Available online at www.sciencedirect.com
SciVerse ScienceDirect
AASRI Procedia 3 (2012) 414 – 420
AASRI Procedia
www.elsevier.com/locate/procedia

2012 AASRI Conference on Modeling, Identification and Control

Structural Analysis and Experimental Research of an CNC Hydraulic Swing-type Plate Shears

Yong Wang[1], Baoping Cui[1,2], Kun Li[1,2], Teng Zhang[1], Zufang Zhang[1,2*]

[1]*School of Mechanical and Automotive Engineering, Hefei University of Technology, NO.193, Tunxi Road, Hefei, 230009, China*
[2]*High-tech Research Institute of Hefei University Of Technology (Maanshan), NO.578, Taibai Road, Maanshan, 243000, China*

Abstract

CNC hydraulic swing-type plate shears is one of the important plate processing equipments, whose safety performance and shearing accuracy are determined by the structure of its frame and tool carrier. The current mechanical property analyses of plate shears mostly only use finite element tools and simply apply uniform load, which can not reflect true work status of the plate shears. By integrating original structures of the frame and tool carrier, two working status' static performances of the plate shears are analyzed with finite element method in this paper, where the uniform load or moving load are separately applied according to its work statue. The analysis results are verified by the later stress-strain tests according to which structural improvements are proposed to provide basis for improving properties of the plate shears.

Keywords: Plate shears, uniform load, moving load, stress-strain test, structural improvement

Introduction

Swing-type plate shears is a metal processing equipment, which fractures plates into required sizes through relative motion and reasonable clearance between the movable blade and static blade. With the rapid development of China's industry, individualized needs of plate shears grow rapidly. The frame and the tool carrier are the major components of the plate shears, whose strength and stiffness directly determine the safety performance and shearing accuracy.

Based on empirical formulas and parameters, majority of current domestic plate shears are designed by abiding the traditional theory and rules. However, stress concentration and overall force distribution of the machines can not be accurately obtained according to the traditional design methods, which leads to high-end performance requirements unsatisfiable[1]. To solve this problem, plate shears mechanical properties are studied at home and abroad. Stiffness and strength of the frame were obtained by Xiangjun Gu according to the linear analysis and modal analysis with ANSYS[2]. Through analytical processing the model established in ANSYS, Xinhua Yang improved structure and mechanical properties of the machine[3]. By programming information of each step of the moving load in ANSYS, deformation, equivalent stress and equivalent strain of the frame were studied by Yingying Liu[4]. But most of the literatures only theoretical analyzed with uniform load, which didn't analyze their

* Corresponding author. Tel.: 13965037586.
E-mail address: zufangzh@163.com.

2212-6716 © 2012 The Authors. Published by Elsevier B.V. Open access under CC BY-NC-ND license.
Selection and/or peer review under responsibility of American Applied Science Research Institute
doi:10.1016/j.aasri.2012.11.065

actual shearing conditions or propose feasible suggestions for improvement.

By taking QC12Y-4×2500 plate shears as an example, static analyses of the frame and tool carrier are carried out separately with uniform load and moving load. According to the later stress-strain tests which verify the analysis results, structural improvements are proposed to improve properties of the plate shears.

1. Bearing Load Calculation of the Plate Shears

1.1 Shearing Force

Shearing force of beveled blade shear can be calculated through the Noshari formula[5]

$$P = 0.6\sigma_b\delta_x\frac{h^2}{\tan\Phi}\left(1 + Z\frac{\tan\Phi}{0.6\delta_x} + \frac{1}{1+\frac{10\delta_x}{\sigma_b Y^2 X}}\right) \qquad (1)$$

where σ_b is the ultimate strength of sheared plate (MPa) ; δ_x is the elongation of sheared plate; h is the thickness of sheared plate (mm) ; Φ is the shear angle (°) ; Δ is the blades clearance (mm) ; C is the distance between binder foot axis and static blade (mm); X is the relative distance of binder foot, where $X = C/h$; Y is the relative clearance of shear lateral, where $Y = \Delta/h$; Z is the bend coefficient which is related to the sheared length L, shear angle Φ, shear plate thickness h and metal elongation δ_x .

Table 1. Value of the required coefficients

coefficient	σ_b (MPa)	δ_x	h (mm)	Φ	Δ (mm)	C(mm)	X	Y	Z
value	450	0.25	4	1.5	1.2	60	15	0.3	0.95

Total shearing force P can be obtained to be 89074N according to the coefficient values as shown in Table 1.

1.2 cylinder thrust, bearing reaction force and shear horizontal thrust

Since forces on the frame and tool carrier are action and reaction, structure of the tool carrier can be simplified to the mechanical model[3] as shown in Fig. 1.

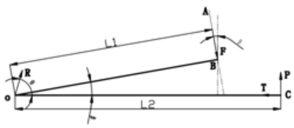


Fig. 1. Diagram of the tool carrier force

According to the location of all forces shown in Fig. 1 and shear horizontal thrust T=0.3P=26722.2N obtained from empirical formula, equations of the torque balance and force balance can be written as follows

$$\begin{cases} F \times L_1 - P \times L_2 = 0 \\ R \times \cos\theta + F \times \sin\beta - T = 0 \\ R \times \sin\theta - F \times \cos\beta + P = 0 \end{cases} \qquad (2)$$

where cylinder thrust F=115098.2N, which is equal to the support-cylinder force. Bearing reaction force R=25204.5N, which is equal to the radial force of support tool carrier.

1.3 Binder Force

Binder force can be determined according to the empirical formula as follows

$$P_y = k \times h \times b \times 10^{-3} t \tag{3}$$

where k is a coefficient, for hydraulic pressure feeder, k=0.8～1.1, for mechanical pressure feeder, k=0.5～0.6, h is the thickness of sheared plate (mm), b is the width of sheared plate (mm). P_y can be obtained by the formula above: P_y=100000N.

2. Finite Element Analysis of the Swing-type Plate Shears

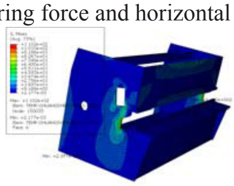
2.1 Finite Element Model Establishment and Pre-processing of the Frame and Tool Carrier

The frame and tool carrier models are separately built by relevant software and then imported into finite element software[6]. Parameters of the frame and tool carrier are selected according to their materials: plain carbon steel Q235, where elastic modulus is $2.1 \times 10^5 \mathrm{MPa}$ and Poisson's ratio is 0.3.

The frame's underside is full-constrained based on the actual situation. Two reference points, whose degrees of freedom are constrained only to rotate around the X axis in local coordinate system, are established at the centers of left and right plate of tool carrier and coupled with two holes separately. In order to remove rigid displacement of the tool carrier, translation constraint of y, z directions are applied to the contact point between the cylinders and tool carrier in the global coordinate system[7]. The frame bed-body is meshed with tetrahedral element, while the beams and tool carrier are meshed with hexahedral element.

2.2 Frame Analysis

2.2.1 Applying uniform load of shearing force and horizontal thrust on frame



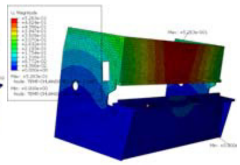


Fig. 2. Frame stress contour; Fig. 3. Frame displacement contour

Based on the analysis result, maximum stress of the frame which occurs at the throat fillet is 110.2Mpa as shown in Fig. 2, while the maximum displacement which occurs at the middle of the front vertical plate is 0.5261mm as shown in Fig. 3.

2.2.2 Applying moving load of shearing force and horizontal thrust on frame

15 points are arranged along the static tool apron and coupled with the underside and front of the static tool apron. Shearing force and shear thrust are respectively applied and divided into 16 load steps. The inheritance between forces are abolished by setting in the load manager to make the forces taking effects in their single steps[8].

According to analysis results, the maximum stresses and displacements of the frame under every load step are listed in Table 2.

Table 2. Maximum stress and displacement table of every load step

Step	1	2	3	4	5	6	7	8
Stress	71.71	110.6	110.3	110	109.7	109.4	109.7	109.9
Displacement	0.1634	0.5264	0.5264	0.5264	0.5264	0.5264	0.5264	0.5264

Step	9	10	11	12	13	14	15	16
Stress	110.2	110.5	110.8	111.1	111.4	111.7	112	112.2
Displacement	0.5264	0.5264	0.5264	0.5264	0.5264	0.5264	0.5264	0.5264

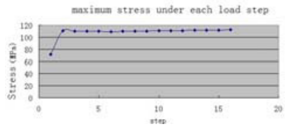


Fig. 4. Variation diagram of every load step stress

Informations can be drawn from the stress and displacement contours as follows:

1) When the shearing force and horizontal thrust haven't been applied in step-1, the minimum stress occurring at the throat fillet is 71.71MPa, while the minimum displacement occurring at the top place in the front vertical plate of the frame is 0.1634mm.

2) It's shown in Fig. 4 that the whole stress change in a slowly upgrade tendency during the shearing position shifting right. The maximum stress which also occurs at the throat fillet achieves 112.2MPa at step-16, relative increased by 1.8% to uniform load condition. Meanwhile the larger stress domain in front vertical plate shifts right. The maximum displacement remains unchanged at 0.5264mm in the middle of front vertical plate. But the deformation domain of the frame horizontal stand shifts right as the load step increasing.

3) In the load step-9, the stress and displacement changes are consistent with uniform load.

2.3 Tool Carrier Analysis

Shearing force pressure and horizontal thrust pressure are respectively applied on the top and back surface of the blade mounted position.

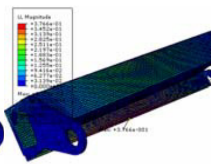



Fig.5. Tool carrier stress contour; Fig. 6. Tool carrier displacement contour

It is shown in fig. 5 that the maximum stress is 57.52MPa, which occurs at the contact position between shear cylinder and wing boards. In addition, bottom of the region where the wing boards being welded with the front plate is a relatively larger stress area.

As it shown in fig. 6 that the maximum displacement is 0.377mm, which occurs at the lower end of the intermediate position of the front plate. The upper end of the intermediate position and the two ribs are also the bigger displacement area.

3. Experimental Verification of Plate Shears for Finite Element Analysis

The stress-strain test on the frame of plate shears is accomplished with DH3816 static strain gauge and 45° rectangular rosette. Fifteen testing points are arranged at the four corners of the cylinder mounting holes on the frame, throat fillet, bearing hole, front vertical plate and lower area of workbench, where larger stresses and displacements occur. Four specifications of shear plates are tested according to four different thicknesses and lengths of sheared plates. The test strain are calculated into stress based on the formulas as follow

$$\begin{cases}\text{Principal strain } \varepsilon_{1,2}=\left(\varepsilon_0+\varepsilon_{90}\right)/2\pm 1/2\left[(\varepsilon_0-\varepsilon_{90})^2+(2\varepsilon_{45}-\varepsilon_0-\varepsilon_{90})^2\right]^{1/2}\\ \text{Principal stress } \sigma_{1,2}=E/2\times\left[\left(\varepsilon_0+\varepsilon_{90}\right)\Big/(1-\mu)\pm 1\Big/(1+\mu)\times\left[(\varepsilon_0-\varepsilon_{90})^2+(2\varepsilon_{45}-\varepsilon_0-\varepsilon_{90})^2\right]^{1/2}\right]\\ \text{the angle between } \sigma_1 \textit{ and } \text{horizontal line } \varphi=1/2\times\tan^{-1}\left[(2\varepsilon_{45}-\varepsilon_0-\varepsilon_{90})/(\varepsilon_0-\varepsilon_{90})\right]\\ \text{Maximum shear stress } \tau_{\max}=E\times\left[(\varepsilon_0+\varepsilon_{90})^2+(2\varepsilon_{45}-\varepsilon_0-\varepsilon_{90})^2\right]^{1/2}\Big/(1+\mu)\end{cases} \quad (4)$$

Corresponding loads of the fifteen testing points are respectively calculated and compared with finite element analysis results based on four different sheared plates. The comparison table between analysis results and calculation data of the five testing points around throat fillet is listed as follows.

Fig. 7. Picture of on line test site

Table 3. Comparison table between testing and calculating data

Specifications	Test point	1.5×2000	3.5×2100	5.8×73	5.8×510
	Calculated value	10.19	36.19	58.61	63.64
5	Test value	7.65	20.83	46.03	61.50
	Error rate	24.93%	42.44%	21.46%	3.36%
	Calculated value	8.56	30.85	52.27	56.17
6	Test value	10.65	26.14	52.34	68.54
	Error rate	24.36%	15.27%	0.14%	22%
	Calculated value	8.18	29.18	47.36	51.45
7	Test value	7.95	17.39	32.07	42.54
	Error rate	2.78%	40.42%	32.29%	17.32%
	Calculated value	10.35	38.79	68.73	73.49
8	Test value	9.03	22.90	50.69	67.40
	Error rate	12.77%	40.97%	26.25%	8.29%
	Calculated value	11.74	44.65	81.29	86.49
9	Test value	9.39	29.66	58.98	75.12
	Error rate	20%	33.57%	27.44%	13.14%

Informations can be drawn from the comparison that the results of analysis and test have value gaps around 20% because of taking the environmental influences from test site into account. Such as the shearing force is calculated based on traditional empirical formula, whose value is bigger than actual value and leads to greater shear stress in later finite element analysis. The overall data are within the allowed range except individual testing points. The conclusion can be drawn that the forgoing finite element method[9] is reliable and the calculation results of maximum stress of throat fillet are also accords with the actual working mode.

4. Implementation of Structural Improvement for Plate Shears

According to the analysis and testing results, structural improvement suggestions for the Hydraulic Swing-type plate shears QC12Y-4×2500 are given as follows.

In addition to the throat fillet of frame and shear cylinder contact position of tool carrier, the mechanical properties of the structural material Q235 can meet the maximum force condition requirements, which means the plate shears has a certain range for weight reduction, for example the frame wallboard and tool carrier wing boards can be appropriate reduced.

Aiming at reducing the stresses of the throat fillet and shear cylinder contact position of tool carrier, reinforcing plates can be welded to partly strengthen the two larger stress positions as shown in Fig. 8 and Fig. 9.

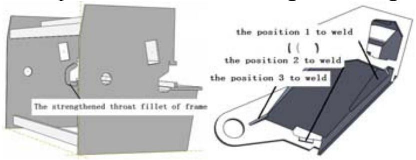


Fig. 8. Sketch of frame improvement; Fig. 9. Sketch of tool carrier improvement

5. Summary

Based on the existing structure of frame and tool carrier of the hydraulic swing-type plate shears QC12Y-4×2500 and the actual shearing process, uniform load and moving load are applied and analyzed by using load step in the finite element analyses. Through comparing the two status, it is indicated that the moving load arrangement more fully reflect the impact from shearing process on its performance. Experimental data of five testing points of throat fillet are extracted and contrasted with the analysis results which are later verified to be reliable.

This swing-type plate shears can be normally used under the existing structure of frame and tool carrier. In order to improve the plate shears properties and reduce weight of the plate shears, the thickness of frame wallboard and the tool carrier wing-boards can be appropriately decreased, and reinforcing plates can be welded to partly strengthen the throat fillet and shear cylinder contact position of tool carrier.

In conclusion, the research in this paper can provide a reference for size design, finite element analysis as well as structural improvements of plate shears, and has engineering reference value for improving shearing quality and guiding the actual production.

Acknowledgments

The paper is funded assistant by Anhui key laboratory construction foundation (2011AKSY1119), innovative service platform for high-end equipment manufacturing technology construction foundation (11Z0201025) and foundation from Maanshan science and technology bureau. Design and testing works presented in this paper are some of the result of a cooperative research effort and the authors would like to acknowledge the valuable discussions and suggestions from the members of group.

References

[1] MAO Zhiqiang, WANG Fei, Finite element analysis of cutter holder on swing shearing machine[J], EQUIPMENT(in chinese), 2012, 01: 31-33.
[2] GU Xiangjun, ZHANG Weiqiang, Analysis for the Cutter Frame Based on ANSYS[J], Science Technology and Engineering(in chinese), 2010, 2(10): 476-478.
[3] Yang Xinhua, Structural Finite Element and Modal Analysis of Plate Shearing Machine[D], Huazhong University of Science and Technology, Wuhan(in chinese), 2009, 05.
[4] LIU Yingying, WANG Qiang, YANG Jinsu, XU Jisheng, Finite Element Analysis of Hydraulic Shears' Tool Carrier Based on ANSYS[J], Modular Machine Tool & Automatic Manufacturing Technique(in chinese), 2009, 6: 35-39.
[5] HUANG Huaqing, Rolling Machinery[M], Beijing, Metallurgical Industry Press(in chinese), 1979.

[6] Zienkiewicz, O.C. The Finite Element Method. 3rd ed., London: McGrew-Hill, 1977.
[7] Xiaojun Huang. Computation on shearing machine Q12y-12×3200 by static and dynamic Finite Element Method and optimal design. Xi'an: Xi'an Jiaotong University(in chinese), 1998.
[8] Yiping Shi, Yurong Zhou. ABAQUS finite element analysis of detailed examples. Beijing: Machinery Industry Press(in chinese), 2006, 6.
[9] Clough R W. The finite element methon in plane stress analysis. Proc. 2nd Conf. Electronic Computation, ASCE, Pittsburg, 1960, 9.