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Two Cases of Forced Vibrations in Grinding Machines

A. A. ARANTES and P. M. R. DA CUNHA

Research Institute of Technology, Sao Paulo, Brazil

SUMMARY. Two cases of forced vibrations are presented, one of a cylindrical grinding machine, the other of a small surface grinding machine. In both of them, unlike in most common cases where rather unbalanced rotating masses or hydraulic systems are the main disturbances, driving Vee-belts caused the main or a considerable part of the trouble, together with unusual resonances and a wrongly dimensioned elastic mounting of the main driving electromotor.

RESUME. Deux cas de vibrations de rectifieuses ont été étudiés et certains effets mal connus ont été observés. Dans le cas d'une rectifieuse cylindrique, les sources habituelles de vibrations, provoquées par des masses en rotation non équilibrées, produisent des effets négligeables. On a trouvé que les courroies de transmission constituaient la source de vibrations la plus importante; elles engendrent des vibrations du moteur monté sur des supports élastiques et dont la masse d'inertie excite par résonance des vibrations de la machine. Le deuxième cas examiné a été celui d'une rectifieuse plane. Les deux vitesses de rotation de la meule correspondaient aux fréquences de résonance des vibrations de la machine. Des vibrations additionnelles étaient en outre imputables à l'ensemble composé par les courroies de transmission et le moteur monté sur supports élastiques.

ZUSAMMENFASSUNG. An zwei Schleifmaschinen wurden erzwungene Schwingungen untersucht. In dem ersten Fall handelte es sich um eine Rundschleifmaschine, bei der die allgemein bekannten Störquellen, wie z.B. Unwuchten von rotierenden Massen oder hydraulischen Pumpen nur einen vernachlässigbaren Einfluß hatten. Die Hauptstörquelle wurde in den V-Riemen gefunden, die den elastisch montierten Hauptmotor zu Schwingungen anregten. Die Massenträgheitskräfte des Hauptmotors verursachten wiederum Resonanzschwingungen der Maschine. Als zweites wurde eine Flachschleifmaschine untersucht. Die zwei möglichen Drehzahlen der Schleifscheibe führten zu einer Resonanz bei den entsprechenden Schwingungsarten der Maschine. Durch die Riemenkräfte und die Gummilagerung des Motors wurden zusätzliche Schwingungen hervorgerufen.

1. INTRODUCTION

IN the newly created department for research of machine tools in the Research Institute of Technology—IPT in Sao Paulo, the basic experimental activity at the present stage of development consists in analytical testing of machine tools delivered mostly by renowned machine tool manufacturers. In measuring and analysing the various characteristics of basic types and designs of machine tools, the staff of the department learn not only experimental techniques but also the relationships between design features and resulting characteristics. Various points of view are taken such as static deformations and their influence on accuracy, thermal deformations, stiffness and damping characteristics of spindle systems, limit conditions of chatter, etc. In one of these projects, forced vibrations of grinding machines have been analysed. Some interesting aspects have been discovered which are believed to be of general interest, which is the reason for presenting them to the C.I.R.P. conference in all the modesty of the youngest among machine tool research laboratories.

2. CYLINDRICAL GRINDING MACHINE

The machine investigated was of a middle size: maximum swing over table 200 mm, maximum distance between centres 1500 mm, power of the grinding wheel driving motor 7·5 kW.

Relative vibrations between the grinding headstock and the workpiece clamped between centres were measured by means of a Philips relative electrodynamic pick-up fixed rigidly to the headstock. Two workpiece sizes have been used in the investigations, one with dia. 42 mm, 480 mm long, the other dia. 90 mm, 470 mm long.

If, as usual, vibrations caused by unbalanced rotating masses are anticipated the following frequencies have to be looked for, corresponding to the speed of rotating masses on the given machine, expressed in rev/sec.

main driving electromotor	29 Hz
electromotor driving the oil pump	58 Hz
oil pump (driven by belt transmission)	55 Hz
electromotor of workhead steplessly variable	2·5–30 Hz
grinding wheel spindle	25 Hz

The supply voltage frequency was 60 Hz. A possible magnetic unbalance of electromotors should therefore result in vibrations with frequency 120 Hz.

For good reasons which will be understood later the frequency with which the Vee belts completed one pass around the pulleys of the motor and of the spindle was found to be 9·5 Hz.

First step

Measuring absolute vibrations of various points of the machine frequencies are found corresponding to some of the mentioned sources, see Fig. 1, where records of displacement of vibrations are reproduced as registered on the Oscilloscript. The record (*a*) shows vibrations on the grinding head resulting from interference between those produced by the motor driving the oil pump and those produced by the pump itself. On record (*b*) vibrations of the workhead are given as being caused by the electromotor of the workhead its speed set at 25 rev/sec. On record (*c*) vibrations of the grinding head are seen to be caused by the unbalance of the grinding wheel. All records have been made with only one of the sources running, so that in case (*a*) it was only the hydraulic pump, in case (*b*) the workhead motor and spindle, in case (*c*) the main motor and the grinding spindle.

Second step

However, once relative vibration between the workpiece and the grinding head are measured, the picture changes, see Fig. 2, reproducing records of relative displacements. Record (*a*) corresponds to

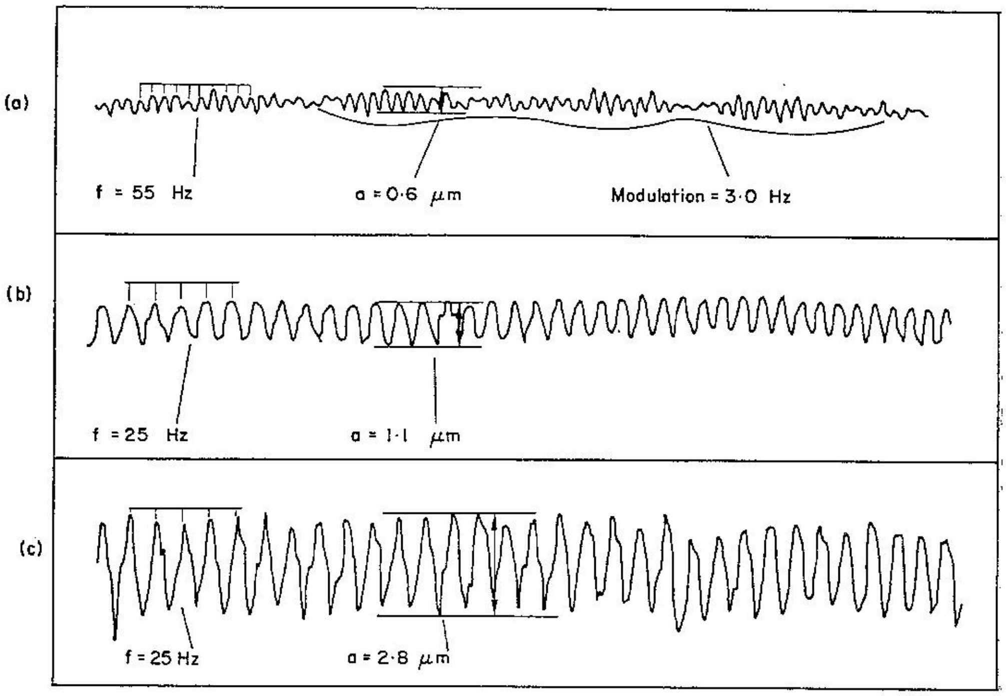

Fig. 1

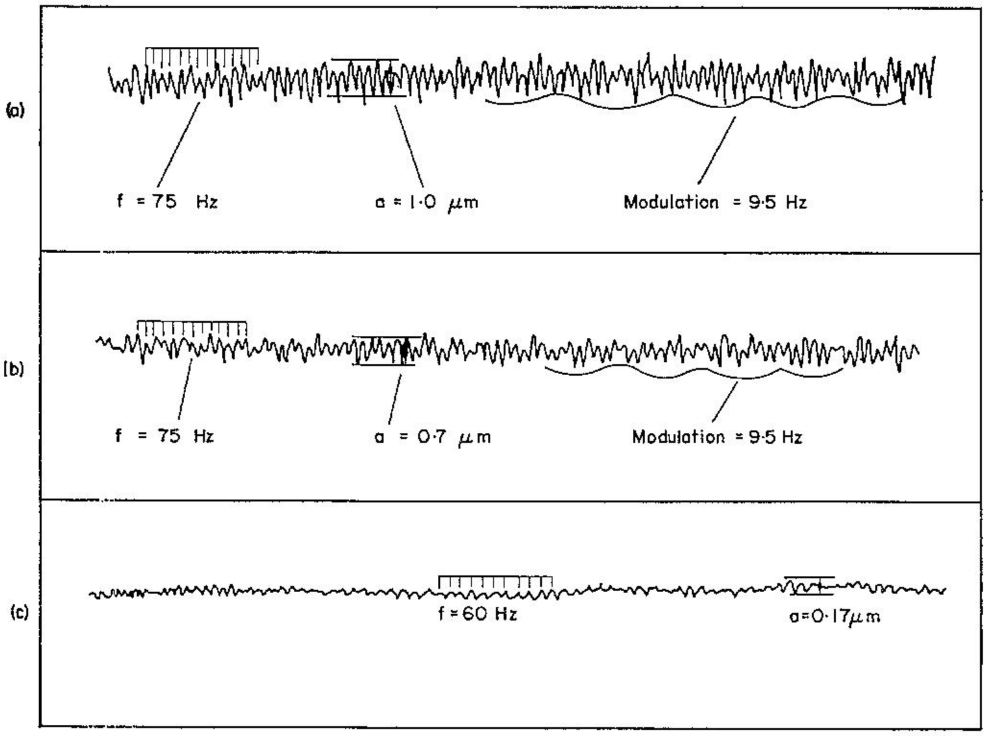

Fig. 2

the case where all possible sources are switched on. In case (*b*) only the main motor and the grinding spindle were running. In case (*c*) all sources were running excepting the main motor which had been switched off. In case (*c*) the picture is similar to case (*a*), Fig. 1, but the vibration amplitude is much smaller. If the main motor with the spindle are on, vibration of 0·7–1 μm amplitude occurs with frequency 75 Hz, amplitude modulated at a frequency 9·5 Hz. It is quite feasible that vibration with this frequency should have been found as absolute vibration of the grinding head, case (*c*), Fig. 1, but of course in the record it is completely covered by large vibrations with frequency 25 Hz.

The result is rather surprising because no source of disturbing force with frequency 75 Hz was anticipated.

Third step

Therefore, some more detailed measurements have been made as reproduced in the following. In order to increase the sensitivity of measurement all following records have been made by registering velocity instead of displacement of vibration. Nevertheless, figures noted below the records indicate again amplitudes of displacement as calculated from velocity. In this way in Fig. 3 the following cases may be observed:

(*a*)—all sources on, apart from 75 Hz; 1·7 μm also vibration 150 Hz; 0·5 μm is found,

(*b*)—as (*a*), but workhead off, similar to (*a*),

(*c*)—only main motor and grinding spindle on, result similar to (*a*),

(*d*)—only main motor running with belts between it and the spindle removed, vibration diminishes drastically.

From the previous evidence it is obvious to look for the Vee-belts between the main motor and the grinding spindle as source of the 75 Hz vibration, although the mechanism of the occurrence of just this frequency is still not yet clear. Some more records are shown to support this assumption.

Fourth step

First of all, in Fig. 4, records are shown which illustrate the influence of preload of belts on the vibration in question. From case (*a*) to (*b*) and further to (*c*) the preload increases and the pertinent vibration of 75 Hz increases also from an amplitude 1·0 μm to 1·5 μm. Further, in Fig. 5 other changes in belts and their effect are shown. In case (*a*) all six belts have been used adjusted for the moment in such a way that joints of all of them were in line. The modulation of the pertinent vibration is very clear corresponding to that point of disturbance running around 9·5 times per sec. Thus the belts are identified as source almost with certainty. Cases (*b*) and (*c*) are such that in both only one belt has been used those being different ones producing different amplitudes of vibration.

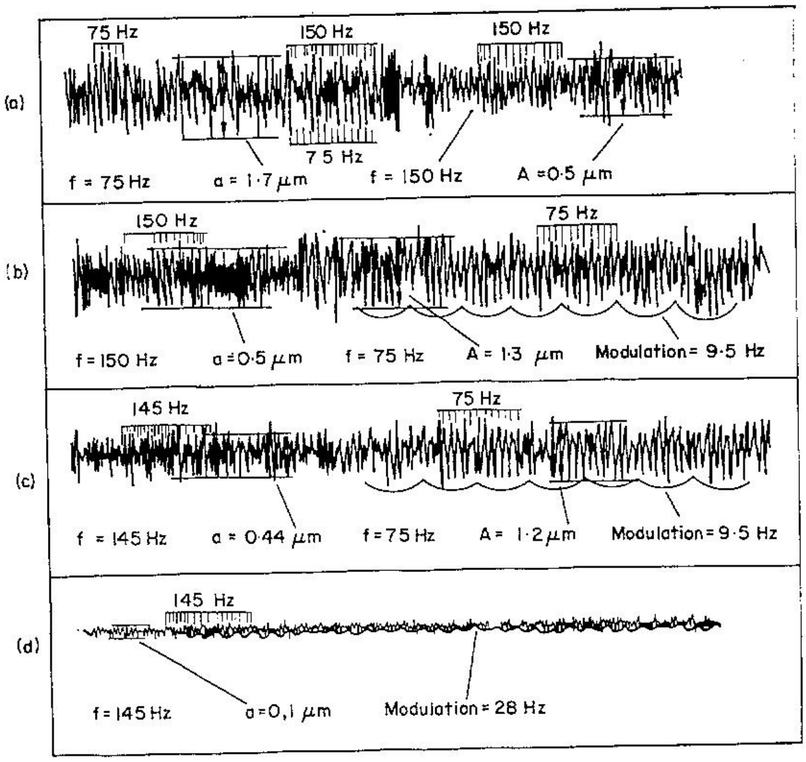

Fig. 3

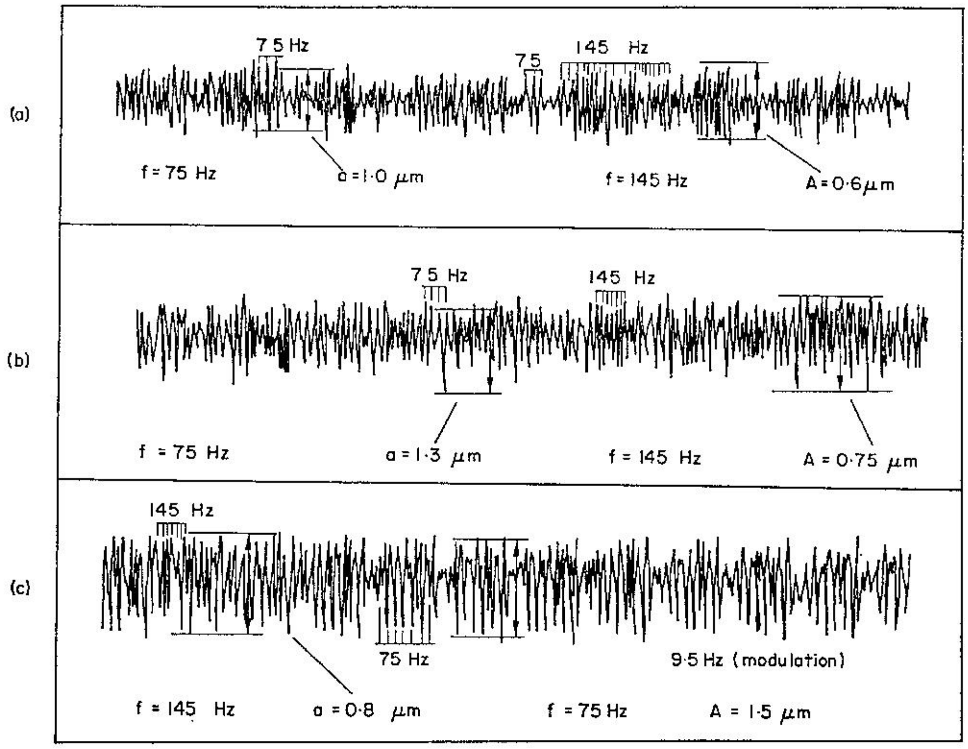
(a)
7.5 Hz
7.5
145 Hz
f = 75 Hz
a = 1.0 μm
f = 145 Hz
A = 0.6 μm
(b)
7.5 Hz
145 Hz
f = 75 Hz
a = 1.3 μm
f = 145 Hz
A = 0.75 μm
(c)
145 Hz
75 Hz
9.5 Hz (modulation)
f = 145 Hz
a = 0.8 μm
f = 75 Hz
A = 1.5 μm

Fig. 4

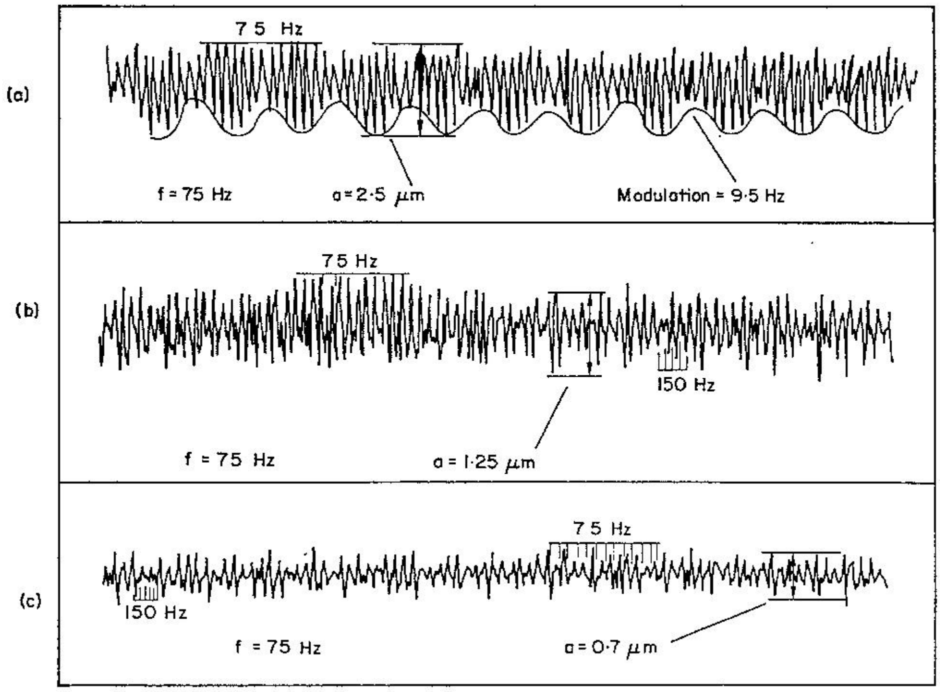
(a)
7.5 Hz
f = 75 Hz
a = 2.5 μm
Modulation = 9.5 Hz
(b)
7.5 Hz
150 Hz
f = 75 Hz
a = 1.25 μm
(c)
7.5 Hz
150 Hz
f = 75 Hz
a = 0.7 μm

Fig. 5

It may quite well be assumed that the width of the belts is irregular and, therefore, the tension acting on both pulleys varies and that this variation of force contains a number of components of different frequencies. The first question arises as to why just the 75 Hz component, and to a smaller extent, also the 145–150 Hz component are those which excite the relative vibration?

Fifth step

In order to find the answer the machine has been excited by means of an electro-dynamic exciter acting between the grinding head and the workpiece and direct response measured on the relative vibration pick-up. The set-up is shown in Fig. 6, where 1 is the exciter, 2 is the pick-up, 3 is the workpiece, 4 is the main motor, 5 is the workhead, 6 is the grinding head, 7 is the tailstock.

The measured frequency response for the bigger workpiece is shown in Fig. 7. Two clear resonance peaks are found at 75 Hz and at 150 Hz in the range of force frequencies 0 to 200 Hz. With the smaller workpiece a very similar response has been found. Thus, it is evident that the frequencies 75 Hz and 150 Hz are excited because they are natural frequencies of the machine corresponding to modes in which there is a great amount of relative vibration between the workpiece and the grinding head. It is also obvious why the other sources which excite considerable absolute vibration at various points of the machine with 25 Hz, or 55 Hz, do not enter practically at all into the picture of relative vibrations. At those frequencies the relative dynamic compliance as shown in Fig. 7 is very small.

In order to understand the case still better, modal shapes have been measured and they are shown in Fig. 8. In observing this diagram in detail it is found that the vibratory system of the machine may be represented in a simplified way, as shown in Fig. 9. There the bed is represented by an elastic plate to which the masses of the workhead 1, the tailstock 3 and the grinding headstock 4 are attached by means of flexible joints. Between the headstock and the tailstock there is the mass of the workpiece suspended on the springs of the centres and of its own elasticity. If both Figs. 8 and 9 are observed simultaneously

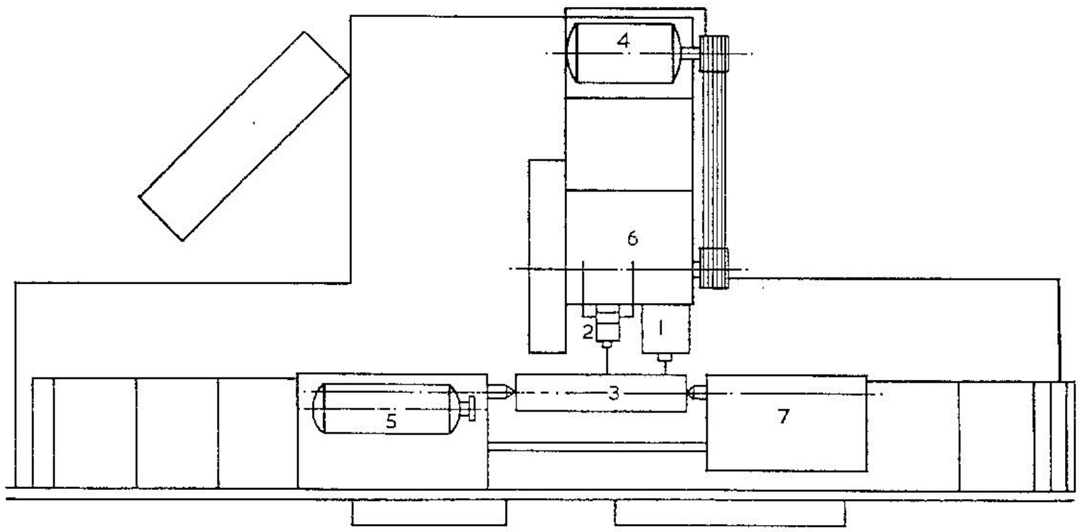

Fig. 6

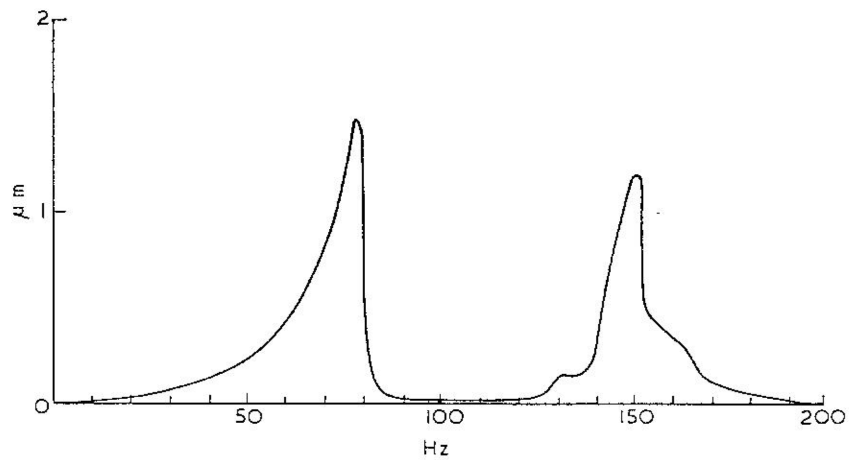

Fig. 7

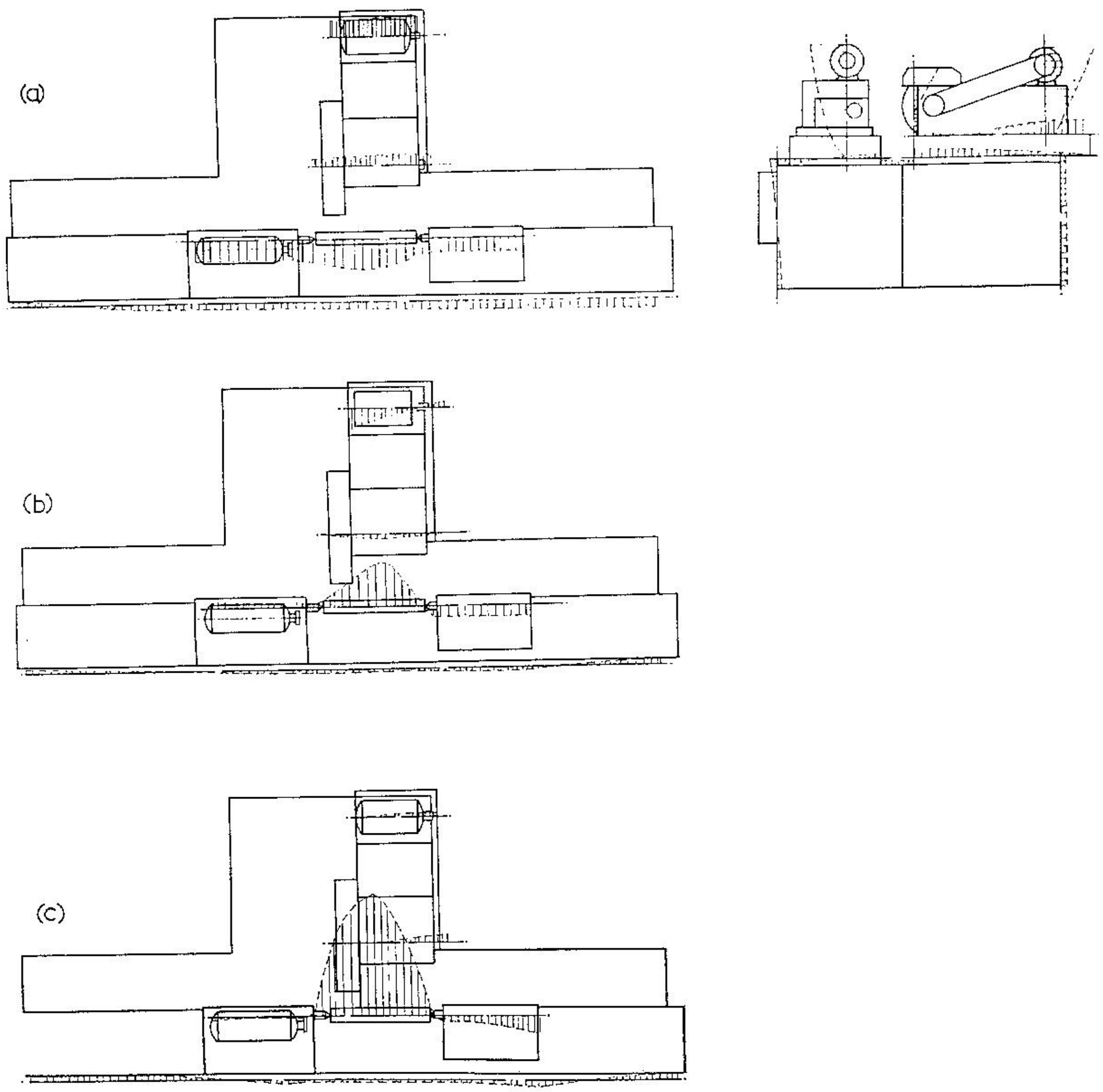

Fig. 8

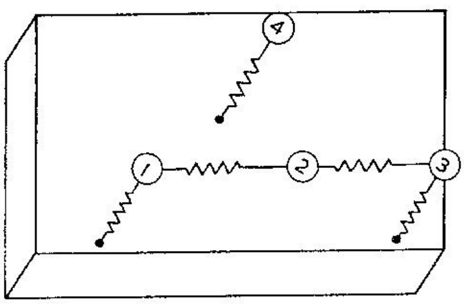

Fig. 9

the modal shapes may be described in the following way: the first mode, with frequency 75 Hz—masses 1, 2, 3 move more or less together in counter-phase to the mass 4; the second mode, with frequency 150 Hz—masses 1 and 2 move in one direction, mass 3 in the opposite direction; maximum amplitude is found on the workpiece and the grinding head moves to a great part again in counter-phase to the workpiece.

There is also a third mode which has not been discovered on the response curve Fig. 8, because it has a frequency above the range shown in the diagram; it is 315 Hz. In this mode the grinding headstock practically does not participate and it is the vibration of the mass 2 against the masses 1 and 3.

The most important mode is the first one. As seen in Fig. 8, the right hand top picture, this mode is represented by deformations of the U shape of the structure; the U shape opens and closes in one cycle; the rear upper masses move in counter-phase to the front upper masses.

This being so it is again not easily understood why the variation of tension in the belts can excite in such a sensitive way vibration of the U shape if that force is basically internal to the rear mass and does not act between the rear and the front masses.

Final solution

Therefore, a further step was necessary in the investigation. The natural frequency of the electromotor as mass on the springs represented by the rubber pads on which the motor was mounted, was surprisingly enough found to be approximately 80–90 Hz. In this way the whole picture becomes clear and may be explained on the basis of the diagram Fig. 10.

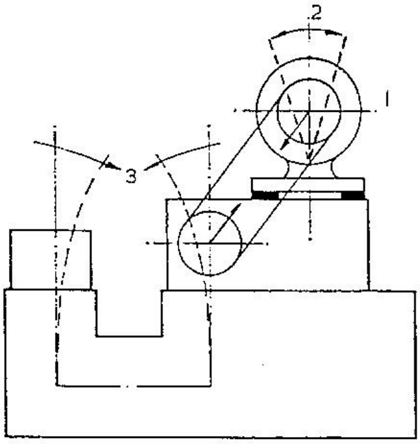

Fig. 10

137 Hz, which shows that the rigidity of the motor legs is $(137/80 \text{ to } 90)^2 = 3$ times higher than that of the rubber pads. Once again relative vibrations between workpiece and grinding head have been measured, see Fig. 11. The record on top shows vibrations of the motor without rubber pads, the record below shows vibrations with the rubber pads. The 75 Hz vibrations decreased twice.

The case illustrates: (*a*) the importance of Vee-belts as source of disturbing force, (*b*) how elastic mounting improperly applied may harm instead of improve the performance of the machine, (*c*) a particular coincidence of natural frequencies and resonances on a machine.

3. SURFACE GRINDING MACHINE

If the preceding case was one of particular coincidence this one is still more such a case. No detailed discussion of the case will be given and only the most striking and surprising aspect of it will be exposed.

It is a case of a simple small surface grinding machine with table area 10 in. by 16 in. and power of the main motor 1·5 h.p.

There are two possible speeds of the grinding wheel spindle of the machine, 1680 rev/min and 2680 rev/min, to be obtained by changing the pulleys of the belt transmission between the motor and the spindle. Any unbalance of the grinding wheel results thus in a harmonic force of either 28 Hz or 44·5 Hz.

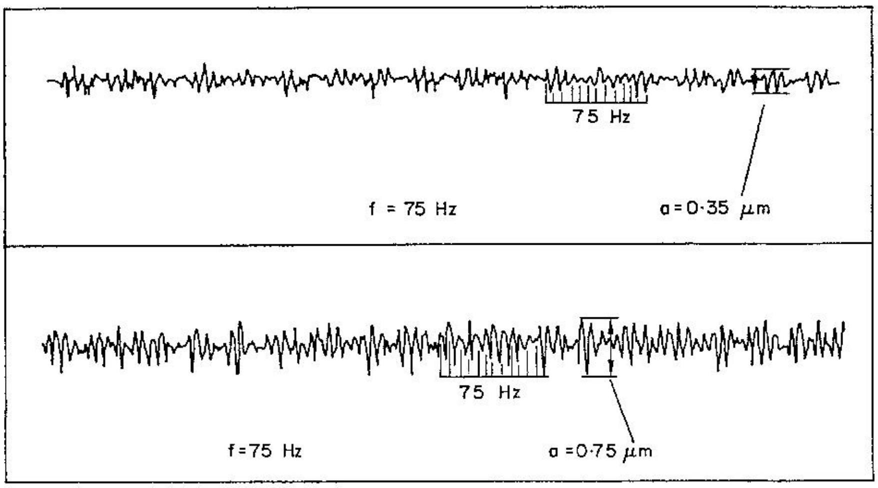

Fig. 11

There are two steps in the 75 Hz vibration. First, the variable component of the belt tension containing among others a frequency of 75 Hz excites vibration of the motor 1 in the shape indicated by arrows 2. The inertia force of the mass of the motor becomes the external absolute disturbing force acting at the right hand top of the basic U shape of the machine and excites its vibration indicated by the arrows 3.

This explanation may be proved. The rubber pads have been removed. The natural frequency of the motor as a mass on its legs as springs increased to 137 Hz (continued above).

The surface ground showed unusually great waviness, see Fig. 12. It has been ground with infeed only 5 μm; the upper part with transversal feed 5 mm, the lower one with transversal feed only 1 mm.

In analysing the reasons the machine has been excited by means of an electro-dynamic exciter acting between the table and the cross-arm (spindle

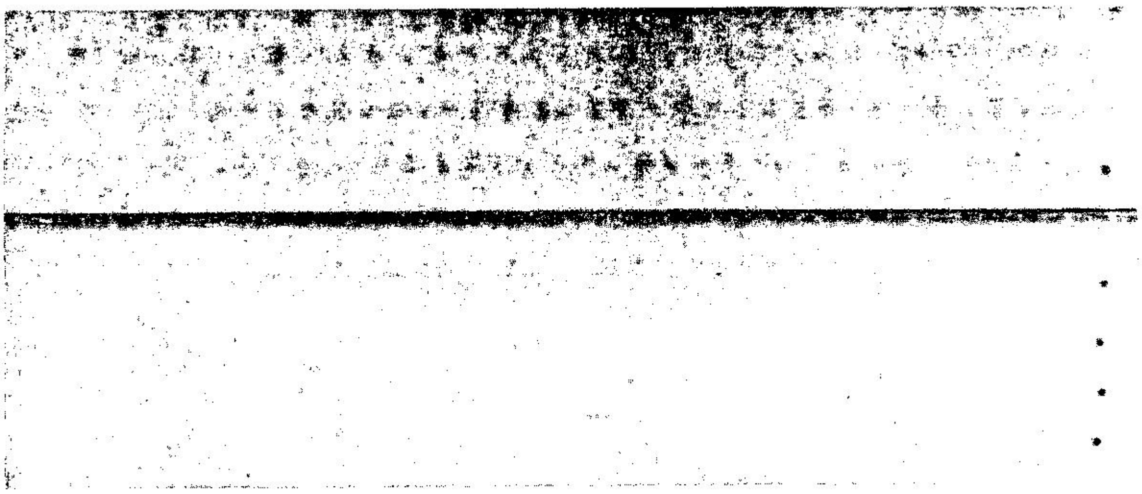

Fig. 12

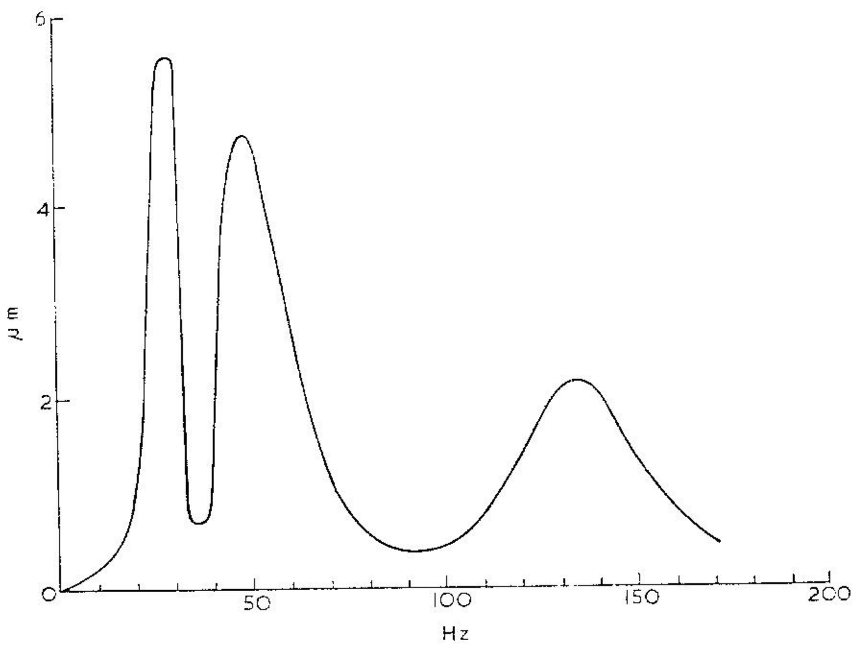

Fig. 13

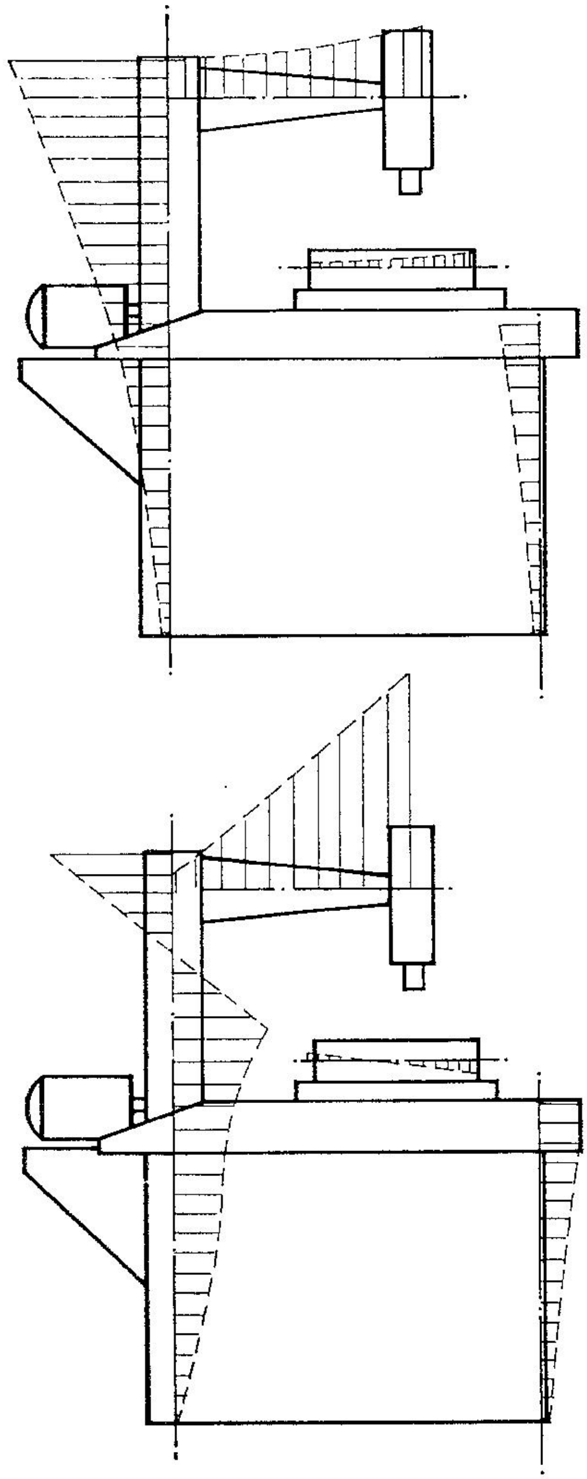

Fig. 14

housing). The obtained receptance is shown in Fig. 13. Two resonance peaks have been found: one at 28 Hz, the other at 45 Hz.

Corresponding modal shapes are shown in Fig. 14.

The case does not need much comment. An extremely improbable case is encountered: a machine designed in such a way that both available speeds are almost exactly in resonance with two basic modes of natural vibrations of the structure. Any slight unbalance of the grinding wheel produces rather great relation vibrations. Apart from vibration caused by an unbalanced grinding wheel, again on this machine vibration produced by the Vee-belt has been found of importance. Again the electromotor was mounted on rubber pads which in no way isolated the disturbance; the natural frequency of the motor as a mass on those pads as springs was just in the middle of the frequency range of disturbing forces.

4. CONCLUSION

Both cases described illustrate that although forced vibrations may be rather easily analysed (and the basic rule always is to avoid resonances) the manufacturers of grinding machines do not always pay enough attention to applying the common rules in the design stage. The first case was a rather intriguing one. The second case was most interesting by the unusual way of unintentionally achieving double resonance which if deliberately aimed at would be very difficult.

DISCUSSION

Dr. Corbach:

How did the authors come to the peak as the column in the deformation diagram of Fig. 14?

Author's Reply:

There is a mistake in the diagram in not showing that the lower part of the deformation line belongs to the bed and the upper one to the column. The column is vertically movable in rolling guideways. Thus, the peak in the deformation line corresponds actually to the deformations in these guideways.

Dr. Kaliszer:

Maybe the reason for the vibration at frequency 25 Hz was due to the frequency of a.c. or may be due to the wheel unbalance since the transmission ratio between the electromotor on the grinding wheel is usually 1 to 1.

Author's Reply:

Vibration with frequency 28–29 Hz really is excited by the unbalance of the wheel and of the motor (running at about

29 *rev/sec—remembering that supply voltage frequency in Brazil is* 60 *Hz). The excitation is at resonance with a* 28 *Hz mode of the machine. The natural frequency of* 28 *Hz was, of course, found in the exciter test, measuring the frequency response of the machine, when neither the motor nor the wheel were running.*

DR. VANHERCK:

Is the mode of 28 c/s not due to the compliance of the foundation?

Author's Reply:

It can be seen not to be so in the diagram depicting the modal shape. The main flexibilities in this modal shape are the column and the cross-arm themselves and the rolling guideways of the column.