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Noise Characteristics of Pumps at Tehran's Oil Refinery and Control Module Design

R. Golmohammadi[a], M. R. Monazzam[b]*, M. Nourollahi[a] and A. Nezafat[a]
[a]Occ. Hyg. Deptt., School of Public Health and Center for Health Research, Hamadan University of Medical Sciences, P.O. Box 4171, Hamadan, Iran
[b]Occ. Hyg. Deptt., School of Public Health, Tehran University of Medical Sciences, Tehran, Iran

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Abstract. Considering industrial excessive noise exposure, sound pressure level of 4 pumps with different applications installed in Isomax Unit of Oil Refinery Centre of Tehran, Iran was studied. The A-weighted sound pressure level and maximum sound pressure level showed that the emitted noise is far above the permissible limits. Installing enclosure around the noise source was found to be the best noise control measure. Results of operational calculating transmission loss of the designed module with a sandwich layer showed that it is possible to provide 19.7 dB (A) reduction in overall sound pressure level and 20 dB reduction in dominant frequency. Designing the module with given specifications and probable leak estimation and prevention gives remarkable and effective results in the studied field.

Keywords: pump noise, noise pollution, noise exposure

Introduction

Exposure to excessive noise is associated with high risk of hearing loss (Golmohammadi, 2007; Esmaeelzadeh *et al*., 2006). National Institute for Occupational Safety and Health (NIOSH) categorized hearing loss as one of the ten most important work related illnesses and estimated that 25% of workers of more than 55 years, who were exposed to excessive noise (higher than 90 dB) suffered from different levels of hearing loss. Workers in petrochemical industries also suffered from noise exposure problems (Cheremisinoff Paul and Allen Ernest, 1977).

An investigation for evaluation of noise pollution in oil refinery fields in Iran was undertaken by Nassiri and Ahmadi (2004). It was found that exposure of workers to noise, in most cases, was far above the permissible limits provided by American Conference of Governmental Industrial Hygienists (ACGIH).

In another study of Iranian petrochemical industry, it was reported that the noise level of the studied sources was so high that the exposed workers could hardly work in that condition and in all cases control measures were required (Gholshah, 1997).

Reduction of industrial and environmental noise pollution has been the subject of many different studies. (Monazzam and Nassiri, 2009; Monazzam and Lam, 2008). Hansen (2005) in a study demonstrated that applying control methods, such as installing enclosures, effectively reduced noise to considerable levels. In another study, Joseph *et al*. (1991) found that ignoring the structural path that caused sound leak from module reduced the effectiveness of the control measure. It was also reported that the exact recognition of noise source and surface specifications surrounding the module, plays crucial role in precise acoustic efficiency assessment. The results demonstrated that the application of a module, with complex layers proportionate to the design requirement of the module, produced 12 to 19 dB reduction in the noise level (Min and Ying, 2008). Another study of noise control showed that use of a multiporous enclosure reduced the sound pressure level by more than 40 dB. Hakimi *et al*. (2006) estimated a 20 dB reduction in the sound level by applying a module in the air outlet.

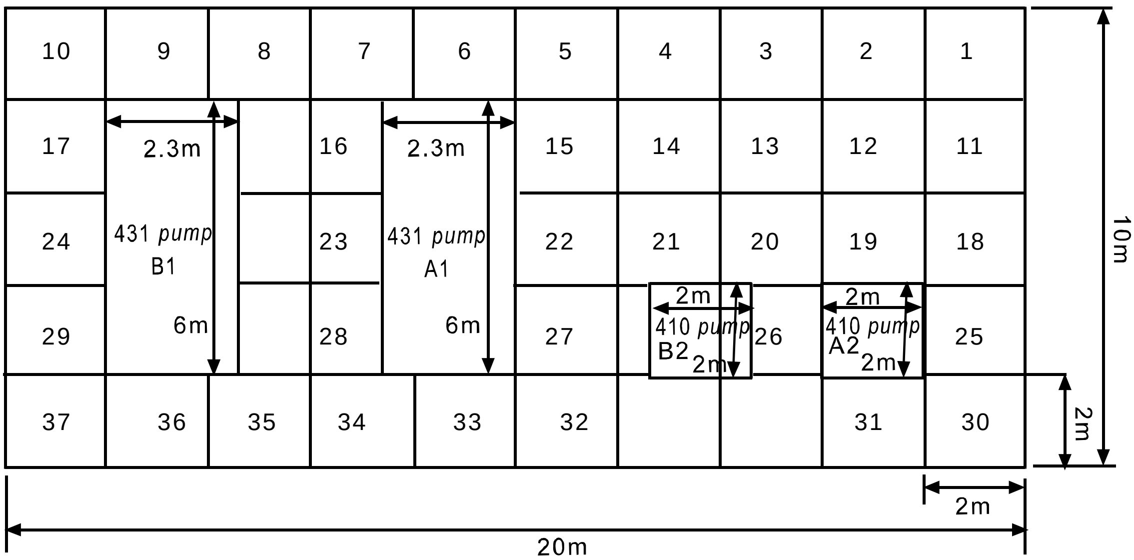
In this paper, results of an extensive theoretical and experimental study on noise propagation character of pumps along with application of control measures in a unit of Tehran oil refinery are presented.

Materials and Methods

A study for sand pressure levels was conducted at Isomax unit of the Oil Refinery Centre of Tehran (ORCT), that had 4 pumps for different applications. Field measurements and calculations were carried out for evaluation and prediction of noise pollution at the site along with the characterization of the noise sources.

Field measurements. The study field comprised of a 20x10 m open site having 4 pumps installed on a rigid floor. Fig. 1 presents a simple plan showing type and location of the pumps

*Author for correspondence; E-mail: mmonazzam@hotmail.com



Power of motors: A_1 = 200 KW: A_2 = 75 KW; B_1 = 55 KW; B_2 = 230 KW

Fig. 1. Sound plan of Isomax pumps.

and 37 measurement points. The workers at this site were carrying out different tasks. Motors of power 200, 75, 55 and 230 KW were distributed over the site, which are labelled as A1, A2, B1 and B2, respectively. There were no control measures for the noise sources at the site.

For field measurement, first, the A-weighted sound pressure level ($L_{p(rms)}$) was measured using calibrated TES-1385 sound level meter; then the maximum sound pressure level ($L_{p(max)}$) at 37 zones of 2x2 meter was measured according to the lattice method. Finally, the crest factor was determined by calculating the difference between $L_{p(max)}$ and $L_{p(rms)}$. For the measurement of sound, standard method ISO 9612 (1997) was followed.

Sound frequency analysis and evaluation were carried out using analyzer sound level meter TES-1385 and standard calibrator B&K 4231 was applied for calibrating the device. The sound level meter time constant was set on slow mode; microphone position was set at 1.5 m above the ground, pointing in the direction of the workers.

Calculations. Applying RPM meter (RPM indicator Model No.RM-20), the RPM of pumps was specified and using the following equation, dominant frequency (f) of the sources was predicted.

$$f = \frac{N_b \text{ x RPM}}{60} \tag{1}$$

where N_b is the number of pump blades.

Results and Discussion

The A-weighted sound pressure level and maximum sound pressure level along with their relevant crest factor in Isomax unit of Tehran Oil Refinery Centre are shown in Table 1.

Using surfer software, a plan for the sound field was drawn (Fig. 2). It can be seen that the highest sound pressure levels are located in the mid-field where two pumps with the highest noise pressure levels (A1 and B2) are placed side by side.

Table 1. A-weighted sound pressure level $Lp_{(rms)}$, maximum sound pressure level $Lp_{(max)}$ and crest factor (CF) of Isomax pumps area

Station	$Lp_{(rms)}$ (dB)	$Lp_{(max)}$ (dB)	CF	Station	$Lp_{(rms)}$ (dB)	$Lp_{(max)}$ (dB)	CF
1	87.8	104	16.2	20	93.1	109	16.9
2	87.7	101.9	14.2	21	94	108	16.7
3	88.5	106.2	17.7	22	94.5	112	17.9
4	88.4	106	17.6	23	93.7	112	18.3
5	88.6	105	16.4	24	87.8	105	17.2
6	88	105.6	17.6	25	90.2	105.6	15.4
7	87.5	104	16.5	26	92.3	108	15.7
8	88.2	105.8	17.6	27	94.6	111	17
9	87.5	103.6	16.1	28	92.5	109	17
10	85.8	101.1	16.3	29	86.4	102	15.6
11	88.5	106.2	17.7	30	88.3	105	16.7
12	89.8	107.7	17.9	31	88.8	106	17.7
13	90	106	16	32	92.3	108	15.7
14	89.5	105.8	16.3	33	92.6	108.4	15.8
15	91	109.2	18.2	34	90.2	106	15.8
16	90.4	106	15.6	35	85.5	105.7	16.2
17	85.5	102.6	17.1	36	88.4	104.5	16.1
18	89.7	107	17.3	37	88.2	103.1	16.9
19	90	106.5	16.5				

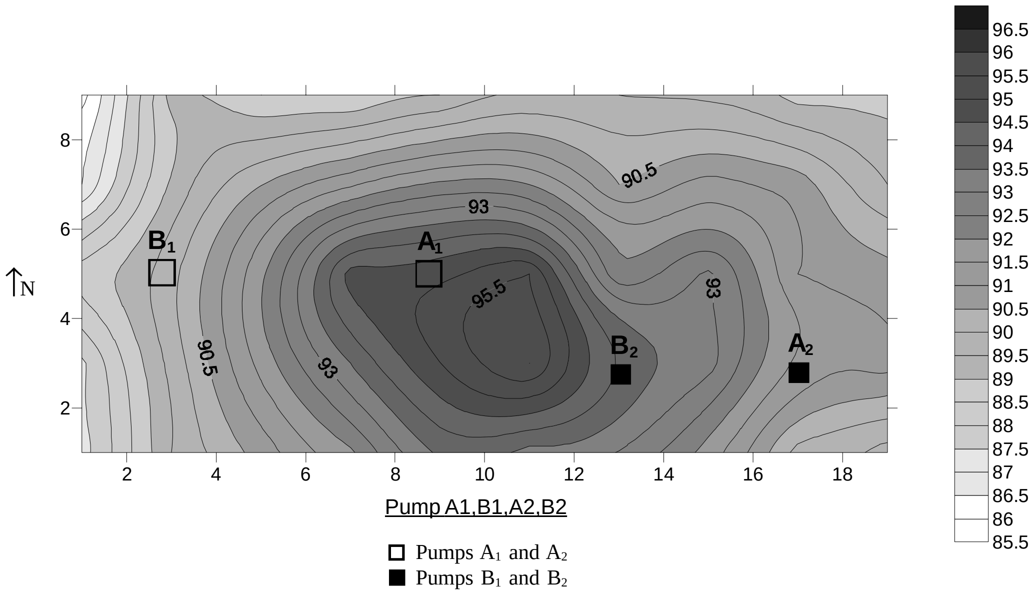


Fig. 2. Noise contours around pump positions.

It may be noted that at all stations, the sound pressure level is well above the standard level (85 dBA). The frequency analysis was carried out in octave band of 63 to 8000 Hz by dividing the site into four station; each station having one pump and other devices at the specified and measurable distance (Fig. 1). The results are given in Table 2.

Table 2. The octave band frequency analysis and dominant frequency of the studied noise sources

Measuring station frequency (Hz)	Station 9	19	21	23
63	83.2	87.3	83.2	81.7
125	77.1	82.4	81.0	84.2
250	76.7	86.6	83.0	81.5
500	77.4	87.4	80.8	86.2
1000	82.8	88.0	87.0	85.0
2000	80.4	89.8	89.6	87.7
4000	83.9	84.0	87.8	89.6
8000	78.6	77.4	84.0	81.3

Prediction procedure. In order to calculate the dominant frequencies of the pumps, their rotation speed was measured using RPM meter. Fluctuations in frequency of devices around the pumps were also accounted for. Rotation speed of pumps A2 and B2 was 2935 and 3000, respectively, and of pumps A1 and B1, 4500 and 5600, respectively. Applying equation (1), the dominant frequency of each source in the studied field was calculated.

The number of blades of pumps A2 and B2 is 50 and so the dominant frequency of these pumps is between 2445 to 2500 Hz; number of blades of pump A1 and B1 is 49 and so the dominant frequency of these pumps ranges between 3675 to 4573 Hz. So, the dominant frequency of pumps with index 2 (A2, B2) is found to be in 2000 Hz and that of the pumps with index 1 (A1, B1) is predicted to be in the range 4000 Hz in octave band scale.

By comparison of the field measurement and the estimation approach, a perfect agreement between these two methods is found. In this case the dominant frequency of noise in stations 9 and 23, which are adjacent to pumps with index 1, is 4000 Hz and the dominant frequency of noise in stations 19 and 21, which are close to pumps with index 2, is 2000 Hz. These results uphold precision of the field measurement.

Control module. *Enclosure design.* For designing enclosure, it is important to determine the critical frequency of the main insulator, (2 mm steel). By applying the well known equation 6.17 for calculating the critical frequency (Lewis and Douglas, 1994), its frequency is predicted to be 8978 Hz which is far above the dominant frequency of our main noise source.

Layout and specification of the module sandwich layers. ***Absorbent.*** In the design, a layer of slag wool with 2.5 kg/m^2 surface density and 25 mm thickness was applied as an absorbent for the considered frequency. Reflective surfaces around the noise source increase the sound pressure level due to multiple reflection of sound. So, applying an absorbent for the sound, particularly in the dominant frequency range, is one of the principal actions in the module and the enclosure design. In this study, slag wool is applied as the appropriate absorbent.

Frame. The sandwich panel was fixed by a wooden frame of 15 mm thickness and surface density of 7 kg/m^2.

Insulator. For insulating the structure - borne noise passing from the panel, 2 mm steel with surface density of 17 kg/m^2 was applied in the centre-line of the panels. Considering the dominant frequency of pumps in the range 2000 to 4000 Hz, minimum surface density of the insulator should be 12 kg/m^2 (Lewis and Douglas, 1994).

External surfaces. For prevention of sound reflection from the external surface of the modules, chipboard of 9 mm thickness and 7 kg/m^2, surface density was applied on the external surface of the module.

Door. A common gash door of dimensions 1.8 x 0.7 m, 43 mm thickness and surface density of 9 kg/m^2 was used for the entrance of the enclosure.

Windows. For giving perfect view of the pumps to the operators, three windows of dimension 1x1 m were designed which were vacuumed, double glazed with 9 mm thickness and 7 kg/m^2, surface density.

The dimension of the designed enclosure was 3×3×3 m. By the application of sandwich layers and using the following equation, the overall panel surface density ($\bar{w}$) of the enclosure was found to be 16.25 kg/m^2:

$$\bar{w} = \Sigma\, w_i \times s_i \,/\, \Sigma\, s_i \quad (2)$$

where w_i and s_i are, the surface density (kg/m^2) and the area (m^2) respectively, of each panel component (Fig. 3).

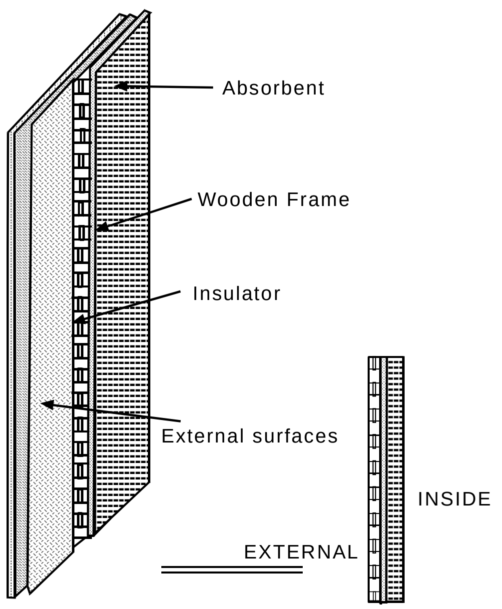


Fig. 3. Detailed structure of the main enclosure panel.

Frequency analysis, TL and NR calculation. Using the above mentioned field measurement results (95 dB (A)) and Iranian noise exposure limit (85 dB (A)), it is easily found that the total noise reduction required is 15 dB (95 dB (A) - 85 dB (A) +5 dB (A)); 5 dB (A) is added to arrive at the practical results. The noise reduction level for dominant frequency (according to the above method) was found to be 20 dB (A).

Total noise reduction achieved by installing the designed enclosure is calculated to be 19.7 dB (by the difference of total outdoor and indoor noise levels). In this case, the overall noise level inside the enclosure was measured to be 95 dB while outside, it was estimated to be 75.3 dB. Fig. 4 provides sound pressure level variations before and after installing the enclosure in octave band centre frequencies.

The architectural plans and related details are designed by AutoCAD software and a cross section of the design enclosure is shown in Fig. 5.

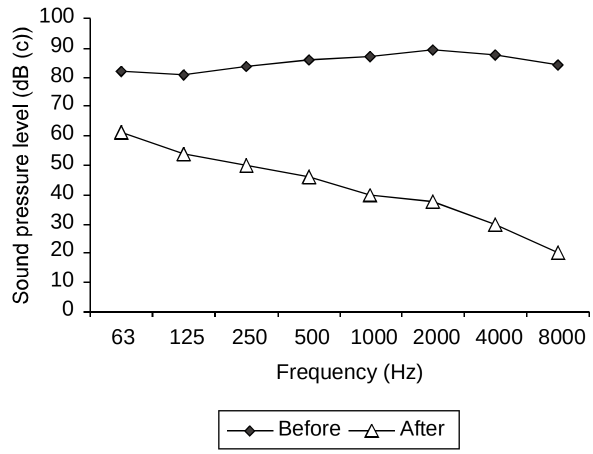


Fig. 4. Comparison between sound pressure level before and after installing the enclosure.

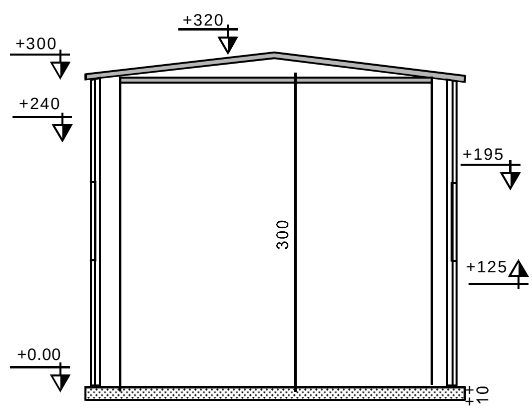


Fig. 5. A cross section of the designed enclosure for Isomax unit.

Isomax unit contains four pumps which operate to feed and lubricate the compressors. Results of field evaluation (Table 1) demonstrated that at all the tested stations, the sound pressure level was far above 85 dB(A). The results of evaluation of sound contour plan also showed that the Iso-sonic contour in the field between pumps A2 (90 dB(A) and A1 with 95 dB (A) is the highest (Fig. 2).

Frequency analysis results of octave band of pumps reveal the dominant frequency of pumps A2 and B2 to be 2000 Hz and that of pumps A1 and B1, 4000 Hz (Table 2). Evaluation of the rotation speed of pumps and determination of the dominant frequency of the pumps due to their technical specifications and calculating procedure, uphold me asurement procedure well. In this case the results of prediction method showed that the dominant frequency of pumps A2 and B2 was in the range of 2445 to 2500 Hz and that of pumps A1 and B1 pumps was in the range of 3675 to 4573 Hz.

It is thus concluded that designing and installation of an acoustic enclosure is the right choice for controlling noise of the pumps.

It was found that use of a layer of steel of 2 mm thickness and critical frequency of 8978 Hz – which is well above the dominant frequency of the noise sources – as insulator is a suitable control measure. However, applying steel as a layer in the module causes multiple reflective surfaces around the source and sound pressure level rises accordingly. Therefore, using absorbent material at the source side of the enclosure is necessary. It is worth adding that the results of evaluation of the average absorbent coefficient of the module, room factor and sound transmission loss in dominant frequencies showed that application of a single layer does not provide the expected transmission loss. Hence, multiple layers were used in the design of the enclosure.

Slag wool with surface density of 2.5 kg/m^2, was found to be the best absorbent. For providing enough distances between the two layers of the absorbent material, a wooden frame of 15 mm thickness was used. This improved the performance of the enclosure in 2000 Hz, which is the dominant frequency of some of the pumps. In order to reduce the reflection from external surfaces of the enclosure, the sandwich panel was finished by a layer of chipboard of 9 mm thickness. To provide straight vision to the pump operators, three vacuumed doubled glazed windows were also designed. Comparison of surface density of combined panel (16.25 kg/m^2) and minimum surface density needed for dominant frequency (12 kg/m^2), it was demonstrated that the designed module is much effective.

Results of the operational calculation of the transmission loss of the designed module with multiple layers showed that by close recognition of sound source and applying the above module, it is possible to provide 19.7 dB (A) reduction in overall sound pressure level and 20 dB reduction in dominant frequency. Lastly, it is concluded that designing the module with the given specifications and probable leak estimation and prevention gives remarkable and effective results in the field of study.

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