allow the design of such assemblies.

Knowing the main gear box casing dynamic behaviour is a very important factor; in fact,

- Due to the vibration of its wall, the casing is a source of noise.

- The vibration energy generated at the source and transmitted to the casing through the bearings will reach the structure through the casing attachment points (main gear box suspension bars and flexible mounting plate).

- Casing supports the shafts and thus ensures proper positioning of meshing gears, hence the risk of coupling between the excitation and casing response.

Modal determination in laboratory.- As for axisymmetric parts, modal determination has been made in the laboratory on SA 330-SA 365 main gear box casings using the laser holography method. Figure 12 shows two examples of mode determination on the SA 360 casing. On the prototype casing, it has been noted that a natural frequency of 1792 Hz was close to the spiral bevel gear meshing frequency of 1850 Hz. A structural change (stiffening of casing through a rib located at midheight) has relocated the natural frequency from 1792 Hz to 1850 Hz and generated a new mode at 1729 Hz. As there is a slippage of natural frequency according to the load (from 1792 Hz to 1850 Hz) and as this has been checked on the prototype casing (fig. 13), the modified casing should not have any longer natural frequencies in accordance with the spiral bevel ring gear meshing frequency. In fact, a gain of some dB's has been noted during the bench testing of this modified casing.

Forced response of complete main gear box on test bench and on aircraft.- To check the results obtained in laboratory tests, a bench accelerometric measurement (see fig. 14 for set-up) has shown there was really a very large response of the main gear box casing at 1770 Hz, and this frequency was moving towards 1850 Hz when torque was getting nearer the nominal load.

A second check on the presence of resonance at 1850 Hz has been obtained in cruising flight by applying a damping product on the main gear box surface. The gain in noise level has been appreciable in the 2 kHz octave (an atten-uation of more than 3 dB for 1 kilogram of damping product) although the attenuation was negligible at the other frequencies.

As for the gear train detail parts, the design of casings using materials that offer a large internal damping seems to be necessary in view of limiting coupling effects between the gear train and casing modes, and also the amplitude of responses at the excitation frequencies. This will be the sub-ject of future research tasks.

Main Gear Box Suspension Bar Dynamic Behaviour

The main purpose of the main gear box suspension bars is to ensure the

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ransfer of lift loads to the structure; the attachments on structure and
ain gear box upper section are made through metal hinge fittings. Therefore,
he main gear box casing vibratory motions are transmitted to the structure
ithout possibility of energy dissipation.

For the SA 360 main gear box bars, the first bending modes, in free-free
onfiguration, have been determined in the laboratory (excitation through B
nd K vibrating pot and accelerometric recording). This mode determination
n the laboratory has allowed the validation of the mathematical model used
o calculate the bending modes and the study of the effect on the bars of
he hinges and weight (concentrated or distributed weights).

Figure 15 shows the results of the calculations made on a SA 360 main
ear box bar. We can see the correspondence between the third bending mode
requency (1850 Hz) and the spiral-bevel gear meshing frequency, together
ith the inportant displacement of the resonant frequencies according to
he type of weights added to the bars. The efficiency of these weights has
een verified in flight as, with 1.3 kilogram of lead distributed on the
our bars, the mean noise level dropped by 4.2 dB SIL (Speech Interference
evel).

IMPROVEMENT OF THE INTERNAL NOISE LEVELS BY OPTIMIZING

THE CABIN ACOUSTIC TREATMENT

Although some improvements have been made in the knowledge of means for
oise reduction at the source, these improvements are not sufficient to
nsuré a satisfactory noise level in the cabin, and a sound-proofing treat-
ent isolating the passenger has to be installed and optimized.

The treatments we have optimized associate the three following effects
fig. 16):

(a) An acoustic screen using the weight effect isolates the passenger
rom the noise source. (Item 1 on fig. 16.)

(b) A damping treatment limits the conversion of the vibratory energy
nto acoustic energy. (Item 2 on fig. 16.)

(c) An absorbing treatment achieved either through HELMHOLTZ resonators
r through a glass wool blanket limits the propagation of accoustic waves
nd the wave reflection effects in the cabin. (Item 3 of fig. 16.)

Figure 17, a section of the SA 360-365 cabin structure, shows the
nstallation of the various elements.

Figures 18 and 19 show the efficiency of the various treatments and
heir weight which is to be compared with the maximum weight of aircraft
f about 3000 kg. It can be noted that the conventional sound-proofing

treatments offer the minimum efficiency from the weight penalty aspect.

Type of action	Modification	Acoustic gain	Weight penalty
On the source	Modification of teeth geometrical characteristics	6.4 dB SIL	2 kg (approx.)
On the load transfer	(1) Treatment of a housing using a damping material	5 dB SIL	1 kg
	(2) Treatment of attachments by means of lead cloth	4.2 dB SIL	4.2 kg
On passenger's isolation	(1) Damping of cabin structure	6.8 dB SIL	26 kg
	(2) Acoustic screen	13.3 dB SIL	50 kg

CONCLUSIONS

The combined application of treatments at the source in the transfer of vibratory energy level and in the optimization of the sound barriers allowed, at high cost (weight, price), the obtainment of interesting results as shown by the narrow- and octave-band analysis of noise signals recorded in a SA 365 aircraft in flight, both with a prototype main gear box without cabin sound-proofing and with a modified main gear box and sound-proofed cabin. (See fig. 20 and fig. 21.)

A reduction of about 53 dB was optained on a pure tone (1850 Hz) which was at the origin of the main annoyance on the prototype aircraft (fig. 19). The overall noise level expressed in dB SIL and dBA has been improved by about 30 dB (fig.21), which ensures a good comfort in this aircraft.

The comparison with MIL specifications shows that it was dangerous to fly without ear protection device in the prototype aircraft and that it is now possible in the treated aircraft to fly for more than 8 hours per day without ear protection.

The comparison with airliner specifications shows that a great deal remains to be done at the mean octave frequencies of 1 kHz, 2 kHz, and 4 kHz. The possibilities of further improving the conventional acoustic treatments seem to be small.

Only an important research effort to improve the knowledge of the
echanisms generating and propagating the noises in helicopter cabins can bring
ome additional gains. Results obtained so far represent the maximum of what
an be obtained in helicopters where the main gear box is located just above
assengers, and at a cost (weight, price) which is the maximum that a
elicopter manufacturer can tolerate.
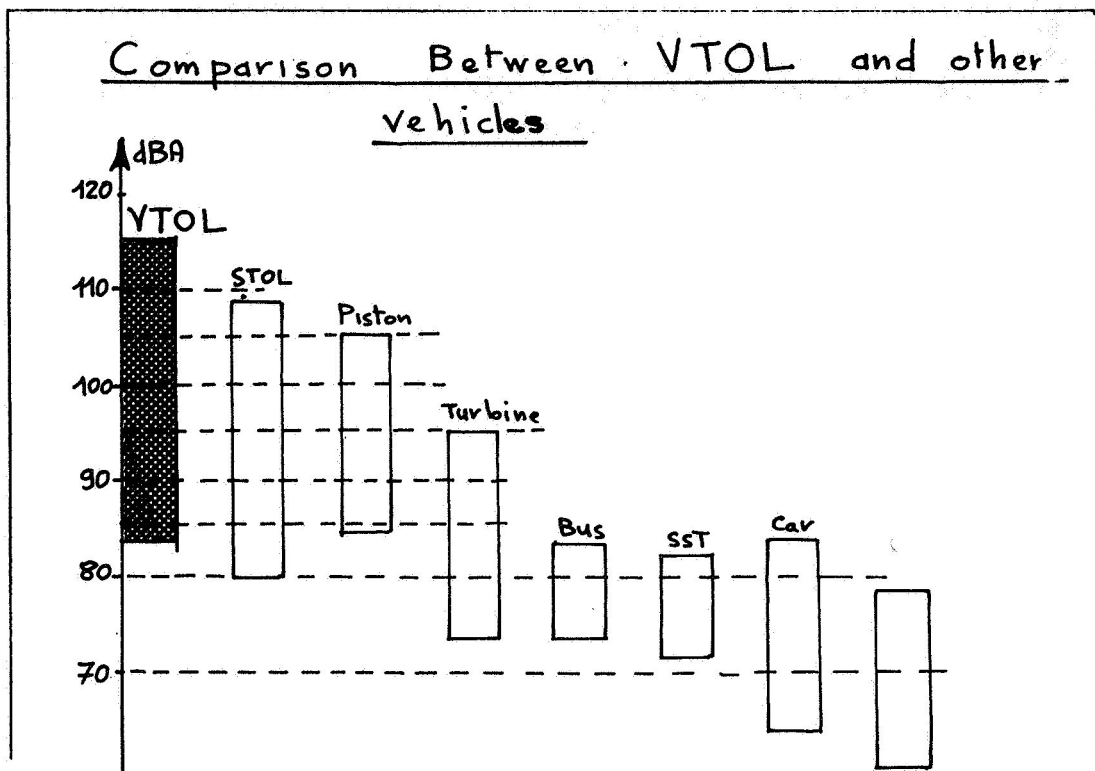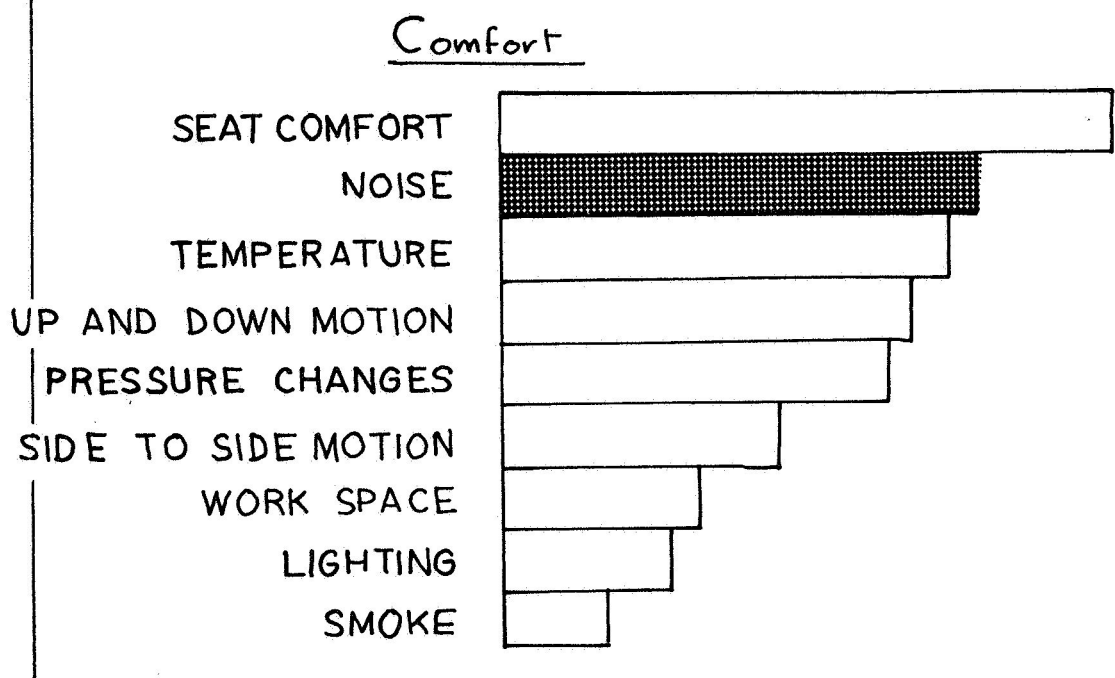
Figure 1.- Interior noise considerations for various transportation vehicles.
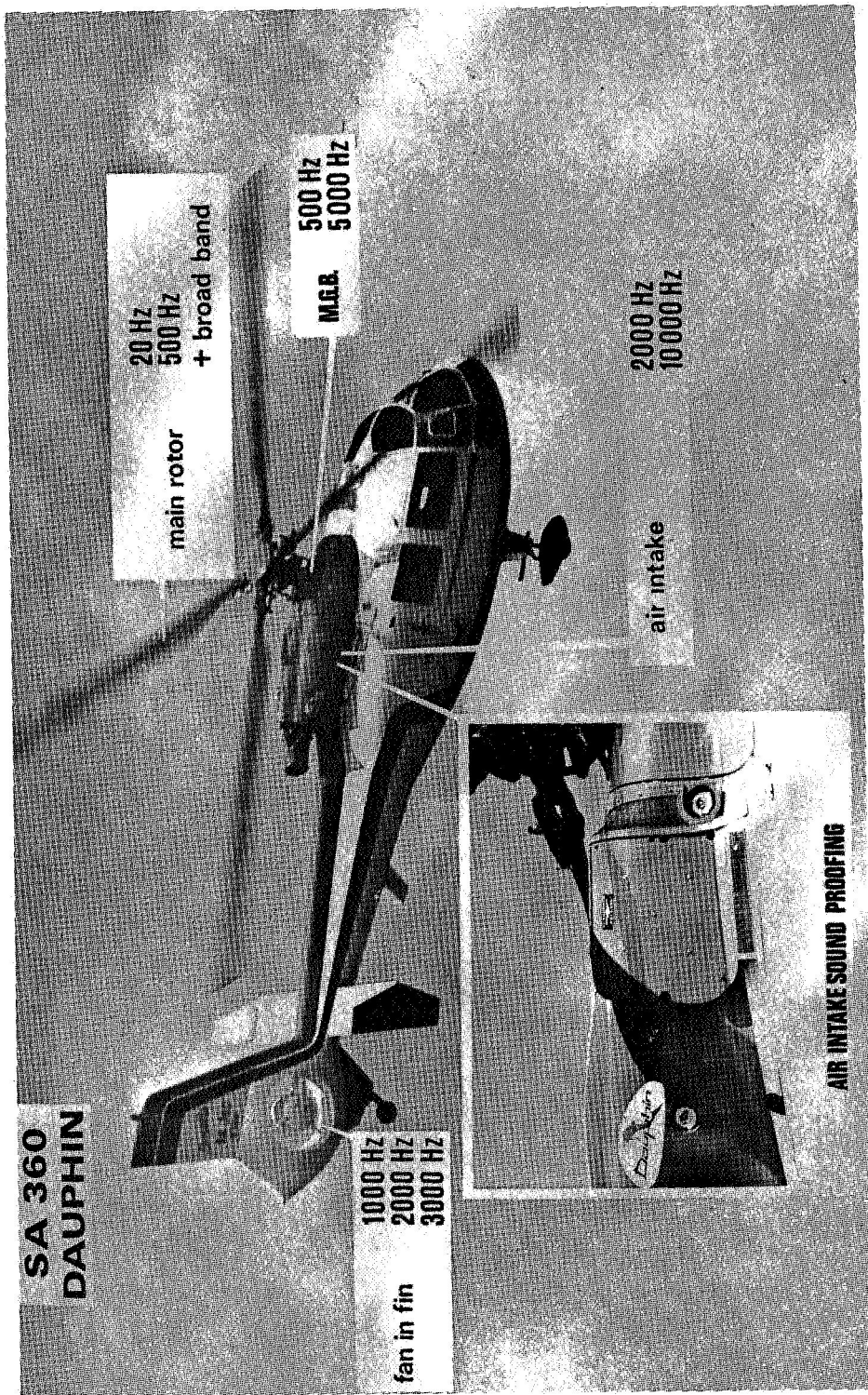
Figure 2.– Helicopter noise sources.

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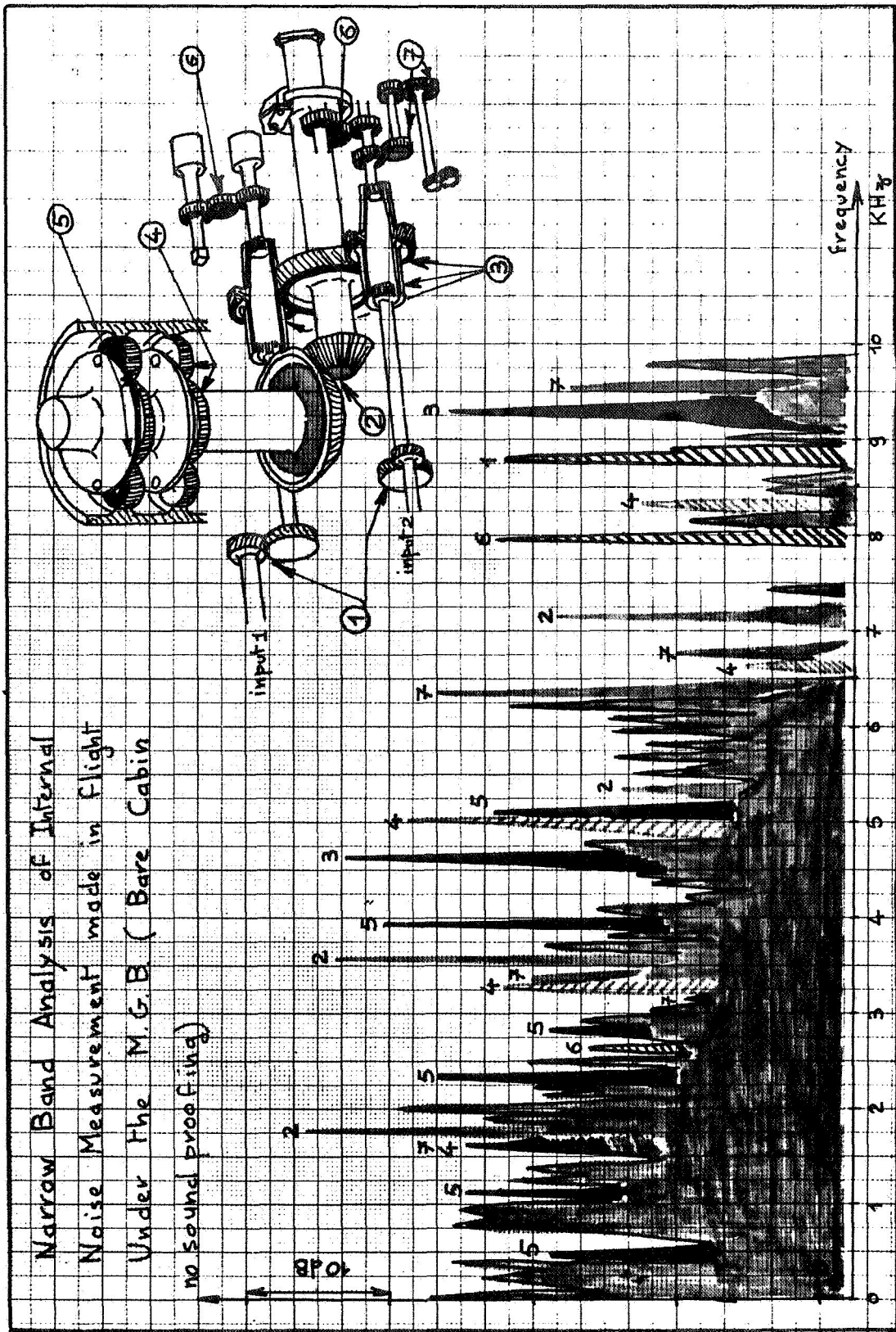
Figure 3.— Example spectrum of main gear box noise.

Perceived noise curves.

NOYS
100
50
20
10
5
2
1
0.5
0.1

dB

Acoustic Pressure Level dB

120 110 100 90 80 70 60 50 40 30 20 10 0

Frequency (Hz)
100 1000 10000

1- Calculation of total perceived noisiness N:

Combine the perceived noisiness n_i for the 24 third octave band:

$$N = 0.85 \, n + 0.15 \sum_{i=1}^{24} n_i$$

2- Calculation of perceived noise level

$$PNL = 40 + 33.3 \log N$$

3- Correction for spectral irregularities

$$TPNL = PNL + C.$$

Total Weighted Levels

Attenuation

dBA
dBD

Frequency (Hz)
1 10 100 1000 10000 100000

Average Levels Within Speech Frequencies.

$$N \, dB \, SIL = \frac{1}{4}\left(N_1 + N_2 + N_3 + N_4\right)$$

N_3
N_2
N_1
N_t

65 125 250 500 1000 2000 4000 8000

Speech Octaves

Octave Frequency

Figure 4.- Conventional units for the quantification of annoyance.

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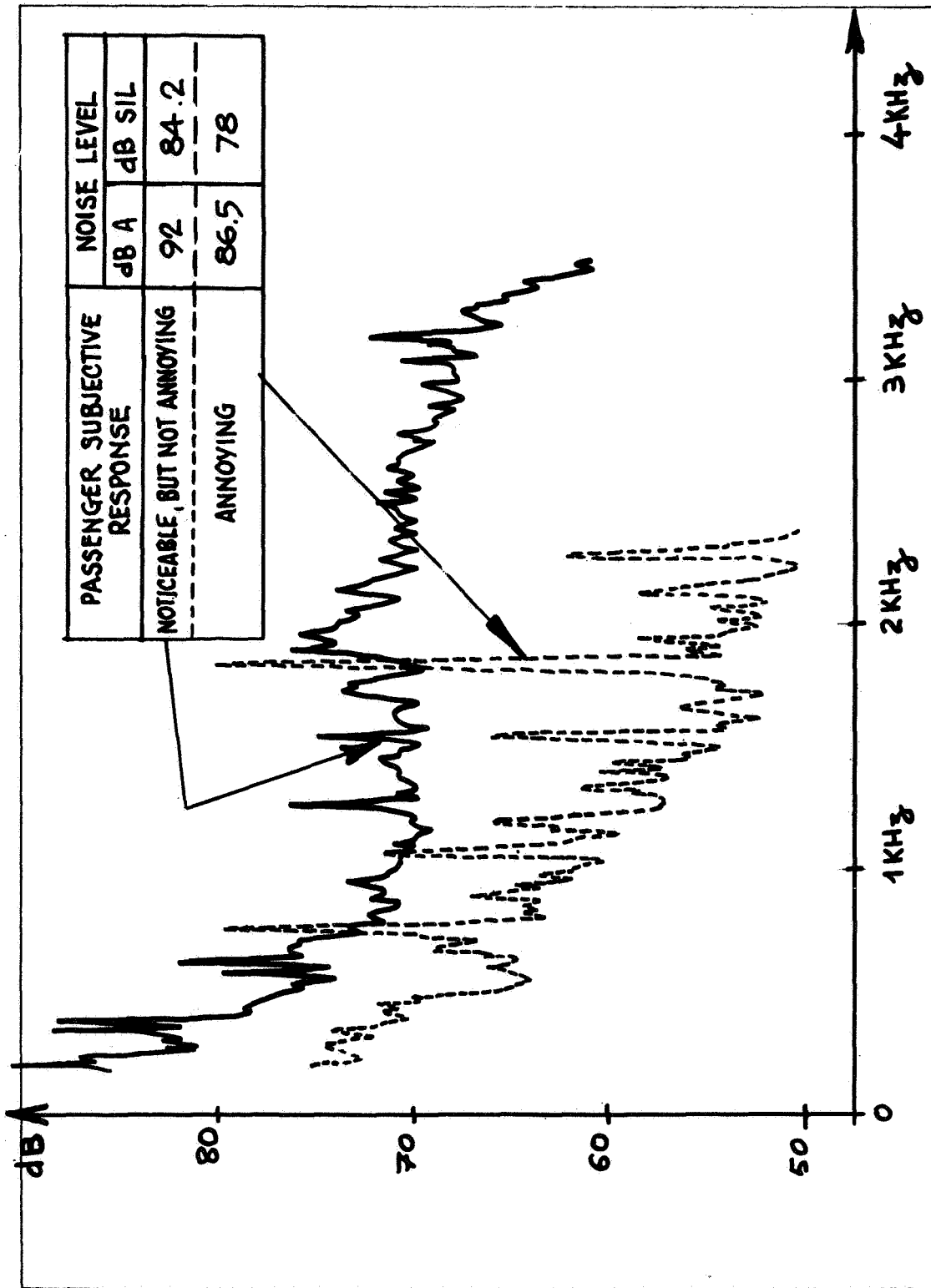
Figure 5.- Comparisons of noise level measures and associated passenger subjective responses in two helicopters.
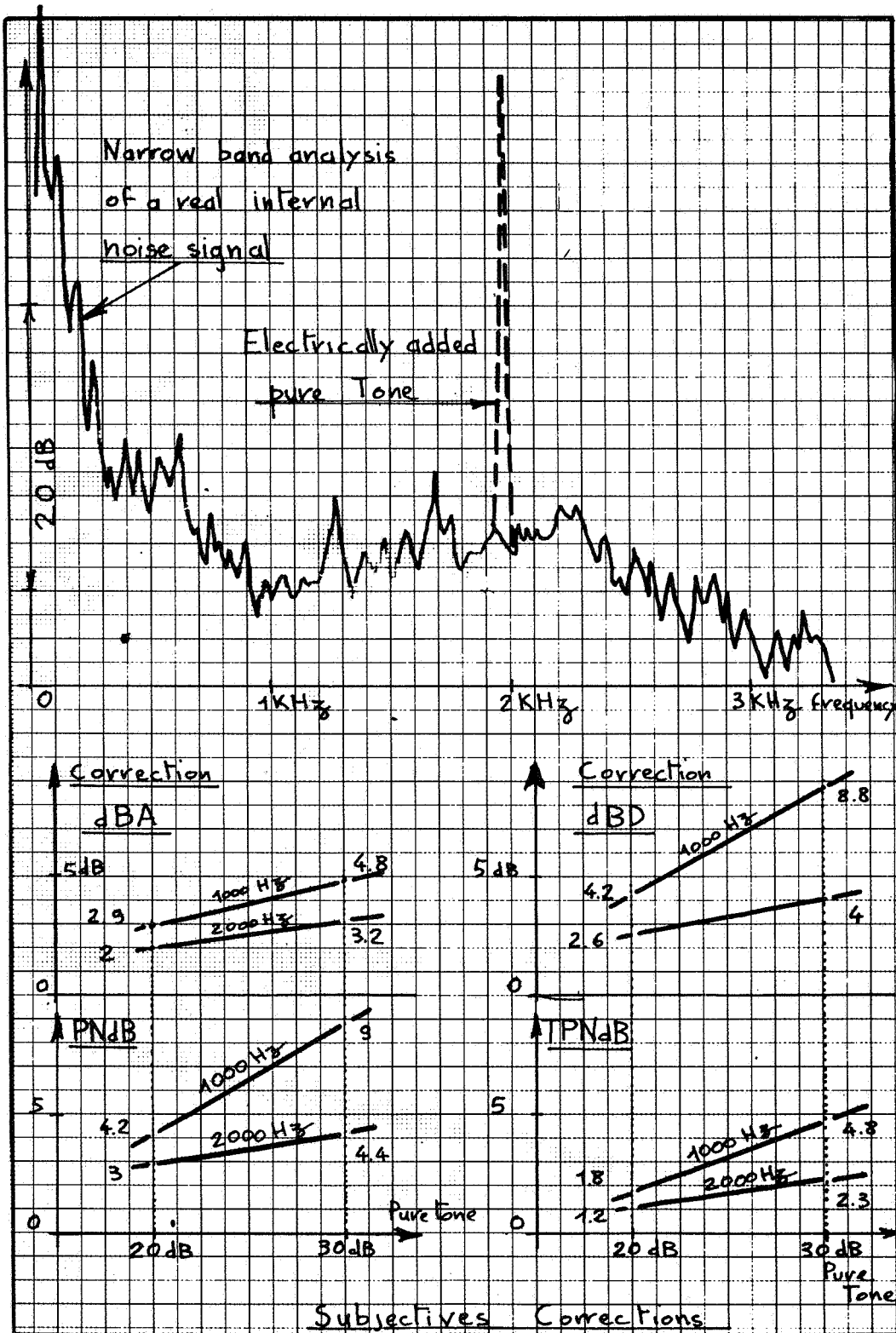
Figure 6.- Psychoacoustic study of the noisiness of pure tone internal
noise of helicopters.
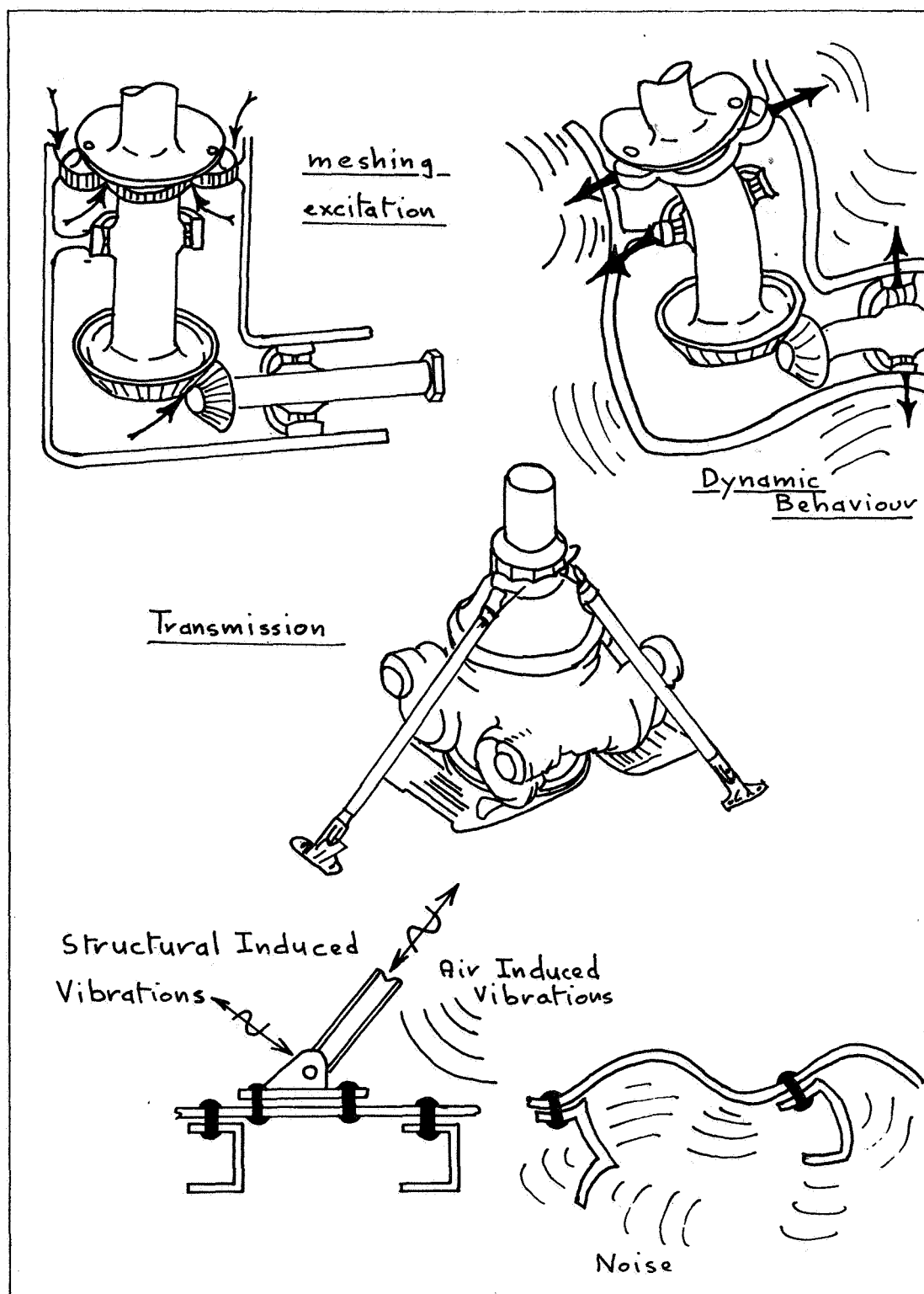
Figure 7.- Gear box noise generation and propagation considerations.
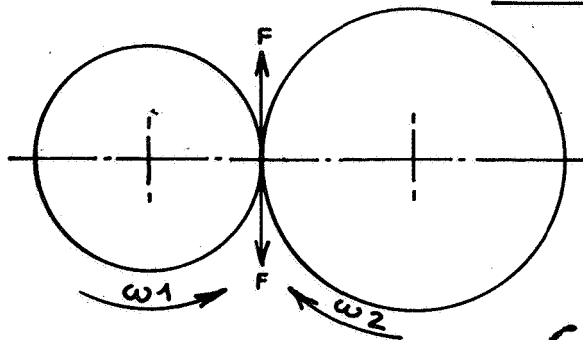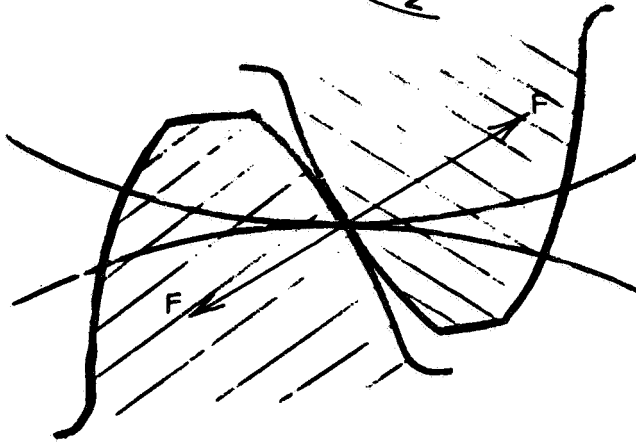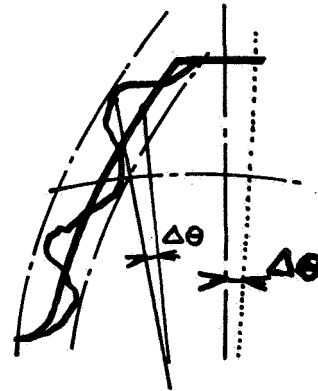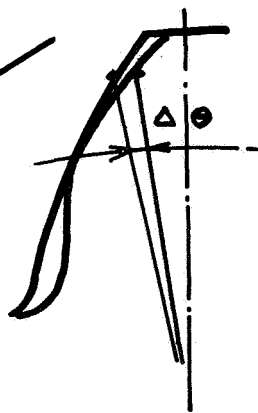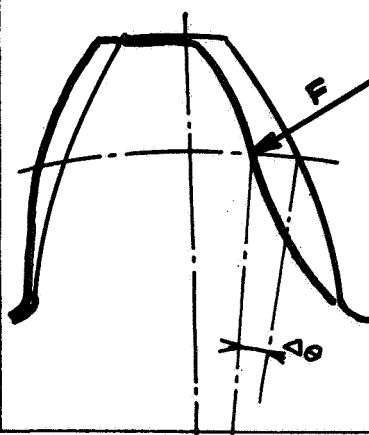
Figure 8.- Gear tooth design factors in the generation of gear meshing noise.
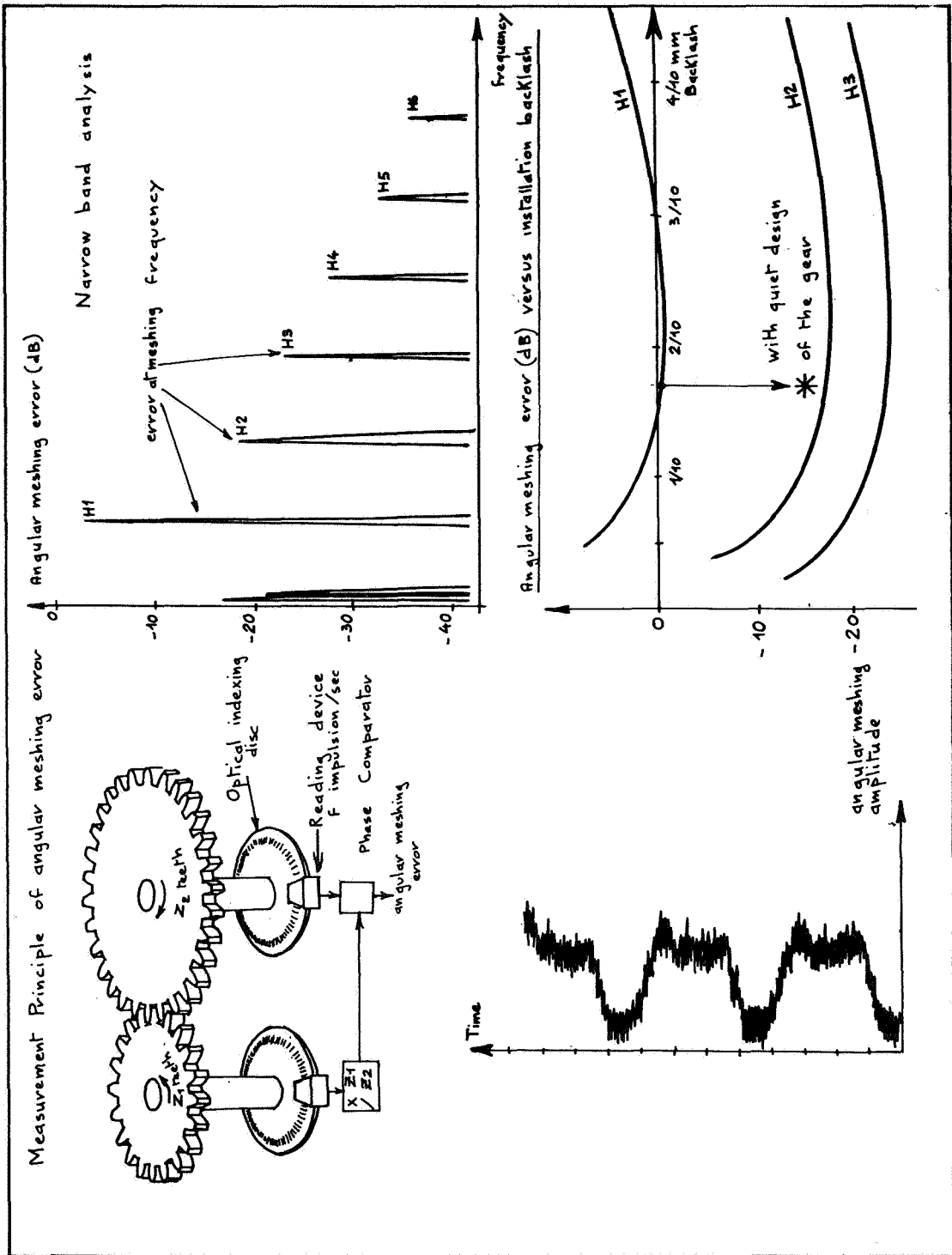
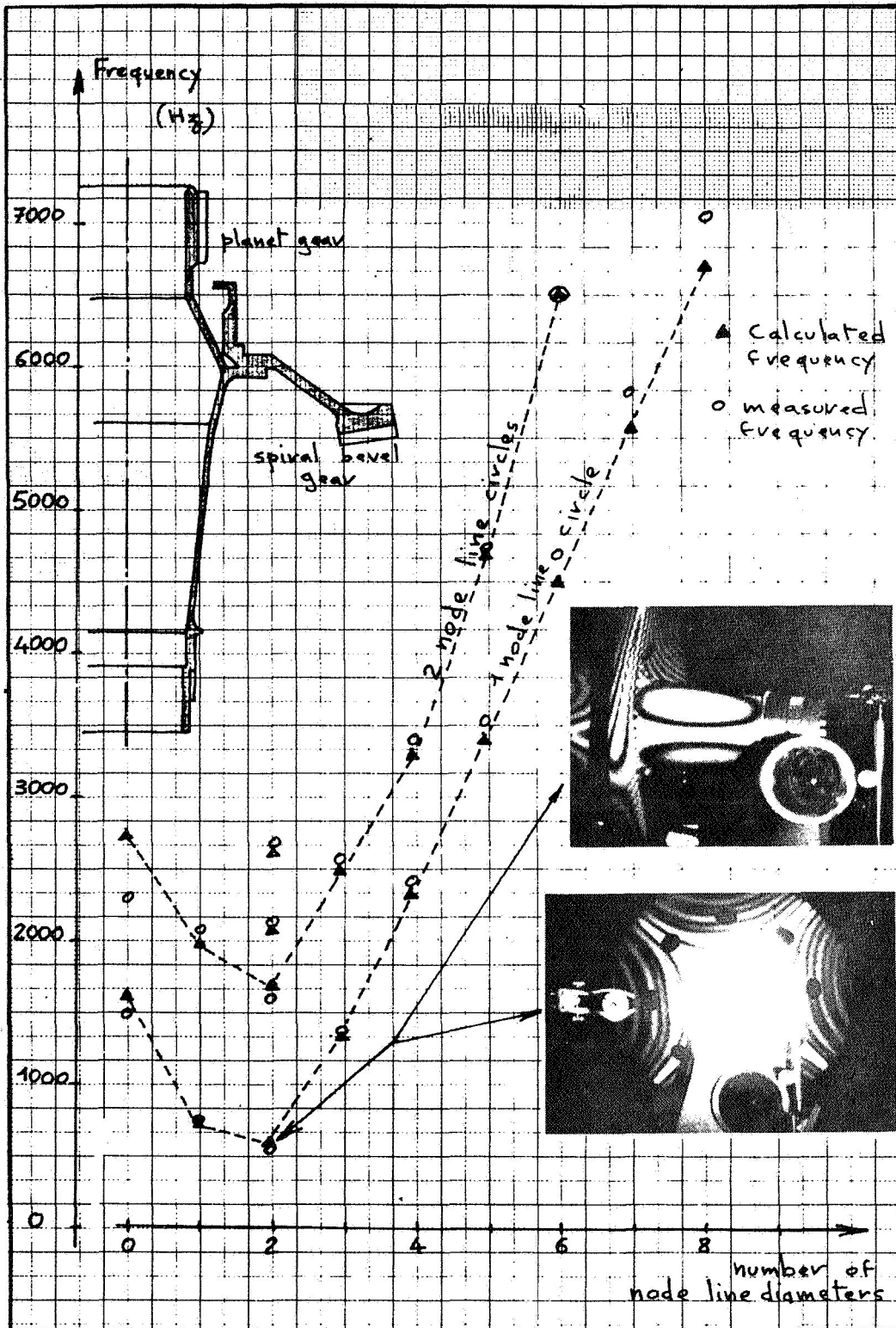Figure 9.— Angular meshing error measurements on a pair of pinion gears.

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Figure 10.- Comparison between calculated and measured natural frequencies
of a spiral bevel pinion shaft on the SA 365 helicopter main gear box.
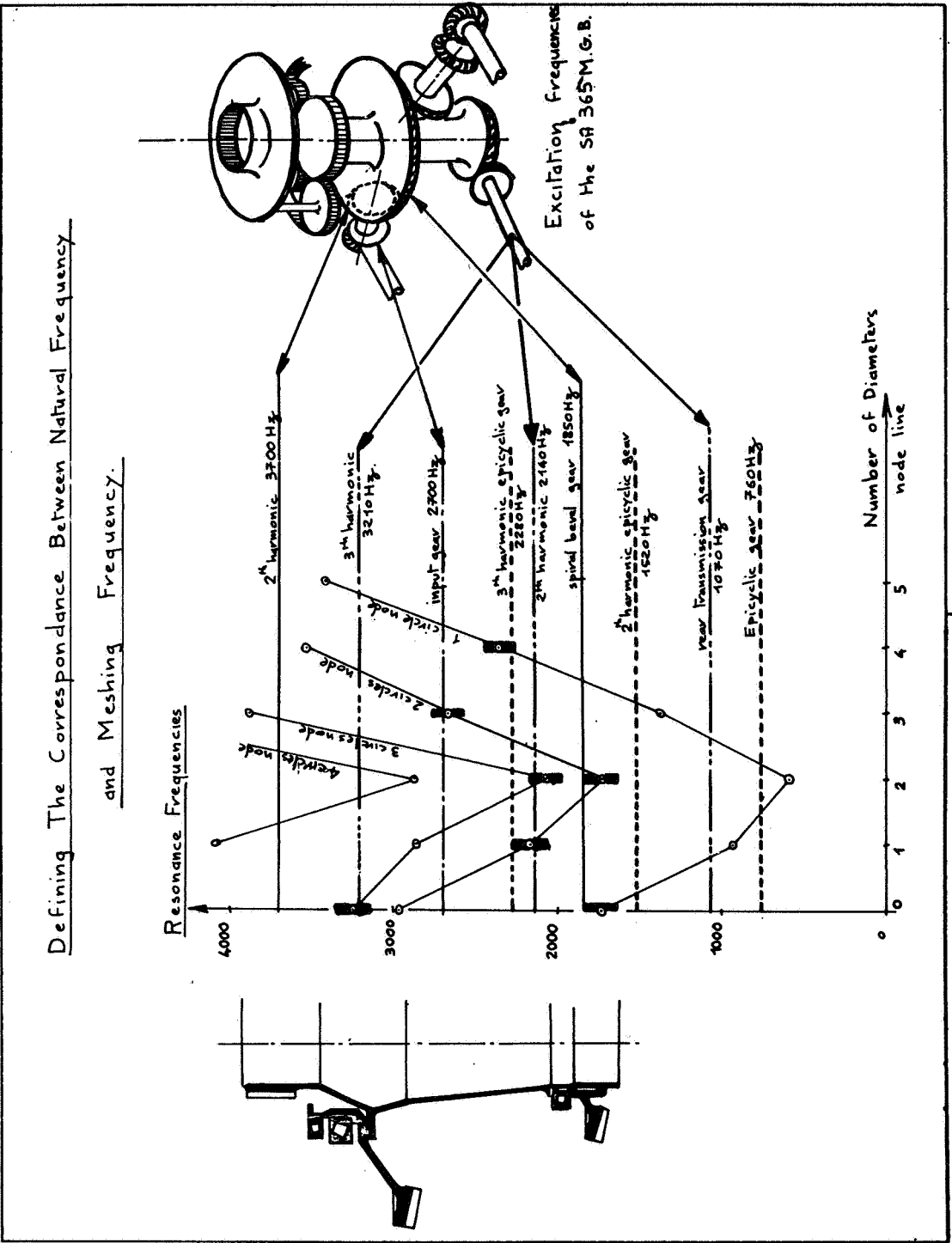
Figure 11.- Comparison of component natural frequencies with gear meshing
frequencies for the SA 365 helicopter main gear box.
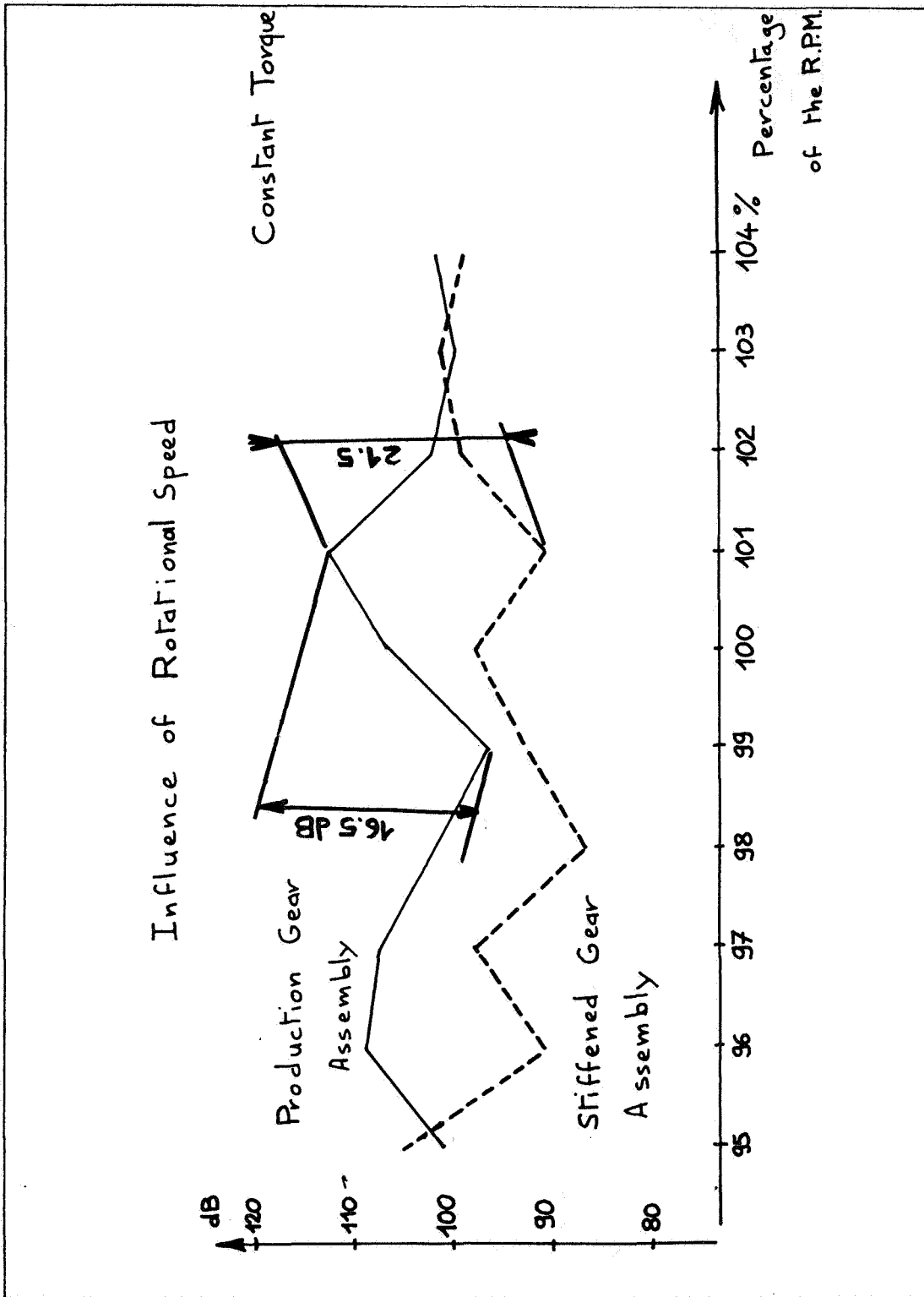
Figure 12.- Influence of stiffening of spiral bevel ring gear on the gear noise (2nd meshing harmonic).

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1- Prototype Gearbox
  Frequency 1792 Hz

2 - Modified Gearbox
  First Frequency 1850 Hz   Second Frequency 1729 Hz
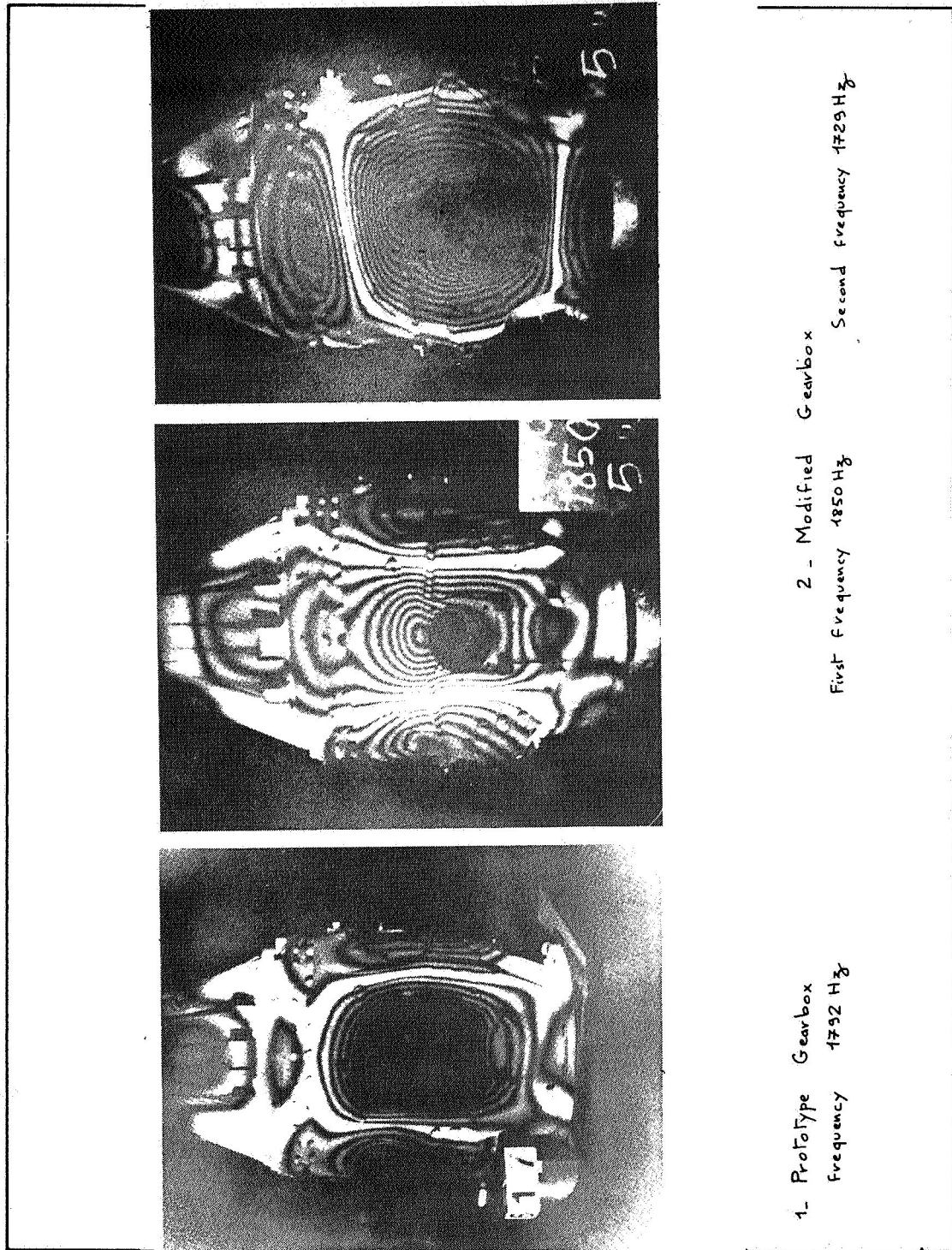
Figure 13.— Gear box housing mode shape representations from holographic measurements.

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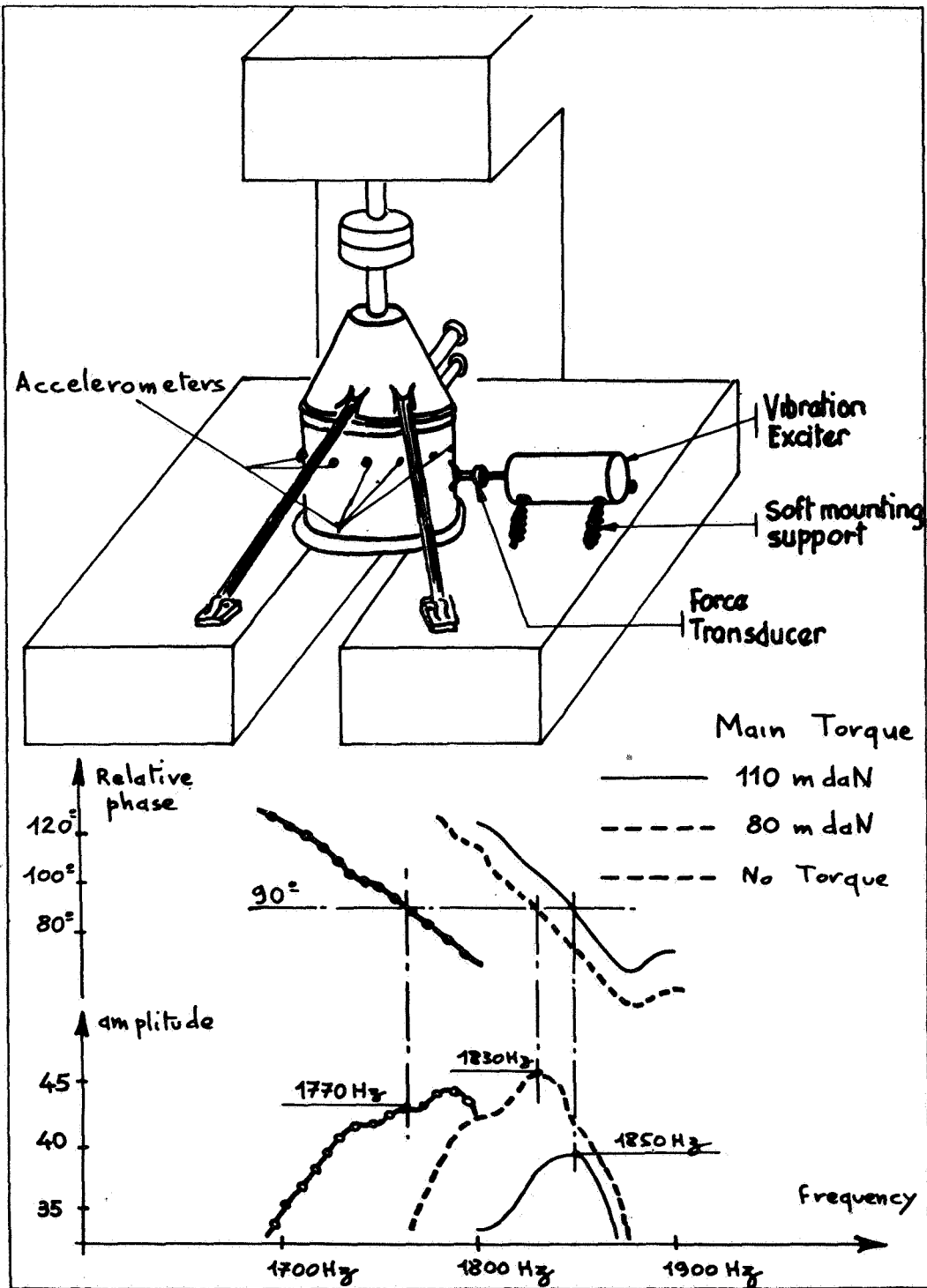
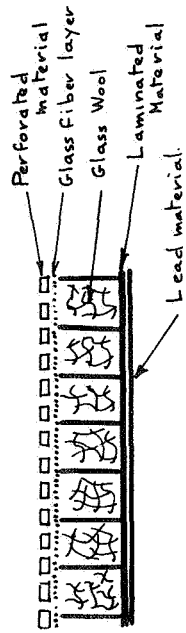Figure 14.- Effect of torque on the dynamic behaviour of the SA 360 helicopter main gear box housing.
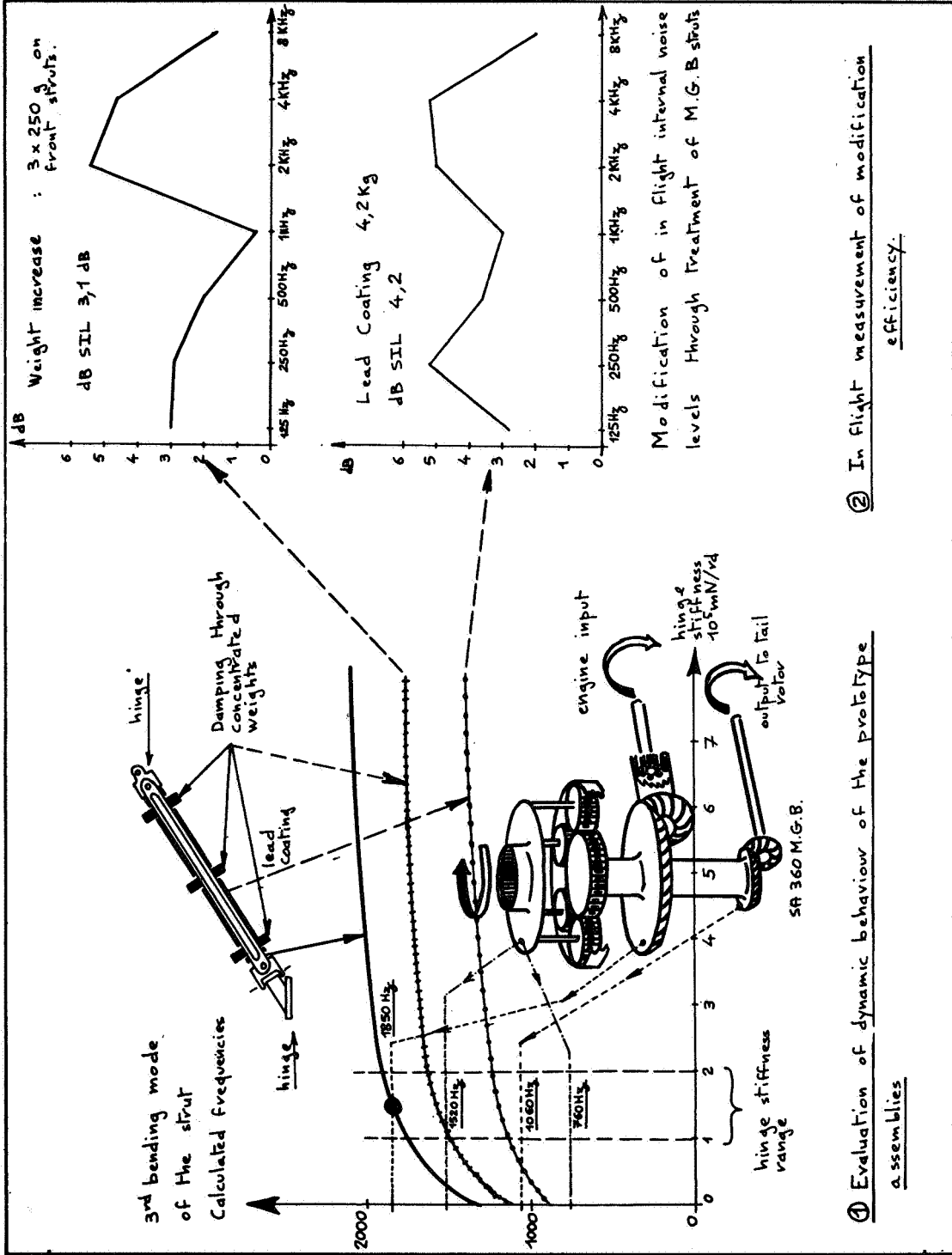
Figure 15.- Effect of dynamic behaviour of the SA 360 helicopter main gear box coupling elements on the internal noise level.
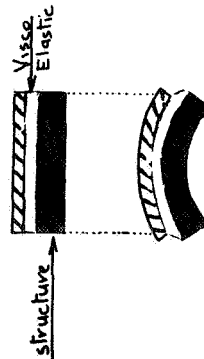
1_ Conventional Acoustic Treatment

Weight Law

$$\Delta_{dB} = 20 \log \frac{\omega \beta}{2\rho c}$$

Acoustic Screen

Graph: ΔdB axis — 60, 50, 40, 30, 20, 10

Curves: 8000 Hz, 4000 Hz, 2000 Hz, 1000 Hz, 500 Hz, 250 Hz, 125 Hz, 63 Hz

x-axis: 1 2 3 4 5 6 7 8 9 10 Kg/m², ρs

2 Treatment of Structures Through Visco-Elastic Materials.

- Utilisation of damping material

Shear Mode

Reduce vibration level of the structure through increased damping of bending motions

Visco Elastic

Structure

3 - Absorbing Panels Resonators.

Perforated material
Glass Fiber layer
Glass Wool
Laminated Material
Lead material.

Research on bending test specimens

Response Curves

before treatment

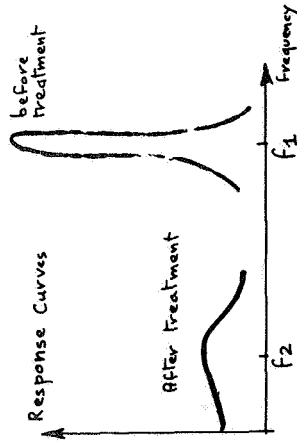After treatment

f_1 f_2 Frequency

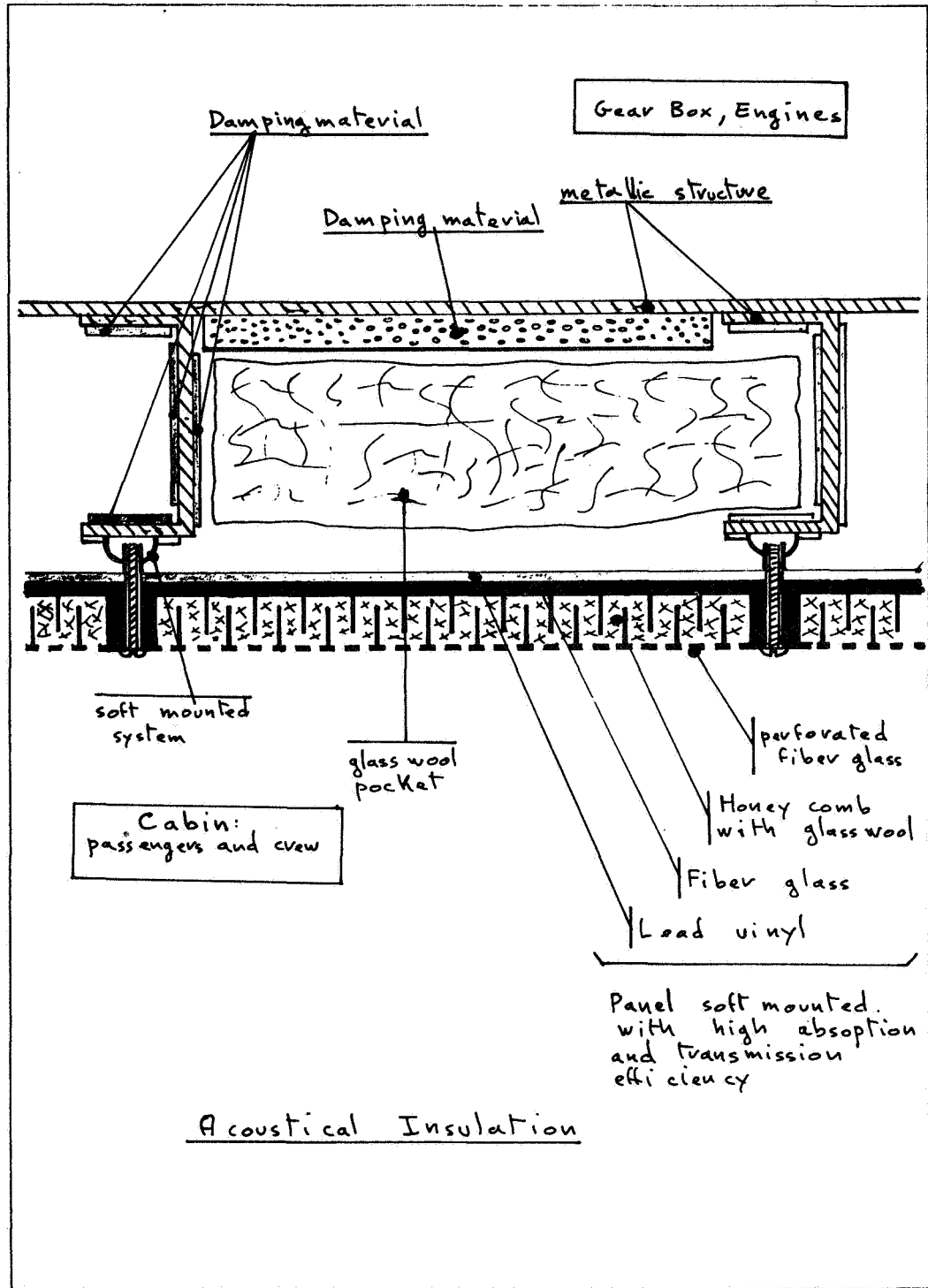Figure 16.- Sound treatment concepts for the control of internal noise.

Figure 17.- Sketch showing installation details of acoustic treatments in the SA 360-365 helicopter cabin structure.
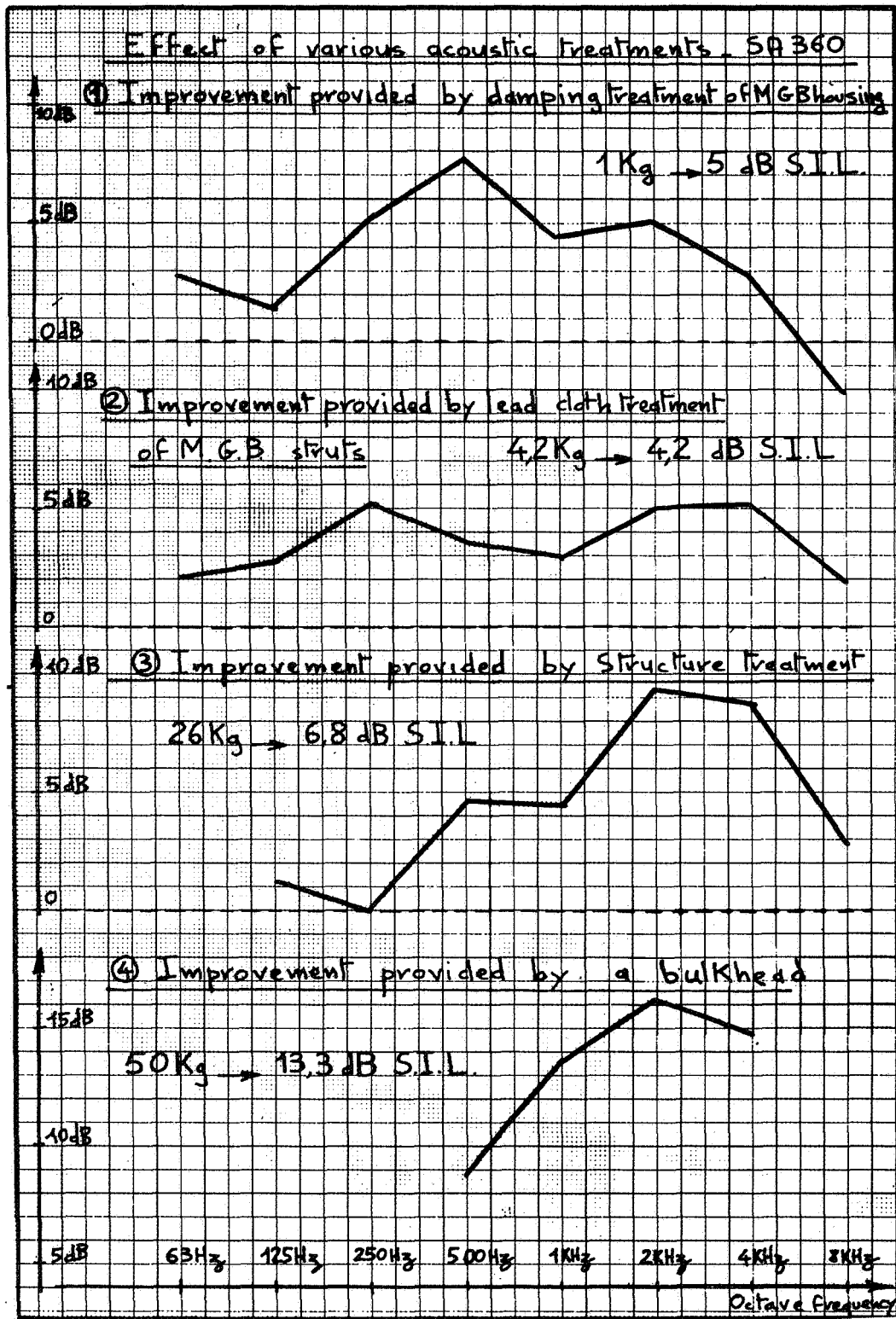
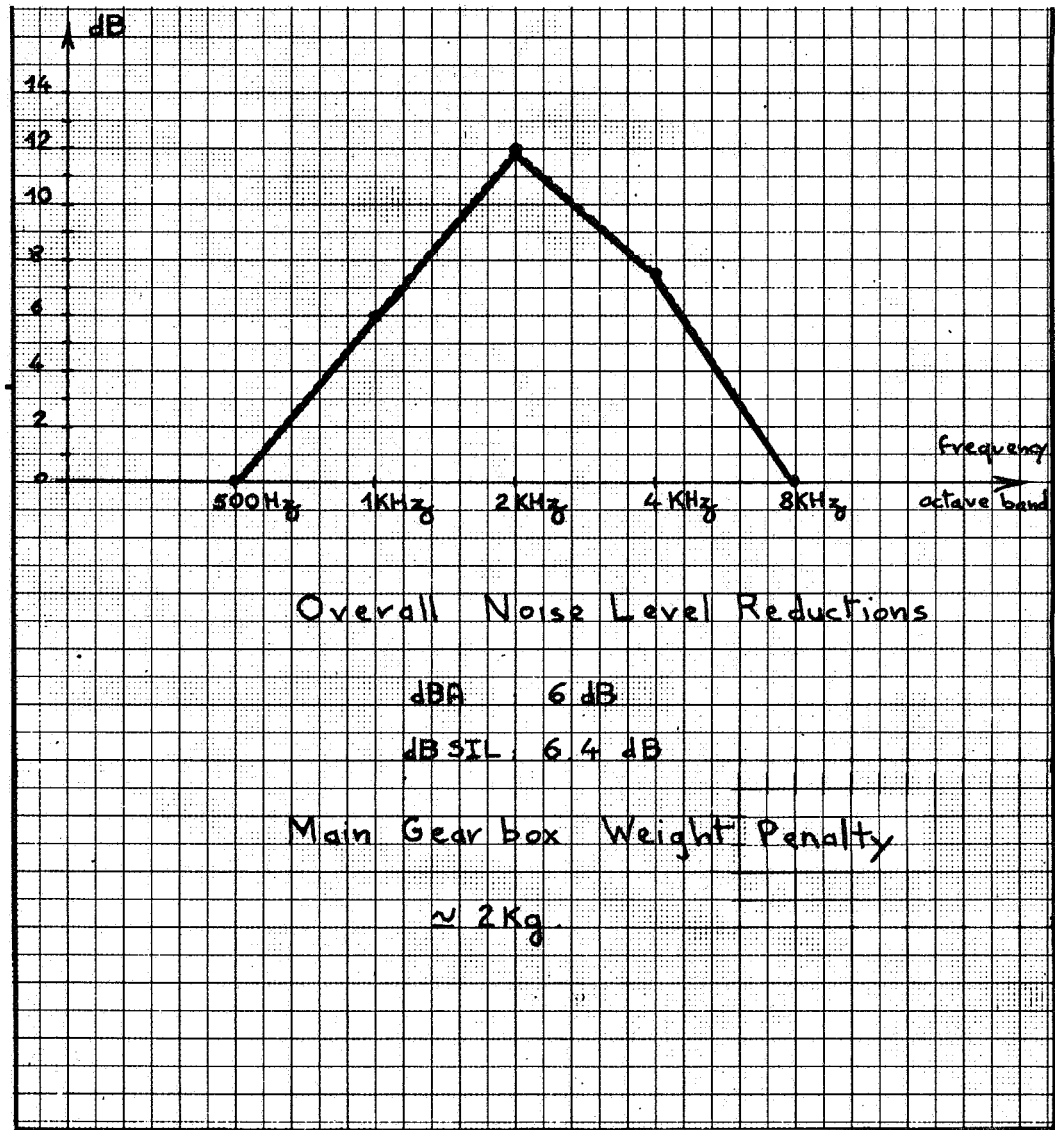Figure 18.- Acoustic gains and associated weight penalties for various cabin acoustic treatments.

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Figure 19.- Influence of spiral bevel gear tooth modifications on the SA 365
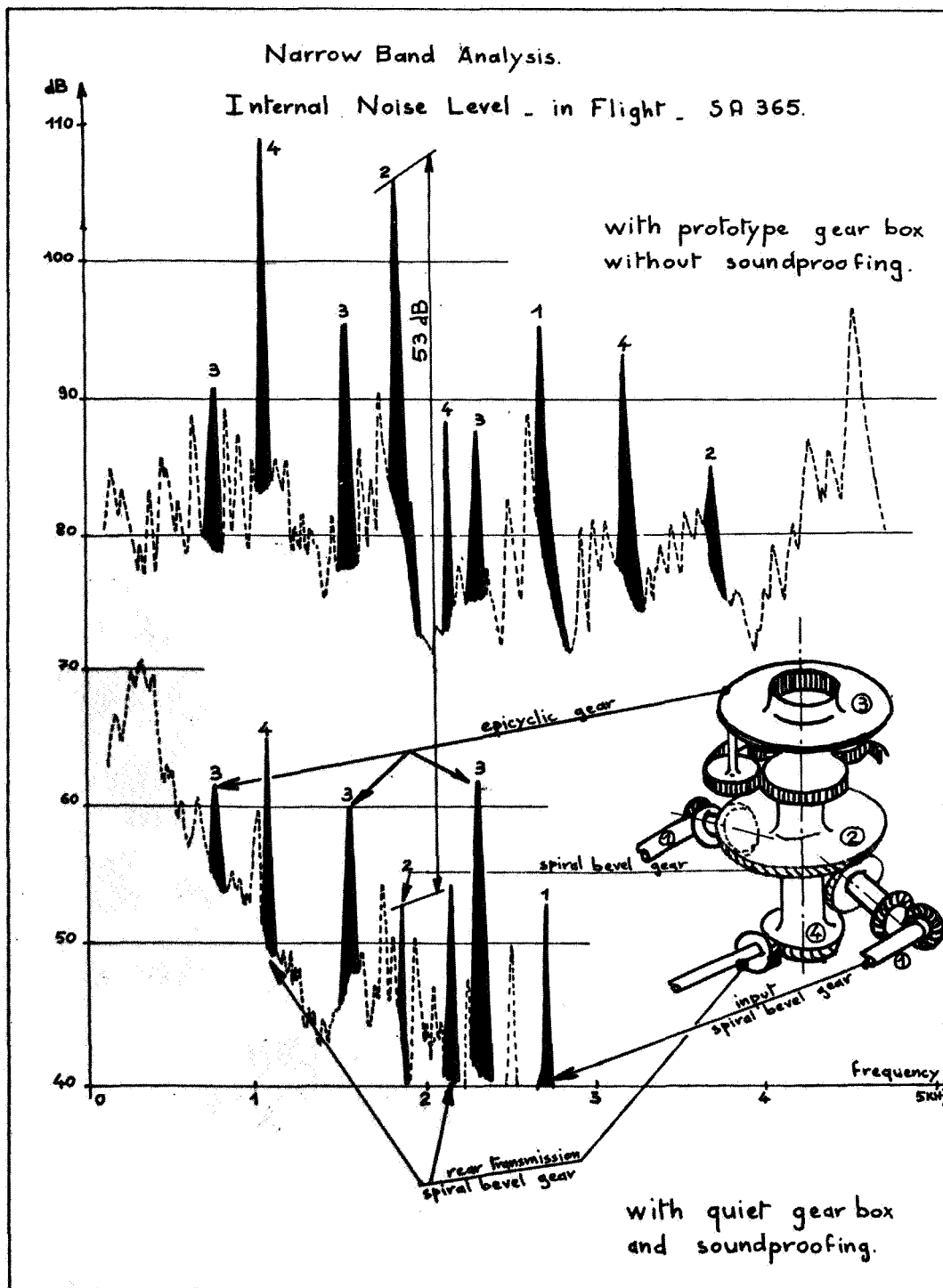internal noise level.

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Figure 20.- The effects of main gear box quieting on the measured internal
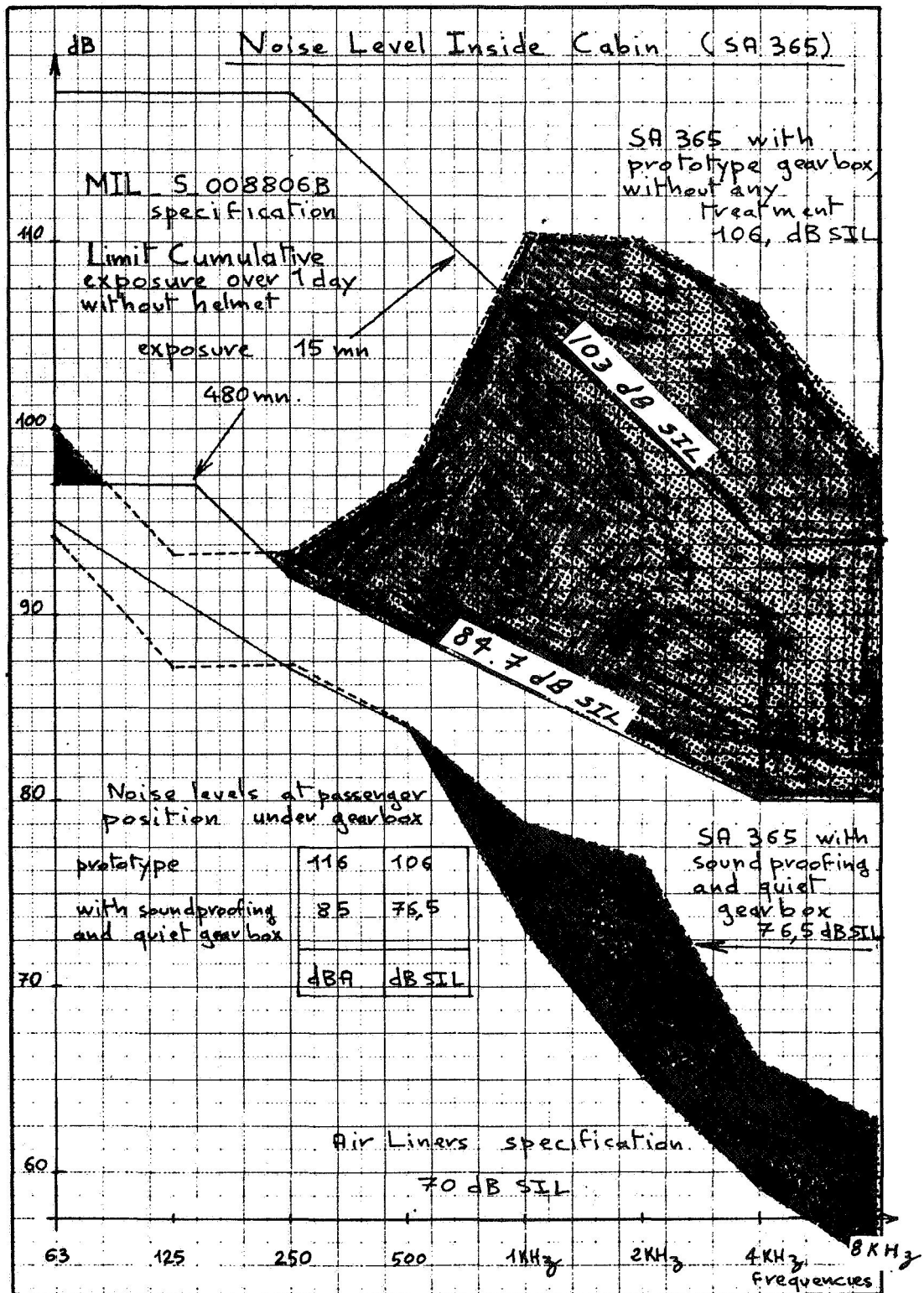noise levels of the SA 365 helicopter in flight.

Figure 21.— The effects of main gear box quieting plus cabin sound proofing
on the internal noise levels of the SA 365 helicopter in flight.