EFFECT OF BACKLASH ON VIBRATIONS OF TWO ROTORS SYSTEM CONNECTED BY GEAR COUPLING

By

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ABSTRACT:

Gear couplings are frequently used to connect two rotors as in feed pump units, compressors and blowers. The effect of backlash and teeth errors on vibrations set up in the rotor system have not yet been throughly investigated.

This paper presents an experimental analysis for the vibrations in the rotor system attributed to the gear coupling mechanism. A test rig is specially designed, which consists of two rotors connected by a gear coupling having different amounts of backlash.

1. INTRODUCTION:

The different amounts of backlash ranging from 0.0067 to 0.0175 rad. are obtained by changing the engagement position of the gear coupling teeth. Torsional and transverse vibrations of the rotor system are recorded during the periods of starting to stopping of the system. Precautions are made to minimize the effect of other factors to which vibrations may be attributed. The critical speeds of the rotor system are theoretically calculated by using Myklestad method (3) and Holzer method (4) in order to avoid resonance conditions.

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Natural frequencies of bearing supports are experimentally determined and an operating speed of 3000 r.p.m. appears to be a reasonable value, since it is well below all speeds of excitation. The harmful inertia effects are dynamically eliminated by testing and correcting the balancing condition of the rotor system on a pivoted - cradle balancing machine.

The computer programs are developed to find the values of the critical speeds of the rotor system, corresponding to transverse and torsional natural frequencies of vibration.

A complete set of experimental results is carried out, analyzed and represented graphically.

Finally, it has been shown that the minimum dynamic torques occur at values of backlash ranging from 0.009 to 0.0125 rad. Also, it may be recommended that the gear coupling unit must be out of work when its amount of backlash reaches 0.011 rad., since the dynamic factor is about 1.5 .

2. EXPERIMENTAL INVESTIGATION:

Experimental investigations are carried out on the test rig shown in Fig. (1).

2.1. Balancing Test:

The dynamic balancing test of the rotor system is carried out on a pivoted-cradle balancing machine. Four symmetrical long bolts are used to connect the two flange couplings of the male and female gears as shown in Fig. (2) in order to ensure perfect alignment of the two rotors on the balancing machine supports.

2.2. Bearing Support Natural Frequency Measurements:

The natural frequency of the bearing support has been measured by using the exciting system shown in Fig. (3).

2.3. Backlash Measurements:

The angular amount of backlash of each engagement position of the gear coupling teeth was measured by using the orientation dividing head as shown in Fig. (4).

2.4. Vibration Measurements:

Vibration measurements included in this section are:
a- Torsional vibration of the rotor system.
b- Transverse vibration transmitted to the bearing support.

Vibrations are measured for each amount of backlash by using the measuring apparatus shown in Fig. (5). Teeth replacement is carried out by means of the female gear flange coupling to give different amounts of backlash.

3. EXPERIMENTAL RESULTS:

The experimental results are obtained by analyzing the recorded waves of the measured signals to give the real values. A sample of the recorded charts is given in Fig. (6).

The effect of backlash on the torsional vibro-characteristics has the same behaviour during the starting and stopping periods of the rotor system as shown in Fig. (7). Torsional vibration during the starting and stopping periods, resulting from the difference between the inertia torques of the driving and driven units, has minimum amplitudes corresponding to amounts of backlash ranging from 0.009 to 0.0125 rad. The dynamic factor of torques for each amount of backlash is determined and illustrated graphically in Fig. (8).

Transverse vibrations of the rotor system in the vertical and horizontal directions, which may related to teeth errors, are shown in figures (9) and (10). Transverse vibration in the two directions has no definite behaviour over the wide range of backlash because it is dependent on the resolution of the impact force in these directions.

CONCLUSION:

It may be concluded that:

1. The minimum torsional vibration amplitudes occur at values of backlash ranging from 0.009 to 0.0125 rad. Out of this range the torsional vibration has high amplitudes.
2. It **may be** recommended that all the engagement positions of the gear coupling teeth which give values of backlash between 0.009 and 0.011 rad. should be marked on both male and female parts, so that the gear coupling unit can be used for longer service life.
3. It may also be recommended that the gear coupling unit must be out of work when the amount of backlash of all the engagement positions reaches 0.011 rad., since the dynamic factor at this value is about 1.5 which is acceptable in machine element design.

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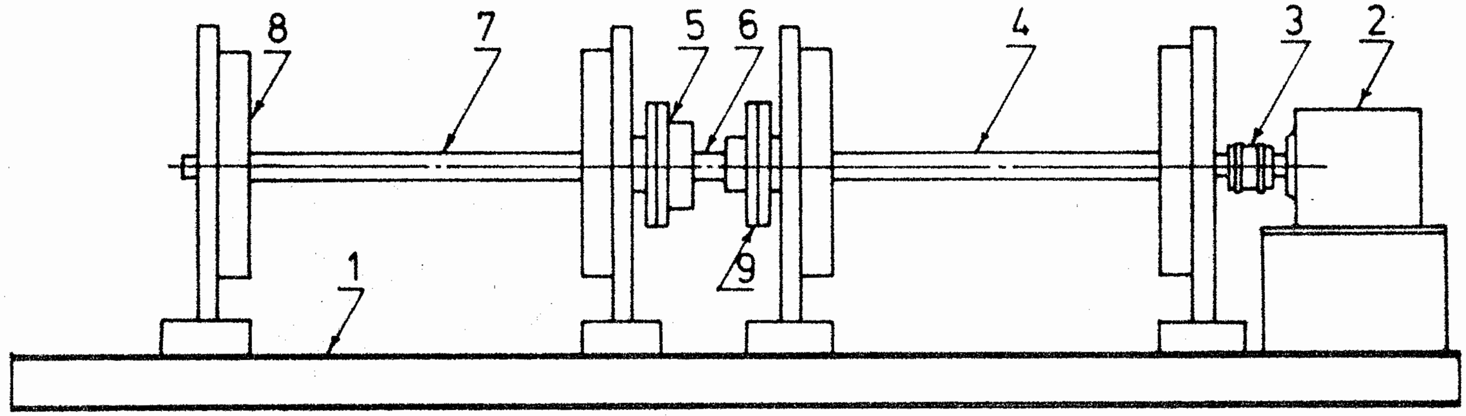
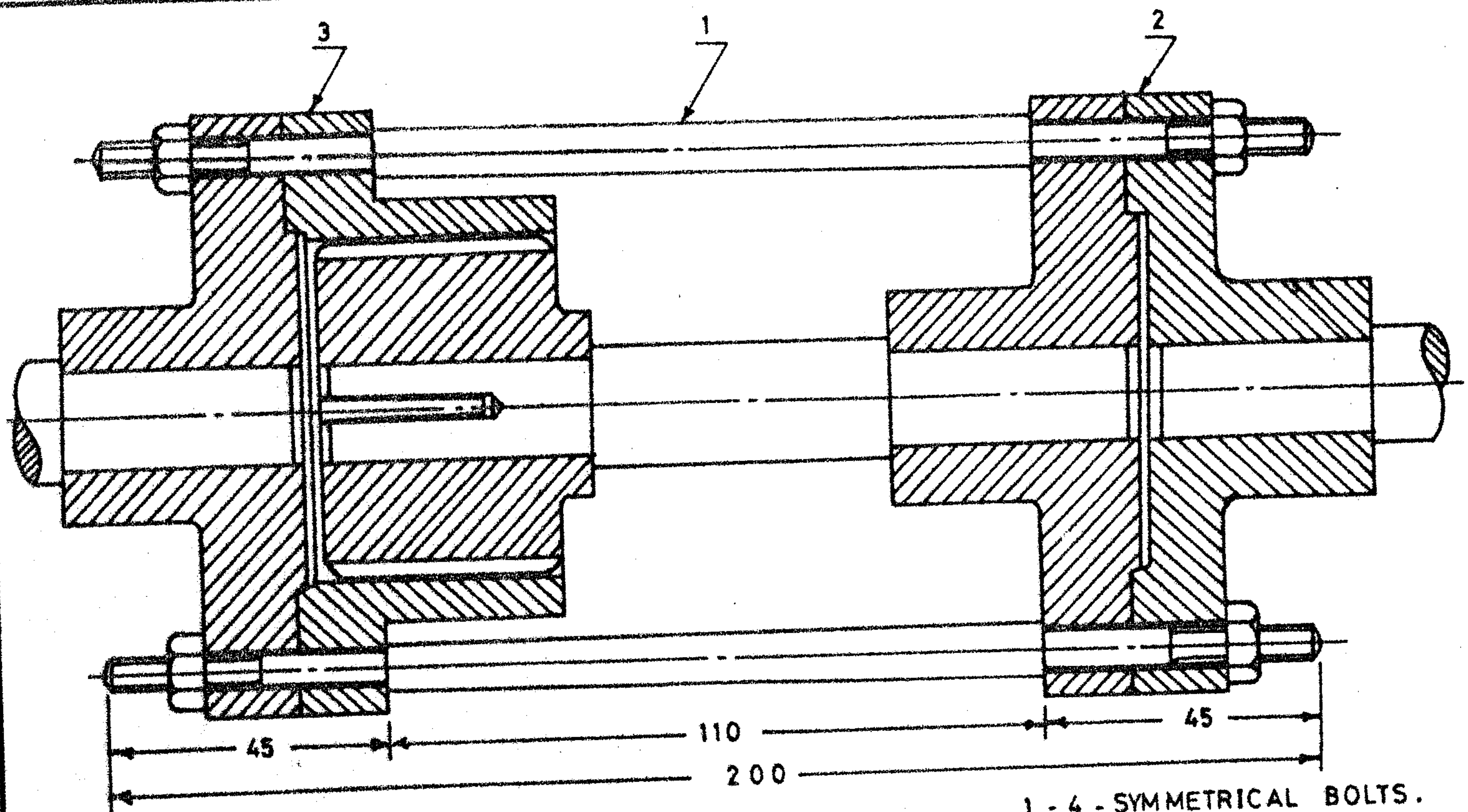


1 - Base.
2 - Electric Motor.
3 - Flexible Coupling.
4 - Driver Rotor.
5 - Gear Coupling.
6 - Coupler Rotor.
7 - Driven Rotor.
8 - Bearing Support.
9 - Flange Coupling.

FIG.(1) TWO ROTOR SYSTEM CONNECTED BY GEAR COUPLING.



1 - 4 - SYMMETRICAL BOLTS.
2 - MALE GEAR FLANGE COUPLING.
3 - FEMALE GEAR FLANGE COUPLING.

FIG.(2) GEAR COUPLING CONNECTION.

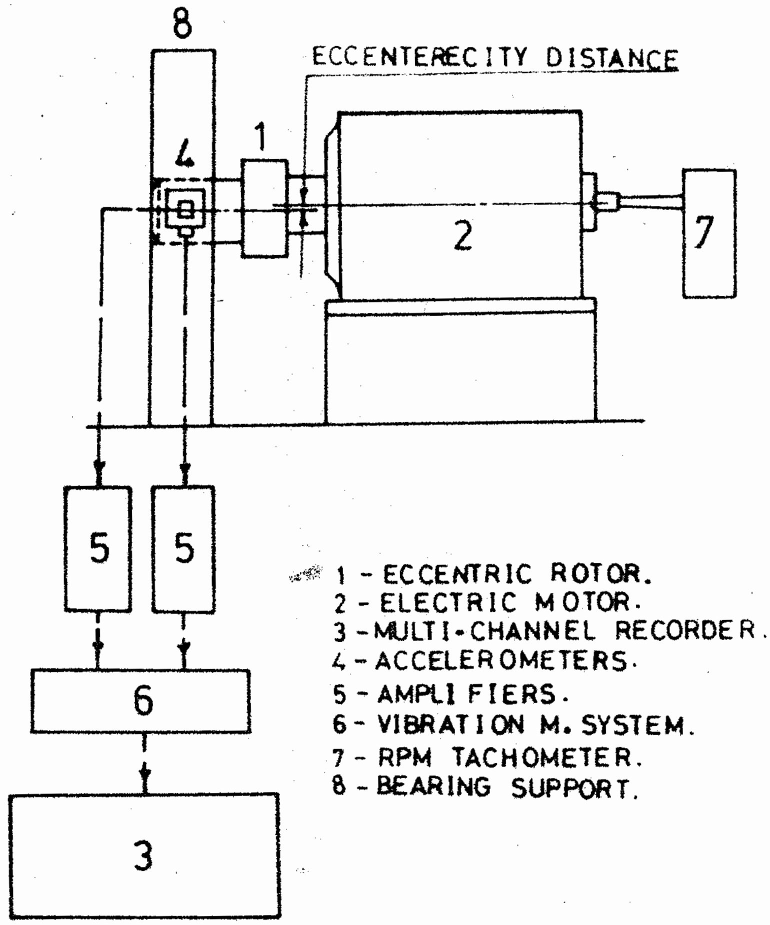
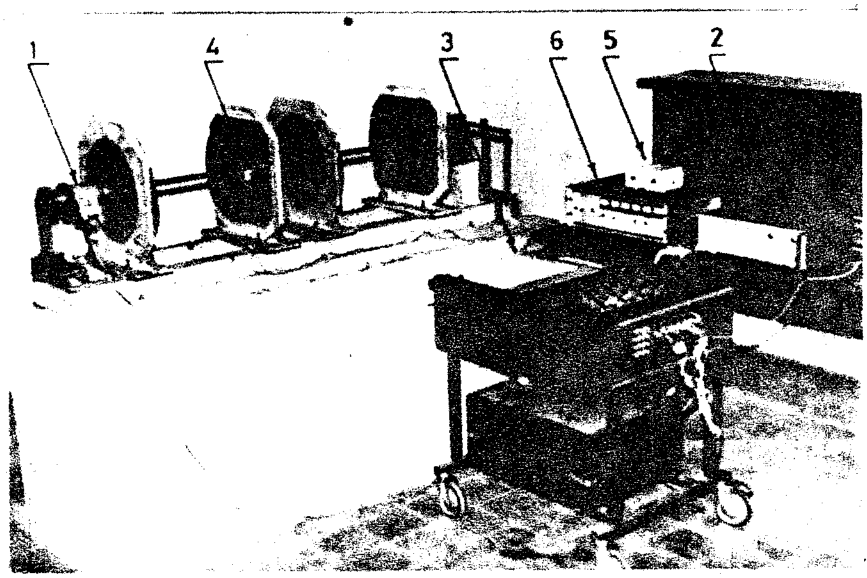


FIG.(4-10) SCHEMATIC DIAGRAM FOR MEASURING NATURAL FREQUENCIES OF THE BEARING SUPPORT.



1 - Torque Transducer.

2 - Bridge.

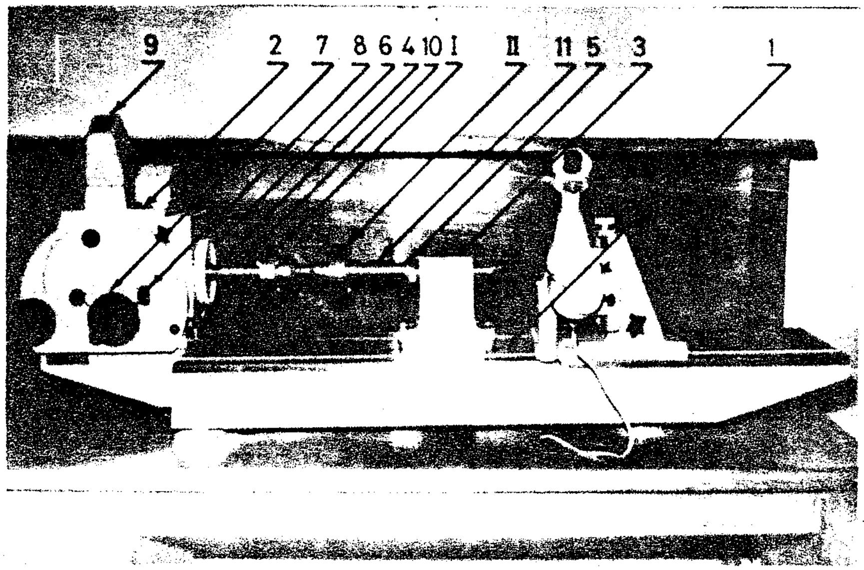
3 - Multi-Channel Recorder.

4 - Accelerometers.

5 - Amplifiers.

6 - Vibration Measuring System.

FIG.(4) MEASURING SYSTEM.



I - Female Gear.

II - Male Gear.

1 - Base.

2 - Optical Dividing Head.

3 - Tailstock.

4 - Dividing Head Centre.

5 - Tailstock Centre.

6 - Orientation Scale.

7 - Handwheel.

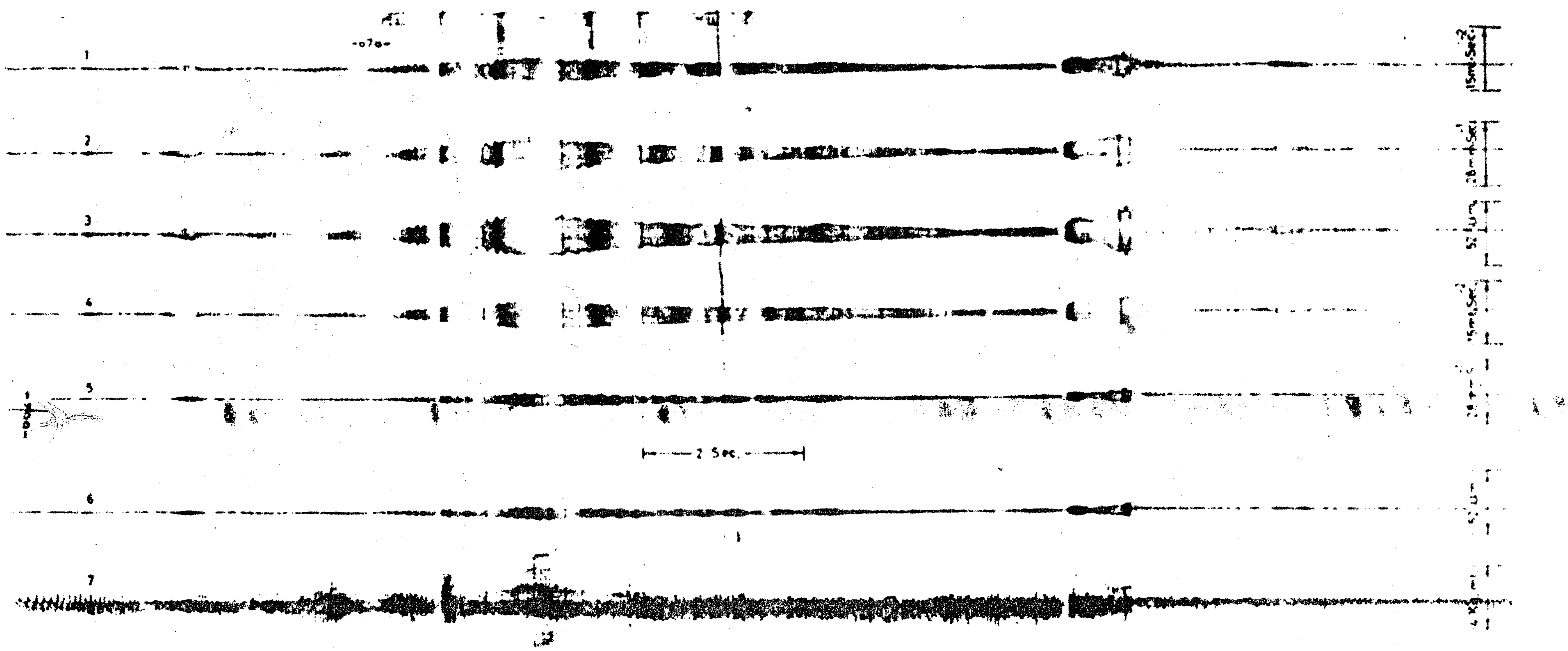
8 - Knurled Knob.

9 - Projection Screen.

10 - Fixture.

11 - Fixture.

FIG.(5) BACKLASH MEASUREMENT.



Back-lash = 0.0'3 rad.
n = 3000 rpm.

1-Acc. in V. direction.
2-Vel. in H. direction.
3-Disp. in V. direction.

4-Acc. in H direction.
5-Vel. in H direction.
6-Disp. in H direction.
7-Tors. Vibrating wave.

FIG.16.

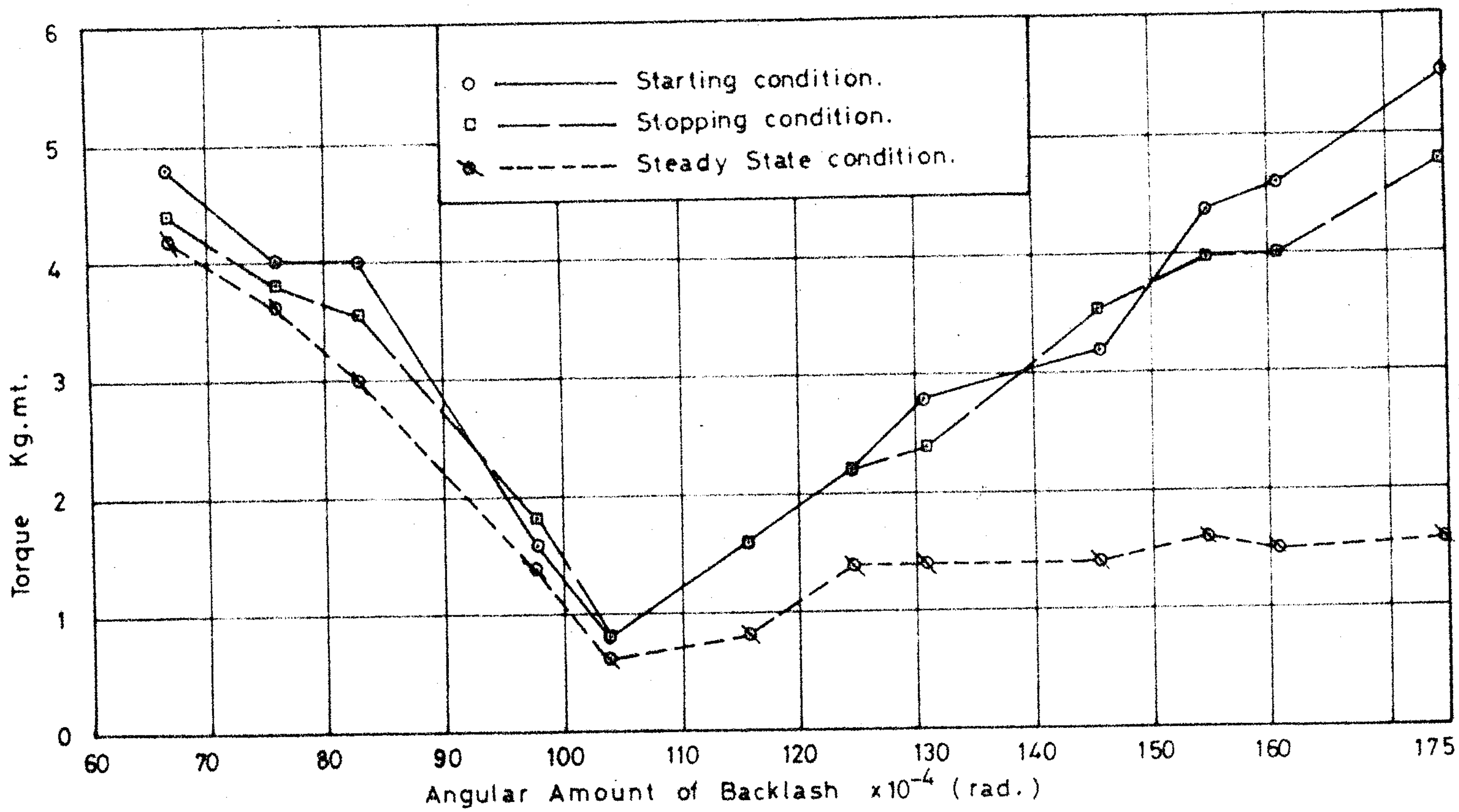


FIG.(7) TORSIONAL VIBRATION AMPLITUDE OF THE ROTOR SYSTEM.

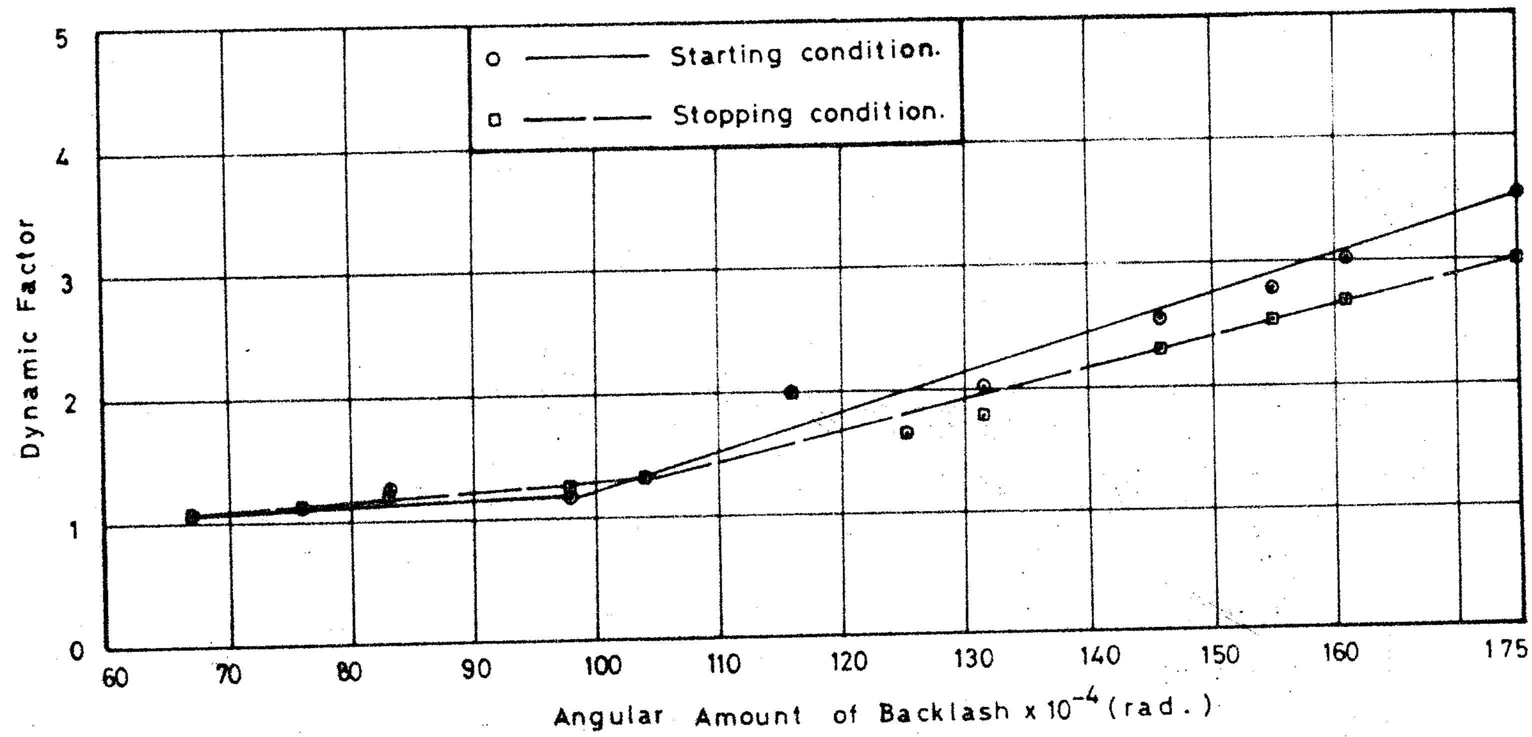


FIG.(8).DYNAMIC FACTOR OF TORSIONAL VIBRATION AMPLITUDES.

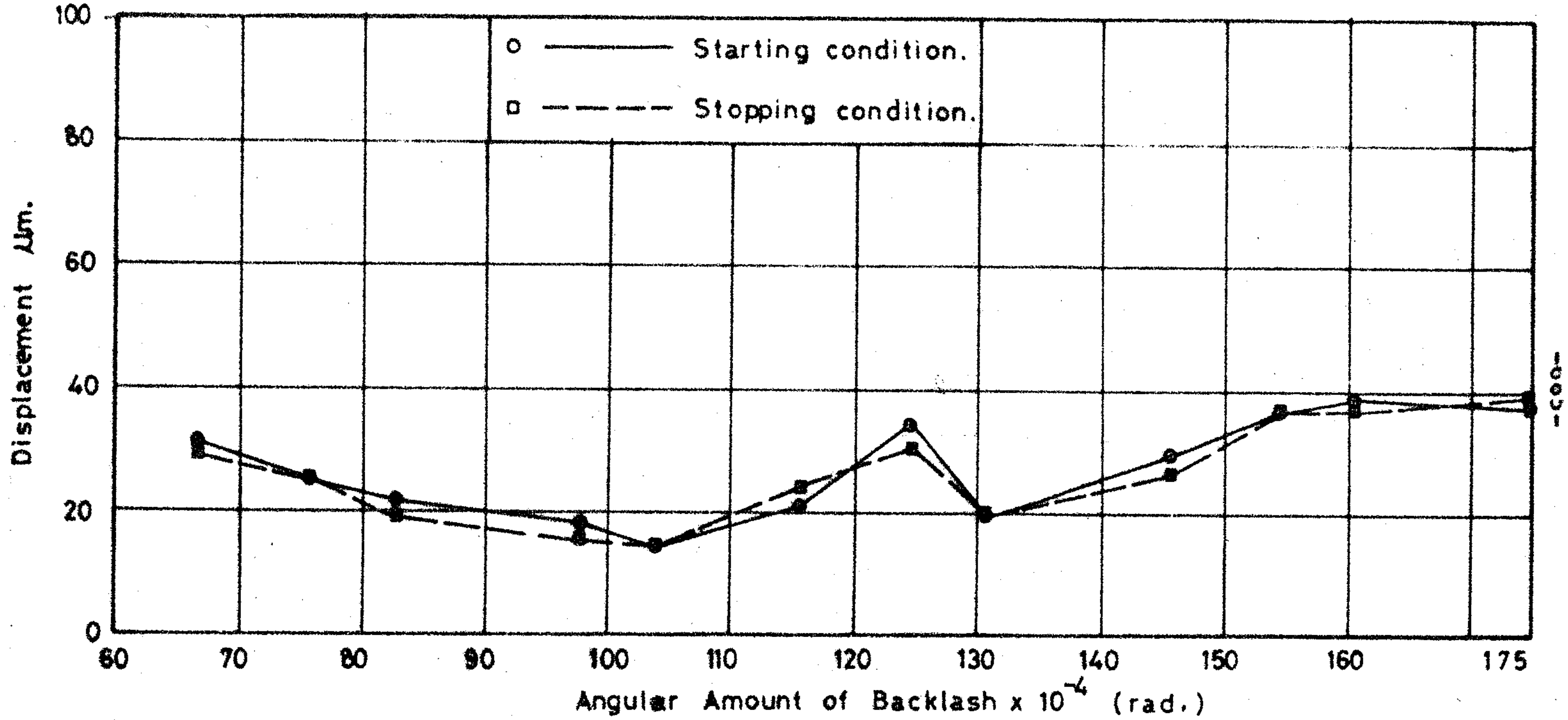


FIG.(9) TRANSVERSE VIBRATION IN VERTICAL DIRECTION.

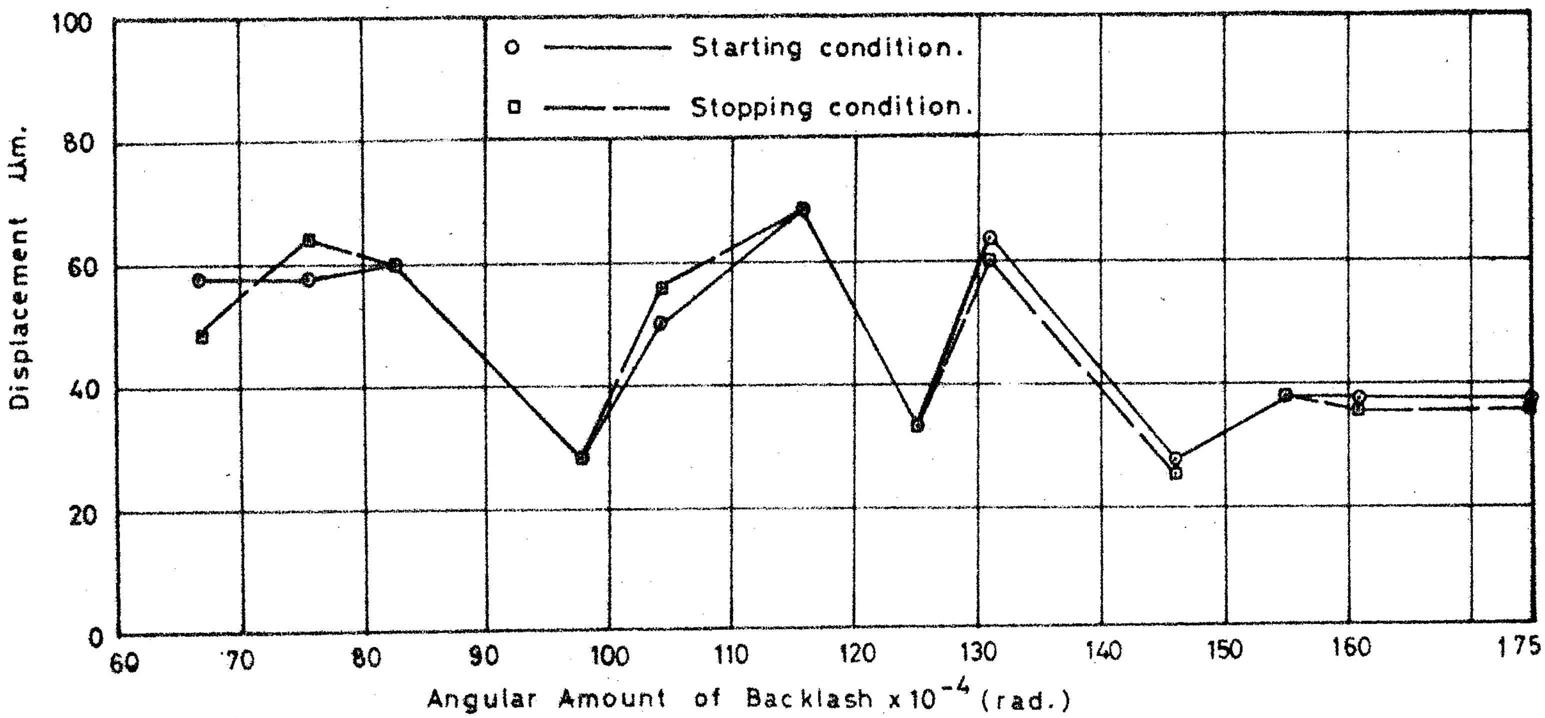


FIG.(10) TRANSVERSE VIBRATION IN HORIZONTAL DIRECTION.

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ملخـــص

تأثير الفوت الزاوى على اهتزازات العمودين الموصلين فيما بينهما بقارنه ترسية

تستخدم القارنات الترسية فى توصيل عمودين كما فى الضخات والضواغط وقد وجد أن الاهتزازات التى تتعرض لها مثل هذه الماكينات تحدث تأثير مباشر على آدائها وعمر التشغيل لها • لذلك اتجهت الابحاث لدراسة العوامل المسببه لهذه الاهتزازات وكذلك ايجـــاد الطرق المناسـبة للتغلب عليهـا •

ونظرا لضرورة وجود الفوت الزاوى (Backlash) بين اسنان جزئى القارنه الترسية فقد اتخذ هذا العامل أهمية خاصة للدراسة بهدف تقليل الاهتزازات الناتجة من القارنـــة الترســية •

ولذلك فقد تم تنفيــذ جهاز معملى للدراسة مكون من عمودين موصلين فيما بينهما بقارنــه ترسية لها قيم مختلفة من الفوت الزاوى (من ٠٠٦٧ر٠ الى ١٧٥٠ر٠ زاوية نصفقطريـــة) قياس الاهتزازات الناتجة عنها معمليا بعد التغلب على العوامل الاخرى المسببة للاهتزازات•

وكنتيجة لهذه الدراسة • تم تحديد قيم الفوت الزاوى التى يحدث عندها أقل قـــيم للعــزوم الناتجة من الاهتزازات الالتوائية وتتراوح بين ٠٠٩ر٠ و ٠١١ر٠ زاوية نصف قطرية استرشادا بالعـامل الديناميكى المسموح به فى تصميم الماكينـات وهو ٥ر١ •

وكذلك نوصى المصممين بوضــع علامات مميزة على أسنان الترسين الداخلى والخارجـــى للقارنــه الترسية بحيث تحدد الاوضاع التى يمكن التشغيل عندها بأمان واستبدال القارنـــه الترسية عنــدما تصــل قيمة الفوت الزاوى لجميــع أوضاع التعشيق ٠١١ر٠ زاوية نصفقطريـــة •