LINEAR BUCKLING ANALYSIS OF CYLINDER RODS USED ON INDUSTRIAL 300 TONS H-TYPE HYDRAULIC PRESS

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Abstract- In this study, buckling analyzes of auxiliary cylinders and ram cylinder which are the most important parts of industrial 300 tons H-type hydraulic press are realized for safety factor calculations. For this purpose; linear buckling analyzes of cylinder rods of auxiliary cylinder and ram cylinder are performed by using finite element method. As a result; maximum Von Misses stress locations, safety factors of static analysis, maximum deformation results and safety factors of buckling of cylinders are determined via ANSYS Workbench software.

Keywords- Hydraulic Press, ANSYS, Buckling Analysis

I. INTRODUCTION

In this paper, an industrial 300 tons H-type hydraulic press of manufacturer given in Fig.1 is chosen and structural analysis of press is realized for geometric optimization. For this aim, linear static analysis of press body parts is realized and maximum Von Misses stress locations, safety factors, maximum deformation results and required optimization locations are determined via ANSYS Workbench finite element software. All obtained results are useful and realistic for chosen press manufacturer company to decrease using raw material for press production.

Metal forming machines and presses are one of the classic applications of hydraulic science and they are used in many branches of the industry for high quality and series production. New materials, products and new manufacturing process are new areas of application for press technology. Major power is provided by using hydraulic in presses for effective and high-volume production. Today, hydraulic presses which are the most important part of the industrial hydraulic, are used in iron and steel industry such as plastering, twisting, extrusion and forging process [1].

However, some mechanical problems occur in the hydraulic press manufacture and use. These critical problems are investigated and listed below:

1) Time to time cracking, plastic deformation and fracture problems are seen in press body and components. This situation prevents the performance of press, it leads to the deformation and breakage of the molds and decreases the efficiency and capacity of hydraulic press.
2) The some values are given about power, capacities and fatigue behavior of the press but these values cannot be validated exactly.

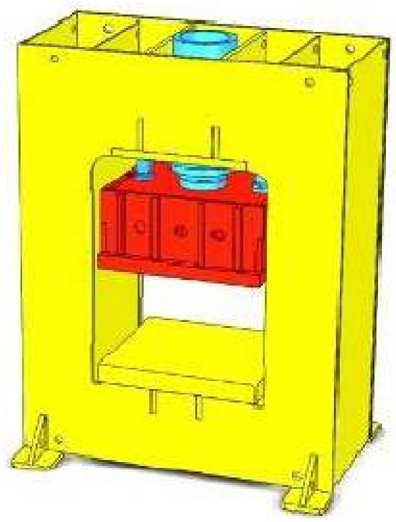


Fig.1. 300 tons H-type hydraulic press

3) Press types are increased because of every manufacturer products different type and size of press for customer desire. But increased production rate disrupt standard production of this kind of machines. Also manufacturers use lots of raw materials to product without using engineering calculation.
In the light of all, these problems are considered and a large market, literature review and relationship with manufacturers are realized to help solving problems. As a result of investigations it can be said that engineering knowledge and realistic methods or formulas have not been used in press industry exactly [2]. In literature review, different studies have been made about the hydraulic presses such as design and mechanical analysis. But the most important and similar studies are considered and given in this study. Arslan [3] studied the structural analysis of the body of an eccentric press using ANSYS software. Yağbasan [4] realized finite element analysis of C-type hydraulic press body. Köseler [5] in his study, examined the design and analysis of three effective high-speed hydraulic press. Raz et al. [6] analyzed

stress-strain of hydraulic press components using finite element method. Zahalka [7] studied the modal analysis of a hydraulic press. Zhang et al [8] have implemented structural optimization of the hydraulic press. Again, Zhang et al. [9] investigated mechanical analysis of the cylinder block of a hydraulic press.

II. BUCKLING ANALYSIS

Many structures require an evaluation of their structural stability. Thin columns, compression members, and vacuum tanks are all examples of structures where stability considerations are important. At the onset of instability (buckling) a structure will have a very large change in displacement {□x} under essentially no change in the load (beyond a small load perturbation).

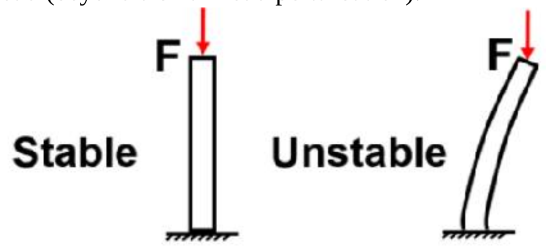


Fig.2. Linear eigenvalue buckling cases

Eigenvalue or linear buckling analysis predicts the theoretical buckling strength of an ideal linear elastic structure. This method corresponds to the textbook approach of linear elastic buckling analysis. The eigenvalue buckling solution of a Euler column will match the classical Euler solution. Imperfections and nonlinear behaviors prevent most real world structures from achieving their theoretical elastic buckling strength. Linear buckling generally yields unconservative results by not accounting for these effects. Although unconservative, linear buckling has the advantage of being computationally cheap compared to nonlinear buckling solutions. Stable and unstable linear eigenvalue buckling cases are shown in Fig.2. For a linear buckling analysis, the eigenvalue problem below is solved to get the buckling load multiplier () and buckling modes () and these are shown in equation (1).

$$([K]+\lambda_i[S])\{\psi_i\}=0 \qquad (1)$$

[K] and [S] are assumed constant, elastic material behavior is assumed linear and small deflection theory is used, and no nonlinearities included. A Static Structural analysis will need to be performed prior to (pre-stress analysis) a buckling analysis. Solid, Surface (with appropriate thickness defined), Line (with appropriate cross-sections defined) geometries supported by Workbench Mechanical may be used in buckling analyses, Although point masses may be included in the model, only inertial loads affect point masses, so the applicability of this feature may be limited in buckling analyses. One structural load which causes buckling, should be applied to the model and, at least Young's Modulus and Poisson's Ratio are required material properties for buckling analysis.

All structural loads will be multiplied by the load multiplier (□) to determine the buckling load (see below). The structure should be fully constrained to prevent rigid-body motion. In a buckling analysis all applied loads (F) are scaled by a multiplication factor (□) until the critical (buckling) load is reached. Buckling load is calculated by the equations (2) and (3).

$$F \cdot \lambda = p'(Buckling\ load) \qquad (2)$$

$$p' = k \cdot \left| \frac{\pi^2 \cdot E \cdot I}{L^2} \right| \qquad (3)$$

The buckling values should be investigated as the rods belonging to the cylinder are columns exposed to the compressive load. Both the theoretical and the finite element analysis have been explained how to investigate the buckling load. In this direction, buckling loads of auxiliary cylinder and ram cylinder were investigated and mode shapes were observed. It is quite simple to theoretically examine the buckling load in linear problems. However, in hydraulic cylinder geometries it is difficult to examine this process with a classical hand account due to both the absence of a straight column and the boundary conditions. It is also possible to observe mode shapes as well as load multiplication via simulation. For buckling analysis, the model must first be analyzed statically.

III. RESULTS AND DISCUSSION

The mathematical model, boundary conditions and loading inputs have been solved for the completed part. Von Mises strain and total displacement results were obtained as static solution results. In Fig.3. the maximum stress value for the auxiliary cylinder is measured as 146 MPa (safety factor 4). The maximum stress value for ram cylinder was measured at 96 MPa (Safety factor 6).

As shown in Fig.4. the total displacement results of the rods have a displacement of only about 0.3 mm due to vertical axial pressure alone. But this is not the result of column and shaft displacements that are actually subjected to pressure and pull loads. Therefore, buckling analysis was performed for the safety factor and the lowest value was accepted as the safety factor.

In Fig.5. the auxiliary cylinder and the ram cylinder have mode 1 buckling results. In the case of static problem solving, there is displacement only in the loading axis in displacement calculations. However, it is actually possible to move in 6 DOF. Mode 1 results are measured as about 1 mm on the "Z" axis between the two supports, but this result is for the load factor (4.9). The load striker is a number of safety coats for value buckling. The safety number for mode 1 of the ram cylinder is 41.

In Fig.6. a displacement in the z-axis is observed relative to the auxiliary cylinder mode 2 buckling result, but this is the case when the visual load is increased by 51 times. The maximum displacement zone occurred between the fixed face and the motion-limited bearing on the "Z" and "X" axes. The safety factor for Mode 2 is 51. Buckling safety factor for mode 2 of the ram cylinder is measured as 450. According to the obtained results,

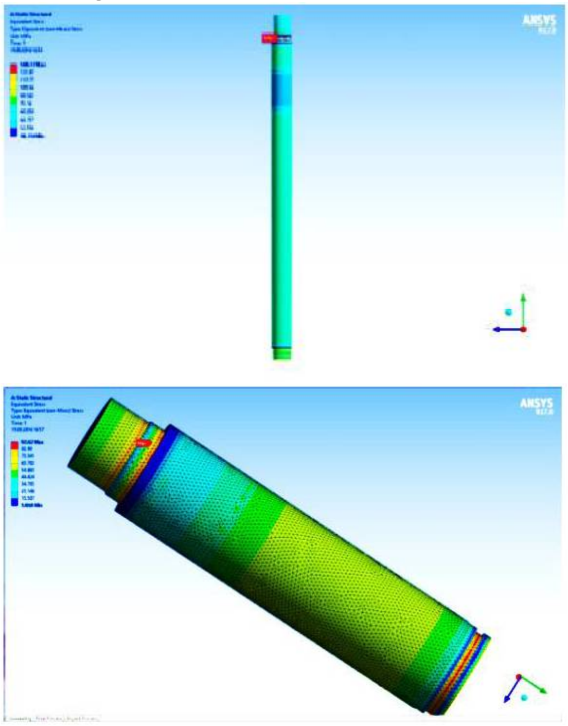


Fig.3. Auxiliary Cylinder (left) and Ram Cylinder (right) Buckling Analysis Von Misses Stress Results.

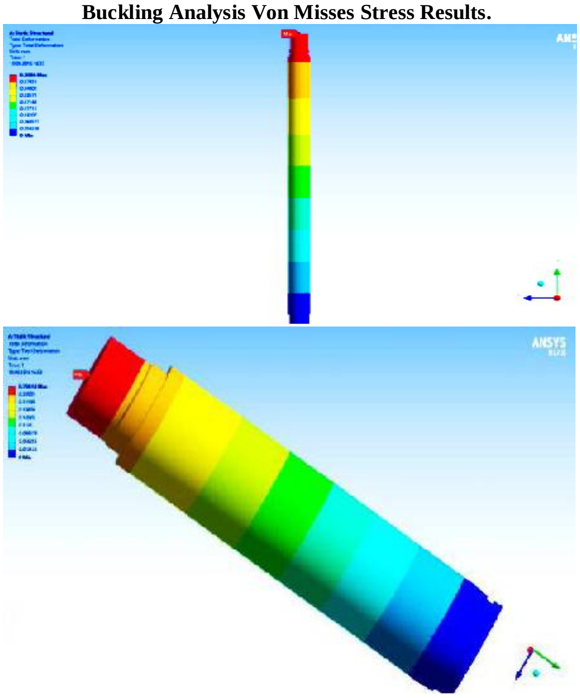


Fig.4. Auxiliary Cylinder (left) and Ram Cylinder (right) Buckling Analysis Total Displacement Results

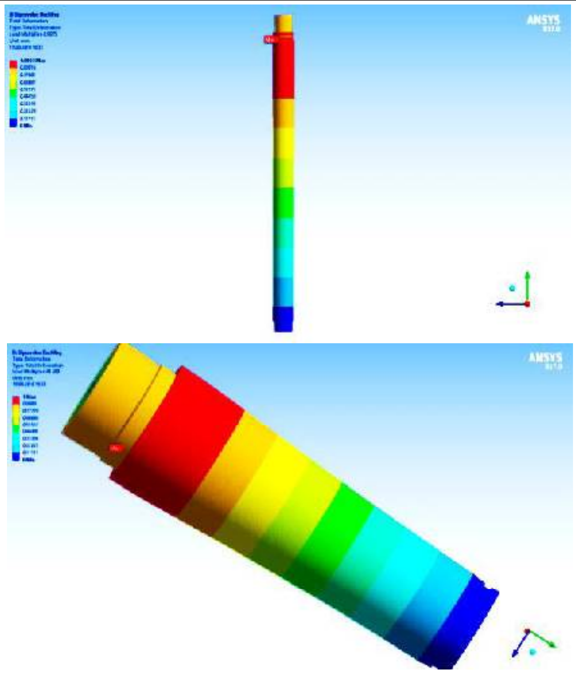


Fig.5. Auxiliary Cylinder (left) and Ram Cylinder (right) Buckling Analysis Mode 1 Shape and Load Factor

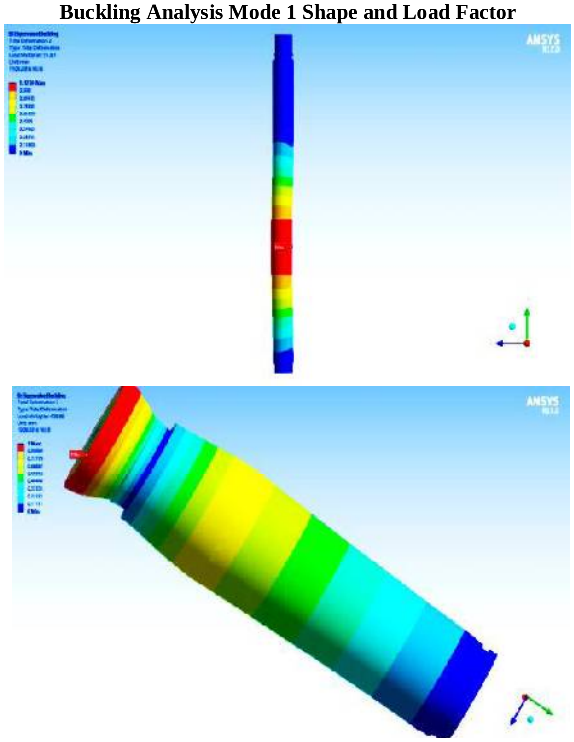
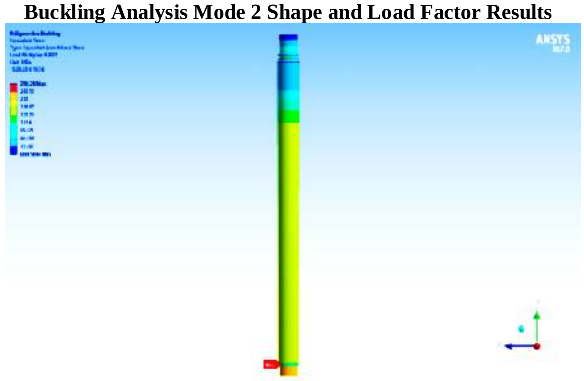


Fig.6. Auxiliary Cylinder (left) and Ram Cylinder (right) Buckling Analysis Mode 2 Shape and Load Factor Results



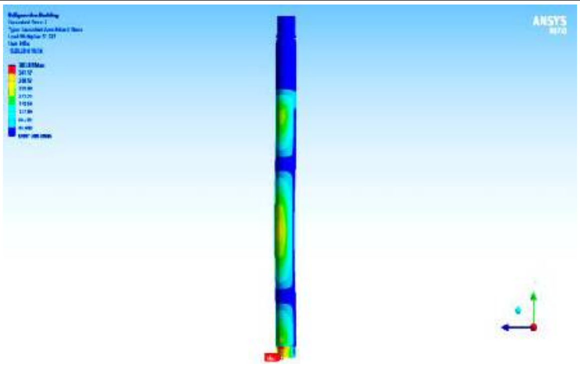


Fig.7. Auxiliary Cylinder Buckling Analysis Mode 1 (left) Mode 2 (right) Von Misses Stress Results

CONCLUSIONS

In this paper, an industrial 300 tons H-type hydraulic press is chosen and structural analysis of cylinder rod is realized for buckling analysis. For this purpose; linear buckling analysis of cylinder rods of auxiliary cylinder and ram cylinder is realized by using finite element method. As a result; maximum Von Misses stress locations, safety factors of static analysis, maximum deformation results and safety factors of buckling of cylinders are determined via ANSYS Workbench software.

All obtained results are useful and realistic for chosen press manufacturer company. The main contribution of the paper is that hydraulic press manufacturer verified analysis results with their experiences, changed design parameters and cylinder rod diameters of same type hydraulic. Also obtained results can be improved according to fatigue analysis in the future works by topological optimization.

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