OPTIMAL DESIGN ON ONE-LAYER CLOSE-FITTING ACOUSTICAL HOODS USING A SIMULATED ANNEALING METHOD

Min-Chie Chiu

Key words: close-fitting acoustical hood, simulated annealing method.

ABSTRACT

High noise levels in a multi-noise plant can be harmful to workers and will not only lead to psychological but also to physiological ailments. Consequently, noise control of industrial equipment becomes vital for workers. The study focuses on shape optimization of space-constrained close-fitting acoustic hoods.

In this paper, a sound insertion loss used for evaluating the acoustic performance of an acoustical hood will be adopted. Numerical assessments of case studies for depressing the broadband noise emitted from motor-driven equipment by optimally designing a shaped one-layer close-fitting acoustic hood within a space constrained situation will be introduced. Additionally, simulated annealing (SA), a robust scheme used to search for the global optimum of a one-layer close-fitting acoustic hood by imitating the metal’s heating process, was used during the optimization process. Before dealing with a broadband noise, the maximization of sound insertion loss (IL) with respect to a one-tone noise was introduced for a reliability check on the SA method. Also, an accuracy check on the mathematical model was performed.

This paper, by using a simulated annealing method, presents an optimally designed one-layer close-fitting acoustic hood to provide a quick and effective method to reduce the noise level of the equipment.

I. INTRODUCTION

Noise control work on equipment with an acoustic hood in industry is vital for workers [1, 6]. Beranak et al. [4] started the study of noise reduction for an acoustical panel using a acoustical mass law. Considering mechanical resistance, London [17] proposed a sound transmission loss (STL) for a rectangular panel. Crocker [7] also proposed a mathematical model for the sound transmission loss of a resonating/non-resonating panel. Oldham et al. [20, 21] assessed the prediction formula of sound transmission loss on the basis of a vibration model. Fahy [8], Beranak et al. [3], and Kinsler et al. [14] analyzed the sound transmission loss for an infinite acoustical panel without boundary conditions. However, the STL cannot easily be used to evaluate practical acoustical performance. Moreover, the vibration mode in an acoustical board will be largely induced by the near-sound-field-effect. Therefore, sound insertion loss (IL) used in evaluating the acoustical efficiency before and after a stiff acoustic hood is installed is adopted. Because of the near-sound-field-effect, the acoustical performance of the acoustic hood is linked to the gap between the equipment and the hood [20, 21]. In addition, Jackson [11, 12] assessed the IL of a close fitting acoustic hood using two pieces of flat plates. His experimental results revealed that the IL is highly influenced by the vibration of the vibrating noise; however, a negative IL is unreasonable. Junger [13] also proposed a theoretical formula to predict the IL; but, the accuracy of the theory and the experimental data is inconsistent. In 1972, Hine [10] approached the theoretical IL using the plate’s vibration model. Yet, the accuracy was still insufficient. Considering the effect of the vibration model on the plate, Moreland [19] predicted noise reduction (NR) in a close-fitting acoustic hood; however, the formula is valid only for noise of low frequencies. Roberts [24] also predicted the IL of the acoustic hood at the critical frequency. Considering the effect of the vibrating mode on the hood, Oldham [20, 21] successfully assessed the IL of a close-fitting acoustic hood within both the simple supported boundary condition and the clamped boundary condition. Results reveal that the accuracy between the theory and the experimental data is in agreement.

Because the constrained problem is mostly concerned with the necessity of operation and maintenance in practical engineering work, there is a growing need to optimize the acoustical performance under a fixed space. However, research of shape optimization on a space-constrained close-fitting acoustic hood by adjusting the design parameters (the panel’s damping ratio, the panel’s thickness, and the gap between the equipment and the hood) has been neglected. In order to efficiently...
depress the noise level, a numerical assessment for finding an optimally shaped acoustic hood in conjunction with a simulated annealing method (SA) will be presented.

This paper provides a quick and effective method to reduce the noise by optimally designing a shaped one-layer close-fitting acoustical hood within a space-constrained situation.

II. MATHEMATICAL MODELS

A one-layer close-fitting acoustic hood made of metal shown in Fig. 1 is adopted for reducing the noise emitted from a piece of motor-driven equipment. The mathematical model for the one-layer close-fitting acoustic hood is described below.

1. The One-layer Close-fitting Acoustic Hood

According to Oldham [20, 21], for a one-layer close-fitting acoustical hood within a clamped boundary condition, the \( IL \) is

\[
IL(\bar{X}) = 10 \times \log_{10} \left[ \cos(kd) + \left( \frac{\pi^2}{4K\rho_{oc}} \right) \sin(kd) \right]^2
\]

(1a)

\[
K = \left[ \frac{1.35}{3.86 + D_1 \left( \frac{129.6}{a^2} + \frac{78.4}{a^b} + \frac{129.6}{b^2} \right) - \omega^2 \rho h} \right]^{1/2}
\]

(1b)

\[
D_1 = \left[ \frac{Eh^3}{12(1-\nu^2)} \right] * (1 + i\eta)
\]

(1c)

\[
\bar{X} = (f, h, d, \eta, a, b, E, \rho, \nu)
\]

(1d)

where \( k \) is the wave number, \( d \) is the distance between the equipment and the hood, \( \rho_{oc} \) is the acoustic impedance, \( E \) is the panel’s Young modulus, \( D_1 \) is the panel’s complex bulk modulus, \( a \) is the length of the panel, \( b \) is the width of the panel, \( \eta \) is the panel’s internal damping coefficient, \( h \) is the thickness of the panel, \( \nu \) is the poison ratio of the panel, and \( \rho \) is the panel’s density.

2. Overall Sound Pressure Level after Using an Acoustical Hood

The silenced octave sound pressure level at the noise testing point shown in Fig. 1 is

\[
SPL_i = SPLO_i - IL_i
\]

(2)

where \( SPLO_i \) is the original \( SPL \) at the noise testing point without an acoustical hood, and \( i \) is the index of the octave band frequency. \( IL_i \) is the sound insertion loss (IL) with respect to the relative octave band frequency. \( SPL_i \) is the silenced SPL (with an acoustical hood) with respect to the relative octave band frequency.

Finally, the overall \( SPL_T \) silenced by an acoustical hood at a specified location is

\[
SPL_T = 10 \times \log \left( \sum_{f=1}^{10} \frac{\sum_{i=1}^{5} \left[ \frac{\left(10^{\left(\frac{SPLO(f=125) - IL(f=125)\right)/10}\right) - \left(10^{\left(\frac{SPLO(f=250) - IL(f=250)\right)/10}\right)}{10}\right]}{10} + \frac{\left(10^{\left(\frac{SPLO(f=500) - IL(f=500)\right)/10}\right) - \left(10^{\left(\frac{SPLO(f=1000) - IL(f=1000)\right)/10}\right)}{10}\right)}{10}} \right)
\]

(3)

3. Objective Function

By using Eqs. (1) and (3), the objective function used in the SA optimization was established.

1) \( IL \) Maximization for a One-Tone (\( f \)) Noise

\[
OBJ_1 = IL(f, h, d, \eta, a, b, E, \rho, \nu)
\]

(4)

2) \( SPL_T \) Minimization for a Broadband Noise

To minimize the overall \( SPL_T \), the objective function is

\[
OBJ_2 = SPL_T(f, h, d, \eta, a, b, E, \rho, \nu)
\]

(5)

An aluminum-made acoustical hood is selected in the numerical assessment, the related ranges of parameters \((h, d, \eta)\) are

\[
h: [0.01, 0.02]; \ d: [0.4, 1.0]; \ \eta: [0.001, 0.1]
\]

(6)

III. MODEL CHECK

Before performing the SA optimization, an accuracy check
of the mathematical model on the one-layer close-fitting acoustic hood is performed using experimental data from Blanks [2]. As depicted in Fig. 2, the trend of the performance curve with respect to the theoretical and experimental data are relatively similar. Therefore, the mathematical model is acceptable. Consequently, the model linked with the following numerical method is used for optimizing the shape of one-layer close-fitting acoustic hoods in the following section.

IV. CASE STUDIES

An aluminum-made acoustical hood \( a = b = 0.8 \) (M), \( E = 69 \times 10^7 \) (Pa), \( \nu = 0.33, \rho = 2700 \) kg/m\(^3\) used to depress a noise from motor-driven equipment is adopted and shown in Fig. 1. The sound pressure level (SPL) at the noise testing point three meters from the motor-driven equipment is shown in Table 1 where the overall SPL reaches 115.5 dB(A). To suppress the noise from the motor-driven equipment, an aluminum-made acoustical hood with a one-layer close-covering is considered. To obtain the best acoustical performance within a fixed space, numerical assessments linked to an optimizer are applied. Before the minimization of a broadband noise is executed, a reliability check of the SA method by maximization of the IL at a targeted tone (500 Hz) has been carried out. To appreciate the acoustic performance, three kinds of material for the cover (aluminum, steel, and acrylic) are accessed and optimized. Moreover, the sensitivity of IL with respect to \( h \), \( d \), and \( \eta \) will be assessed.

<table>
<thead>
<tr>
<th>f(Hz)</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1k</th>
<th>2k</th>
<th>Overall</th>
</tr>
</thead>
<tbody>
<tr>
<td>SPL –dB(A)</td>
<td>100</td>
<td>103</td>
<td>113</td>
<td>110</td>
<td>104</td>
<td>115.5</td>
</tr>
</tbody>
</table>

Fig. 2. The performance curve with respect to the theoretical and experimental data [\( a = b = 0.293 \) (m); \( q = 0.33; d = 0.012 \) (m); \( h = 0.013 \) (m)] [2].

V. SIMULATED ANNEALING METHOD

Various methods used for solving optimization problems can be classified into three categories — enumerative, deterministic, and stochastic. The first technique is best applied to problems that are defined by a few discrete decision variables only [16, 22]. The second technique mainly incorporates problem domain knowledge to reduce the size of the search space. However, during the optimization process [23, 25, 26], the gradient methods, deterministic techniques, require a good starting point and a mathematical derivation that is calculated in advance. Evolutionary Algorithms (EAs) belong to the group of stochastic search methods, also referred to as random search. Evolutionary Algorithms have been widely developed for two decades. Many good EAs have been established. Simulated Annealing [9] is one of the best stochastic search methods. Here, sensitivity analyses is not necessary for choosing the starting design data, which is required in classical gradient methods of exterior penalty function method (EPFM), interior penalty function method (IPFM) and feasible direction method (FDM) [5]. Therefore, SA is adopted as an optimizer and used in the muffler’s shape optimization.

The basic concept behind SA was first introduced by Metropolis et al. [18] and developed by Kirkpatrick et al. [15]. The scheme of SA is a variation of the hill-climbing algorithm. All downhill movements for improvement are accepted for the decrement of the system’s energy. In order to escape from the local optimum, SA also allows movement resulting in solutions that are worse (uphill moves) than the current solution. The pseudo-code implementing the simulated annealing heuristic is listed in Table 2. To imitate the evolution of the SA algorithm, a new random solution \((X')\) is chosen from the neighborhood of the current solution \((X)\). If the change in the objective function (or energy) is negative \((\Delta F \leq 0)\), the new solution will be acknowledged as the new current solution with the transition property \((pb(X'))\) of 1. If the change is not negative \((\Delta F > 0)\), the probability of making the transition to the new state \(X'\) will be a function \( pb(\Delta F/CT) \) of the energy difference \( \Delta F = F(X') - F(X) \) between the two states and a function of the global time-varying parameter \( T \). The new
transition property \( (pb(X')) \) varied from 0–1 will be calculated by the Boltzmann factor \( (pb(X') = \exp(\Delta F/CT)) \) as shown in Eq. (7)

\[
\begin{align*}
 pb(X') &= \begin{cases} 
 1, & \Delta F \leq 0 \\
 \exp(-\Delta F/CT), & \Delta F > 0
\end{cases} 
\end{align*}
\]

(7a)

\[
\Delta F = F(X') - F(X) 
\]

(7b)

where \( C \) and \( T \) are the Boltzmann constant and the current temperature. Moreover, compared with the new random probability of \( rand(0,1) \), if the transition property \( (pb(X')) \) is greater than a random number of \( rand(0,1) \), the new solution (worse solution) which results in a higher energy condition will then be accepted; otherwise, it is rejected. Each successful substitution of the new current solution will lead to the decay of the current temperature as

\[
T_{\text{new}} = kk \ast T_{\text{old}} 
\]

(8)

where \( kk \) is the cooling rate.

The process is repeated until the predetermined number \( (\text{iter}) \) of the outer loop is reached.

VI. RESULTS AND DISCUSSION

1. Results

1) Sensitivity Analysis

Before the optimization process of an aluminum-made acoustical hood used in the noise elimination of motor-driven equipment is performed, the sensitivity analysis of the IL of three parameters \( (h, d, \eta) \) is assessed. The results of the sensitivity analysis are shown in Figs. 3–5. As indicated in Fig. 3, the span of the IL curve will be widened when \( d \) decreases. In addition, Fig. 4 reveals that the IL of the hood will increase when \( h \) increases. However, as depicted in Fig. 5, the sensitivity of the IL with respect to \( \eta \) is small. To accurately search for a best shaped hood, three design parameters \( (h, d, \eta) \) used in the optimization process are selected.

2) Optimization

The accuracy of the SA optimization depends on the cooling rate \( (kk) \) and the number of iterations \( (\text{iter}) \). To achieve a good optimization, both the cooling rate \( (kk) \) and the number of iterations \( (\text{iter}) \) are varied step by step

\[
kk = (0.91, 0.93, 0.95, 0.97, 0.99); \text{iter} = (50, 100, 500) 
\]
M.-C. Chiu: Optimization on One-Layer Acoustical Hood by SA

Table 3. Optimal design data for an acoustical hood (targeted tone at 500 Hz).

<table>
<thead>
<tr>
<th>SA parameter</th>
<th>Design parameters</th>
<th>Performance</th>
</tr>
</thead>
<tbody>
<tr>
<td>iter</td>
<td>kk</td>
<td>h</td>
</tr>
<tr>
<td>50</td>
<td>0.91</td>
<td>0.01423</td>
</tr>
<tr>
<td>50</td>
<td>0.93</td>
<td>0.01987</td>
</tr>
<tr>
<td>50</td>
<td>0.95</td>
<td>0.01078</td>
</tr>
<tr>
<td>50</td>
<td>0.97</td>
<td>0.01147</td>
</tr>
<tr>
<td>50</td>
<td>0.99</td>
<td>0.01884</td>
</tr>
<tr>
<td>500</td>
<td>0.99</td>
<td>0.01822</td>
</tr>
<tr>
<td>1000</td>
<td>0.99</td>
<td>0.01717</td>
</tr>
</tbody>
</table>

Fig. 6. The IL with respect to frequencies at the SA parameter (kk) at iter = 50 (targeted tone: 500 Hz).

Fig. 7. The IL with respect to frequencies at the SA parameter (iter) at kk = 0.99 (targeted tone: 500 Hz).

The results of two kinds of optimizations (one, a pure tone noise; the other, a broadband noise) are described as follows.

A. Pure Tone Noise Optimization

By using Eq. (4), the maximization of the IL with respect to a one-layer close-fitting acoustical hood at the specified pure tone (500 Hz) was performed first. As indicated in Table 3, seven sets of SA parameters are tried in the acoustical hood's optimization. Obviously, the optimal design data can be obtained from the last set of SA parameters at (kk, iter) = (0.99, 500). Using the optimal design data in a theoretical calculation, the resultant curves of the IL with respect to various SA parameters (kk, iter) are depicted in Figs. 6~7. As revealed in Fig. 7, the IL is precisely maximized at the desired frequency.

B. Broadband Noise Optimization

To realize the influence of the IL with respect to various hood material, the investigation into the influence of the acoustical performance with respect to aluminum, steel, and acrylic is also assessed during the optimization process. Using the formulas of Eq. (5) and the SA parameters of (kk = 0.99, iter = 500), the minimization of the sound pressure level of the noise emitted from a piece of motor-driven equipment at the noise testing point is performed. The optimal result is obtained and shown in Table 4. Using these optimal design data in a theoretical calculation, the resultant curve of the SPL_T with respect to the original SPL are plotted in Fig. 8. As illustrated in Tables 1 and 4, for a steel-made acoustical hood, the sound pressure level at the noise testing point will be improved from 115.5 dB(A) to 52.01 dB(A) by using an optimal acoustical hood. In addition, for an aluminum-made acoustical hood, the sound pressure level at the noise testing point will be improved from 115.5 dB(A) to 59.19 dB(A) by using an optimal acoustical hood. Furthermore, for an acrylic-made

Table 4. Optimal design data for an acoustical hood (broadband noise) (iter = 1000; kk = 0.99).

<table>
<thead>
<tr>
<th>Material</th>
<th>Design parameters</th>
<th>OBJ</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>h</td>
<td>d</td>
</tr>
<tr>
<td>Acrylic</td>
<td>0.01833</td>
<td>0.8999</td>
</tr>
<tr>
<td>Aluminum (Al)</td>
<td>0.01833</td>
<td>0.8999</td>
</tr>
<tr>
<td>Steel (Fe)</td>
<td>0.01833</td>
<td>0.8999</td>
</tr>
</tbody>
</table>

Fig. 8. Optimal IL for acoustical hoods (broadband noise).
acoustical hood, the sound pressure level at the noise testing point will be improved from 115.5 dB(A) to 68.09 dB(A) by using an optimal acoustical hood. Obviously, the acoustical performance of a steel-made acoustical hood is superior to others. Moreover, an acrylic-made acoustical hood has the worst acoustical performance of the IL.

2. Discussion

To achieve sufficient optimization, the selection of the appropriate SA parameter set is essential. As indicated in Table 3 and Figs. 6–7, the best SA set with respect to a one-layer close-fitting acoustical hood at the targeted tone of 500 Hz is shown. Fig. 7 reveals the predicted maximal value of the IL is precisely located at the desired frequency. Therefore, the usage of the SA optimization in finding a better design solution is reliable; moreover, in dealing with the broadband noise using a one-layer close-fitting acoustical hood, Tables 1 and 4 indicate the overall sound insertion loss of the optimally shaped acoustical hoods with respect to three kinds of hood material (steel-made, aluminum-made, and acrylic-made) reached 63.5 dB(A), 55.3 dB(A), and 47.4 dB(A), respectively. As indicated in Fig. 8, the IL curve of the steel-made acoustical hood can provide a more efficient noise reduction in lowering the whole SPL curve.

VII. CONCLUSION

It has been shown that the one-layer close-fitting acoustical hood in conjunction with an SA optimizer can be easily and efficiently optimized within a constrained space. As indicated in Table 3, two kinds of SA parameters (kk, iter) play essential roles in the solution’s accuracy during SA optimization. As indicated in Fig. 7, the IL is precisely maximized at the desired frequency; therefore, the tuning ability established by adjusting design parameters of the acoustical hood is reliable. In addition, the appropriate acoustical performance curve of the acoustical hood in decreasing overall broadband noise using three kinds of hood material (steel-made, aluminum-made, and acrylic-made) has been assessed and shown in Fig. 8. As indicated in Fig. 8, the acoustical performance of a steel-made acoustical hood having a IL of 63.5 dB(A) is superior to the others. Moreover, the investigation into the influence of acoustical performance with respect to h, d, η indicates that the span of the IL curve will be widened when d decreases; in addition, the IL of the hood will increase when h increases. Consequently, this approach used for optimally designing the shaped acoustical hoods is easy and quite effective.

NOMENCLATURE

This paper is constructed on the basis of the following notations:

- \( a \): the length of the panel
- \( b \): the width of the panel
- \( C_o \): sound speed (m s\(^{-1}\))
- \( d \): the distance between the equipment and the hood
- \( D_1 \): the panel’s complex bulk modulus
- \( E \): the panel’s Young’s modulus
- \( F \): cyclic frequency (Hz)
- \( h \): the thickness of the panel
- \( k \): wave number \((= \frac{\omega}{c_o})\)
- \( k k \): cooling rate in SA
- \( O B J_i \): objective function
- \( p b (T) \): transition probability
- \( S P L_T \): the sound pressure level after adding an acoustical hood
- \( T \): the temperature in simulated annealing
- \( \rho_c \): the acoustic impedance
- \( \eta \): the panel’s internal damping coefficient
- \( \nu \): the poison ratio of the panel
- \( \rho \): the panel’s density.

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