PHILOSOPHICAL TRANSACTIONS OF THE ROYAL SOCIETY A
MATHEMATICAL, PHYSICAL & ENGINEERING SCIENCES

Noise Control on Textile Machinery

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IV. NOISE CONTROL OF FACTORY PLANT

Noise control on textile machinery

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Introduction

In the textile industry as a whole the pattern of machine layout and operation has been changed little over the years. Banks of machines, housed in large factory areas, are crowded together to preserve floor space and optimize production in both the natural and man-made fibres industries. Noise in the mills and factories is increasing, and one of the main reasons is that in the face of keen competition and already on a 24 h basis, machines are being operated at speeds higher than ever before to increase production rates. This is possible to a greater extent with the man-made fibres; continuous threads can be produced and wound more quickly because of their inherent strength and other advantageous properties compared with natural fibres. As a direct result of the speed increase, out of balance forces and vibration increase and more energy is released as sound.

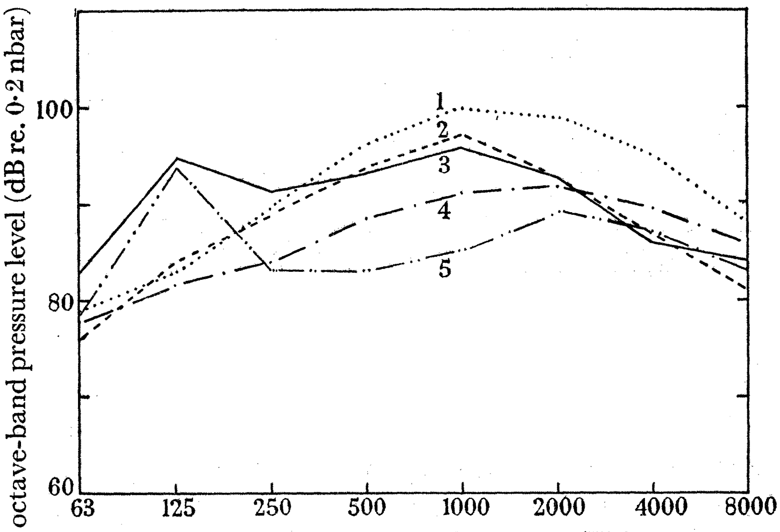

Figure 1. Noise spectra of five processes in the synthetic fibres industry. 1, staple fibre cutter; 2, spinning; 3, drawtwisting; 4, weaving; 5, uptwisting.

There may be many machines of one kind in a particular area with the result that each process has its own noise spectrum. In figure 1 typical average spectra for five processes in the synthetic fibres industry indicate how the frequency content and levels change from one area to another. The noise levels in each process can vary considerably depending on the machine speeds, thus giving a spread of levels for a particular process. Generally, some textile areas are inherently quieter than others.

In almost all the areas in the industry the machine operators are exposed to the noise levels throughout the duration of their work period of about 8 h per day. Much work has already been done on the effect of long-term hearing exposure to textile machine noise and other environments with similar noise spectra (Taylor, Pearson, Mair & Burns 1965; Burns, Hinchcliffe & Littler 1964; Nixon & Glorig 1961; Atherley 1964). Spurred on by the Wilson Committee (1963) Report attention has been focused on the problem of the reduction of industrial noise in many textile environments in this country. Over-all noise levels throughout the industry where high-speed machinery is operating are in the region of 90 to 105 dB (re. 0·2 nbar) with the main energy peaking around 1000 to 2000 Hz. Recommended deafness risk criteria (Burns 1965; Glorig, Ward & Nixon 1961) are sometimes exceeded in several octave bands in many processes, supplying the need for a noise-reduction programme.

This paper discusses noise control in several processes and in detail the noise associated with the drawtwisting of nylon or Terylene, its sources and their suppression. Much of the work has direct application to other textile machinery and to high-speed rotating machinery in general.

In the manufacture of continuous filament synthetic yarns there are several basic mechanical processes involved—spinning, drawing and winding. In the spinning process the thread is made by extruding the molten polymer through small disks, the spray cooling into fine separate threads and the one, two or many threads are wound together loosely into the normal yarn. At this stage the chemical chains are randomly oriented and the undrawn yarn has relatively little strength. To orient the chains in the direction of the yarn and hence give nylon its well-known strength, the yarn is stretched and wound on to a bobbin in a drawtwist machine; the loose thread, unwound from the 'cheese' is fed by slowly moving feedrolls to much faster moving drawrolls. The speed differential stretches the threads to several times their original length. A degree of twist is imparted to the yarn as the drawn yarn is wound on to a bobbin rotating about a vertical axis, a winding traverse mechanism being used to guide the threadline evenly along the length of the bobbin.

Drawtwist noise

In the drawtwist areas the machines to not necessarily produce the same noise levels. Heavy yarns are produced at slower speeds than light yarns and the noise levels vary according to machine speeds. Even machines operating in the same process under identical conditions can produce noise levels that vary considerably. The spectral shape remains the same but the levels can be different, particularly for discrete frequencies. As speeds increase the excitation forces increase and small differences in response of machine elements to the vibration excitation are accentuated and the resulting noise levels vary from machine to machine. Figure 2 shows the spread from machine to machine of the maximum noise level to the minimum noise level for machines of the same type running under similar conditions. A representative idea of the spread of levels of machines running under different production conditions, obtained by running machines one at a time during a factory shutdown period, can be seen in table 1. The spread of levels from 88 to 103 dB (A) indicates that the loudness or subjective variation is very great. The table also shows that much of the noise depends on the type of spindle used.

The process of drawtwisting demands that the operator should be close to his machines throughout the work period. He is in the near field radiation of the machines for much of the time. Even when he is at a rest point he is within 10 to 15 ft. from a bank of machines. Since he has to observe the winding process and remove broken threadlines, it would be

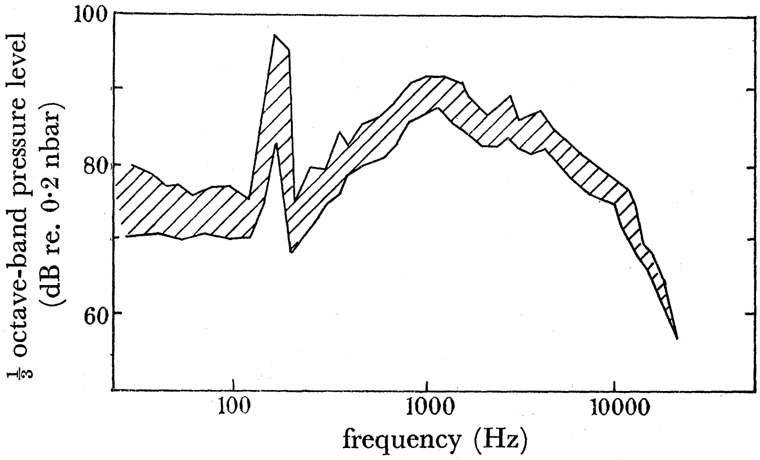

Figure 2. Intermachine variation.

Table 1. Variation of over-all noise levels of drawtwist machines running under different conditions

Each noise level is associated with a different speed, machine configuration, or process phase.

machine type	spindle type	noise level (dB(A))
I	A	102
		97
		103
		97
		101·5
		97·5
		100·5
		97
		94
I	B	93·5
		94·5
		95·5
		92
		90·5
I	C	88·5
II	B	92·5
		91·5
		88

extremely difficult to design suitable acoustic enclosures for a drawtwist machine in its present form. Ear protection could provide an answer but there are objections from hygiene, safety and comfort considerations (Rice & Coles 1966). The solution then is the reduction of the noise at source, which, although often difficult to achieve, is the most desirable solution. A necessary prelude to noise reduction at source is a detailed knowledge of the source mechanisms, transmission paths and noise-radiating surfaces.

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A schematic diagram of a drawtwist machine is shown in figure 3. The main parts are the spindle and bobbin assembly, the drawbox—a long gearbox driving the drawrolls, and the main headstock drive unit. The narrowband noise spectrum of a drawtwist machine in figure 4 can be attributed to noise mechanisms from these three main parts. The loudest of these is the spindle and bobbin assembly.

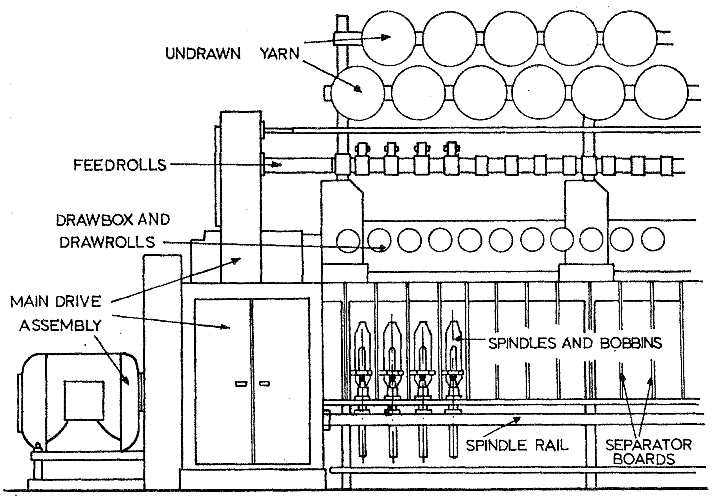

FIGURE 3. Schematic diagram of drawtwist machine.

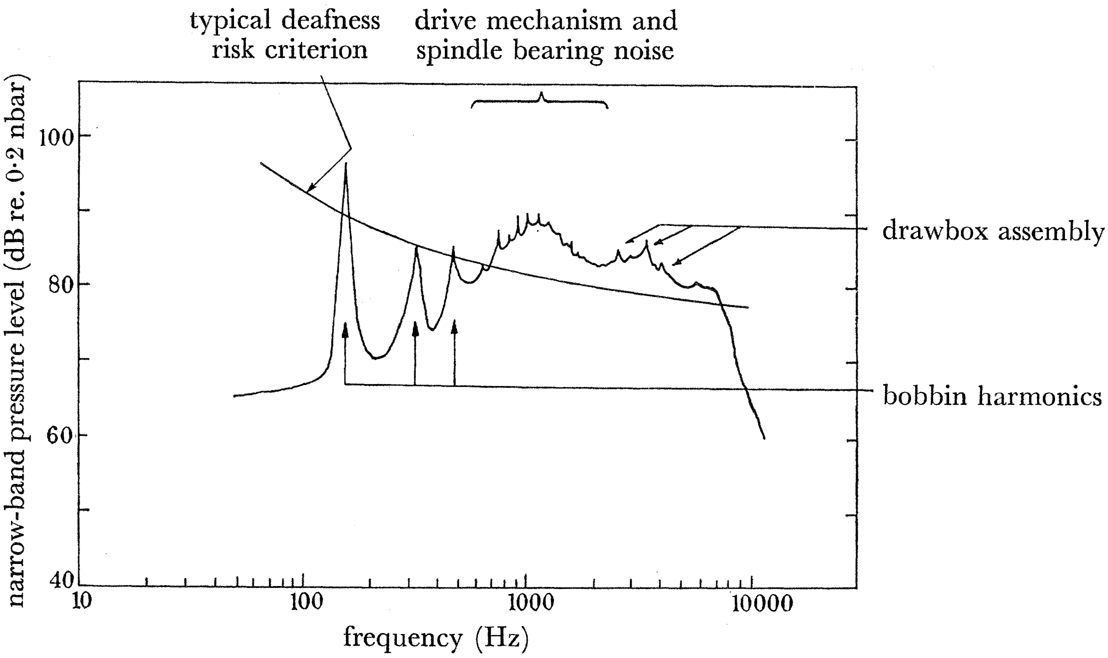

FIGURE 4. Narrow band spectrum of drawtwist machine.

As the operator walks around a drawtwist machine the subjective emphasis changes as he hears the different sources. In the centre of the machine he hears the drawrolls and spindles, nearer the headstock he hears also the noise of the various drives. Throughout the area he is conscious of a low-frequency 'beating' at the spindle fundamental frequency. Floor vibration also adds to the noise. Directionality patterns of a drawtwist machine are shown as octave band noise contours in figures 5 and 6. These were obtained by running

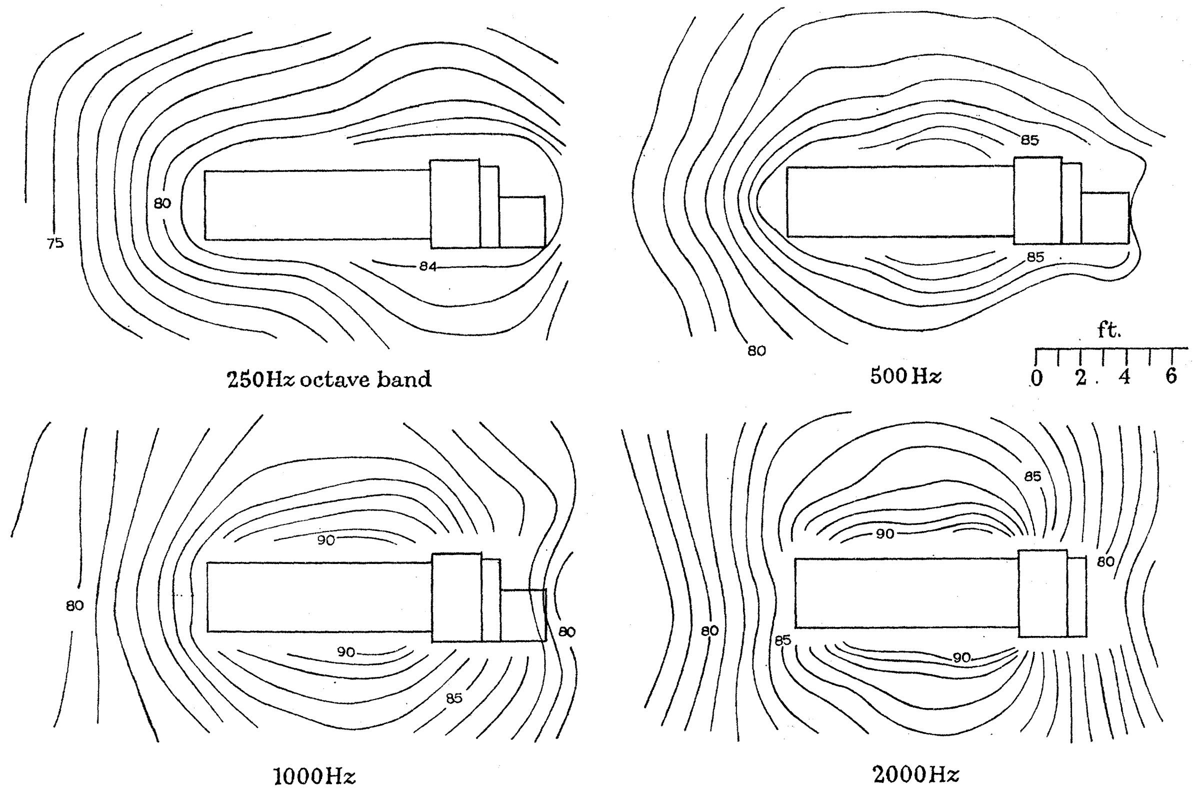

FIGURE 5. Noise contours (dB re. 0·2 nbar) of drawtwist machine. Spindle speed 10 500 rev/min; drawroll speed 4150 rev/min.

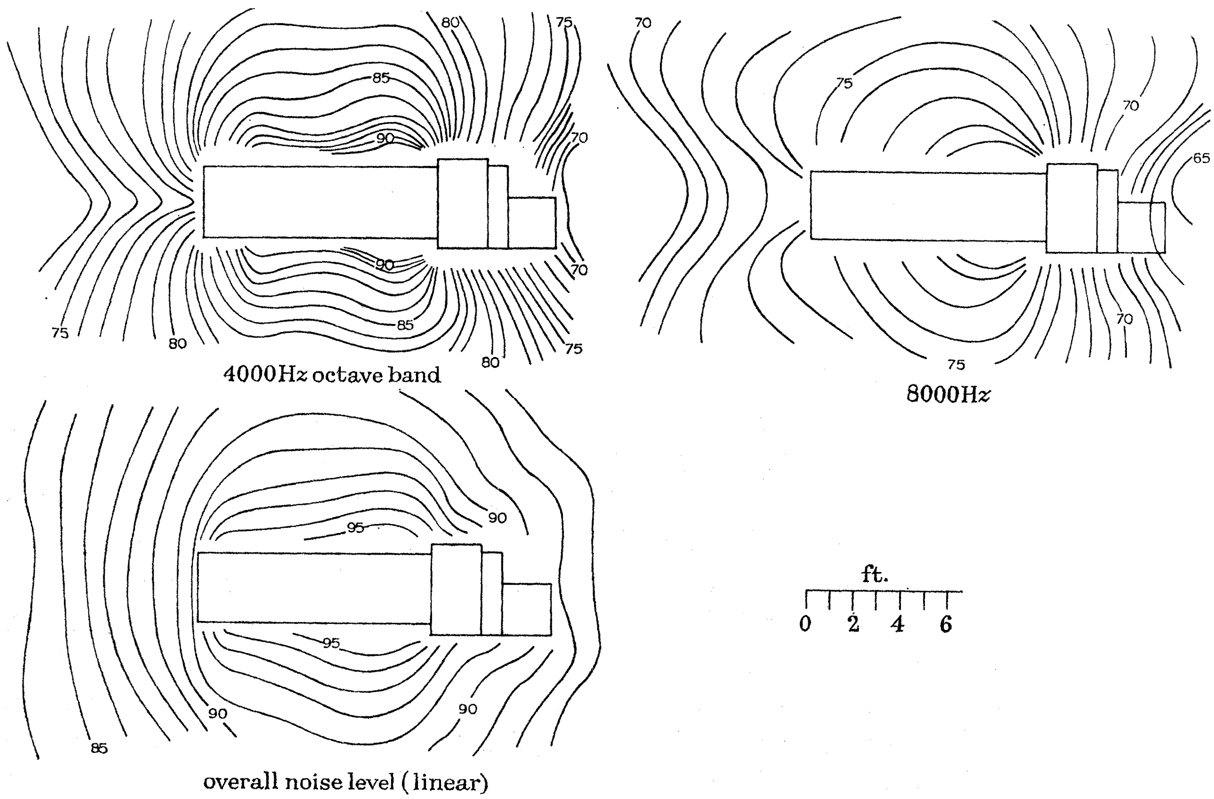

FIGURE 6. Noise contours (dB re. 0·2 nbar) of drawtwist machine. Spindle speed 10 500 rev/min; drawroll speed 4150 rev/min.

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a drawtwist machine in a semi-anechoic room that was used for many of the noise measurements throughout the programme. The noise contours are shown for a machine without bobbins and the measuring datum is at operator head height. Noise levels in the lower frequency octave bands rise considerably when full bobbins are used and the spindle harmonic noise is much in evidence.

Spindle and bobbin noise

In order to understand the mechanisms of noise production in textile spindles a knowledge of the mechanical features of the spindles is required. This is best described in conjunction with figure 7.

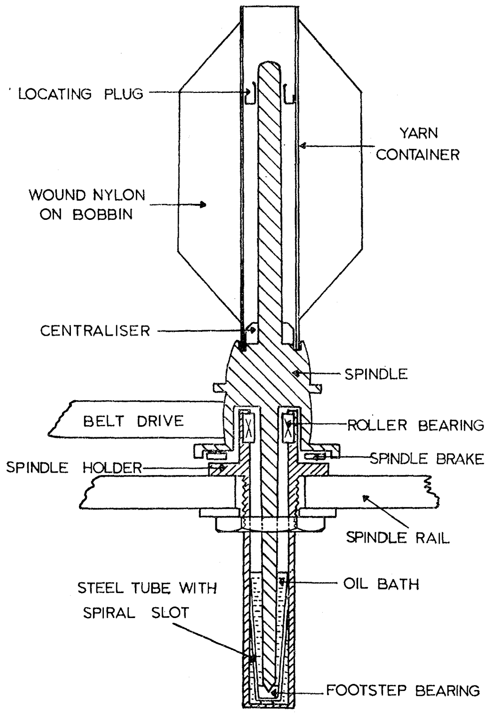

Figure 7. Schematic diagram of spindle.

A rigid one-piece balanced spindle runs in two bearings mounted in the spindle holder. The lower or footstep bearing is a plain bearing that takes the weight of the spindle and bobbin and is mounted with a degree of horizontal flexibility provided by a spiral slot cut in the supporting spring steel tube. The top bearing is a roller bearing fixed rigidly to the spindle bolster and the ground spindle blade is used as the inner bearing surface. Splash lubrication is provided by an oil bath in the spindle bolster and damping is supplied by a series of concentric sliding steel cylinders in the oil bath around the footstep bearing. The spindle holder is bolted rigidly to the supporting cast steel rail, running the length of the machine. Spindles are driven from a central drive-shaft running the length of the machine, by short friction-drive canvas or Terylene flat belts, two spindles on each side of the machine being driven by one belt. Brakes are incorporated into each spindle so that it may be stopped independently of the others. During this operation the friction drive belt slips on the spindle whorl. The bobbin or yarn container is a metal tube that fits over the top of the spindle and is located by driving dogs and guide centralizers. When this spindle is run at speed there are two-main noise sources: (*a*) the noise originating from the out-of-round of the bobbin; (*b*) the noise originating from the bearings.

Comparison of spindle-noise spectra of bare spindles and spindles with full bobbins indicate that most of the bobbin noise is discrete frequency noise corresponding to the rotational speed of the spindle and its harmonics. Bearing noise appears more broadband in character and predominates over the spindle harmonics at frequencies higher than 700 to 800 Hz and will be discussed later.

Discrete frequency noise

If the empty bobbins were initially running concentrically and the yarn wound concentrically, there would be very little mechanisms to generate discrete frequency noise from the bobbin. If a degree of eccentricity were introduced in the running of the bobbin, then noise would be radiated from the sides of the bobbin. This appears to be the main source of the discrete frequency noise at the spindle fundamental rotational speed.

An eccentric bobbin can be likened to a short cylinder rotating about an axis parallel to its own axis, and offset from it by a distance or eccentricity b. As the cylinder rotates the eccentricity will displace a volume of air. This displacement may be Fourier analysed into a series of pressure components so that noise is radiated at the fundamental rotational speed of the bobbin and its harmonics. The radial motion of an element of air at the surface is assumed to have the same radial motion as the cylinder surface itself and experiences a displacement of $\pm b$ per revolution. This implies that boundary layer and turbulence effects are neglected, an assumption that is valid at normal spindle speeds.

In a previous paper on spindle noise (Crawford 1967) expressions were derived for the discrete frequency acoustic intensity and sound power generated by eccentric rotating bobbins on the basis of both a rotating dipole model and also a ring source.

The intensity may be written

$$\left.\begin{aligned} &\text{For a rotating dipole} && I = \frac{\rho \Omega^6 a^4 b^2 l^2}{2\pi^2 s^2 c^3} \sin^2 \psi \quad (a), \\ &\text{or for a ring source} && I_m = \frac{m^4 \rho \Omega^4 a^2 b^2 l^2}{8 c s^2} J_m^2 \left(\frac{m\Omega a}{c} \sin \psi\right), \quad (b) \end{aligned}\right\} \tag{1}$$

the notation being:

- ρ density of air,
- Ω bobbin rotational speed,
- b bobbin eccentricity,
- l bobbin length,
- a bobbin radius,
- c speed of sound,
- s distance from field point to bobbin,
- m order of harmonic,
- ψ angle subtended at the centre of the bobbin by the axis of the bobbin and field point,
- J_m Bessel function of first kind of order m.

Since $m\Omega a/c \ll 1$ in bobbin applications $ka \ll 1$, we get for small values of m and if $\psi = 90°$ (at right angles to the bobbin axis)

$$J_m(z) \doteqdot \frac{z^m}{2^m m!}$$

substituting in equation (1 *b*) then

$$I_m = \frac{\rho b^2 l^2 c^3}{s^2} \frac{M^{2(m+2)}}{a^2} \frac{m^{(2m+2)}}{m!m!2^{2m+3}}, \tag{2}$$

where $M = \Omega a/c$ is the peripheral speed Mach number. For $m = 1$ this reduces to (1 *a*).

Immediately the dependence of the acoustic intensity of the rotational speed harmonics on the peripheral speed Mach number is seen. For the first, second and third harmonics, i.e. for $m = 1, 2, 3$ the power indices are 6, 8 and 10. Dokuchaev (1967) obtains the same power indices for the radiation of sound waves of a body moving in a circle. Computation of the ratio of intensities of the harmonics to the intensity of the fundamental, $m = 1$, show that almost all the energy is radiated at the fundamental frequency at low speeds (table 2). As the Mach number rises the intensities of the harmonics increase relative to the fundamental and thus at higher speeds subjectively the first few harmonics are important.

TABLE 2. VARIATION OF THE RATIOS OF THE DISCRETE FREQUENCY INTENSITIES ($m = 1, 2, 3$) WITH SURFACE SPEED, CALCULATED FROM EQUATION (2)

peripheral speed Mach no.	I_1/I_2 (dB)	I_1/I_3 (dB)	approx. rotational bobbin speed (rev/min)
0·05	14·0	31·9	2600
0·10	8·0	19·9	5200
0·15	4·4	12·8	8000
0·20	1·9	7·8	10400
0·30	−1·6	0·8	16000

To supplement the theory, the discrete frequency noise radiated by eccentric cylinders rotating at speeds up to 16000 rev/min was experimentally measured and is shown in figures 8 and 9. These cylinders were balanced so that no mass out-of-balance was present, only a geometric out-of-round. The indices for m are in good agreement with theory and the intensity levels for the fundamental $m = 1$ can be predicted to within 1 to 3 dB. There is a larger discrepancy for the intensities of the higher harmonics, the theory predicting intensities that are too high by several decibels.

Eccentricity of bobbins arises from several factors: the locating guide plugs in the yarn container are often misalined. Yarn containers are mass produced and replaced in vast numbers so that close manufacturing tolerances would, therefore, be expensive. Generally, containers are not fitted tightly on the spindles for production reasons and this play, together with the misalined centralizers, allows the bobbin to move on the spindle blade and rotate eccentrically. This eccentricity is highly undesirable and is the cause of the sound produced at the fundamental rotational speed of the bobbin.

Discrete frequency spindle noise reduction has been effected by the use of flexible bearing mounts so that mass centring of the spindle and bobbin combination takes place. The footstep bearing already has a degree of horizontal flexibility as previously described, and by mounting the top roller bearing in a resilient bush of the correct stiffness then at speeds above the critical speeds the mass centre of the rotating system moves on to the axis of rotation, which is also coincident with the geometric axis of the spindle and bobbin combination (K. Shotbolt, unpublished work). On the assumption that the yarn is evenly wound on the initially eccentric container then the mass out-of-balance is in phase with the geometric out-of-round. Ideally, centralization takes place, the bobbin runs concentrically, and discrete tones are not longer heard. However, there is always a residual out-of-round due probably to the yarn-winding process, spindle tilting caused by belt tension, play in the bearings or of the bobbin on the spindle blade so that the discrete tones are still generated.

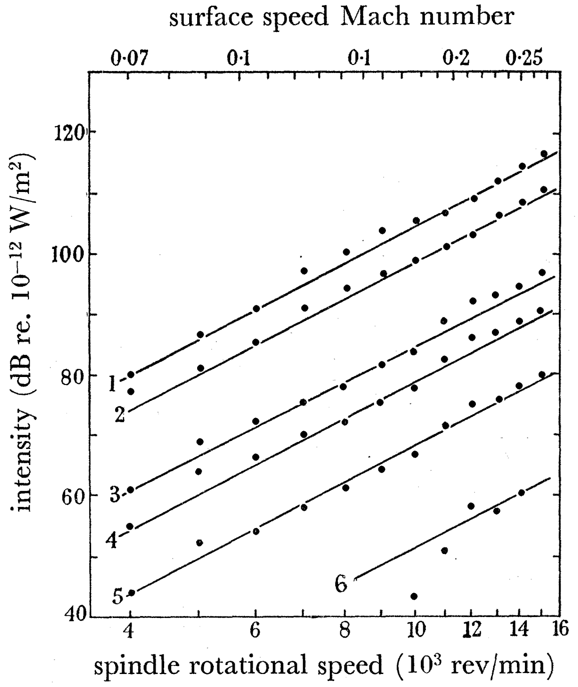

FIGURE 8

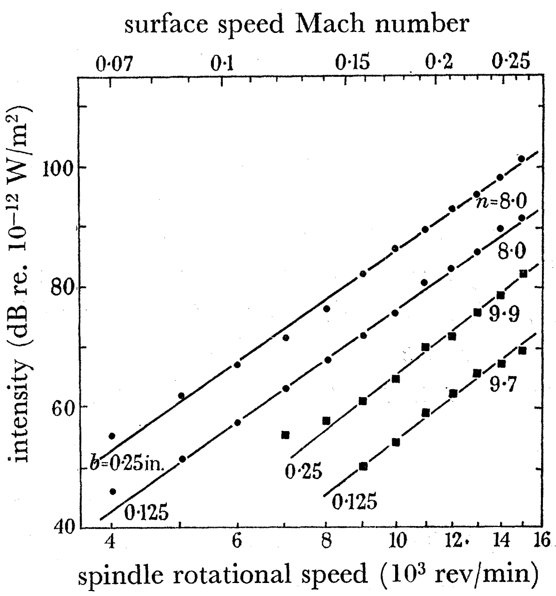

FIGURE 9

FIGURE 8. Measured and calculated discrete noise $m = 1$. l (in.): 8·5; a (in.): (1 to 5) 2·25, (6) 0·87; b (in.): (1) 0·25, (2) 0·125, (3) 0·020, (4) 0·010, (5, 6) 0·003; s (ft.): (1, 2) 4, (3 to 6) 3·28; ψ: 90°.

FIGURE 9. Measured discrete noise. $m = 2$ (●), $m = 3$ (■). l (in.): 8·5; a (in.): 2·25; s (ft.): 4; ψ: 90°.

Tests carried out on a laboratory scale and on full-size production machines using various types of flexible spindle mountings, showed that in practice there was a net reduction in eccentricity. A 10 dB reduction of the spindle fundamental tone was obtained with the top bearing flexibly mounted relative to the bearing holder and 6 to 7 dB reduction when the complete bearing holder was flexibly mounted relative to the spindle rail. This can be seen at the rotational speed frequency in the spectrum in figure 12.

Throughout the drawtwist areas the low-frequency noise generated at the spindle rotational speed waxes and wanes randomly. The cloth belt spindle drive is a non-positive type of drive and particularly at high spindle speeds, the bobbins do not rotate at the same speed, because the tension differential between the tight and slack sides of the belt causes slip on the bobbin whorl. When viewed by a stroboscope, the bobbins can be seen to rotate relative to one another at speeds that depend mainly on the belt tension. As a result the phenomenon of 'beats' is observed, the rate of beating depending on the relative rotation of the bobbins. Figure 10 shows the time history of the discrete frequency noise of the fundamental. Since there are many bobbins involved, with randomly varying phase, the waxing and waning appears as a random phenomenum with noise levels varying by up to 30 dB.

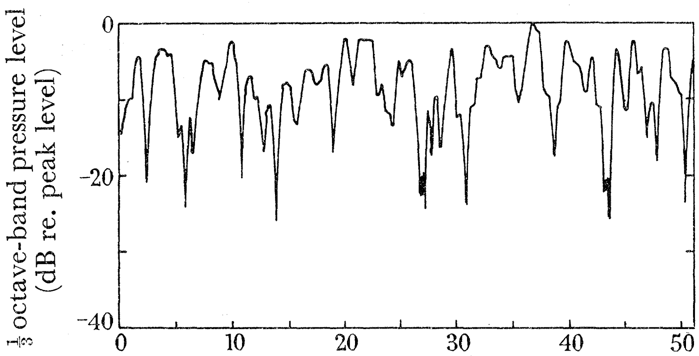

FIGURE 10. Interaction of bobbin sound fields. Analyser centred on 160 Hz.

Associated with the rotation of the bobbins is the vibration excitation of the spindle rail caused by the rotating out-of-balance vectors. When they are in phase, torsional vibration of the rail occurs which leads to larger bobbin vibration amplitudes and hence more noise. Again because of the mass centring, with the flexibly mounted bearings the out-of-balance forces are reduced and the excitation of the spindle rail is less.

BROADBAND NOISE

There are several sources of broadband noise in the rotating bobbin problem. Vortex shedding and air turbulence associated with surface roughness and shape of the bobbin occur to some extent, particularly at very high spindle speeds, but by far the most important is the bearing noise, mainly from the top roller bearing. The plain footstep bearing in the oil bath is relatively quiet. Calculations of the vibrational frequencies and harmonics produced by the geometrical shape and size of the top bearing at a constant spindle speed indicate that over the audio spectrum the energy will be distributed at many discrete frequencies, very close together. Dimensional variation of the bearing components due to manufacturing tolerances and clearances, and distortion or rotational speed variations due to belt slip, will cause changes in these frequencies and when considered together with the vibrations produced by impact, rubbing and slipping of the bearing parts, the spectrum appears broadband in character with the main energy peaking around 1000 to 3000 Hz. The roller-bearing vibrations are transmitted to the spindle holder and spindle rail so that noise is radiated from these parts.

The transmission of bearing vibrations to the radiating surfaces, and therefore the noise radiation from them, is reduced by the same flexible mounting that is used to allow the spindle to mass centre. There is considerable isolation of the high-frequency bearing vibrations from the radiating surfaces as is shown in figure 11. For complete machines (figure 12), the noise reductions for the different spindle mountings are shown. The middle frequency reductions are less than laboratory tests predict, due to the masking noise of belt guide and tensioning pulleys, and main drive shaft noise.

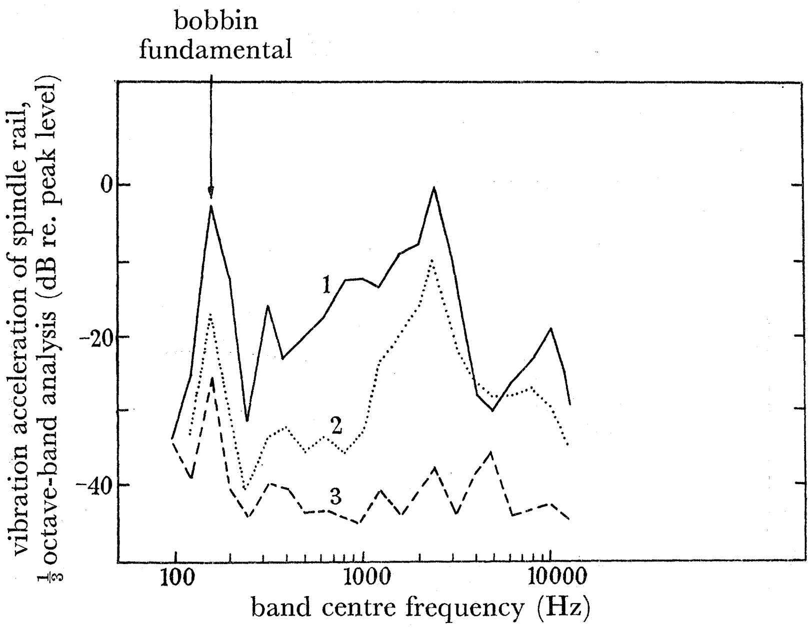

FIGURE 11. Vibrations of spindle rail. Spindle speed 10 000 rev/min. 1, Rigid mounting of bearing and holder on beam; 2, rigid mounting of bearing with isolated holder; 3, rigid mounting of holder with isolated bearing. Over-all levels (dB re. peak): 1, +4·5; 2, −6·5; 3, −23·0.

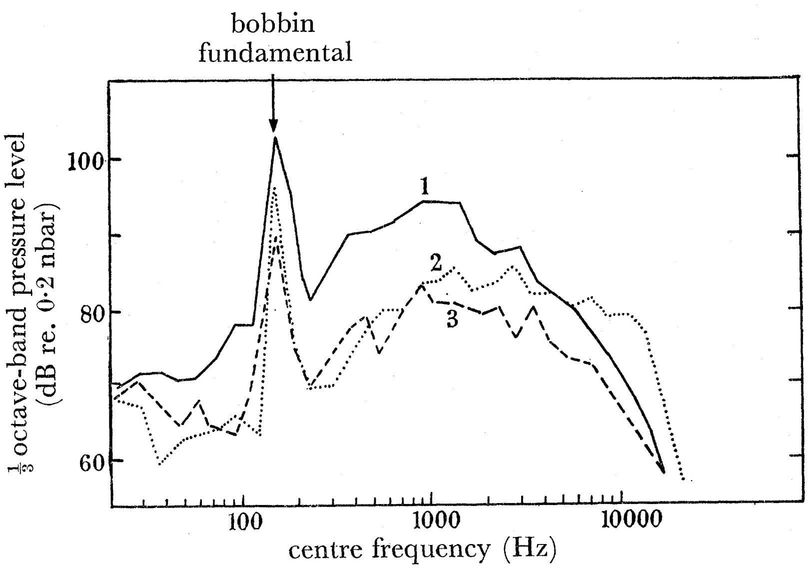

FIGURE 12. Noise spectra of real drawtwist machines. 1, Rigid mounting of bearing and holder on beam; 2, rigid mounting of bearing with isolated holder; 3, rigid mounting of holder with isolated bearing.

Experiments have shown that needle roller bearings were quieter by 2 or 3 dB than comparable roller bearings in spindle applications. Plain bearings of various materials and design, oil lubricated have been examined (G. Elliot, unpublished work) and were found to be quieter *per se* than needle roller bearings provided they are also flexibly mounted to reduce the out-of-balance forces. Their acoustic advantages were quickly lost by the masking noise of the bobbin harmonics and belt noise. Their mechanical disadvantages can be great at high speeds. Once the top needle roller bearing was flexibly mounted and the bearing noise reduced, the noise of the friction drive belts became significant at a speed of about 11 000 rev/min. At higher speeds up to 18 000 to 20 000 rev/min range, which incidentally were possible only with the flexibly mounted spindles, the noise emphasis shifted to the bobbin harmonics and the belt noise.

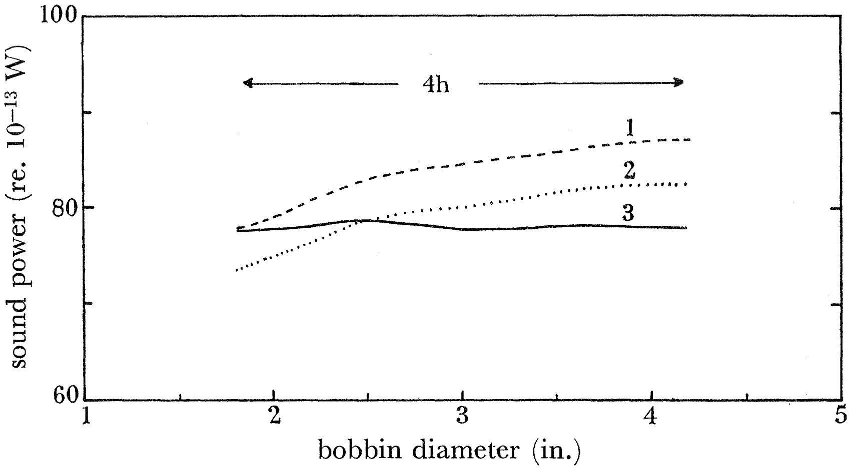

FIGURE 13. Programmed spindle speed noise. 1, 11440 rev/min; 2, 9600 rev/min; 3, slowing from 11400 to 8100 rev/min.

The residual out-of-round of the bobbin may take the form of a cylinder with a single or many lobes on the surface. If there are many lobes, then the higher harmonics of the bobbin noise increase in intensity and can become greater than the fundamental. Very little out-of-roundness is required to produce noise. For example, from equations (1) and figure 8 for typical bobbin dimensions, an eccentricity of only 0·010 in at 15 000 rev/min generates a tone of about 90 dB. The inherent out-of-roundness of bobbins clearly is indicative of a limit in the noise reduction at source for high-speed spindles and bobbins.

It is obvious from equations (1) that a reduction in the rotational speed or the peripheral speed would be advantageous. Modern drawtwist machines incorporate a programmed spindle speed whereby the spindle speed is high at the beginning of the wind and as the bobbin builds up, increasing in radius, the rotational speed is reduced, keeping a fairly constant peripheral speed and thread-line tension. The older machines run at a constant spindle speed so that the discrete frequency noise increases as the wind progresses according to equations (1). There is a noise reduction of approximately 9 dB of the fundamental discrete tone when the spindle slows down to about 70 % of the initial spindle speed compared to a constant spindle speed, as shown in figure 13. Since intensity is proportional to $M^{2(m+2)}/a^2$ it is better to wind large diameter bobbins running at slower rotational speeds. Subjectively, this also keeps the frequencies of the harmonics below the most sensitive hearing region of the ear.

The belt noise increases at high speeds due to the increased vibration of the guide and tensioning pulleys. The passage of the belt joint over these components and around the spindle whorl is easily heard. The use of endless belts is a great advantage from a noise and vibration viewpoint.

Drawbox noise

The second major noise producer in a drawtwist machine, is the drawbox. The drawbox (figures 1 and 14) consists basically of a long layshaft running the 30 ft. length of the machine, in 4 ft. sections. At right angles to the layshaft there are short cross-drawroll shafts driven in pairs by means of spiral helical gears. Thus, every 7 in. along the drawbox there is a cluster of three spiral helical gears. The drawrolls, approximately 1 ft. in circumference, are bolted to the end of the short drawroll shafts. The drawrolls at the

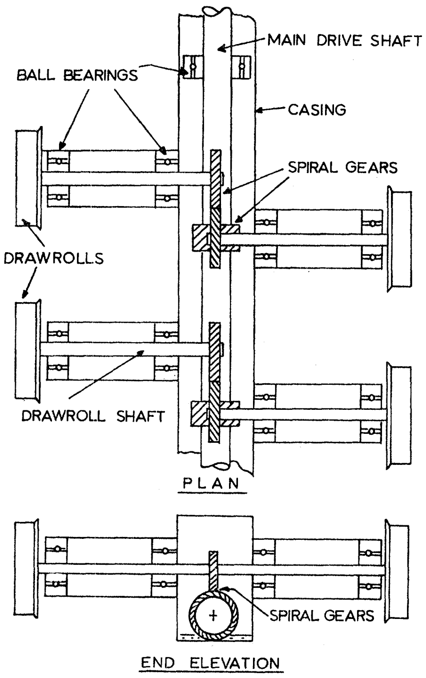

Figure 14. Schematic drawbox.

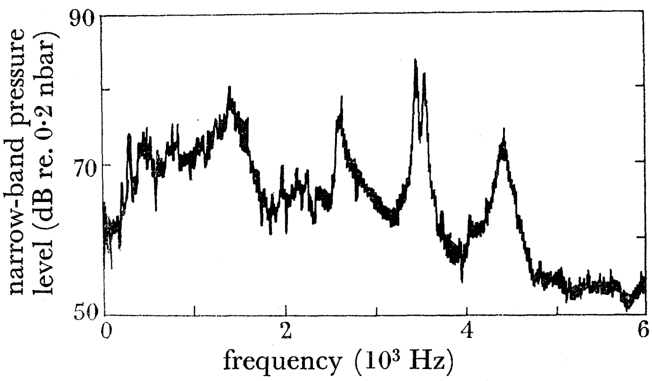

Figure 15. Drawbox spectrum. Measured 2 ft. from the centre of the drawroll, (5000 rev/min).

present rate of production run at 3000 to 4000 rev/min. The gears are a steel-brass-steel train and are splash lubricated from an oil reservoir in the bottom of the drawbox. Apart from high starting torques, the loading on the gears is light. The noise spectrum from a standard drawbox running at 5000 rev/min is shown in figure 15. Identification of the various peaks in the spectrum was done by examining the spectra of a single drawroll driven in various configurations and at different speeds. Vibration excitation of the drawroll shaft assembly by an electro-magnetic shaker confirmed that the major noise peak at 3500 Hz was due to a longitudinal natural frequency of the drawroll shaft assembly excited by the meshing forces of the gear teeth. The spiral helical gears have an axial component of thrust as well as a torsional component. Other peaks at 1500, 2600 and 4500 Hz, correspond to bending modes of the shafts, again excited by the gear meshing, presumably due to alinement and distortion problems. These major frequency peaks have no bearing on the shaft rotational speed. Torsional vibration modes did not appear on the noise spectrum. The broadband noise is due to the metallic noise of the tooth impacts. If the gears were perfect then rolling contact would take place with a minimum of noise. Owing to manufacturing tolerances, clearances, and perhaps more to tooth wear and hence poor tooth profile, both rubbing and sliding take place with a significant amount of noise generation. The bearings in this type of drawbox did not generate much noise at the speeds involved. Subjectively, the drawbox noise spectrum is one of medium to high frequency with a major 'ringing' noise predominating at 3500 Hz. The face of the draw-roll moves in and out like a loudspeaker diaphragm and radiates the ringing noise in the 4000 Hz octave band with a directionality pattern that is maximum at the operator's head.

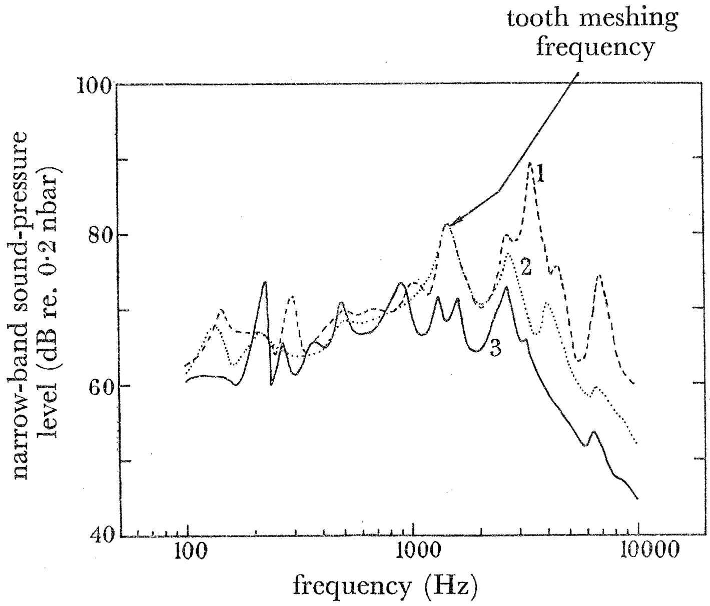

FIGURE 16. Noise spectra of modified drawbox for different gears at 5000 rev/min: 1, normal steel and brass gears; 2, flexibly bushed gears; 3, cast polyurethane gears.

Several methods have been used to quieten the drawbox, each with its characteristic features. The drawroll gear was vibration isolated from the shaft by a metal sleeve cast in polyurethane in the centre of the gear. This places a resilient link between the gear teeth and the shaft and reduces the excitation of the high-frequency shaft modes. Noise reduction of the major peaks is shown in figure 16. Similar reductions were obtained with resilient links in the shafts themselves. Other approaches included the use of damping processes. Drawroll shafts were bored out, filled with lead shot, steel shot or a manganese copper inset. A complete drawroll shaft was also manufactured from manganese copper alloy, an alloy of high damping capacity. Some benefit was gained from these approaches, but the ringing was still evident. Although the vibration from each tooth impact is damped rapidly by the alloy shaft, the impacts follow each other very quickly so that the following impact occurs before much damping can occur. Thus, there are still appreciable peaks at the natural frequencies of the shaft assembly coming directly from the gear meshing excitation.

Although the main noise peaks are attenuated, these efforts have a common drawback, in that there is still much noise from the metallic tooth meshing that is radiated from the drawbox.

The best solution so far has been the use of 'plastic' or non-metallic gears. At first many materials were tried with varying degrees of success. Gears were cut or moulded, not without difficulty, in materials that included nylon, delrin, tufnol, polythene and polyurethane rubber. Apart from manufacturing difficulties and other problems due to expansion, oil absorption or backlash clearances, the harder the material that was used, the more noise was generated, and generally the softer the material any noise reduction was counterbalanced by shortened gear life. The one exception was polyurethane (isocyanate) rubber, formed by a novel chemical process (Payne & Scott 1960; Davey & Payne 1965). An extensive development programme by I.C.I. Fibres has produced quality moulded gears that are adequate for drawbox applications (A. P. Julian, unpublished work). Provided that the correct grade of hardness and rebound resilience is used, the material is flexible enough so that the teeth deflect to give surface contact rather than line or point contact. Tooth loading is, therefore, reduced. The polyurethane has a degree of inherent damping capacity and combined with its flexibility, i.e. its vibration isolating properties, the tooth impacts are not transmitted to the shaft, resulting in large reductions of the noise associated with the axial and bending modes of the shaft. The non-metallic tooth noise is much lower than the comparable steel-brass tooth meshing noise (figure 16). The life of these gears is much longer than the other 'plastic' materials used because of the phenomenal abrasive and tearing resistance and very high tensile strength of the polyurethanes (Payne & Scott 1960; Davey & Payne 1965). An over-all noise level reduction of 8 dB and a 20 dB reduction in the ringing noise at 3500 Hz are obtained with the polyurethane gears as shown in figure 16. Subjectively, the drawbox is quiet because the reductions have been effected mostly in the sensitive region of the ear. As is often the case in a noise-reduction programme, some of the noise reduction may be sacrificed for a speed increase, the speed increase depending on the noise level that can be tolerated. The steel-brass-steel normal combination will not run at these increased speeds without excessive noise and vibration and eventual stripping of gear teeth.

Headstock drive noise

The drawbox and spindle assemblies are significant noise generators and although they are at the same time not very complicated, their noise sources are many. The headstock is a much more complicated mechanical arrangement incorporating drive systems for the feedrolls, drawrolls, spindles, winding transverse mechanisms and ancillary oil pumps.

Basically the power is supplied by a 40 h.p. electric motor and through duplex chains, triplex chains and gears, the various directions of rotation and speed ratios are attained. Noise was not considered when these machine drives were designed and much can be done to improve them. Although the outside casings are fairly massive providing sound transmission loss from the noise sources, the levels involved are high.

In particular, the high-speed duplex and triplex chains driving the spindle drumshaft and drawroll layshaft are very noisy. Chain drives are inherently noisy particularly at the higher frequencies, owing to teeth-meshing impacts and vibration of the small chain elements. The approach has been to redesign the drive mechanisms using separate electric motors for the individual functions. This type of operation is required for the programmed spindle speed in any case. Chain drives have been abandoned, power being transmitted by wide flat belts and, in some cases, toothed belts. In this way much of the original straight cut gearing is also discarded. In many of the older machines the gear shafts were in pods bolted to the frame casting and often the gears were on the end of a cantilevered shaft with the two shaft bearings on one side of the gear. This allows more movement of the gear with a consequent increase in noise. In the newer machines, the gearing is of a higher quality with the usual precautions of using fibre gear teeth for the intermediate gears and many of the gears are mounted between bearings.

The result of headstock design with noise considerations in view, is a combination of principles. Vibration and out-of-balance forces are cut to a minimum by the use of better quality gears and shafting arrangements. Noisy chain drives are replaced by variable speed motors and belt drives. The drive system is an integrated unit inside a casing that more or less forms a complete enclosure around the headstock. Since there are now parts that do not have oil lubrication, the addition of sound absorptive material within the headstock casing is possible and prevents the build-up of sound within the casing. Circulation of cooling air is required for the electric motors and acoustic considerations can be applied to the flow vents at the design stage.

Secondary noise radiation

Other noise generating parts in drawtwist machines are mainly large radiating surfaces that are excited by vibrating machine elements. Between each bobbin there is a separator board, usually of aluminium alloy, that ensures that one threadline does not interfere with its neighbour should it break. These separator boards are attached to a spindle guard rail that is rigidly attached to the spindle rail at several points. When the mass out-of-balance spindle rotating vectors are at a maximum, the separator boards resonate with a harsh penetrating metallic sound. A simple vibration isolator between the guard rail and separator rail reduces the vibration transmission and the resonating is completely eliminated. Headstock and end-cover plates have also been vibration isolated or replaced with damped materials, keeping them from rattling or responding to frame vibration.

Acoustic absorption

The drawtwist areas are very long and wide with relatively low ceilings. Conventional Sabine room acoustics are not obeyed in these areas because of their dimensions and the scattering of noise by the machines themselves. Further, the arrangement of the machines and their dimensions virtually ensures that the sources are distributed throughout the area. As a result the operators are in the near field or non-reverberant sound field most of the time. However, the operators do spend some of their time in the open areas, some feet from the banks of machines, that are used for storage, rest points, etc., and some relief is afforded by a sound absorptive ceiling treatment. Very little absorption is present in the original buildings, as they are of typical factory construction of painted brickwork, glass, concrete or varnished wooden floors often with metal partitions between the areas. Some areas have false ceilings with the air-conditioning ducting above, leaving a relatively easy area to treat. Others have the maze of ducting directly over the machines and with the added lighting problem, the alteration of existing areas is often difficult.

PHILOSOPHICAL TRANSACTIONS OF THE ROYAL SOCIETY A MATHEMATICAL, PHYSICAL & ENGINEERING SCIENCES

Results have been obtained for one drawtwist area where a false ceiling of woodwool slabs was installed. Although the absorption coefficients are not very high ($\bar{\alpha} = 0{\cdot}5$) some benefit was obtained objectively and subjectively. Tests were carried out with broadband noise from a point source and the intensity fall off with distance from the source was measured in the open areas of the treated and untreated rooms. Comparison with the noise fall off from real machines and with a predicted fall off assuming that the walls of the room are so far away that their contribution to the reverberant field is negligible (Sabine 1952) is shown in figure 17. It is seen that the predicted fall off of noise is in good

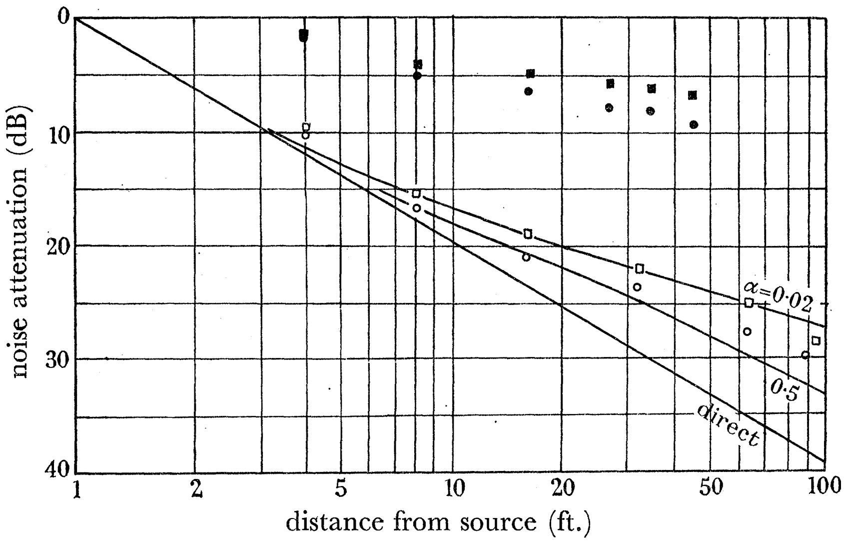

FIGURE 17. Fall off of noise with distance in drawtwist areas. Wide-band noise (point source): □, bare ceiling; ○, woodwool ceiling. Machine noise (dB(A), distributed sources): ■, bare ceiling; ●, woodwool ceiling. The solid lines are predicted attenuations for different ceiling absorption coefficients.

agreement with the measured values for a point source, but the noise fall off from the machines is much less. There is the same amount of noise reduction of 1 to 3 dB, due to the absorbent ceiling for the point source and for the machines. The large discrepancy between the point source and the machines is attributed to the distribution of the machine sources in the area. Although this is only a small benefit in terms of decibels subjectively the background noise is 'pushed back' and conversation can be carried on more readily as the speech interference levels are reduced a little.

Modified machine noise

When all the sources have been attacked and the modifications carried out, i.e. flexibly mounted spindles, polyurethane gears, redesigned headstock and various vibration isolation of parts, the noise contours change and the levels are reduced to meet the recommended levels at the operator's position. In figure 18 octave-band levels for the modified machine are compared to the average drawtwist noise and a deafness risk criterion, n.r. 85. There is a reduction of noise from 128 sones to 77 sones, i.e. a reduction in loudness of 40 %. These noise levels are for the modified machine running with full bobbins at spindle and drawroll speeds higher by 11 and 20 % respectively than the maximum production speeds commonly used. In other words, these are the maximum noise levels likely to be encountered in the production of light denier yarns at present speeds. Heavy denier yarns drawn at lower speeds produce lower levels and if programmed spindle speeds are used, the noise levels will be further reduced. The deafness risk criterion is thus met for the severest drawtwist noise conditions at present.

Throughout the industry, new methods are constantly evolved to produce yarns that may or may not use the principles established over the years. One thing is common in that once the thread has been made, it has to be wound on to some kind of container. The noise of high-speed rotation has and will become even more important and thus the experience gained in the bobbin problem is very relevant.

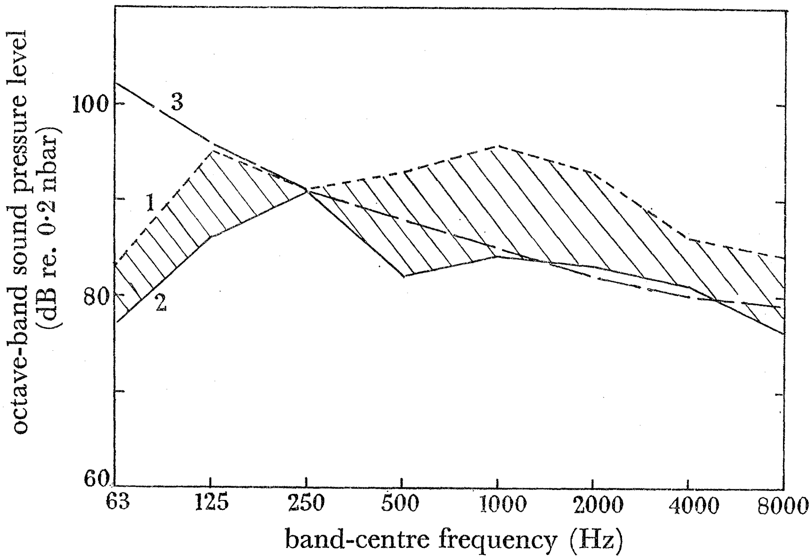

FIGURE 18. Octave-band levels at centre of machine for modified drawtwist machine. 1, Average drawtwist noise; 2, modified machine noise; 3, n.r. 85 criterion.

Surface speeds are fairly high and as even small eccentricities act as strong noise sources, harmonics of the rotational speed will be important. As indicated earlier, it is better to use large diameter containers running at slower speeds—discrete tones are reduced in amplitude and in frequency and are consequently subjectively quieter. Bearing noise too is less at lower speeds.

FUNDAMENTAL WORK

Irregularities in the winding, initial out-of-round, etc., lead to a non-cylindrical shape, generating both discrete and broadband noise.

This type of radiation is being studied on a model rig where experiments have been carried out by spinning short cylinders with sinusoidal lobes on the surface at speeds up to 15000 rev/min. The noise spectra show that for low numbers of lobes (n) the discrete tone generation is governed according to a modified equation (1) in which n the number of lobes, is incorporated. A wound bobbin is fairly well represented by an eccentric cylinder, i.e. a one-lobed cylinder, and defects in the cylindrical shape have been represented by a number of sinusoidal lobes on the cylinder surface. This may not be a true representation, but the approach is relatively simple and leads to the generation of both discrete and broadband noise and has some merits on these accounts.

Once the number of lobes exceeds about seven, at least for diameters up to about 1 ft. the discrete tones are masked by the broadband noise. The aerodynamic surface drag is increased and the broadband noise rises. A broadband peak in the spectrum occurs just below the blade passage frequency and may have a Strouhal number dependence on the peripheral speed. The broadband noise generation is thought to be due to vortex shedding and boundary-layer turbulence and the far field intensity radiation varies approximately as U^6, where U is the peripheral velocity. This type of noise is similar to the windage noise of the rotors of large electrical machines where empirical formula have been developed for the noise radiation (Heubner 1963). It is hoped to further the investigation of the flow around the cylinder and perhaps explain better the broadband noise generation.

Noise in other textile environments

Uptwisting

The noise spectrum of uptwisting (figure 1) is similar to the drawtwisting spectrum. Yarn is rewound from small diameter high-speed bobbins on to surface-driven containers running at relatively low rotational speeds so that a high degree of twist is imparted to the yarn. The drive to the high-speed bobbins is via a single snake belt, kept in contact with the bobbin whorl by idler pulleys. Uptwist noise investigations were carried out several years ago by Wormsbecher (1957), and he obtained an over-all reduction of about 10 dB by redesigning the idler pulleys and the bobbins. The limiting factor was belt noise. Nowadays in the nylon industry as new higher speed processes evolve uptwisting is being used less and less so that modifications to uptwisters are probably not an economic proposition.

Spinning

In the spinning areas producing continuous-filament synthetic fibres, much of the noise stems from the main gearbox, reciprocating traverse bars and surface-driven winders. Partial acoustic screens have been used successfully for noise reduction from the main gearbox and drive assembly (E. Klemm, unpublished work) but a fundamental study of the noise mechanisms in the spinning area is necessary if noise control of the complete process is to be effected.

Staple fibre cutters

Staple fibre cutters are high intensity noise sources, but this process is automatic and requires only periodic attention. Siting of these machines in a suitable area is the simplest answer, but some noise reduction is obtained by an intake silencer.

Weaving

The weaving industry has had noise problems ever since its inception. A conventional loom has changed very little since the Industrial Revolution. Much of the noise arises from the impact and general clatter of the shuttles, picking arms and the harness. (The harness and healds are those parts that alternately separate or 'shed' the warp threads so that the shuttle can pass between them.) The spectrum is mainly high frequency (figure 1) and the sources are difficult to eliminate, mainly because impact is inherent in the process. Noise control of shuttle and picking arm noise in the jute industry (Stott 1965) has quietened a loom by up to 10 dB. This has been achieved by replacing metallic picking points, headle balls and picking cones with polyethylene parts. Nylon driving pinions and nylon bushes have been used in lieu of cast iron. Improvement of the shuttle design by the use of plastics has also been a factor. With these approaches, not only is the noise emission less, but fire risk has also been reduced by the elimination of friction points on the loom. However, radical techniques are probably required in the process itself if noise is to be further reduced. For instance, a water jet loom has been developed in which the picking arm and shuttle are replaced by a small jet of water on which the weft thread rides across the warp. This is used only for man-made fibres, but can operate at a much higher speed than a

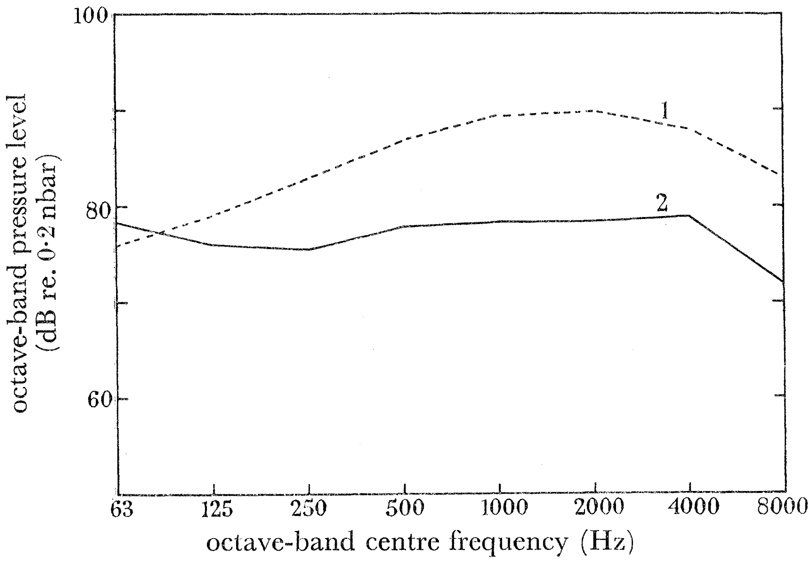

FIGURE 19. Noise comparison of conventional and water jet loom. 1, Shuttle loom, 205 picks/min, 94·5 dB(A); 2, water jet loom, 400 picks/min, 84·5 dB(A).

conventional loom and is still much quieter than it. The spectra for a conventional loom and a water jet loom are shown in figure 19. The high-frequency noise of the water jet loom is due to harness noise and this could easily be reduced even further at the operator position by a semi-cylindrical acoustic screen over the top of the harness. Again, acoustic absorption in a weaving shed is an advantage—probably more so than in a drawtwist area. Weaving is basically impact noise and by reducing the reverberation time in the area the peak impacts of the other machines are reduced so that an operator hears the noise of his own machine and less of the others in the area (Sabine 1952). Subjectively, this is more pleasing.

Conclusions

It has been shown that many textile processes should and can be quietened. A limited amount can be achieved by intuition and much more if the basic noise generating mechanisms are examined. A systematic study and treatment of the sources leads not only to a reduction of the noise at source to acceptable levels, but also has been shown to lead to a better design of the machines themselves and speed increases have been possible, balanced against the available noise reduction.

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