Isolation and auto-balancing techniques for portable machines

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Isolation and auto-balancing techniques for portable machines

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The portable, engine powered strimmer is used as a case study of the application of vibration analysis and control techniques. Frequency analysis is an important tool in understanding how vibration is generated and transmitted. Modal analysis extends this to show how components can be modified and where they should be connected for controlling vibration. The theory of vibration isolation is discussed and tested. Auto-balancing is tested, and the theory of auto-balancing is reviewed to show why it cannot be applied to all machines with rotating unbalance.

The strimmer has two sources of vibration, the engine and the rotating unbalance of the cutting head. These are identified by frequency analysis, and resonant modes of components are shown to exist in the range of working frequencies. It is shown that better locations are possible for attaching the handles. Suggestions are given for maintaining strimmers to ensure low vibration.

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EXECUTIVE SUMMARY

The portable, engine-powered strimmer is identified as a type of machine whose operator can be subjected to large magnitudes of hand-arm vibration. This is shown to have two sources. One is the engine. The other, more important on those machines with the worst vibration, is the rotation of an out-of-balance component, in this case the cutting head.

The latter makes the machine a suitable example to test the wider application of auto-balancing, which has already been shown to be very effective for portable grinders. The fact that high levels of vibration are found even on those models that are fitted with vibration isolators to the handles also suggests that more comprehensive analysis techniques may be needed. These show where the present designs could be improved, and why.

Auto-balancing is shown not to be a solution for this type of machine. The reason for this is to be found in the dynamics of the machine, which has multiple resonances at frequencies below the operating range. An appendix that describes the literature and basic theory of auto-balancing provides the explanation for this. As rotating unbalance is a common source of vibration in portable machines, this technique does have potential for wider application, but it is not a universal panacea.

The key to understanding vibration is frequency analysis. This is used to identify the sources of vibration in the strimmer, and to indicate where resonant frequencies of components could increase transmitted vibration.

Modal testing and analysis is a more powerful tool, based on frequency analysis, and the strimmer provides several examples of where this can be useful. As well as providing the explanation of why auto-balancing does not work in this case, modal tests on the main shaft showed where the handles should be connected to it for minimum transmission of vibration. Modal testing also showed how one of the handles could be stiffened so as to reduce transmitted vibration by up to 35%.

The basic theory of vibration isolators is described briefly. This shows the importance of the impedance, i.e. the mass and stiffness, of the "source" and "receiver" as well as the flexibility of the isolating elements themselves. Some tests made with mass loading of the strimmer shaft ("source") showed that scope exists for reducing transmitted vibration, in this case by 40%.

The techniques used to understand the how vibration is generated and transmitted in the case of the strimmer can be applied to many other hand-held and hand-guided machines.

The strimmer is a machine that exhibits very large variation in vibration magnitudes. This study showed how most of this variation is caused by the way in which the cutting head is mounted on the machine. In addition to complicating the testing associated with this project, this needs to be taken into account when standard vibration "emission" values are obtained, and when worker exposure is measured.

Five points are also suggested whereby maintenance of strimmers could help to reduce operator exposure to vibration:

- Engine tune should eliminate misfiring
- Engine tune should limit possibilities for over-speed operation
- Operators need to be observant for incorrect operation of automatic heads, which should be tested during routing servicing.
- Operators need to be observant for increased unbalance of heads, which should be tested during routing servicing
- Anti-vibration mounts should be renewed regularly.

1. INTRODUCTION

Many types of powered portable machines generate vibration of sufficient magnitude that their daily use has to be curtailed so as to avoid exposing their operators excessively, with the associated risk of Hand-Arm Vibration Syndrome (HAVS). The HSE has for some time provided guidance for control of vibration exposure and the risk of HAVS (Health & Safety Executive, 1994)[1], and the requirement for control has recently been strengthened by the European Physical Agents (Vibration) Directive (EU, 2002) (PAVD). The design of machines for inherently low vibration is therefore of interest. Common sources of vibration within these machines are: the forces generated by internal combustion (ic) engines, as in chain-saws; rotation of out-of-balance masses, as may be intentional in pavement compacters or unintentional as in some grinding machines; and interaction of the tool with the material being worked on, as also contribute to vibration in chain-saws and grinders.

Various techniques are available to the designer of portable machines that can assist in controlling vibration at the handles. These include methods for analysing the vibration itself, and for analysing the dynamics of the machine and its components, the introduction of anti-vibration mounts, and in some cases the addition of an auto-balancing mechanism. These techniques are discussed in more detail below, followed by a description of their effectiveness when applied to one particular example of portable machine.

The example of a portable machine chosen for this study is the ic engine powered strimmer or brush-cutter. The distinction between "strimmer" and "brush-cutter" varies somewhat, but in this case the machines are powered by 32cc ic engines and use string or cord as the cutting element rather than fixed blades. They are used in considerable numbers by local authorities, government agencies and contractors for controlling grass and other growth on steep verges and riverbanks (Figure 1).

Figure 1: Typical strimmer usage.

[1] See list of references.

Magnitudes of vibration in work of between 4 and 15 ms^{-2} have been reported (Allsop *et al*, 2000). These magnitudes are for machines that already incorporate some vibration isolation, and so a study in greater depth may be needed to effect an improvement, making this type of machine a suitable case study for vibration control techniques.

The basic elements of a strimmer are a motor and cutting head, connected by a shaft on which are fitted handles and controls for the operator. The shaft is of such a length, and the handles so disposed, that the cutting head can be held conveniently at ground level in front of the operator (Figure 2, Figure 3). It is also usual that the mass of the motor counterbalances the extended length of the shaft with the cutting head at the end. There are some designs for which the operator carries the motor on a back-pack, connected to the shaft by a flexible drive, but with these it is difficult to provide the operator with a well balanced unit. In order for the cutting head to rotate in a plane parallel to the ground, the drive has to change direction at the end of the shaft. For some machines for light use, this is achieved by means of a curved shaft with a flexible drive. More usually it entails a bevel gear. Strimmers for domestic use are often powered by electric motors. Professional machines for use on larger areas are powered by lightweight internal combustion (ic) engines, usually single cylinder two-stroke units. Cutting heads are interchangeable and may be of two basic types: those with flat blades of plastic or steel, and those with cord or string. The former are often decribed as “brush-cutters”, the latter as “strimmers”. And finally there are two basic arrangements for the handles. These are known respectively as “Loop Handle” and “Cow-horn” and are shown in Figure 2 and Figure 3 respectively. Note that in order to provide a balanced support system for the operator, the Cow-horn design is provided with a harness to transfer most of the weight to the operator’s shoulders, leaving the handles for positional control. This latter system is better suited to large, relatively level areas. The Loop handle design is better for steep slopes and working around fixed obstructions.

Figure 2: Typical loop handle machine

Figure 3: Typical cow-horn machine

It is already known that there are two primary sources of vibration excitation in strimmers of this type (Author’s unpublished report to the Environment Agency). These are the ic engine at one end of the shaft, and the rotating, cutting head at the other. It can be shown by analysing the vibration at the handles that for those machines that have the most severe vibration, the cutting head is the dominant source. It is also obvious that the machines could be highly resonant, and this can be observed by any user who holds the handles while allowing the speed to run down from switching off at maximum speed. In these conditions, as with some other types of machine, the speed passes through brief periods of strong vibration as a succession of resonances is excited.

Given the importance of rotating unbalance, it is useful to note (Miwa *et al*, 1984, Stayner, 1996) that vibration arising from this source in a grinder can be greatly reduced by fitting an auto-balancer. It would therefore be of interest to know if such a device could be applied to other machines, both for simple unbalance, and possibly to reduce vibration from the ic engine.

This project was therefore devised as a practical demonstration of techniques for the understanding and control of vibration, particularly in powered hand tools, using the example of one type of tool, the strimmer, for which consistent low levels of vibration are presently difficult to achieve. After a brief introduction to some of the techniques, the report covers how the vibration at the handles of the machines were characterized, and some basic investigations of the likely sources. There is some concentration on the main source, which in these machines is in the cutting head, followed by an experiment with auto-balancing. (The history and theory of auto-balancing is covered briefly in an appendix). This experiment was not successful, and the next chapter describing some modal tests helps to explain this, and also relates to the other modifications proposed, for the anti-vibration mounts and the design of the loop handle. Finally there is some discussion of what factors in a maintenance programme might be significant for vibration control, of how the information gained could inform measurement and testing of these machines, and of how the techniques could have wider application.

2. TECHNIQUES FOR VIBRATION CONTROL

Techniques for the control of vibration can be described in terms of three categories:

- Reduce the source of excitation
- Break the transmission path between the source and the sensitive part of the structure ("receiver"), in this case the handles.
- Control resonances

2.1. AUTO-BALANCERS

(See also chapter 6)

As used successfully in the case of grinders, an auto-balancer provides a means of controlling the excitation effectively at the source. It consists of a toroidal cage, like that of a ball bearing, whose internal annulus is only partially filled with balls. The principle whereby these balls migrate to a position where they oppose the inherent unbalance of the part to which they are attached is described in more detail in chapter 6, below. The device may be attached to a machine as a separate object, or may be built into the machine. The only proviso is that the plane of the annulus in which the balls are free to circulate should be as close as possible to the plane of the inherent unbalance. An auto-balancer can be used in conjunction with other techniques for reducing vibration.

2.2. FREQUENCY ANALYSIS

Frequency analysis of vibration at the handles of a portable machine has two uses. First, where sources of vibration rotate at different speeds, or otherwise have different frequency characteristics, it may be possible to distinguish which source predominates, and therefore which source or transmission path requires attention.

And secondly, given knowledge of the machine dynamics (see below) and the operating speed range of the machine, it can indicate where elements of the machine design could be modified with best effect.

Note that in order for frequency analysis to be useful as a design or development tool, it is necessary to distinguish peaks in narrow bands of frequency. Frequency analysis in one-third octave bands is of limited value for this purpose.

2.3. MODAL ANALYSIS AND TESTING

Modal analysis of a machine or of its components concerns the dynamic response to excitation at resonant frequencies. The classic example is of a bell, which when struck rings at its fundamental resonant frequency. In the process of ringing, the body of the bell distorts periodically, its shape varying between two maximum deflected shapes, or mode shapes. As described by these mode shapes, some points on the body will show large periodic deflections, and some will show small deflections. In practice, the bell will ring with a combination of its fundamental frequency and higher harmonics of that frequency. The purpose of modal analysis is to separate the mode shapes for each natural, resonant frequency. It is therefore an important tool in controlling the effects of resonances.

There are various methods of modal testing. All involve defining a mesh of points to describe the component. Some involve exciting one point and measuring at all points. Some involve measuring at one point and exciting each point in turn. And some involve exciting more than one point simultaneously and measuring at each point. This last usually also involves sinusoidal excitation, or swept sine, whereas the others allow sine, swept sine or impulse excitation.

Impulse excitation with a fixed measuring point is the quickest method for highly resonant components, and is the method used in this project. As with any of the methods, this generates a family of frequency response functions which contain information about the response “shape” of the component throughout the range of frequencies measured. Software is available for extracting the resonant frequencies and associated mode shapes, such as Modent/Modesh (ICATS, 2002).

The results can be used to identify components whose resonant frequencies lie within the range of excitation frequencies, and also to show where there are areas of low response that would be suitable for connecting to other components of the machine or structure.

When used in conjunction with finite element modeling (FEM) modal analysis enables predictions to be made of the effects of changes in mass or stiffness within a component. However, for this project reliance is placed only on measured modal information and some basic principles of vibration isolation (see below).

2.4. ANTI-VIBRATION MOUNTS

One of the most useful techniques for reducing vibration reaching the handles of a machine is to insert flexible isolators, or anti-vibration mounts between the source of excitation and the handles. In some cases this may take the form of mounts for the engine, in others it is the handles themselves that are flexibly attached to the machine.

It is widely understood that reducing the stiffness of mounts and increasing the mass of the part to be isolated are both ways to increase the efficiency of isolation. For a combination of rigid bodies this is simply the case that a larger mass on a softer spring has itself a lower natural frequency, and so responds less to excitation at frequencies higher than twice that natural frequency.

This is actually a special case of a more general relationship. For machines with flexible components, and particularly when frequencies of excitation are higher than a few tens of Hz, the more general relationship applies:

$$E \approx [M1 + Ms + Mr] / [Ms + Mr]$$

Where E represents the efficiency of isolation, $M1$ the Mobility of the isolator itself, Ms the Mobility of the source or component on the excitation side of the isolator, and Mr the Mobility of the receiver or component on the handle side of the isolator. This expression would be inverted if mechanical Impedances were used instead of Mobilities. The expression can be interpreted as follows.

- Increasing the mobility of the isolator itself, i.e. making it softer, increases the efficiency, as already indicated.
- Reducing the mobility of both the source and receiver increase the efficiency. This means increasing the local mass and/or stiffness at the mount position.

2.5. APPROACH TO THE EXAMPLE OF THE STRIMMER

In the example of a strimmer, used for this study, the approach to the problem of excessive vibration was as follows:

- Preliminary checks to confirm that the large magnitudes of vibration experienced elsewhere were replicated with the tools under examination. (Chapter 3)
- Analysis of vibration frequencies of handles to confirm the relative importance of engine and cutting head components of excitation. (Chapter 3)
- Exchange of components to identify which part of the mechanism was most strongly associated with the excitation.(Chapter 4)
- Direct approach to understanding the sources of unbalance of cutting head (Chapter 5)
  - Changing relative orientation of component parts
  - Variations in packing the cutting string (cord)
- Modal testing to identify components likely to benefit from changing stiffness or mass, and to indicate any possible improvements in mounting positions for the vibration isolators. (Chapter 7)

The potential modifications then considered were:

- Fitting an auto-balancer to the cutting head (Chapter 6)
- Revised anti-vibration mount position (Chapter 8)
- Change of impedance at mount position (mass loading) (Chapter 8)
- Change the mass or stiffness of one variant of handle. (Chapter 9)

In the course of the work, the auto-balancer was tried before the modal testing. This was done because of the potential benefit and wide application had it proved successful. As will be seen, auto-balancing was not successful, and modal testing subsequently helped to explain why not. Modal testing also provides indicators to the other potential modifications, which are associated with improving the effectiveness of vibration isolators, or with controlling the resonance of the loop handle.

3. BASIC VIBRATION LEVELS AND FREQUENCY ANALYSIS

Two versions of strimmer were used, a Loop handle and a Cow-horn machine, as described in chapter 1 (Figure 2, Figure 3). Initially both machines were given an hour's use cutting grass by way of running-in. They were then tested under free-running conditions, using different lengths of the cutting cord to provide two, or three engine speeds when operated at full throttle, which was the only repeatable setting. The cord lengths were initially 160mm, 120mm and 80mm, but because, the loop handle machine had already reached a magnitude of 20 ms^{-2} when tested with the 120mm cord length, the shorter length was discontinued. Vibration was measured in three axes, but all values are given as approximations to the vector sum, i.e. the Root-Sum-of-Squares (RSS).

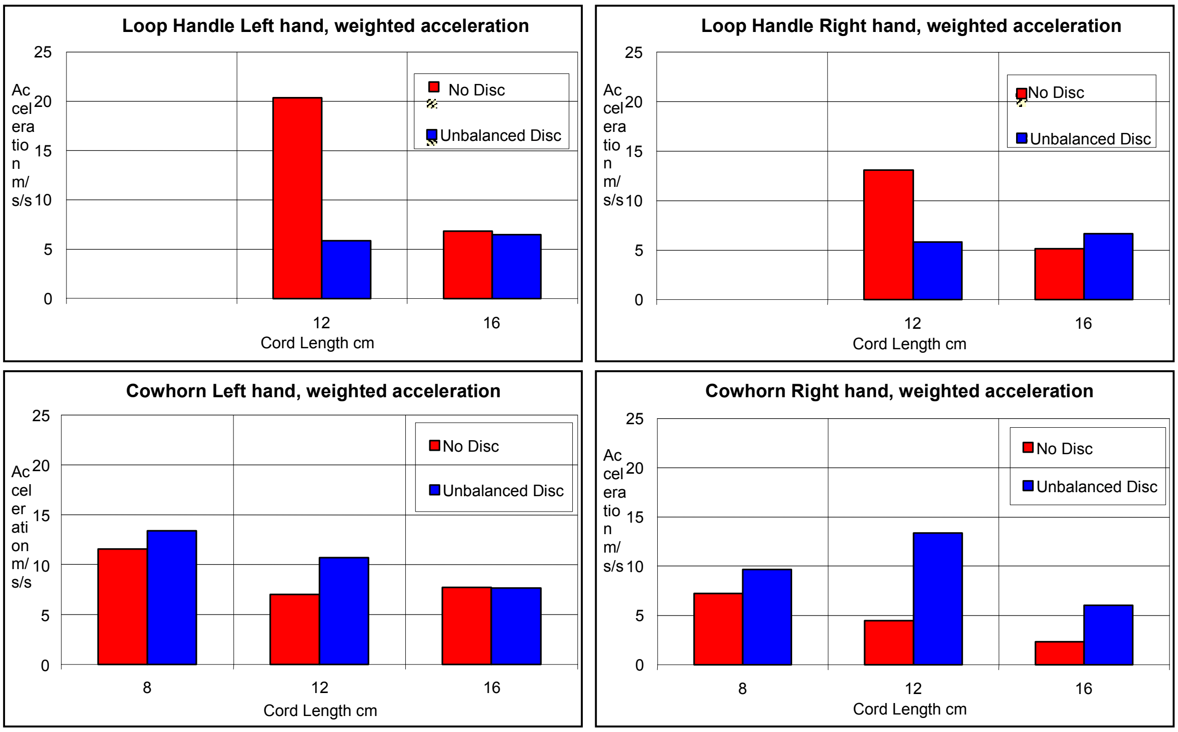


Figure 4: Vibration magnitudes (weighted rms acceleration, RSS) at the handles of two types of strimmer – Upper loop handle; Lower cow-horn.

The measurements were made according to the protocol outlined in Appendix A, and they were repeated with the addition of the unbalanced disc, also described there. Figure 4 illustrates several points:

- The magnitudes of vibration, at 5 to 20ms^{-2} were similar to those obtained in a previous study (Author's unpublished report to the Environment Agency)
- The Cow-horn machine showed a consistent increase in vibration with speed.
- The Cow-horn machine showed higher vibration with the additional unbalanced disc.
- The Loop-handle machine showed a very marked increase of vibration with speed.
- The Loop-handle machine showed a lower vibration with the additional unbalanced disc.

These suggest that there was relatively little unbalance in the head of the cow-horn machine before the addition of the artificial disc, but that there was unbalance in the head of the loop handle machine which was, by chance, largely balanced out by the artificial disc. They also suggest that there was a resonance in the loop handle machine, but not the cow-horn, that was excited at the higher speed, and that it was excited by the unbalanced head.

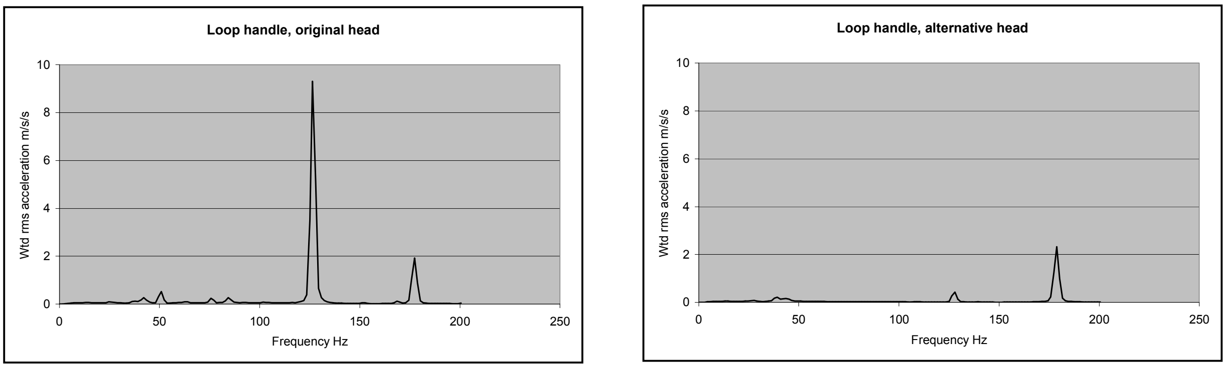


Figure 5: Frequency analysis of handle vibration: Loop handle with two different heads..

These findings are confirmed by the frequency analyses (Figure 5), which show clearly the separate peaks for engine and head rotation, the latter being at about 0.71 of the frequency of the former. The frequencies and engine speeds for each setting of cord length are given in Table 1 below:

Table 1: Frequencies of engine and head components at different cord lengths

Cord length mm	Engine speed rev/min	Engine frequency Hz	Head frequency Hz
160	9000	150	107
120	10800	180	128

4. INVESTIGATION OF COMPONENTS

Before analysing the machines in depth, some experiments were made by exchanging or adjusting components. These had three aims:

- To investigate whether the source of head excitation was in the rotating moulding or in the spigot on which it was mounted.
- To investigate how much of the high vibration of the loop handle machine was due to the handle itself.
- To investigate how sensitive the handle vibration was to the amount of compression of the anti-vibration mounts.

4.1. EXCHANGE OF CUTTING HEAD AND DRIVE GEAR

Each of the loop handle and cow-horn machines was tested free-running at a speed of 10,500 to 11,000 rev/min. In order to avoid misfiring, this required the cord to be lengthened slightly, to 130mm. Each machine was tested with the original head and drive gear, and then with every possible combination of head and drive gear.

From Figure 10, which shows the drive spigot, it can be seen that there are six possible orientations of the head on the drive gear, mating with the hexagonal body of the spigot. Measurements were made for all six, and results for the average values are compared in Table 2a and b, below. These tables show magnitudes for the head component only.

Table 2 a: Vibration magnitudes (head component) with different head units and drive gears fitted to the loop handled strimmer, free-running with 130mm cord.

Head unit	Gear unit	Wtd rms ms^{-2} LH	Wtd rms ms^{-2} RH	Engine rev/min
Loop	Loop	18.5	11.1	10,500
Loop	Cow-horn	20.4	12.8	10,650
Cow-horn	Cow-horn	1.6	1.1	10,750
Cow-horn	Loop	5.0	3.0	10,500

Table 3b: Vibration magnitudes (head component) with different head units and drive gears fitted to the cow-horn strimmer, free-running with 130mm cord.

Head unit	Gear unit	Wtd rms ms^{-2} LH	Wtd rms ms^{-2} RH	Engine rev/min
Loop	Loop	13.6	11.5	10,800
Loop	Cow-horn	11.8	11.5	10,800
Cow-horn	Cow-horn	4.8	1.1	11,000
Cow-horn	Loop	5.0	4.5	11,000

From these results it can be seen that the moulded head unit for the loop handled machine has far more unbalance than the equivalent unit for the cow-horn machine. Visual examination could determine no differences, so some further tests were made, reported in the next chapter.

Any differences between the drive gears were smaller and not consistent.

4.2. EXCHANGE OF HANDLES

The loop handled machine was tested with its original head and drive gear, but fitted with the handle from the cow-horn machine. This involved exchanging the complete handle moulding (Figure 6) so that the engine controls could be connected. Table 3 shows vibration magnitudes for both head and engine components with each handle fitted.

Figure 6: Handle moulding

Table 3: Effect of fitting cow-horn handle to loop handled machine, weighted rms acceleration for engine and head components of vibration, free-running with 130mm cord.

Handle	Engine component LH	Engine component RH	Head component LH	Head component RH	Engine rev/min
Loop	2.7	2.0	18.5	11.1	10,500
Cow-horn	3.8	0.6	10.6	10.2	10,600
Cow-horn machine with loop head and gear:					
Cow-horn	4.2	0.8	13.6	11.5	10,800

These results suggest that it is the loop handle itself that is responsible for the excessive magnitude of head-excited vibration (18.5 ms^{-2}) compared with the cow-horn handle (13.6 ms^{-2}).

Concerning the engine vibration, the left end of the cow-horn handle responds far more strongly than the right end. The asymmetry of the cow-horn handle can be seen clearly in Figure 15 (below).

4.3. COMPRESSION OF ANTI-VIBRATION MOUNTS

The handle moulding (Figure 6) has a clearance of ±6 mm between its internal bore and the main shaft of the strimmer. It is located and supported between the engine unit and a bracket which clamps to the shaft, by means of two elastomeric bobbins at each end (Figure 7). The amount of pre-load on these mounts is determined only by whoever assembles or services the machine at the moment when the screw that clamps the mounting bracket is tightened.

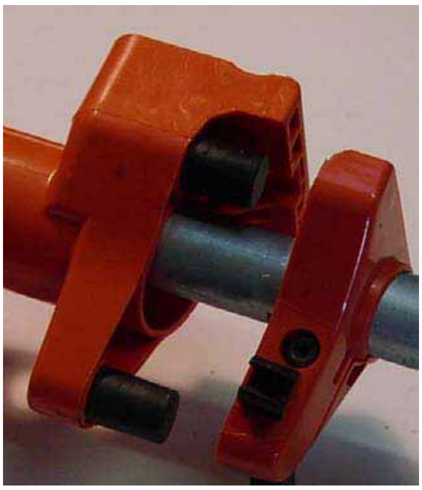


Figure 7: Anti-vibration mount detail showing elastomeric bobbins

In order to determine the sensitivity of the handle vibrations to the pre-load of the mounts, and hence the amount of care necessary by those who service the machine, a short series of tests was made on the loop handle machine in a high vibration condition, i.e. with the original head and gear units and the cord length set to 130mm.

- For the first of these the bracket was loosened, and with the shaft held vertically allowed to drop by its own weight on to the mounts when it was tightened to give the “loose” condition.
- For the “tight” condition, the bracket was pushed against the mounts manually with what was considered to be excessive force.
- An intermediate position mid-way between loose and tight
- The original position, marked before the first separation of the unit.

The results were as shown in Table 4, for head and engine components separately.

Table 4: Effect of pre-load of anti-vibration mounts, loop handle machine, 130mm cord.

Pre-load	Engine component LH	Engine component RH	Head component LH	Head component RH
Loose	3.6	2.4	21.4	12.4
Tight	3.4	2.7	27.5	18.2
Intermediate	3.1	2.3	22.2	14.1
Original	3.2	2.1	24.1	14.3

From these results it appears that assembly of the anti-vibration mounts does not require great precision. It is necessary only to avoid excessive force during the initial compression, which could result in a 20% increase in transmitted vibration. Setting the pre-load too loose does not have a deleterious effect on vibration, but it could affect the handling of the machine and would probably be avoided for that reason.

4.4. CONCLUSIONS ABOUT COMPONENTS

- Some head units can be much better balanced than others. In these tests, that fitted to the cow-horn machine was clearly better than that supplied with the loop handle machine.
- The loop handle itself contributes significantly to the high level of vibration experienced by the operator.
- The fitting of the anti-vibration mounts is not particularly sensitive, provided that extremes of tightness and looseness are avoided.

5. DETAILS OF HEAD UNBALANCE

It was found (4.1 above) that there could be large differences in the unbalance of different head units. Those tests included measurements for all 6 possible orientations of the head on its drive spindle. Any variation between these positions might help explain some of the large measurement variability found with this type of machine. The way in which the cord is wound onto the bobbin could obviously affect the unbalance of the complete unit, so this was tested. And finally, it was observed that the internal “bobbin” on which the cord is wound could have any of 4 orientations relative to the external case of the head unit, so tests were made to evaluate the significance of this.

5.1. ORIENTATION OF HEAD ON SPINDLE

Figure 8 shows one example of how the orientation of the head on its drive spindle can affect the vibration magnitude significantly. This is an apparently extreme example, showing a range of more than 2 : 1 between highest and lowest. Other examples show much smaller variations, although this may be because the troughs are very narrow and do not coincide with any of the available orientations.

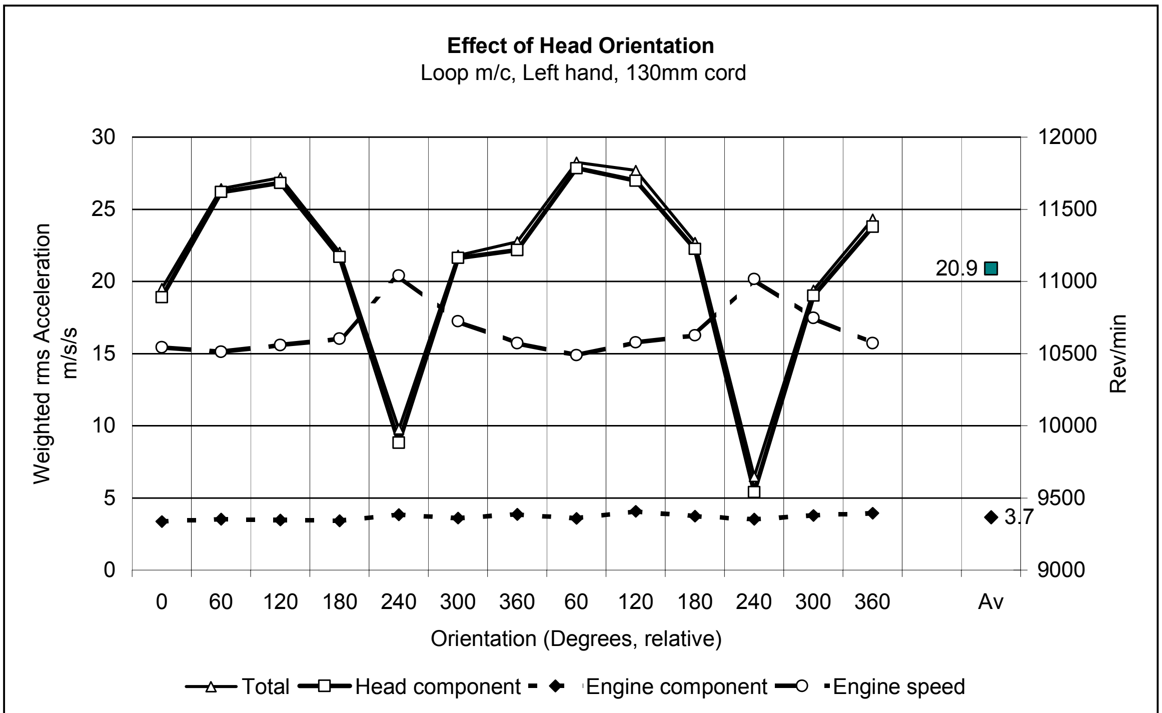


Figure 8: Effect of orientation of cutting head on vibration, showing total, engine and head components.

It may be observed that engine speed varies in inverse proportion to the vibration magnitude. These measurements do not enable a distinction to be drawn between cause and effect. However, it seems likely that the increases in engine speed at the low vibration points, which occur quite consistently at 360^{o} intervals, arise because less energy is being dissipated in vibration, and more particularly in the arms of the operator.

Subsequent measurements associated specifically with head rotation were made for all 6 orientations, and the results averaged. Only for those cases where a constant head excitation could be assured were measurements made with fewer orientations, e.g. for anti-vibration mount pre-load as above.

5.2. LOADING OF CORD WITHIN HEAD

Two operators were asked to load the bobbin with a minimum of 1.5m of cord, then about half fill it (approximately 5m), and then to fill it (just less than 10m). One operator was given no other instructions. The other was asked to load the cord as evenly as possible, pulling it tight every few turns. The results of free-running tests with the loop handle machine and 130mm free cord length are shown in Table 5 as RSS values for the head component of vibration on each handle.

Table 5: Effects of cord loading, RSS ms^{-2} of head component of handle vibration

Operator	Load	Wt of head, g	LH	RH
1	Minimum	218.1	17.5	11.9
1	Half	230.5	21.6	13.7
1	Full	254.5	23.8	16.3
2	Minimum	217.2	18.6	12.0
2	Half	231.0	20.9	13.4
2	Full	253.0	20.0	14.0

The values in the table are averaged over 12 orientations of the head on the spindle, but take no account of the internal orientation of the bobbin (see below). They apparently provide some indication that the heavier the bobbin, the greater the unbalance, and that there is not a great difference between the winding techniques.

5.3. ORIENTATION OF BOBBIN WITHIN HEAD

This potentially confounding variable was tested on the loop handle machine, in its original condition, with a cord length of 130mm, and a quite small amount of cord on the bobbin. Results are shown in Table 6, below.

Table 6: Effect of bobbin orientation within casing of cutting head, RSS ms^{-2} of head component of handle vibration.

Orientation (Deg)	**LH** mean *(range)*	**RH** mean *(range)*	Engine rev/min
0	21.7 *(16-27)*	14.3 *(10-17)*	10,500
90	21.8 *(8-28)*	15.1 *(5-20)*	10,600
180	14.5 *(5-22)*	8.4 *(3-13)*	10,600
270	22.8 *(16-28)*	14.2 *(10-18)*	10,600

The implications of these figures for the reproducibility of vibration measurements are clearly of considerable concern. There is a range of 1.5 to 1 (14.5 to 22.8) for the left hand arising from the bobbin orientation, in addition to a range of up to 4 to 1 (5 to 22) from orientation on the spigot. Equivalent values for the right hand are 2 to 1 (8.4 to 15) and 4 to 1 (5 to 20). Overall the range is about 6 to 1 (5 to 28 or 3 to 18)

Although the orientation on the spigot was included in the trials of cord loading, bobbin orientation was not controlled, so the results in 5.2 above are of doubtful value.

Concerning the design of these machines for lower vibration, these results suggest that the root cause of head unbalance, which is associated with those that have more severe vibration, may lie in the location of the bobbin in relation to the casing, and therefore in relation to the axis of rotation.

5.4. CONCLUSIONS RE HEAD UNBALANCE

- For a head with high unbalance, the combination of possible orientations of the head and bobbin can be responsible for a range of 6 to 1 in the handle vibration. This has important implications for testing and evaluation
- The investigation of loading the cord was not conclusive.

6. AUTO-BALANCERS

6.1. DESIGN OF EXPERIMENTAL AUTO-BALANCER

Two similar devices were designed to fit behind the cutting head of the strimmers (Figure 9, Figure 11). One of these had a track for 12mm diameter balls, the other a track for 6mm balls. The overall size of the devices was similar to that of the head itself. However, the chosen material, which was steel, meant that the devices added 1.6 kg to the mass of the heads. The devices were fitted with bushes that provided a close fit on a register on the driven flange at the head. An alternative spindle was made to accommodate the thickness of the central web of the device (Figure 10), also having a register of the diameter to be a push fit in the locating bush. The cover of the toroid was attached by set screws to allow changing the number of balls between tests.

A preliminary test (see chapter 4, below) had shown that an unbalance of 5g at 50mm radius was of a similar order of magnitude to the actual head unbalance, and would easily be balanced by the balls within these devices.

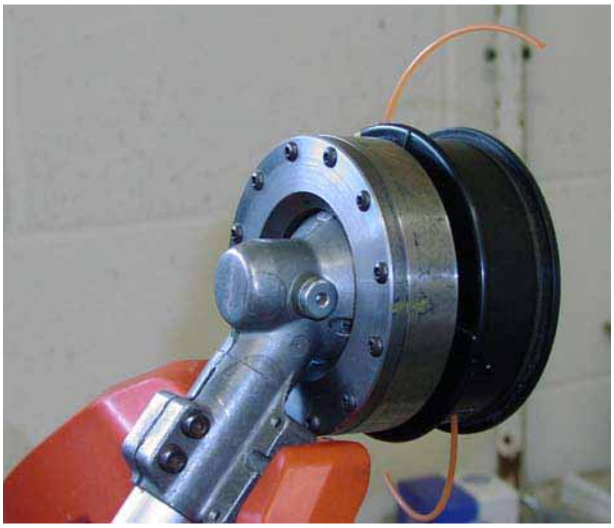


Figure 9: Experimental auto-balancer fitted behind cutting head

Figure 11: Components of experimental auto-balancer – Left, 6mm balls; Right, 12mm balls

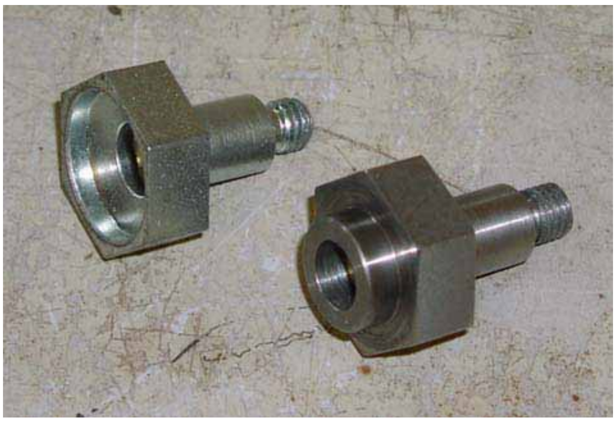


Figure 10: Spigots for cutting head – Left, original; Right, modification for auto-balancer

6.2. TESTS OF EXPERIMENTAL AUTO-BALANCER

The devices were tested on the loop handle strimmer, using the methods described in chapter 4, below. These gave total (Root-Sum-of-Squares or RSS) weighted rms accelerations for each handle, together with separate components identifiable with the engine and the head rotation. The operating condition was free-running at a speed that had been found to give severe vibration (see Chapter 3). Tests were also made with the addition of the artificial unbalanced disc (Chapter 3).

The device with 12mm diameter balls was tested with the cutting head set to an orientation that gave high magnitudes of vibration in the earlier test (First row in Table 7). It was tried first with no balls, and then with 2 balls and 4 balls. With no balls the vibration magnitudes, measured directly as total weighted rms acceleration, were very much lower then before, probably due to the additional mass and inertia reducing the transmission of energy into the whole system. The addition of balls generally had the effect of increasing the vibration (Table 7 below). This can be seen most clearly for the head component, in 5 out of the 6 cases tested, including the 12mm ball unit, the 6mm ball unit, and the 6mm ball unit with artificial unbalance. Figure 12 (below) shows the test arrangements.

Table 7: Tests with auto-balancers, rms weighted acceleration 3-axis RSS

Test unit	No of balls	No of averages	Left hand			Right hand		
			Head	Engine	Total	Head	Engine	Total
None	N/A	1	18.7	2.5	18.9	9.7	2.6	10.1
12mm balls	0	4	6.1	3.1	6.9	3.8	1.8	4.4
	2	4	5.6	3.6	6.7	4.7	1.9	5.3
	4	2	8.9	3.4	9.8	8.2	1.9	8.6
	8	2	6.5	3.9	7.8	10.1	1.5	10.5
6mm balls	0	2	1.8	4.3	4.9	3.1	1.9	3.9
	12	2	2.3	3.8	4.6	1.9	1.6	2.9
6mm balls + disc	0	1	7.3	4.8	8.9	7.9	1.2	8.2
	12	1	8.1	3.3	8.8	9.3	1.2	9.4

Comparison with the range of magnitudes without the balancer (first line of Table 7) shows that the mass of the test units greatly reduces the effect of the head component. However, the effect of the balls is to increase vibration and not to reduce. This is further examined in measurements of the head response, below.

Figure 12: Head arrangements:Left – standard, Centre – with balancer; Right – also with artificial unbalanced disc

6.3. DISCUSSION OF AUTO-BALANCER PERFORMANCE

It is clear that, in the form tested, the auto-balancer did not counteract the inherent unbalance of the rotating head of the strimmer. It is possible that this was the result of inappropriate lubrication for the balls in the annulus. However, there is a much more likely explanation that has implications for the application of the device to other tools.

This is to be found in the explanation of operation, in which it is seen that the unbalance component must be rotating at a frequency above the first resonance of the unit on its effective suspension. When that is the case, it is whirling "light side out", and the balls migrate towards the light side, so opposing the inherent unbalance.

It has already been mentioned that, in allowing the machine to run down from its operating speed more than one resonance is passed. The modal tests (Chapter 7) confirm that there are at least two resonances below the operating range. Furthermore, the speed used for these tests, at the top of, or just above the normal operating range, coincides with another resonant mode.

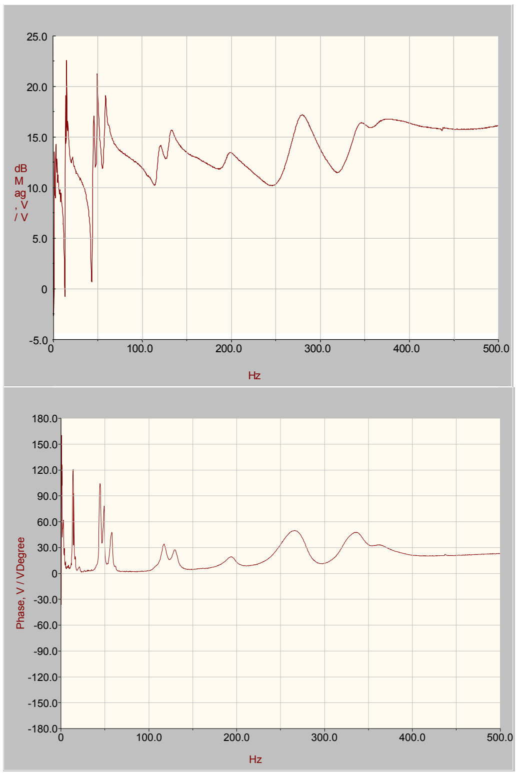
In these conditions, if the forced response of the head end of the strimmer shaft reflects these resonances, the balls are unlikely to be directed to a point exactly out of phase with the inherent unbalance, and could even add to it, as in the case of the washing machine at low speeds (see Appendix A). Figure 13 shows the forced response of the head in various conditions. This was measured with the impact hammer and accelerometer also used in the modal tests. Similar results were found using swept sine and random excitation provided by an electro-dynamic shaker. These show the two resonances below 100 Hz, with the third around 120 Hz to 140 Hz. The test speed used gave a head rotational frequency of 125 Hz to 130 Hz. This would therefore have been in a region where the phase between unbalance force and displacement approaches 90^o, i.e. a transitional region and clearly not whirling "light side out", which is the requirement described in Appendix A**.** It is thus unlikely that at this speed the auto-balancer would have worked.

Further testing would be needed to discover whether the right conditions would be found between the first and second, or between the second and third resonances, and therefore allow the device to work at lower speeds within the operating range.

In theory it might be possible to modify the machine to alter the response of the head end of the shaft such that there was a wider frequency gap between resonances, together with the appropriate phase relationship to ensure that the head would whirl "light side out" throughout the operational speed range. In practice this could be very difficult to achieve without comprehensive changes to the machine. The need to keep the weight down, and particularly the weight at the extremity of the head end of the shaft, renders this approach to be of only academic interest.

At one stage in the project, it was thought that the auto-balancer might also reduce the engine component of vibration. If that were possible, then it would have a very wide range of application. However, even if the engine excitation could be represented as a simple rotating out-of-balance mass, the dynamics of the strimmer shaft would make successful balancing unlikely for the same reason as for the head unbalance. There are some directly out of balance forces in most engines that are light enough to be suitable for portable power tools. However, the engine firing forces, which are probably more important, are translational in nature, and would have to be modeled by a pair of contra-rotating masses. It was not thought that this line of experiment would be worth pursuing.

Case S1: Normal assembly, freely suspended, Transverse



Case S2: Normal assembly, hands on handles, Transverse

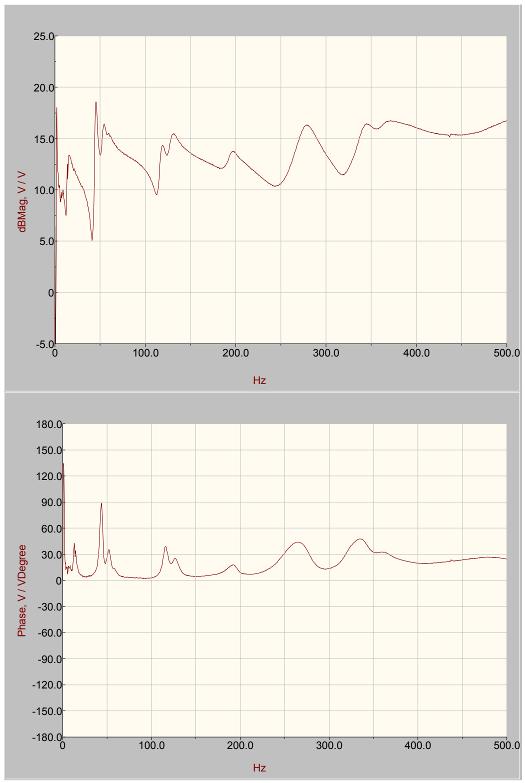


Figure 13: Impact response of head support housing

6.4. CONCLUSIONS ABOUT AUTO-BALANCING

- An auto-balancer fitted to the head of a strimmer did not reduce the unbalance.
- The probable reason lies in the multiple natural frequencies associated with the long shaft separating the head from the handles.
- Auto-balancers are unlikely to be useful for engine balancing.
- There are other types of portable powered tool for which auto-balancers could be effective.

7. MODAL TESTING

Impulse excitation was used to measure frequency response functions necessary for modal test analysis of components, including the loop and cow-horn handles and the shaft of one strimmer with all components attached, and later with the handle moulding removed. Both handles and shaft were tested with and without operators' hands holding the grips, and the shaft was also tested with the additional mass of the balancing device attached. The test points (nodes) are shown in Figure 14-16.

Figure 17 shows the impact hammer and accelerometer which were connected to a Data Physics Signalcalc Ace acquisition system to obtain the sets of frequency response functions. These were then analysed using the ICATS "Modent" suite of modal analysis programs (ICATS, 2001). The measurement range was set to 0 – 500 Hz. The figure also shows the elastic cord used to suspend the machine for these tests.

Figure 14: Nodal points on loop handle

Figure 15: Nodal points on cow-horn handle

Figure 16: Nodal points on shaft (some points on cow-horn handle also shown)

Figure 17: Impulse hammer and accelerometer for modal tests

The objectives were to obtain information that might explain the failure of the balancer, and to indicate whether the present mounting points for the handles are optimum or could be improved to remove them from points of large dynamic response of the shaft. A further objective was whether or not there is scope to improve the response of the handles, particularly the loop handle.

7.1. RESONANT FREQUENCIES AND MODE SHAPES OF SHAFT

The first tests were conducted only on the shaft as part of a complete machine (cow-horn unit). Tests more relevant to optimizing mounting points required the machine to be dismantled, and were made later in the programme. Mode shapes are shown in Figure 18 for the length of shaft between the handle clamping bracket and the cutting head, for the freely suspended condition. Measurements were also made with hands on the handles, and with the balancer unit attached to the head. Resonant frequencies for all three conditions are shown in Table 8, below. The mode shapes were similar in all cases.

Table 8: Natural frequencies of strimmer shaft

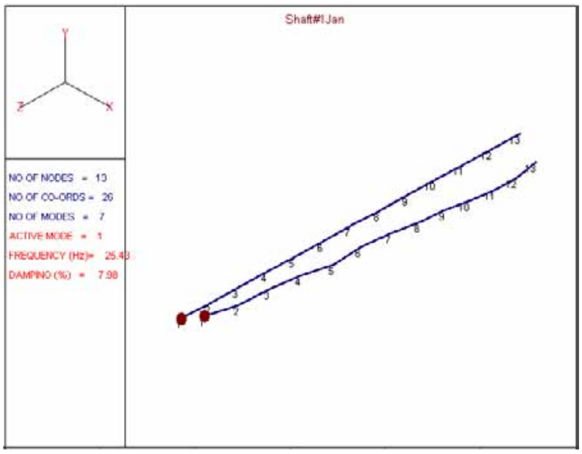
Modal description	Frequency Hz		
	Freely suspended	Hands on grips	Hands with balancer
Rigid body	25.4	27.3	
First bending (anti-nodes at ends)	49.4	53.1	53.9
Second bending (anti-node in centre)	127.7	137	110.5 146.5
Third bending (two anti-nodes)	203.5	233.4	238.4
Fourth bending (3 anti-nodes)	336.1	324.5	332

What is apparent from these frequencies is that the "second bending mode" of the shaft is probably of particular importance in the increase of vibration of the loop handle machine when the speed is raised above 9,000 rev/min. With the 120mm and 130mm cord length, the head excitation almost coincides with this resonance. Furthermore, this resonance (at 127.7 Hz or 137 Hz) is associated with quite large deflections at the handle end of the shaft (point 1 in the figure).

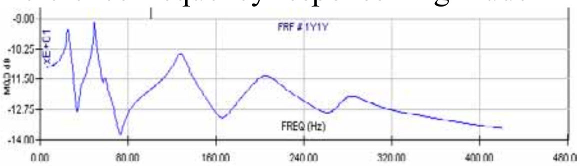
It is not clear from this why the cow-horn handles do not also show a sharp increase in vibration between 160mm cord length and 120mm cord length.

For the tests with the balancer, the excitation frequency of head rotation coincided with a minimum in the frequency response between the peaks at 110 Hz and 146 Hz. This may have contributed to the generally lower magnitudes of handle vibration whether there were any balls or not contained in the device, but it does not explain the lack of effect of the balls. The multiple resonances which affect the shaft, including the head end, are confirmed as the most likely explanation, as already discussed.

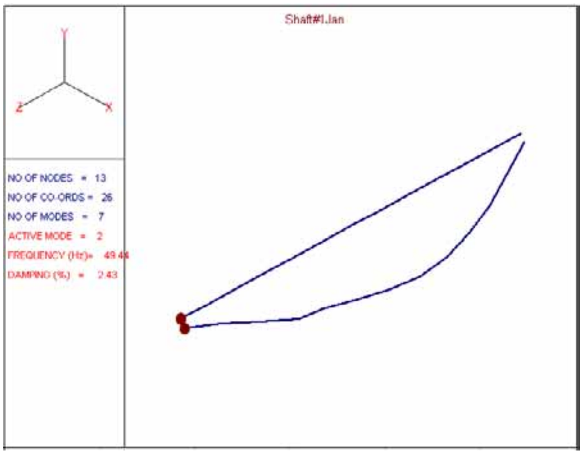
Mode 1: 25.4 Hz



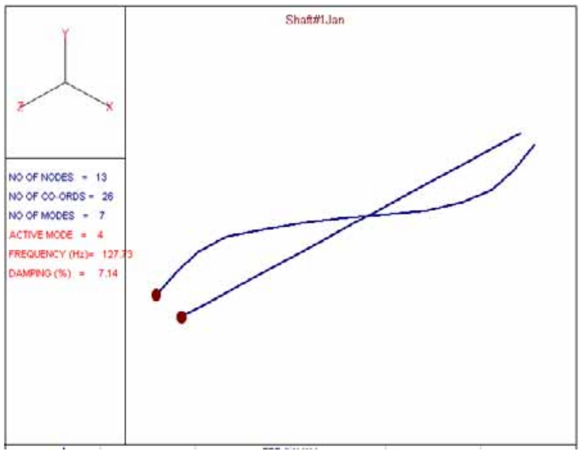
Reference frequency response magnitude



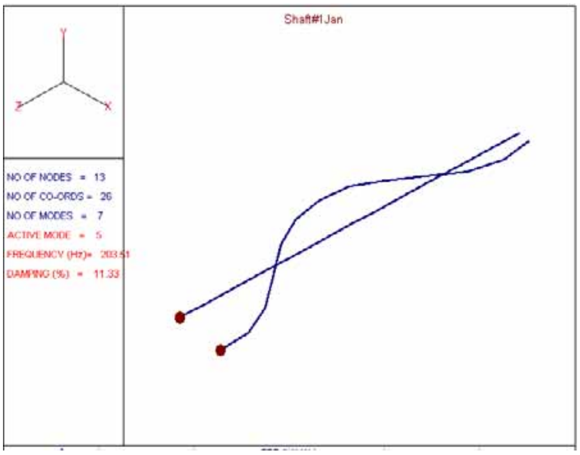
Mode 2: 49.4 Hz



Mode 3: 127.7 Hz



Mode 4: 203.5 Hz



Mode 5: 336.1 Hz

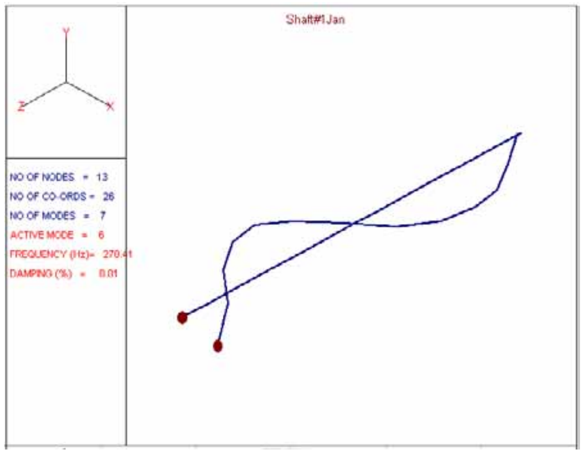
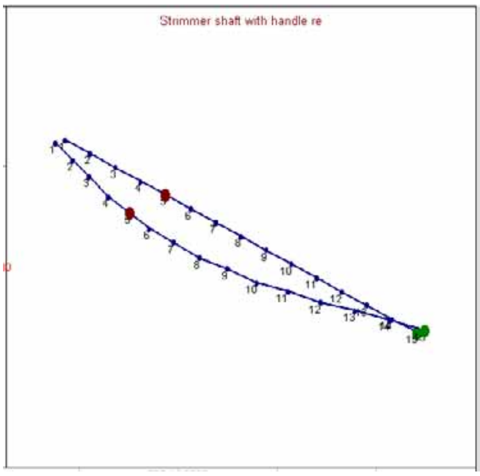
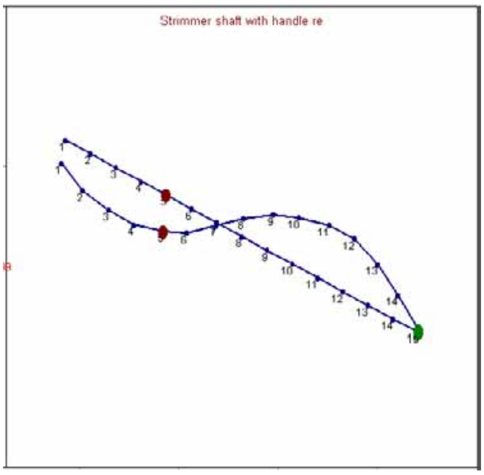


Figure 18: Mode shapes for set-up: Cowhorn machine, freely suspended, standard head, data for shaft between handle and head.

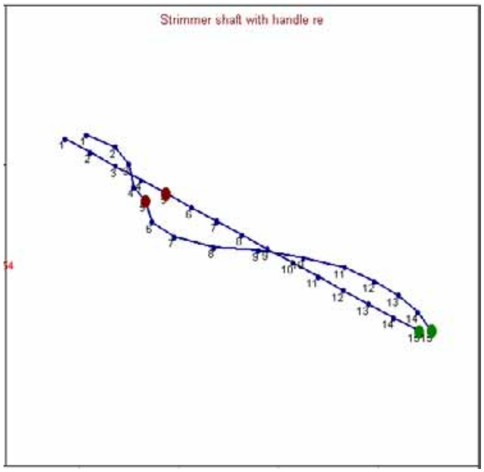
a. Fundamental (18.8Hz)



b. Second harmonic (59 Hz)



c. Third harmonic (129.5 Hz)



d. Fourth harmonic (217 Hz)

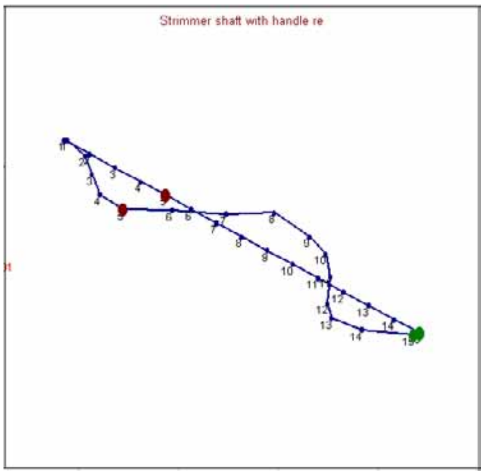


Figure 19: Mode shapes, strimmer shaft, handle mouldings removed, engine and head in place

Modal results with the handle clamp and moulding removed (Figure 19 above) show slight deviations from those of the fully assembled machine. In these figures, point 1 is close to the engine and point 15 is at the head gear casting. The handle bracket would be clamped at point 5 (highlighted) in the complete assembly.

What is now clearly the second harmonic of the shaft, at 59Hz instead of 49Hz, is not constrained to have its node at the clamp, and hence has a shorter wavelength and higher frequency.

More important is the third mode, which appeared to be the second mode when constrained by the handle clamp. The natural frequency of this at 129.5 Hz is close to the rotational frequency of the head when operating with the shorter cord (120-130 mm). The free mode shape would have relatively large displacement at the clamp position (node 5), but would have low vibration at a point between nodes 3 and 4 and close to node 9.

The fourth mode is outside the frequency range of interest, even for the engine excitation.

7.2. RESONANT FREQUENCIES AND MODE SHAPES OF LOOP HANDLE

The loop handle was tested with and without hands on the loop itself and on the control handgrip. In both cases there was a single dominant resonance (Figure 20, Figure 21), with motion of the top of the handle along the axis of the drive shaft, involving pivoting about the mounting bolts (points 1, 15), but also flexure of the lower sections in torsion (elements 3-4, 12-13). The frequency **increased** from **70 Hz** to **95 Hz** with the hand gripping, associated with a change from parallel motion to a twisting motion about the corner gripped by the hand.

These frequencies are significant. For the “free” loop, i.e. without the impedance of the operator’s hand, 70 Hz is not far below the lower end of the operating range. However, with the hand present, the resonant peak approaches 100 Hz, with amplification extending beyond 120Hz. This possibly explains why excitation of the third mode of the shaft feeds through to give the high levels of vibration observed for the loop handle with the 130mm cord length.

There are two possible remedies:
Either add mass to the handle to reduce the natural frequency
Or increase the stiffness to raise it.
These will be considered further below.

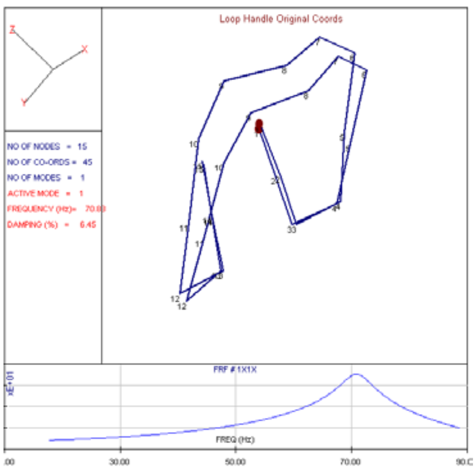


Figure 20: Primary mode shape for loop handle without hand. [Top of grip moves in z-direction (axis of shaft)pivoting about mounts (1,15) and twisting sections 3-4 and 12-13].

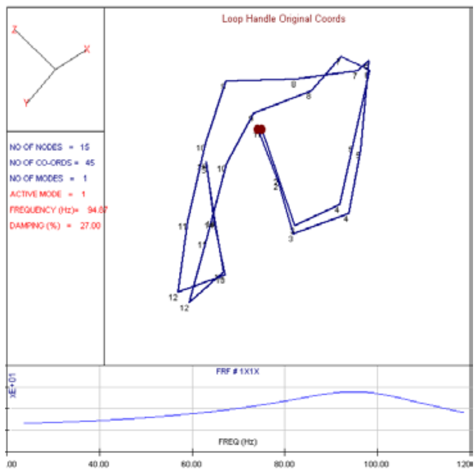


Figure 21: Primary mode shape for loop handle with hand gripping corner (point 7). [Top of grip pivots about hand, in x-z plane, twisting sections 3-4 and 12-13 about 1-3 and 13-15 (point 7 was measured with improper hand grip)].

In Figure 20 and Figure 21, the X-axis is along the axis of the shaft, with positive X towards the engine end.

7.3. RESONANT FREQUENCIES AND MODE SHAPES OF COW-HORN HANDLE

Modal tests were made on the cow-horn handle both freely suspended and with hands gripping. Up to 5 resonances were found in the frequency range 20-250Hz, but there were none between 50Hz and 150 Hz. This explains how the cow-horn was less susceptible to excitation at the third mode of the shaft, this falling in a zone of low response. However, the resonance at 157 Hz (146 Hz with hands) brings higher response close to the engine frequencies, so that magnitudes of the engine component of vibration are not as low as might otherwise have been expected. And the mode shape below also shows how pivoting of the handle about the mounting clamp leads to lower vibration on the right hand-grip than on the left.

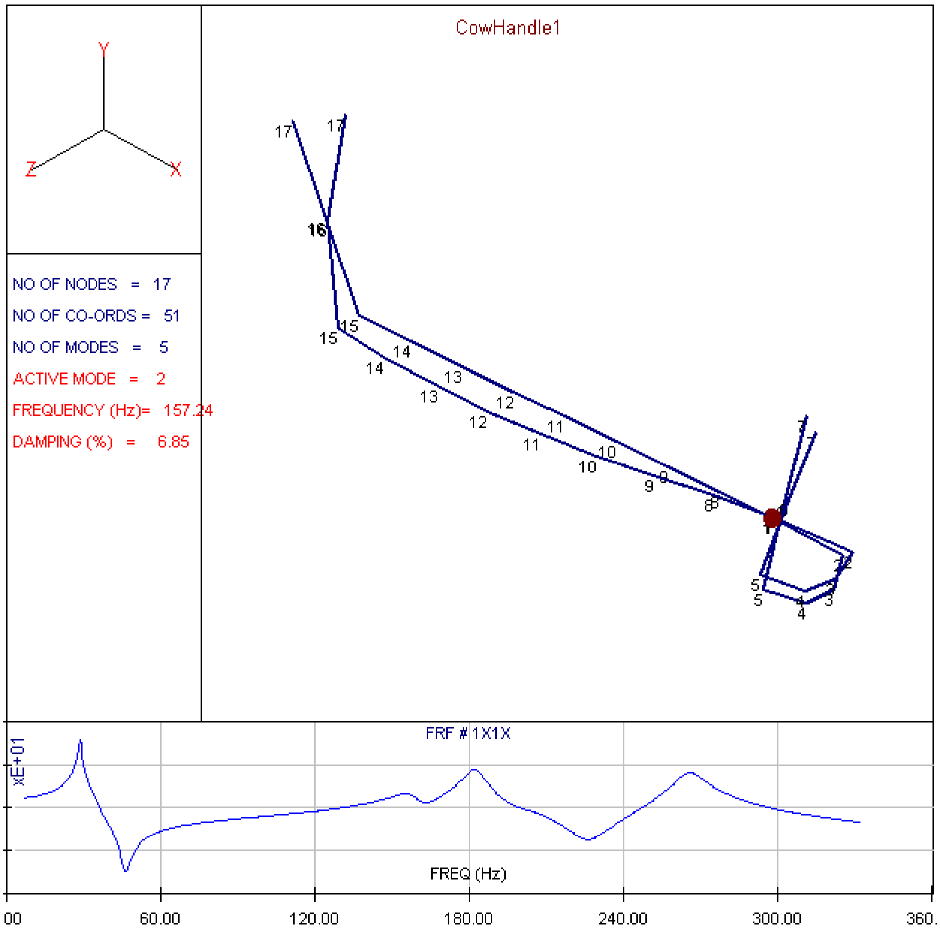


Figure 22: Shape of second resonant mode of cow-horn handle. (Handle moves in x-z plane, pivoting about clamp and grips twist longitudinally. Z-axis is along main shaft, positive towards engine.)

7.4. CONCLUSIONS RE MODAL TESTING

- The shaft of the strimmer shows classic bending modes, one of which has a natural frequency of 129-130Hz, which is responsible for a sharp increase in vibration at the handles of a machine with head unbalance when the machine over-speeds to 10,500-11,000 rev/min.
- The deflected mode shape shows the optimum points for mounting the handle for minimum transmission of vibration at this frequency.
- The loop handle is very flexible, and with the presence of a hand, the first natural frequency is high enough for this to amplify both head and engine excitation within the range of operating speeds. A solution could be either by stiffening the handle, or by adding mass.
- The cow-horn handle has several resonant modes, but most of these are outside the operating range of frequencies. The only exception is one at 157 Hz which may affect transmission of engine vibration, but not that from head unbalance.
- Modal testing provides a useful insight into the dynamic behaviour of the machine and its components.

8. ANTI-VIBRATION MOUNTS

With the discovery that the auto-balancer would not provide a solution to the problem of unbalance of the cutting head of the strimmer, attention was turned to making use of information gained from the investigations into frequency analysis, effects of changing components and natural modes of vibration of the strimmer components. In the first place, there are two aspects of the optimization of the anti-vibration mounts that may be worth pursuing, namely the location of the mounts on the main shaft of the machine, and the impedance presented by the shaft and the handle moulding at the mount position.

8.1. LOCATION OF MOUNTS

It is first necessary to consider which of the natural frequencies (resonant frequencies) of the bare shaft might be relevant to vibration at the handles in normal operation. For this it is necessary to consider the frequency ranges of head rotation and engine firing. Estimating that the engine speed may drop as low as 7,000 rev/min with a long cord (160mm) when under the load of cutting grass, and could rise to nearly 11,000 rev/min when the cord reduces to 120mm and the load is removed, we have frequency ranges as in Table 9, below.

Table 9: Likely frequency ranges for head and engine components of vibration in normal work

Engine rev/min	Frequency of head rotation Hz	Engine firing frequency Hz
7,000	83	117
11,000	130	183

We are therefore most interested in the third bending mode of the shaft, at 129.5 Hz, which could be excited by the head under over-speed conditions, and by the engine in mid-range.

The existing mount locations are presumably dictated by practical considerations. The mounting arrangement depends on compressing four elastomeric bobbins between the ends of the handle moulding and, at one end the engine cover, at the other a bracket clamped to the shaft. The position of this clamp is in turn set by the shortest length of handle moulding that will allow an ergonomically sound hand position for the operator. That is one in which the machine is naturally balanced, and the cutting head can be controlled near to the ground, without need for the operator to exert any great force, nor to adopt a stressful posture.

There are two further practical requirements in the design of the handles and associated mouldings. These are to (1) provide a protected path for wires and cable between the engine and the hand controls, and (2) space the mounts far enough apart along the shaft to provide adequate control of the position of the cutting head.

As already mentioned in section 7.1 (above), transmission of vibration at or near to the frequency of this mode could be minimized if the mounts were located elsewhere from their present positions. In the Figure (Figure 19c), the optimum locations are between nodes 3 and 4, and close to node 9. The first of these is between the existing mount points, roughly where the operator's control grip is situated on the loop handle machine. The second is about 400mm nearer to the head end of the shaft than the present clamping bracket.

These locations would provide adequate spacing for control of the machine, but would need considerable development of the mount details, and a revised design of handle moulding. This amount of work was beyond the scope of the present project, but is not beyond the bounds of what is practical. It might also allow the details of source and receiver impedance to be addressed more effectively (see below).

8.2. IMPEDANCE OF "SOURCE" AND "RECEIVER"

As shown in 2.4, above, the isolation effectiveness of an anti-vibration mount system, for all but very low frequency motion, depends on having soft mounts between points of high impedance on the source and the receiver, i.e. parts that are locally either massive or stiff.

In the present case, the flexibility of the mounts themselves is constrained by the need to provide stable control of the long shaft extending beyond them, although the possibility of a system having softer mounts should not be altogether discounted.

Of more interest, the source points, as well as being unfavourably located with regard to natural modes of vibration of the shaft, are not both at positions of high mechanical impedance. That at the engine end may be well designed from this point of view, being close to a center of mass, and having a relatively well-supported plastic moulding. However, that moulding might be shown to separate the mounts from the mass to an undesirable extent. This has not been tested.

The other source point is in a highly flexible part of the shaft, and comprises a very light plastic moulding, weighing less than 40g. There is clearly scope for increasing the source impedance for this mount, by adding mass to the bracket, such as by using metal instead of plastic.

The handle moulding which forms the receiver is necessarily light, to avoid adding unnecessary mass to a machine that has to be easily portable. As a result, its impedance at either end is restricted to what can be achieved through the rigidity of the moulding. As for the bracket, above, there are possibilities for adding some mass in these areas.

The scope of the project allowed only for testing one of these possibilities, and the simplest was to add some mass to the shaft immediately adjacent to the mounting bracket (Figure 23). The added mass, of 400g, was in the form of a split steel annulus, strapped tightly to the shaft. The result of a simple comparison under the severe conditions of free-running with 130mm cord length is shown in Table 10 below.

Figure 23: Mass loading to increase shaft impedance at mounting bracket.

Table 10: Effect of added mass at bracket for anti-vibration mount, RSS weighted acceleration, Left hand (Loop), 130mm cord length

Condition	Head component	Engine component	Total
Original	13.2	2.6	13.5
Added mass	7.6	4.0	8.9

These are values from a series of brief tests in which the head, and hence the head excitation, was not disturbed. Values for the "original" condition were repeated before and after the whole series, and changed by about 10%. That change was probably due to the speed changing from 10400 to 10500 rev/min. Results of the other tests relate to the next section.

The improvement at the head rotation frequency of 35% is to some extent offset by higher vibration transmission at the engine firing frequency. Clearly the modification needs more thorough testing, and possibly some optimization could be made. However, this result is sufficiently encouraging to suggest that there is some benefit in increasing the impedance of the source and/or receiver associated with an anti-vibration mount.

8.3. CONCLUSIONS RE ANTI-VIBRATION MOUNTS

- There is scope for reducing vibration by modifying the handle moulding to enable it to be attached to the shaft at points of lower vibration than at present.
- The vibration transmission across the anti-vibration mounts could be reduced by increasing the mass of either the moulding or the shaft close to the mounts themselves.
- Neither of these potential improvements has been tested conclusively.

9. MODIFICATIONS TO LOOP HANDLE

It was shown in Chapter 5 that, despite every other detail being the same, the loop handle had very much higher levels of vibration than the cow-horn handle. It was shown in Chapter 7 that the cow-horn handle had no resonant modes that were likely to be excited by head rotation, but that the loop had one mode that could be so excited. It was suggested, in Chapter 7, that there were two possible approaches to changing the response of the loop handle, by increasing either its mass or its stiffness.

Because the addition of the mass (impedance) of the operator's hand actually served to ***increase*** the resonant frequency (as described), it is doubtful whether adding a lumped mass would have a beneficial effect. However, it was simpler to do this than to increase the distributed mass of the handle, so a test was run with 200g added beneath the operator's hand.

The loop is subjectively felt to be very flexible, with most movement as shown by the mode shape (Figure 20) being along the axis of the main shaft. It is very easy to increase the stiffness in this direction radically. This may be done by triangulating the top of the loop to the mounting bosses on each side (Figure 24).

Figure 24: Loop handle with added stiffness.

These two basic options were tested in conjunction with the test of source impedance in 8.2 above, with the results shown in Table 11, below.

Table 11: Effect of modifications to loop handle, RSS weighted acceleration, Left hand (Loop), 130mm cord length

Condition	Head component	Engine component	Total
Original	13.2	2.6	13.5
Added mass	22.0	2.0	22.1
Added stiffness	8.8	2.8	9.3

Adding mass in a single position close to the point of the hand-grip does not have an advantage for head excited vibration, although it is possibly beneficial at higher frequencies (see engine component). However, this does not exclude the possibility that adding a distributed mass might not be useful in lowering the first resonant frequency.

Increasing the stiffness clearly does have some potential benefit, but the design would need to be considered carefully because of potential interference with the operator's ease of changing the hand-grip position.

10. MAINTENANCE

It is often suggested that lack of maintenance can lead to deterioration in power tools such that the severity of vibration increases. With regard to the ic engine powered strimmers in this study, there are not many ways in which that can happen. Some possibilities are discussed below.

First, with regard to the state of tune of the engine, there are two aspects that could affect the vibration at the handle: misfiring and over-speeding.

Misfiring was not addressed in the tests described above. Misfiring would introduce sporadic vibrations of a much lower frequency than the normal rotation and firing frequencies. As a result of the frequency weighting for human response, these lower frequencies would assume greater importance. Maintaining the engine and its settings so as to avoid misfiring is important for all machines powered by ic engines.

Over-speeding, on the other hand, was included in the tests, specifically because of the important effect on handle vibration, particularly in the case of the loop handle machine. This has been shown to arise because of a resonance of the strimmer shaft at a frequency excited when over-speed occurs. It is therefore important that engine settings should not be modified to allow over-speeding to occur with the lengths of free cord which are typical of normal use.

Although not strictly a matter of maintenance, another way of reducing the likelihood of over-speeding is to ensure that the free cord length is not allowed to become too short in operation. This might be achieved by ensuring that the operator made frequent adjustments, but with the simplest type of cutting head, that is time-consuming. Therefore the use of automatic or semi-automatic (“bump”) heads is preferable. With “bump” heads, the spooling out of fresh cord is under the operator’s control. With fully automatic heads it is dependant on the correct operation of those devices themselves, and it has been found in other trials (Author’s unpublished report to the Environment Agency) that these do not always prevent over-speed to the extent required. It is therefore a matter of the operator being sufficiently observant to know when a replacement head is needed, or of the maintenance agent having a suitable test for operation of these heads when the machines are serviced.

Similarly, the elimination of heads with severe unbalance, which is the main problem highlighted by this study, needs either operator vigilance or a suitable test for the maintenance agent to use to check this at service intervals.

It is conceivable that damage to the final drive gear at the head end of the shaft, or wear therein, could contribute to an elevated level of head excited vibration. This has not been studied in these tests, which were confined to new machines. However, it has been shown that quite excessive magnitudes of vibration could be excited with no obvious contribution from this gear. It would need a survey of machines after extensive use to ascertain the importance of this and then to suggest any remedial action.

With regard to the anti-vibration mounts, in their present form these are made from a material that can harden with use. They are sufficiently simple, and therefore presumably cheap, to be replaced at regular intervals during routine servicing.

There are therefore 5 points to be made with regard to the contribution of maintenance to the control of operator exposure to vibration:

- Engine tune should eliminate misfiring
- Engine tune should limit possibilities for over-speed operation
- Operators need to be observant for incorrect operation of automatic heads, which should be tested during routine servicing.

- Operators need to be observant for increased unbalance of heads, which should be tested during routine servicing
- Anti-vibration mounts should be renewed regularly.

11. COMMENTS ON MEASUREMENT AND TESTING

The measurement of vibration on the handles of strimmers has been found to be a very imprecise science. This study has discovered two reasons for this.

First, the main source of excitation in what might be termed “problem” machines, i.e. the unbalance of the cutting head, can change according to the location of the head on the drive spigot, and according to the orientation of the cord bobbin in relation to the casing of the head unit.

And secondly, the highly resonant nature of components such as the shaft and the handles of this type of machine can lead to sudden changes in vibration transmission when small changes of operating speed cause excitation frequencies to coincide with natural modes of vibration.

With regard to orientation of the cutting head, location on the spigot can cause large changes when the head unit is re-mounted after loading fresh cord. Orientation of the bobbin can cause large changes when the bobbin is turned in the process of adjusting for shortening of the cord during use.

Both of these can probably be eliminated, or at least greatly reduced by discarding heads with high unbalance, although this has yet to be tested conclusively

With regard to measurement of machine “emission” levels, as in type testing, care is needed first to ensure that the chosen test speeds do not bring the excitation close to any particular resonance. One possible way of checking this would be to test at 2 or 3 closely spaced speeds, differing by perhaps ±5%, instead of increasing the number of replicates at a single speed.

It would then be necessary to ensure that unbalance of the head is within acceptable limits. This would require testing with at least two orientations of the head. Such a test (for head unbalance) would be more sensitive if the test speeds did actually coincide with a resonance. Also, such a test could be used to eliminate bad heads, either on receipt of new machines, or at service intervals.

Unless manufacturers can show that they have reduced head unbalance to very low levels, it must be necessary to include an allowance for this when estimating the “k” values for variability of emission values, to add to the measured mean acceleration magnitude. Otherwise, external quality checks are likely to present machines that do not conform to manufacturers’ claims.

The tests considered thus far were all made under steady state conditions, for which short measurement periods, of the order of tens of seconds, are sufficient. With regard to measurement of operator exposure in the field, it is generally accepted that longer measurement periods are needed to include representative ranges of operating conditions. The existence of the variations highlighted by this study serves to extend further such measuring periods. It must be necessary to include enough time for the bobbin in the cutting head to pass through all of its available orientations. A suitable duration might be the time it takes for one complete loading of cord to be used up, provided that all 3 axes on each handle can be measured together. Even this will not take into account the possible variations arising from different mounting of the head on its spigot. Furthermore, if an adequate sample for a group of machines is required, then it will be necessary to make measurements on several machines to include several different heads, unless and until the source of variation between heads has been eliminated.

12. APPLICATION OF TECHNIQUES TO OTHER TYPES OF MACHINE

Other makes of strimmer and brush-cutter use similar designs to those of the models described above, particularly with regard to location of mounts in "low vibration" models. The techniques and processes are therefore applicable to the machine type rather than just the make selected.

When it comes to different types of machine, there are some very specific characteristics of strimmers that would be found only rarely. However, some or all of the techniques are likely to be applicable to controlling the vibration at the handles or grip surfaces of almost any portable or hand-guided powered machine.

All such machines will have sources that excite vibration, generally in the form of the driving motor or a rotating or reciprocating working piece or the interaction between the working piece and the material being worked on. Many of them, even while appearing to be rigid bodies, exhibit dynamic flexibility at the frequencies of their exciting sources. And many again incorporate flexible isolators for the motor, for the handle(s), or somewhere in the system between these. Understanding how the various components in this system react under dynamic excitation is the first step towards reducing the levels of vibration.

Drive gearing often produces excitation at more than one frequency. Frequency analysis of vibration can then remove much of the trial and error from any exercises aimed at identifying or separating sources of excitation. In conjunction with tests at different speeds, frequency analysis can also show the influence of any resonances in the machine, and may show the effectiveness of any isolation systems that are already used.

In cases where machines exhibit large variation in measured vibration magnitudes, the methodical exchange of components between examples with high and low magnitudes can narrow down the part at the root of the problem. Often, restricting the comparison to a sensitive frequency band can increase the sensitivity of these comparisons.

Where rotating unbalance is an important source of vibration, tests of the relevant component on the machine may be the simplest way of separating acceptable from unacceptable specimens. Eventually this could lead to specification of limits of acceptability in terms of unbalance measured on a test rig, as part of quality control, as is the case with the discs of grinding machines. A test rig was not used to check the unbalance of strimmer heads in this project, but may become necessary to control production quality.

Also where rotating unbalance is important, the application of auto-balancing should be considered. This may not be useful for flexible machines with multiple resonances, but it has proved very effective for hand-held grinders. Specialist advice on the application of these is available to add to the basic theory contained in Appendix A.

Modal analysis of components can identify resonant frequencies. Used in conjunction with frequency analysis of vibration, this provides further information about where improvements may be sought. Modal analysis also shows which parts of components have the potential for the greatest movement, and which have the least. Those parts with the least movement are where connections with other parts should be made for low energy transfer. Walk-behind machines with long tubular handles can be designed so that the hand-grip is in an area of low energy for resonant modes near to the main excitation frequency. On some plate compacters this is achieved by adding mass near to the hand-grips. Such an approach is more acceptable for pedestrian-controlled machines than for those that are purely portable.

The low energy points identified by modal analysis are also points of high impedance (around the relevant resonant frequencies) and so are the points best suited for the use of anti-vibration mounts to join components. The basic principles of vibration isolators are applicable to any machine, not least the need for either the source, or the receiver, but preferably both, to possess high mechanical impedance. Flexibility of the mounts themselves is not enough, as the simplest theory shows. In some cases, the introduction of flexibility can increase vibration.

And finally, modal analysis, or any other means of investigating which parts of a structure move most freely under vibration, shows how best that structure can be stiffened, if stiffening is the best way of reducing response to the driving excitation.

13. CONCLUSIONS

13.1. DESIGN POSSIBILITIES

13.1.1. Control vibration at source (unbalance of cutting head)

- Unbalance of the cutting head is a major source of handle vibration in strimmers.
- It cannot be controlled by simple application of an auto-balancer such as do improve grinding machines.
- It is important to eliminate heads with large unbalance by selection
- Auto-balancing could still be useful for other types of machine

13.1.2. Optimisation of anti-vibration mount (impedance of shaft and bracket)

- The efficiency of anti-vibration mounts depends on the impedance of the structures to which they are attached.
- Increasing the impedance of the strimmer shaft by adding mass close to the handle attachment reduced vibration at the handle by up to 40% in some cases.
- A softer anti-vibration mount would have to be designed with care to ensure good controllability of the machine, and adequate durability.

13.1.3. Use of modal information (1): Choice of mounting position for handle on shaft

- It would be possible to mount the handles of the strimmer at points on the shaft where resonant vibration is much lower than the present location, but the benefit of this has not been tested.

13.1.4. Use of modal information (2): Stiffening of loop handle.

- It is possible to reduce vibration on the loop handle by up to 30% by adding stiffness.
- This would make it similar in response to the cow-horn handle.
- The design would need to be optimized to minimize conflict with ease of use.

13.2. MAINTENANCE

There are 5 points to be made with regard to the contribution of maintenance to the control of operator exposure to vibration:

- Engine tune should eliminate misfiring
- Engine tune should limit possibilities for over-speed operation
- Operators need to be observant for incorrect operation of automatic heads, which should be tested during routing servicing.
- Operators need to be observant for increased unbalance of heads, which should be tested during routing servicing
- Anti-vibration mounts should be renewed regularly.

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APPENDIX A: AUTO-BALANCERS – LITERATURE REVIEW

In order to reduce the effects of unbalance in rotating machines, the textbook advice is to add (or remove) balancing mass ***in the plane of the out-of-balance component*** (e.g. Macduff & Curreri 1958)[2]. For machines whose unbalance arises as a result of tolerances in manufacture, and which does not change greatly throughout their working life, this is treated by a once-for-all dynamic balancing of the rotating components. In the case of automotive wheels and tyres, whose unbalance may change gradually, but in any case changes whenever tyres are replaced, rapid balancing techniques have been developed and are in widespread use.

Thearle (1931) devised a "novel type of dynamic-balancing mechanism", with the object of simplifying the once-for-all production balancing of rotating machines. This was probably the first use of self-aligning balls to achieve balance (Figure A1). The purpose was to speed up the measurement of rotor unbalance, so that balancing could be applied to smaller and cheaper machines than was at the time the practice. The automatic balancer was part of the dynamic balancing test rig, and incorporated a clutch ("D" in the figure) which clamped the balls once balance had been achieved. From their position, a simple chart enabled the operator to add the necessary mass to the component being balanced.

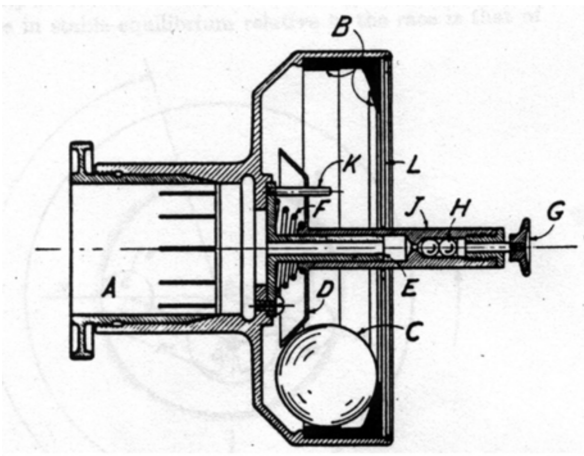


Figure A1 Section of automatic dynamic balancer (Thearle 1932)

The basic operation of Thearle's automatic balancer is described with reference to Figs A1, A2, as follows, and depends on the balancing rig using a soft suspension for the rotating components, endowing the unit with a low natural frequency. Figure A2 shows *"the familiar amplitude vs speed and phase angle vs speed for a system such as this . . .What is most important to us here is the relation between the phase angle ø and speed as shown by the upper curve in Fig A2. Since at constant speed, both the radial displacement p of the point S and the phase angle ø are constant with respect to time, the motion of the rotor is pure rotation about the axis z. When the speed of rotation is somewhat below the resonant speed, the phase angle ø is small, hence the displacement of the shaft is in the general direction of the unbalance w. An instantaneous end view of the rotor would appear as shown in Fig A3(a) where its motion is pure rotation about the axis z. The shaft may be said to whirl with its "heavy side" farthest from the axis of rotation. . . . when the rotor is running at a constant speed somewhat above that of resonance, the phase angle ø is slightly less than 180 deg. And the shaft end is displaced in a radial direction about opposite to that of the unbalance weight w as shown in Fig A3(b). The shaft may now be said to whirl with its "light side" "out", or farthest from the axis of rotation. This motion provides forces which serve to operate an automatic balancing device."* (Thearle, 1931)

[2] See Reference list for main text.

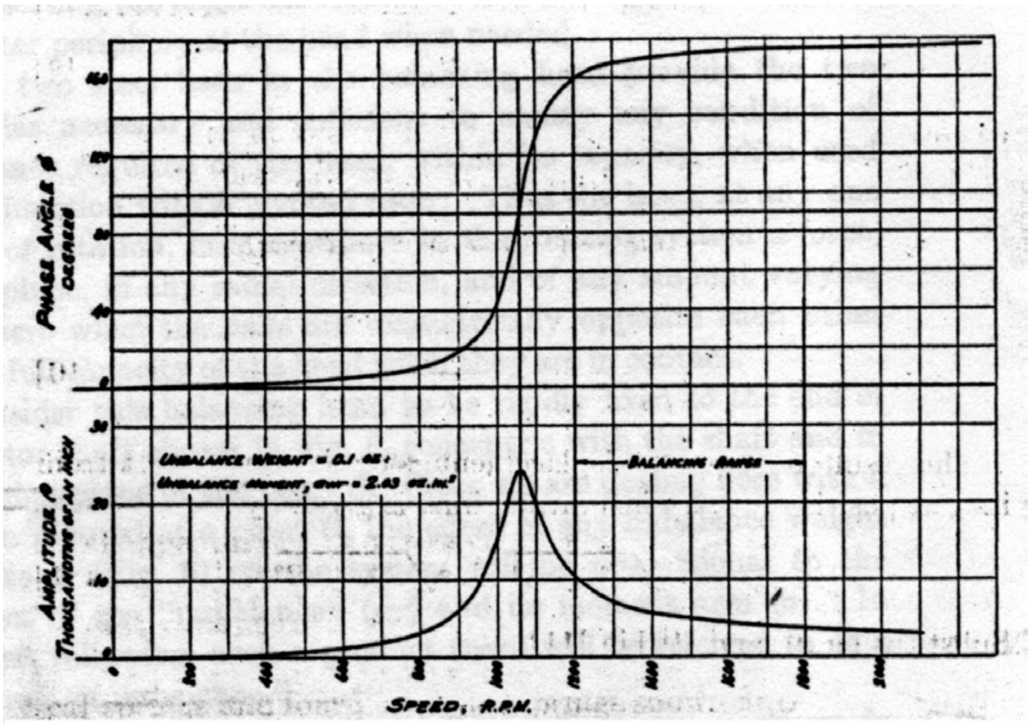


Figure A2: Amplitude and phase as functions of speed (Thearle, 1932)

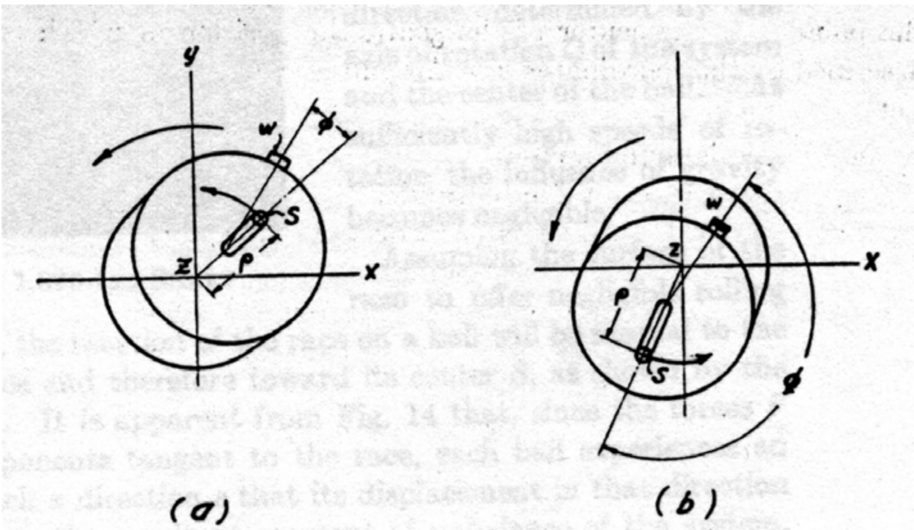


Figure A3:: Centre of rotation (a) low speed, (b) high speed (Thearle, 1932)

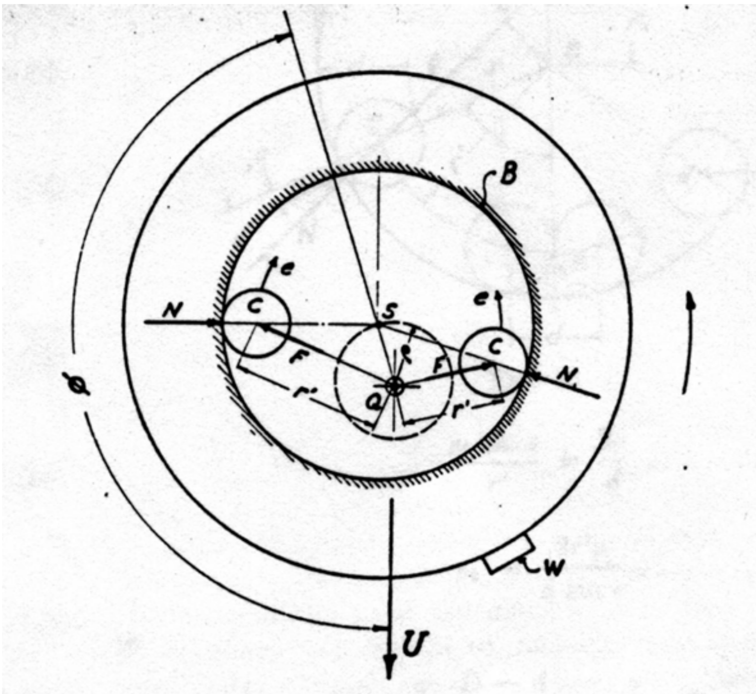


Figure A4: Showing balls before balance with center of rotation displaced towards "heavy side"

(Thearle, 1932)

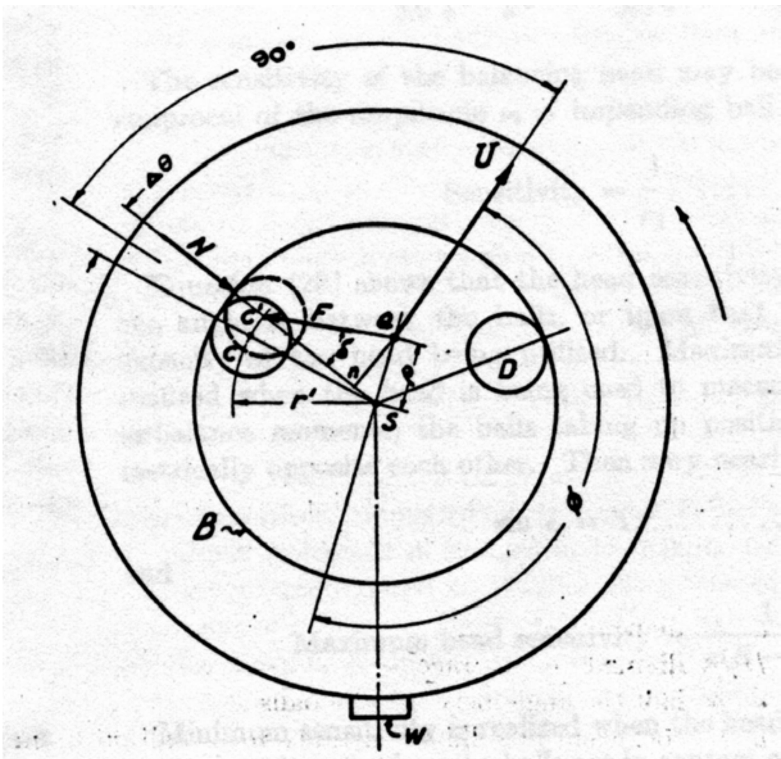


Figure A5: Showing position of balls after balance (Thearle, 1932)

Thearle's automatic balancer is realized in the form of a ball "race" (Fig A1) containing only two balls, that is attached concentrically to the rotor under test. When this is rotating above resonant speed, i.e. "light side out", the centrifugal forces acting on the balls is such as to force them towards the outer side, i.e. towards the light side and hence in a direction that causes them to reduce the resultant unbalance. Fig A4 shows the center of rotation "Q", before balancing occurs, displaced towards the inherent unbalance "w". Fig A5 shows the position of the balls when balance has been achieved. Thearle's 1931 paper is thereafter concerned with the masses and dimensions required to maximize the sensitivity of the device as a measuring instrument, and the type of indicator needed to direct the subsequent addition of balance weight.

In later years, Thearle applied this same auto-balancing principle to the problem caused by the unbalanced distribution of clothes in washing machines and spin-driers (Thearle 1950a, b, c). In the first two of this series of papers, he describes the Leblanc liquid balancer and mechanical *"self-balancers"* which included ring, ball and pendulum types. He concluded that only the ball type would provide a practical solution to the case of the washing machine. In the case of the washing machine, operations require different speeds, some of which are above the resonant speed (defined by the rotating mass and the tub suspension springs), and some below it. The ball auto-balancer would clearly operate to **increase** the unbalance at the lower speeds. Thearle therefore devised a system using a larger number of balls which would be constrained to a smaller diameter track at low speeds, completely filling this, and so adding nothing to the state of balance. The transition from low speed to high speed is achieved by means of a ramp (Fig A6). The slope of this was designed so that the centrifugal force would overcome gravity at just above the resonant speed. This design works only for machines with vertical shafts, common in the U.S.A. (top-opening). Front-loading machines common in Europe cannot use this type of balancer, and consequently reply on adding mass to the frame of the machine, often in the form of concrete, in an attempt to control the vibration. The third part of Thearle's 1950 publication is concerned with the design of a liquid balancer for the low speed operations of the washing machines, and is of no immediate interest for portable power tools.

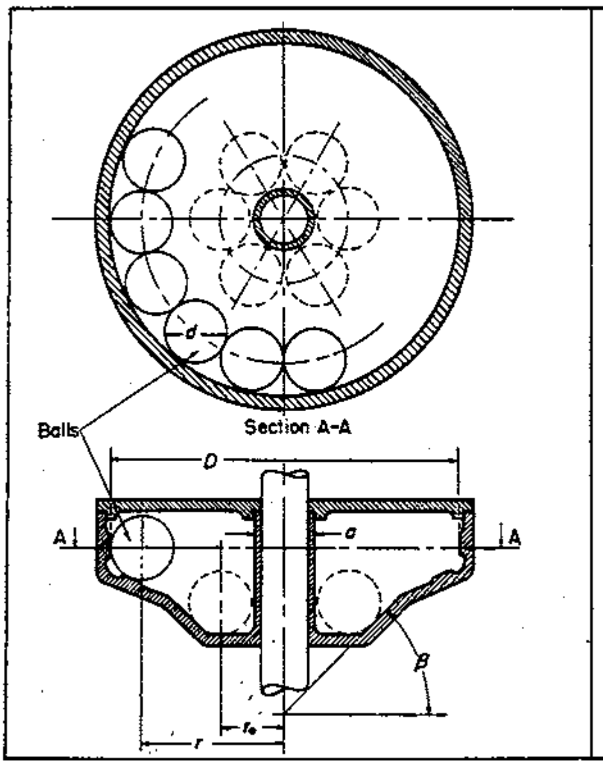


Figure A6: Six-ball balancer for vertical rotor showing low-speed (inactive) position (Thearle 1950b)

Thearle's ball-type auto-balancer re-appeared in the 1980's when some Japanese researchers applied it to the problem of hand vibration in portable grinders. In this case, it was recognized that the vibration was caused mainly by unbalance of the grinding wheel, and that this could change rapidly as the wheel was worn down. Auto-balancing might therefore provide an alternative or complementary solution to suspensions, as these on their own could compromise the ease of use of the machines. Miwa *et al* (1984) approached the problem experimentally, mounting a balancer closely behind the grinding wheel (Fig A7), and trying different numbers of balls to achieve the best result for each of two grinders (B and C in Fig A8).

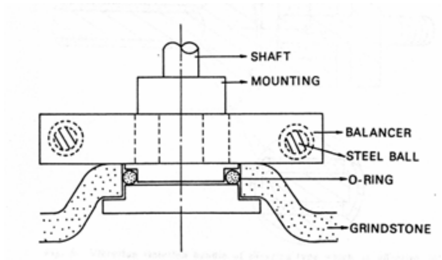


Figure A7: Auto-balancer for grinding machine (Miwa *et al*, 1984)

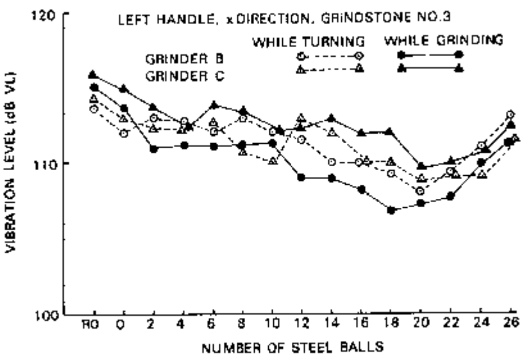


Figure A8: Experimental results for different numbers of balls (Miwa *et al*, 1984)

The results of the Japanese study were not greatly encouraging. However, Atlas Copco have since introduced a range of portable pneumatic grinders that use similar types of auto-balancers, developed by SKF-Autobalance Systems AB, which have been found to be very effective (e.g. Stayner 1996).

Considering the grinders, it may be understood that the "suspension system" is provided by the hand-arm system of the operator, so that the resonant frequency is much lower than the rotational frequency of the grinding wheel. It may also be that the machine is essentially a rigid body with regard to the motion between the operator's hands and the wheel, although this is perhaps questionable because of resonant modes of the casing of the machine (Stayner 1996). However, because the grinding wheel has been balanced successfully, it must be concluded that the conditions of whirling "light side out" are available.

It is an important part of the present study to discover whether such conditions exist at the cutting heads of strimmers, thus allowing automatic balancing of these according to the same principle.

APPENDIX B: MEASUREMENT AND ANALYSIS OF FREE-RUNNING VIBRATION

Free-running tests were used to measure handle vibration during several of the investigations that comprise this report. Similar measurement and analysis protocols were used for both the loop handle machine and for the one with cow-horn handles.

Accelerations were measured in three axes on both left and right handles simultaneously, using ICP accelerometers connected to a pair of Larson Davis HVM 100 Human Vibration Meters (Figure B1).

For one hand, a triaxial accelerometer (PCB model 356A24) was used. For the other a combination of three single-axis accelerometers (two PCB model 352C22 and one PCB model 353B16) was used. This arrangement was chosen so that the connecting cables could all be run along the hand-grip. All the accelerometers had a nominal sensitivity of 10mV/g.

The accelerometers were mounted on small aluminium blocks for attachment to the hand-grips and to provide some local support to the cables.

The accelerometer units were attached to the hand-grips under the palm position, by cable ties, and covered with protective fibre-glass moulds (Figure B2).

Figure B1 Loop handle machine showing Measuring equipment

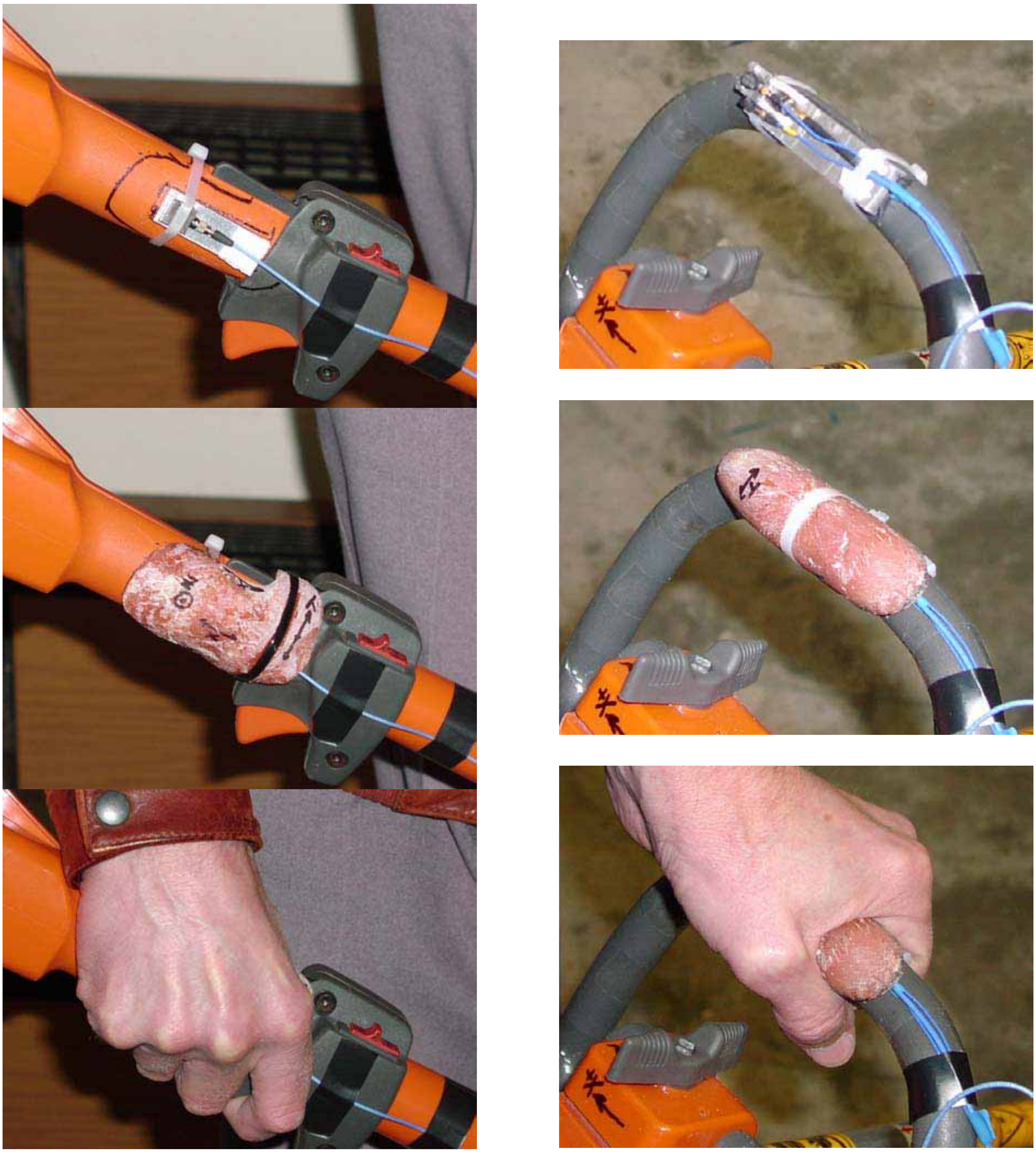
Direct readings of frequency-weighted acceleration were taken from the Larson Davis meters. But in order to provide frequency analysis, the analogue outputs of these meters were connected to the acquisition side of a PC-based HvLab system (ISVR).

The complete measurement chain, including both Larson Davis and HvLab rms acceleration magnitudes, was calibrated by back-to-back tests on an electro-dynamic shaker with a traceable accelerometer system as reference.

Each measurement consisted of a series of 10 second samples, acquired into HvLab at 3 kHz. Each sample provided overall weighted rms levels through both the Larson Davis meters and HvLab. The HvLab protocol included the following stages:

- Frequency analysis and display of PSD graphs for selection of ranges associated with engine and head fundamental frequency.
- Extraction of engine and head fundamental components. A frequency band of about 30 Hz around each of the two PSD peaks was put into a separate file for further calculation. This is only practicable for steady state operation of machines that exhibit narrow peaks in their vibration characteristics (e.g. Figure 5, in Chapter 3).
- Calculation of weighted mean square values for engine and head fundamental components from the extracted sections of the power spectra.

These components were calculated for each axis separately. Spreadsheets were used to combine axes and so obtain RSS rms values for presentation in the various comparative charts.



(a) (b)

Figure B2: Transducer mountings, (a) shaft grip, (b) loop handle grip
Top: Transducer block; Middle: plastic cover; Bottom: hand grip

For each set of comparative measurements, only one operator was used. However, it was not the same operator for all sets of measurements, and so comparisons can be made only between contemporaneous measurements

Figure B3: Artificial discs, "balanced" and with 5gm unbalance

In order to make a first estimate of the amount of inherent unbalance in the cutting heads, a pair of additional discs was made. One of these had notionally no unbalance. The other was deliberately unbalanced by 5g at 50mm. It was of a similar pattern to those used in tests of grinders (Stayner 1996), and the amount of unbalance was measured in the same way as the grinding discs. The steel bushes made a close fit on the splines of the head drive spindle.

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