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# MODELLING OF SCREW JOINTS WITH ITS EXPERIMENTAL VERIFICATION

# MODELOWANIE POŁĄCZEŃ ŚRUBOWYCH Z WERYFIKACJĄ PRZYJĘTYCH MODELI

The paper presents the analysis of modelling methods of screw joints taking into account the estimation of the effects of a screw and joined elements' load. It may be an instrument for the prediction (identification of conditions of occurrence) of possible unfavourable phenomena, e.g. cracking during installation and exploitation. The individual model of such type of joint using FEM, has been devised in the course of investigations. Further, the model was subjected to experimental verification using a testing machine.

W pracy zaprezentowano analizę sposobów modelowania połączeń śrubowych z punktu widzenia możliwości oceny efektów obciążenia śruby i elementów łączonych jako narzędzia do przewidywania (identyfikacji warunków występowania) możliwych niekorzystnych zjawisk, w tym np. pękania podczas montażu i eksploatacji. W ramach przeprowadzonych badań opracowano własny model połączenia a następnie dokonano jego weryfikacji poprzez realizację badań na stanowisku zainstalowanym na maszynie wytrzymałościowej.

## 1. Introduction

Possibility of identification of conditions of unfavourable phenomenon occurrence e.g. cracking during assembling and exploitation of joint construction elements is important from the point of view that it has not been taken into consideration in theoretical research so far.

In practice, screw joints are usually calculated in the traditional manner, according to simplified formulae, which leads to considerable divergence in calculations, and therefore, to a possible risk of failures of various kinds resulting directly from the applied approximations and simplifications. The theoretical bases of approximate calculation methods come down to isolating a simple screw joint from a real assembly and substituting it with a simple axisymmetric model. In another approach the screw joint is treated more generally as a complex contact problem involving the mechanics of a system of deformable bodies. Its solution consists of the definition of the contact zone of the main elements of the joint, as well as the interacting forces, stress fields and strains in individual elements. These problems are solved by FEM. The factors affecting the load capacity of a screw joint are: the connection surfaces, the method of realizing and controlling forces in screws, the manner of the joint load and fitting of the bolt shanks in holes.

## 2. Characteristics of screw joints and methods of analysis

The method of force-elongation-characteristics is one of the ways to analyze a screw joint. Here the calculations come down to isolating a single screw joint from a real assembly substituting it with a simple axisymmetric model [1, 2].

It show in [1] force-elongation-characteristics determined for the models of the screw and connected elements, as well as the whole joint.

The force-elongation-characteristic is analyzed in the two states of load and deformation of a joint:
– assembly state, i.e. the joint is preloaded with the initial force;
– exploitation state, taking into consideration different loads occurring during this stage.

The method of force-elongation-characteristic can be employed for different cases of the load.

Numerous authors have dealt with the analysis of the presented models of joint. Their work, preserving the

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classical calculation system, took into account as many parameters and load cases as it was possible. While the determination of the bolt characteristic does not pose a problem, the correct determination of the interaction zone between the bolt and the connected elements as well as the element characteristic creates various difficulties.

The paper [3] presents the evaluation of the most frequent hypotheses applied to calculate the rigidity of connected elements and the zone of deformations of the clamped part, paying special attention to the effect of the accuracy of the assumed values on the load capacity of the frictional joint.

The article [4] discusses the effect of the change in the work temperature of the joint on the achieved connection parameters based on the example of the gear housing joints made from aluminium or magnesium alloys. For the aluminium wrought alloy, the reduction in pre-tension varies in the range of 10…25%, depending on the applied dynamic load and the initial pre-tension. The aluminium cast alloy gear housing with the steel screw joint shows a decrease of pre-tension of about 2% at room temperature and of about 10% at the temperature of 120°C. In the case of the magnesium gear housing, the situation is more serious. At 120°C, the pre-tension is reduced by 40% in the best case (high strength aluminium screw), and by 70% in the worst case (steel screw) within a time interval of 100 h. The thermal expansion of screw and clamped parts plays a central role for the loss of pre-tension. This problem is also discussed in the paper [1].

Basing on the above presented connection model the paper [5] shows the relationship between the size of the deformation zone, thickness of connected elements and the washer diameter (Fig. 1)

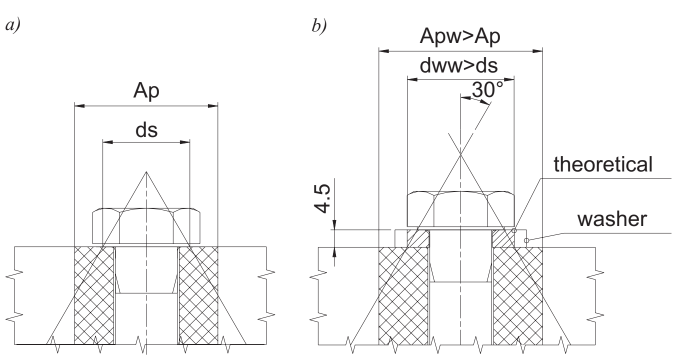



Fig. 1. Influence of washer on deformation zone $A_p$: a) deformation zone without washer, b) deformation zone with washer

Program POLS-2 has been designed relying on the theoretical bases described in the article [6]. The program is intended for the multi-variant structural calculation of coaxial and non-coaxial loads of one-screw connections. The program has been written in the FORTRAN 77 language. The described program has been checked in project classes in the Bases of Machine Construction and it can provide assistance to a project engineer. Other programs, making use of the VDI 2230 norm, such as SR1 V13.2 [7] and MITCalc [8], realize similar computational functions.

In some papers screw joints are treated in more general terms as a complex contact problem concerning the mechanics of a system of deformable bodies. Its solution consists in the definition of the contact zone of the main elements of the joint, as well as the interacting forces, stress fields and strains in individual elements. Finite element methods are applied for solving these questions.

The authors of the article [9] outline the result analysis of the determination of elastic elongation of the elements connected in an axisimmetrical screw joint. These results are obtained from different computational methods and FEM modeling. Wide scatter of results has been found as well as their divergence with the results obtained experimentally.

This emphasizes the necessity of the elaboration of an improved model, which takes into account a greater number of factors as well as the non-linear flexibility of elements.

The FEM method has also been used to calculate screw joints in wheel rim bearings [10, 11]. In such studies, the traditional model of connection was subject to verification by building a FEM model of connection, which allowed the correct calculation of the rigidity of the bolts and clamped parts.

The factors affecting the load capacity of a screw joint are:

- the method of preparing the connection surfaces
- the methods of realizing and controlling forces in screws
- the manner of the joint load
- fitting of the bolt shanks in holes.

The papers [12, 13] show an attempt at the presentation of functional relationships occurring between physical and mechanical parameters during the installation of high resistance screws.

The mutual dependence between forces occurring in bolts and the assigned moment of tightening as well as the angle of nut rotation.

The analysis of butt connections without preliminary compression is presented in [14]. It has been found that the occurrence of definite clearances (clearance = 2 mm) causes the growth of stress in connection elements. The NASTRAN program was applied to model a typical T-joint. The numerical analysis and experimental research on compressed two- bolt strap connections is presented in [15]. This was a stage leading to the elaboration of the method to calculate the joint of low deformability and high load capacity.

The ABAQUS program was used to carry out the FEM analysis, assuming the elastic-plastic and isotropically hardened model of material. The presented model consisted of 11 elements: lap-plates, a middle plate, bolts, washers and nuts. A series of research dealing with natural scale models has been conducted in order to verify the numerical model. A satisfactory convergence of the results has been obtained allowing, after generalization, practical applications of this type of joint.

The manners of realizing the joint load are very important, what have bee analysed in [16, 17]. These papers describe the effects of tightening above the yield point on fatigue strength and loosening in bolt-nut connections as well as the influence of set-up conditions on created stresses in bolts. The work [16] deals with research on fatigue strength, loosening and vibro-creeping of bolts of 8.8 material class. This research allows one to ascertain that tightening of bolts over the yield point causing plastic deformation in thread turns, increases fatigue strength of a joint approximately threefold in comparison to tightening in elastic range. They affect the stable moment of tightening before and after a dynamic load of a joint.

A similar problem is described in [18]. The standard NC50 connection, used in gas and oil installations, has been considered. It is a joint with a conical thread. It has been established that exceeding the moment of tightening by about 50%, compared to the recommended moment of tightening, causes a drop of the ratio flank load (RFL), both with the load and without it. A more favourable (more uniform) distribution of load can improve the fatigue strength of a connection.

On the other hand, the article [17] states that such actions as oiling the thread, obeying the sequence of tightening and using washers and nuts made from different materials, have the greatest influence on the value of stretching stresses in the bolts fixing the engine head. Failing to keep technical requirements may lead to the failure of the connection, namely leakage or even engine damage.

Numerous sources [19, 20, 21] are devoted to the elaboration of FEM modelling of typical connections used in building engineering. The authors concentrate mainly on the elaboration of various FEM models of screw connected steel constructions elements, e.g. the analysis of strains and stresses in T-type connections of different geometric parameters. The analysis were carried out using different programmes and different models of materials, and next the results were verified by experimental methods achieving satisfactory convergence of simulations with the results of empirical research. This proves that the FEM method is highly useful for predicting mechanical behaviur of a joint in real construction.

## 3. Numeric analysis of the proposed axisymmetric FEM model of a screw joint

The analysis of stresses and strains in the elements of a screw joint was performed on the example of a cover joint of a wheel loader driving axle [22].

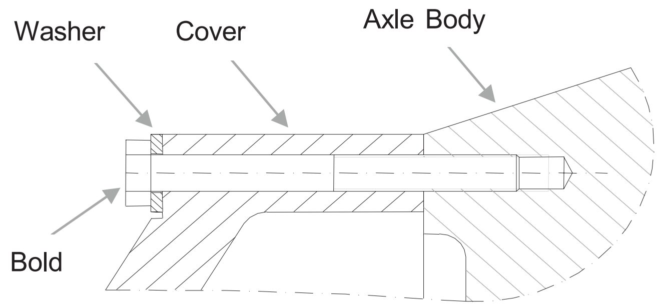



Fig. 2. Joint of cover with body driving axle

The connection is realized by means of 18 bolts M16×140 class 12.9 and 18 hardened washers ∅18. The mechanical properties of bols are as following:

– tensile strenght $R_m$ = 1220 MPa
– hardness HRC = 38–44
– yield strength $R_{e0,2}$ = 1100 MPa
– elongation A5 min 8%
– material 40HNMA

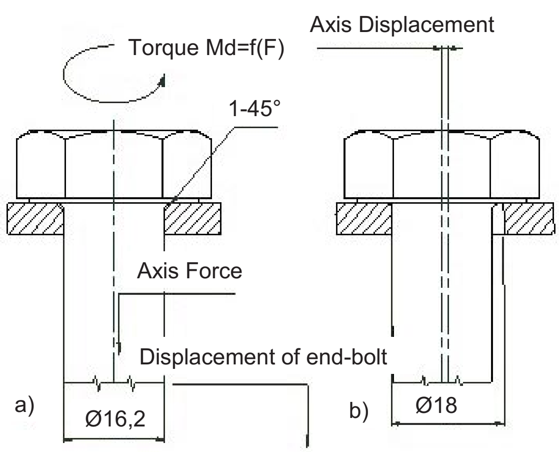



Fig. 3. Scheme of screw joint: a) screw M16×140 + washer with chamfer 1-45°, hole in washer ∅16,2, b) screw M16×140 + washer without chamfer with hole ∅18, axis of washer move differ of screw 0.55 mm

The mechanical properties of washer are defined by hardness 36–42 HRC and material is steel 65. Two cases of joints, which were analyzed, are shown schematically in (Fig. 3). The (b) case can come into being when the hole in the washer is bigger than the bolt shank and the bolt screwing proceeds in horizontal position. Then the axis of the washer is displaced in relation to the bolt axis, which results in uneven distribution of stresses and strains in the contact zone of the bolt and the washer edge. In addition, the sharp edge of the washer undercuts the bolt head in the passage zone from the shank to the head. In consequence of the above conditions the joint works incorrectly, which may lead to the destruction of the bolt. In the (a) case this phenomenon does not occur.

A simplified assumption was accepted in the preliminary period of numerical analysis. The assumption consisted in the modelling of the connection as the axisymmetrical bolt-washer system. It allowed to obtain considerable simplification of the FEM model, so that only a limited model of half a bolt and washer is needed to be elaborated (Fig. 4). Considering the fact that the bolt-washer contact occurs on the surface of the bolt shank and the passage from the shank to the small neck under the head, the bolt head was accepted as cylindrical, and not hexagonal.

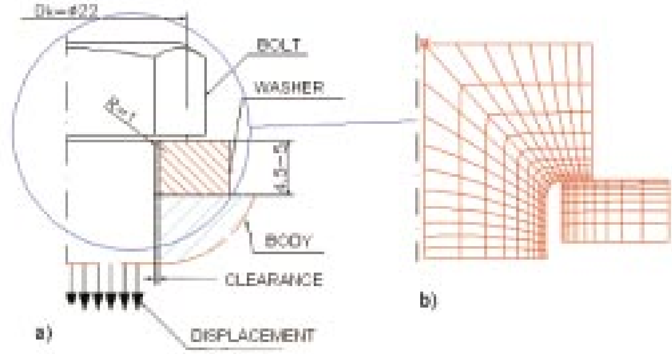



Fig. 4. Scheme for FEM: a) sketch, b) finite-element model

The analysis was carried out using the ADINA program [23]. Mechnical behavior of the material of the bolt is described by the equation:

$$\sigma = 1005.8\varepsilon^{0.204}. \tag{1}$$

The tightening of the bolt is simulated as displacement (pitch) of the bolt head of the value from 0.1 for 0.2 mm.

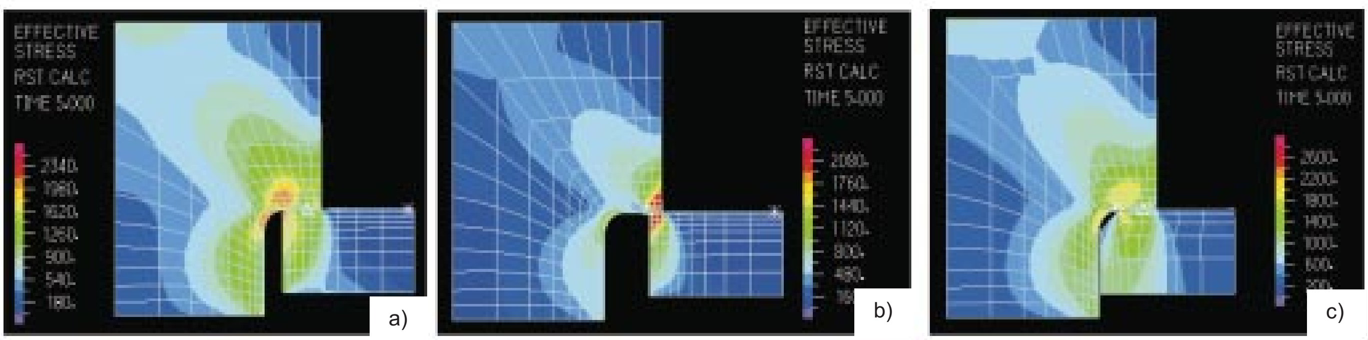



Fig. 5. Distribution of stresses in screw and washer for different montage casses: a) axisimmetrical position of washer ∅18 (clearance 1 mm), b) max. axisimmetrical position of washer ∅18 (clearance 2.38), c) fit washer (cleareance 0.2 mm) [22]

Tightening of bolt simulate as displacement (pitch) bolt head about value from 0.1 for 0.2 mm. (Fig. 5) shows exemplary results of modeling of loads for different assembly cases.

The model of joint suggested in the preliminary analysis does not reflect the actual state of the joint.

As it does not take into consideration additional loads resulting from the lack of symmetry in the real object. The washer with a certain clearance in relation to the bolt shank does not have to fit symmetrically in relation to the bolt axis. However, it adequately shows the influence of the clearance between the washer hole and the bolt shank on the distribution of stresses and strains, and thereby on the parameters of the joint's work.

A three-dimensional model of connection with one symmetry plane was worked our during the next stage of modelling (Fig. 6, [24]).

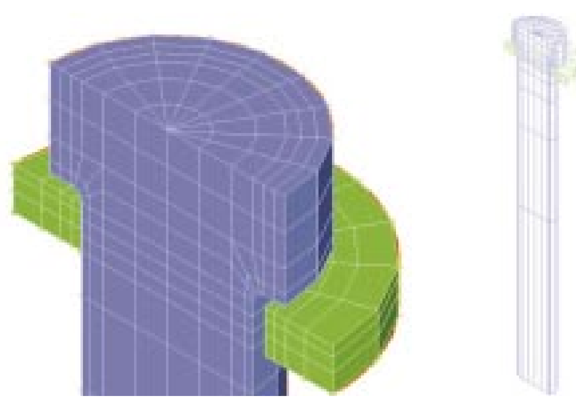
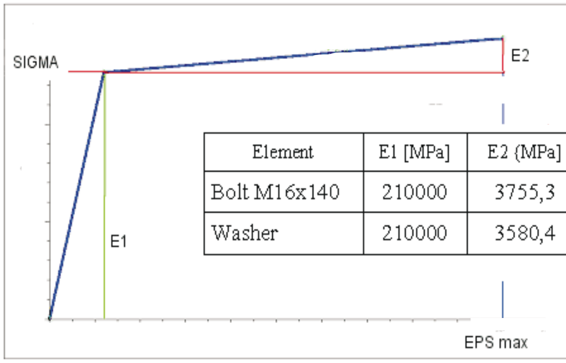


Fig. 6. Finite-element model

The installation of joint (screwing in the bolt causes axial force in the bolt) was simulated by the displacement of the bolt end as follows:

– $\Delta l$ = 5 mm, for the material model without the determination of permissible strain;

– $\Delta l$ = 3 mm, for the material model with the detemination of permissible strain.




| Element | E1 [MPa] | E2 (MPa] | SIGMA [MPa] | EPS max.*) |
|---|---|---|---|---|
| Bolt M16x140 | 210000 | 3755,3 | 1261,8 | 0,1177 |
| Washer | 210000 | 3580,4 | 1059,4 | 0,1115 |

Fig. 7. Parameters of elastic-plastic material model

The model of bilinear-plastic body with linear hardening and maximum permissible strain (Fig. 7) was accepted for calculations of the bolt and washer material. This model is described by the following parameters:
E1 – elastic modulus
E2 – plastic modulus
SIGMA – preliminary stress (stress value in the intersection point of straight lines)
EPS max (option) – maximum permissible strain.

Fig. 8, Fig. 9, Fig. 10 present results of FEM analyses in characteristic points on bolt (PL, PP, PS) at the non axisymmetric position of the washer.

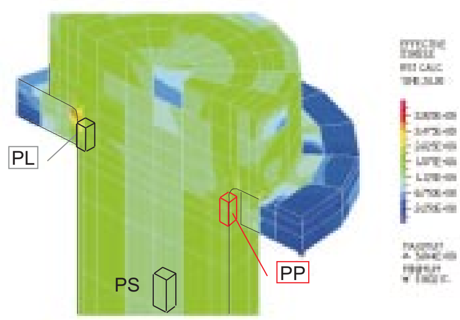



Fig. 8. Distribution of stresses in screw and washer for noncoaxial placement

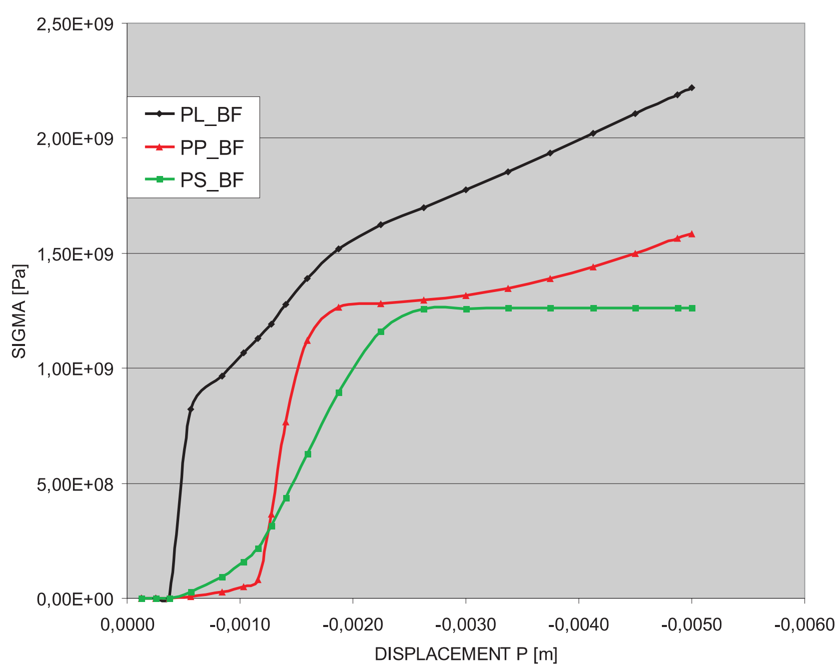



Fig. 9. Relationship between stress and bolt elongation (end-bolt displacement)

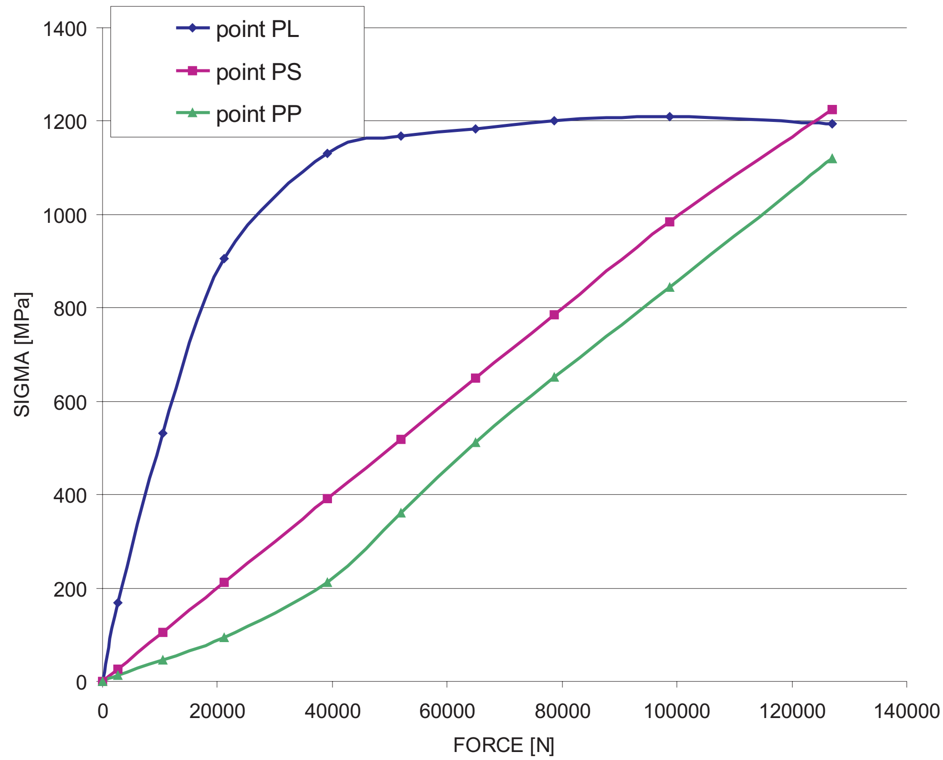



Fig. 10. Relationship between stress and tensile force determined in points: PL, PP, PS (noncoaxial placement of washer)

## 4. Verification of FEM model

Experimental research of M16 class 12.9 bolts and a special washer for high – loaded connections was carried out for the verification of the FEM analysis results of a screw joint. This research was conducted using the tensile machine, WPM HECKERT, which recorded the variability of the measuring gauge strains placed in the zone of anticipated piling up of stresses. The second measuring gauge, which was the point of reference, was placed in the zone of uniform stresses.

Experimental research was carried out for different possible cases of mutual position of cooperating elements.

Concepts of analyses of mechanical behaviour of connection elements subjected to the axial stretching were employed for verification of the accepted model.

The experiment was limited to a small range of elastic deformations on account of the work range of the tensile machine. The bolt was subjected to axial stretching of up to 60 kN. Fig. 11, Fig. 12 shows the measurement set scheme.

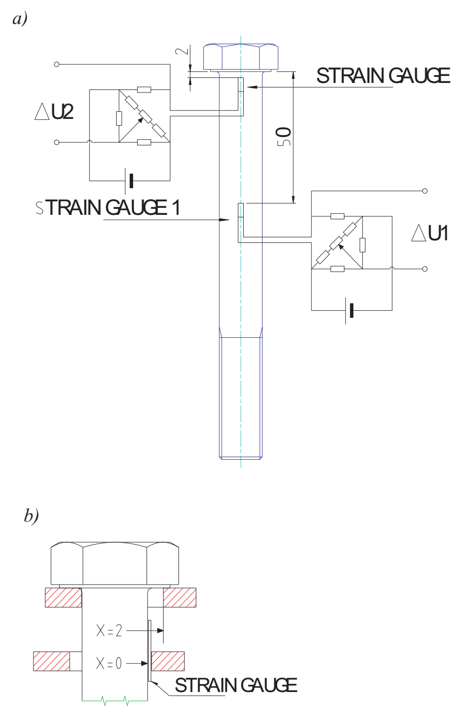



Fig. 11. Position of mesurement gauge on bolt (a) and position washer in bolt (b)

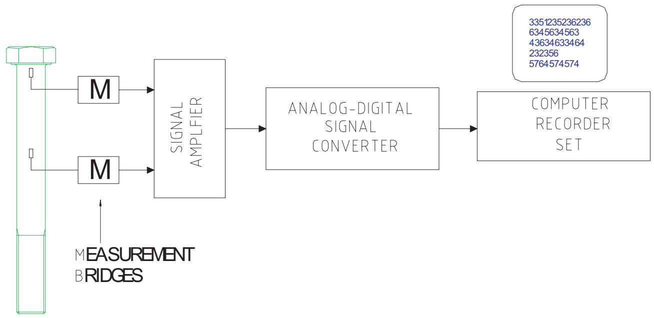



Fig. 12. Measurement set (scheme)

## 5. Results of experimental work

During the experiment the examined bolt was subject to axial stretching force. The variability of the signals received from the measuring gauge was recorded.

The experiment was conducted in two stages. During the first stage the signal was recorded from the measuring gauge placed about 2 mm below the bolt head and from the bridge measuring the deformation of the piston of the hydraulic actuator in the tensile machine.

During the second stage another measuring gauge was added half-way through the non-threaded bolt shank. For each series of measurements the position of the washer was changed in relation to the bolt, ranging from the position by the bolt shank (X=0) to the position of maximum distance from the bolt shank (X=2). The range of the stretching force was changed from 0 kN up to 60 kN.

Fig. 13, Fig. 14, Fig. 15. show relationships calculated on bases of graphs of the signal.

The received results of the variation of stresses under the bolt shank prove the influence of the position of the washer (distance from the bolt shank) on the value of stresses in the tested bolt zone. While in the examined load range it is not obviously visible (60 kN load is 30% of the maximum possible load for the 12.9 bolt class), the curves indicate that the difference increases with the increase of the connection load, which in turn may lead to exceeding permissible stresses.

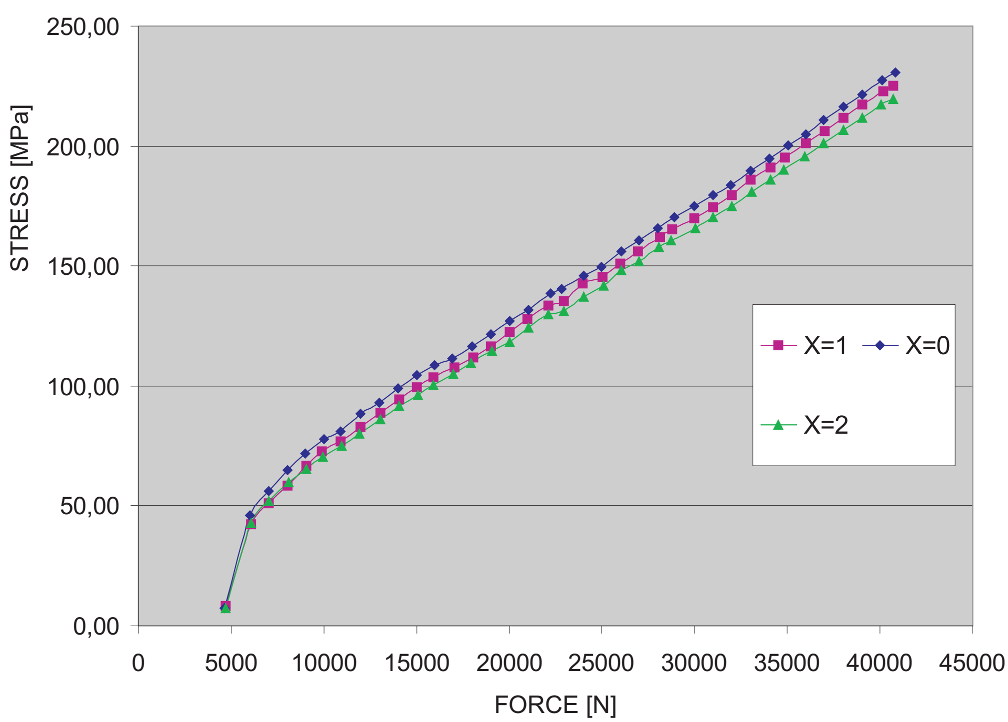



Fig. 13. Stresses under the screw head depending on the elongation force for position x = 0, 1, 2

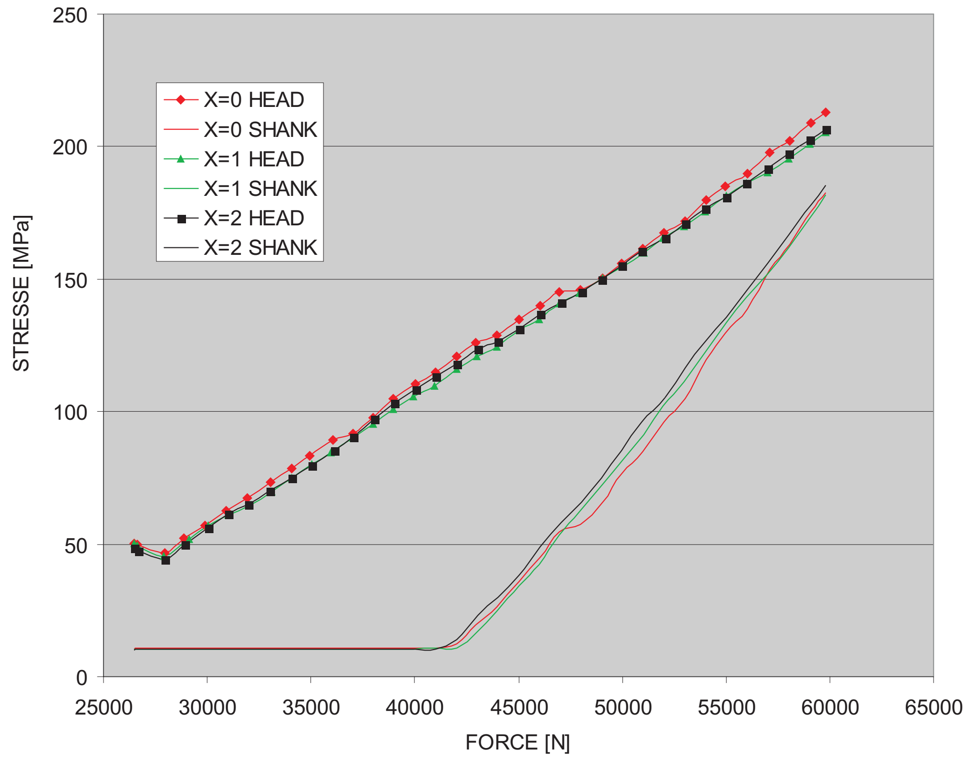



Fig. 14. Stresses under the screw head and screw shank depending on the elongation force for position x = 0, 1, 2

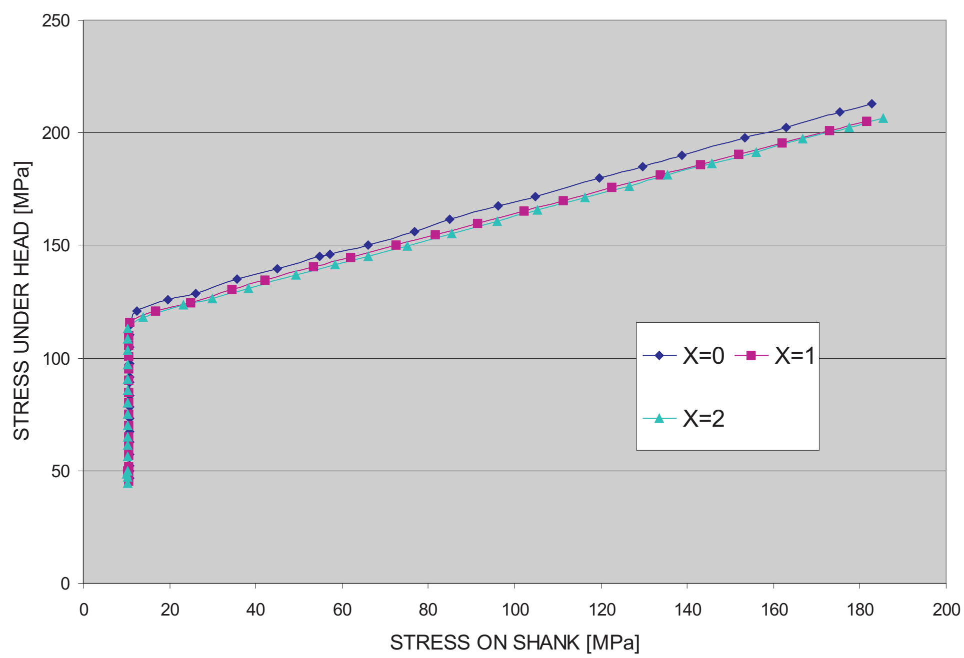



Fig. 15. Stresses under head of the bolt depending on stresses on the shank for position x = 0, 1, 2

Experimental research has confirmed differences between stresses under the bolt head and the ones taking place in the bolt shank, these differences increasing during the increase of load.

## 6. Conclusions

1. The presented models and methods of analyses of joints do not include all parameters which can occur in screw joints and resulting from the applied design solution, production engineering, technology of installation and conditions of exploitation. The suggested model introduces the next important factor for analysis, namely the interaction between a bolt and washer.

2. Analysis of mechanical behaviour of joint's elements subjected to axial stretching were employed for verification of the accepted model. This method is relatively simple and easy to realize, but with certain limitations resulting from difficulty of applying proper measuring gauges as well as the limited tested zone.

The obtained results of research show certain convergence with the results of the FEM analyses and they allow one to predict the behaviour of individual connection elements by means of the model analysis.

3. Taking into account the limited character of the experiment, a relatively small range of load and elastic deformations, further experimental and modelling research is being conducted with the use of elastoptical methods.

4. The results of modelling and experimental verification research include areas which have not been researched so far. The results of this work indicate necessity of continuation research work concerning further optimization of connection elements, their shape and production engineering in order to obtain higher exploitation parameters and more reliability.

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