The Design and Analysis of Ultrasonic Roll Welding Head and Tool

Kuen-Ming Shu[1], Wen-Hsiang Hsieh[2], Yu-Guang Li[3], Chi-Wei Chi[4], Yi-Shen Li[5]*

[1] Graduate Institute of Mechanical and Computer-aided Engineering, National Formosa University

Email: kmshu@nfu.edu.tw

Abstract - Solar panels conduct electricity through aluminum strips on substrate surfaces. Ultrasonic roll welding can weld the conductive aluminum strips onto the glass substrates. This paper illustrates vibration characteristics and optimal design of amplitude horns used in the ultrasonic welding roll. Based on theoretical equations, this study used the ANSYS software to establish the parametric model according to design requirements. With the parametric model as the initial design, this study conducted modal analysis and harmonic analysis to obtain the vertical mode and disc bending mode of the horn, and measured the resonant frequency, amplitude amplification rate and stress distribution. Finally, this study implemented and verified the optimal coupled disc tool of the ultrasonic horn.

Keywords: **ultrasonic roll welding, horn design, vibration characteristics, optimal design**

I. INTRODUCTION

The bonding methods of conductive aluminum strips with solar panels currently available on the market can be divided into two types: the traditional method uses silver plastic bonding, and the other green technology uses the ultrasonic vibration system for welding. The latter method is newly emerging, cost-saving and environmentally friendly. The principle of ultrasonic roll welding is to generate vibration by transmitting high frequency sound waves to piezoelectric ceramic and resonate with the disc by transmitting the vibration in the form of vertical waves through the amplitude horn mounted on the front end of the transducer. The amplitude is amplified in the form of horizontal waves inside the disc to achieve friction welding. Through the overall pivotal movements, the roll welding of the solar panels on the conveyor can be realized. In addition, the transducer displacement amplitude during the process of converting electrical energy into mechanical energy is usually 4~10μm. Meanwhile, horns of various ratios of gains can be designed according to different purposes with various output amplitude values.

This study connected the vibration amplitude horn for welding metallic materials with the front-end tool (bending vibration disc) to design an ultrasonic horn of longitudinal bending vibration mode suitable for continuous plane rolling movements.

II. LITERATURE REVIEW

Most of past studies on ultrasonic welding focused on the vertical vibration horns for welding plastic materials without considering the front-end tool. In 1994, Ahmed et al.[1] conducted optimal design of double cone-shaped horn using the ANSYS software to maximize its amplitude amplification rate. In 2007, Vinod et al.[2] studied the horn for ultrasonic milling, and applied the finite element software to simulate the differences of horns of 20~24kHz in terms of axial stress, confirming that horn of higher frequency has higher horn stress consequently. In the same year, Chu [3] conducted the harmonic analysis of the 20kHz milling tools using the Marc finite element analysis software. The amplitude requirements for milling was in the range of 8~16μm. With force as the amplitude input source, the exciting force as suggested by repeated tests was 6000N and the output amplitude was amplified to 15μm. The input force can be effectively transmitted to the output end surface and thus the milling tool satisfying the requirements was successfully developed. In 2009, Yongbo et al. [4] used the finite element software to develop a pivotal ultrasonic vibration system integrating transducer, concentrator, horn and tool, and measured the natural frequency and displacement using the Doppler laser meter. The developed system was in line with system demands, proving that it is feasible and efficient to apply the finite element software in the design of ultrasonic vibration system.

III. ANSYS SIMULATED OPTIMAL DESIGN

As the finite element analysis results are very close to design objectives, the variable with the maximum impact on the design objectives was selected from the six design variables for fine tuning. According to the gradient approach, Variables x3, x4, and x5 have the greatest impact on disc outer ring amplitude, overall stress and resonant frequency respectively. As a result, they are selected as the adjustment factors to define the design objectives and constraints:

1) The resonant frequency of the disc tool coupled by horn is 35000+9=35009Hz.
2) The range of amplitude of disc outer ring vibration is ±5μm~12μm.
3) Because it is a vibration system, the safety factor is set as 2.5, and the allowable stress should be less than 138MPa.

This is a problem of non-linear planning with constraints, and it can be represented by the following equation:

(1)

$$
\begin{aligned}
&Minimize \quad f(x_3, x_4, x_5) = F - 35009\\
&subject\ to \quad g_1 = 29.7266 \le x_3 \le 33.7266\\
&\qquad\qquad\quad g_2 = 43 \le x_4 \le 47\\
&\qquad\qquad\quad g_3 = 32.5 \le x_5 \le 36.5\\
&\qquad\qquad\quad g_4 = 0.005 \le |A_c| \le 0.012\\
&\qquad\qquad\quad g_5 = \sigma_{\max} - 138000000 \le 0
\end{aligned}
$$

By inputting the above objective function, design variables and constraints into the ANSYS setting, this study obtained three groups of feasible designs closest to the design objective, as shown in Table 1, using the random search method of 500 times of iteration. It set the feasible solution closest to design objective as the initial value. In the previous sensitivity analysis, the level of impact of three variables on disc outer ring amplitude, overall stress and resonant frequency is x5 > x4 > x3 by the given order. As fewer design variables are better, x3 is set fixed and the rest two variables are fine-tuned to search the optimal solution using the sub-problem approximation method. The allowable value of design variables is set as 0.1, the status variable allowable value is set as 0.0001, and the target function allowable value is set as 0.1. Figure 1 illustrates the optimization objective function's iterative process, which converges in the 10th iteration and its optimal size is as shown in Table 2. The optimal design's modal analysis and harmonic analysis results are as shown in Table 3. The improvements before and after optimization are as shown in Table 4.

Table 1 Feasible design

Set	x_3	x_4	x_5	F (Hz)	A_c (μm)	σ_{max} (MPa)
12	31.198	43.088	34.313	34983	5.0447	93.00
76	32.666	45.842	33.808	34959	5.9143	87.55
449	31.763	46.803	33.835	34911	5.1518	87.96

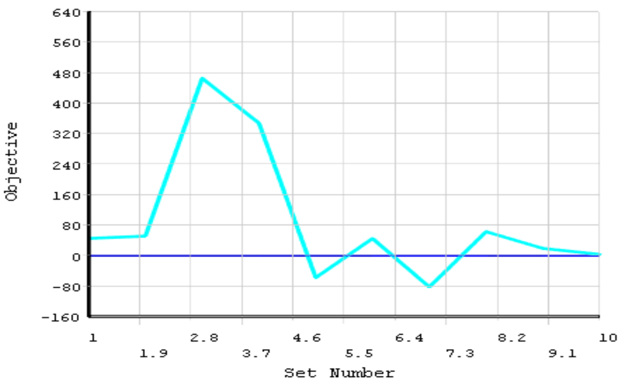


Figure 1 Converging curve of objective function

Table 2 Sizes of optimal design

D_1 (mm)	D_2 (mm)	l_1 (mm)	l_2 (mm)	r (mm)	h (mm)
24	16	31.198	43.123	34.287	4.8

Table 3 Results of modal analysis and harmonic analysis of optimal design

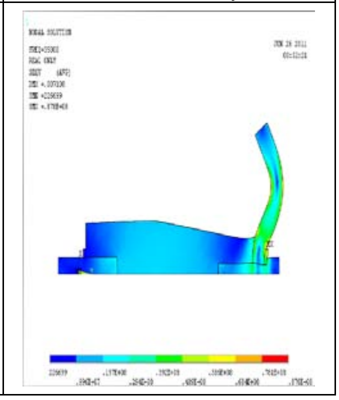
Optimal design		
F (Hz)	A_c (μm)	σ_{max} (MPa)
35016	5.1	87.8
Modal Analysis	Harmonic Analysis	
		

Table 4 Improvement results of optimization

	F (Hz)	A_c (μm)	σ_{max} (MPa)
Original design	34355	4.9	82.6
Optimal design	35016	5.1	87.8

By converting the optimized sizes into a physical 3D model via Solid95 elements, coupled with the previously reserved wrench gripping position, the resonance frequency obtained was 34902Hz. By comparison with the simulation by Plane42 elements of the 3D axisymmetric 35016-9=35007Hz, the error is only 0.3%. The optimized 3D vertical bending resonance mode is as shown in Figure 2.

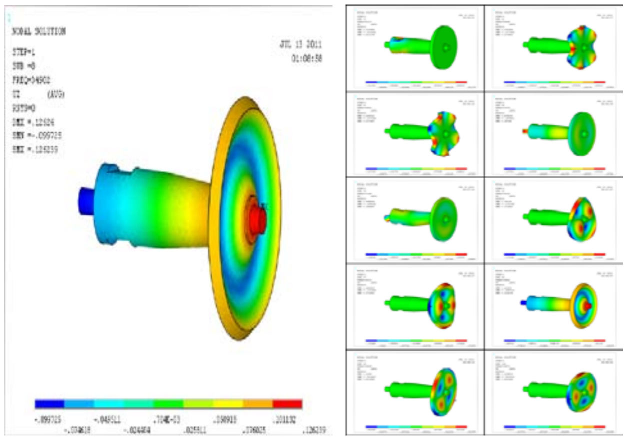


Figure 2 Optimized vertical bending resonance mode

IV. EXPERIMENTAL METHOD AND PROCEDURES

This study used CAD software to draw the three-view diagram, which was delivered to the King Ultrasonic Co., Ltd for manufacturing. Figure 3 illustrates the engineering drawings of the optimal design, and Figure 4 illustrates the finished product after processing.

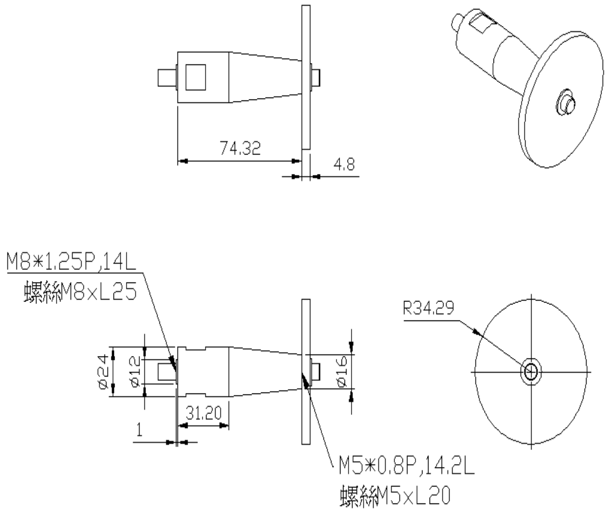


Figure 3 The optimal design engineering drawings of disc-coupled uniform section cone-shaped horn

Figure 4 Finished product by optimal design of disc-coupled uniform section cone-shaped horn

A. Experimental Equipment

The experimental equipment consists of audio signal generator, ultrasonic vibration group, digital frequency counter, ultrasonic oscillator, and fiber optic displacement meter etc. The photos of the experimental and measurement equipment are as shown in Table 5.

1) GW GAG-809 audio signal generator: it is used to send the ultrasonic vibration group at 35 kHz sine wave signal to vibrate the horn.
2) K-SONIC WAS-0935 ultrasonic vibration group: the overall structure contains the transducer (piezoelectric ceramic), concentrator, horn, functioning for the assembly of horn.
3) GW GFC-8010H digital frequency counter: it is used to bridge the audio signal generator, piezoelectric ceramic and digital frequency counter to observe the resonance load display for the measurement of horn resonance frequency.
4) K-SONIC KW AS-0635ultrasonic oscillator: it is purchased from King Ultrasonic Co., Ltd for the purpose as stated in (1).
5) MTI-2100 fiber optic displacement meter: it is used to measure the horn displacement amplitudes at the input end, the output end and the disc outer ring to calculate the amplitude amplification rate.

Table 5 Experimental materials and measurement equipment

Audio Signal Generator	Ultrasonic Vibration Group
	
Digital Frequency Counter	35kHz Ultrasonic Oscillator
	
Fiber Optic Displacement Meter	Quartz Sand
	

B. Resonance Frequency Measuremnet and Modal Experiment

This study assembled the audio signal generator and digital frequency counter by bridging, and connected input end of the digital frequency counter with the positive and negative poles of the vibration group. The established instrument is as shown in Figure 5. The relevant parameters and observed the resonance load display during the fine-tuning process were set. When the resonance load display is zero, the displayed figure on the screen of the digital frequency counter represents the natural frequency of the horn.

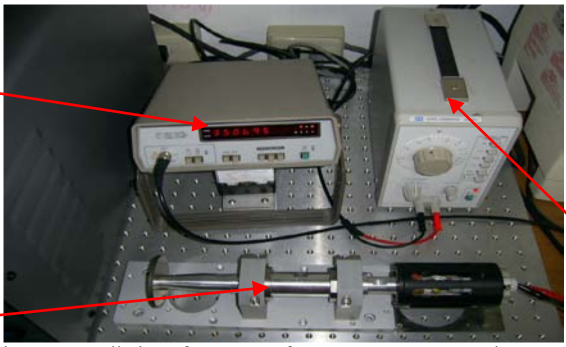


Figure 5 Installation of resonance frequency measurement instrument

V. RESULTS AND DISCUSSION

The measurement results suggested that the horn-coupled disc resonated at 35070Hz and the ANSYS simulated value of the optimal design is 35007Hz. The error is only 0.2%, and is judged as caused during the manufacturing process. After confirming the resonance frequency of the horn-coupled disc around 35kHz, it can be connected with the purchased 35kHz ultrasonic oscillator. For the convenience of the follow-up modal verification experiment, the vibration group was first located vertically on the load slide support as shown in Figure 6. Quartz sand was spread on the disc tool to observe the disc vibration. As shown in Figure 7, quartz sand appeared in Chladni patterns of two circles. Places of evenly concentrated quartz sand represent the displacement amplitude is zero, namely, the nodal and places without quartz sand distribution represents the places of maximum displacement amplitude (i.e., the anti-node). The experiment proved that the disc tool is doing the second bending vibration, which is the same with the mode of the ANSYS harmonic analysis of the optimized example design.

Figure 6 Vertical installation of the vibration equipment group

Figure 7 Pitch circles of vibration disc

A. Displacement Amplitude Measurement

As the input amplitude of the ultrasonic vibration system cannot be guaranteed as 4μm, the experimental method measured the displacement amplitude at the concentrator end (horn input end) using the fiber optic displacement meter before measuring the displacement amplitude at the horn output end. The division of the output end amplitude and the input end amplitude can lead to the amplification rate of the horn amplitude. The actual measurement is as shown in Figure 8. In addition, the disc outer ring plane is divided into eight parts and marked as shown in Figure 9. The average value of the eight amplitude values is the displacement amplitude of the disk outer ring too and the amplitude loss rate is defined as (η):

$$\eta = \frac{A_{output} - A_c}{A_{output}} \times 100\% \qquad (2)$$

Figure 8 Measurement of displacements at input/output ends of the horn

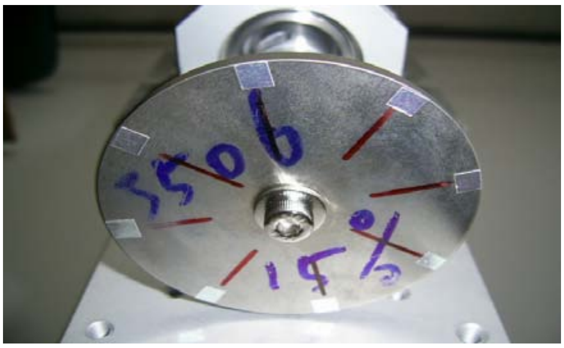


Figure 9 Disc outer ring measurement points

the measurement results suggested that the horn input end displacement is 3.7μm, the output end displacement is 5.5μm, namely, the horn amplitude amplification rate is 1.5, the average disc outer ring amplitude is 4.5μm, the amplitude loss rate is about 18%. Compared with the amplitude loss rate at 15% of the ANSYS harmonic analysis, the error is 3%. The amplitudes at the eight measurement points at the disc outer ring are as shown in Table 6.

Table 6 Distribution of disc outer ring amplitudes

Measurement Point	Disc Outer Ring Amplitude(μm)
1	4.6
2	4.2
3	4.3
4	4.4
5	5.0
6	4.2
7	4.9
8	4.7
Avg.	4.5

B. Comparison of Optimal Simulated Values

The results of comparing the vibration measurement experiment and the ANSYS optimal simulation values suggested that the overall resonance frequency and disc mode are consistent with the simulated values. However, the disc outer ring amplitude error is relatively large. This is due to too high sensitivity of the fiber optic displacement meter. In addition, as the disc tool was processed by heating and the material parameters cannot be guaranteed as 100% with the ANSYS settings.

VI. CONCLUSIONS

This study discussed the design and analysis of the welding head and tool of ultrasonic roll welding and the conclusions are as follows:

1) The proposed systematic ultrasonic horn and tool design and analysis method is proved as accurate and feasible by simulation and experiments.

2) The theoretical design equations of horn and tool are proved as accurate by the vibration analysis. This study also summarized the first order axial resonance sizes of uniform section connection with cone-shaped horn and the finger-shaped horn for the reference of manufacturers and researchers in the future.

3) This study could develop the optimization model of the horn coupled tool rapidly to replace the complex mathematical model inference process through the optimization model provided by ANSYS.

4) The design focuses should be the horn resonance length and the radius of the disc tool in both cases of removable or one-piece design.

VII. ACKNOWLEDGEMENTS

The authors would like to the support of the Ministry of Education Industry-University Cooperation Program (Project 100M-43-011) and the assistance of Murano Industrial Co., Ltd.

REFERENCES

[1] Amin S.G., Ahmed M.H.M., and Youssef H.A.," Computer-aided Design of Acoustic Horns for Ultrasonic Machining Using Finite-element Analysis," *Journal of Materials Processing Technology*, Vol. 55, pp.254-260, 1995.

[2] Yadava V. and Deoghare A.," Design of Horn for Rotary *Ultrasonic* Machining Using the Finite Element Method," *The International Journal of Advanced Manufacturing Technology*, Vol. 39, pp.9-20, 2008.

[3] Chu, H.C., A Study on Fundamentals and Applications of Ultrasonic Milling Technology, doctoral dissertation, Nanjing University of Aeronautics and Astronautics, 2007.

[4] Wu.Y., Yokoyama S., Sato T., Lin W. and Tachibana T., "Development of a New Rotary Ultrasonic Spindle for Precision Ultrasonically Assisted Grinding," *International Journal of Machine Tools & Manufacture*, Vol.49, pp.933-938, 2009.