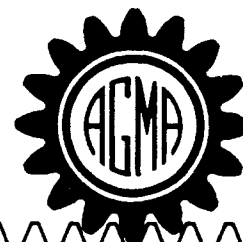


# Low Vibration Design on a Helical Gear Pair

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American Gear Manufacturers Association



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## TECHNICAL PAPER

# Low Vibration Design on a Helical Gear Pair

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[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

## Abstract

This paper shows a detailed knowledge of how to design the tooth-surface-modification to realize a quiet helical gear pair where face width is finite in length.

The performance of gears with bias-in and bias-out modification are discussed considering the effect of the shaft parallelism deviation.

To clarify the effect of the tooth deviation types on the vibration behavior of helical gear pairs, performance diagrams on vibration are introduced and created, in which the acceleration level of gear pairs are shown by contour lines on the contact ratio domain.

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# LOW VIBRATION DESIGN ON A HELICAL GEAR PAIR

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## 1. Introduction

### 1.1 The short history of the researchs on helical gear pairs :

The first paper titled as "Helical Gears" was reported by Harry Walker at 1946 July 12 [2]. In the paper, he wrote "new aspect of gear technique had been seen in the air craft industry, where spur reduction gears were now carrying loads that were not long ago unthought of. The author (H. Walker) believes that a similar technique was desirable for the more difficult subject of helical gears." He concluded that there was an increase in pressure at each end of contact corresponding to about three times the mean rate of loading.

H. Walker's paper at 1946 was the first report discussing on the load distribution along the contact line of a helical gear. Then the history of the researchs on the helical gear pairs is very short as only 50 years.

A. W. Davise [4] 1950, reported on the marine gear that the load distribution along the contact line were plus 60 per cent and minus 20 per cent of average load, totalling 80 per cent to the mean value at the both ends with the precise experiments by the tooth whose DP was 8/10, pressure angle was  $16^\circ$ , helix angle was  $30^\circ$ .

On the helical gear with narrow face width, E. Mönch and A. K. Roy [5] 1957, succeeded in observing directly and precisely the load distribution along the contact lines of the gear module was 10, gear teeth numbers were 25/25, pressure angle was  $20^\circ$ , helix angle was  $27.5^\circ$ , face width was 81mm, with the use of 3 dimensional photo-elasticity method. These measured results showed that the load distribution along a contact line shaped as like the back of an Arabian camel, that is, the distribution were gently convex and did not inclese at the both ends.

Also In 1957, M. D. Trobojevic [6] reported same results as Roy's results on the load distributions with the use of the rubber tooth whose module was 15mm, face width was 300mm.

In 1962, K. Hayashi [8] showed theoretically the load distribution along the contact line of a helical gear tooth results in the second kind integral equation of Fredholm-type. However the kernel of the integral equation means the deflections due to a concentrated load on a tooth with finite width, which should have been obtained as the solutions of the differential equation on bending deformation of a rack shaped cantilever plates shown by R.G.Olsson [1] in 1934. For marine gears, in 1950, T. J. Jaramillo [3] succeeded in solving the deflections due to a transverse concentrated load acting an arbitrary point of an infinitely long cantilever plate of constant width and thickness.

G. Niemann and T. Hösel [9] examined precisely and reported the relation between noise level and, overlap and transverse contact ration, in 1966.

In 1970 G. Niemann and J. Baethge [10] reported that the behaviors of the driven gear under loading could be substituted with the changing of the total length of contact lines of a helical gear pair in 1970.

D. R. Houser and A. G. Seireg [11], [12] examined the dynamic load formula for spur and helical gears with the use of four sets of specially designed gears, mounted with forty-one strain gages each, in 1970.

T. F. Conry and A. Seireg [13] presented a mathematical programming method for design of elastic bodies in contact in 1971. And they [14] reported a mathematical programming technique for the evaluation of load distribution and optimal modifications for parallel gear systems in 1973. However, in this study, they used Jaramillo's solution of an infinitely long cantilever plate and the mirror image law proposed by E.J. Wellauer and A. G. Seireg [7] 1960, to determine the deflections due to bending of a gear tooth being finite in length.

In the early 1970s, main frame computer and computer aided technology became available to all research field, of course to the researchs on helical gears, and lead the studies more precise.

## 1.2 Introduction of this paper :

It was fortunate for the author that the author could start and continue studying a helical gear pair in the same age when main frame computers were developed and evolved explosively.

From the viewpoint on the vibration performance, power transmission helical gear pairs with narrow face width, are classified theoretically into three categories over the contact ratio domain. And it has been verified with experiments. From the experimental results, at the same time the relation between the vibration magnitude and the shaft parallelism deviation have been investigated.

To clarify the effect of the tooth deviation types on the vibration behavior of helical gear pairs, performance diagrams on vibration are introduced, in which the acceleration level of gear pairs are shown by contour lines on the contact ratio domain.

Finally, the performance of gears with bias-in and bias-out modification are discussed considering the effect of the shaft parallelism deviation with use of the developed simulator on a helical gear unit. It becomes clear that there is an asymmetrical feature on the relation between the vibration magnitude of a gear pair and the direction of each deviation.

## 2. Three Categories of a Parallel Pair :

It has been confirmed theoretically and experimentally from the viewpoint on the vibration performance that power transmission helical gear pairs with comparatively narrow face-width can be classified into three categories over the proposed contact ratio domain whose abscissa is the transverse contact ratio and whose ordinate is the overlap contact ratio.

## 2.1 The Helix Angle and the Transmission Behaviors of a Driven Gear [19] :

The author solved numerically the deflections of a thick plate with finite width [15] and a rack shaped cantilever [16] under a concentrated load shown by R.G.Olsson[1] by using the finite difference method. Furthermore, the load distribution along the line of contact and the compliance of a helical gear tooth pair from the start of meshing to the end of meshing have been revealed [17]-[19].

Under the condition that the width of gear face is constant, i.e. three times the whole depth, the relation between the helix angle and the calculated behaviors of the driven gear under loading is shown in Fig.1. These results are analyzed under the unique condition that the normal pitch  $P_m$ , normalized with the whole depth, is 0.6. Then the whole contact ratio is calculated for each helix angle. And the overlap ratio is calculated from this whole contact ratio and the assumption that the transverse contact ratio is  $\epsilon_\alpha = 1.4$ . This contact ratio was calculated for a spur gear pair when the normal pitch  $P_m$  is 0.6.

When the helix angle is  $14^\circ$  in Fig.1, the sum of the transverse and overlap contact ratio is smaller than 2. So this pair of gears transmits load alternately with one pair and with two pairs of mating teeth. It is observed that the load sharing ratio for this pair of gears varies more smoothly than that of spur gears. But sometimes there occur some knobs in this curve when the meshing condition transits from one pair meshing to two pairs meshing.

When the helix angle is  $20^\circ$  or  $30^\circ$  as shown in Fig. 1-(iii) or (iv), the total contact ratio is over 2. So the gears transmit load alternately with two and three mating pairs of gear teeth. Under these conditions the load sharing ratio varies

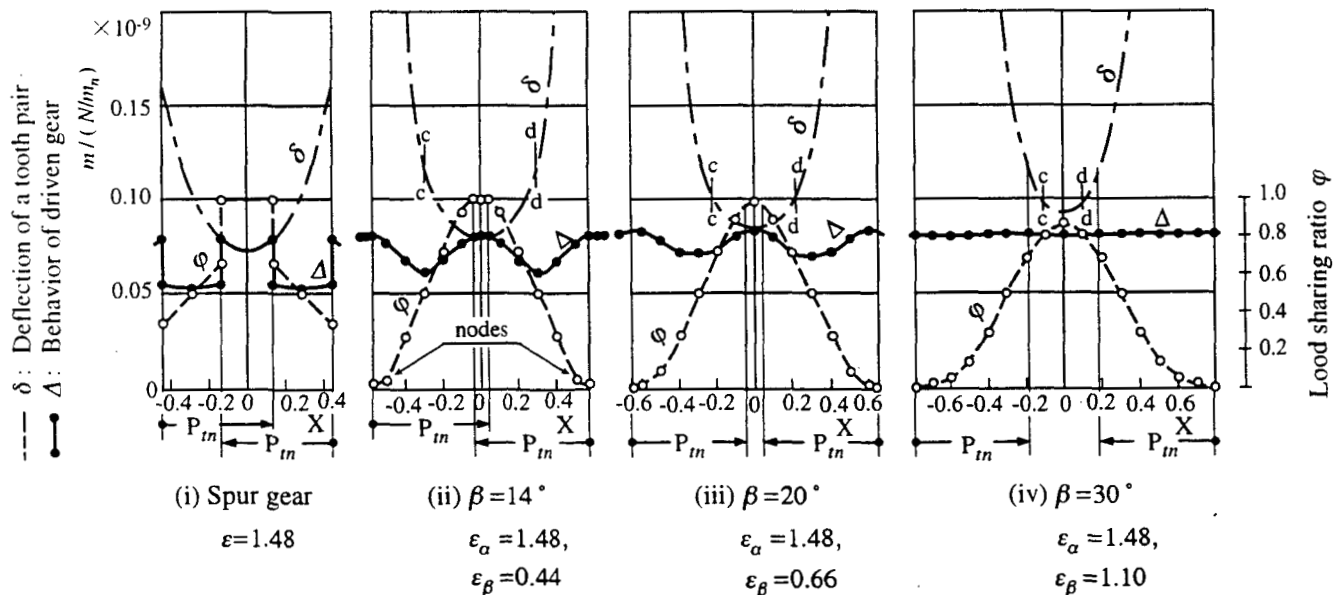


Fig.1 The behaviours of the driven gear and the helix angle with  $b/h=3.0$

smoothly. The behavior of the driven gear, or the transmission error, also varies smoothly.

Especially when the helix angle is 30°, the overlap contact ratio is over 1.0, the behavior changes very smoothly with little fluctuation as shown.

## 2.2 Three Categories of a parallel axes gear pair [20] :

The theoretical and experimental study on the static meshing behavior under loading has proved that a power transmission parallel gear pair can be classified into three categories on the contact ratio domain based on its facility for reducing the vibration as shown Fig.2 (refer to Fig.1). They are categorized as:

- K(I) Total contact ratio is less than 2.0.  
Poor performance.
- K(II) Total contact ratio is over 2.0 and overlap ratio is less than 1.0.  
The tooth modification is recommended.
- K(III) Overlap ratio is over 1.0.  
Good performance without tooth modification.

## 3. The Vibration Magnitude and the Shaft Parallelism[21] :

The proposed three classifications of a parallel gear were verified experimentally by dynamic meshing test under loading. At the same time the relation between the vibration and the parallelism of axes was investigated for three kinds of helical gear pairs classified into three categories. Two kinds of shaft misalignment were implemented in this study, i.e., the in-plane and the out-of-plane deviation. For realizing the out-of-plane or the in-plane parallelism, the pedestal of the driving gear shaft was tipped in the vertical plane or in the horizontal plane, respectively. The vibration was measured by two accelerometers attached directly to the driven gear blank surface.

### 3.1 Dimensions of test gears and test apparatus :

Test gear pairs are designed to belong to each category classified over the contact ratio domain, and are named H1, H2, H3 and H4 as shown in Fig.2. Dimensions of each gear pair are displayed in Table.1.

All test gears were hardened about Hrc55, and finished by the MAAG 30-BC Gear Grinder. Tooth profile and tooth traces are made with as little deviations as possible.

### 3.2 Shaft misalignment set up :

The realization of shaft misalignment in the actual experimental set up is shown in Fig.3. It was carried out by inserting several thickness gages either on the surface of the base plate or on its side surface for the out-of-plane or in-plane deviation, respectively. In this set up the thickness gages used were of 0.1mm-0.4mm and the angular deviations realized were about  $0.5 \times 10^{-3}$  rad and  $1.1 \times 10^{-3}$  rad. The amount of

the out-of-plane and the in-plane deviation were measured with two dial indicators.

Each gear shaft misalignment was introduced for not only the leading side bearing but also the trailing side bearing.

## 3.3 Influence of the out-of-plane deviation :

The relation between the rotational vibration response vs. speed and the out-of-plane deviation is shown in Fig.4 for the gear pair H3. In the case of the proper alignment condition

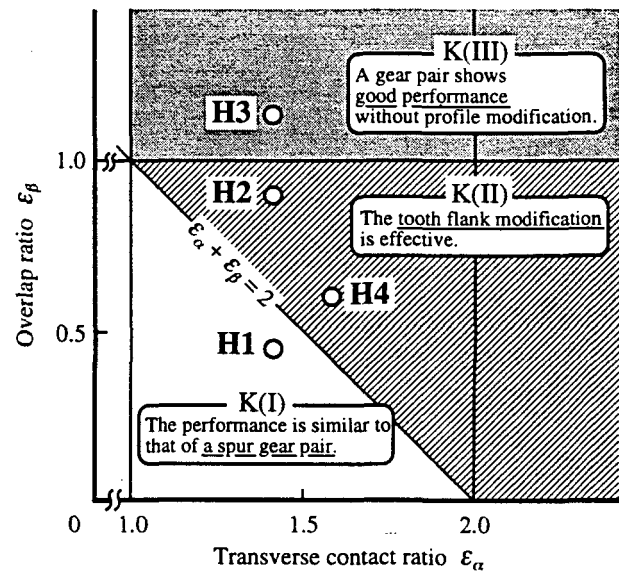


Fig.2 Classification of parallel pairs

Table 1 Dimensions of test gear pairs

| Gear pair                         | H1    | H2   | H3   | H4   |
|-----------------------------------|-------|------|------|------|
| Normal module                     | 3.5   |      |      | 4    |
| Number of teeth                   | 30    |      |      | 29   |
| Helix angle [deg]                 | 30    |      |      | 15   |
| Pressure angle [deg]              | 20    |      |      |      |
| Face width [mm]                   | 10    | 20   | 25   | 28   |
| Addendum modification coefficient | -0.17 |      |      | 0    |
| Transverse contact ratio          | 1.4   |      |      | 1.57 |
| Overlap contact ratio             | 0.45  | 0.91 | 1.14 | 0.58 |

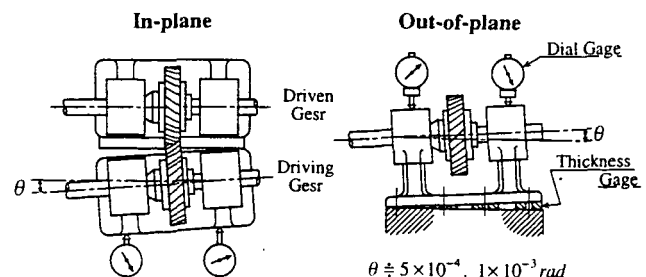


Fig.3 Shaft misalignment setup