NOISE GENERATED BY LATHES - EFFECTS AND CONTROL AT SOURCE

J.R. Rosinski

Design Unit, Department of Mechanical, Materials & Manufacturing Engineering, University of Newcastle upon Tyne, Claremont Road, Newcastle upon Tyne, NE1 7RU, England.

1. INTRODUCTION

The challenge created by the Open European Market in 1992 will force many machine tool manufacturers to build their machinery with significantly reduced noise levels.

A survey of noise generated by different makes of lathes indicates that there are many sources of noise generation common to these machines. One of the typical problems experienced is the intermittent gear noise which occurs randomly on nominally identical newly built units. The most common action undertaken in such cases is to increase gear accuracy and to carefully design profile modifications to minimise transmission errors under certain loading conditions. An improvement in gear accuracy, however, rarely brings the expected noise reduction.

In this paper vibroacoustic diagnostics is demonstrated to be a very useful tool for noise troubleshooting in lathe drive systems.

A case study is presented in which different noise sources are positively identified, thus allowing the noise problem to be controlled at source.

2. TYPICAL NOISE PATTERN

Noise acceptance criteria during quality control procedure for lathes require noise measurements at idle and under load conditions at different speed ratios. It is very often found that noise generated from gearboxes is intermittent and could vary by 5-10dB for nominally identical newly built units. By randomly selecting the gears from the same batch and adopting a trial and error method, satisfactory noise levels can be obtained. This is an expensive method and can not be accepted in volume production.

Measuring noise and vibration narrow band spectra and identifying a system's dynamic behaviour seems to be the best diagnostic procedure which is capable of identifying problems and allows the noise problem to be controlled at source.

3. VIBROACOUSTIC DIAGNOSTICS

It is generally accepted that transmission errors under load and other non-linear effects (tooth separation, mechanical looseness and gear rattle) are the main reasons for gear noise excitation [1,2,3]. Very rarely, however, the gear noise is heard or measured directly from the meshing gears. Transmission loss through the cast-iron or steel fabricated gearboxes is usually high enough to significantly reduce direct sound propagation, making the secondary noise sources predominant. In this way, gear excitation is modified by a structural path, e.g. shafts, bearings, bearing housings, gearbox case, etc. Very often it is difficult to identify whether gears excite the structure with high excitation levels or whether the structure is dynamically "weak" (or resonant) and responds excessively to otherwise normal excitation levels. These structural elements can be selected to measure structural response and identify vibration path properties. One method of identifying the dynamic properties of a structure is to carry out Experimental Modal Analysis.

Before carrying out full modal analysis it is recommended to measure the Frequency Response Function (FRF) between the secondary noise sources and the excitation points within the gearcase. This can be obtained by applying a well-defined excitation at or near primary sources of vibration and measuring the response of the lathe structure. The easiest method of exciting the structure is by means of impact excitation where the structure is excited by a suitably instrumented impact hammer. This method can also be used to carry out a simplified modal analysis by measuring the amplitude of the imaginary part of FRF which, for linear systems, is proportional to the modal displacement (the so-called quadrature picking method).

A very effective method of vibration and noise control is one which deals directly with its source. Narrow-band noise spectra, measured on different makes of lathes, confirmed that the primary source of noise in the lathe systems is the gear train giving significant peaks in meshing gears tooth contact frequencies. Once the noise source is positively identified, the question remains as to whether the high level of gear excitation is due to excessive dynamic factor K_v (poor gear accuracy, high transmission error), due to sources external to gearing giving rise to gear modulation (these can be expressed by the application factor K_A [4]) or due to other effects like mechanical looseness, gear rattle, etc.

Detailed analysis of narrow-band spectra, combined with cepstrum analysis [5] and envelope analysis gives a powerful tool for gear vibration diagnostics.

The case study will be presented for the lathe drive system which demonstrates how vibroacoustic diagnostics can be used to identify reasons for gear noise generation allowing noise control procedures to be undertaken at the very source.

4. CASE STUDY

Vibration and noise measurements were performed on a newly built lathe assembly to establish the reason for the higher noise level which was present on some lathes. Two lathes were of interest and these will be referred to as the Noisy and the Quieter one.

Preliminary sound intensity and vibration measurements indicated that the upper part of the gear case exhibits the highest vibroacoustic emission of all the covers and side panels. To check the vibration path between the spindle shaft and the top cover, an impact test was carried out while the machine was not running. An instrumented impact hammer kit and a dual channel signal analyser BK2032 were used to measure Frequency Response Function and the result is presented in Fig.1. From the magnitude of the FRF measured for the two lathes it can be seen that the main resonances in the gearbox assembly were at frequencies of approx. 530Hz and 2000 Hz. This was, however, not problematic since both noise and vibration measurements did not indicate high RMS values at these frequencies.

Vibration and noise measurements were taken for the two speeds of the spindle, e.g. 800 rpm and 2000 rpm for which large differences in the noise levels were experienced between the noisy and quieter lathes. Since emission of noise for the two spindle speeds exhibited clear differences in the noise pattern, they are considered separately.

800 rpm Spindle Speed

The spectrum of acceleration measured on the top gearbox cover is shown in Fig.2a and Fig.2b for the noisy and quieter lathe respectively. From the narrow band acceleration spectra it is seen that there are distinct peaks in the spectra which cause high overall noise and vibration levels. The most significant peak was in all cases at a frequency of 1468Hz. From the kinematic model of a gear train presented in Fig.3 this peak was identified as a second harmonic of the first mesh tooth contact frequency (TCF). It can be seen that the first mesh TCF and its higher harmonics give the most predominant peaks for both lathes in the frequency range 0-3.2kHz. This indicates that the gear pair F-G cause the majority of the noise emission for both lathes at a lower spindle speed.

The quality standard adopted for use on the lathe gears was studied and did not indicate any potential problems. All errors measured seemed to be well balanced and tip/root relief of approx. 6μm was appropriate for lightly loaded gears.

Since the results of gear geometry inspection did not indicate major geometry problems it was expected that higher dynamic excitation could be caused by other factors, like gear misalignment, eccentricity, and unbalance. These factors very often cause gear modulation which can normally be detectable by very narrow-band analysis and cepstrum. The results from narrow-band analysis zoomed around 1468 Hz centre frequency with 100Hz frequency span and are presented in Figs. 4a and 4b for the noisy and quieter lathes respectively.

Sidebands around the tooth meshing frequencies indicate modulation at a frequency corresponding to sideband spacing. From zoomed frequency spectrum at first and second harmonics of first TCF it can be seen that there is a very srong modulation at a frequency of 7.03Hz, for which the sidebands have an amplitude exceeding 20dB adjacent to the average spectrum level. At this stage a conclusion was drawn that the most likely reason for first TCF modulation at a frequency of 7.03Hz was too high a clearance of shaft "F" assembly which gave rise to modulation at shaft subharmonics. The spindle shaft "G" was fitted with a spring loaded anti-backlash bearing, and under normal operating conditions should not be considered as a potential source of radial clearance.

2000 rpm Spindle Speed

For 2000 rpm spindle speed there are distinct differences between the acceleration levels generated by the noisy and quieter lathe. Average acceleration levels, measured on the top cover of the gearbox, are presented in Figs. 5a and 5b for the noisy and quieter lathes respectively. It can be seen from these figures that all spectra exhibit a maximum of vibroacoustic activity at a frequency of approx. 1743Hz. From the kinematic model and forcing frequencies presented in Fig.3, it can be seen that the predominant peak in vibration spectra matches exactly the first TCF and, similarly as for 800 rpm spindle speed, is the major source of noise for both the noisy and quieter lathes. It can also be seen that for 2000 rpm the peak in the first TCF is shifted up by very strong modulation at a frequency of 33.625Hz. This is clearly visible in the zoomed noise level spectrum presented in Fig.6. The modulation frequency of 33.625Hz is easily identified as shaft "F" rotational frequency which indicates misalignment of the shaft and/or errors in gear concentricity. This leads to the similar conclusion of poor assembly of the gear 17T on the shaft "F", as encountered for the 800 rpm spindle speed.

Careful zoom analysis around the first TCF with a different frequency span allowed the identification of additional modulation frequencies, of secondary importance, e.g.

i) Δf = 20.2Hz (unbalance or misalignment of shaft "A")
ii) Δf = 26.85Hz (unbalance or misalignment of shaft "E")
iii) Δf = 6.7Hz (belt drive problems).

Recommendations

A simple vibroacoustic diagnostics carried out for the noisy and quieter lathes allowed precise noise source identification, thus allowing the noise control procedures to be focused at the very source. The original arrangement of the shaft "F" assembly is schematically shown in Fig.7a. The shaft was fixed in the housing and the gear was supported on the two DU type plain bearings. An alternative arrangement in this assembly was suggested to give better alignment of the gear assembly, lower clearances and hence a more silent transmission:

i) use of direct plain bearings between the shaft "F" and the gear 52T. Tolerance H6/f5 and a case carburised shaft would give better alignment and acceptable sliding conditions for changing gears.

ii) introduction of ball bearings at both supports for the shaft "F". In such an arrangement shaft "F" would always rotate together with the gear as there would be much less friction at the shaft bearings.

The modified assembly is schematically presented in Fig. 7b. Finally it was recommended that the minor problems with unbalance or misalignment at shafts "A" and "E", and any problems with belt drive quality should be carefully checked and rectified.

5. CONCLUSION

Simple vibroacoustic diagnostics was demonstrated to be a very useful tool for noise troubleshooting in lathe drive systems. As a result of noise source identification procedures firm recommendations could be suggested to tackle vibration and noise control problems at the very source. Procedures outlined in this paper were successfully used on different makes of lathes, allowing many structural changes to be recommended and utilised. Similar procedures could be employed to tackle other machine tool vibration problems.

6. REFERENCES

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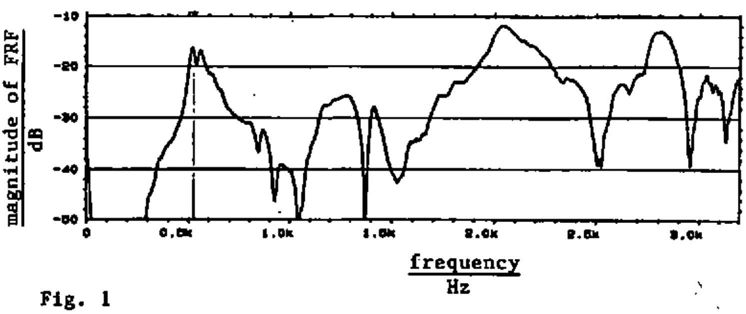

Fig. 1

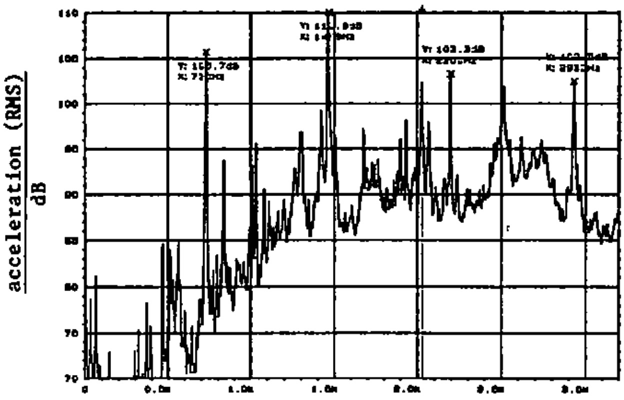

Fig. 2a frequency
Hz

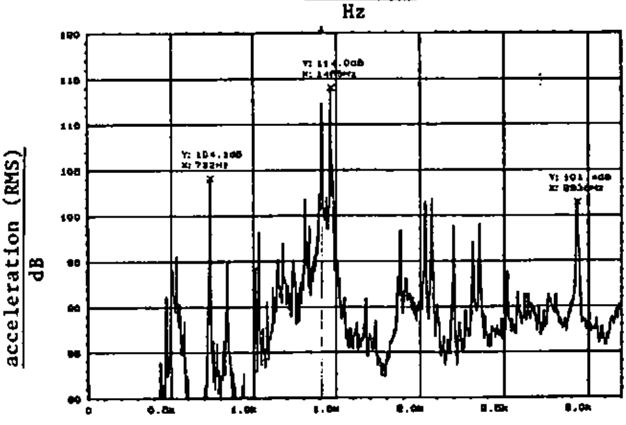

Fig. 2b frequency
Hz

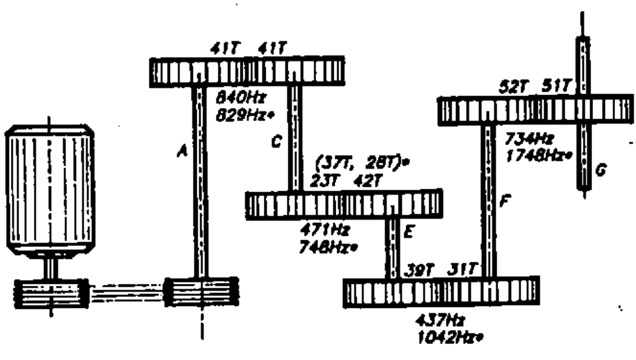

Fig. 3

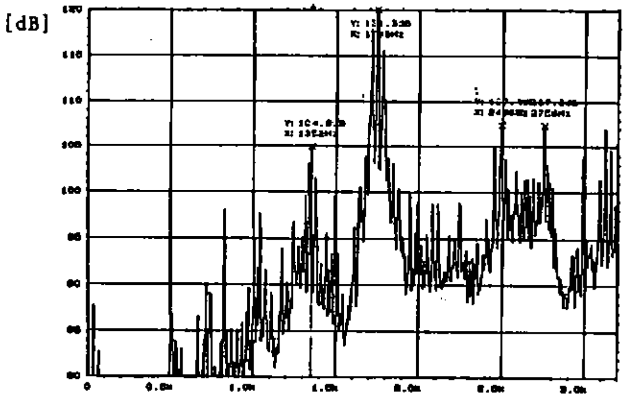

Fig. 5a frequency / Hz

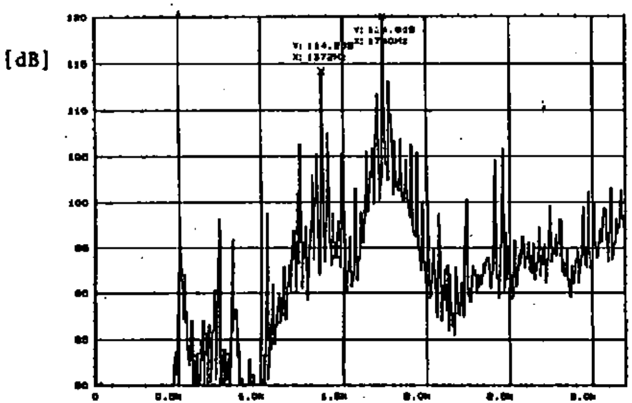

Fig. 5b frequency / Hz

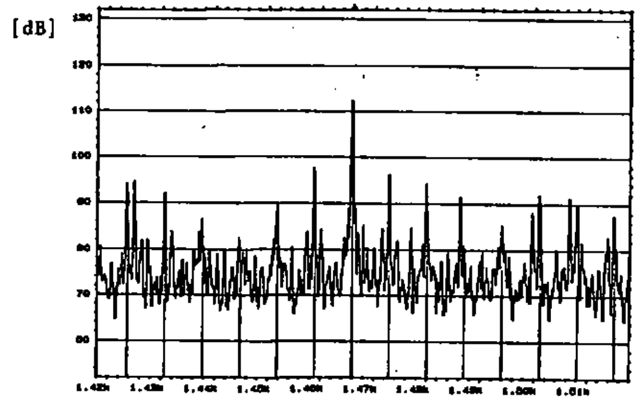

Fig. 4a frequency / Hz

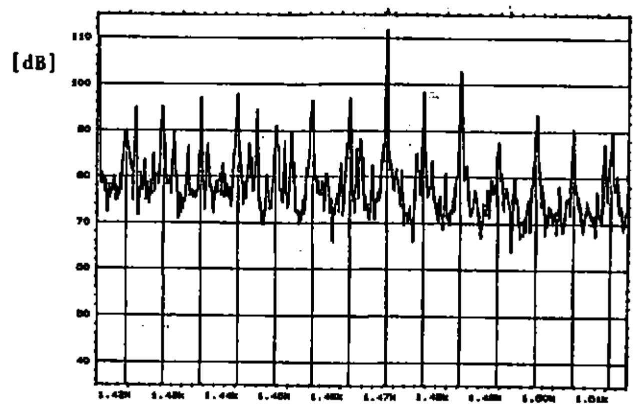

Fig. 4b frequency / Hz

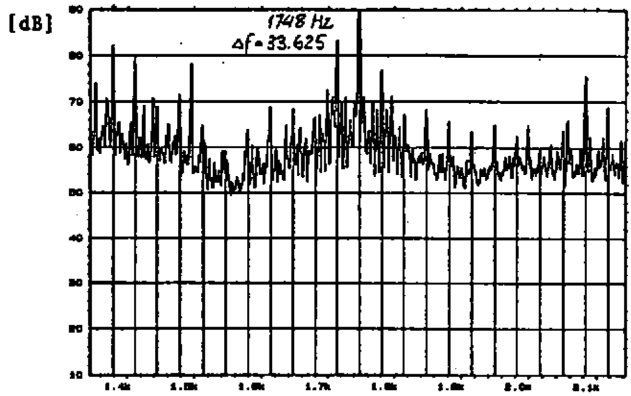

Fig. 6 frequency / Hz

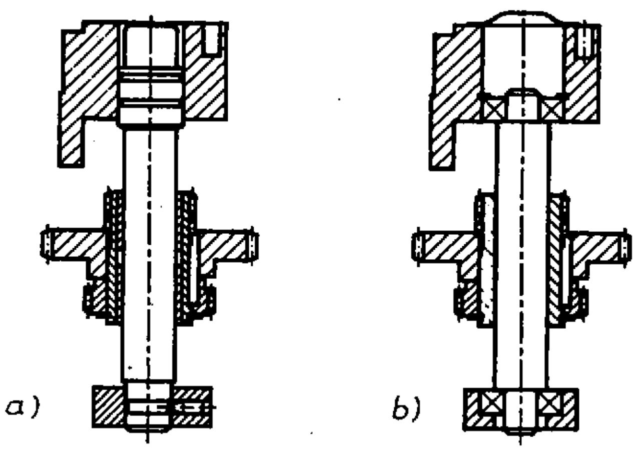

Fig. 7