Journal of Marine Science and Technology

Volume 22 | Issue 2 Article 12

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

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Recommended Citation
Chiu, Min-Chie (2014) "OPTIMAL DESIGN ON ONE-LAYER CLOSE-FITTING ACOUSTICAL HOODS USING A SIMULATED ANNEALING METHOD," *Journal of Marine Science and Technology*: Vol. 22: Iss. 2, Article 12.
DOI: 10.6119/JMST-013-0503-1
Available at: https://jmstt.ntou.edu.tw/journal/vol22/iss2/12

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Acknowledgements

The author acknowledges the financial support of the National Science Council (NSC100-2622-E-235-001-CC3 and NSC102-2221-E-235-001, Taiwan, ROC).

Journal of Marine Science and Technology, Vol. 22, No. 2, pp. 211-217 (2014)
DOI: 10.6119/JMST-013-0503-1

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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

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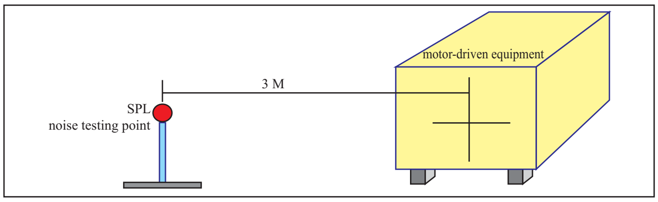
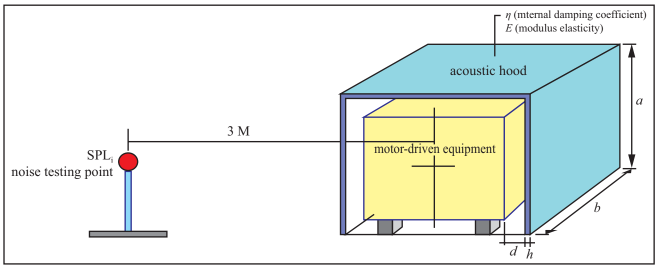


(A) without acoustical hood



(B) with acoustical hood

Fig. 1. Motor-driven equipment before/after adding a one-layer close-fitting acoustical hood.

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

$$IL(\bar{X}) = 10*\log_{10}\left[\left(\cos(kd) + \left(\frac{\pi^2}{4K\omega\rho_o c}\right)*\sin(kd)\right)^2\right] \tag{1a}$$

$$K = \frac{(1.35)}{\left[3.86*D_1\left(\frac{129.6}{a^4} + \frac{78.4}{a^2b^2} + \frac{129.6}{b^4}\right) - \omega^2\rho h\right]} \tag{1b}$$

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

$$\bar{X} = (f, h, d, \eta, a, b, E, \rho, \nu) \tag{1d}$$

where k is the wave number, d is the distance between the equipment and the hood, $\rho_o c$ 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, η is the panel's internal damping coefficient, h is the thickness of the panel, ν is the poison ratio of the panel, and ρ 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

$$SPL_i = SPLO_i - IL_i \tag{2}$$

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

$$SPL_T = 10*\log\{\sum_{i=1}^{5} 10^{SPL_i/10}\}$$
$$= 10*\log\left\{\begin{array}{l} 10^{[SPLO(f=125)-IL(f=125)]/10} + 10^{[SPLO(f=250)-IL(f=250)]/10} \\ +10^{[SPLO(f=500)-IL(f=500)]/10} + 10^{[SPLO(f=1000)-IL(f=1000)]/10} \\ +10^{[SPLO(f=2000)-IL(f=2000)]/10} \end{array}\right\} \tag{3}$$

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

$$OBJ_1 = IL(f, h, d, \eta, a, b, E, \rho, \nu) \tag{4}$$

2) SPL_T Minimization for a Broadband Noise

To minimize the overall SPL_T, the objective function is

$$OBJ_2 = SPL_T(f, h, d, \eta, a, b, E, \rho, \nu) \tag{5}$$

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

$$h: [0.01, 0.02];\ d: [0.4, 1.0];\ \eta: [0.001, 0.1] \tag{6}$$

III. MODEL CHECK

Before performing the *SA* optimization, an accuracy check

Table 1. The spectrum of a original sound pressure level (*SPLO*) at the noise testing point without adding an acoustical hood.

f(Hz)	125	250	500	1k	2k	Overall
SPLO –dB(A)	100	103	113	110	104	115.5

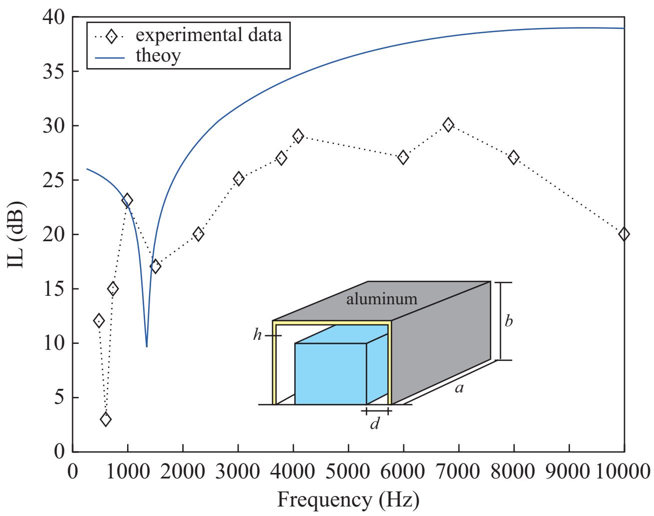


Fig. 2. The performance curve with respect to the theoretical and experimental data [a = b = 0.293 (m); η = 0.33; d = 0.012 (m); h = 0.013 (m)] [2].

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 = $69*10^9$ (Pa), ν = 0.33, ρ = 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-fitting cover is considered. To obtain the best acoustical performance within a fixed space, numerical assessments linked to an *SA* 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 η will be assessed.

Table 2. The pseudo-code implementing the simulated annealing heuristic.

```
T := To
X := Xo
F := F(X)
k := 0
while n < iter
  Xn' := neighbour(Xn)
  ΔF=F(Xn')-F(Xn)
  if ΔF ≤ 0 then Xn'= Xn ; T'n=kk*Tn ; n := n + 1
     elseif random() < pb(ΔF /C*Tn) then
        Xn'= Xn ; T'n=kk*Tn ; n := n + 1
return
```

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 (*EA*s) 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 *EA*s 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 ($\Delta F \le 0$), 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

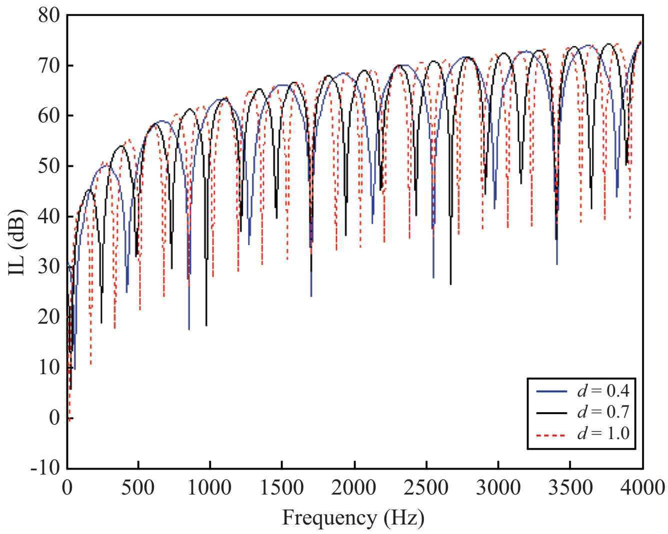


Fig. 3. Sensitivity of the *IL* with respect to *d*.

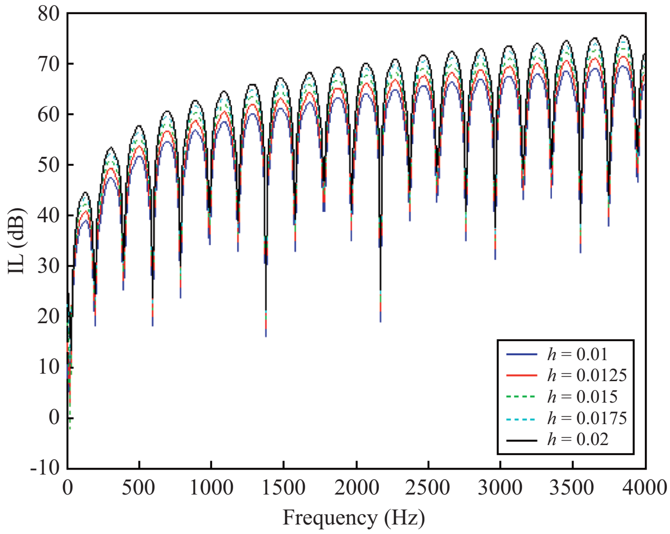


Fig. 4. Sensitivity of the *IL* with respect to *h*.

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)

$$pb(X') = \begin{cases} 1, \Delta F \le 0 \\ \exp(\dfrac{-\Delta F}{CT}), \Delta f > 0 \end{cases} \quad (7a)$$

$$\Delta F = F(X') - F(X) \quad (7b)$$

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

$$T_{new} = kk * T_{old} \quad (8)$$

where kk is the cooling rate.

The process is repeated until the predetermined number (*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, η) 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 η is small. To accurately search for a best shaped hood, three design parameters (h, d, η) used in the optimization process are selected.

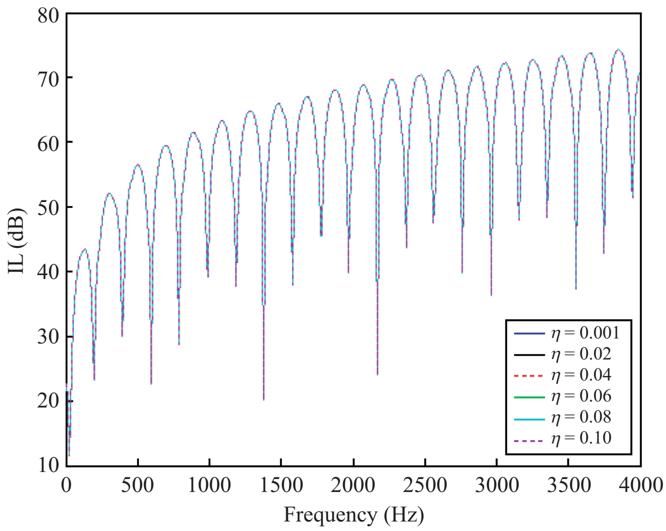


Fig. 5. Sensitivity of the *IL* with respect to *η*.

2) Optimization

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

kk = (0.91, 0.93, 0.95, 0.97, 0.99); *iter* = (50, 100, 500)

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

SA parameter		Design parameters			Performance
iter	*kk*	*h*	*d*	*η*	*IL* (dB)
50	0.91	0.01423	0.6537	0.04286	42.3
50	0.93	0.01987	0.9921	0.09869	45.7
50	0.95	0.01078	0.4469	0.008731	50.63
50	0.97	0.01147	0.4884	0.01559	52.58
50	0.99	0.01884	0.9304	0.08852	54.43
500	0.99	0.01822	0.8933	0.08240	56.2
1000	0.99	0.01717	0.8299	0.07194	56.11

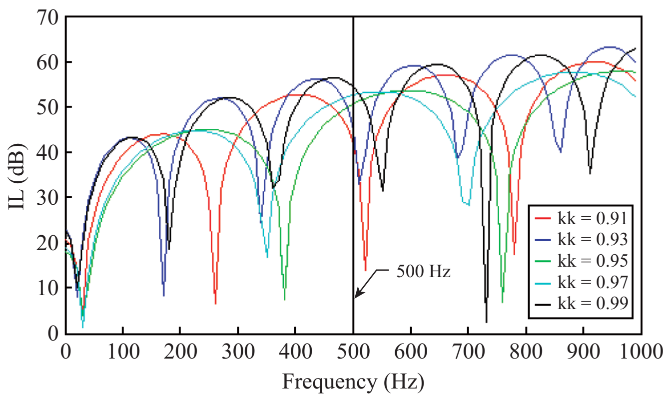


Fig. 6. The *IL* with respect to frequencies at the *SA* parameter (*kk*) at *iter* = 50 (targeted tone: 500 Hz).

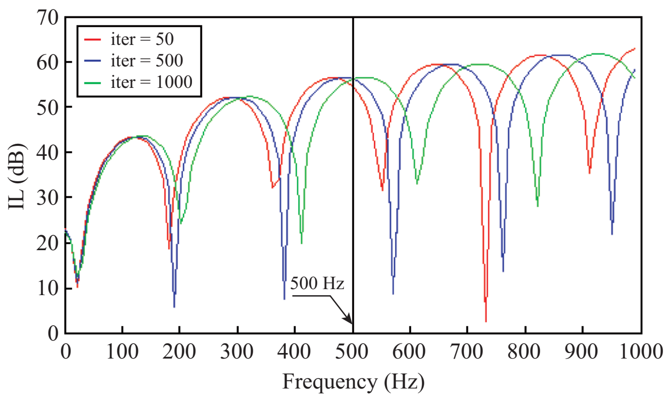


Fig. 7. The *IL* with respect to frequencies at the *SA* parameter (*iter*) at *kk* = 0.99 (targeted tone: 500 Hz).

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

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

Material	Design parameters			*OBJ*
	h	*d*	*η*	SPL_T –dB(A)
Acrylic	0.01833	0.8999	0.08348	68.09
Aluminum (Al)	0.01833	0.8999	0.08348	59.19
Steel (Fe)	0.01833	0.8999	0.08348	52.01

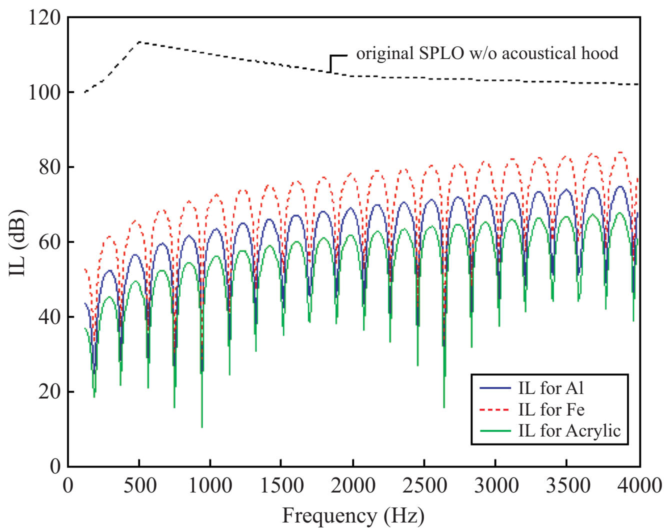


Fig. 8. Optimal *IL* for acoustical hoods (broadband noise).

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 a 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 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
$iter$	the maximum iteration
IL_j	the sound insertion loss for the j-th acoustical hood (dB)
k	wave number (= ω/c_o)
kk	cooling rate in *SA*
OBJ_i	objective function
pb(T)	transition probability
SPL_T	the sound pressure level after adding an acoustical hood
T	the temperature in simulated annealing
$\rho_o c$	the acoustic impedance
η	the panel's internal damping coefficient
ν	the poison ratio of the panel
ρ	the panel's density.

ACKNOWLEDGMENTS

The author acknowledges the financial support of the National Science Council (NSC100-2622-E-235-001-CC3 and NSC102-2221-E-235-001, Taiwan, ROC).

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