THE INFLUENCE OF EXTERNAL VIBRATIONS ON FRICTION-INDUCED NOISE

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ABSTRACT

As a trial to understand the problem of friction, noise and vibration, this study investigates the relation between friction and noise under self-excited vibration with external excitation. The measurements were made on a pin-on-disc set-up. The noise levels were recorded up to frequency of 6.4 kHz under different loads and speeds with and without external vibration excitation. A statistical analysis of the experimental results is performed under different test conditions. The results indicate that the average normal force has a significant effect on generating noise under self-excited vibration; different noise levels are observed as the average normal force is changed. The experimental data demonstrated that introducing external excitation to the system decreased both the average coefficient of friction and the sound pressure levels in comparison with the results of self-excited vibration case.

Keywords: Friction-induced noise, Vibration

INTRODUCTION

Vibration and noise in the environment or in industry are caused by particular processes where dynamic forces excite structures. Noise is simply a part of the vibrational energy of a structure. In any given situation of excessive noise or vibration levels, there are always three responsible factors which affect the problem: Source, Path and Receiver, any of these may be the cause of the problem, and can be investigated to find the corresponding reasonable solution.

FRICTION AND VIBRATION

Different types of vibrations induced by friction have been reported in the literature depending on the normal load, sliding speed, and the nature of the surfaces in contact. These vibrations may be generally classified into the following three classes : stick-slip, vibrations induced by random surface irregularities, and quasi-harmonic self-excited oscillations. Papenhuyzen [1] accurately classified friction-induced vibration into two general types, stick-slip vibration and quasi-harmonic oscillation. Godfrey [2] studied the effect of normal vibration on friction and emphasized the importance of avoiding external vibration during measurement. He reported that, vibration reduces the apparent friction force during lubricated and unlubricated sliding, vibration eliminated scuffing and increased plastic deformation. Brockley *et al.* [3] presented a theory which emphasizes the importance of the time-dependence of static friction in relation to stick-slip vibration. The analysis yields an expression for a critical velocity that limits the incidence of vibration. This critical velocity depends on damping normal load, stiffness and friction characteristics, which vary with time and velocity.

In most studies of friction-induced vibration, it is assumed that stick-slip exists because static friction is higher than sliding friction. Aronov *et al.* [4-6] conducted a

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series of tests using pin-on-disc type models. The measured friction force and velocity of the pin along the friction force were found to change randomly with time. Three different regions of operation were observed. The first is stable steady state friction process with no fluctuations. This region occurs when the normal load is low and below critical value. The second is a region of unstable intermittent self-excited vibration. The friction force suddenly increases, with high frequency modulated oscillations at 889 Hz (squeal), then decreases to a low level with a low frequency of 240 Hz (chatter). This region exists when the normal load increases beyond a certain critical value. The third region is characterized by high-frequency self-excited oscillations. Dweib and D'Souza [7], indicated that the friction force depends on the normal load for a constant sliding speed. They found that increasing the normal load causes unstable sliding motion. It was concluded that the mechanism that causes self-excited vibrations in the pin-on disc type system is coupling between its degrees of freedom. Adams [8], investigated the steady sliding of two flat elastic half-spaces against each other with a constant friction coefficient. Dynamic instabilities in the form of self-excited oscillations were found to exist for a range of material combinations, friction coefficients, and sliding velocities (including very low speeds). The mechanism of stability is that of destabilization of interfacial (slip) waves.

Ibrahim [9 and 10] presented a considerable amount of background study for dynamists studying friction-induced vibration, chatter, squeal and chaos. Different types of complex dynamic characteristics depending on the relative velocity, normal load, temperature and surface conditions. The motion may not be continuous, but may be intermittent and followed by a process of stick due to the higher static friction between the surfaces, and slip due to the lower kinetic friction during the slip itself. His survey papers showed that the vibration characteristics are system dependent in addition to the degree of coupling among friction and other degrees of freedom. Under certain conditions, the friction force acts as a medium to transfer energy from the steady state sliding motion to excite and sustain periodic limit cycle oscillations. The self-excited vibrations can cause surface damage, excessive wear, and fatigue failure of components and objectionable noise levels. Hence, a fundamental knowledge of self-excited vibrations is essential for the design of machines with frictional joints to prevent their occurrence.

FRICTION AND NOISE

Friction-induced noise varies greatly depending on temperature, brake pressure, speed and other conditions. Yokoi and Nakai [11-14] concluded that frictional noise could be classified into two categories: rubbing noise, which is generated when the frictional force between sliding surfaces are relatively small, and squeal noise which occurs when those forces are high. They have observed quasi-harmonic vibrations which they named squeal. In their investigation of the squeal noise generation mechanism of higher modes showed that squeal noise of both fundamental mode and higher one occur. Squeal noise of higher modes occurs when the area of contact is relatively large, and when the longitudinal vibration increases in rubbing noise. They found that, the frictional noise and the lateral acceleration cause higher harmonic and sub harmonic resonance because of the non-linearity such as loss of contact, between the rod and the disc, and also the sound pressure level of the frictional noise becomes higher with an increase of the surface roughness. They found that, the sound pressure level becomes higher and the loss of contact occurs with an increase in surface roughness and revolutions of the disc. They showed that, when the rod is inclined to the same direction of the rotation of the disc, rubbing and squeal noise occur and their frequencies become larger with an increase of the rod angle, when the rod is inclined to the reverse direction of the revolution of the disc, chatter noise of a high sound pressure level occurs. Nakai and Yokoi [15] showed that the friction coefficient and noise level in dB generated with the passage of friction time or the friction distance for a peripheral velocity of 0.7 m/s and normal load 14.7 N. It is seen that the noise level is relatively small, 65 dB, and is random of the rubbing type. This is generated for a small friction coefficient. As the friction coefficient increases (three or four times the initial value) for a friction distance greater than 1100 meters, the noise level suddenly jumps to about 94 - 97 dB (a screaming noise).

The objective of this investigation is to reduce the noise generated from the dynamic contact between two surfaces such as brake systems by using active noise control.

EXPERIMENTAL SET-UP AND INSTRUMENTATION

The experimental set-up illustrated in Figure 1 was constructed to study the relation between friction and noise levels with and without external vibrations. The pin tip was rotated by means of a variable speed (0-2500 rpm) 3/4 hp DC type electric motor. The flange was attached to hold the fixture. An optical encoder (4) giving signals for every revolution mounted to the motor disc (3). The optical encoder provides signals to a 10 MHz counter (HP model 5301A) to measure rotational speed in revolutions per minute (rpm). The motor is connected to AC/DC converter and speed controller. The fixture consists of a battery holder/counter weight for balancing purpose (5). An electrical four-channel slip ring coupling (6) and a spring (7) which loads contact point with and through the I-beam (8). The spring has a stiffness of 15.44 N/mm^2. The construction of the pin fixture is such that a threaded bar with key way is connected with the motor disc by means of a thread. A movable hexagon nut (11) compresses the upper surface of the spring. The lower surface of the spring rests on a square base (35×35×7 mm) which is welded to a shaft with a pin. The threaded bar moves up and down with the key way as a guide due to the movement of the spring. The square base is connected to 1045 steel I-beam via 4 small screw bolts. The 80-mm I-beam has cross-section area (50 mm^2). The ball holder is connected to the end of the I-beam via 4 small bolts. An industrial tool ball (10) with hardness of 58:62 HRC and a diameter of 12.7 mm, was welded at the center of the ball holder (20×20×5 mm). The center contact between the ball and disc surface is 47 mm away from the center of the disc. The specimen consists of a circular disc of 1018 cold rolled steel with diameter of 127 mm and a thickness of 20 mm. The specimen surface (10) was hardened to (58:62) HRC and surfaced and hand polished with 600 grit wet in a circular pattern to a mirror finish of 0,025 R_a (roughness average). The specimen resting on the specimen holder (14) of 160 mm diameter and 40 mm thickness, with 3 dowel pins to prevent rotation, but allowing vertical movement for the shaker (15) which is connected to the specimen. The height adjusting brackets holding that specimen are moved using 3 threaded rods (2) to adjust parallelism to the motor flange with an accuracy of 0.5 μm and then to firmly lock in place with 3 set screws and jam nuts (16). The parallelism between the specimen surface and the motor disc is adjusted using dial-gage with mini stand base. The entire apparatus is leveled and bolted to a fixed stand heavy base (17).

Figure 2 shows the block diagram of the instrumentation. The friction and normal forces at the contact point are measured via four strain gages, installed on the two faces of the I-beam. Two of the strain gages are sensitive to bending and are used for measuring the frictional force and the other two are sensitive to compression and are used for measuring the axial force. The input power supply to the strain gages comes through two wires stuck to both of the pin fixture and the motor disc and connected to the balancing weight which contain three 9 volt batteries. Two batteries have been connected to the strain gage excitation circuits to provide 5-volt excitation for the strain gages. Excitation voltage is kept constant with a voltage regulator. The third battery provides power to the axial strain gage amplifier circuit.

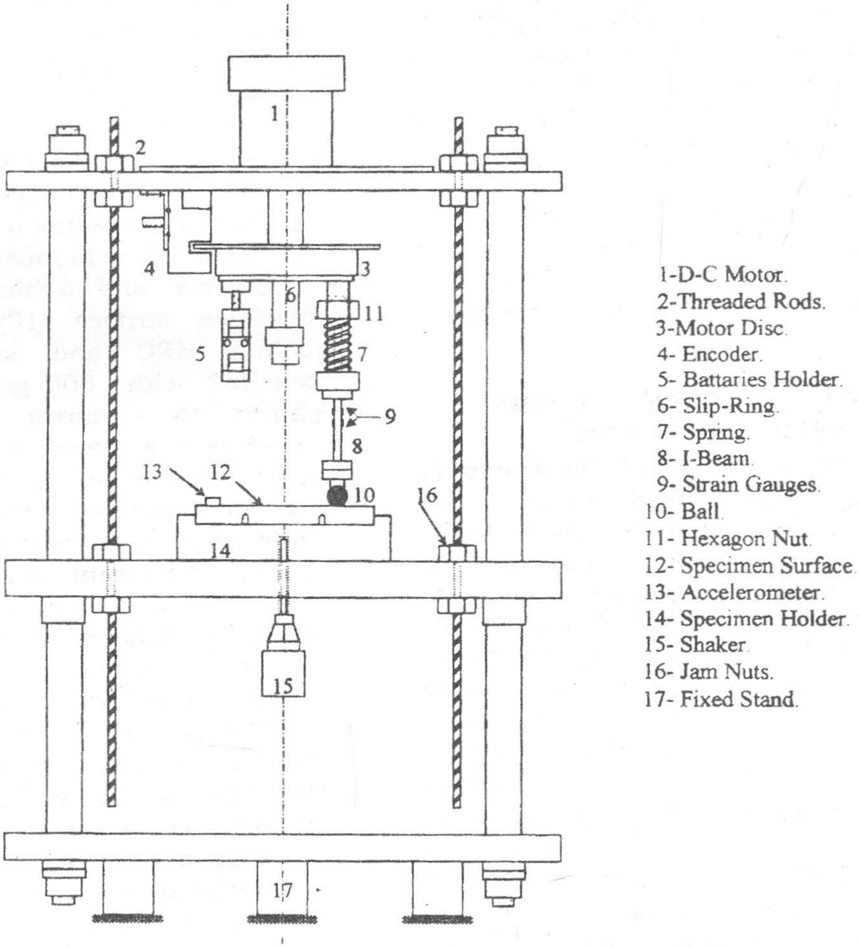

Figure 1 Pin-on-disc Experimental Set-Up

The axial strain gage signal was amplified via an AD624 amplifier with a gain of 500. The output of the strain gages goes through a slip ring, which has two parts, a rotating part connected to the motor disc and stationary part. The signals are transferred from the rotating part to the stationary part of the four-channel slip ring. The output of the strain gage signals will go via wires passing through a specific hole on the specimen surface and to the data 6100 digital oscilloscope. The axial and bending signals represent the required information about the applied normal load and the friction force. An IBM computer together with the oscilloscope, recorded the signals and stored than in computer files. The output signals were later plotted in time domain.

All measurements were conducted in an anechoic chamber, to avoid the effect of back ground noise on measurements.

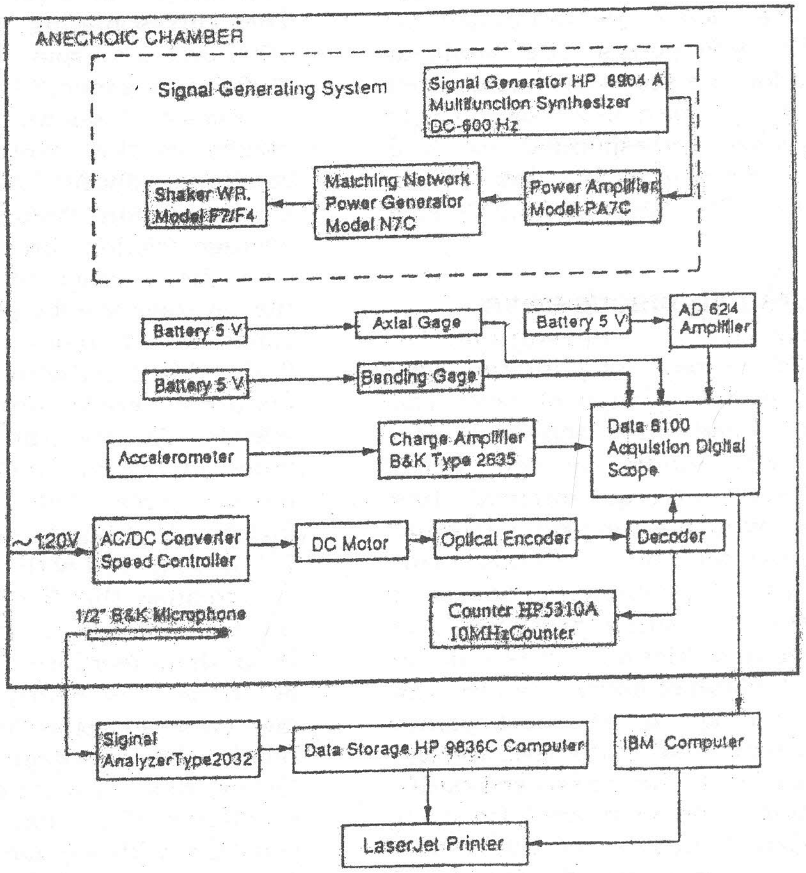

Figure 2 Block diagram of the experimental set-up

For measuring the noise presented here by sound pressure level (SPL), 1/2 inch B&K microphone was held at a distance of 10 cm from the ball, and 50 cm from the ground of the anechoic chamber. The microphone was connected to a B&K signal analyzer model 2032, which recorded the noise in both frequency and time domains. The HP9836C computer read the data and generated plots which were printed out by an HP Laser-jet printer and IBM computer. An electromagnetic shaker system has been attached to the set-up, for introducing external vibrations at different amplitude and frequencies.

The Model F4 Shaker was fed from one of the two low impedance output channels of the PA7C power amplifier. The PA7C power amplifier had a 15 Hz (-3 dB points) high pass filter. The HP8904, a multifunction synthesizer DC to 600 Hz, was connected to the Power Amplifier to set the frequency and the amplitude of vibrations. To record both the self-excited and external vibrations, an accelerometer B&K model 4393 has been mounted on the disc surface. The accelerometer signal was carried via its cable, which pass through a specific hole on the disc surface and away from the pin's track. The cable was connected to a charge amplifier B&K model 2635. The output of the charge amplifier went to a data 6100 digital scope, as shown in figure 2. To record both the sound pressure and the strain gage output signals; a triggering circuit has been used. The change in rpm required changing the time periods to match the same starting triggering point.

The triggering circuit assumed that the data were collected from the same 37.5

degree sector of the disc surface. The time periods for the scope are calculated for collecting 512 data points. The sampling times needed for the scope were calculated to be 0.244, 0.122, and 0.061 ms. These sampling periods corresponded to 37.5 degrees disc sector and 512 data points were taken at 50, 100, and 200 rpm respectively.

RESULTS AND DISCUSSIONS

This experimental investigation of friction-induced noise and vibration was conducted under two groups of tests. The first group contained three series of tests; these series were under the self-excited vibration. The second group contained three series of tests, which were under external vibrations generated by a shaker with different ranges of frequencies. For each series of tests, the sound pressure level versus normal and friction forces will be presented. Each value of normal and friction force is the average of 512 data points acquired simultaneously with the average sound pressure level. The measured results will be examined and compared for each group. A statistical analysis of the results will be used to analyze the distribution of normal and friction forces and sound pressure levels under different conditions. The applied normal load was in the range 15-98 N. The friction force depends on the normal force for a constant rotational speed. The corresponding surface speeds to 50, 100, and 200 rpm, are 0.246, 0.492, and 0.984 m/s respectively.

Figure 3 shows the scattering of data points of the average forces at different rotational speeds. At 50 rpm a nonlinear friction region was observed in which the average friction force increases nonlinearly with the average normal force. In this case, the average coefficient of friction was not constant. It fluctuated between 0.37 and 0.80. This fluctuation is due to the surface texture, average normal force and rotational speeds. At 100 rpm, the average friction force increases linearly with the average normal force. The average coefficient of friction fluctuated in a small range, 0.496 to 0.758 compared with the range at 50 rpm.

Because the fluctuation range is small, the relationship between the average forces is in the linear form. The average coefficient of friction is nearly constant which may mean a steady-state friction. At 200 rpm, the relationship between the average of the forces was also linear, and the average coefficient of friction was very close to be constant with a range of 0.399 to 0.538. The three ranges of the average coefficient of friction are decreased as a result of increasing the number of revolutions from 50 to 100 to 200 rpm.

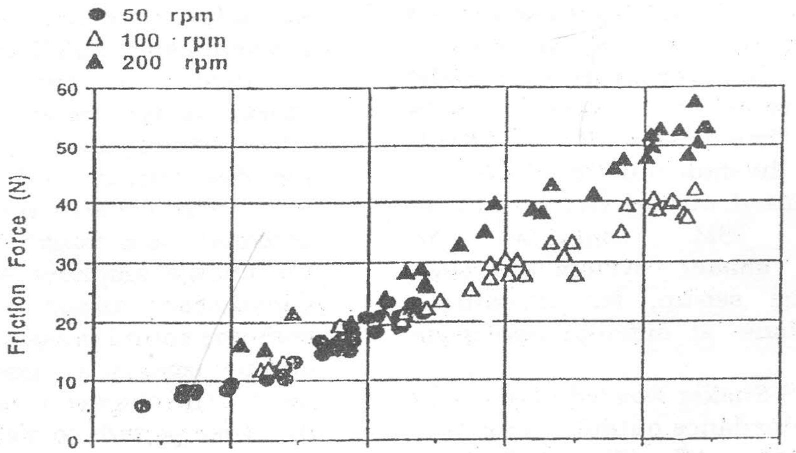

Figure 3 Average friction and normal forces at 50, 100 and 200 rpm.

Figures 4 and 5 show the scatter-plot of sound pressure level versus the average normal and friction forces at different rotational speeds. At 50 rpm, as the average normal forces increase from 7 to 48 N, the average friction forces increase approximately linearly from 5 to 25 N. For those ranges of the average normal and friction forces, the sound pressure levels vary from 69 to 99 dB(A) in a nonlinear manner. At 100 and 200 rpm, as the average normal forces increase, the average friction forces and the sound pressure levels increase linearly in the sound pressure range of 100 to 107 dB(A) at 100 rpm, and from 91 to 106 dB(A) at 200 rpm.

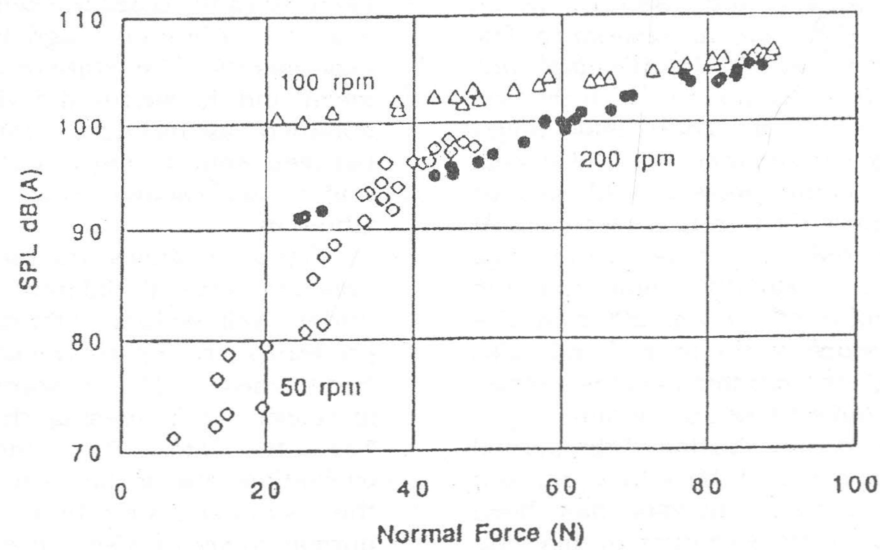

Figure 4 SPL dB(A) versus average normal force at different rotational speeds.

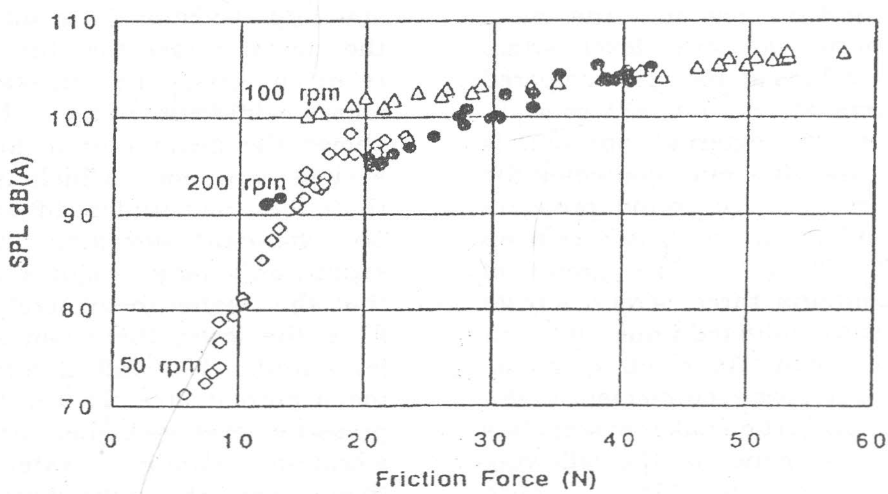

Figure 5 SPL dB(A) versus average friction force at different rotational speeds.

Frequency spectra of the noise signals from the microphone for the range 0-6400 Hz for 50, 100 and 200 rpm are given in Figure 6. In the range 1000 - 2300 Hz, there are two peaks. These two peaks are seen at 50 and 100 rpm, the first peak at 1085 Hz and the second at 2200 Hz. The amplitude difference between the two peaks is very small. At 200 rpm, the peaks obtained at 50 and 100 rpm are missing, but other sharp peaks appear at 970 and 1885 Hz, which may indicate other system resonance. The first peak is shown at 970 Hz with amplitude lower by 12 dB(A) compared with the first peak at 50 and 100 rpm. These peaks might be the resonance frequency of the system set-up broad band noise could also be observed to have risen in this spectrum. At 100 rpm, the broad band noise in the range below 6400 Hz is slightly higher than the broad band noise at 50 and 200 rpm. The main contribution to the broad band noise which express the random noise is a result of the friction and surface interaction.

To analyze the distribution of the normal and friction forces and the sound pressure levels, a statistical analysis has been conducted to show the variation at different rpm, and the mean of the sound pressure level corresponding to both forces also at 50, 100, and 200 rpm.

Figures 7 and 8 show that the mean forces and sound pressure level attain relatively high values at 100 rpm compared with the others at 50 and 200 rpm. To investigate how the external vibration is going to affect the dynamic contact of the system, with respect to sound pressure level, the second group of tests have been conducted at 100 rpm. This group of investigation contains three series of tests and each one was conducted under different excitation frequencies. An electromagnetic shaker system has been connected to the bottom of the disc. The shaker system has been adjusted to generate the following frequencies 135, 185 and 235 Hz. These frequencies have been chosen after running a lot of tests under different ranges of frequency. These particular excitation frequencies gave a reduction of the sound pressure for this particular test rig. One of the difficulties was the excitation voltage of the shaker; by increasing this voltage above 5 volts, the shaker itself generates considerable noise even at low frequencies.

The excitation voltage of 5 volts has been selected and held constant during all tests. This excitation gave a constant displacement for the chosen frequencies. An accelerometer mounted on the disc surface, has been used to record the self-excited and the external vibration signals through the experiments. The statistical results of the mean and the standard deviation of the data obtained at 100 rpm show the difference between sound pressure level in two cases; under self-excited and under external vibration.

Figure 9 shows the mean of the sound pressure level at different frequencies and under self-exciting vibration. The mean pressures are nearly the same for the three frequencies. The standard deviation increases by increasing the frequency from 135 to 235 Hz. Under self-excited oscillation, the mean is slightly higher and the standard deviation is smaller. The normal force as a significant parameter has been investigated from the same point of view. Figure 10 shows the mean of the normal force under 135 to 235 Hz and its standard deviation. The difference between the mean under the three frequencies is relatively small and the standard deviation shows different ranges. Figure 10 also shows the mean normal force under self-excited vibration, which is slightly lower than the ones under different frequencies. The standard deviation in this case is significantly large. Figures 9 and 10 show that the higher the generated mean normal force the lower the mean sound pressure level under 135 and 235 Hz. Although the mean normal force is lower, the mean sound pressure level is higher under self-excited vibration. Under external excitation frequencies, the range of mean normal force is high and generated lower mean of sound pressure level compared with the one under self-excited vibration. The relationship between the average normal and friction forces at different frequencies at 100 rpm are shown in Figure 11. The three nonlinear friction regions are a response of the external excitations 135, 185 and 235 Hz. The differences between these three frequencies are small and limited. The average coefficient of friction under the different excitations fluctuate in a nearly steady-state range: 0.385 to 0.551.

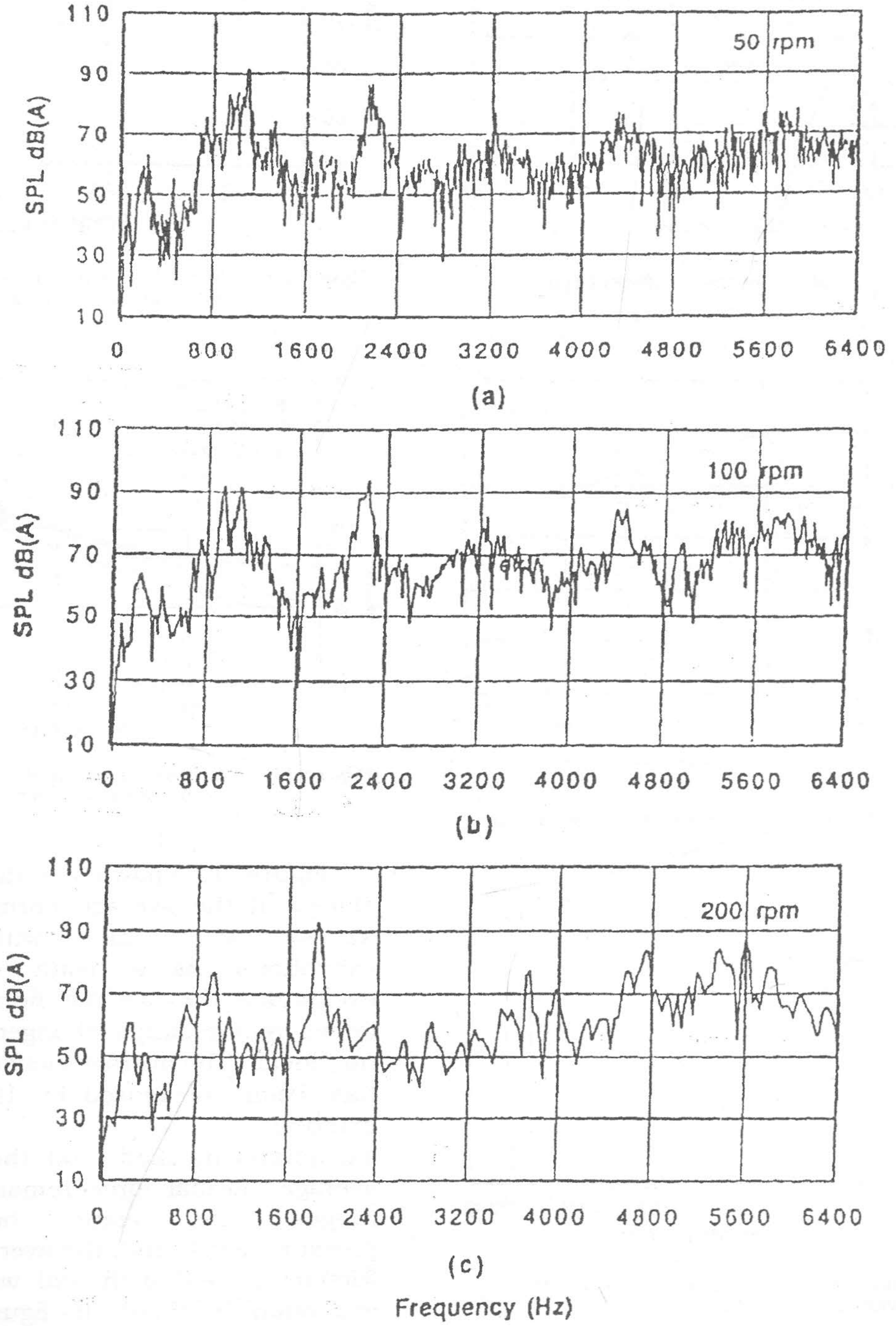

Figure 6 Sound pressure level versus frequency at different rotational speeds.

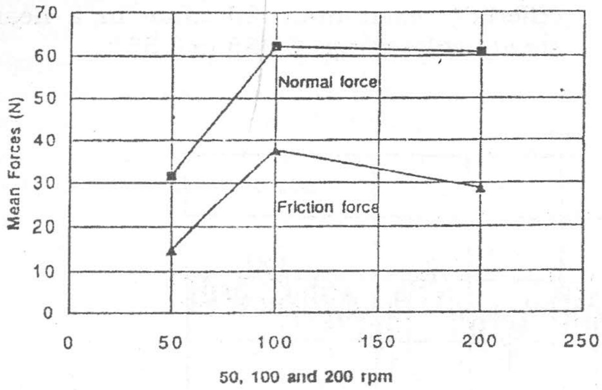

Figure 7 Mean forces at different rpm.

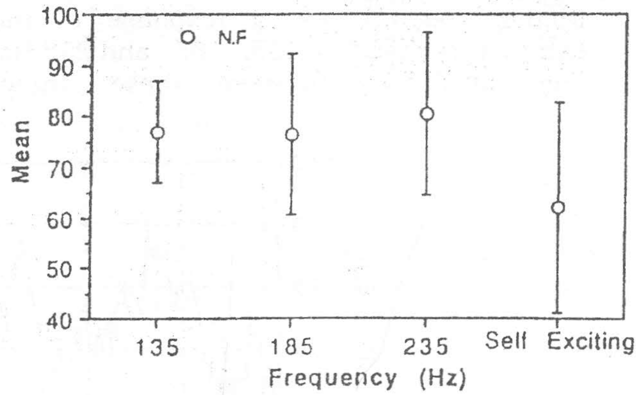

Figure 10 Normal force mean and standard deviation at 100 rpm.

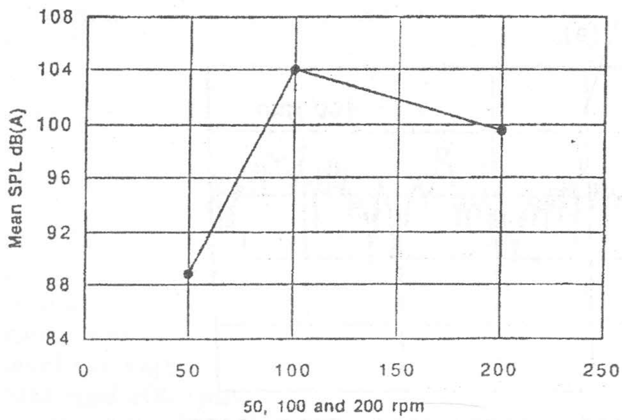

Figure 8 Mean sound pressure levels at different rpm.

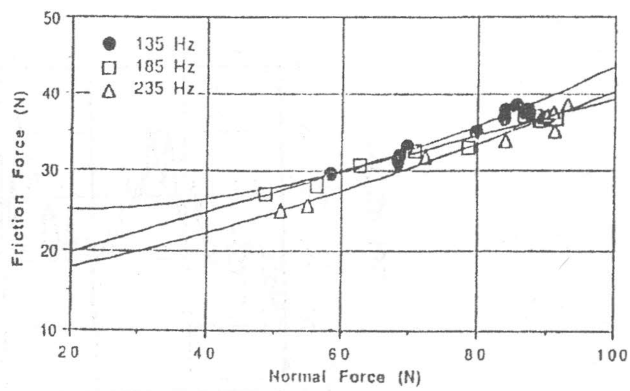

Figure 11 Average friction and normal forces at different frequencies.

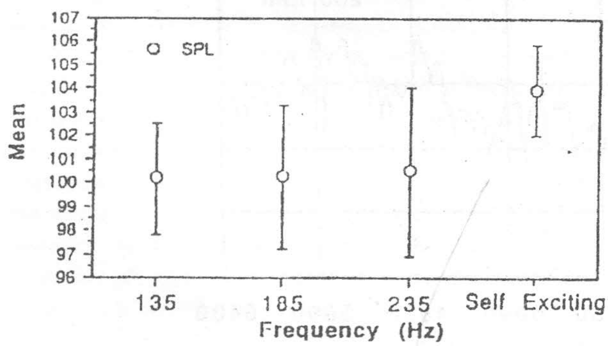

Figure 9 Sound pressure level mean and standard deviation at 100 rpm.

Figure 12 shows the difference in the slopes of the average normal and friction forces, with and without external excitations. As a result of the external excitations the average normal and friction forces relationships changed from linear to nonlinear. The average coefficient of friction has been decreased by 19-25 % as the external

excitations invoked, and the values of the average normal force remain at the same ranges. The relation between sound pressure level and, the average normal and friction forces with and without external excitation is shown in figures 13 and 14 respectively.

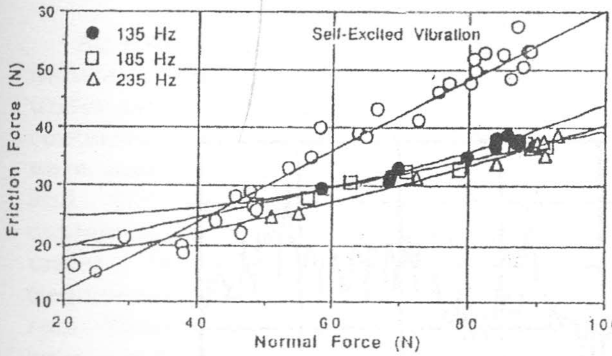

Figure 12 Average friction and normal forces with and without external excitation.

From Figure 13 and for the range of average normal force, the three external excitations generated sound pressure levels in the range of 94 - 103 dB(A). On the other hand, the average normal force, which was under self-excited vibration generated sound pressure level in a range of 102 - 107 dB(A). Invoking the external excitations at the frequencies 135, 185 and 235 Hz decreased the sound pressure level compared to the one under self-excited vibration. Based on the experimental tests, the reduction in the overall sound pressure level was in the range 3 - 8 %. The output signals of the accelerometer were also recorded in addition to the signals from the strain gages. All the signals are in frequency domain and at 100 rpm. Frequency spectra of the noise signals from the microphone for the range 0 - 6400 Hz for the three external excitation frequencies of 135, 185 and 235 Hz for the case of 100 rpm are presented in Figure 15. In the 1880-5720 Hz frequency ranges, there are three peaks. These three peaks are seen at 1886, 3771 and 5714 Hz. The amplitude difference between the three peaks is very small. Although the external excitation introduced by three different frequencies, the trends of the three spectra are the same. This may indicate that, addition of the shaker did not change the resonance frequency of the system set-up as the number of revolutions did. Broad band noise can also be observed to have the same trend in these spectra.

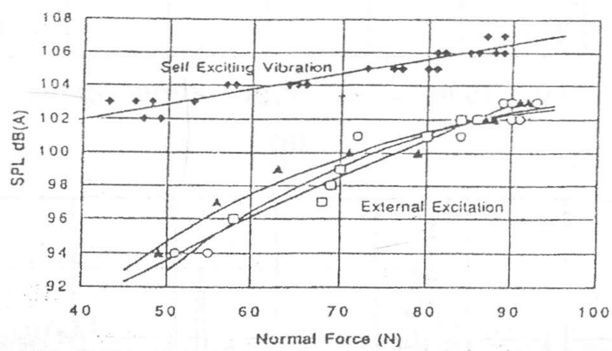

Figure 13 SPL dB (A) versus normal force with and without external excitation.

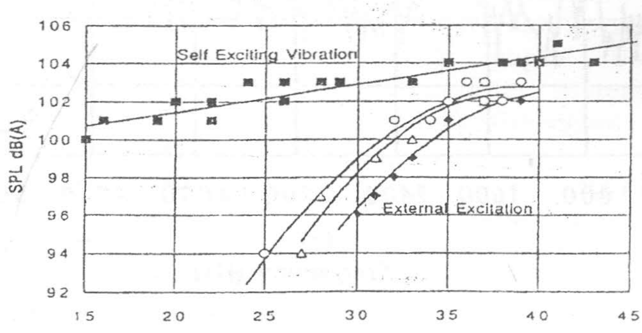

Figure 14 SPL dB (A) versus average friction force with and without external excitation .

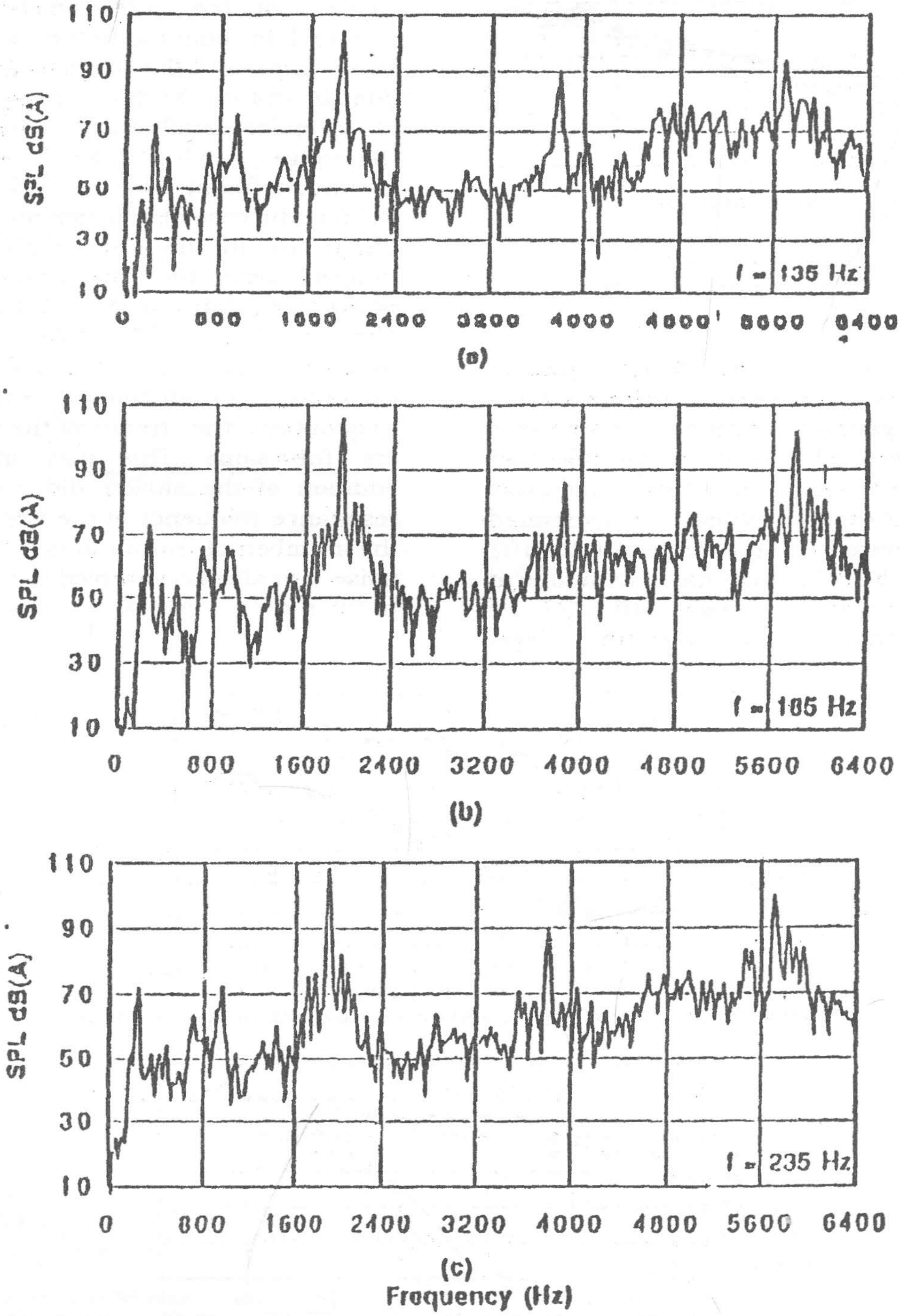

Figure 15 sound pressure level versus frequency at different external exciting frequencies at 100 rpm.

CONCLUSIONS

An experimental investigation of friction-induced noise and vibration was conducted under two groups of tests. The first group contained three series of tests, these series were under self-excited vibration at 50, 100 and 200 rpm. The second group also contained three series of tests, which were under the three external excitation frequencies of 135, 185 and 235 Hz at 100 rpm. These particular excitation frequencies gave a reduction of the sound pressure level for this particular test rig. The conclusions concerning this study are as follows:

1- The three external frequencies 135, 185 and 235 Hz were chosen to excite the disc with constant 5-volt amplitude. Compared to the results of the first group at 100 rpm, the results of the second group showed that the average coefficient of friction under the three different excitation frequencies fluctuate in a nearly steady state range. On the other hand, as a result of the external excitations the average normal and friction forces relationships changed from linear to nonlinear. Also, the average coefficient of friction was decreased by 19-25 %. Under the external excitation, the generated sound pressure levels were less than the one generated under self-excited vibration at 100 rpm by 3 - 8 %.
2- The statistical analysis showed that, under external excitation, the higher the mean normal force, the lower the mean sound pressure level. Although the mean normal force is lower, mean sound pressure level is higher under self-excited vibration.
3- The addition of a shaker did not change the resonance frequency of the test set-up. Under self-excited vibration, changing rotational speeds changed the resonance frequency of the system set-up.
4- The external excitations may reduce the contact between the peaks of the asperities of the metal in the dynamic contact. This reduction in contact may decrease the area of contact between the two surfaces, which reduce the range of the average coefficient of friction. As a result of the decrease in the average coefficient of friction, the sound pressure levels decreased. Comparing the results between the self-excited vibration tests and external excitation tests also showed that with the external the excitations the chance to increase the average normal force with lower sound pressure level is higher than the condition of self-excited vibration.

RECOMMENDATIONS

The present study can be extended to future research for a better understanding of friction-induced noise. Some suggestions are:

1- Investigation of the materials of the brakes under external excitations.
2- Searching for suitable excitation frequencies and amplitudes which might decrease the squeal level.
3- How the external excitations may be introduced to the brakes system.
4- Response of the dynamic contact when lubricant (liquid, solid and spray form) is introduced to the system.
5- Effect of the dirt on the performance of the dynamic contact and noise.

There is a strong need for further research to promote our understanding of the various friction mechanisms and to provide designers of sliding components with better guidelines to minimize the deteriorating effects of friction in the future. Some modifications to the set-up are recommended such as using piezoelectric force transducers or laser techniques to measure the dynamic forces instead of the strain gages. Using a wireless radio transmitter instead of the slip ring can transfer the signals from the rotating pin.

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تأثير الاهتزازات الخارجية على الضوضاء المولدة بالاحتكاك

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ملخص البحث

فى محاولة لتفهم مشكلة الاحتكاك والضوضاء والاهتزازات، فان هذه الدراسة تبحث فى العلاقة بين الاحتكاك والضوضاء في حالتى الاهتزازات الذاتية والأهتزازات القسرية. أخذت القياسات من جهاز التجربة وهو عبارة عن (قلم يتلامس عموديا مع قرص). سجلت مستويات مختلفة للضوضاء تحت أحمال عمودية وسرعات مختلفة فى مجال تردد يصل إلى ٤,٦ كيلو هرتز.

كذلك أيضا تم تسجيل قيم القوى العمودية والاحتكاكية فى كلا من حالتى الاهتزازات الذاتية والاهتزازات القسرية.

تم عمل تحليل إحصائي للنتائج العملية التى أنجزت. أوضحت النتائج أن متوسط القوى العمودية له تأثير واضح على توليد الضوضاء فى كل من حالتى الاهتزازات الذاتية والاهتزازات القسرية حيث لوحظ أن مستويات الضوضاء تتغير بتغير متوسطات القوى العمودية.

أوضحت النتائج العملية التأثير بأهتزازات قسرية على القرص يقلل كلا من متوسطات معامل الاحتكاك ومستويات الضغط الصوتى (الضوضاء) بالمقارنة للنتائج التى سجلت فى حالة الاهتزازات الذاتية.