Reduction of Noise and Vibration of Spur Gear by Using Asymmetric Teeth Profiles with Tip Relief

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ABSTRACT

Reduction of noise and vibration in spur gear experimentally by using asymmetric teeth profiles with tip relief was presented. Both of classical (symmetric) and asymmetric (with and without tip relief) spur gears are used in this work. Gear test rig was constructed to achieve torsional vibration measuring, and two modified cutters are designed and manufactured to achieve tooth profile modifications. First to cut asymmetric gear tooth with pressure angles (14.5^{o}/25 ^{o}) without tip relief for loaded and unloaded tooth sides respectively, and second to cut asymmetric gear tooth with pressure angles (14.5^{o}/25 ^{o}) for loaded and unloaded tooth sides respectively with tip relief to achieve best dynamic performance. Dynamic load factor, transmission error and noise level are carried out in this work. Final results showed improvement in dynamic load factor and noise level for asymmetric gear (with and without tip relief) compared with classical spur gear .

Key words: dynamic load factor, tooth transmission error, asymmetric tooth profile , tooth tip relief.

اختزال الضوضاء والاهتزاز للمسننات العدلة بواسطة جانبيات الاسنان غير المتناضرة مع تشذيب الحواف

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الخلاصة

تم تقديم اختزال الضوضاء والاهتزاز للمسننات العدلة عمليا بواسطة استخدام جانبيات اسنان غير متناضرة مع تشذيب الحواف في هذا البحث. كلا من الاسنان المتناضرة والغير متناضرة (مع تشذيب الحواف وبدونها) تم استخدامه في هذا العمل. منصة اختبار المسننات بنيت خصيصا لتحقيق قياس الاهتزاز الدوراني وتم تصميم وتصنيع قاطعات مسننات محورة لتحوير جانبيات الاسنان. الاولى لقطع مسنن غير متناظر ذو زوايا ضغط (25/14.5) درجة بدون تشذيب الحواف. والثانية لقطع مسنن غير متناضر ذو زوايا ضغط (25/14.5) درجة ذو خاصية على تشذيب حواف المسنن وذلك لضمان افضل اداء

ديناميكي . في هذا البحث تم استخراج معمل الحمل الديناميكي وخطا النقل الديناميكي ومستوى الضوضاء . النتائج النهائية اظهرت تحسين في معمل الحمل الديناميكي ومستوى الضوضاء للمسننات غير المتناضرة (بوجود وعدم وجود تشذيب لحافات الاسنان) مقارنة بالاسنان التقليدية .

الكلمات الرئيسية : معامل الحمل الديناميكي , خطأ النقل لسن المسنن ,اسنان غير متناضرة الجانبيات , تشذيب حواف الاسنان.

1. INTRODUCTION

Dynamic loads, vibration and noise level have been considered as a major problem in transmitting machines especially at high speeds and heavy loads .Tooth transmission error (Tm) and non linear mesh stiffness have been represented a major excitation sources for vibration and noise in gear drive system, as the tooth transmission error defined as the difference between actual and ideal position of driven gear . The aim of this paper is reducing of dynamic load factors, transmission error and noise level by modifying gear tooth profiles experimentally by using asymmetric teeth profiles with tip relief, where relief defined as removing metal from the tip or the root of the teeth or from both, **Smith, 1999**. Little literature attempted to investigate dynamic characteristics of spur gear system experimentally. **Munro, 1962** measured dynamic transmission error in spur gear pair, he selected high precision spur gears with manufacturing errors much smaller than tooth deflection, and he applied tooth profile modifications to achieve minimum static transmission error at design load. **Kubo, 1962** designed gear test rigs which were heavily damped, he measured dynamic root stress and then estimated the dynamic factor. **Smith, 1999** used two smaller optical rotary encoder made by Heidenhain Ltd. which had become readily available and used extensively for rotary positioning system on tooth geared system. His results indicated error below 0.1 second of arc at operating speed. **Kang ,2009** indicted the shaft compliance effect on dynamic Tranmission error, he used tangential accelerometer for measuring the gear motions in rotational, torsional and translitional axis, he used spur and helical gears in his experimental work. In this work gear test rig was built to achieve torsional vibration measuring, and two modified cutters was designed and manufactured to achieve tooth profile modifications such as asymmetric tooth profile with and without tip relief to improve dynamic load factor, transmission error and noise level.

2. PROBLEM FORMULATION

Simple dynamic model with single degree of freedom has been employed in this work ;therfore , equation of motion will be, **Hasan, et al,2014** :

$$m_e \ \ddot{x} \ + c_m \dot{x} \ + k_m \ (t) \ x = \ F \quad (1)$$

Where *x(t)* refers to dynamic transmission error of a gear pair along its line of action which defined as:

$$x(t) = \ r_{b1} \, \theta_1 - r_{b2} \, \theta_2 - e(t) \quad (2)$$

r_{b1} & r_{b2} represented base radius for pinion and gear respectively , θ_1 & θ_2 represented angular displacement for gear and pinion respectively, and *e(t)* represented periodic static transmission error , m_e equivalent mass, k_m non linear mesh stiffness , c_m non linear mesh damping and F static load .

Dynamic load factor in gear drive system defined as the ratio of dynamic mesh load to static load under operating speed :

$$DLF = \ \frac{F_d}{F} \quad (3)$$

Dynamic mesh load in Eq. (3) defined as, **Duboswky and Freudenstein, 1971**:

$$Fd = c_m \dot{x} + k_m(t)\, x \quad (4)$$

Sub. Eq. (4) in Eqs. (1) and (3). Dynamic load factor has been written in terms of second derivative of transmission error as:

$$DLF = 1 - \frac{m_e \ddot{x}}{F} \quad (5)$$

Where ($\ddot{x}$ & x) measured by gear test rig experimentally as shown in next section .

3. GEAR TEST RIG

Test rig was constructed to analyze dynamic performance in gear drive system such as dynamic load factor, transmission error and noise level. Three types of steel spur gear pairs with a unity speed ratio (1:1) and a fixed center distance of 98 mm had been selected to be under the test. The first type was symmetric spur gear with classical pressure angles ($20^o/20^o$). The second type was asymmetric spur gear with pressure angles ($14.5^o/25^o$) for loaded and unloaded tooth sides respectively. Third type was asymmetric spur gear with pressure angles ($14.5^o/25^o$) for loaded and unloaded tooth sides respectively but with tip relief. All design parameters of the symmetric and asymmetric spur gear drives with and without tip relief are listed in table (1). The chemical composition of the spur gears (pinion and gear) and shafts material are inspected in "Specialized Institute for Engineering Industries/Ministry of Industry & Minerals/Iraq" that was listed in Appendix (A) .

A gear milling operation had been adopted to manufacture the test specimens (prototypes) of all spur gears by using standard and modified gear milling cutters.

A standard HSS gear milling cutter of disc type had been adopted to cut symmetric prototypes of spur gears (pinion and gear). According to the design specifications of DIN 3972, the cutter number can be selected by depending on the gear design parameters to be cut, where each cutter number was designed to cut a range of teeth number for certain pressure angle and module, therefore; with gear design parameters $\phi = 20^{\circ}$, $m_o = 7$ mm and $z = 14$ teeth, cutter No.2 in Appendix (B) had been selected, where this cutter can be used to cut three gears with different number of teeth ($z = 14$, 15 and 16 teeth) .

A modified HSS gear milling cutter of disc type with asymmetric involute profiles with optimal fillet radii and without tip relief had been designed by **Abdullah and Jweeg, 2012,** depending on the tooth profiles geometry. Asymmetric tooth profiles with tip relief had been design depending on reduction in static transmission error by selecting appropriate amount and location of tip relief on asymmetric tooth profile (tip amount 40 micron , start relief lied on 0.33 base pitch on involute profile) as shown in **Fig.1** ,amount and extent of tip relief achieved theoretical basis of spur gear profile relief design that established by **Munro et ,al ,1990** and **palmer ,1999** .

The workshop drawings and manufacturing of these cutters had been achieved by "Acedes Gear Tools Company (division of Furzeland Ltd.) / UK-England". Final products of these cutters are shown in **Fig.2**. Test rig (gear-shaft-bearing system) as shown in **Fig.3** had been constructed precisely to indicate dynamic load factor, transmission error and noise level under certain external load and rotational speed values for symmetric , asymmetric and asymmetric with tip relief spur gear . Two steel gears (pinion and gear) of the same diameter of 98 mm are fixed on two steel shafts with diameter of 49 mm; where two ball bearings supported each shaft. AC motor (3 KW, 2880 rpm, Three-phases, 220~230 Volt, 50~60 Hz, MeZ Electric Motors Ltd./Czech Rep.) was used to provide sufficient amounts of the required torque to overcome the overall inertias of mechanical components of test rig and steel flywheels with masses (5 kg , 15 kg , 25kg , 35 kg) respectively. Speed control was automated by using a controller speed device

(Variable Frequency Drive , Power range: from 0.4 to 3.75 KW, Three phase, 220~230 Volts, 50~60 Hz, Delta Industrial Systems Co. Ltd./ Taiwan) . A thick steel plate with dimensions (700 × 300 × 20 mm) was used as a platform for fixing the motor by using four bolts with diameter of 10 mm and the fixed four housings by using four bolts with diameter of 12 mm for each housing. The measurement system was designed to perform measuring torsional vibration components ($\theta 1$ and θ_2) for both pinion and gear. Uni-axial tangential accelerometer was devised. Accelerometers (PCB Piezotronics, Model: 353B18, sensitivity: nearly 10 mV/g, frequency range: 0 to 10 kHz) can be mounted tangentially at pinion and gear respectively as shown in **Fig4**.

The accelerometers data are fed to the fixed frame through the end-of-shaft by using slip rings (Jinapat SR, Model: LPT038). Signals transmitted from the slip rings are fed into a multi-channel signal conditioner (PCB Piezotronics ICP Model 482C64) to condition and amplify the data. Then, the signals are fed to Digital storage Oscilloscope that digitizes the analog signals at a user defined sampling rate and monitoring signals in voltage amplitudes, **Fig. 5** shows the flow chart for measurement system .

Sound intensity levels can be measured by using a sound level meter *(*Model: SL-4022, IEC 61672 class 1, Microphone type: Electrical condenser, Measurement range: from 30 to 130 dB, Lutron Electronic Enterprice Co. Ltd./Taiwan*)* by putting meter microphone vertically with a distance 15 Cm above the gears engagement location to obtain purest sound signal as shown in **Fig. 6** .

4. RESULTS AND DISCUSSION

Digital storage oscilloscope shows accelerometers signals, but in voltage amplitudes, so these data should be converted to acceleration values in (m/s^2) by using accelerometer conversion scale. Dynamic transmission error defined as the difference between axial displacement of pinion and gear ; therefore, Sigview software (V 2.6) analyzed storage signals and integrated it twice to carry out dynamic transmission error and dynamic load factor.

Fig. 7 shows the variation of noise levels (sound intensity level in dB) with rotational speed of motor for both of asymmetric gear teeth with and without tip relief. Generally Tip relief in asymmetric gear teeth reduced the noise levels of spur gear by around (10 -20) % in low speeds (less than 1000 rpm) and by around (2 -5) % in high speeds (more than 3000 rpm).

Fig.8 shows dynamic load factors with rotational speed of motor for symmetric gear teeth with pressure angles ($20^o/20^o$) and asymmetric gear teeth with pressure angles ($14.5^o/25^o$) without tip relief . Generally asymmetric gear tooth with tip relief shows improvement in dynamic load factors at all speeds except at super harmonic frequency which occurred at 1000 rpm .In high speeds (more than 2000 rpm) the dynamic load factor became greater compared with low speeds and symmetric gear tooth recorded high peak of dynamic load factor at 2200 rpm which approximate 6 for dynamic load factor .

Fig. 9 shows comparison between dynamic load factors with rotational speed of motor for asymmetric tooth gear with and without tip relief , this figure shows convergence in results but with improvement for asymmetric tooth without tip relief at low speeds (less than 1000 rpm) and improvement for asymmetric tooth with tip relief at high speeds (more than 1200 rpm) .

Fig. 10 shows the effect of different inertial loads on dynamic load factors for asymmetric gear tooth with tip relief. (5kg, 15 kg and 25 Kg) fly wheels are used in gear test rig. In low speeds (less than 1500 rpm) both of (5 and 25 kg) flywheels showed improvement in dynamic load factors compared with 15 kg flywheel . In high speeds (more than 1500 rpm) 15 kg flywheel showed improvement in dynamic load factors compared with 25 kg flywheel which was increased while 5kg inertia still better compared with both of (15 &25 kg) flywheels.

Fig. 11 shows dynamic load factors for symmetric gear tooth with loaded pressure angle (20^0) and asymmetric gear tooth with tip relief and loaded pressure angle (25^0) with rotation speed . Results showed improvement by around (51%) in asymmetric gear tooth at most speeds (higher than 500 rpm) although loaded pressure angle in symmetric gear tooth lower than asymmetric gear (25^0) due to modified loaded tooth side by medium tip relief.
Fig.12 shows dynamic transmission error for symmetric and asymmetric tooth gear with rotation speed, it's clear that asymmetric gear tooth showed improvement by around (50 %) in dynamic transmission error compared with symmetric tooth gear at all speeds except (1000 rpm) .
Fig. 13 shows improvement by around (50%) in dynamic transmission error for asymmetric gear tooth without tip relief compared with asymmetric gear tooth with tip relief at low speeds (less than 1100 rpm) ,for other speeds tip relief modification in asymmetric tooth profile improve dynamic transmission error in spur gear by around (20-66 %) .
Fig. **14** shows the effect of different inertias on dynamic transmission error for asymmetric tooth gear with tip relief , and three regions were remarked in this figure. In low speeds (less 1000 rpm) (5kg and 25 kg) flywheels showed improvement in results compared with 15 kg flywheel , at speeds (1000 -2000 rpm) (15kg and 25 kg) flywheels show improvement in results compared with 5 kg flywheel , at high speed (more than 2000 rpm) 15 kg flywheel shows improvement in results compared with (5 kg and 25 kg) flywheels .
Table 2 showed maximum dynamic load factors and maximum dynamic transmission errors that were recorded in previous figures.

5. CONCLUSION

Noise level, Dynamic load factor and dynamic transmission error are carried out experimentally in this paper. Results showed that the noise radiated from asymmetric gear drive system was improved when asymmetric profiles modified by appropriate tip relief by around (10 -20) % in low speeds and by around (2 -5) % in high speeds .

Asymmetric gear teeth with and without tip relief showed improvement in dynamic load factor and dynamic transmission error compared with symmetric gear tooth where percentage improvement reach (50 – 80) % in different rotations speed. Minimum flywheel (5 kg) used in asymmetric gear drive showed lowest dynamic load factor compared with other flywheels while (15) kg flywheel showed lowest dynamic transmission error compared with other flywheels.

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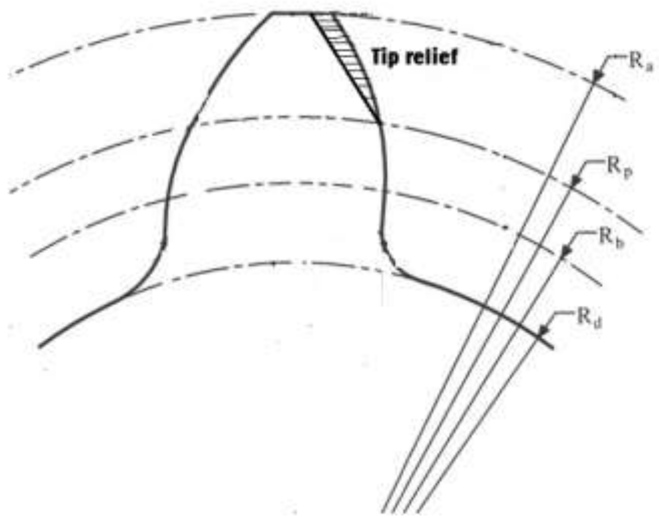

Figure 1 . Asymmetric tooth profile with tip relief.

Figure 2. Asymmetric gear milling cutter.

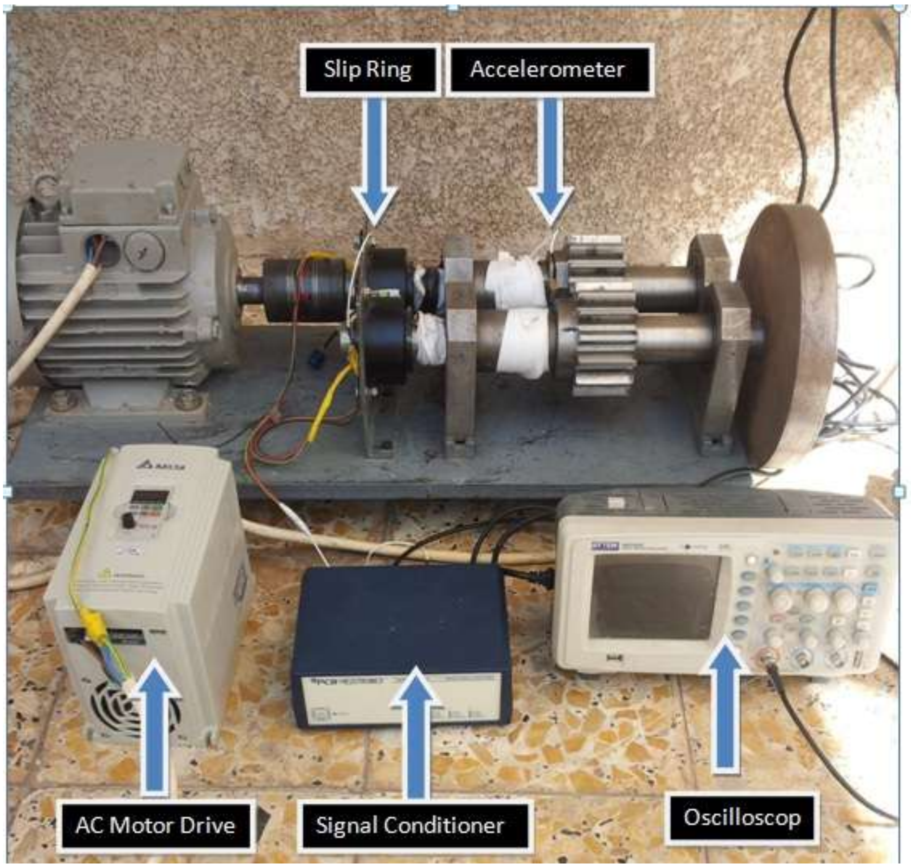

Figure 3. Gear test rig.

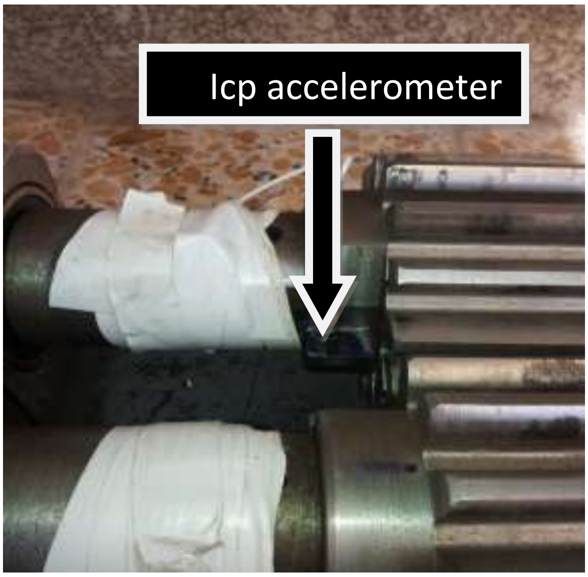

Figure 4. Tangential accelerometer.

Figure 5 . Noise measurement .

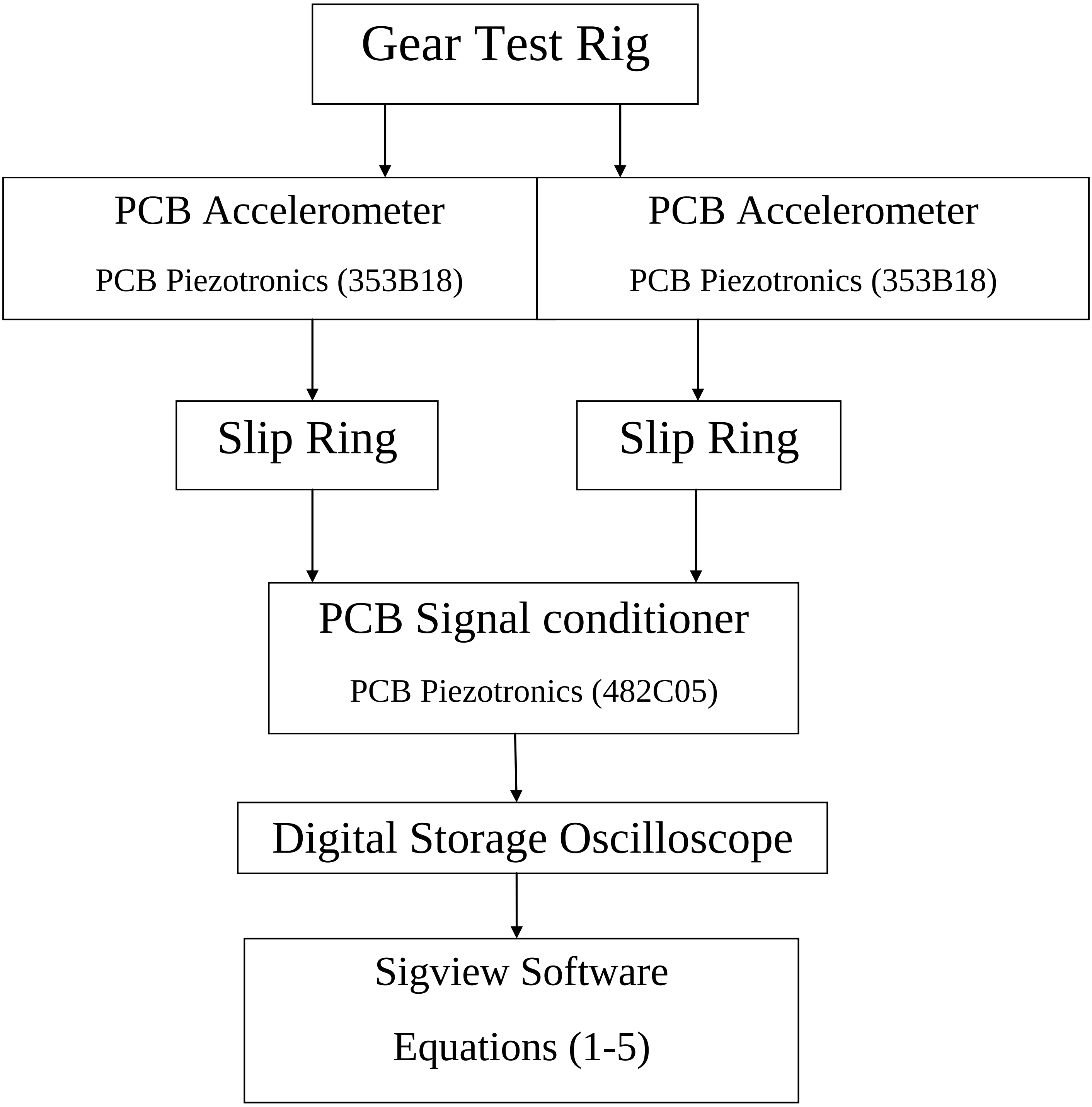

Figure 6. Flow chart for measurement system.

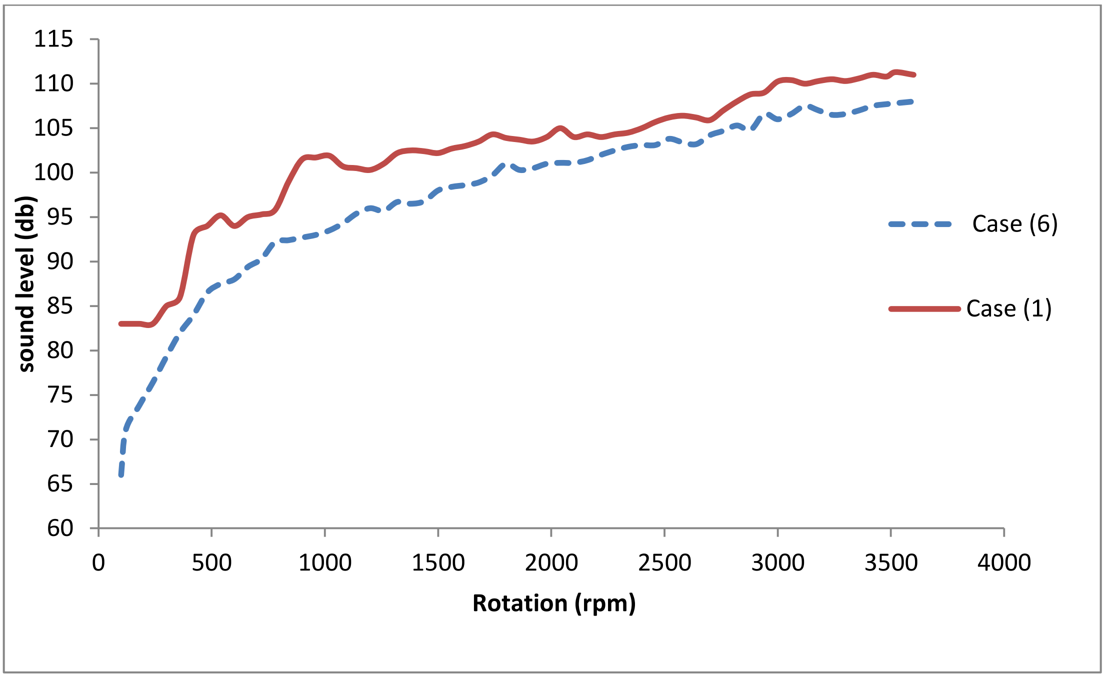

Figure 7.The Variation of sound intensity level for asymmetric gear with and without tip relief.

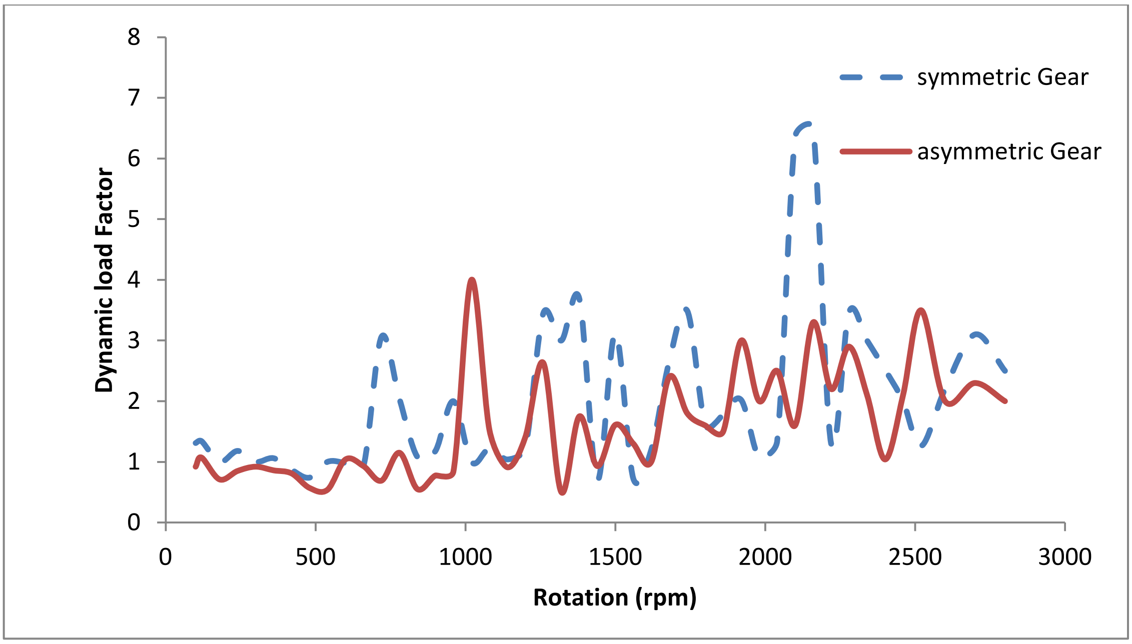

Figure 8. Dynamic load factor for symmetric& asymmetric gears.

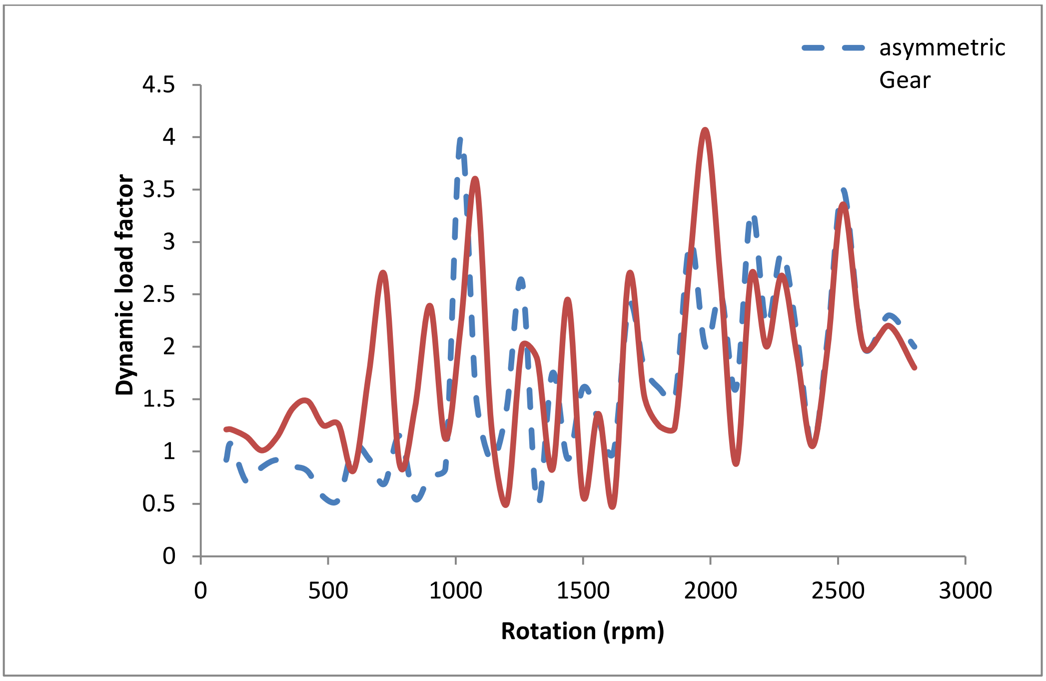

Figure 9. Dynamic load factor for asymmetric gears with and without tip relief.

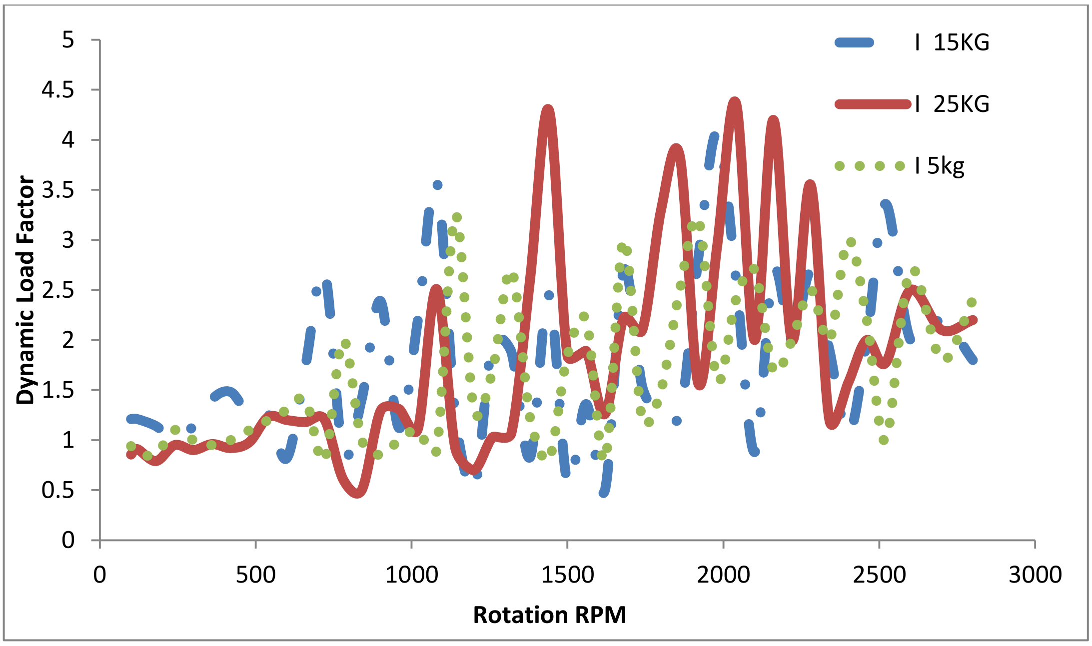

Figure 10. The effect of different inertia on dynamic load factor for asymmetric tooth gear with tip relief.

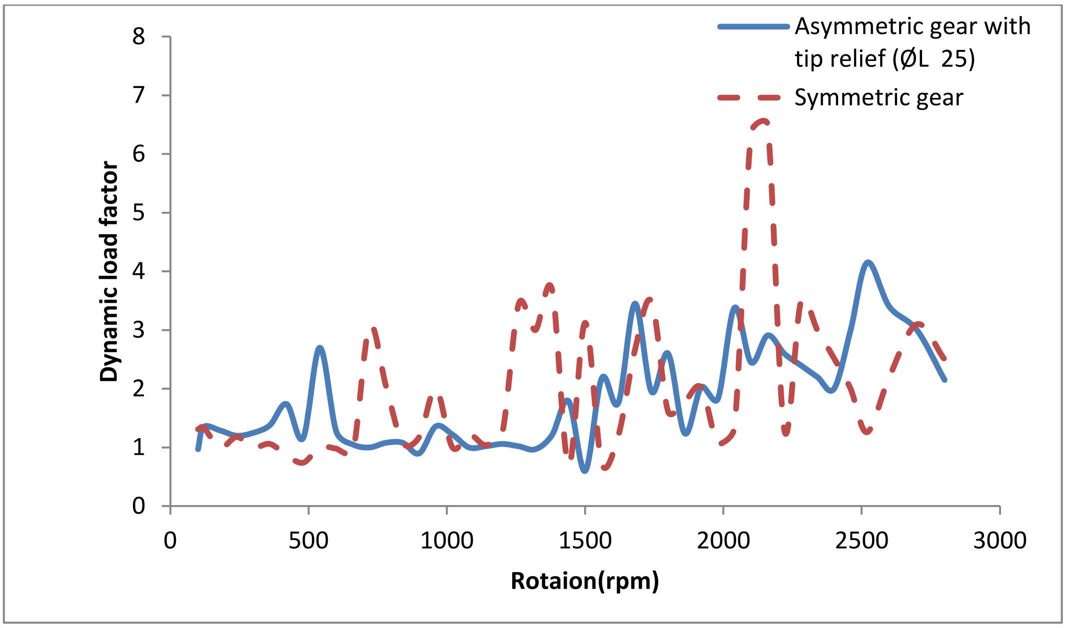

Figure 11. Dynamic load factor for symmetric gear and asymmetric gear with tip relief and loaded pressure angle (25^0).

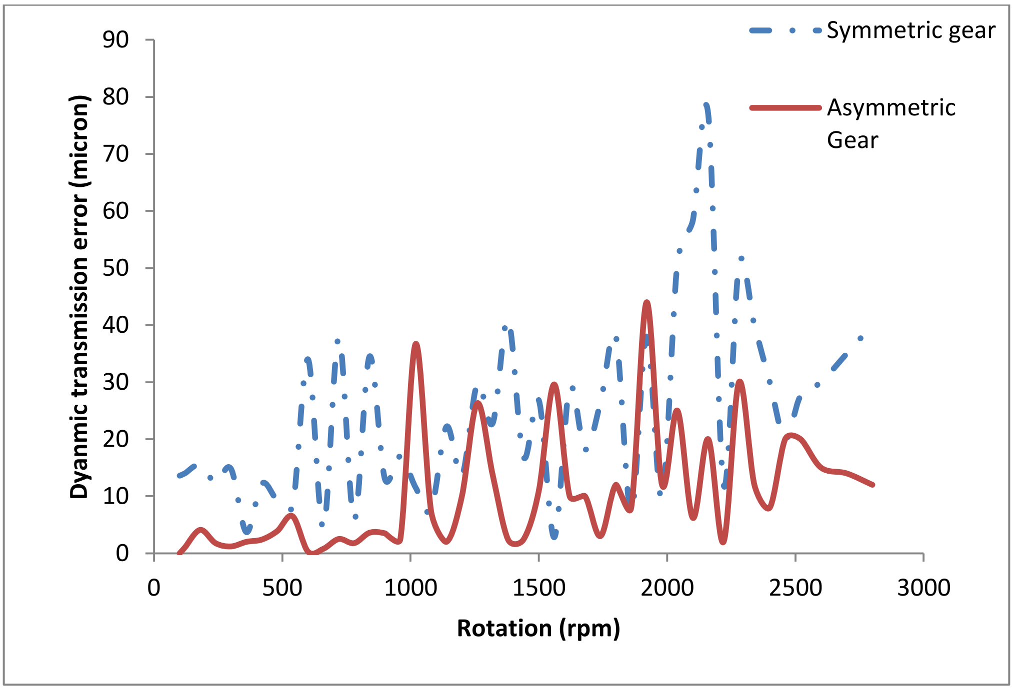

Figure 12. Dynamic transmission error for symmetric and asymmetric gear.

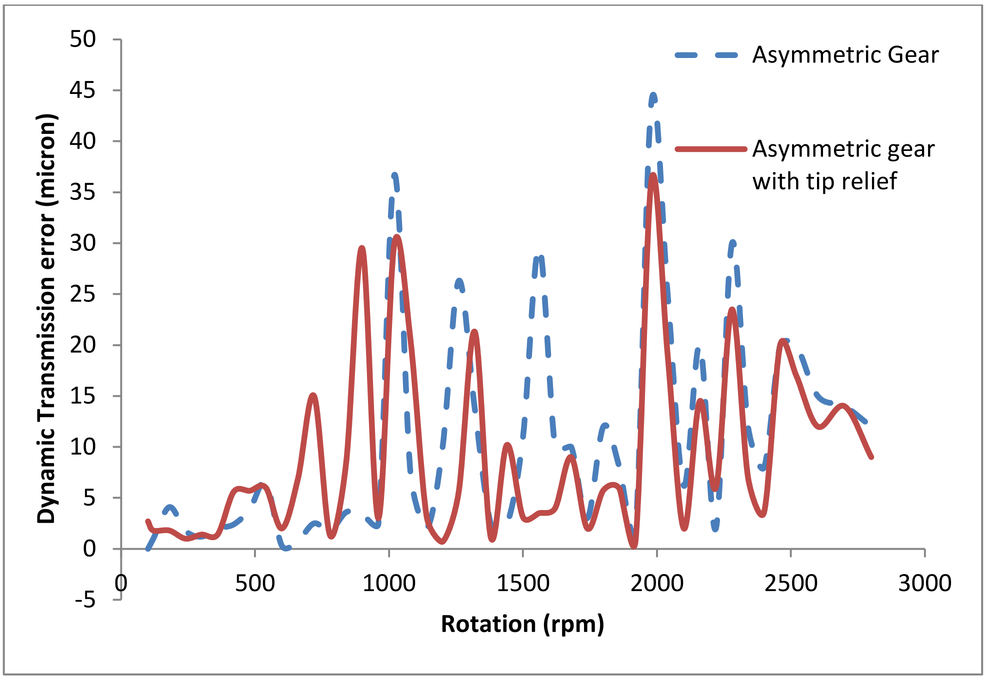

Figure 13.Dynamic transmission error for asymmetric gears with and without tip relief.

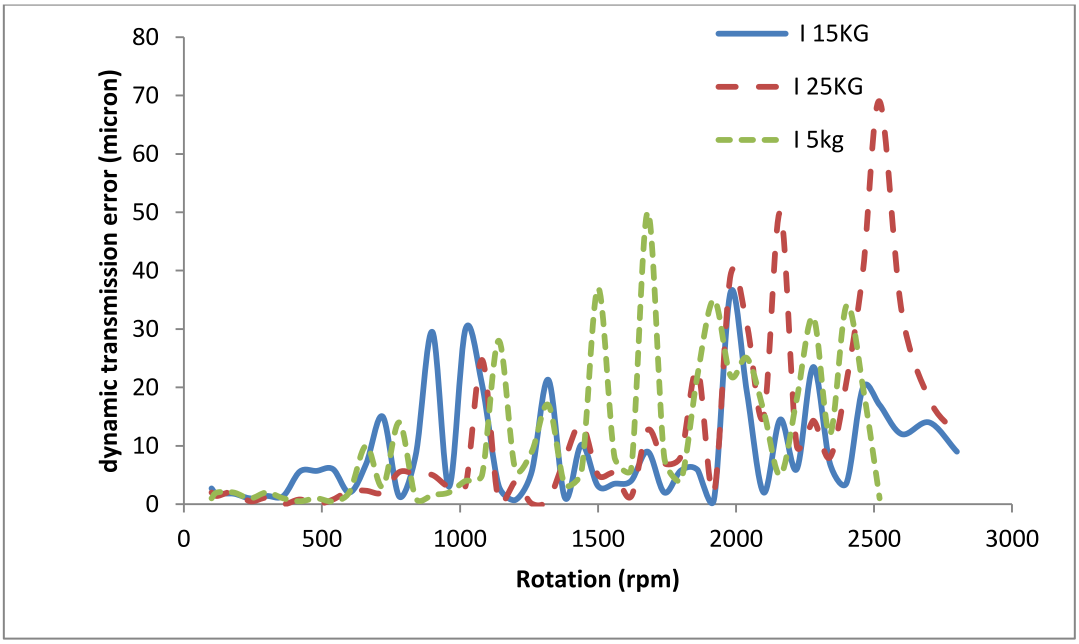

Figure 14. The effect of different inertia on dynamic transmission error for asymmetric tooth gear with tip relief.

Table 1. Design parameters of symmetric, asymmetric and asymmetric with tip relief of spur gear drive.

Gear parameter	Symmetric gear drive (pinion and gear)	Asymmetric gear drive (pinion and gear)	Asymmetric gear drive with tip relief (pinion and gear)
Pressure angle	$20^{o}/20^{o}$	$14.5^{o}/25^{o}$	$14.5^{o}/25^{o}$
Module	7 mm	7 mm	7mm
Face width	60 mm	60 mm	60 mm
Teeth number	14	14	14
Tooth addendum height	7 mm	7 mm	7 mm
Tooth dendum height	8.16 mm	8.16 mm	8.16 mm
Fillet radius	2.1 mm	2.1 mm	2.1 mm
Relief amount	-	-	40 micron
Relief specification	-	-	0.33 pb

Table 2. Maximum dynamic load factor and maximum dynamic transmission error for different gear types.

Gear type	Maximum Dynamic load Factor	Maximum Dynamic Transmission Error (micron)
Symmetric	6.1	75
Asymmetric	4	40
Asymmetric with tip relief (I 5kg)	3.2	45
Asymmetric with tip relief (I 15kg)	3.3	35
Asymmetric with tip relief (I 25kg)	4.2	65
Asymmetric with tip relief (I 35kg)	3.4	32