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THE 75 dB(A) THRESHOLD LEVEL OF THE PHYSICAL AGENTS DIRECTIVE: A FLAWED EVOLUTION

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1. INTRODUCTION

In 1993, the Council of the European Communities proposed a Directive dealing with the exposure of workers to physical agents, one of which is noise (CEC, 1993; amended by CEU, 1994). This draft Physical Agents Directive is intended to repeal the current noise-at-work regulations (CEC, 1986) and to introduce measures which are more protective. As a matter of social and employment policy, the risks arising from noise exposure must be reduced to the lowest achievable level, ultimately to below a Threshold Level, "the exposure value below which continuous and/or repetitive exposure has no adverse effect on the health and safety of workers". In respect of noise-induced hearing loss, the proposed Threshold Level is set at an $L_{EX,8h}$ value of 75 dB(A).

In the light of the definitions above, the various Articles of the proposed Directive require certain general and specific actions on the part of the employer.

1. Taking account of the availability of measures to control noise at its source, the risks arising from noise exposure must be reduced to the lowest achievable level; the aim is to get workers' daily noise exposure below 75 dB(A).
2. If noise exposure exceeds the Threshold Level, then exposed workers must be informed of the potential risk of noise-induced hearing loss.

Obviously, the Threshold Level will assume considerable importance for industry, becoming the target for any noise control programme. If that ideal cannot be achieved, then exposed workers must be informed that their hearing may be at risk from noise at work.

The draft Physical Agents Directive was accompanied by an Explanatory Memorandum which gave background on the aim of the proposal and the need for action, relying upon a draft International Standard (International Organization for Standardization, 1982 and 1985). The Memorandum stated that, during the period leading up to the 1986 Council Directive on noise at work, "scientific and technical knowledge was already sufficiently advanced to make it possible to ascertain precisely the harmful effect of noise on hearing capacity. The scientific community had already established that, from 75 dB(A) on, the risks run by workers were far from negligible." These statements may be challenged on several counts.

Over the few years before the 1986 Directive, the exposure level 75 dB(A) was being canvassed in international circles as the threshold at which *no effect whatever* would be observed in human hearing, even at the most noise-sensitive frequency 4 kHz. To imply that hearing risk, from that exposure level upward, would be "far from negligible" is an unwarranted overstatement. The effect is progressive: there is no sudden jump from no effect to "far from negligible" as the threshold is crossed. A more accurate perception may be gained by considering the Threshold Level from both directions as a transition zone or band of uncertainty *above which hearing damage is measurable and significant, and below which no effects are discernible*. This idea is supported by a number of occupational hearing loss studies involving low-level noise exposures (eg Passchier-Vermeer, 1988; Flottorp, 1995). Male workers exposed to a daily average noise level of 80 dB(A) or less showed no hearing loss except that expected for their age. Where noise exceeded an $L_{Aeq,8h}$ of 85 dB(A), a risk of noise-induced hearing loss emerged. As workplace noise increased to higher levels, the risk of hearing damage became plain.

This paper examines the background of the 75 dB(A) exposure threshold of ISO 1999, which is the foundation of the Threshold Level of the proposed Physical Agents Directive. However, it will be shown that ISO 1999 is flawed. There is considerable uncertainty in the predicted magnitude of noise-induced threshold shift, particularly for low noise exposure levels; this uncertainty persists in the Threshold Level of the draft Directive.

2. THE ISO STANDARD

The various editions (1982, 1985, 1990) of the standard ISO 1999 enable the prediction of Hearing Threshold Levels resulting from noise exposure and natural ageing. Formulae are given for the calculation of noise-induced permanent threshold shift (NIPTS) for the frequencies 500 Hz to 6 kHz. The noise is deemed to last 8 hours per day, over a period ranging from 0 to 40 years, with daily exposure $L_{EX,8h}$ ranging from 75 dB(A) to 100 dB(A).

The lower limit of applicability of the standard, an $L_{EX,8h}$ of 75 dB(A), is implicit in the calculation method. For exposure levels above 75 dB(A), and durations ranging from 10 to 40 years, the median NIPTS values, $N_{0.50}$ in dB, for both sexes are given by the equation:

$$N_{0.50} = [u + v \log q\,][\,L_{EX,8h} - L_0]^2$$

where L_0 is a cut-off sound pressure level defined as a function of audiometric frequency,
q is the exposure time in years, and
u and v are parameters tabulated as a function of frequency.

The values of u and v (appropriate to the different frequencies) are of no interest for present purposes; the values of L_0 are reproduced below in Table 1.

Table 1: Values of L_0 for each audiometric frequency

Frequency, kHz	L_0, dB
0.5	93
1	89
2	80
3	77
4	75
6	77

Note that L_0 assumes a minimum value at 4 kHz, the audiometric frequency at which noise-induced hearing loss usually appears first. For an $L_{EX,8h}$ of 75 dB(A), the squared term in the equation above equals zero, therefore the median NIPTS is zero. For $L_{EX,8h}$ less than 75 dB(A), NIPTS is defined as zero. Other fractiles of the NIPTS distribution are also calculated using the term $[\,L_{EX,8h} - L_0\,]^2$. Therefore, NIPTS at 4 kHz is zero for all members of the population exposed to 75 dB(A) or less. In fact, a non-zero value of NIPTS at 4 kHz is not predicted until the exposure reaches 78 dB(A).

To show the influence of the quantity L_0 upon NIPTS, predicted threshold shifts are given in Table 2 for the frequencies 1–6 kHz, and for various values of $L_{EX,8h}$. Of interest here is hearing loss due to relatively low noise exposures, therefore NIPTS values are given to represent the median of the noise-exposed population and also the most noise-susceptible decile. An $L_{EX,8h}$ of 75 dB(A) yields NIPTS values of zero. Exposures of 76 and 77 dB(A) give non-zero NIPTS values for 4 kHz but rounding to the nearest integer decibel still gives entries of zero.

For an $L_{EX,8h}$ of 78 dB(A), a non-zero value of NIPTS at 4 kHz emerges for the 'tender-ear' decile after 10 years of exposure. After 20 years of exposure, the threshold shift at the median reaches 1 dB. Further exposure duration up to 40 years does not enlarge the 1 dB of NIPTS at 4 kHz, and does not produce NIPTS at any other frequency. A threshold shift of 1 dB is virtually undetectable, and is imperceptible to the person with such a loss. Even if the predicted values of NIPTS are accepted as true, the Threshold Level of the proposed Directive should not be 75 dB(A), but closer to 80 dB(A).

Still referring to Table 2, it may be seen that an $L_{EX,8h}$ of 80 dB(A) is another landmark: non-zero values of NIPTS are now exhibited for the frequencies 3 and 6 kHz, as well as for 4 kHz, the most noise-sensitive frequency. The predicted NIPTS values are, however, very small. As the noise exposure assumes higher values, 85 and 90 dB(A), it may be seen that NIPTS grows quickly over the exposure duration period up to 10 years, and then less quickly over the period 10–40 years.

Table 2. Noise-induced permanent threshold shift, in dB, by ISO 1999 calculations.

Exposure duration		10 yr		20 yr		30 yr		40 yr	
	Fractile	50%	10%	50%	10%	50%	10%	50%	10%
$L_{EX,8h}$ dB(A)	Freq. kHz								
78	1	0	0	0	0	0	0	0	0
	2	0	0	0	0	0	0	0	0
	3	0	0	0	0	0	0	0	0
	4	0	1	1	1	1	1	1	1
	6	0	0	0	0	0	0	0	0
80	1	0	0	0	0	0	0	0	0
	2	0	0	0	0	0	0	0	0
	3	0	1	1	1	1	1	1	1
	4	1	2	1	2	2	2	2	2
	6	0	1	0	1	0	1	1	1
85	1	0	0	0	0	0	0	0	0
	2	1	1	1	2	1	2	2	2
	3	3	5	4	6	4	7	5	7
	4	5	7	6	8	6	9	7	9
	6	3	4	3	5	3	6	4	6
90	1	0	0	0	0	0	0	0	0
	2	2	6	4	8	5	9	6	10
	3	8	13	10	16	11	18	12	19
	4	11	15	13	18	14	19	15	20
	6	7	12	8	14	9	15	10	15

The behaviour of NIPTS across frequencies, exposure levels and exposure durations is determined by the workings of the equation for $N_{0.50}$ as seen above. Its derivation is based upon work undertaken for the ISO committee by Johnson (1978), forcing a synthesis of two incompatible data sets.

2.1 UK Data

Hearing surveys of noise-exposed British workers were conducted during the 1960s by the Medical Research Council working with the National Physical Laboratory. The findings of this work were condensed into a set of convenient tables (Robinson and Shipton, 1977) giving a form of NIPTS for a normal population with noise exposure every working day for a given number of years. Provision is made for various fractiles of the population, frequency (0.5–6 kHz), and sex. Adding a standard age correction gives overall Hearing Threshold Level (HTL). This NPL data formed one half of the input to Johnson's 1978 synthesis

To allow convenient use of the NPL tables, the statistical variation of both age and noise effects is loaded on to the noise component alone and *cannot* be partitioned between the two components. Johnson, nevertheless, attempted just such a separation in order to force the British data into a format compatible with other information available at the time.

2.2 Dutch Data

Johnson's other input data were contained in a report by Passchier-Vermeer (1977). The hearing loss model was given in the form which Johnson desired, as separate distributions (90%, 50% and 10%) of NIPTS and age components, with overall HTLs being obtained by summing at corresponding fractiles.

Although being in the desired format, the data were far from satisfactory. For the frequency 4 kHz, expected to show the first sign of damage by noise, the NIPTS component actually *decreased* with increasing years of exposure for the 'tender-ear' decile. Such a notion is, of course, false: longer noise exposure is not beneficial to the hearing. A related fault indicated that, for some durations, the 'tender-ear' decile had a smaller NIPTS component than the 'tough-ear' end of the distribution. Despite such obvious shortcomings, the Dutch data was included in the ISO 1999 model for noise-induced hearing loss.

2.3 The ISO Model

The data derived from the two sources described above were not in close agreement; in some cases, the corresponding values differ by over 20 dB. However discordant, the values were simply averaged to provide a data field which was then modelled mathematically. The curve-fitting procedure introduced two arbitrary assumptions regarding the noise exposure threshold.

1. Noise exposures of 75 dB(A), no matter for how long, produce no hearing loss. This threshold value has its origin elsewhere, as will be shown in following sections.
2. NIPTS is proportional to the excess of actual exposure over the 75 dB(A) threshold. This model for the growth of NIPTS has no foundation in experiment.

In addition to these assumptions regarding the noise exposure threshold, the mathematical model for estimating noise-induced permanent threshold shift is based upon two incompatible data sets, one of which has been mistreated, while the other is simply incorrect. The hearing loss calculation method for ISO 1999 is flawed: there is a large underlying uncertainty of the magnitude of NIPTS, particularly for low noise levels near the noise exposure Threshold Level.

3. THE WORLD HEALTH ORGANIZATION

The World Health Organization (1980) criteria on noise gave information on the estimation of hearing impairment risk, and might be thought of as an authoritative source for the ISO 75 dB(A) noise exposure threshold. A summary statement was given: "For the working environment, there is no identifiable risk of hearing damage in noise levels of less than 75 dB(A) L_{eq} (8-h). For higher levels, there is an increasing predictable risk and this must be taken into account when setting occupational noise standards." On the specific subject of noise-induced permanent threshold shift, the important text was: "... the percentage of people who suffer an NIPTS of 5 dB (the smallest amount measurable) at the most sensitive frequency (4000 kHz) may be defined as a function of an equivalent 8-h level. [Here, the WHO document refers to a diagram attributed to US Environmental Protection Agency, 1974.] From this diagram, an 8-h equivalent level of 75 dB(A) can be identified as the limit for protection against significant NIPTS." Whilst adding support to the ISO exposure threshold, it is clear that the WHO recommendation is a simple restatement of information from the US Environmental Protection Agency.

4. THE EPA "LEVELS DOCUMENT"

The 75 dB(A) noise exposure value featured in an important social policy document from the US Environmental Protection Agency (1974), to establish sound levels which would not adversely affect public health. Any measurable loss of hearing sensitivity, at any audiometric frequency, was deemed unacceptable. A noise-induced permanent threshold shift of 0 dB was thought ideal, but not entirely appropriate: there was no evidence to suggest that a NIPTS of 5 dB or less would be perceptible by an individual with such a hearing loss. Therefore, an imperceptibly small threshold shift at the most noise-sensitive frequency 4 kHz was thought to be of no practical significance.

The EPA recommendation was based upon a synthesis (earlier work by Johnson, 1973) of results from three occupational hearing loss surveys, one Dutch, one British, and the last American. These survey results were summarized to give the predicted maximum NIPTS for the better-hearing decile, median, and worse-hearing decile of a noise-exposed population after a 40 year working lifetime in a range of average noise levels.

Table 3. NIPTS (dB) at 4 kHz

$L_{Aeq,8h}$	Fractile		
dB(A)	90%	50%	10%
75	0	1	6
80	2	4	11
85	5	9	19
90	11	15	28

These predicted NIPTS values were the result of extrapolation below the observed exposure levels in all three surveys, down to 75 dB(A). To use the Dutch NIPTS data, a second extrapolation was made across percentiles. In the American survey, the threshold data were inflated by Temporary Threshold Shift. (It is worth noting that the US data were not included in Johnson's 1978 work for ISO.) Despite the uncertainties of the predicted threshold shift data, these values were used to define a curve representing 5 dB NIPTS at 4 kHz; see Figure 1.

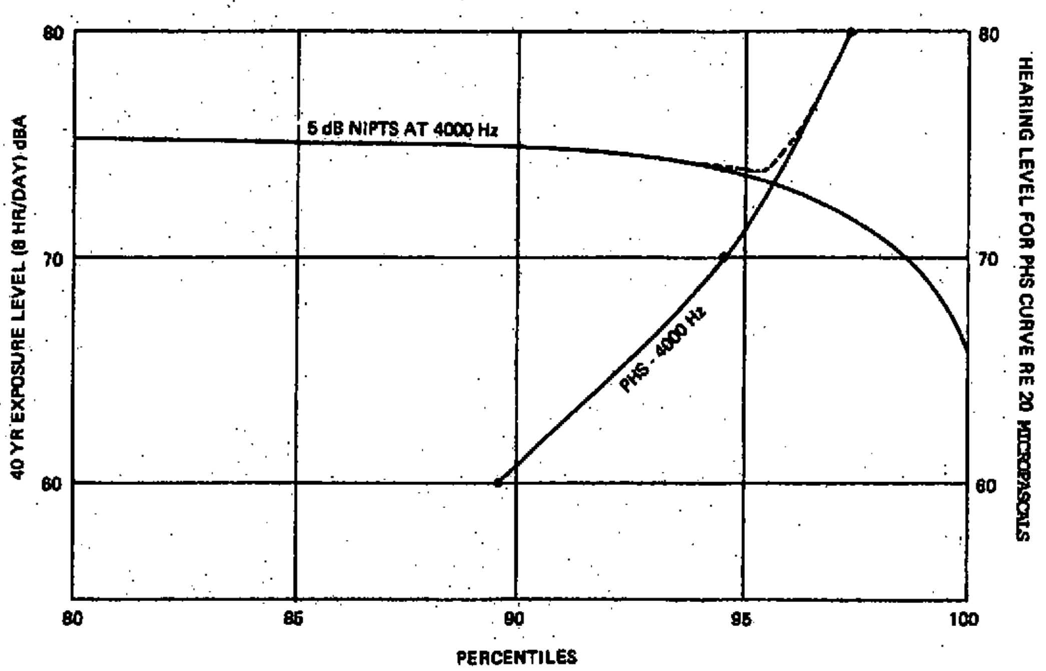

Figure 1. This diagram appeared in the Environmental Protection Agency document of 1974 bearing the caption: "Exposure Level and Hearing Level as a Function of Population Percentile, Showing the 5 dB NIPTS Curve Merging with the PHS 4000 Hz Curve". The 5 dB NIPTS curve is set against the left-hand vertical axis of '40 yr exposure level (8 hr/day) dBA'. The PHS 4 kHz curve represents data from US Public Health Service for women aged 60 years; this line must be read using the right-hand vertical axis ' Hearing Level for PHS curve re 20 micropascals'.

A further step involved the seemingly reasonable limiting condition that one's hearing cannot be damaged by a sound not loud enough to be audible. The worse-hearing extreme of the distribution of 4 kHz HTLs in 60 year old women was used (wrongly) to estimate what sound levels would be inaudible to younger workers. This information is superimposed on the 5 dB NIPTS line of Figure 1 by the simple expedient of equating the A-weighted exposure level of a broadband noise in free air to the magnitude of a pure tone, presented by earphone and measured in terms of dB above 'audiometric zero'. If one accepts this numerical operation, then a conclusion may be drawn from the Figure: If occupational noise exposure is limited to 75 dB(A) or less, then no US worker will suffer a NIPTS greater than 5 dB at 4 kHz.

Despite the uncertainties of this line of reasoning, the EPA recommended that an $L_{Aeq,8h}$ of 75 dB(A) would protect the hearing of the American public "with an adequate margin of safety". In fact, the recommended occupational noise threshold value was intentionally conservative to protect the *entire* American working population from an imperceptible hearing loss due to noise, even after a working lifetime of 40 years.

5. CONCLUSIONS

There are a number of issues embedded in the noise exposure Threshold Level of the European Union draft Physical Agents Directive. The Threshold Level was set at $L_{EX,8h}$ = 75 dB(A), administratively marking "the exposure value below which continuous and/or repetitive exposure has no adverse effect on the health and safety of workers". This is a statement by the Commission of the European Union of a new goal of social and employment policy, in respect of hearing damage by occupational noise.

It appears that the value 75 dB(A) originates with the International Organization for Standardization, but consideration of the contemporary literature indicates otherwise. The US Environmental Protection Agency judged that an $L_{Aeq,8h}$ of 75 dB(A) would protect the hearing of the American public "with an adequate margin of safety". In fact, this recommended occupational noise threshold value was intentionally conservative to protect the *entire* working population from even minimal hearing injury at the most noise-susceptible audiometric frequency. The WHO endorsed this goal of social policy.

Since the EPA, WHO and ISO documents were published, there have been a number of occupational hearing loss studies involving low-level noise exposures. The results suggest a region of transition between no noise effect and a clear effect. Below an $L_{EX,8h}$ value of 80 dB(A), the daily occupational noise seems to have little if any effect upon the 4 kHz threshold of even the most susceptible workers; male workers exposed to a daily average noise level of 80 dB(A) or less showed no hearing loss except that typically expected for their age. As workplace noise becomes greater, the risk of hearing damage becomes evident. Where noise exceeds an $L_{Aeq,8h}$ of 85 dB(A), there is a chance of acquiring noise-induced hearing loss. Above 90 dB(A), the risk of hearing damage becomes plain and progressively more severe for populations with higher noise exposures.

The concept of an noise exposure Threshold Level is worthwhile, indeed necessary, but the $L_{EX,8h}$ value 75 dB(A) is unnecessarily restrictive. The scientific evidence suggests that the Threshold Level could be higher without sacrificing the aim of no noise-induced hearing loss, even in the most noise-susceptible portion of the population exposed for a working lifetime.

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The Control of Tonal Noise - A Structured Approach
A case study at an Aluminium Recovery Plant

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1. INTRODUCTION

This paper describes an investigation by SRL into the cause of tonal noise alleged to be emanating from an aluminium conversion plant in Wales. The case was politically very sensitive almost resulting in a court hearing. Consequently the name of the factory and other references are omitted from this paper.

SRL's investigation resulted as an appeal from the factory management against the extent of the remedial work that the Local Authority required, as a result of the report from their consultant.

2. BACKGROUND

The factory which was the subject of the Noise Abatement Notice is one which recycles scrap aluminium and recasts it into billets. The entrance for vehicles is at the font or east side of the factory and the rear or west side of the factory faces a residential estate some 200m distance. The factory building which is constructed largely of profiled steel cladding houses most of the plant including the furnaces. At the rear of factory, there is some external plant comprising a dust and fume extraction system and several cooler units.

For several years some of the residents of the estate had been complaining about their perception of unpleasant low frequency noise which appeared to emanate from the factory. The noise was identified as having tonal in nature. A previous study demonstrated that here were measurable tones at 12 Hz, 16 Hz and 38 Hz at the complainants properties. The notice served by the local Environmental Directorate demanded a reduction at 38 Hz or specifically "in the 40 Hz one-third octave band".

3. THE PLANT

The plant layout is shown in Figure 1. It shows the 4 melters and homogeniser cooler within the building and the coolers, caster coolers and fume extraction plant sited externally on the southwest side of the building.

This paper deals largely with the fume extraction and filtration system. It comprises three bag plants. Two of the bag plants, numbers 1 and 2 are served by centrifugal fans which discharge into a common stack. The third unit bag plant 3 was of a different, newer design and was shown not to contribute to the problem.

The filter bags are mounted at high level and are periodically shaken by mechanical excitation in the "Scavenger Area". This area is mounted above the plant. There are of course two cyclones which collect the larger articulate emissions. It is essential that the efficiency of the plant is unimpaired to satisfy the demands of the Factory Inspectorate for the employees and the Environmental Agency for the local residents.

4. REQUIREMENTS OF THE NOTICE

The Noise Abatement Notice required that the noise measured at the complainants was reduced to 48dB in the 40 Hz one-third octave band. This was based on the level described in the German Standard DIN 45680. The typical level existing at the complainants when the plant is operating was 56-57 L_{eq}dB.

Much discussion was had with the client, the Environmental Inspectorate and their consultant about the merits of using a German Standard when it could be argued that BS 4142 used in conjunction with BS 7445 part II would be appropriate. However, this aspect is being covered in a paper by Mike Hewitt of AVT at this conference.

In addition to specifying noise limits, the Notice contained a schedule of work split into two phases as follows:

Phase 1

- ☒ Acoustically treat the scavenger area on bagging plant 1 and 2
- ☒ Fit / replace inlet and exhaust silencers on the main fans of bagging plants 1 and 2 to attenuate low frequency noise
- ☒ Acoustically clad the fan casing
- ☒ Stiffen and clad the existing / replaced outlet silencer casing
- ☒ Fit silencer to the homogeniser cooler

Phase 2

- ☒ A secondary stack silencer shall be introduced
- ☒ A frequency tuned expansion box should be fitted inside the factory to control low frequency duct born noise from the melters
- ☒ The cyclones should be sealed and stiffened to prevent drumming.
- ☒ The platform which sits close to the cyclones should be isolated using anti-vibration mounts.

5. SRL INVESTIGATION

SRL was not convinced about the need to do all of the work set out in the schedule nor about the priority given to each item. A structured approach to analysing the noise source was adopted.

In particular, noise measurements were made under a number of different operating conditions to isolate the source of the low frequency noise from the offending areas. The two surveys we completed included the three bag plants, No 5 stack, the roof area and the ductwork from melters 3 and 4 to the bagging plant. We also measured noise from the homogeniser cooler fans on the roof and on the caster 2 cooling tower. DAT recordings were made for subsequent post analysis using a laboratory-based narrow band analyser. Some vibration measurements on the cyclones and the scavenger area building were also taken with particular attention being given to vibration in the 40 Hz region.

An aerial platform was used to gain access to the top of the bagging plant stacks and the high level extraction ductwork.

6. SUMMARY OF FINDINGS

6.1 NUMBER 1 AND 2 BAG PLANTS

Both plants together - peaks at 38 Hz and 34 Hz. With plant 2 off, the peak at 38 Hz remained, regardless of which melter fume extract was diverted through plant 2.

6.2 BAG PLANT 3 STACK

No discrete tone in the 40 Hz one-third octave band and an L_p of 10 dB less than that for bag plant 2 at 40 Hz.

6.3 DUCTWORK FROM MELTERS 3 AND 4

Close field measurements made of the ductwork linking bag plant 1 to the melters showed significant levels at 40 Hz only close to fan and the closed off section of ductwork. Thus we concluded that the noise radiated from the ductwork at 40 Hz was due to the fan and not low frequency transfer from the melters.

6.4 ROOF

No significant radiation in the 40 Hz one-third octave was detected over areas of the roof.

6.5 STACK FOR MELTERS

Measurements at 10m distance from the stack showed no tonal component at 40 Hz.

6.6 BAG PLANTS 1 AND 2 - MAIN DUCTWORK FAN TO BAGGING PLANT

Clear tones at 38 Hz were observed on the ductwork which runs from the fans to the bagging plant itself.

6.7 SCAVENGER AREA

Readings were taken in the Scavenger Area, on the Scavenger fans and on the steel cladding forming the scavenger enclosure. Narrow band analysis of the readings without the fans running showed a significant component of 15-25 Hz with a very small peak at 38 Hz. With the scavenger fans running the peak at 38 Hz did not increase. Clearly the scavenger fans were not contributing to the 40 Hz one-third octave band level.

6.8 CYCLONE BAGGING PLANTS 1 AND 2

Measurements taken at the top of cyclones on bagging plants 1 and 2 showed a clear peak at 32 Hz but not at the 40 Hz one-third octave band. On top of the cyclone for bagging plant 2 a peak at 30 Hz was discovered. The cyclones are not a significant source of noise in the 40 Hz one-third octave band.

6.9 BAG PLANT 3 FAN

Although a clear tone could be heard emanating close to the fan on bag plant 3, measurements shown occurred in the 50 Hz one-third octave band with a lesser peak at 38 Hz which most likely originated from the nearby ductwork from bagging plant 1 and 2.

6.10 HOMOGENISER FAN AREA

Measurements made over the inlet and exhaust opening of the homogeniser fans revealed a broad band noise spectrum only with no obvious tonal components. It was concluded that although the fans contribute to the overall noise level produced by the plant they were not responsible for the tonal component which was the subject of the Notice.

6.11 CASTER 2 COOLING TOWER

A clear tonal component at 38 Hz was determined at the base of the Caster 2 cooling tower. However, it had been observed that complaints occurred whether or not the Caster 2 cooling tower was in operation.

7. NOISE LEVEL PREDICTIONS

Having performed the measurements set out in 6 above, SRL established the sound power level of the individual plant items in one-third octaves and predicted the sound pressure level in the 40 Hz one-third octave band which would have occurred at the claimants property as follows:

PLANT ITEM	PREDICTED LP IN 40 Hz ONE-THIRD OCTAVE BAND
Bag Plants 1 & 2 Stack	49
Bag Plant 1 Main Ductwork	49
Bag Plant 2 Main Ductwork	53
Caster 2 Cooler Main Body	45
Caster 2 Cooler Fans	41
Total at Complainants Property	*56*

Thus the predicted sound pressure level at the complainants property in the 40 Hz one-third octave band is very similar to that measured in practice.

8. DISCUSSION

One of the problems experienced throughout this project was that the fans operating by Plants 1 and 2 could not be switched off. It was known at the outset that the rotational speed of the centrifugal fan on No 1 Bag Plant was 2280 rpm which of course coincides with a frequency of 38 Hz. We had been assured by the factory management that the fan had been regularly balanced and was in good order. The fan was belt driven so the prospect of misalignment being a cause of unacceptable vibration at the rotational frequency was unlikely. We were left with the possible conclusion that the 38 Hz tone was due to a strange effect in the ductwork. A closed off section of ductwork having a length of 4.5 metres was, coincidentally, approximately 1/2 a wavelength long of noise at 38 Hz.

As a result of the measurements we were able to provide a revised scope of work which differed considerably from the schedule of works included in the notice.

First of all our measurements had determined that the scavenger area on the bagging plants was not a cause of noise in the 40 Hz one-third octave band so this item was eliminated.

Fitting or replacing inlet or exhaust silencers on the main fans on bagging Plants 1 and 2 was physically not possible because of space restrictions and implications of additional pressure drop. The additional pressure drop could not have been tolerated either by the Factory Inspectorate or the Environmental Agency. Acoustically cladding the fan casing was an option but for the fan casing to be adequately clad it would have required cladding of some 400 mm thickness. We believed that stiffening of the ductwork was a good first step and this work was implemented as is shown in Figure 3.

We were able to avoid the need to fit a silencer to the homogeniser cooler as the factory management made a commitment not to use the unit.

Thus all of the items suggested in the first phase of the schedule of works were considered not to be necessary except perhaps stiffening of the ductwork.

The second phase required that a secondary stack silencer should be introduced. We believe that this measure should be the first priority as it would have the effect of reducing noise emanating from the exhaust stack of bagging plants 1 and 2 albeit the reduction being broad band in nature. Consequently a silencer in the form of a 4 m x 300 mm diameter pod was inserted into the stack and a reduction of 5 dB at 38 Hz plus a broad band reduction in noise was obtained. There was however still a much reduced but noticeable tonal component in the 40 Hz one-third octave band which was found to be radiating from the stiffened ductwork.

As this stage of the exercise it was possible to shutdown the bagging plant and we were able to inspect the rotor of the fan on bagging plant 1. To our surprise and delight we found that the fan rotor had been repaired rather crudely and the blades of the rotor were very asymmetrical. Although a fan rotor may be in balance, if it is asymmetrical one naturally would expect that a tone at the rotational frequency would be generated. In finding this we advised the client to install a new fan with a symmetrical, well balanced rotor. This was done and the fan was chosen so that it had a different rotational speed to the original one but was still able to perform the same duty.

9. RESULT

The result of changing the fan to the new type, installing the stack silencer and stiffening the ductwork was such that the noise level in the 40 Hz one-third octave band was reduced to 46 dB thus achieving the Environmental Directorates target even though there had been some dispute about the standard to be used. It is important to remember that this was achieved at a fraction of the cost that would have been involved had the client followed the schedule of works recommended by the Environmental Director Consultant. The message is that we as consultants should never adopt the "broad brush" approach to a noise issue. By careful investigation and analysis problems can be solved without incurring unnecessary expense.

Finally, figure 4 shows the one-third octave band spectrum at the complainants property before and after the implementation of the few noise control measures described above.

FIGURE 1 - SITE PLAN OF FACTORY UNDER INVESTIGATION

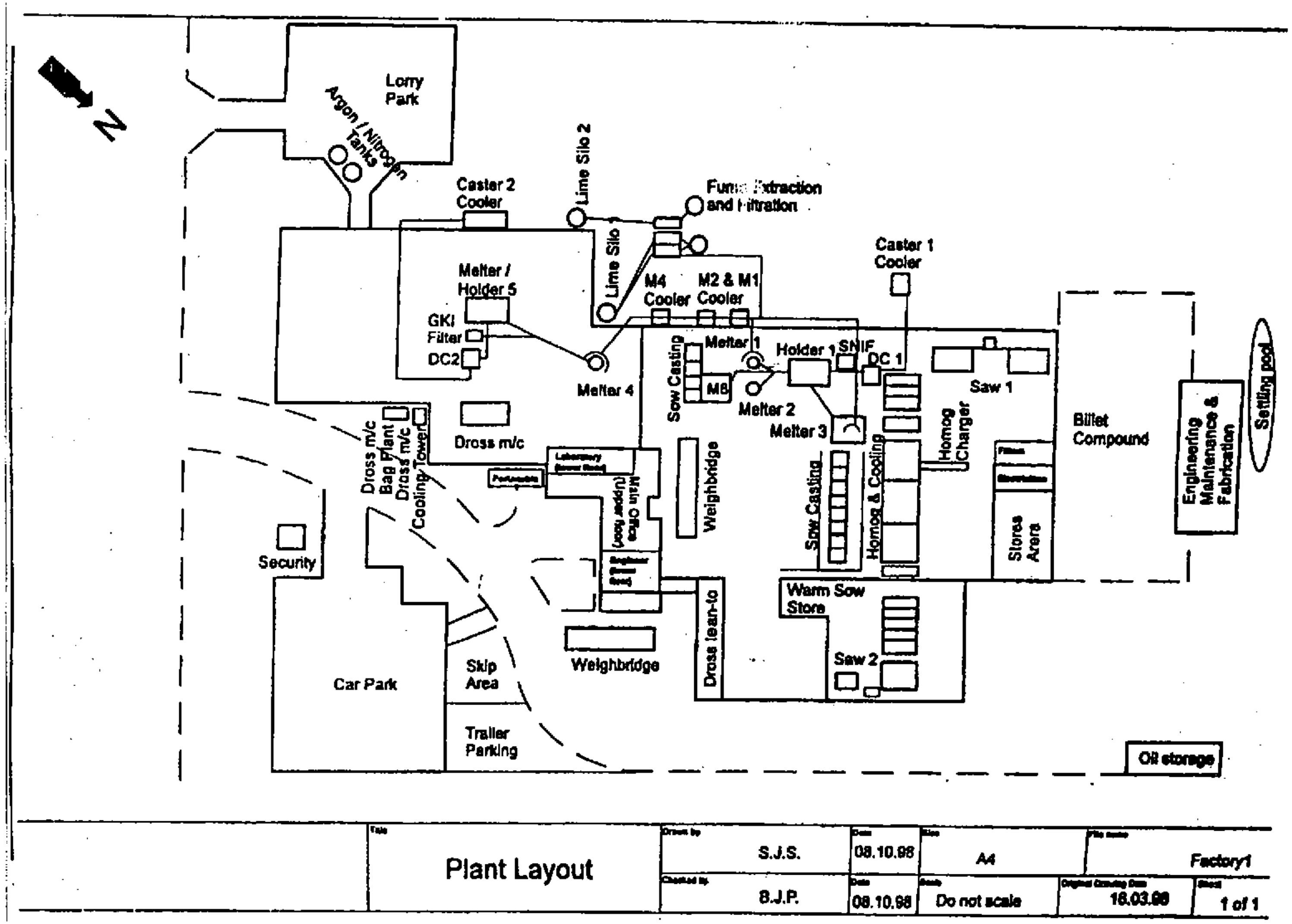
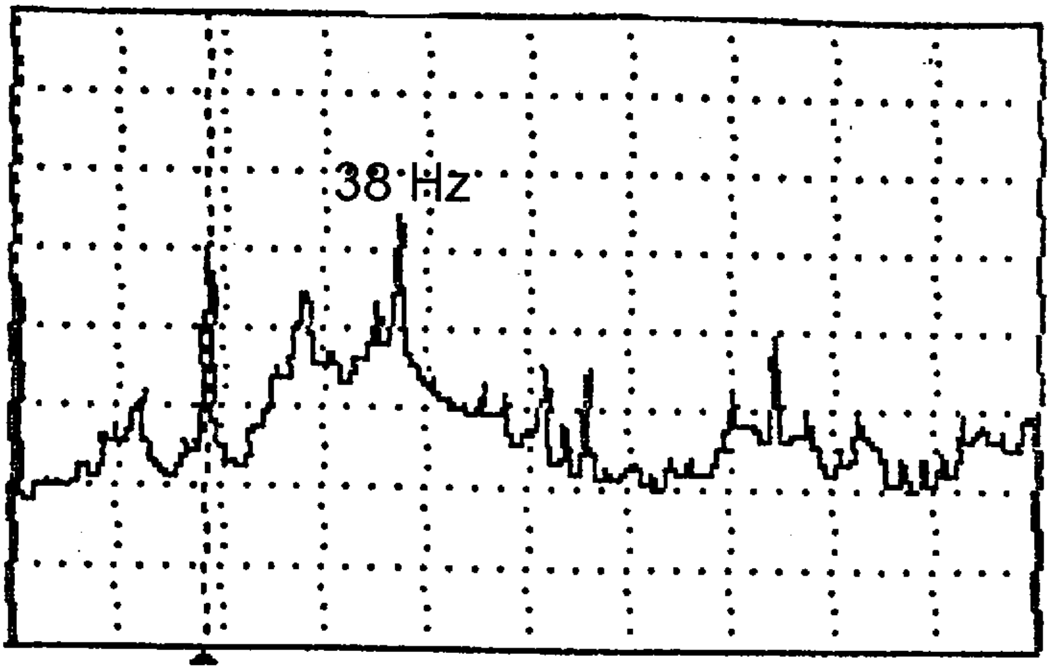


FIGURE 2 – COMPARISON OF PLANT AND BOUNDARY NARROW BAND ANALYSIS RESULTS

Narrow band analysis of Bagging Plant 1 (fan casing)



Narrow band analysis of levels 35m from boundary fence

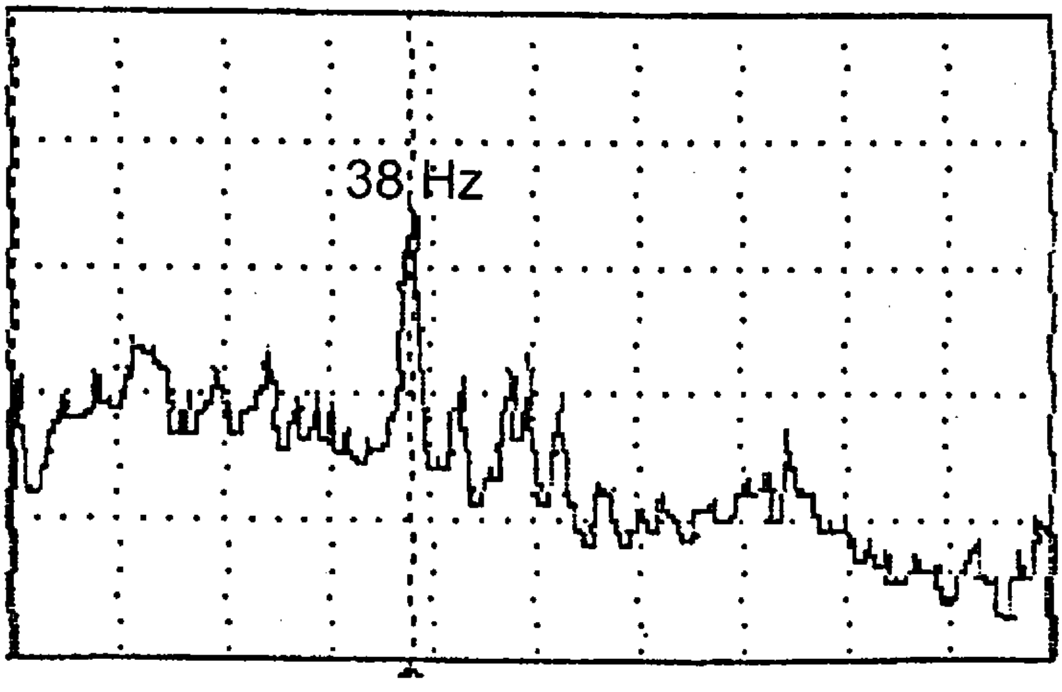


FIGURE 3 – INITIAL NOISE CONTROL MEASURES : DUCT STIFFENING

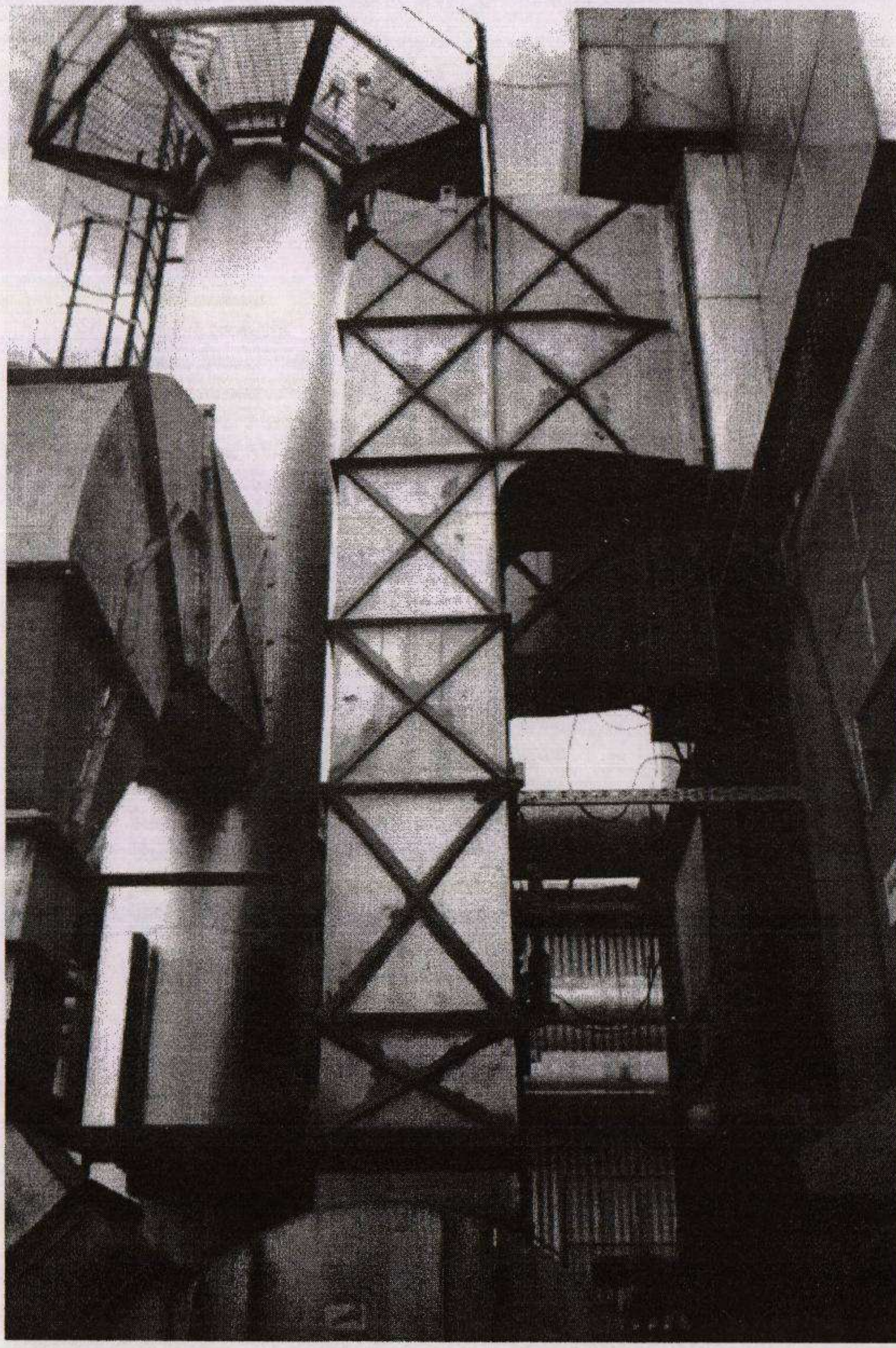
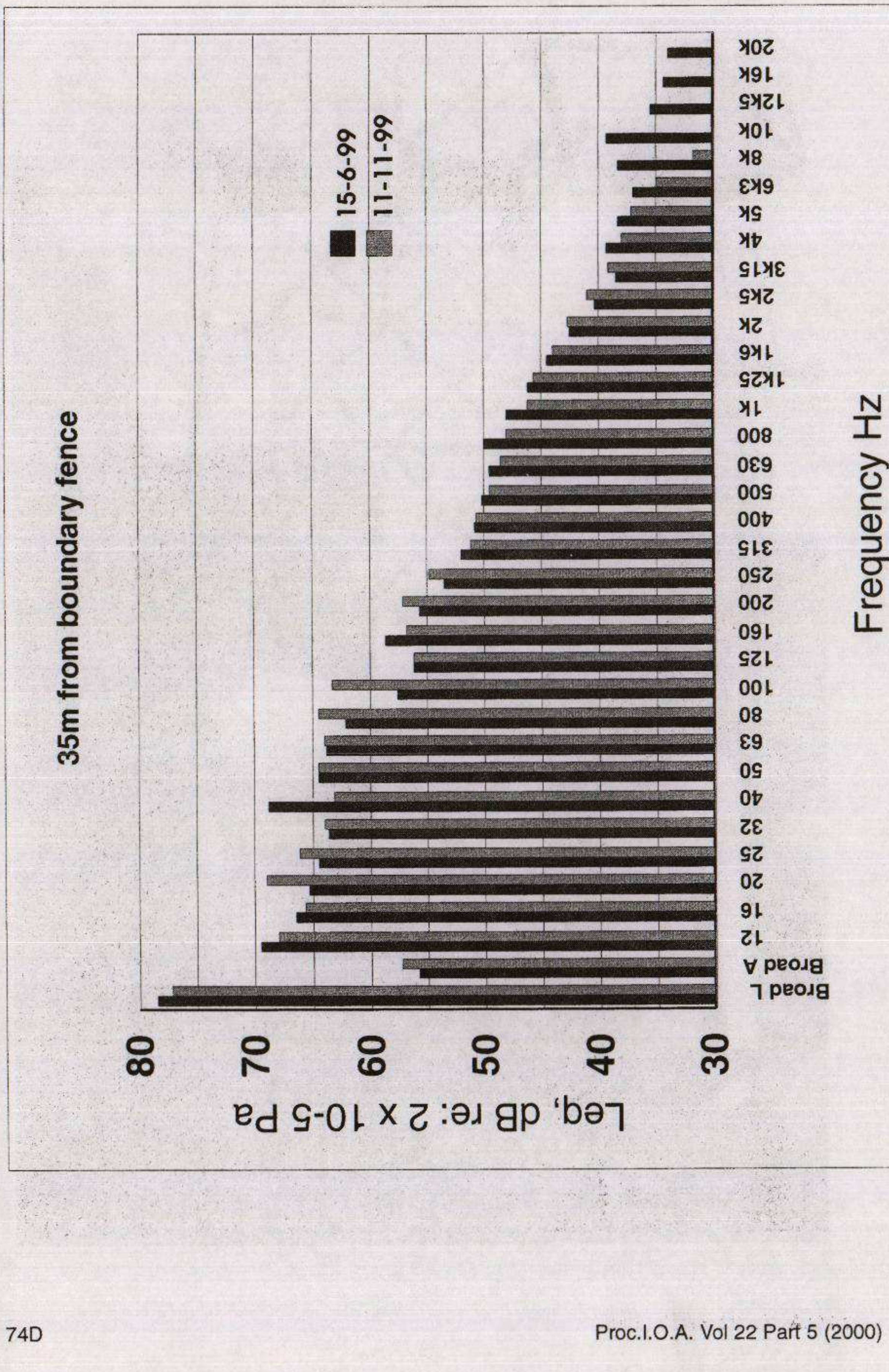


FIGURE 4 - COMPARISON OF THIRD OCTAVE BAND SPECTRUM BEFORE AND AFTER CONTROL MEASURES



DESIGN OF A SKID-MOUNTED GAS TURBINE ENCLOSURE

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1. BACKGROUND

The growing demand for a quieter environment has led governments to create legislative framework that reduces excessive noise exposure on people working in close proximity to machines such as gas turbines. Power plant developers respond to this growing environmental concern and have to comply with noise control guidelines and more demanding legislation by implementing various noise control measures [1]. The gas turbine casing is the major noise contributor within a gas turbine plant. Skid-mounted acoustic enclosures have many advantages over free-standing enclosure and are widely used to control gas turbine casing noise. Current research trends show an increased awareness on transmission of sound through partitions [2]. Little has been done to advance cost-effective simple solutions based on sound design principles. The traditional noise control attitude taken toward enclosure design is "build a box with absorptive lining around the machine". Production and operational problems inevitably arise when this lack of engineering concern for enclosure design is taken. This paper presents the design challenges that were faced when designing a high performance skid-mounted acoustic enclosure for a GE Frame 6 gas turbine (rated at 42.1MW).

2. DESIGN PROCESS

2.1 Design goals

The design process used is summarised in table 1. In order to clarify what was wanted from this design, we interviewed many people including sales and marketing staff, potential customers, commissioning and maintenance engineers. The result of this survey identified the following motivations driving the desire for a new enclosure design.

- Improved acoustic performance
- Compact
- Cost effective
- In compliance with the relevant health and safety requirements
- Inexpensive to manufacture
- Easy to install
- Easy for shipment
- Easy access for inspection and maintenance
- Resistance to various environmental conditions

2.2 Functional Specification

Having established customers and other relevant parties requirements, a general functional specification including topics such as noise targets, accessibility, fire rating, ventilation flow, structural support loading, lighting, temperature, material, environmental conditions, overall maximum dimensions, constructability, hazardous area classification, health and safety requirements was prepared. The specification consists of parameters which defined in a measurable way the design requirements. It provides the basis for evaluating and controlling the design. There are various ways in defining the noise reduction goals that were to be achieved by the new enclosure. The most common way being to specify the Insertion Loss (IL) of the enclosure determined as the difference between the noise level produced by the un-enclosed machine at the specified location and the sound pressure level with the enclosure fitted. However, experience has shown that the acoustic performance of an enclosure may behave differently depending on its mounting arrangement and the characteristic of the noise source. Consequently, an insertion loss target of 35 dB(A) was imposed. With the gas turbine operating at base load, the following noise limits were selected :

- an average sound pressure level limit of 80 dB(A) at 1m – (dB re 20 micropascal),
- a maximum sound pressure level limit of 85 dB(A) at 1m – (dB re 20 micropascal),
- a sound power level limit of 100 dB(A) – (dB re. 1 pico Watt).

Assessment was based on measurement being carried out in accordance with the relevant company and ISO standards and procedures.

2.3 Site investigation

The first technical task was to collect basic data on a gas turbine package with a standard skid-mounted acoustic enclosure. Sound pressure level measurements supplemented by intensity and vibration measurement were undertaken both inside and outside the enclosure. Sound intensity measurement enables the sources of noise to be located so that the relative contributions of noise from various parts of the gas turbine unit could be evaluated. Vibration measurement serves a similar purpose. From the knowledge of the radiation characteristics of the unit, the dominant sources and transmission paths of noise were identified. The result of our measurement reveals that both the vibration level of the enclosure wall and the radiated noise consists of a strong tonal component at the 2.5 kHz band. It was identified to be the 1st stage compressor blade passing frequency. The measured sound pressure level was about 10 dB(A) above our design target. The rather poor performance of the enclosure was mainly due to structural borne paths whereby vibrations were transmitted from the gas turbine to the enclosure walls. The result confirmed that unless the structural borne noise was adequately attenuated, the use of expensive material and complex construction for the enclosure panels would not noticeably improve the overall performance of the enclosure.

2.4 Cost model

With increasing competition in the markets, there is greater emphasis on cost reduction. One of the main objectives of the design exercise was to design for minimum total unit cost within the required specification. A realistic cost target was established at the outset of the design exercise. The first estimate of manufacturing costs was completed by drafting a list of all the components and using the purchase price or fabrication cost for each part. The cost estimate was used as a performance model, but instead of predicting the value of technical performance, it predicted cost performance. The cost estimate remained useful throughout the design process and was updated regularly to monitor actual costs against the current status of the estimated cost. Based on the cost model, the team eliminated various costly design options throughout the design process.

2.5 Panels design

Different composite materials for the enclosure panel were considered during the initial design stage. Results of our feasibility study indicated that steel was the most economical material to meet our functional specification. The transmission loss (TL) is a measure of the sound power attenuation of an item when tested under laboratory conditions. It is important to determine the right combination of transmission loss of enclosure panel and interior sound absorbing treatment required to achieve the specified overall performance. TL levels of conventional panel constructions at the mass controlled region can be predicted fairly accurately by classical theory. A number of resonances are set up when a finite panel is subjected to the excitation of bending waves caused by the approaching sound wave. Normally, the first resonance frequency is below the audio range and the second resonance occurs when the wavelength of oncoming sound is equal to the length of the bending wave in the panel, at what is known as the "Coincidence frequency". In the coincidence region the transmission loss is limited by the damping of the material. This deficiency can be minimised by laminating different thickness of material combined with various layers of absorption materials. Laboratory test was carried out to identify a cost-effective combination. Based on our study, additional damping material on the enclosure wall panels did not result in cost effective improved performance [3].

The actual installed performance is usually less than the test data because it is not possible to eliminate all structural borne noise via flanking paths. Flanking occurs through enclosure structural supports. This was to become more of a challenge because one of the requirements in the functional specification is the enclosure need to provide primary structural support for the gas turbine inlet ductwork. To minimise the effect of structural borne noise, all the possible flanking paths were identified and appropriate isolation was introduced.

2.6 Other design considerations

Accessibility is one of the criteria in our specification. Adequate space and openings are required for maintenance work. Consequently, the panels on the sides of the enclosure were designed to be de-mountable. When major overhaul of the engine is required, the entire length of the engine is accessible. Access into the turbine compartment for routine and preventive maintenance is provided by standard acoustic doors built into the panels. Additional panels exist along the base for suppressing breakout noise from the gas turbine base.

Each component used in an enclosure (e.g. wall panels, acoustic doors, roof panels, window, ventilation exhaust and inlet system) may have different transmission losses. The overall acoustic performance of an enclosure, therefore, results from the interaction of many components with different transmission losses. The acoustic performance and size of the weakest component, therefore, determines the overall attenuation achieved. The down rating in overall performance being normally caused by a lack of attention to detail engineering such as door seals, pipe penetration design etc. The loss of acoustic performance would only require a small gap on the enclosure wall. Particular attention was given to all the potential weak paths.

The latest Health and Safety requirements have been taken into account for the new enclosure design including the re-location of instruments and gauge panel for full external access thereby removing the need for operating personnel to enter the enclosure while the gas turbine is running. The construction of the turbine gauge and instrument panel along with the control and power supply junction boxes was modified to offer similar attenuation as the rest of the panel system. This arrangement allowed improved access within the enclosure for other mechanical equipment while also giving a more compact enclosure space envelope without reducing acoustic performance.

As a result of an incident which occurred at Teeside combined-cycle gas turbine power plant in June 1996, the health and safety inspectorate raised concerns about the safety of gas turbine enclosures. The large number of flanges and joints, combined with high pressures, presents an explosion hazard within the enclosure in the event of fuel leaks, in gas turbine plants. The Health and Safety Executive (HSE) recently issued a guidance note [4] covering a range of health and safety issues for gas turbine enclosure. To minimise the risk of explosion and the extent of any possible damage, following design measures which conform to the HSE's recommendations were introduced in the design.

- Ensure the ventilation flow is adequate to dilute any possible gas leaks by computational fluid dynamics simulation study [5].
- The provision of high performance gas leakage detectors.
- Provide a positive airflow to push leak gas toward the detectors.
- Carry out smoke tests to confirm there were no dead spots in the enclosure.

The new enclosure was designed recognising all of these requirements. Figure 1 shows an isometric view of the new acoustic enclosure.

2.7 Site measurement

In order to establish and prove the basic acoustic design of the new enclosure, a program of site noise measurements were undertaken which included surveys of similar gas turbines fitted with the previous design of skid-mounted enclosure, free-standing enclosure and finally the new high performance skid-mounted enclosure. This testing included near field sound pressure level, sound intensity level and panel vibration measurements with the gas turbines operating at similar loads. The measurements taken from the first new skid-mounted enclosure shows that it meets its functional specification and the specified limits [6]. To gauge the effectiveness of this gas turbine enclosure design, the sound power levels of the new enclosure was compared with the other previous skid-mounted enclosure and free-standing enclosure design results. As can be seen in figure 2, the sound power level of the new enclosure is significantly lower than the previous enclosure and its performance is very similar to a free standing enclosure. Figure 3 shows a comparison of the panel vibration levels between the new design and a previous design. The vibration level difference between the old enclosure wall and the new enclosure wall is well over 35 dB at the critical frequency (i.e. 2.5kHz band). This represents a significant improvement on controlling the structural borne noise.

3. CONCLUSIONS

Full awareness of the commercial aims of the new enclosure and the constraints imposed by the manufacturing processes proved to be invaluable to the overall success of the design exercise. With the help of the cost model, the cost target of the enclosure was met with a comfortable margin. Acoustic enclosures can be designed that are operator friendly, allowing easy access and are cost effective by careful thought, attention to details of all engineering disciplines and above all, having a good understanding of the noise source and its transmission characteristics.

4. REFERENCES

[1] Noise Control for Gas Turbine Plant - E. Fung , Euro-Noise 98 Conferenece,
[2] Sound and Structural Vibration, Radiation, Transmission and Response – Frank Fahy, Chapt. 4
[3] Investigation into the effect of visco-elastic damping treatment on gas turbine enclosure panels (Internal Report re. AT-1365)

[4] Guidance Note PM 84 : Control of safety risks as gas turbines used for power generation – Health & Safety Executive
[5] CFD Analysis of Frame 6 gas turbine enclosure fuel gas leak hazard (Internal Report re. FS/4392/1)
[6] Noise Assessment of the New Gas Turbine On-base Enclosure (Internal Report re. DDM424)

Phase	Mission	Tasks
1	Establish design goals	• Collect site data. • Establish target specification. • Identify resources required. • Establish design programme.
2	Generate product design concept	• Outline design. • Choose material. • Define major subsystems and interfaces.
3	Cost modelling	• Establish cost (including material cost, fabrication cost and fitting cost) of all the components. • Establish engineering man-hours and resources.
4	Detail design and manufacturing	• Refine design concepts. • Design for easy manufacture. • Detail design.
5	Testing and refinement	• Field testing. • Implement design changes.

Table 1. Design Process

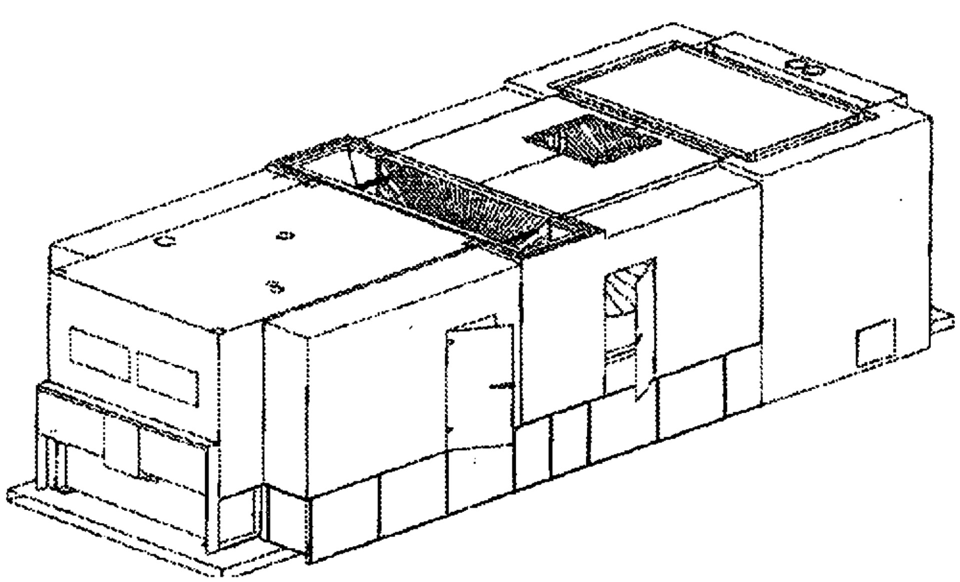

Figure 1 : Isometric view of the gas turbine enclosure

Fig. 2 - Comparison of the enclosure sound power levels

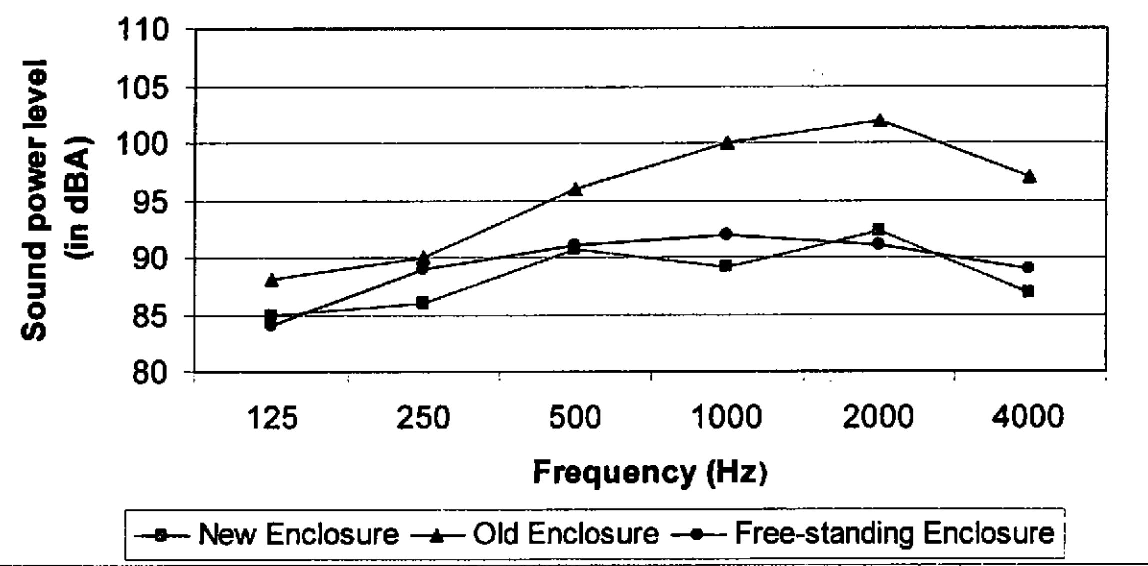

Figure 3 - Comparison of the vibration levels