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Investigation on the Performance of a Spur Gear System under Variance Load and Frequencies

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Abstract

Knowing the operating conditions of a gear system is important to its durability and longevity. When the maximum specified system load is exceeded during the operation, there is a possibility of a complete system breakdown. In this paper, a single spur gear system was used by applying three different loads to simulate changes in operating frequencies. Three loads and four rotational speeds have been applied to the gear system, representing three load levels, simple, medium, and high. Vibration is measured by a 3-axis sensor, "MPU-6050", via the Arduino connection, in which the acceleration is measured in the X, Y, and Z directions in two stages. The first stage is done by taking measurements of the frequency in normal operation and comparing them to the operation with the loads in the second stage. The FFT function converts the acceleration signals produced by the sensor in the time domain into Hz frequency signals with the software "SigView". The results clearly showed the direct effect of increased loads on the rotary gear system by increasing the operating frequency values. The increase in frequency resulting from the increased load applied to the system was associated with the increase in the rotational speed. The results showed that the frequencies are increased by about twenty percent when placing high loads with high rotational speed, which reflects negatively on the entire system.

Keywords: Rotational Speed, Spur Gear, Frequency, Single Shoe Brake System

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التحقق من أداء منظومة مسنن عدل تحت حمل متغير والترددات الطبيعية

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الخلاصة

إن معرفة ظروف تشغيل نظام التروس يعد أمراً مهماً لديمومتها وزيادة عمرها. عند تجاوز الحمل الاقصى المحدد للنظام اثناء التشغيل هناك احتمالية انهيار كامل للنظام عندها. في هذه الورقة البحثية، تم استخدام نظام تروس عدل ذو مرحلة واحدة من خلال تطبيق ثلاثة أحمال مختلفة لمحاكاة التغييرات في ترددات التشغيل. تم تطبيق ثلاث أحمال وأربع سرعات دورانية على نظام التروس، وهو ما يمثل ثلاثة مستويات للحمل، بسيطة ومتوسطة وعالية. يتم قياس الاهتزازات بواسطة مستشعر ثلاثي المحاور ، "MPU-6050" ، عبر وصلات Arduino ، حيث تم قياس التسارع في اتجاهات X و Y و Z على مرحلتين. تتم المرحلة الأولى عن طريق أخذ قياسات التردد في التشغيل العادي ومقارنتها بالتشغيل بالأحمال في المرحلة الثانية. تعمل وظيفة FFT على تحويل إشارات التسارع التي ينتجها المستشعر في المجال الزمني إلى إشارات تردد بوحدة هرتز بواسطة برنامج "SigView". أظهرت النتائج بوضوح التأثير المباشر لزيادة الأحمال على نظام التروس الدوارة عن طريق زيادة قيم ترددات التشغيل. ارتبطت الزيادة في الترددات الناتجة عن زيادة الحمل المطبق على النظام بزيادة سرعة الدوران. وأظهرت النتائج

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[1] **المؤلف المراسل**

معلومات البحث
تأريخ النشر : 2022

زيادة الترددات بنحو عشرون بالمائة عند وضع أحمال عالية مع سرعة دوران عالية مما ينعكس بصورة سلبية على كامل النظام.

الكلمات المفتاحية : سرعة الدوران، الترس العدل، التردد، نظام كبح فردي

1. Introduction

Gear systems consist of several factors that help to define their functions, such as rotational speed, the direction of transmission, gear sizes used, and other factors. Often the gear systems need to be operated under a variable load that varies from a simple, medium, or even high, several researchers presented several studies examining the effect of loads on a gear. [1], which shows the fluctuating load conditions on the gear system. The experimental data calculated during constant, sinusoidal, and other situations of a damage gear system. [2], determines the effects of machining error, component errors, and gear tooth modifications on loading ability. Besides, the load sharing ratio and the transmission error of a gear system by using the finite element method software. [3], taking polyamide-based gears and detecting the thermal damage of the gear tooth surfaces over gear meshing operation due to the heaping up heat in the tooth body. The teeth have been modified by drilling cooling holes at several locations on the gear tooth body that leads to a lowered temperature of the tooth surface and drives to an increase in the load carry capacity. [4], using a model of non-uniform load allocation over the side contact between teeth, allowing analytic measurement of the load per-unit of length at each point of the line of communication between gear teeth. The results have been compared with some studies that work out by the finite element method FEM. [5], used a non-uniform model of load allocation over the line of contact of gear teeth for spur and helical gears concerted with the use of linear elasticity equations. [6], investigated the meshing stiffness of the teeth of a spur gear. The load sharing ratio is measured and compared with the former results found from the hypothesis of "minimum elastic potential energy", by considering the tooth deflections and deserting the Hertzian deflections. Several studies have taken important aspects of the effect of loads on rotating gear systems. Some of them shed the light on the teeth of the gears and some of them focus on some of the operational factors associated with the design of gears. In this research paper, a similar approach will be followed for these studies, with an explanation of the effect of overloading on the operating frequency of the system as a whole, according to the increased rotational speed.

2. Theoretical and Experimentation Work

2.1 Spur Gear System

The open gear system consists of a pair of spur gear coupling with a three-phase electric motor, shows that Figure (1). A single-shoe brake system was manufactured and installed on the big gear shaft, consisting of an iron disc with a diameter of (150 mm) and 25 mm thickness. Above the iron disk, the braking arc has an angle of 60 degrees has installed, the brake arc has a material with a high friction content that facilitates stopping the disk when rotating. The braking system has an arm that can be added under different loads easily on it, shows that Figure (2). Table 1 has more details about the geared system used.

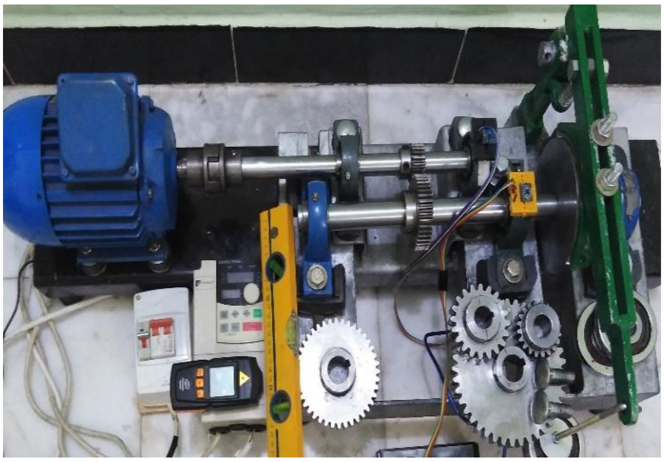


Figure 1. Manufactured Spur Gear System

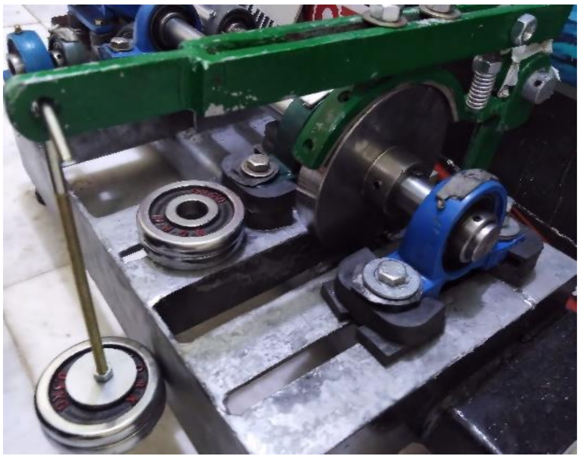


Figure 2. Manufactured Brake System

Table 1. System Details

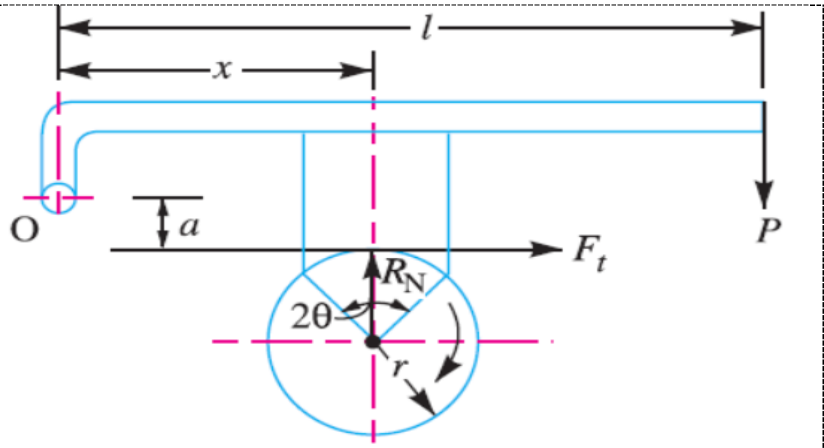
Part Name	Details	Quantity
Pinion Bearings	UCP205 Pillow-Block Bearing	2
Gear Bearings	UCP206 Pillow-Block Bearing	3
Pinion	Module 3 mm 20 teeth	1
Gear	Module 3 mm 40 teeth	1
Pinion Shaft	(350 mm) length, (25 mm) diameter	1
Gear Shaft	(500 mm) length, (30 mm) diameter	1
Speed Controller	Powtran type (Pl8100a R75G3)	1
Electrical Motor	Three-Phase 0.75 kW, Max. Speed (2850 rpm)	1

2.2 Vibration Measurements

The MPU-6050 sensor [7], was used to measure acceleration in the three axes X, Y, and Z simultaneously. The sensor connects with a computer via the Arduino Mega-2560 card to deliver the computed signal. The resulting signal from the sensor was handled by the LabVIEW program as a time-domain signal, and then this signal was converted by SigView program to the frequency signal in hertz units.

2.3 Brake System Calculations

The single-shoe braking system is of a rather simple principle [8], whereby loads are added at the end of the arc brake arm. The placed loads create a torque opposite to the torque generated as a result of the rotational speed of the system, as it is affected by an increase or decrease depending on the value of the added weight in addition to the dimensions associated with its place. Figure (3), represents a detailed scheme of the braking system.



(a) Clockwise rotation of brake wheel.

Figure 3. Scheme of the Brakes System

l	450 mm	a	10 mm	2θ	60°
x	200 mm	r	75 mm	μ	0.3

For equilibrium, taking moments about the fulcrum (O):

$$R_N \cdot x + F_t \cdot a = Pl \quad \text{or} \quad R_N = \frac{P.l}{x + \mu.a} \qquad \ldots (1)$$

P_1= 1 kg × 9.81 m/s² = 9.81 N then $R_{N1} = \frac{9.81 \times 0.45}{0.2 + (0.3 \times 0.01)} = 21.74$ N ... (2)

P_2=2 kg × 9.81 m/s² = 19.62 N then R_{N6} = 43.5 N ... (3)

P_3 =6 kg × 9.81 m/s² = 58.86 N then R_{N7} = 130.477 N ... (4)

To find the friction force ($F_r = \mu.R_N$) and the torque ($T_B = Fr.r$):

F_{r1}=0.3×21.74=6.522 N then T_{B1}= 6.522 × 0.075 = 0.49 N.m. ... (5)

F_{r2} =0.3×43.5=13.05 N then T_{B2}= 13.05 × 0.075 = 0.98 N.m. ... (6)

F_{r3} =0.3×130.477=39.14 N then T_{B3}=39.14× 0.075 = 2.935 N.m. ... (7)

The results (Table 2 and Figure 4):

Table 2. Brake Measurements

Weight / kg	Force applied/ N	Friction Force / N	Brake Torque / N.m
1	9.81	6.522	0.49
2	19.62	13.05	0.98
6	58.86	39.14	2.935

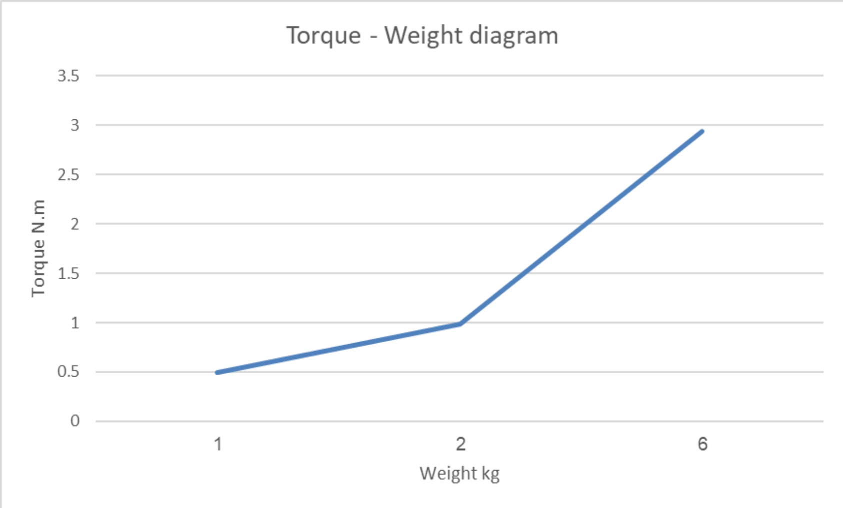


Figure 4. Brake Torque

From Figure (4), it can be seen that an increase in the impacted load leads to an increase in the reverse torque applied to the rotary disc, which in turn is the braking torque of the gear system.

3. Results and Discussion

The practical readings were taken for two shafts everyone is alone, as well as taken at four rotational speeds, which are 500, 1000, 1500, and 2000 rpm and at three loads. The frequency measured in the three directions X, Y, and Z, the resulting frequencies will be taken before the loads are placed and after that.

3.1. Practical Frequency Results by the Normal Operation

The frequency resulting values from operating the spur gear system without any loads or during normal operation can be listed in Table 3**.**

Table 3. Normal Operation Frequencies

Rotational Speed	25 mm Shaft Normal operation Frequency / Hz			30 mm Shaft Normal operation Frequency / Hz		
	X-axis	Y-axis	Z-axis	X-axis	Y-axis	Z-axis
500 rpm	110.84	125	145.02	140.38	123.05	148.93
1000 rpm	341.31	336.91	331.98	348.63	344.97	342.41
1500 rpm	514.16	514.16	514.16	510.25	489.26	510.25
2000 rpm	586.67	583.5	586.67	598.14	596.68	585.94

3.2 Practical Frequency Results Loads

The frequency resulting values from operating the spur gear system under 1 kg, 2 kg, and 6 kg loads can be listed in Tables 4, 5, and 6**.**

Table 4. Frequency under 1 kg Load

Rotational Speed	25 mm Shaft / Frequency / Hz			30 mm Shaft /Frequency / Hz		
	X-axis	Y-axis	Z-axis	X-axis	Y-axis	Z-axis
500 rpm	134.93	135.79	136.96	138.67	139.89	140.14
1000 rpm	334.84	333.66	334.98	339.98	339.33	339.75
1500 rpm	527.34	527.34	527.05	527.01	531.74	528.91
2000 rpm	626.83	626.83	628.42	628.91	625.84	627.54

Table 5. Frequency under 2 kg Load

Rotational Speed	25 mm Shaft / Frequency / Hz			30 mm Shaft /Frequency / Hz		
	X-axis	Y-axis	Z-axis	X-axis	Y-axis	Z-axis
500 rpm	150.98	150.27	150..76	146.88	145.31	148.32
1000 rpm	351.12	351.12	353.91	346.68	346.68	346.43
1500 rpm	541.21	538.67	538.25	543.04	545.39	545.7
2000 rpm	671.48	672.66	671.48	677.61	676.76	690.23

Table 6. Frequency under 6 kg Load

Rotational Speed	25 mm Shaft / Frequency / Hz			30 mm Shaft /Frequency / Hz		
	X-axis	Y-axis	Z-axis	X-axis	Y-axis	Z-axis
500 rpm	158.98	161.5	160.97	155.57	156.25	158.45
1000 rpm	366.36	366.39	367.68	366.58	355.71	364.01
1500 rpm	581.16	580.63	580.81	588.65	585.33	596.19
2000 rpm	706.84	703.42	701.37	709.77	710.16	712.89

3.3 Relation between Load and the Frequencies

3.3.1 For Pinion Shaft (Figure 5)

500 rpm / 3 mm Module / Pinion Shaft

	6 kg	2 kg	1 kg
X	158.98	150.98	134.93
Y	161.5	150.27	135.79
Z	160.97	150.76	136.96

1000 rpm / 3 mm Module / Pinion Shaft

	6 kg	2 kg	1 kg
X	366.36	351.12	333.84
Y	366.39	351.12	333.66
Z	367.68	353.91	334.98

1500 rpm / 3 mm Module / Pinion Shaft

	6 kg	2 kg	1 kg
X	581.16	541.21	527.34
Y	580.63	538.67	527.34
Z	580.81	538.25	527.05

2000 rpm / 3 mm Module / Pinion Shaft

	6 kg	2 kg	1 kg
X	706.84	671.48	626.83
Y	703.42	672.66	626.83
Z	701.37	671.48	628.42

Figure 5. Relation between Loads and the Frequency for Pinion Shaft

3.3.2. For Gear Shaft (Figure 6)

Figure 6. Relation between Loads and the Frequency for Gear Shaft

The practical frequency values are recorded in addition to the drawn diagrams record clear results of a significant increase in frequency before and after adding loads to the gear system. At 1 kg load, the frequency increase was 8 percent versus 20 and 28 percent for 2 and 6 kg loads, respectively, at 500 rpm. At 1 kg load, the frequency increase was 2 percent versus 6 and 10 percent for 2 and 6 kg loads, respectively, at 1000 rpm. At 1 kg load, the frequency increase was 3 percent versus 5 and 13 percent for 2 and 6 kg loads, respectively, at 1500 rpm. At 1 kg load, the frequency increase was 7 percent versus 15 and 20 percent for 2 and 6 kg loads, respectively, at 2000 rpm. It can be seen that the increase in operating frequency is high at low speeds due to the instability of the system about the imposed loads. The increase is also clear at higher speeds due to the high gear meshing frequency. The average speeds showed good stability and a fairly acceptable increase in operating frequencies.

4. Conclusions

- Diagnosing the condition of systems that contain gears is one of the most important matters that must be addressed in an analytical and detailed way from several aspects. In this research paper, the aspect related to the overloading gear system was taken and related to the finding of the frequencies in three axes according to an ascending pattern in terms of the increased load. A single-shoe braking system was used, where it was installed with the gear system, three loads, and four rotational speeds were selected to find the frequencies for the three axes of the two gear shafts.

- On the theoretical side, it has been shown conclusively that the torque applied by the braking system is progressively increased according to the applied load. Where the torque was calculated according to the principle of the exerted force by the set weights, which, in turn, transformed into a frictional force between the disc and the arc of the brake system to create a counter torque to the rotational motion.

- On the practical side, practical measures were taken by the acceleration sensor in three directions simultaneously, where the frequencies of the gear system were taken in the normal position to run the system without an effect by any loads, and then the frequencies were measured after applying three different loads and four rotational speeds on the gear system.

- The relationships show that increasing the loads leads to an increase in the calculated frequency value. It can be seen that the increase in acceleration in the three directions is about seeming in one increasing level, as well as that the increased acceleration at the higher rotational speed has a rise greater than that of the low rotational speed when calculating the resulting at the three loads.

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