FIFTH INTERNATIONAL CONGRESS ON SOUND AND VIBRATION

DECEMBER 15-18, 1997
ADELAIDE, SOUTH AUSTRALIA

THE EFFECT OF WORKPIECE TORSIONAL FLEXIBILITY ON CHATTER PERFORMANCE IN CYLINDRICAL GRINDING

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ABSTRACT

Most of the steady state grinding force models for cylindrical grinding show a dependence on the rotational speeds of both the workpiece and the grinding wheel. If, due to the flexibility of the workpiece drive, a torsional oscillation was superimposed on the mean rotational speed of the workpiece, then according to the cutting force models the cutting forces would vary about their mean. Any variation in the magnitude and/or phase of the cutting forces must have an influence on the chatter performance of the machine. Mathematical models are used to predict that such would be the case when the natural frequencies of the machine tool structure and the workpiece drive (torsion) are close. An experimental programme undertaken to test the idea is described. An artificially flexible workpiece drive was arranged and the chatter of the machine monitored. It was found that the torsional flexibility prevented 'workpiece' chatter from arising and, under certain conditions, 'grinding wheel' chatter was suppressed. The paper briefly outlines the hypothesis, describes the experimental arrangement and presents the results showing the beneficial effects. Some practical applications are provided.

INTRODUCTION

As with other machine tools, the cylindrical grinder is potentially dynamically unstable. Furthermore, as there are two possible regeneration paths (the workpiece surface and the grinding wheel surface), the dynamic system is considerably more complex than other machining processes such as milling. Research into chatter in cylindrical grinding processes now has a 45 year history **[1,2]** and it has been thought that the major phenomena are well understood.

A significant assumption made in all previous work is that the rotational velocity of the workpiece and the grinding wheel remain constant. Clearly this assumption cannot be strictly true as load variations, from for instance a variation in feed rate, would alter the speed. Generally in chatter models this assumption is adequate provided the cutting forces do not vary significantly with cutting speed.

It is well known that in the case of cylindrical grinding the tangential grinding force does vary as a function of both the workpiece speed and the grinding wheel speed and hence the constant speed assumption requires critical review. Consider the scenario where, due to torsional flexibility of the workpiece drive and a source of excitation, a torsional vibration were superimposed on the mean rotation. This would induce cutting force magnitude variations which would necessarily influence the chatter response of the process. To consider the potential effects of such torsional vibrations, a model needs to be constructed which includes the torsional degrees of freedom of the workpiece and grinding wheel drives. That work has been undertaken by the authors **[2,3]** and is reported briefly below.

GRINDING FORCE MODELS

Many workers have proposed cutting force models. An extensive review was undertaken by Tonshoff *et al.* **[4]** which compared fifteen models presented in the literature from 1953 to 1989 and suggested a composite form

$$P_t = C_w C_g \left(\frac{V_w}{V_g}\right)^{a_1} \delta^{a_2} d_{eq}^{a_3} \tag{1}$$

where C_w, C_g are constants to account for the workpiece and grinding wheel properties respectively, δ is the depth of cut, a_1, a_2, a_3 are constants and d_{eq} is an equivalent grinding wheel diameter. Of particular note in the present context is that the tangential grinding force is dependent on the workpiece surface speed, V_w, and the grinding wheel surface speed, V_g.

$$P_t \propto \left(\frac{V_w}{V_g}\right)^{a_1} \tag{2}$$

A THREE DEGREE OF FREEDOM MODEL

A mathematical model was developed which included a degree of freedom which allowed the workpiece and the grinding wheel to move apart (simulating the flexibility of the machine tool structure), a torsional degree of freedom between the workpiece drive motor and the workpiece and a similar degree of freedom for the grinding wheel. Each of these degrees of freedom was modelled as a linear spring-mass-viscous damper system. The boundary of stability as a function of the width of a plunge cut was then located and compared to the response where the torsional degrees of freedom were removed. It was discovered that when the workpiece torsional degree of freedom was tuned close to the natural frequency of the structure and rotational speed fell below some 'cutoff' point, regenerative chatter did not occur. Figures 1 shows the effect on the stability boundary.

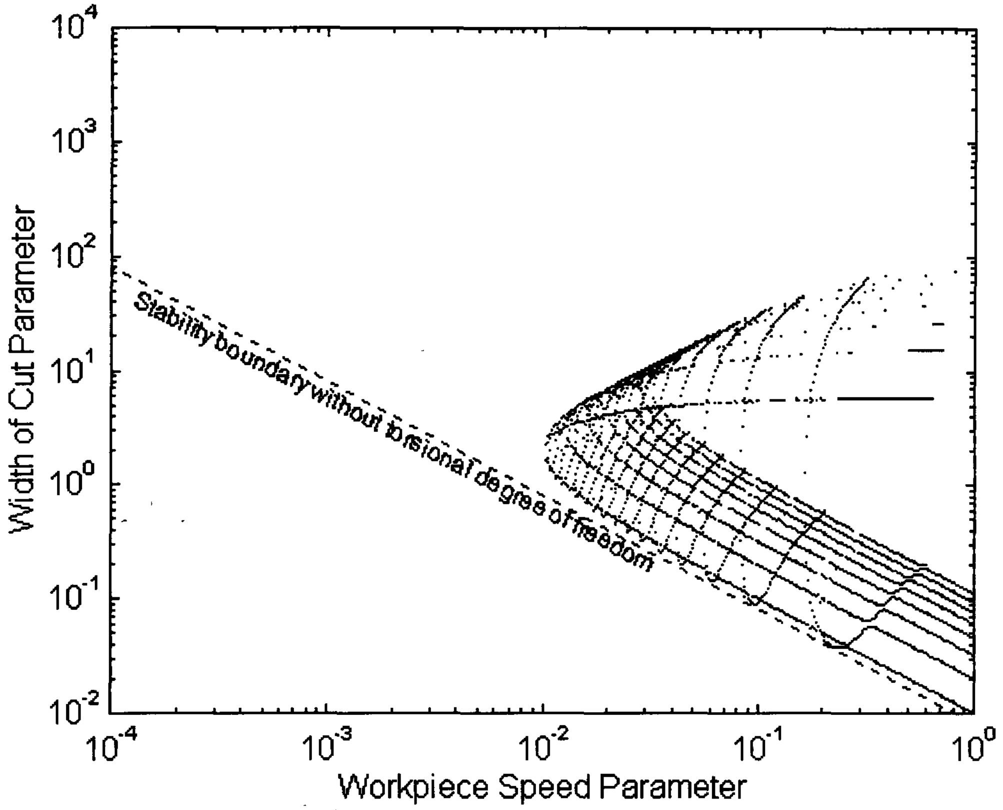

Figure 1. Stability boundary modified by the addition of workpiece drive torsional flexibility

It is also well known **[5]** that most cylindrical grinding operations are operating beyond the stability boundary and rely on the low growth rate of the self-excited vibrations which allows sufficient time to achieve some useful output before the amplitude is no longer acceptable. The mathematical model was aimed at determining the instability or otherwise of the system and assumed that all vibration amplitudes were small. It was therefore necessary to test experimentally whether the workpiece drive torsional flexibility affected the chatter response of a cylindrical grinding machine when detectable chatter amplitudes existed.

EXPERIMENTAL AIMS

While the model predicted that torsional compliances are of significance, many of the assumptions and limitations inherent in the model are yet to be studied and verified individually. It was a 'first generation' model to test the hypothesis that torsional effects can be significant. Being 'first generation' it is unlikely that it reflects reality with sufficient accuracy to be directly verifiable. Recognising this, the experimental programme did not attempt to do so. Rather, it tests the influence that torsional compliance has on a real grinding system to assess its importance in chatter.

EXPERIMENTAL SETUP

The grinding machine available to undertake the experimental programme was a Jones - Shipman Model 310 Cutter and Tool grinder with a 200mm swing and 480mm between dead centres. It was readily configured for cylindrical plunge grinding as a motorised work drive was available. The grinding wheel drive motor was rated at 1 hp (0.75 kW) and pulleys were available which provided grinding wheel rotational speeds of 3600 and 5000 rpm. Unfortunately, plunge feed was provided by a manual hand-wheel only. It was desirable to be able to adjust the natural frequencies of the governing transverse mode and the workpiece torsional mode. The grinding wheel spindle is of precision design and it was therefore undesirable to attempt to incorporate a tunable natural frequency device into the drive train. The grinding wheel drive system was not altered in any way. Hence, the aim was to modify and adapt the machine to enable it to undertake cylindrical plunge grinding with the following facilities:

- automatic and continuously variable plunge feed rate
- tunable structural natural frequency
- tunable workpiece torsional natural frequency
- continuously variable workpiece speed.
- variable width of cut

Instrumentation was necessary to measure and record the following parameters:

- workpiece transverse displacement
- mean normal contact force
- mean tangential contact force
- grinding wheel mean speed
- grinding wheel instantaneous speed
- workpiece mean speed
- workpiece instantaneous speed
- feed rate

The tunable workpiece structural flexibility was achieved by mounting the workpiece on a separate carrier whose bearings were mounted on leaf springs. The thickness of the springs could be altered and thus alter the structural natural frequency. The workpiece carrier was driven via a torsionally flexible shaft. The torsional natural frequency could be adjusted by replacing this shaft with others of varying diameters. The instantaneous shaft speeds were measured via laser torsional vibrometers. A view of the grinding setup is shown in Figure 2.

RESULTS

Recalling that the aim of the experimental programme was to determine if variation of the torsional tuning of the workpiece system affected the chatter response, the results presented in this section are from two tests which show the altered response clearly.

Table 1 shows the parameters for these two tests. All of these parameters were kept as close as possible for the two tests; the only modification was to the tuning of the torsional natural frequency of the workpiece system. In the first case (Case 1) its freely rotating natural frequency was 262 Hz while in the second case (Case 2) this was lowered to 40 Hz. Note that, as the moment of inertia of the workpiece remained constant, this alteration implies a significant reduction in the torsional stiffness of the rotating system.

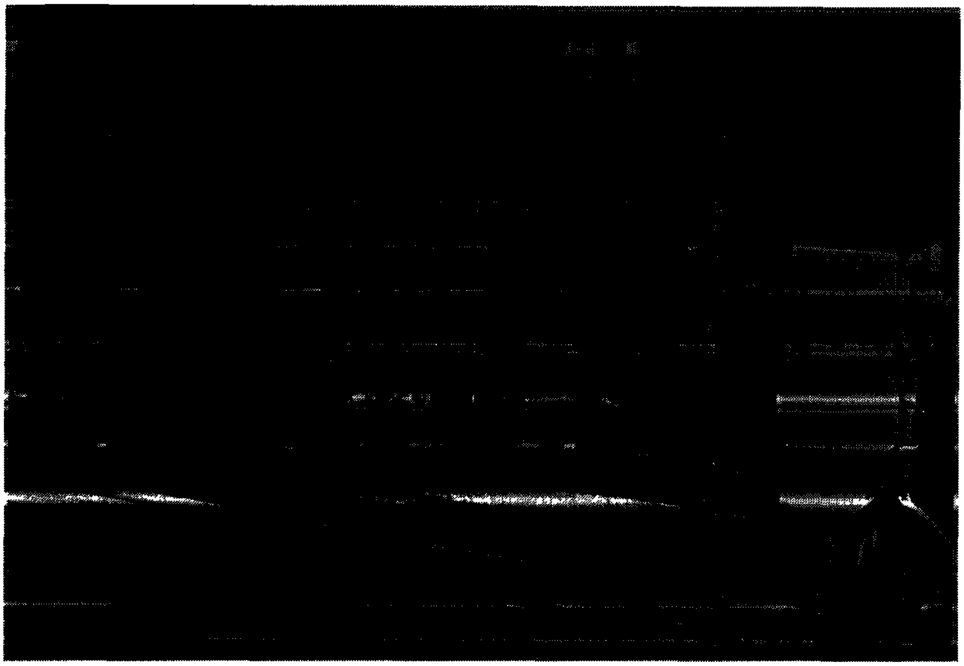

Figure 2. View of grinding test rig.

Test	Case 1	Case 2
Workpiece torsional natural frequency, Hz	262	40
Workpiece tuning parameter, F_W	1.46	0.22
Workpiece mean speed, rpm	158.9	163.9
Grinding wheel mean speed, rpm	3867	3871
Feed rate, μm/rev	7.61	7.42
Workpiece diameter - start, mm	89.36	89.46
Workpiece diameter - 1000 seconds, mm	75.64	75.76
Grinding wheel diameter, mm	149.16	145.04
Width of cut, mm	1.81	1.76
Mean tangential force, N	2.97	2.56

Table 1. Grinding parameters for representative tests.

Figures 3 and 4 show the spectrograms for the signal from the left hand pedestal and the workpiece torsional vibration respectively for test Case 1. The corresponding diagrams for the Case 2 test are shown in Figure 5 and 6.

The results of the two tests presented here show clearly that the chatter performance of the grinder has been altered by changing only the torsional tuning of the workpiece. Comparing the spectrograms it is clearly seen that the former has a growing pedestal vibration amplitude at 260 Hz accompanied by significant torsional velocity amplitude at the same frequency. Conversely, no such motion is present in the latter. The 260 Hz vibration has been suppressed by the increased workpiece drive flexibility.

A broad conclusion may therefore be drawn; *torsional motions of the workpiece have been shown to modify the chatter response of a cylindrical grinding machine tool.*

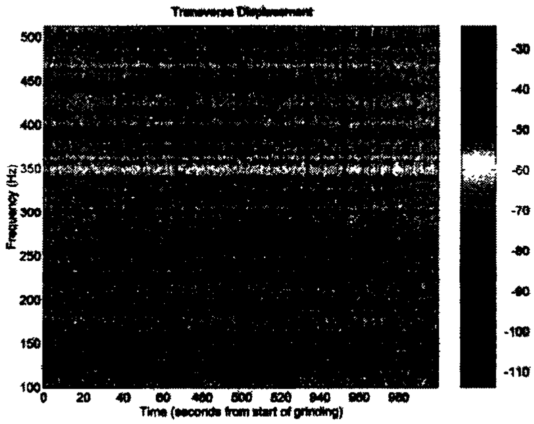

Figure 3. Transverse displacement with workpiece drive tuned to 262 Hz.

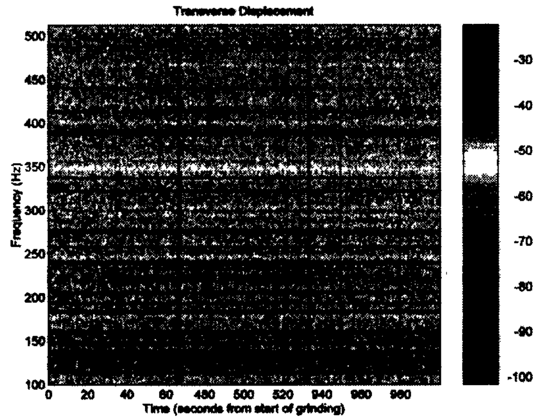

Figure 5. Transverse displacement with workpiece drive tuned to 40 Hz.

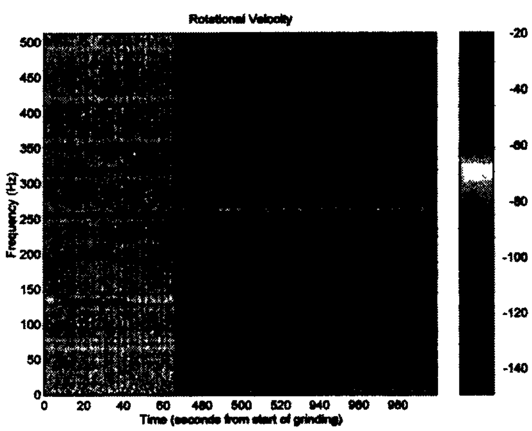

Figure 4. Rotational velocity of workpiece tuned to 262 Hz.

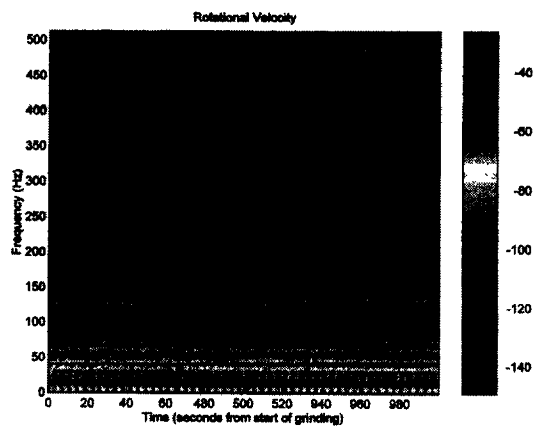

Figure 6. Rotational velocity of workpiece tuned to 40 Hz.

POSSIBLE MECHANISM OF CHATTER SUPPRESSION

The experimental tests produced chatter which was exclusively due to grinding wheel modes (i.e. lobes wearing on the periphery of the grinding wheel) even though plunge grinding was being undertaken. The predictions of the simulations were that workpiece modes would grow rapidly and be the most unstable. Cognisant of the fact that the tests were conducted beyond the stability boundary, it is apparent that some mechanism was active which prevented the growth of the workpiece regenerative modes. The obvious candidate was wave filtering due to contact zone deformations and geometric interference. Following some tests at high workpiece speeds it was concluded that contact zone filtering was not preventing the workpiece modes from arising. The following mechanisms were then considered as alternative possibilities.

PHASE SHIFT LIMITING OF AMPLITUDE

The mathematical model assumed that the amplitudes of vibration were sufficiently small that the angular motion did not alter the rotational period of the workpiece (or grinding wheel). Clearly, as the amplitude of chatter vibration rises this assumption is no longer valid. The experimental work detected only those vibrations with amplitudes sufficient to be resolved by the instrumentation. Hence a possible mechanism for the amplitude limiting of the workpiece chatter modes was that as the torsional amplitude of the workpiece increases, the feedback phase relationship required for unstable amplitude growth (and achieved with constant rotational speed) is sufficiently disturbed such that the system dynamics become non-linear and amplitude limited. Due to the system's non-linearity under such conditions, this mechanism could only be modelled in the time-domain.

PHASE SHIFT DUE TO EXCITED TORSIONAL RESONANCE

It was observed that in all cases the natural motion of the workpiece torsional system was excited even when no transverse motion was detected at that frequency. This implies that the oscillating grinding force is not sinusoidal and contains a broad range of frequencies. This torsional motion at the natural frequency would produce a phase shift of the regenerated workpiece surface waves in a fashion similar to that described in the previous section. Further, occurring at a frequency which is probably harmonically unrelated to any transverse chatter vibrations present or the workpiece mean rotational speed, it would alter the regeneration phase in a pseudo-random fashion. It is well known that regeneration can be suppressed by workpiece speed variation, albeit on a 'macroscopic' scale. The mechanism proposed here is similar excepting that the speed variations are occurring at a relatively high frequency. Without the benefit of a time-domain model to test the above hypotheses, the authors prefer the latter two mechanisms as being the effects which either suppress or amplitude limit the workpiece chatter modes in cylindrical plunge grinding where workpiece system flexibility is present.

PRACTICAL APPLICATION

A simple carrier design incorporating adjustable torsional stiffness is shown schematically in Figure 7. A ring surrounds and is clamped to the workpiece (usually a 'V' rest with a single clamping screw). Rigidly attached to and projecting radially from the ring is a flexure which replaces the relatively rigid arm in standard designs. The flexure is in turn driven by a modified driving pin. The pin can be clamped to the driving plate at any desired radius and it straddles the flexure so that drive can be imparted in both senses. This could be as simple as two pins with parallel axes disposed symmetrically on each side of the flexure with little clearance.

The torsional stiffness, and hence frequency, may be altered by changing the radius at which the drive pin is placed or by exchanging the flexure for others with various flexural rigidities. It is envisaged that this arrangement would be tuned in conjunction which some instrumentation which would detect the natural torsional frequency. It would also be possible to incorporate some torsional flexibility by arranging for the cantilevered drive pin to also be a flexure.

This has the advantage that the torsional stiffness relative to the workpiece axis is an inverse function of the cube of the length of the flexure. It would thus have a much wider adjustment range than the carrier design described. However, the incorporation of length adjustment may be difficult given that the attachment location of the carrier along the workpiece axis may be restricted.

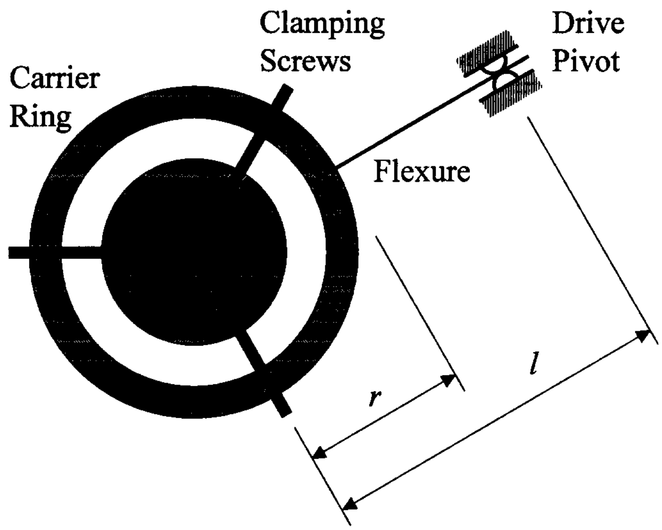

Figure 7. Conceptual design for a torsionally compliant work carrier.

CONCLUSIONS

Recapitulating, the aim of the experimental programme was to determine whether the presence of workpiece torsional compliance influenced the chatter characteristics of a plunge grinding system operating beyond the limit of stability. The results summarised in this paper show that the torsional compliance does alter the chatter performance of the system and, in a small sub-set of cases tested, suppressed chatter growth completely. This is an observation which has not been previously reported in the literature **[2]**.

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