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SMALL ALTERNATOR COOLING FAN NOISE

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1. INTRODUCTION

Legislation has been introduced in Europe to limit the overall sound level emitted by motor vehicles while undergoing a standard "drive by" test. In successive Directives, the Council of Ministers for the Economic Community of Europe have reduced the maximum permissible sound level from most classes of road transport vehicle. The individual steps were typically 2 to 4 dB(A), but the overall effect has been to reduce the noise from public service vehicles (psvs) and many trucks by nearly 10 dB(A) over a period of 16 years.

The test procedure specified requires the engine to be operated near the top of its speed range, in such a gear that the vehicle is travelling at 50 km/hr at the start of the test. During the test, the vehicle is driven at maximum acceleration until it has completed a distance of 20 metres plus its own length, whereupon it is decelerated. A microphone is placed opposite the centre point of the test track, 7.5 m from the centre line of the vehicle, and 1.2 m above the (acoustically reflecting) test track surface. With such a procedure, noise from the engine, and any engine-driven auxiliary components, will be near its maximum level, while the noise from the tyre/road interaction, and the drive train will be appropriate to 50 km/hr or slightly faster. Thus if any engine-driven auxiliaries generate high levels of noise, this could affect the overall sound level from the vehicle. In general, engine and vehicle manufacturers require noise from auxiliaries to contribute insignificantly to the engine noise under drive by test conditions.

When the first Directive specifying maximum permissible vehicle levels and the test procedure was published, the engineers at Lucas Industries Noise Centre calculated target levels for the alternators used in truck and psv applications, and identified those machines which required noise reduction, as reported previously (ref. 1). Automotive alternators are required to generate sufficient current to satisfy the vehicle needs while the engine is idling, which determines the drive pulley ratio. When the engine is running at high speeds (2800 to 6000 rev/min) the alternator can be rotating at 6000 to 16000 rev/min, and sometimes more, depending upon the application. Since fan noise is greatly influenced by fan speed, fan noise can be a considerable problem in these alternators.

A great deal of work has been done to quantify and reduce noise from fans, particularly ventilation fans and jet engines (refs. 2, 3 and 4). This work has been summarised by S Glegg (ref. 4) who lists five mechanisms by which fans generate noise, two "classical", and three additional ones identified by Morfey (ref. 5). Of the classical fan noise mechanisms, fluctuating loads placed upon the blades by variations in pressure or velocity of the air is the most important source of noise for many fans, causing noise to increase with the sixth power of fan blade tip speed. However, if the fan blade tip speed approaches the speed of sound, other mechanisms rapidly become dominant.

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The cooling fans which are fitted to alternators, and some other small rotating electrical machines, operate under very different conditions to those specified in the standard texts, for example:-

1. The fan operates over a wide speed range.

2. Space for the fan within the machine, or between the machine and the drive pulley is limited.

3. Space for the air entering the fan is often obstructed, leading to disturbed and often turbulent flow at the fan inlet.

Comparisons between production alternator samples, with external centrifugal fans, from different manufacturers show a surprising degree of similarity. Figure 1 shows comparisons between two machines in each of two frame sizes with no electrical load on the machines. The differences are less than 2 dB(A) over most of the speed range; in the case of the larger frame size the extra noise is due to the residual magnetic field in one machine.

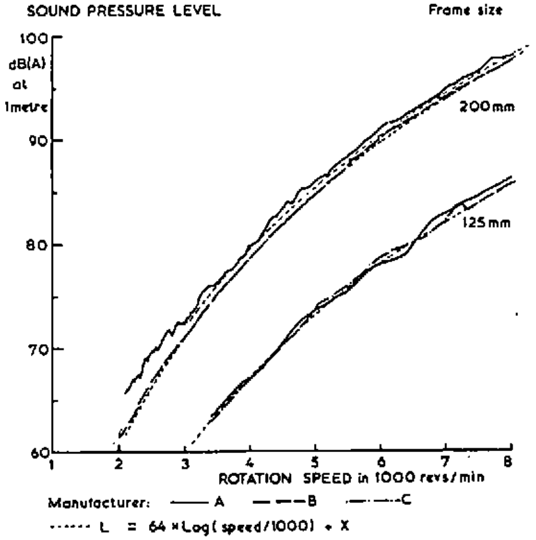


FIGURE 1 Comparison between fan noise from different alternator manufacturers

In Figure 1, the dotted line shows the best fit power law to the experimental measurements. For both frame sizes, the sound level increased with rotational speed raised to the power 6.4. Thus the measurements could be approximated by the equation:-

$$L_A = 64 \times \text{Log}_{10}\,(\text{rotation speed}/1000) + X$$

where X is the sound level at 1000 rev/min.

Clearly to meet increasingly stringent legislation, the value of X for the particular fan should be reduced and, if possible, the power law index should be reduced to minimise noise at high rotation speeds.

2. EFFECT OF RESONANCES

In many machines, the steady increase in sound level with rotational speed is interrupted by sudden increases in level where mechanical or acoustic resonances are excited. In Figure 2, two resonances are illustrated from a particular machine and fan combination which exhibited both acoustic and mechanical resonances. The overall increase of noise with speed approximated to the power 5.4. Between 2600 and 3000 rev/min residual magnetic fields in the salient pole rotor, caused magnetic flux variations as the poles passed under the slots in the stator, and the consequent force variations excited mechanical resonances in the stator. In Figure 2, the increase in sound level was 7 dB(A), but increases of 15 dB(A) have been measured when machines are under load.

In some machines it is possible for the changes in magnetic flux, or consequent vibration, to excite the fan disc into resonance, and this is usually evident as a peak in the noise-v-speed trace when measured on the axis of the machine opposite the pulley.

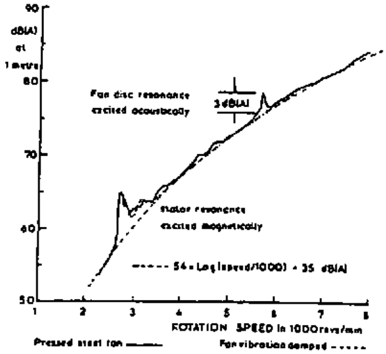


FIGURE 2 Magnetically and aerodynamically excited resonances

Figure 2 shows also a peak in the trace caused by a fan disc resonance which is excited acoustically. Cast metal fans often derive useful damping from the joints between the fan and the rotor or pulley. If the fan disc structural resonance has a node at this radius, then the structural resonance can be excited sufficiently to radiate significant sound levels by fan blade-passing frequency forces.

The cavities and air-paths within the machine provide the potential for numerous acoustic resonances within the fan blade-passing frequency range. Their effect may be minimised if the area for cooling air is kept as constant as possible throughout the machine and up to the fan entry.

3. EFFECT OF FAN DIAMETER

The influence of fan diameter has been shown by running fans of different diameters, but with the same blade configuration, in controlled conditions on the same machine. Figure 3 shows results for two frame sizes of machine, both with external fans (fan mounted between the drive pulley and the drive endshield). The results in Figure 3(a) are a compilation of measurements from several different designs of fan fitted to the same machine, encompassing different numbers of blades, different blade sizes and different axial clearances between fan and endshield. From these results, comparisons have only been made between fans which are similar in design except for their diameter. These results indicate a more general trend than the results for a particular fan in Figure 3(b). In both figures, the dashed line indicates where the measurements should lie if sound level increased with fan diameter to the power 5.4. The measurements show that no greater power is appropriate, and so it would seem to indicate that the increase in sound level with diameter is somewhat less than that with speed.

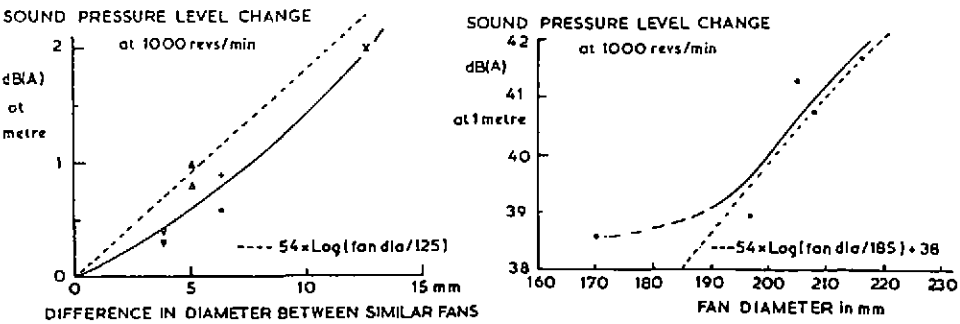


FIGURE 3 Effect of fan diameter on sound level

4. NUMBER AND SIZE OF BLADES

For two different machine designs, which were cooled by external centrifugal fans, experimental fans were constructed to determine the optimum number of blades to maximise the flow of cooling air through the machine. The cooling air flow increased with the number of blades up to 14 blades. When fans with more blades were tested, it appears that the constriction at the fan blade entry reduced the air flow and offset any further gain. Obviously this constriction could be removed by placing the blades further out on the fan disc, however this would increase the fan diameter, and therefore increase the noise as described in the previous section. Thus the number of fan blades, and the size of blades, becomes a compromise between total blade area and air flow area at blade entry; and that compromise design has to fit within the confines of the machine envelope.

Experimental fans have been constructed with backward-sloping blades instead of radial blades, for both the machine frame sizes mentioned previously. No improvement was found in cooling air flow or noise, and it was convenient to retain radial blades to allow operation in either direction of rotation. Despite these results, it ought to be possible to improve the fan efficiency by a better blade design but the gains appear to be small by comparison with changes in other dimensions.

When designing a fan to fit between the drive pulley and the drive-end endshield, there is a temptation to extend the blades towards the machine to fill the available space. This may be acceptable if the blades are shrouded, or if they abut a plane face of the endshield without any irregularities, holes, endshield legs, or other disturbance to the air flow. However, the need to reduce the diameter of the fan to minimise noise can cause the blades to be extended into the region where the air flow is disturbed by the endshield legs, and here the clearance between the endshield and the fan blades becomes a critical factor in determining fan noise. Figure 4 shows the effect of increasing axial clearance between the endshield (spider legs) and the fan blades for unshrouded radial blades on an external fan. The sound level from the fan seems to change with the ratio of axial clearance rather than absolute axial clearance.

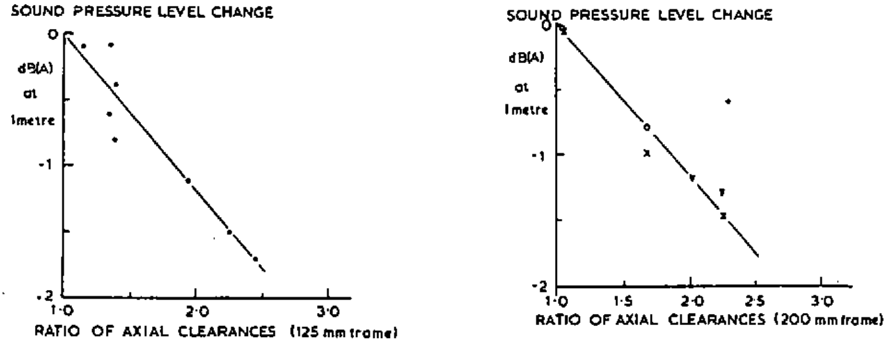


FIGURE 4 Effect of axial clearance between fan and endshield on sound level

With some care an external fan design can be produced which has a large axial clearance between any static obstructions upstream of the fan and the fan blades themselves, but which can run close to the endshield face towards its outer diameter. The fan blade area must be sufficient to impart the required energy to the cooling air to draw it through the machine against the pressure drop across the machine.

5. INLET AND OUTLET CONDITIONS FOR INTERNAL FANS

If the fan design can be made so efficient that the fan can be fitted inside the endshield(s), as shown in Figure 5, then the fan inlet and outlet conditions need to be reassessed.

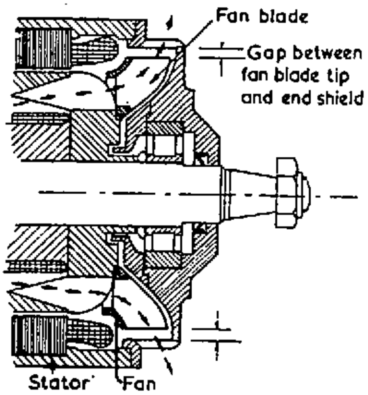


FIGURE 5 Air flow through salient pole machine with an internal fan

In most rotating machines, the air between the rotor and the stator is itself rotating at an angular velocity about the machine axis which is intermediate between the rotor speed and the stator speed. In the particular case of automotive alternators, which have salient poles on the rotor, the angular velocity of the air must approach that of the rotor itself. If this angular velocity can be maintained without loss at the entry to the fan blades, then the fan blade can be reduced in size and the radial clearance between the fan blade tips and the stator can be increased. However, if the fan is separated from the rotor, allowing the cooling air to expand into a space before entering the blades, the angular velocity will reduce and turbulent flow may be induced by the expansion and flow over sharp edges.

Experimental internal fans have been produced which have six extended blades to abut the salient poles of an alternator rotor. The area for air flow has been maintained constant from the rotor through the fan inlet. As a result, the flow of cooling air has been increased from a fan with the same diameter as the original design.

Traces of noise-v-speed were produced for the machines with internal fans, in the same way as those described earlier for machines with external fans. Figure 6 shows how the sound level varied with speed, and also how the sound level varied with clearance between fan blade tips and the air outlets in the endshield. The noise-v-speed traces seemed to approximate best to a power law whereby sound level increased with rotation speed to the power 5.3. In the subsequent comparisons between fans, the sound level at 1000 rev/min for the power law approximation to measured traces is used as the characteristic noise parameter.

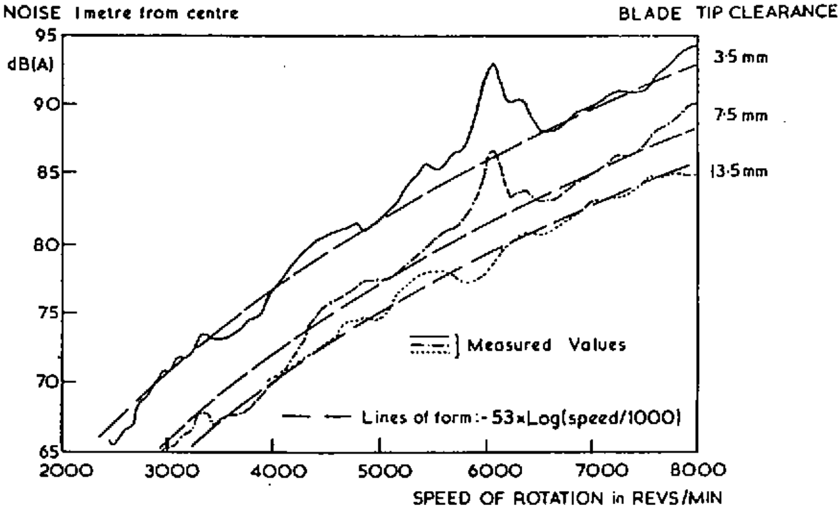


FIGURE 6 Sound levels from an internal fan with three radial clearances between blade tips and air outlets in the endshield

Figure 7 shows the comparison between a shrouded fan design with short blades around the disc periphery, which had been designed to allow the air to mix and settle before entering the fan blade ("s" in Fig. 7), and an unshrouded fan with longer blades which was measured with a series of diameters. The characteristic sound level from the power law approximation to the measured traces has been plotted against cooling air flow. The effect of extending the blades towards the salient pole rotor has been to increase the air flow by approximately 20 per cent, even when the unshrouded fan was slightly smaller. This increase in air flow can be traded off for a 3 dB(A) improvement in noise, giving an overall improvement of nearly 5 dB(A) for the same flow of cooling air. This set of results confirms the well established "rule of thumb" that the radial clearance between blade tips and endshield outlets should be no less than $^1/_6$ of the radius of the fan.

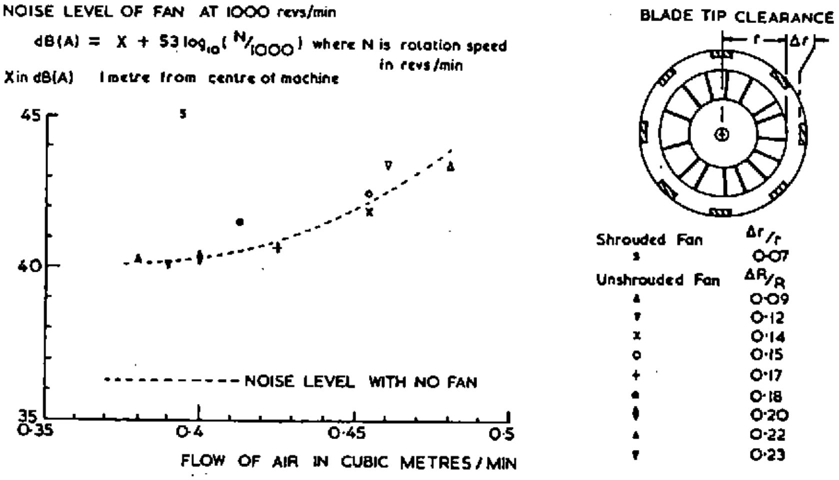


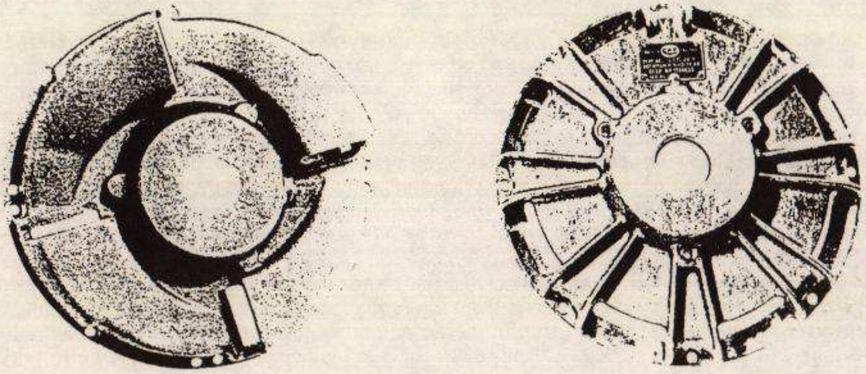
FIGURE 7 Reduction of noise and air flow as the diameter of an internal fan is reduced

7. MULTIPLE VOLUTE ENDSHIELD

As machine outputs increase, and rotor/stator air gaps decrease, the pressure head required to draw cooling air through the machine, and through the diode pack, has increased which requires changes to the cooling fan. Most centrifugal fans in heating and ventilating systems are combined with volute diffusers to increase the pressure head developed by the fan, but there is no space around automotive alternators for such devices to be incorporated. As an alternative, four parallel diffusers were developed out of the face of the drive endshield to provide some improvement in the pressure head of the fan without significantly changing the machine envelope. The arrangement is sketched in Figure 8.

The new endshield was fitted to a large alternator for a public service vehicle application and the increase in air flow was dramatic. Figure 9 shows the results in Figure 7 to a more compressed air flow scale, in addition to those from the same machine with the multiple volute endshield, to maket this comparison. For all diameters apart from the smallest, the flow of cooling air increased by 40 per cent or more, and the air flow from the shrouded fan increased by nearly 50 per cent. The optimum trade-off between noise and air flow occurred with the radial gap between blade tips and endshield which was $^1/_6$ of the fan radius as before, but it was now possible to reduce the fan diameter still more to achieve another decibel of noise reduction while passing a greater flow of cooling air.*

* Patent applied for.



Multiple volute *Standard*

FIGURE 8 Photographs of endshields for the same machine

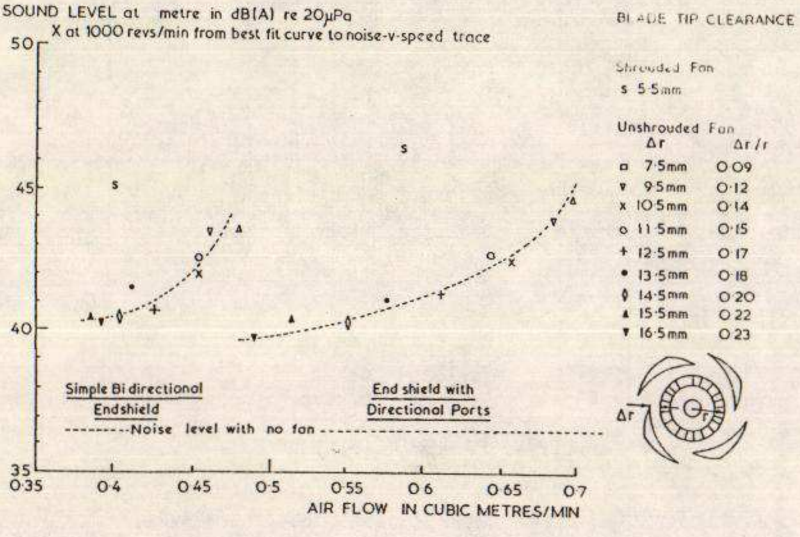


FIGURE 9 Effect of multiple volute endshield on sound level versus air flow trade-off

7. IMPROVING NOISE QUALITY

The primary requirement for alternator cooling fan noise was to reduce the overall level to minimise the contribution to engine and vehicle noise during the drive by test. In addition there is a requirement to minimise any strong tonal components which may be obtrusive, disturbing or annoying to the occupants of the vehicle. The tonal component comes from the "wake chopping" at blade passing frequency and the lowest common multiple of blade passing frequency and the number of endshield legs. In the heavy duty Lucas products fitted to public service vehicles and trucks, it has been common practice to space the blades irregularly around the fan disc.† This is easy to arrange when moulded plastic and cast aluminium fans are used, but requires some more ingenuity when pressed metal fans are employed. The irregular spacing of the blades breaks up the "whine" produced by wakes shed from blade tips being chopped by the endshield legs between the outlets for the cooling air. This technique does not greatly affect the overall sound level from the fan, although moving some of the sound energy from frequencies close to the peak in the A-weighting to lower frequencies does reduce the level in dB(A) slightly. The main improvement is to replace one strong tone with its harmonics by several harmonics of rotation frequency, which distributes the energy more evenly across the audible frequency range. Figure 10 shows the sound spectra from two similar fans running at 6000 rev/min; one with regular blades, where the blade passing frequency (b) and the first and second harmonics are clearly evident; and the second fitted with randomly-spaced blades which shows a more even spectrum with no dominant blade-passing frequency peak.

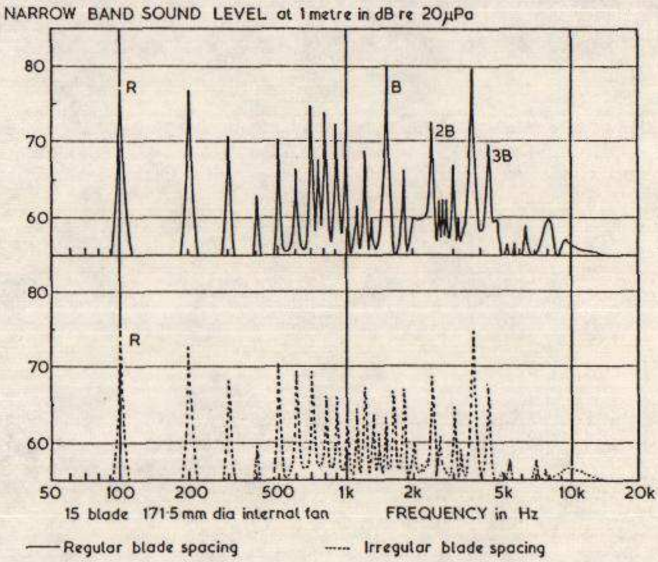


FIGURE 10 Spectra of noise from two similar fans at 6000 revs/min

Subjectively, spacing the blades irregularly produces a "burr" which does not obtrude above the noise from the rest of the vehicle.

Sudden increases in the sound level of a pure tone can prove disturbing to vehicle passengers, especially in public service vehicles, when the increase is greater than 5 dB(A). It is rare for the alternator cooling fan to become a dominant source of noise inside such a vehicle, so greater variations in sound level can be tolerated in practice, however irregular blade spacing provides an increase in noise quality in two dimensions, both reduction of whine and reduction in sudden increases in sound level. This technique is also useful if mechanical or acoustic resonances cannot be treated by damping. The randomly-spaced blades cause less excitation of resonances and so do not cause large and sudden increases in sound level as the machine speed is increased. An example of this effect is shown in Figure 11, where a particular machine exhibited a large number of resonances which were excited by fan blade passing frequency, and the undulations in the noise-v-speed trace were significantly reduced when an irregular blade spacing was employed.

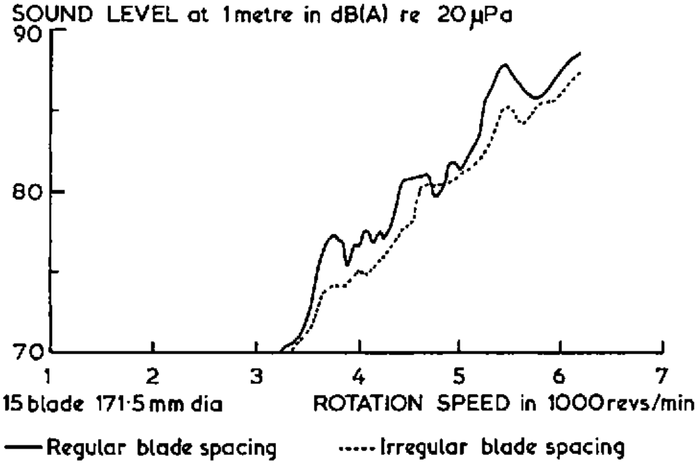


FIGURE 11 Effect of irregular blade spacing on A-weighted sound level

8. SUMMARY

Work on small automotive alternators and electric motors has shown that the noise generated by their centrifugal cooling fans can be controlled by:-

1. Reducing the fan rotational speed; noise has been shown to increase at a rate of 53 to 64 dB/decade increase of speed.

2. Reducing the rate at which sound level increases with speed from the power 6 to 5.5 or less.

3. Ensuring that the design includes an adequate flow-settling distance between the fan blades and any static obstructions to minimise "wake-chopping":-

    (a) For external fans the axial clearance between blades and stator legs or other aerodynamic obstructions should be at least equal to the width of the endshield legs.

    (b) For internal fans, the radial clearance between the blade tips and the stator should be at least one sixth of the blade tip radius.

    (c) All unnecessary flow-obstructions such as casting imperfections, counterbored holes for bolt heads, brackets, lugs, etc., should be removed.

4. Maintaining a constant area for air flow through the machine, particularly at the fan inlet, where small modifications can have large effects on air flow.

5. Utilising any angular momentum imparted to the air flow by the rotor, by integrating the fan blades with the rotor.

6. Matching the pressure head provided by the fan to that required to move the cooling air through the machine with controlled expansion of the air flow leaving the fan.

7. Spacing the blades irregularly around the fan disc to improve the subjective quality of the noise radiated by fans.

10. ACKNOWLEDGEMENTS

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