Vol 7 (2007) 1-9

YEMEN ENGINEER
Faculty of Engineering-University of Aden

1 Experimental Study of Misalignment Effect on Vibration of Helical Gears

Moh'd Aqeel Alattas*

Abstract

A helical gear pair can transmit power with low level vibration and noise as compared with a spur gear. Many studies have been performed on the vibration and noise problem of the spur gear pair, but there are few studies on helical gear pair, especially one with a narrow facewidth. This paper presents an experimental study on the relation between the vibration and parallelism of axis for three kinds of helical gear pairs. Results obtained have made clear that the deviation exerts an influence on the characteristics of gear vibration more strongly than the inclination, and that the deviation as well as the inclination which leads to the leading side bearing exerts an influence on those more strongly than which leads to the trailing side bearing.

Keywords: Misalignment, Vibration, Helical Gears.

الملخص

تمتاز التروس (المسننات) الحلزونية (Helical gears) بقدرتها على نقل الطاقة باهتزاز وإزعاج ذو مستوى أقل مقارنة بالتروس العدلة (Spur gears). لقد أجريت العديد من الدراسات لتحديد مستوى الاهتزازات والإزعاج على المسننات العدلة، بينما هناك القليل من تلك الدراسات على المسننات الحلزونية وخصوصاً تلك ذات أوجه قليلة العرض.

يقدم هذا البحث دراسة تطبيقية (مختبريه) لتحديد العلاقة بين مستوى الاهتزازات وتوازي المحاور لثلاثة أنواع من المسننات الحلزونية المزدوجة.

لقد أظهرت النتائج بوضوح تأثير طبيعة ومستوى الاهتزازات للمسننات بمدى الانحرافات أكثر من تأثرها بمدى الميلان. وأيضا كلاً من الانحرافات والميلان اللذان يؤديان إلى صله جانبيه أمامية (leading side bearing) لها تأثير أكبر من تلك التي لها صلة جانبية متأخرة (trailing side bearing).

1. INTRODUCTION

Recently, the more quiet gear pair is required for the automobile, the machine tool and so forth for all industrial installation. It is well known that a helical gear pair can transmit power with the lower level vibration and the more silent sound as compared with a spur gear pair [Randall, 1982 and Alattas, 1994].

Many studies have been performed about the vibration of a spur gear pair. But few studies have been performed on the vibration and noise problem of the helical gear pair, especially a pair with the comparatively narrow facewidth. This reason bases on the difficulty to analyze the meshing stiffness of the helical gear pair, because the contact line of the helical gear pair inclines to the axes [Umezawa, 1986; Walsh, 1992; Peng and Kerssissogolou, 2003 and VanDyck and Cotterill, 2005].

Authors made clear about the behaviours of the meshing stiffness of a helical gear pair

* Mechanical Engineering Department, Faculty of Engineering, University of Aden, Yemen

with comparatively narrow facewidth theoretically and experimentally, after theoretical analysis on the load distribution along the contact line changing its length through meshing [Umezawa, 1974; Droskack and Houser, 1977; Stewart, 1980 and Özguven and Houser, 1988]. They have clarified that the helical gear pair with comparatively narrow facewidth could be classified into three categories in relation to the transverse contact ratio and the overlap ratio comparing the performance of the vibration of the pair, and that the tooth flank modification should be effective for decreasing the vibration in the category, where the total contact ratio is over 2.0 and the overlap ratio is less than 1.0.

Literature has investigated the influence of the dimensions of gear and the profile error on the root stress and dynamic load. And it has been reported that the tooth end bearing enlarged the dynamic load of the gear pair with the concave tooth profile [Droskack and Houser, 1977]. Also, it has been investigated the influence of the tooth end bearing on the root stress and noise for helical gears used for the train. Moreover, it has been reported that the tooth end bearing changed the maximum root stress distribution along the tooth trace and enlarged the noise [Umezawa, 1986].

Now it has not been made clear how each kind of the error affect the vibration of the helical gear pair. And there are various uncleared relation between the amount of the error and the vibration behaviour. Therefore, the study about the relation between the error and the vibration is necessary for decreasing the vibration and the noise of a gear unit.

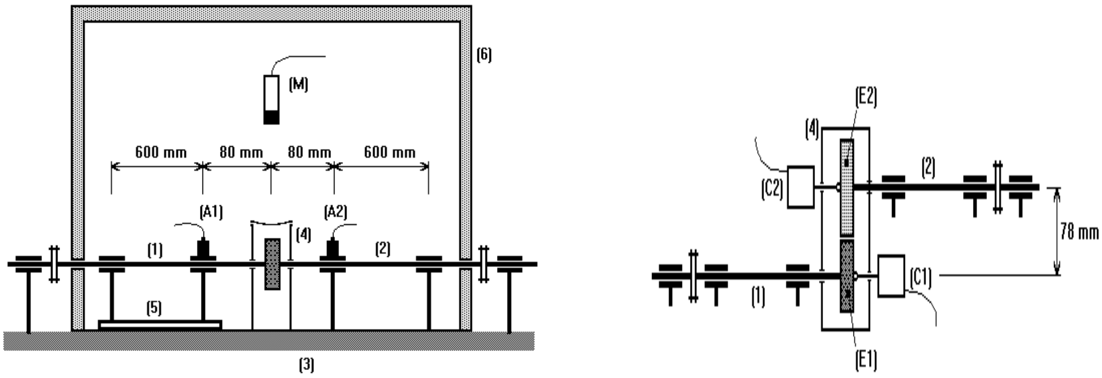
In this paper, the relation between the vibration and the parallelism of axes has been experimentally investigated for three kinds of helical gear pairs classified into three categories. And for realizing the deviation or inclination the pedestal of driving gear shaft was tipped in the vertical plane or in the horizontal respectively. The vibration was measured by two piezoelectric type accelerometers.

It is made clear that the deviation exerts an influence on the characteristics of gear vibration more strongly than the inclination, and that the deviation as well as the inclination which leads to the leading side bearing exerts an influence on those more strongly than which leads to the trailing side bearing.

2. EXPERIMENTAL APPARATUS

3.1 Test Rig

The test rig is a gear simulator constituted of two shafts (1, 2) as shown in Fig. (1), each of 60 mm diameter and mounted on two bearings. Tested gears (E1 and E2) are clamped with nuts at the operating end of each shaft and centered by involute splines so as to limit variations of eccentricity. After gear clamping, a special nose is fixed at each shaft end, in order to measure angular positions during motion with optical encoders (C1- C2). The shaft (1) is fixed on a special mounting, made of thick intermediate plates. It permits to impose small misalignments between the two shaft axes. Test gears were lubricated by an oil jet of the well-known mobile oil II. The input shaft (2) is driven by a 120 kW DC motor. The output shaft (1) is braked by a DC motor. Rotating speed is varying between 0 and 10000 rpm and is feedback controlled. The input torque varies independently from 0 to 150 Nm. The active part of the apparatus is fixed on a 7 tons rigid frame (3) made of steel and concrete. An isolating drum (6) permits to decrease the effect of the environment noise [Alattas, 1994].



(M) microphone,(A1) and (A2) accelerometers, (C1) and (C2) optical encoders, (E1) and (E2) gears, (1) and (2) shafts, (3) rigid frame, (4) oil box, (5) misalignment plate, (6) isolating case

Fig. 1: Schematic arrangement of the test gear apparatus.

2.1 Test gears

Dimensions of test gear pairs are shown in Table 1. Each test gear pairs is designed to belong to each of the categories classified by the transverse contact ratio and the overlap ratio, and is named H1, H2 and H3 . The gear pair HI belongs to the category where the performance of the vibration is similar to that of a spur gear pair(s). That is, the profile of the pair should be modified around tip area of both driving and driven gear. The gear pair H2 belongs to the category where the rotational vibration of the gear pair can be decreased under wide load condition with the tooth flank modification which relieves only the leading side tip area of driven gear. The gear pair H3 shows good performance in the static meshing test under loading and dynamic test without profile modification.

All test gears are hardened about Hrc55, and finished by the MAAG 30-BC gear grinder. Tooth profile and tooth trace of those are made with less error as far as possible and shown in Fig. 2.

Table 1: Dimensions of test gear pair

Gear pair	H1	H2	H3
Facewidth	10 mm	20 mm	25 mm
Number of teeth	30		
Normal module	3.5		
Normal pressure angle	20^0		
Helix angle	30^0		
Reference diameter	121.2 mm		
Addendum modification coefficient	-0.172		
Transverse contact ratio	1.40		
Overlap ratio	0.45	0.91	1.14
Total contact ratio	1.85	2.31	2.54

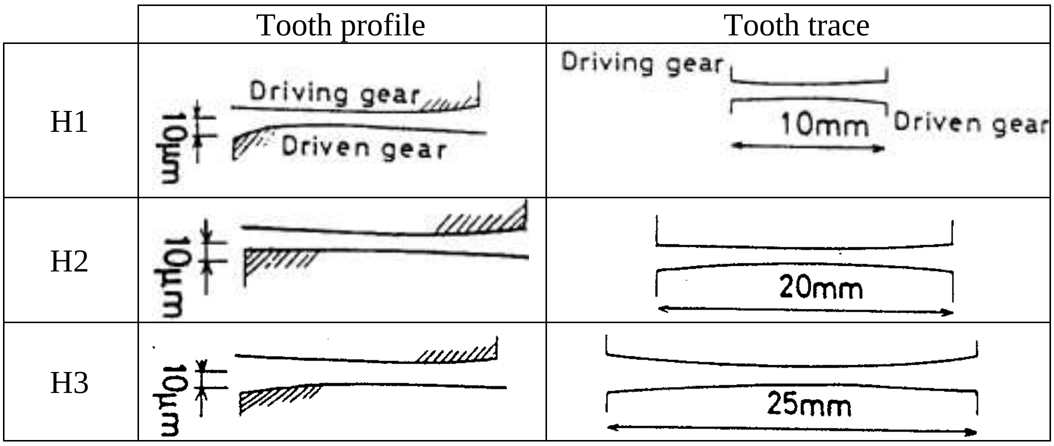


Fig. 2: Tooth profile and traces of test gear pairs

2.3 Measuring system

The block diagram of measuring system of rotational vibration is shown in Fig. 3. The measured signals from two piezoelectric type accelerometers (A1 and A2) are sent to a differential amplifier to remove the radial component through a slip ring and a low pass filter (cut off frequency of 10 kHz). Two types of data are obtained from these signal measurement systems, i.e., the waveform and the root mean square value (RMS value) which indicates the vibration level. When the waveform is measured, signals are recorded by a data recorder at the fixed rotational speed under constant torque and duplicated on a pen recorder after speed reduction.

When the RMS value is measured, the output of differential amplifier is measured by a digital multimeter and its digital output signals are processed by a mini computer. The results (RMS value) are presented on a X-Y plotter. Meanwhile the rotational speed of the output shaft varies from 500 rpm to 3400 rpm with constant speed increasing rate (10 rpm/sec) under the constant torque.

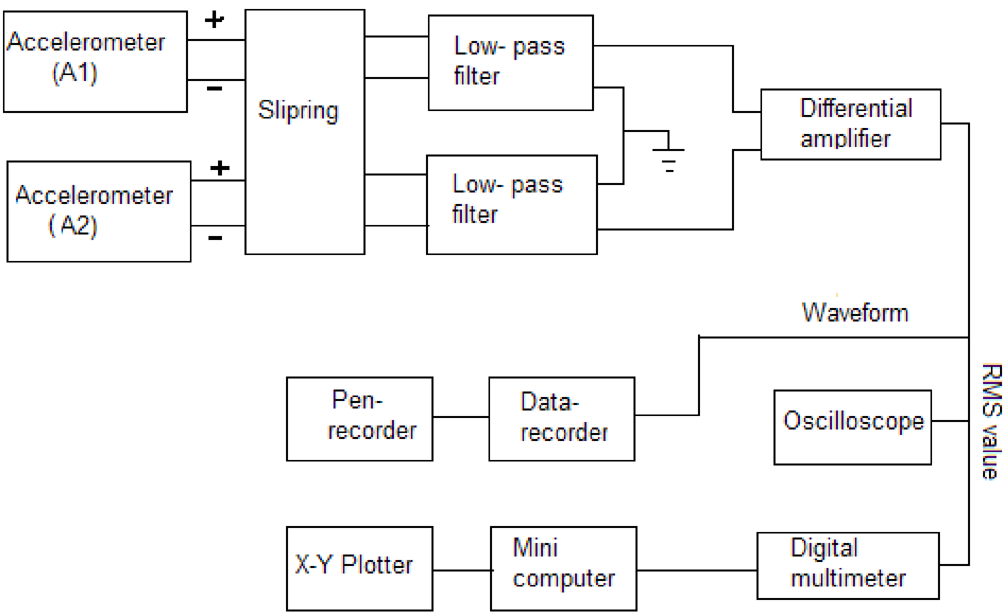


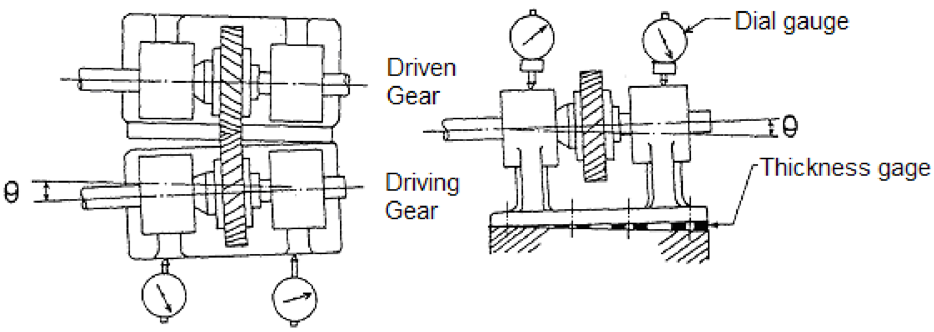
Fig. 3: Block diagram of measuring system

2.4 Shaft Misalignment Set-up

Two kinds of shaft misalignment are implemented in this study, i.e., the shaft inclination (Fig. 4a) and the shaft deviation (Fig. 4b).

The realization of shaft misalignments in the actual experimental set up is shown in Fig. 4. It is carried out by inserting several thickness gages either on the surface of the base plate or on its side surface for the deviation or the inclination, respectively. In this set up the thickness gage used are of 0.1mm- 0.4 mm and the angular errors of about 0.5 x 10^{-3} rad and 1.1x $l0^{-3}$ rad are set. The amount of shaft deviation and shaft inclination are measured with two dial indicators.

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



(a) Inclination (b) Deviation

Fig. 4: Shaft misalignment setup

3. EXPERIMENTAL RESULTS

3.1 Influence of the Shaft Deviation

Experiments have been performed to clarify the influence of the shaft deviation and the shaft inclination on the vibration of the three kinds of helical gear pair.

The relation between the rotational vibration and the rotational speed is shown in Fig. 5 for the gear pair H1. The error which leads to the leading side bearing is represented with the positive sign, and one leads to the trailing side bearing is represented with the negative sign in the figures shown in this paper.

In the case of proper shaft alignment where the full facewidth contact is realized (as shown by the solid line), the acceleration level becomes high as the rotational speed increases. Two peaks are observed at about l600 rpm and 2400 rpm, which are ascribed to the higher harmonic resonance of the meshing resonance frequency by the higher harmonic of tooth meshing frequency, i.e., the third and the second harmonic resonance. It is observed that the positions of peaks shift slightly toward lower rotational speed (lower tooth meshing frequency) by both plus and minus shaft deviation whose values are about l0 μm. This phenomenon indicates that the meshing resonance frequency is lowered by the error.

There is not any other influence of the shaft deviation in the experimental data on this gear pair. Because the arranged error is comparatively small in comparison with the other gear pairs by being narrow facewidth. And the vibration level of this gear pair is higher than the other gear pairs by about 2 or 3 times.

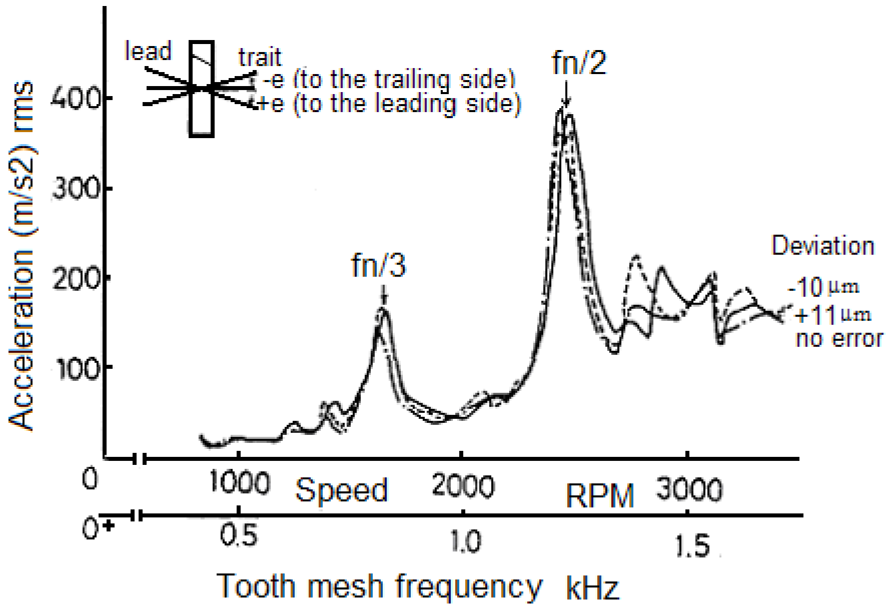


Fig. 5: Influence of deviation on vibration of gear pair H1 (Torque = 150 Nm)

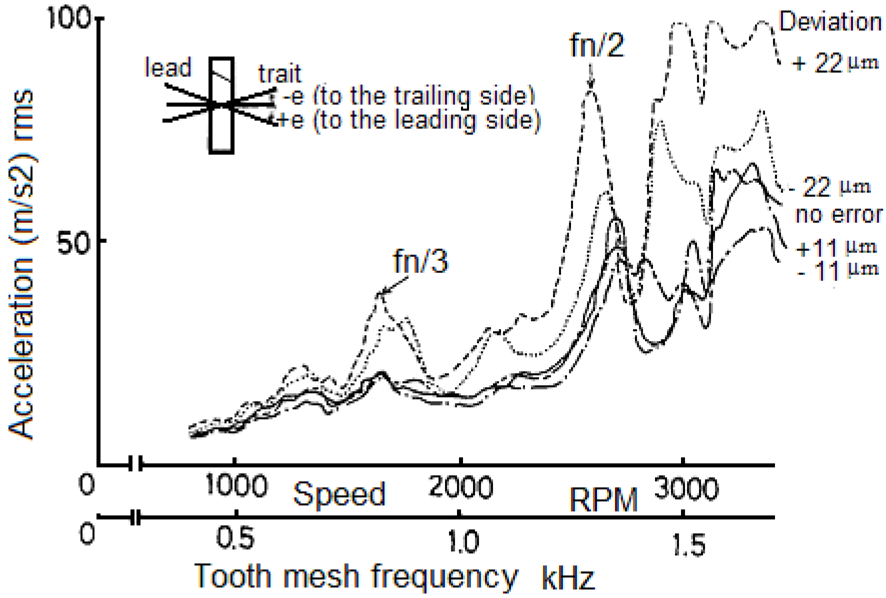


Fig. 6: Influence of deviation on vibration of gear pair H2 (Torque = 150 Nm)

The relation between the rotational vibration level and the rotational speed is shown in Fig. 6 by the parameter of the shaft deviation for the gear pair H2.

In the case of the proper condition (as shown by the solid line) marked as no error in the figure, the acceleration becomes large with increasing the rotational speed. Two peaks are observed at about 1700 rpm and 2600 rpm, which are ascribed to the higher harmonic resonance. And peaks are observed at about 2900rpm and 3300 rpm, which are ascribed to the resonance of the test apparatus.

When the error of 11µm exists at the leading side of meshing shown with plus sign, the acceleration becomes large on the whole rotational speed. As the error increases to 22 µm (as shown by the broken line), the acceleration becomes much larger.

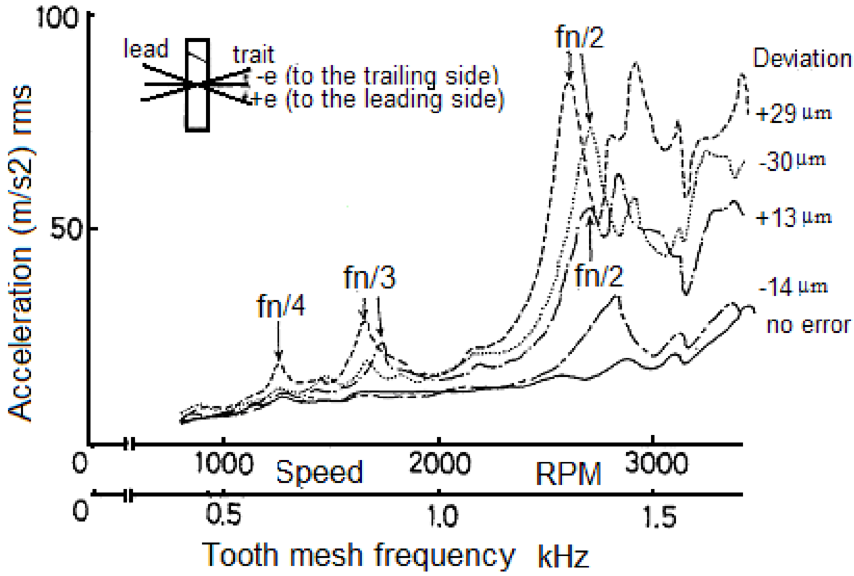


Fig. 7: Influence of deviation on vibration of gear pair H3 (Torque = 150 Nm)

Especially this tendency is remarkable in the region of peaks. On the other hand, when the error of 11µm exists at the trailing side shown with the minus sign, the acceleration level does not change in comparison with the proper condition. However, as the error increases to 22 µm, the acceleration level becomes high, especially in the region of peaks as well as the error at the leading side.

The peak at about 2500 rpm shifts toward lower rotational speed, which is ascribed to the second harmonic resonance, as the error increases.

The relation between the rotational vibration and the rotational speed is shown in Fig. 7 for the gear pair H3. In the case of the proper alignment condition, the acceleration increases monotonously as the rotational speed increases. Observed in the cases of the gear pair HI and H2, the peak can not be recognized for this gear pair.

When the error of 13 µm exists at leading side, the peak appears at about 2700 rpm, which are ascribed to the second harmonic resonance. As the error increases to 29 µm, the acceleration level becomes higher on the whole rotational speed in the experiment and the peak shifts toward lower rotational speed. On the other hand, when the error of 14µm exists at the trailing side, there is not the remarkable variation of the acceleration level. When the error increases to 30 µm, however, the peaks appear which are caused by the higher harmonic resonance, and the acceleration level becomes high.

From these facts, it can be concluded that the deviation which leads to the leading side bearing influences the vibration of a gear pair more strongly than which leads to the trailing side bearing.

3.2 Influence of the shaft inclination

The influence of the shaft inclination on the vibration can not be remarkably recognized for the gear pair Hl, by the same reason with the case of shaft deviation.

The relation between the rotational vibration and rotational speed is shown in Fig. 8 by the parameter of the shaft inclination for the gear pair H2.

When the error of l2 µm exists either at the leading side or at the trailing side, the increase in the vibration level is little. As the error increases to 23 µm at the leading

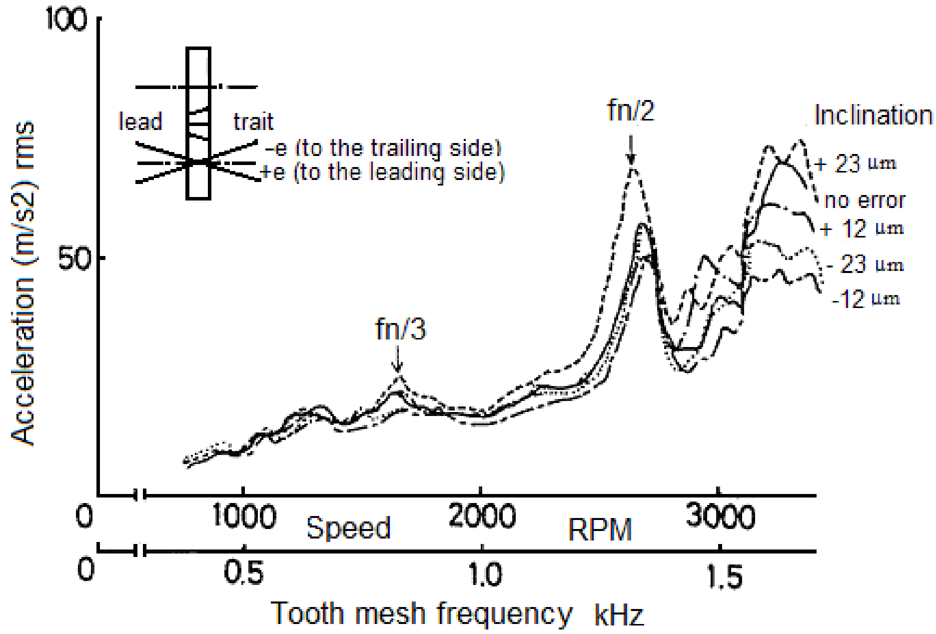


Fig. 8: Influence of inclination on vibration of gear pair H2 (Torque = 150 Nm)

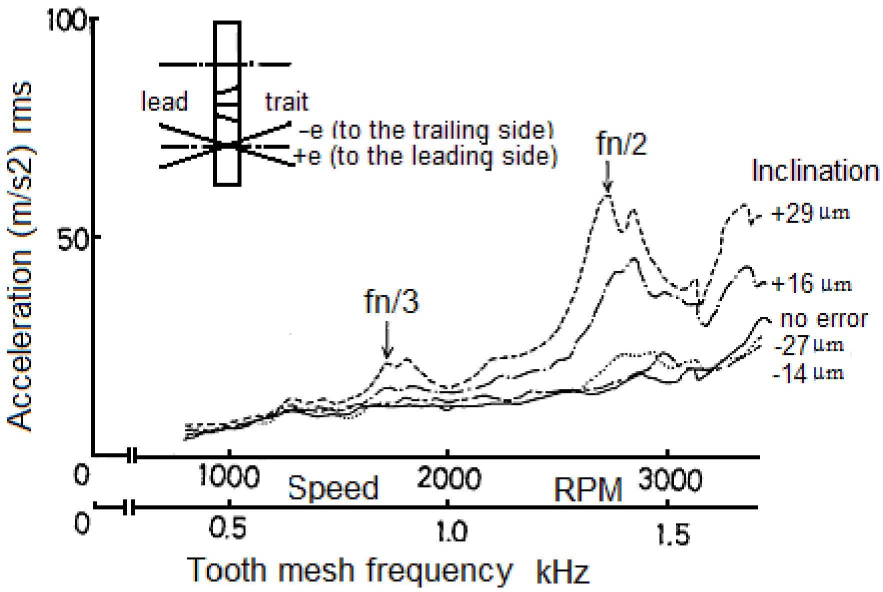


Fig. 9**:** Influence of inclination on vibration of gear pair H3 (Torque = 150 Nm)

side, however, the acceleration becomes large on the whole rotational speed, and the peak shifts toward lower rotational speed. For the gear pair H3, the relation between the rotational vibration and rotational speed is shown in Fig. 9. In the case of the error existence at the trailing side, the vibration of this gear pair is not influenced by the error. On the other hand, it being at the leading side, the acceleration level becomes high, and peaks appear, which are ascribed to the higher harmonic resonance. However, the vibration behaviour of the pair is not so influenced in comparison with the shaft deviation.

4. CONCLUSION

Misalignment is a common cause of machinery malfunction. A poor aligned machine can cost any industrial installation in machine down time, replacement parts, inventory, and energy consumption. A large payback is often seen by regularly aligning machinery. Operating life is extended and process conditions are optimized.

In this study the influence of the parallelism on the vibration has been experimentally investigated for the power transmission helical gear pair with comparatively narrow facewidth. And following results are obtained:

(1) The shaft deviation and the shaft inclination can be approximately considered as the lead error on the plane of action of helical gear pair.

(2) The shaft deviation exerts an influence on the vibration of a pair more strongly than the shaft inclination.

(3) The shaft deviation as well as the shaft inclination which leads to the leading side bearing exerts an influence on the vibration of a pair more strongly than which leads to the trailing side bearing.

(4) The meshing resonance frequency of a pair becomes lower as the misalignment increases.

ACKNOWLEDGEMENTS

The author would like to thank P. Soleilhac, assistant engineer in CASM Laboratory of INSA –Lyon -France, for his efficient help during the entire experimental program.

References

Alattas, M., (1994). Maintenance des machines tournantes: signature de defauts d'engrenage droits et helicoidaux, Ph D thesis, I.N.S.A – Lyon- France.

Alattas, M., J. Mahfoudh, D.Remond and D. Play, (1994). Experimental study of faults influence on vibration and noise measurements in gear transmission mechanism, Gearing International Conference, Newcastle.

Droskack, M. J. and D. R. Houser, (977). An experimental and theoretical study of the effect of simulated pitch line pitting on the vibration of a geared system, ASME 77-DET 123.

Özguven, N. and Houser, D. R., (1988). Mathematical models used in gear dynamics - a review, Journal of Sound and Vibration, Vol .3, No. 121.

Peng, Z., and N. Kerssissogolou, (2003). An integrated approach to fault diagnosis of machinery using wear debris and vibration analysis, Wear, vol. 255.

Randall,R. B., (1982). A New method of modeling gear faults, ASME, J. Mechanical Design, No. 104.

Stewart, R. M., (1980). The specification and development of a standard for gearbox monitoring, Proc. Instr. Mech. Eng.

Umezawa, K., (1974). The meshing test on helical gears under load transmission, Bull. JSME, vol. 17, no. 112.

Umezawa, K., and T. Suzuki, (1986). Vibration of power transmission helical gears (approximate equation of tooth stiffness), Bulletin of JSME, Vol. 29, No. 251.

VanDyck, A., and G. Cotterill, (2005). Improved machine condition monitoring, Maintenance Journal, Vol. 18, No. 2, Australia.

Walsh, C. T., (1992). Condition monitoring of machine systems for the 1990s and Beyond, J. Machine vibration, vol. 1.