Journal of the Technical University of Gabrovo
https://mc04.manuscriptcentral.com/jtug

DESIGN AND EXPERIMENTAL ANALYSIS OF A TEST MACHINE TO DETERMINE BUCKLING IN THE RODS OF HYDRAULIC CYLINDER

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ARTICLE INFO

Article history:
Received 26 September 2023
Accepted 31 October 2023

Keywords:
buckling, chrome plated rod, comparator, hydraulic cylinder

ABSTRACT

The issue of buckling is one of the subjects that should be especially emphasized in the design and dimensioning processes of engineering. In the past, this issue was not of much importance in construction and machine elements. Because the strength of the existing materials at that time was low and as a result, they were resistant to buckling due to the large cross-sectional areas of the structure and machine elements. However, in the course of time, the cross-sectional area of the building and machine elements also decreased. As a result of the reduction of the cross-sectional area, besides the advantage of getting rid of large and heavy elements, it also brought the disadvantage of buckling. In our seminar study the buckling caused by fixing the shaft at different stroke and pressure values was determined by 4 comparators and the buckling values on the shaft surface were examined.

1. INTRODUCTION

Buckling is an important factor in the design of machine parts and structural members operating under compressive load. If a structural element whose length is much larger than its other dimensions is forced into axial compression, it may be damaged in two ways; material yielding and buckling. The critical buckling force and stress for a built-in element with one end fastened and the other end free are given in Equations 1 and 2. (Euler's relation). When the critical load value is exceeded, the material is subject to buckling [1]. (Uluköy, A. 2016)

$$K = \frac{\pi^2 \cdot E \cdot l}{Sk^2} \tag{1}$$

$$\sigma_{CR} = \frac{K}{A} \tag{2}$$

If the ratio of the cross-section to the length of a thin column that is forced to be compressed from its axis is small, buckling may occur in this column. The load at which buckling occurs depends on the stiffness of the structure, not its strength. This incident is a stability problem. For buckling to occur, the calculated stress in the part does not necessarily have to exceed or approach the stress limits. For this reason, buckling may occur well below safe strength values. Since the geometry of the part will never be ideal and the force will never show its effect exactly from the center of gravity axis, buckling may always occur under compressive loads.

In order to under stand the subject of buckling well, it is necessary to know the mathematical equivalents of the words "stiffness", "long" and "slenderness". Stiffness for a material is either its modulus of elasticity E or its shear modulus related to:

$$K = \frac{G}{E(2+2v)} \tag{3}$$

There are many studies in the literature for buckling analysis of rod and frame systems. In most of these studies, approximate solutions were emphasized instead of the exact solution of the problem.

Chajes A., examined the stability problem of rods, frames and continuous beams with straight or variable cross-sections and investigated the approximate solution of the problem, especially using different matrix methods. In his solutions using the finite element method, he obtained the element and system matrices using the equation known as the total potential energy expression in the literature, and by using the sematrices, he obtained the critical buckling load and buckling mode shapes of different types of systems [2]. (continuousbeam, frame, etc.) (Chajes, A., 1974)

Chassie G. G. et al. examined the stability of circular cross-section bars under a constant compressive force and torsion.The transport matrices method was used in the numerical solution of the problem, and various charts were prepared depending on the results obtained [3]. (Chassie G. G., Becker L. E ve Cleghorn W.L., 1997)

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http://doi.org/10.62853/VKYD6280

Pflüger A. worked mostly on the linearization of special value problems. He particularly emphasized the difficulties of solving the problem precisely and stated that in addition to classical methods, approximate methods such as Ritz and Galerkincould be used. The author has presented his own research results on solutions to different types of buckling problems in diagrams and tables. [4]. (Pflüger, A., çev: Tameroğlu S., Cinemre V., Özbek T., 1970)

Tafresri and Bailey studied the stability of composite cylinders under combined loads. They investigated the buckling values when the cylinder was not perfect [5]. (Tafresi, A., ve Bailey C. G., (2007)

Akbulut and Ural examined the buckling behavior of composite plates with circular notches on all four edges. On a notched plate; They investigated how the notch radius, loading condition, plate thickness, reinforcement angles, number of layers and elasticity modulus ratios affect buckling with the help of the finite element method. They found that the notches opened on the edges of the plate increased the buckling strength of the plate, and the critical buckling value decreased in case of bi-directional loading [6]. (Akbulut, H. ve Ural, T., (2007)

Shadmehri et al. calculated the primary buckling load of conical composite shells under axial pressure with a semi-analytical solution method. They solved the equations they derived using the minimum total potential energy theorem using the Ritz method. They found that for thin and short conical shells, increasing the conicity angle reduces the buckling value [7]. (Shadmehri F., Hoa S.V., Hojjati M. (2012)

2. MATERIAL AND METHOD

In this study,it is aimed to determine the buckling parameter values of the Ø28 mm shaft, which is coated with at least 20 µm hard chrome with f7 tolerance, at different stroke lengths and different pressure values.

Test Stand

The GGG40 material was designed and utilized for fixing the hydraulic cylinder and the shaft end at various stroke lengths.

Hydraulic Test Unit

A hydraulic cylinder test unit was used in our experiments. The working pressure of the cylinder is 100 bar. When the shaft end joint is gradually stroked and fixed, the pressure is increased to 150 bar.

Double Acting Hydraulic Cylinder

The thrust force in the cylinders is different in forward (exit) and reverse (return) movements. While the piston area (A) is fully effective in forward motion, there is a loss equal to the cross-sectional area of the piston rod in rotation. In order to increase the force that the cylinder can apply, at least one of the pressure and area values must be increased.

The hydraulic cylinder pipe inner diameter used in the experiment is Ø40mm, the shaft diameter is Ø28mm and the cylinder stroke is 500mm.

Chrome Plated Rod

In the experiments carried out, it was produced from Ck45 material with a surface coated with at least 25 µm hard chrome, with a diameter of Ø28 mm and a tolerance of f7.

Table 1 *Chemical Properties of C 45 Steel (%)*

C	***Si***	***Mn***	***Ni***	***P***	***S***	***Cr***	***Mo***
0.43-0.5	*max 0.4*	*0.5-0.8*	*max 0.4*	*max 0.045*	*max 0.045*	*max 0.4*	*max 0.1*

Comparator

The values of the surfaces obtained in the trial studies were taken with a 0.01 comparator.

Digital Caliper

Measurements of stock materials during the processing of samples were also used as an aid to bias when shifting stock during processing. 500-707-20 IP67 model of Mitutoyo brand was used.

3. METHOD

In thisstudy, C45 material with a surface coated with at least 20 µm hard chrome with a diameter of Ø28 mm and a tolerance of f7 was used. The cylinder rear cover is articulated and fixed to the floor. It will be fixed to the ground at each stage by inserting a pin into the shaft end joint. In the first stage of the experiments, after the cylinder stroke was made to 150 mm, the shaft end was fixed to the ground. 4 comparators were placed equally around the shaft at ½ points of the stroke shaft, and the pressure was increased in 3 stages and buckling was observed in the shaft. This process was also carried out at stroke 300 mm and stroke 450 mm. The buckling values occurring in the shaft at 3 different completed stroke lengths and 3 different pressure ranges were measured on a 0.10 x 0.1 ear comparator device.

Measurement Distances of Comparators Depending on Stroke

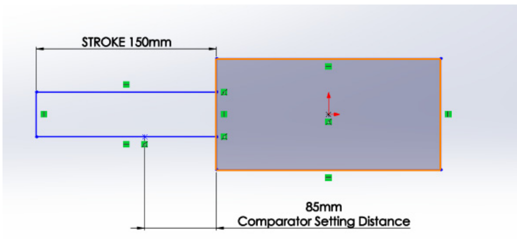

Fig. 1. Distance picture of dial comparators at stroke 150 mm

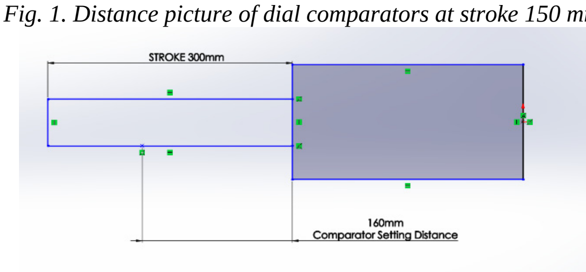

Fig. 2. Distance picture of dial comparators stroke 300 mm

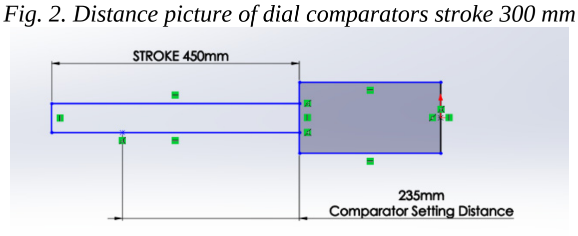

Fig. 3. Distance picture of dial comparators at stroke 450 mm

Position of Comparators on the Cylinder

Comparators are placed at equal intervals on the cylinder shaft.

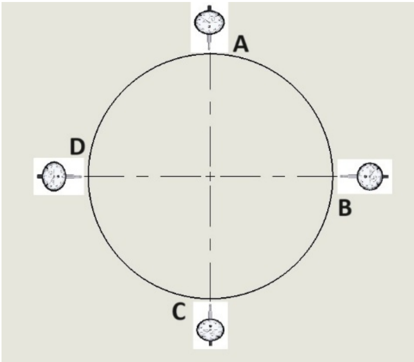

Fig. 4. Positioning of comparators when looking at the cylinder rod from the front view

Formulas Used in Calculations

$$F_{push} = P \times A(\ Newton\ ) \quad (4)$$

$$A = \frac{\pi \times D^2}{4}(\ mm^2\ ) \quad (5)$$

Formula Buckling Load:

$$K = \frac{\pi^2 \cdot E \cdot I}{Sk^2}$$

K - Buckling Load

$E - 2.1 \times 10^6\ N/mm^2$ (For Steel)

Formula Moment of Inertia:

$$I = \frac{\pi\left(D_d^4 - D_i^4\right)}{64}(\ mm^4\ ) \quad (6)$$

D_d - Cylinder tube outer diameter (mm)

D_i - Cylinder rodinner diameter (mm)

Sk^2 - Free Buckling Length (mm)

Formula Degree of Slenderness:

$$\lambda = \frac{s}{i} \quad (7)$$

λ - Degree of Slenderness

$s(\ cm\ )$ - RodFree Buckling Length

$i(\ mm\ )$ - Inertia Radius

Inertia Radius:

$$i = \sqrt{\frac{I_{min}}{A}}(\ mm\ ) \quad (8)$$

$I(\ mm\ )$ - Smallest axial moment of inertia of the rod

$A(\ mm^2\ )$ - Area

$$A = \frac{\pi\left(D_d^2 - D_i^2\right)}{4}(\ mm^2\ ) \quad (9)$$

Buckling Stress Formula in the Tetmajer Region For Steel:

$$\sigma_{BK} = 310 - (1.14 \times \lambda)(N/mm) \quad (10)$$

4. RESULT AND DISCUSSION

Buckling values of the hydraulic cylinder connected to the test device were measured at different pressures and different length ranges. Measurements were carried out in a hydraulic testing unit. The results of the values obtained as a result of the process were transferred to the tables (Table 2-4).

Table 2 *Buckling values obtained at stroke 150 mm*

STROKE 150 mm				
	A	B	C	D
100 bar	0.02mm	0.02mm	0.12mm	0.08mm
150 bar	0.02mm	0.02mm	0.14mm	0.05mm
200 bar	0.04mm	0.01mm	0.18mm	0.05mm

Table 3 *Buckling values obtained at stroke 300 mm*

STROKE 300 mm				
	A	B	C	D
100 bar	0.05mm	0.94mm	0.20mm	0.84mm
150 bar	0.07mm	0.95mm	0.25mm	0.76mm
200 bar	0.08mm	0.99mm	0.20mm	0.71mm

Table 4 *Buckling values obtained at stroke 450 mm*

STROKE 450 mm				
	A	B	C	D
100 bar	0.02mm	0.12mm	0.17mm	0.88mm
150 bar	0.02mm	0.17mm	0.11mm	0.87mm
200 bar	0.04mm	0.16mm	0.12mm	0.92mm

Buckling Calculation:

$F_{push} = P \times A$ **(100 bar)**

$F_{push} = 1.28 Ton = 12552.512 N$

$F_{push} = P \times A$ **(150 bar)**

$F_{push} = 1.92 Ton = 18828.768 N$

$F_{push} = P \times A$ **(200 bar)**

$F_{push} = 2.56 Ton = 25105.024 N$

$$A = \frac{\pi \times D^2}{4}(\ mm^2\ )$$

Comparator insertion distance at stroke *150 mm: 85mm = 8.5 cm*

Elastic buckling in the Euler zone $K = \dfrac{\pi^2 \cdot E \cdot I}{Sk^2}$ calculated from the formula. However, first the buckling area must be determined.

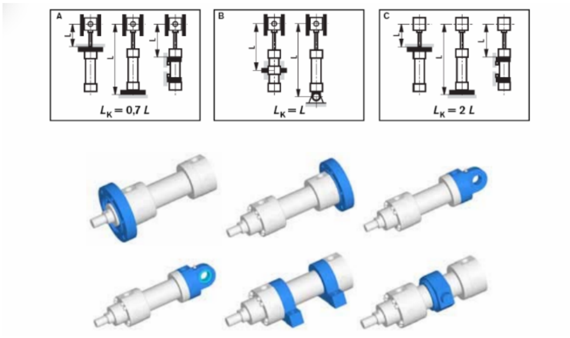

Fig. 5. Values of buckling length in terms of free length l, depending on the shape of the joint in shafts trying to buckle

To determine the buckling zone (Euler or Tetmajer zone) the degree of slenderness must be calculated.

$$I = \frac{\pi\left(D_d^4 - D_i^4\right)}{64} = \frac{\pi\left(40^4 - 28^4\right)}{64} = 95443.44\,mm^4$$

$$A = \frac{\pi\left(D_d^2 - D_i^2\right)}{4}$$

$$A = \frac{\pi\left(D_d^2 - D_i^2\right)}{4} = 640.56\,mm^2$$

$$i = \sqrt{\frac{I_{min}}{A}} = \sqrt{\frac{95443.44}{640.56}} = 12.20\,mm$$

In order to have an idea about whether the buckling will be elastic or not, the degree of slenderness (λ) must be calculated. For this reason, it can be seen that the second figure ($L_K = L$) in the buckling length table is suitable for our system.

s = RodFree Buckling Length(cm) = 1×1

$L = 85\,mm$

$s = 85 \times 1 = 85\,mm$

$$\lambda = \frac{s}{i} = 6.96$$

Since $\lambda \geq \lambda_0$ in the elastic region, our region is not an elastic buckling region (Euler) but an inelastic buckling region (Tetmajer).

If buckling occurs due to compressive stress in shafts with $\lambda \geq \lambda_0$, the shaft can not return to its previous state after the compressive force is eliminated, because the buckling is not elastic. In inelastic buckling, the stress that will cause buckling is calculated according to the values obtained in experiments.

For C 45 material, the slenderness degree is 100.

Buckling stress formula for steel in theTetmajer region:

$$\sigma_{BK} = 310 - (1.14 \times \lambda)[N/mm]$$

$$\sigma_{BK} = 310 - (1.14 \times 6.96)$$

$$\sigma_{BK} = 302.06\,MPa$$

$$F_{BK} = \sigma_{BK} \times A$$

$$F_{BK} = 302.6 \times 640.56 = 193487.55\,N$$

$F_{BK} = F_{push}$ for this reason, the cylinder is safe against buckling. In this calculation, the shaft remained in the elastic region. However, even though the elasticity values seen in the experimental studies (Table 2) were found, the shaft was calculated to be safe against breakage.

These calculations were also applied to stroke 150 mm and stroke 450 mm.

Comparator insertion distance at stroke 300 mm: 160mm = 16 cm

Comparator insertion distance at stroke 450 mm: 235mm = 23.5 cm

According to the formulas written above, the stroke is calculated at 300 mm:

$$F_{BK} = 295.05 \times 640.56 = 189000.17\,N$$

According to the formulas written above, the stroke is calculated at 450 mm:

$$F_{BK} = 288.04 \times 640.56 = 184507.54\,N$$

Since $F_{BK} > F_{push}$ in both stroke lengths (stroke 300mm and 450mm), the cylinder is safe against buckling. In this calculation, the shaft remained in the elastic region. However, even though the elasticity values seen in the experimental studies (Table 3 and Table 4) were found, the shaft was calculated to be safe against breakage.

5. CONCLUSION

In this study, the buckling parameters were optimized by working with the buckling values measured at different stroke values, and the pressure and stroke length with the lowest buckling value were found (Table 1-3). After the buckling calculations were made and the buckling zones were determined, the slenderness degrees were calculated. An idea was obtained whether the buckling would be elastic or not. Since the shaft will be subjected to buckling in the inelastic region, the calculations were obtained through stress experiments and the values were prepared in a table. Also. As a result of the buckling test, it was determined that the samples did not break in any part within the determined pressure ranges.

When we examine the charts in this study, the region with the least buckling in the experiments is the upper part of the shaft. (Area A) In this study, the highest buckling rate was found at the longest stroke and highest pressure value.

As the stroke and connection distance of the cylinder increases, the buckling force decreases and therefore the safety of the cylinder decreases. In the study, when the stroke of the cylinder increased, the buckling force decreased.

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