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Design of a Bend Testing Machine for Determination of Residual Stresses by the MTS-3000 Ring-Core System

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Abstract The design and manufacture of bend testing machine for testing the properties of materials is addressed by several reputable companies. However, their portfolios include commonly used devices with a similar design [1,2]. However, determination of the residual stresses by the MTS-3000 Ring-Core system requires atypical measurement conditions, so it is not possible to use a commonly produced and available bend testing machine. For this reason, there was a need to design a bend testing machine that would meet all the required measurement conditions.

Keywords: *residual stresses, Ring-Core method, bend testing machine*

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1. Introduction

There are several experimental methods to evaluate residual stress. The most well-known method for determination of the residual stresses is the hole-drilling method. The Ring-Core method is derived from the hole-drilling method, but on the examined specimen is not formed a hole, but a ring. Although the Ring-Core method causes more destruction of the specimen, it also eliminates some limitations of the hole-drilling method. The Ring-Core method can evaluate both homogeneous and non-homogeneous residual stresses along the depth of the material [3,4].

However, the Ring-Core method is not currently at the same level as the hole-drilling method. Verification experiments have to be carried out, and for this reason we have been designing our own bend testing machine to simulate internal stresses in the structure.

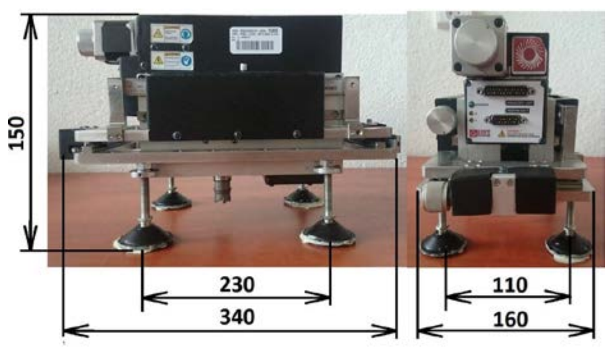


Figure 1. MTS - 3000 Ring-Core system

The task was to design a bend testing machine suitable for the measuring system SINT MTS - 3000 Ring - Core (Figure 1). The measuring system must be placed above the measured specimen and a ring of the required depth is drilled into the loaded specimen.

The design of the bend testing machine must take into account the location of the measuring system Ring-Core above the sample and the need to generate a uniform stress on the measured sample. Four-point bending will be considered where two members will support the measured specimen and the two actuators positioned symmetrically between the supports will apply a predetermined force on the specimen to produce stress, which is required for the measurement.

Before designing, it is necessary to define the requirements that the device will have to meet in order to be able to measure.

• Provide sufficient space for the location and handling of the MTS-3000 Ring-Core.

• The ability of the bend testing machine to generate a stress up to 100 MPa at the measuring point of the specimen, thereby simulating the conditions under which the material is exposed in a real environment.

2. Specimen Analysis

A rectangular beam will serve as a specimen (Figure 2). The design considered the maximum possible dimension of the specimen that the designed bend testing machine should be able to load. Specimen assembling will be considered theoretically with a hinge on one side and a sliding articulation on the other side [5].

Due to the construction of the MTS-3000 Ring-Core and the way how the machine makes ring core, it is not possible to load the specimen directly in the middle, but load forces must be placed next to the measuring device. For this purpose, the load shown in the Figure 3 was chosen, where the bending and reaction forces are also defined, thus defining the position of the working bending member as well as the position of the supports on which the specimen is to be placed.

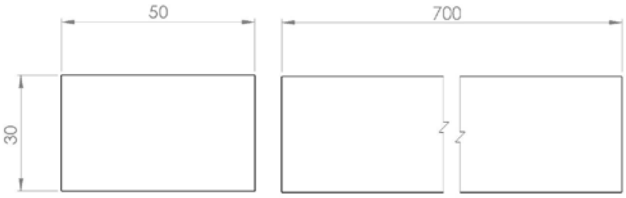


Figure 2. Specimen: front view, side view

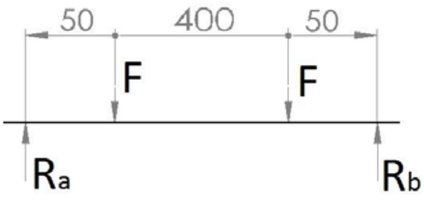


Figure 3. Location of load forces

In this case, it is a simple bend on the prismatic beam. With regard to the simplicity of the equilibrium conditions and the way how the moment is applied to the beam, the mathematical derivation was omitted and the beam was described directly by the picture (Figure 4), from which it is obvious that the reactions will be directly proportional to the force acting on the beam. In the BC section there are only bending moments.

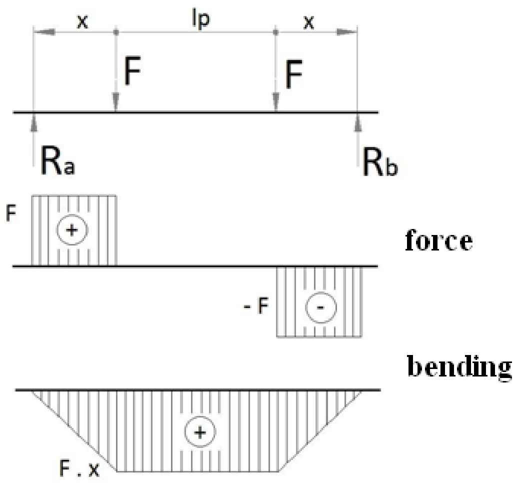


Figure 4. Graphical interpretation of the loaded beam

2.1. Calculation of the Necessary Forces and Reactions in the Supports

The following section describes the calculation of the required forces which have to be applied to the specimen to produce the desired stress on the measured surface [5].

$$\sigma_o = \frac{M_o}{W_o} \tag{1}$$

where:
$\sigma_o [MPa]$ – required bending stress [100 MPa],
$M_o\ [Nmm]$ – bending moment,
$W_o\ [mm^4]$ – second moment of area.

Bending moment:

$$M_o = F.x \tag{2}$$

where:
$F\ [N]$ – force,
$x\ [mm]$ - position vector from the reference point.
Second moment of area:

$$\mathrm{W_o} = \frac{\mathrm{J_y}}{\mathrm{z_{max}}} \tag{3}$$

$$z_{\max} = \frac{h}{2} = \frac{30mm}{2} = 15mm \tag{4}$$

where:
$J_y\ [mm^4]$ – quadratic moment to the neutral axis,
z_{max} – maximal distance from the neutral axis.
Quadratic moment to the neutral axis:

$$J_y = \frac{b.h^3}{12} = \frac{50mm\ .30mm}{12} = 112500mm^4 \tag{5}$$

From the equation for the bending stress, we express the magnitude of the force.

$$\sigma_o = \frac{M_o}{W_o}$$

$$\sigma_o = \frac{F.x}{\frac{\frac{b.h^3}{12}}{\frac{h}{12}}} = \frac{F.50mm}{7500mm^2} \to F = \frac{7500mm^2.100MPa}{50mm}$$

$$F = 15000N.$$

The calculation shows that the magnitude of the required force is F = 15 000 N.

3. Design of the Frame of the Bend Testing Machine

The frame is the basic carrier in the construction and engineering units. This is the part of the structure that forms the basic supporting part (skeleton) [6]. The frame has the task of transferring forces, moments and creating a suitable platform for fastening and clamping other parts of the construction.

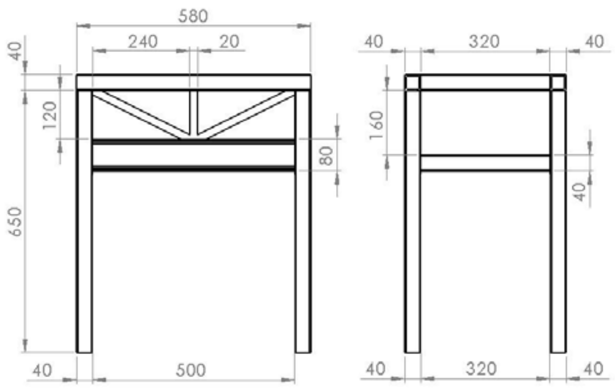


Figure 5. Frame dimensions of the bending device

The frame (Figure 5) is designed from several standardized profiles. Used profiles and their STN standards:

• square profile, seamless, hot-formed steel tube: TR 4 HR 40 x 4 - 11523 - STN 425 720

• I profile hot-rolled steel bar: I 8 / B STN 42 5550 - 11523 - STN 42 0135.21

• Flat hot-rolled steel bar: 40 x 20 STN 42 42 5522.11 - 11550 - STN 42 0209.50

Subsequently, frame load analysis was performed by using finite element method in the software SolidWorks (Figure 6) [7,8].

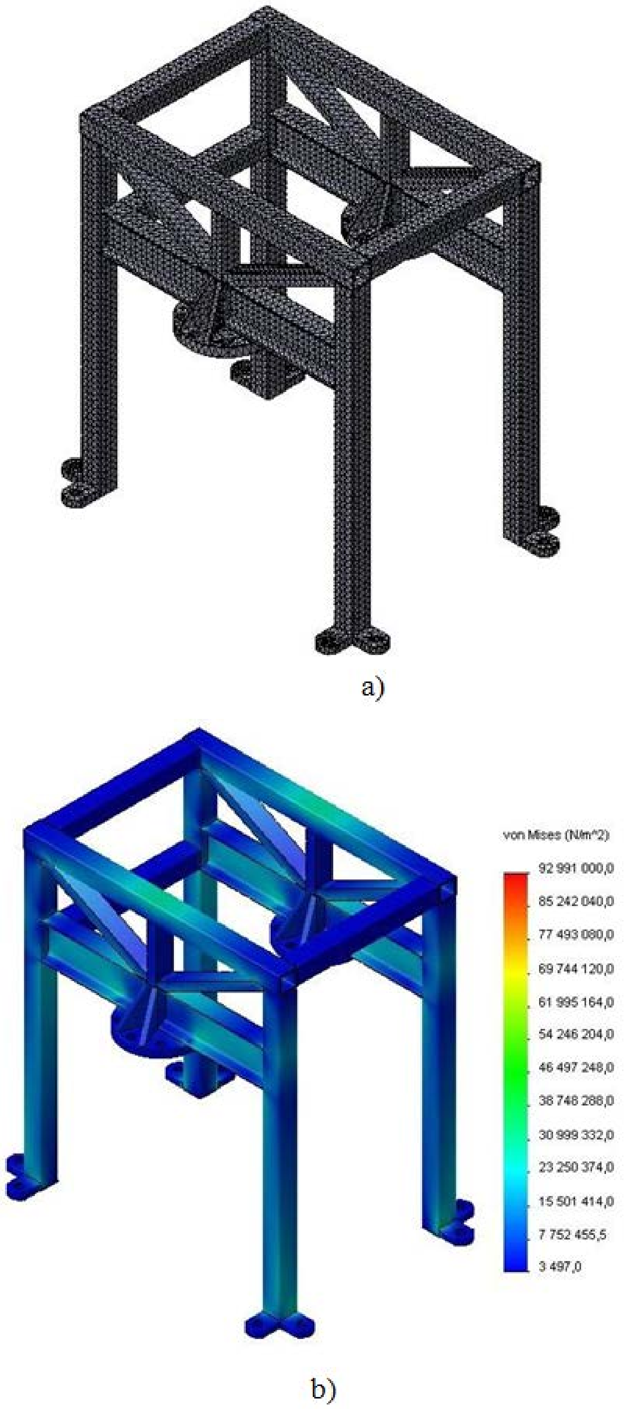


Figure 6. a) Mesh model of the frame b) Simulation of the stresses of the frame

The displacements generated by the forces on the frame are shown in the Figure 7.

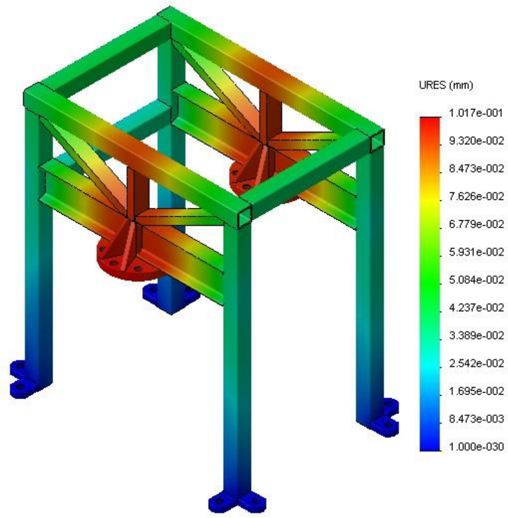


Figure 7. Simulation of the displacement on the frame

4. Bearing of the Specimen

With regard to the bearing of the specimen and that the force generated by the piston on the specimen is equal to the force acting on the specimen, a bearing, which can withstand twice the static load value has been selected. A TASE35-N roller bearing was selected, which can transmit a radial dynamic load of = 27 500 N and a radial static load of 15 300 N (Figure 8) [9].

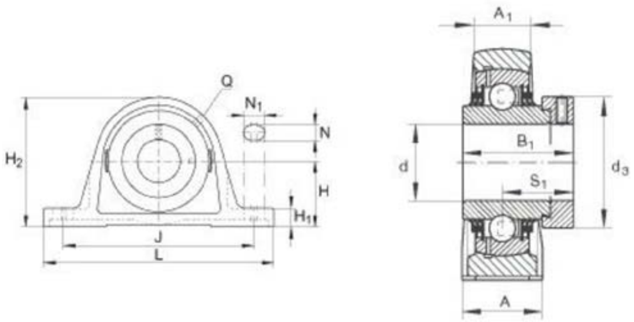


Figure 8. TASE35-N

5. Design of the Bending Tool

As a working tool was selected prismatic roller made of tool steel class 19 and was positioned on the bending frame. The bending frame is solidly connected to the guide bars. The layout of each element is shown in the Figure 9.

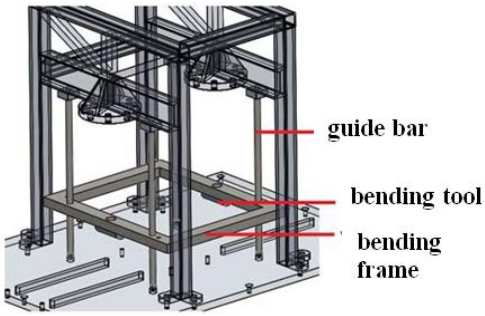


Figure 9. Simulation of the displacement on the frame

6. Selection of the Direct Hydraulic Motor

As a drive unit was selected a hydromotor. The Hydromotor has been selected from the available offer from company HYDRAULICS s.r.o., which deals with the production, development and servicing of hydraulic equipment and their components. On the basis of the minimum displacement force requirement F = 15kN, was selected hydraulic motor ISO 6022 model MF4 (Figure 10) from the catalog of HYDRAULICS s.r.o. company [10].

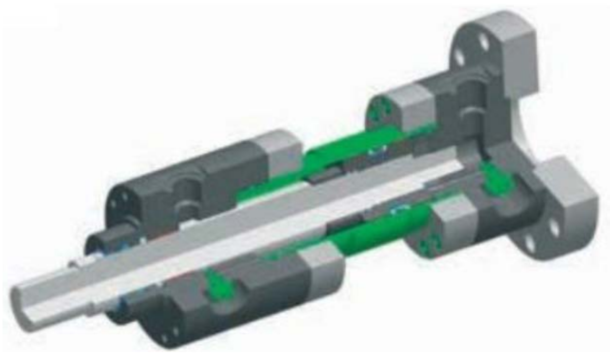


Figure 10. Direct hydraulic motor ISO 6022 MF4

Figure 11 shows the final design of the bend testing machine.

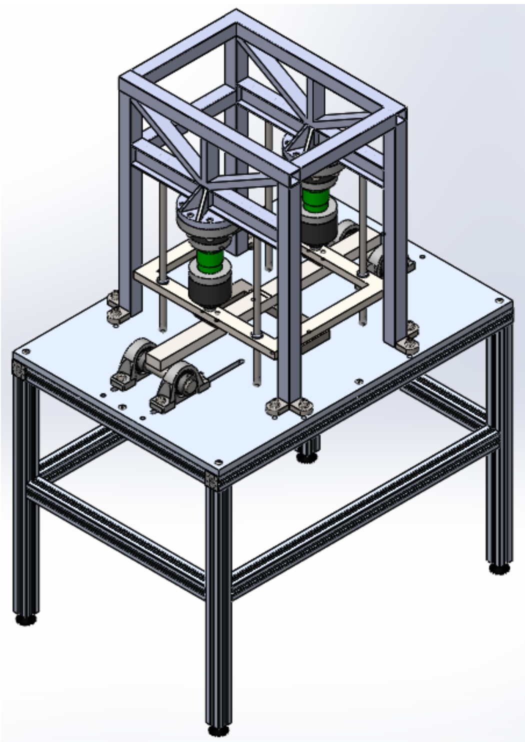


Figure 11. Final design of the bend testing machine

7. Design of Hydraulic Circuit Elements

Basic part of each hydraulic circuit is the pressure source – a hydraulic power pack consisting of a drive unit, a tank, a distribution of hydraulic fluid, valves and filters. The propulsion unit is composed of a hydraulic pump and an electric motor [11,12].

Calculation of the working pressure in the cylinders of the hydraulic pump:

$$p_{jt} = \frac{F_{tl}}{\frac{\pi . d^2}{4}} \tag{6}$$

$$p_{jt} = \frac{37680N}{\frac{\pi .50mm^2}{4}} = 19.2MPa$$

where:
p_{jt} $[MPa]$– pressure in the cylinder of the hydraulic pump,
F_{tl} $[N]$– force,
d^2 $[mm]$– piston diameter.

Calculation of the flow rate delivered to the hydraulic pump:

$$\begin{aligned} Q &= \frac{\pi . d^2}{4}.v \\ Q &= \frac{\pi .0.05mm^2}{4}.0.1m.s^{-1} \\ &= 1.963.10^{-4} m^3.s^{-1} \\ &= 11.77510^{-3} m^3.min^{-1} \end{aligned} \tag{7}$$

where:
Q $[m^3.s^{-1}]$ – pump flow rate,
v $[m.s^{-1}]$ – pump speed

Calculation of the geometric volume of the hydraulic pump:

The rotation speed of the hydraulic pump is the same as that of the electric motor $n = 950\ min^{-1}$.

$$V_g = \frac{Q}{n} \tag{8}$$

$$\begin{aligned} V_g &= \frac{11.77510^{-3} m^3.min^{-1}}{950\,\mathrm{min}^{-1}} \\ &= 12.39cm^3 \end{aligned}$$

where:
V_g $[mm^3]$ – geometric volume of the hydraulic pump,
n $[min^{-1}]$ – rotation speed of the electric motor.

Input power:

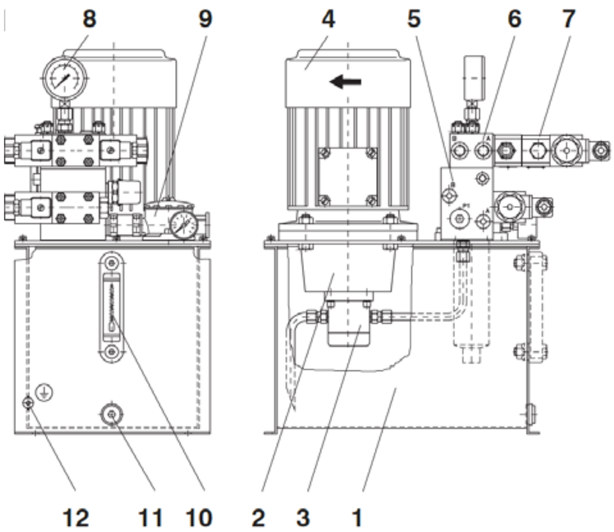
$$P = \frac{Q_j \cdot p_{jt}}{t} \tag{9}$$

$$\begin{aligned} P &= \frac{3.43.10^{-4} m^3.s^{-1}.19.2.10^6 MPa}{0.84} \\ &= 7847W \end{aligned}$$

where:
P $[W]$ – input power,
Q_j $[m^3.s^{-1}]$ – pump flow rate,
η_t $[-]$ – total efficiency.

According to the Argo Hytos company catalog, we chose the SA4-45U class J15, which complied all the calculated parameters [13,14].

Schematic view of the chosen hydraulic power pack SA4-45U is shown in the Figure 12 [13].



1 – tank, 2 – drive/bell housing, 3 – pump, 4 – electric motor, 5 – base block (safety block of the accumulator), 6 – horizontal stacking assembly, 7 – vertical stackling assembly, 8 – pressure gauge, 9 – return filter with by-pass, integrated air breather/filler and clogging indicator, 10 – continuous level gauge, 11 – magnetic drain plug, 12 grounding point

Figure 12. Hydraulic power pack SA4-45U

8. Conclusions

In the present article a design of a bend testing machine is described. The frame of the bend testing machine was designed with regard to the requirement for sufficient space to work with the measuring system MTS-3000 Ring-Core. Using the FEM method in the SolidWorks program, simulation of the stresses and displacements created on the frame in the loaded state was performed. From the obtained values it was concluded that the frame meets the requirements imposed on it.

Prismatic cylindrical supports have been designed to accommodate the sample so that no undesirable stresses are produced on the sample. Accordingly, bending tools were also selected.

As an actuator of the bend testing machine, two hydromotors were selected, which were to be able to draw sufficient pressure on the sample. Hydraulic power pack was also designed.

Acknowledgements

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