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Original Article

Evaluation of Noise Propagation Characteristics of Compressors in Tehran Oil Refinery Center and Presenting Control Methods

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ABSTRACT

Background: The adverse effects of noise are well known and noise problems due to industrialization of communities are increasing over the time. Oil industries due to the process and nature of production; contain many noise sources such as compressors, turbines, and pumps, which cause excessive noise exposure. The objective of this study was to evaluate the noise characteristics of compressors in Tehran Oil Refinery and study on visible control measures.
Methods: To get to the appropriate control method, procedures such as basic theories, measuring sound parameters, frequency analysis, related diagrams and noise propagation schemes due to the measurement results, equivalent noise exposure level ($L_{eq(8h)}$) and exposure noise dose and technical specification of compressors are considered in this paper. Considering field and analytical results, module enclosure with particular specifications (like absorbent layer, specific wall, window and door design etc.) is predicted to be the best control method.
Results: Calculation results of multiple layer density of the enclosure (W = 16.5 kg/m^2) and needed density for the dominant frequency of the source (W = 12 kg/m^2) demonstrated that the designed enclosure satisfies the goal.
Conclusion: Results of designing sandwich layers' module demonstrated that installing the designed enclosure causes 20 dB(A) reduction in total sound pressure level of the source's dominant frequency.

Introduction

Nowadays harmful noise effects are well known and noise problems due to industrialization of communities are increasing over the time. Of harmful noise effects can point to masking noise, relation with second kind diabetes and some psychological disorders [1]. In addition, effects on visual organ (interference in collation control and detecting items and reduce eye reaction to light) and equilibrium system (Nausea, confusion and walking interference) are other harmful effects of excessive noise exposure.

Compressors are one of the main noise sources in industries, which cause noticeable damages to working community annually. Oil industries due to the process and nature of production; contain many noise sources such as compressors, turbines, and pumps, which cause excessive noise. In this case, Nassiri et al. reported that noise exposure in the studied oil fields were far more than Iranian permissible levels [2].

On another study, excessive noise exposure was detected and so necessitated engineering noise controls were outlined [3]. Evaluation of noise pollution on oil field determined that compressors are one of the main noise sources in the field. In addition, this study resulted that applying enclosures to some noise sources would cause 14 to 19 dB noise reduction [2].

Esmail Zadeh et al. studied on noise pollution of compressed air conditioning unit on a factory and showed that compressors noise were excess of the allowable limit and control strategies on all the sources such as air outlet pipes and air inlet vents are the only way for reducing the noise and installing silencer and muffler sound is useful in the field [4]. From this context, Hakimi by applying a module in the air outlet has estimated a 20 dB reduction in the sound level [5].

Other survey done by Speich on controlling compressors' noise showed that for its noise control program noise should be evaluated on the view point of technical and spectral specifications [6].

Knight et al. denoted enclosure compressors with an enclosure composed of soft synthetic layers instead of applying hard layers would reduce the noise of compressors. Applying the acoustic enclosure reduces the compressor noise level by 9 dB [7]. Other evaluation for reducing compressors noise to permissible level specified that acoustic enclosure as the best method [8].

Joseph et al. survey entitled control of shear cutting noise effectiveness of enclosures showed that ignoring structural paths, which generated sound leaks from the controlling device, reduces the efficiency of enclosure. In addition, it was determined that precise recognition of the noise source and the field surfaces plays a great role in assessing the acoustic efficiency of device [9]. Applying multi purpose enclosure can reduce sound pressure level up to 40 dB [10].

Knight et al. study conducted to soft synthetic multi layer and absorbent module design instead of applying hard layers for enclosure air compressor [7]. This study showed that installing soft synthetic multi layer and absorbent material could reduce sound pressure level about 9 dB(A). Nathak et al. stated that the best result is achieved by applying soft synthetic multi layer and absorbent materials [8].

The control measures presented in this study are in accordance with similar studies stated before [8]. In this study, hydrogen gas pressure enhancer compressors were applied in the field (3 devices including A, B, C, 2C-401) to provide needed hydrogen in the unit with increase hydrogen pressure received from hydrogen generated unit or catalyst exchange unit around 2854 psig. Compressors, which were used in Isomax unit of Tehran Oil Refinery center, were composed of three parts: main compressor body (first section), reducer gear (second section), and turbine (third section).

In this paper, firstly the result of evaluated characteristics of noise generated by the compressors and then a recommended control method will be introduced.

Materials and Methods

First cooperating with HSE department of Tehran Oil Refinery center, basic information such as plans, noise source place, and some other technical information like device longevity, size, component, and task or function of the device was collected. In the plan of the unit, most important noise sources were pointed out, circled, and coded. In addition, number and task position of workers exposure to excessive noise were determined.

Second step was to measure sound parameters like SPL_{rms}, SPL_{max} on specified zone. TES-1385 sound level meter was applied in this study and it was calibrated by B & K 4231 standard calibrator. For this purpose, 21 measuring points, which are shown in Figure 2, were considered. The overall dimension of the investigated unit was 32 by 15 meter. It is worth noting that the measuring points were chosen by the net method and in fact, they are the centers of squares of 5 by 5 meter in the field. In this step, frequency analysis was done on C-weighted and results were given to excel and SURFER 7 software. Then related diagrams and noise propagation schemes were obtained. Standard method ISO 9612 (1997) is used for measurement [11]. Finally, in this step, reviewing routine procedure in the unit and following personal exposure time, equivalent noise exposure level ($L_{eq\ (8h)}$) and noise exposure dose were calculated.

On third step, to determine the dominant frequency of compressors by theoretical method, their technical specifications such as rate per minute were evaluated and estimated. Applying results of compressor's noise measuring, sound contours of the unit was drawn (Figure 1).

Finally, a control measure for the dominant noise source using both experimental and theoretical findings is presented and the overall performance of the designed enclosure is predicted.

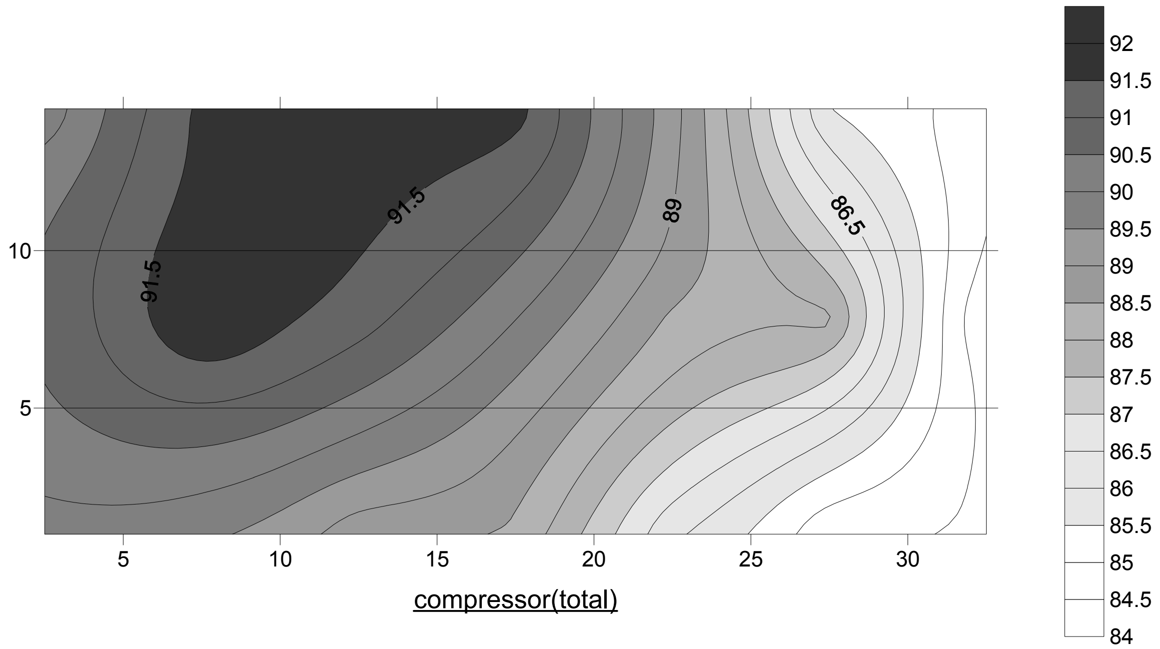

Figure 1: Noise contours in Isomax unit

Results

Compressor's specification of Isomax unit

There were three reciprocating compressors in Isomax unit. Two compressors work simultaneously and another one is left stand by. The results LA_{rms} and LA_{max} of 21 measuring points are shown in Table 1. The locations of the above-mentioned measuring points are showing in Figure 2. Using the above results, the crest factor was calculated by subtracting rms sound pressure level of maximum sound pressure level. The results are also shown in Table 1.

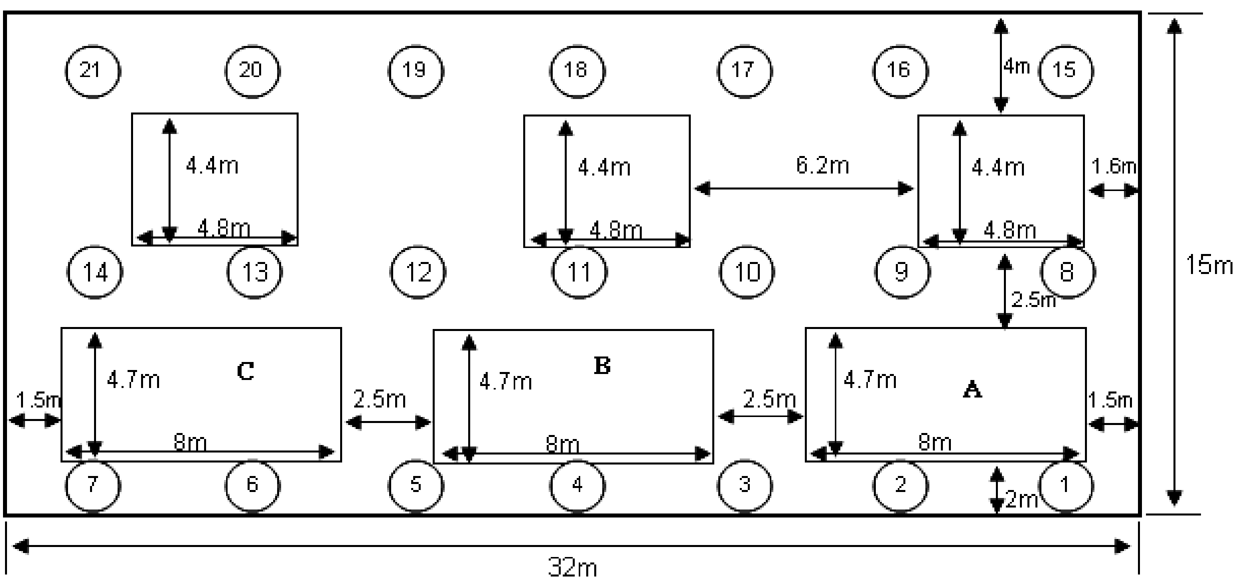

Figure 2: Noise zone plan of Isomax compressors, Compressor A is off and both compressors B and C are on

Table 1: The A-weighted sound pressure level and maximum sound pressure level along with the crest factor of Isomax compressors (dB)

Measuring point	L_{Arms}	L_{Amax}	CF
1	84.4	98.1	13.7
2	84.6	99.6	15
3	86.1	100	13.9
4	88.4	103.5	15.1
5	88.7	103	14.5
6	89.6	104.5	14.9
7	89.8	104.8	15
8	84	99.2	15
9	88.1	102	14
10	88.4	104.1	15.7
11	90	105.4	15.4
12	91.2	107	15.8
13	92	108	16.4
14	90.6	107	16.6
15	84.7	100	15.4
16	85.5	102.2	16.7
17	89.2	104.6	15.4
18	91.7	108	16.3
19	91.5	107.4	15.9
20	91.6	108.9	17.3
21	89.7	105.9	16.2

The noise was analyzed in octave band center frequency band in six measuring point. The results of turbine, gear, and compressor's side in Isomax unit are shown in Table 2 and Table 3. Point numbers is the tables are shown in Figure 2.

Table 2: Octave band center frequencies of turbines' side in Isomax unit

Measuring point	Points		
Frequency(Hz)	1	4	6
63	80.4	84.7	84.4
125	79.1	83.6	82.7
250	78.4	81.4	82.3
500	75.3	77.6	79.8
1000	76.7	82.5	81.6
2000	72.7	77.8	79.9
4000	69.2	76.1	77.5
8000	*62.7*	*69*	*71.2*

Table 3: Octave band center frequencies of gear and compressor's side in Isomax unit

Measuring point	Points		
Frequency (Hz)	16	18	20
63	79.5	79.4	78
125	83	83.1	82.1
250	81.3	82.2	81
500	78.2	79.8	78.4
1000	82.5	81.7	82
2000	81.8	81.4	81.8
4000	87	86.8	87.3
8000	*84*	*83.3*	*84*

Dominant noise frequency of Isomax compressors

The rotation speed of the three compressors' component (including compressor, gear and turbine) of Isomax unit was measured using a RPM meter. The rotation speed of the turbine was 4000 RPM with 49 blades and the rotation speed of gears and compressor were 1/13 of that of the turbines with 8 blades. Therefore using the well-known equation (see equation 3-1 Ref. No. 12) the dominant frequency for the turbine 3266.6 Hz and that is 41 Hz for the gears and compressor. In an octave band analysis the above frequencies is located respectively in 4000 and 63 Hz center frequencies. Therefore, very good agreement can be seen by comparison between the above prediction results and the field measurement results as shown in Table 2 and Table 3.

Equivalent workers' exposure level

To determine the equivalent exposure level, the exposure time of related workers with different sound pressure levels were specified. Workers usually work on three shifts with durations of 2.5, 1 and 4 h and the sound exposure levels in each section are respectively 71, 65 and 92 dBA. Therefore the $L_{Aeq(8h)}$ was found to be 89.5 dBA and the received dose according to Iran standard levels (85 dB per 8 h work shift and criterion shift parameter of 10) was 282%. Therefore, based on Iranian standard limits, allowable exposure limit for workers is just 2.8 h per day.

Control measure

Designing a noise enclosure for the Isomax compressors are as follows:

1- Critical frequency of main insulator of the enclosure (2mm steel)

For designing enclosure, it is firstly important to determine the critical frequency of the main insulator, which is considered 2mm steel. By applying the well-known equation of calculating the critical frequency (see equation 6-17 Ref. No. 12), this frequency is predicted to be 8978 Hz which is far above the dominant frequency of our main noise source.

2- Layout and specification of needed module sandwich layers

A: Absorbent: For the purpose of this design, a layer of slag wood with 2.5 kg/m^3 surface density and 25 mm thickness applied as an absorbent for the considered frequency.

B: Frame: A wooden frame with15 mm thickness and surface density of 7 kg/m^3 was used

C: Insulator: A 2 mm steel with surface density of 17 kg/m^3 was applied in the center-line of the panels for insulating the structure born noise.

D: Filler: A 20 mm polyurethane foam was used as a filler within the panels.

E: External surfaces: A 9 mm chipboard with surface density of 7 kg/m^3 was used for covering the external surfaces of the enclosure

F: Door: A common gash door with 43 mm thickness and surface density of 9 kg/m^3 was used for the entrance of the enclosure. The dimension of the door was designed to be 1.8 by 0.7 m.

G: Windows: 8 double layer glass windows with 9mm thickness and surface density of 7 kg/m^3 and dimension of 1.7 by 1.7 m were used for the enclosure.

3- Calculating the least surface density for the dominant frequencies (63, 4000 Hz)

The least surface density for the dominant frequencies was calculated to be 12 kg per cubic meter [12].

4- Size and area of Enclosure

The dimensions of the designed enclosure were 3×5×5.5 m and as it was mentioned 8 windows with size of 1.70×1.70 m and a 1.80×0.70 m door were used in the design. Total surface of the enclosure was 118 square meter and using the following equation the overall panel density was found to be 16.5 kg per cubic meter.

$$\bar{w} = \frac{\sum w_i \times s_i}{\sum s_i} \qquad \text{(Eq.1)}$$

Where w_i and s_i are the surface density and the area of each panel component respectively (Figure 3).

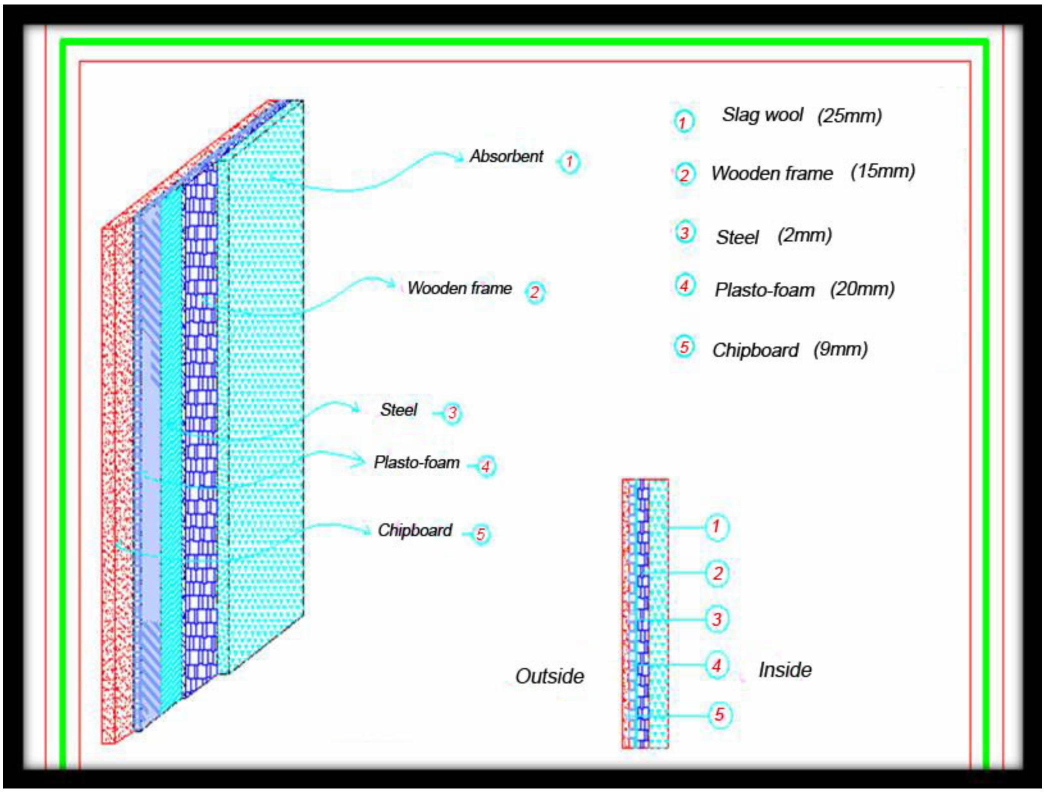

Figure 3: Detail structure of the main enclosure panel

5- Frequency analysis, and TL and NR

Using the above mentioned field measurement results [92 dB(A)] and Iranian noise exposure limit [85 dB(A)], it is easily found that the total noise reduction required is 12 dB [92 dB(A) - 85 dB(A) +5 dB(A)] and in that of dominant frequency is 16 dB. The 5 dB is mostly added to achieve the practical results.

Total noise reduction achieved by installing the designed enclosure was calculated about 20.1 dB, which is gained by subtraction of total outdoor and indoor noise levels. In this case, the overall noise level inside the enclosure was measured to be 92 dB while the noise level outside the enclosure was estimated to be 71.9 dB. Figure 4 provides sound pressure level variations before and after installing the enclosure.

The architectural plans and related details were designed by AutoCAD software and are shown in Figure 5 to Figure 7.

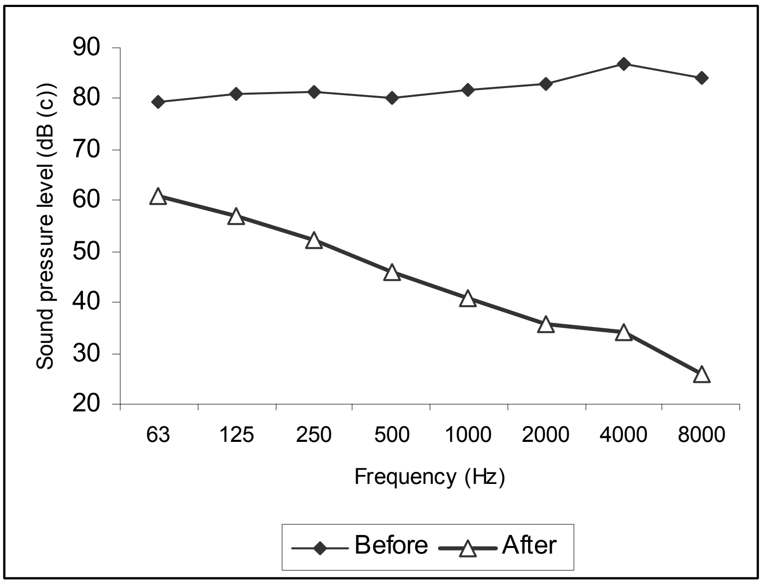

Figure 4: Comparison between field sound pressure level before and after installing the enclosure

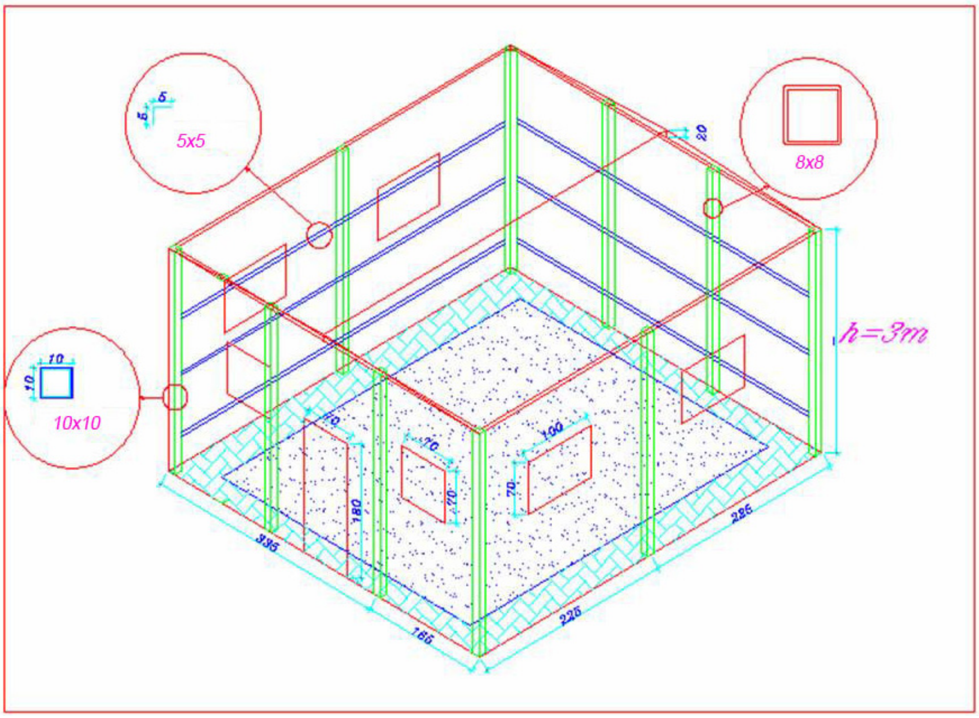

Figure 5: A detailed three dimensional plan of the designed enclosure (the unit of undefined numbers are based on cm)

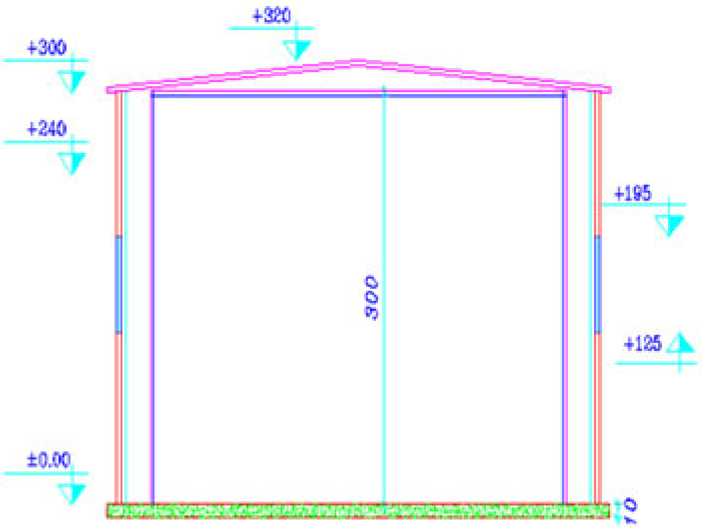

Figure 6: A cross section of the designed enclosure for Isomax unit

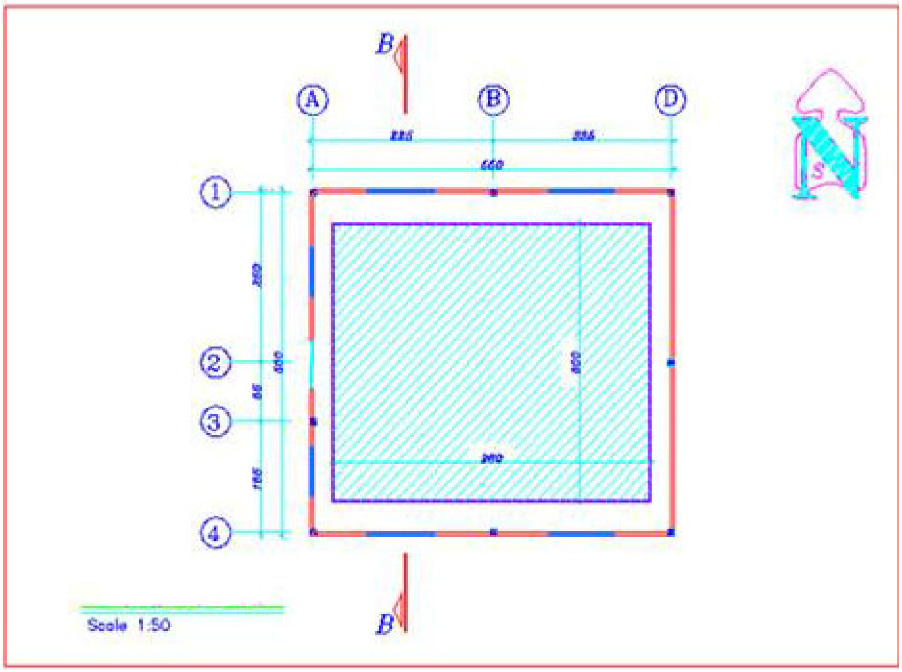

Figure 7: The horizontal plan of the designed enclosure for Isomax unit

Discussion

Results of field measurement of compressors' noise (Table 1) indicates that in 17 of 21 stations evaluated, sound pressure level was above 85 dB(A) and just in 4 stations, the measured noise was just 1 dB(A) bellow Iranian standard. These four stations were close to the stand by compressors (Figure 2).

Results of sound plan presented in compressors' zone showed that the same level contours which placed between two running compressors contains maximum sound pressure level [91.5 dB(A)]. Also evaluating sound counters plan illustrated that sound pressure level around compressors' placement was above Iranian standard levels, which is in consistent with previous studies [2, 4] (Figure 1).

Equivalent exposure level of workers indicated that workers noise exposure was above allowable level and due to standard limits, allowable working shift should be reduced to just 2.8 h, also daily exposure dose results showed that exposure dose was about 282% and there should applied control methods to reduce the effects of harmful noise.

The results of both theoretical prediction and field frequency analysis demonstrated that the dominant frequency at the turbine side was 63 Hz, while that was 4000 Hz for gear and compressor side of the noise sources (Table 2 and Table 3).

To control the compressor noise and reduce its harmful effect with less noise exposure level, it was decided to design and apply an insulating enclosure.

Considering the large scale of the needed enclosure for controlling the noise and also with regard to results achieved by frequency evaluation limit of insulation applied for the enclosure, it was demonstrated that the critical frequency of a 2 mm steel insulator was far above the dominant noise frequency of the source (f_c = 8978 Hz). So the steel panel was used as a main insulator of the enclosure. It is worth noting that using steel panel with thickness more than 2 mm will reduce the critical frequency bellow the dominant frequency of the source. This obviously reduces the performance of the control measure.

Calculation

To avoid the multiple reflections from the hard surfaces of the insulators, utilizing an efficient absorbent material is unavoidable.

In addition, results of evaluating absorbent coefficient of the enclosure and minded transmission loss in dominant octave band frequencies indicated that applying a layer in module designing, would not get to the considerate transmission loss. Therefore, a multiple layers (sandwich layers) were applied.

Sound absorbent evaluation also showed that the slag wool was a suitable absorbent for the control measure. Fiberglass was also another suitable choice but using perforated sheet or metal lace is necessary for installing a fiberglass on a panel, which in this study and due to the large size of the enclosure, using this absorbent increase metal surface area of the enclosure and so increase the reflection of the sound inside the enclosure which leads to increase the sound pressure level inside the enclosure. To reduce the noise reflection of the outside of the enclosure, a 9 mm chipboard is used as a last layer.

Results of multiple layer density of the enclosure (W = 16.5 kg/m^3) and needed density for the dominant frequency of the source (W = 12 kg/m^3) demonstrated that the designed enclosure satisfies the goal. In fact, the findings listed above are in agreement with the results of the other studies in the literature [7-9].

Results of designing sandwich layers' module demonstrated that installing the designed enclosure causes in 20 dB(A) reduction in total sound pressure level of the source's dominant frequency.

Finally, it should mention that this article is only presented according to prediction and designs and the results should be validating with field research.

Acknowledgements

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