

COMPARATIVE STUDIES OF THE SEATINGS OF PROPULSION PLANTS AND AUXILIARY MACHINERY ON CHOCKS MADE OF METAL AND CAST OF RESIN PART I. MOUNTING ON STEEL CHOCKS

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ABSTRACT

This paper presents a description and the results of experimental studies of the deformation, friction and structural damping occurring in foundation bolted joints of propulsion plant components and auxiliary machinery that is rigidly mounted on sea-going ships. The rigid mounting of these devices to the ships' structural foundations can be implemented in a traditional way, i.e. on chocks made of metal (usually of steel), or in a modern way, i.e. on chocks cast of resin, specially designed for this purpose. The main goal of this study is to perform a comparative analysis of these two solutions and to give a scientific explanation for why chocks cast of resin perform better in machinery seatings than the steel chocks traditionally used for this purpose. The paper consists of two parts. Part I presents the details of the rigid mountings of machinery to the foundations, and contains the results of experimental studies performed on a model of a foundation bolted joint with a traditional steel chock. Part II contains the results of similar studies carried out for a model of a bolted joint with a modern chock cast of resin. Next, a comparative analysis and evaluation of the results obtained for both investigated bolted joints was carried out, and conclusions were formulated to highlight important aspects of the problem from the point of view of science and engineering practice.

Keywords: sea-going ships, propulsion plants, auxiliary machinery, seating, bolted joints, chocks

INTRODUCTION

The intensity of vibration, quality of operation, reliability and durability of propulsion plants and auxiliary machinery installed on sea-going ships depend not only on their design and workmanship quality; to a large extent, they also depend on the type and quality of their seatings onto the structural foundation. In the modern approach to this problem, the plant components, their foundations and fastening systems should form an integrated mechanical (dynamic) system, with a stable structure and stable parameters.

This problem will be discussed here using the example of the seating of a main propulsion engine on a sea-going ship's structural foundation, due to the special role of these objects and the particularly high requirements which these engines must meet. The issues discussed here are, to

a greater or lesser extent, also relevant in relation to the seatings of other propulsion plant components and other types of auxiliary machinery installed on sea-going ships, and in particular those that generate large dynamic forces and vibrations.

The mounting of a large propulsion engine onto a ship's structural foundation must be done in accordance with the installation instructions of the manufacturer and the regulations set by the classification society supervising the construction of the ship. According to instructions set out by the engine manufacturer Wärtsilä [9] and the regulations of Germanischer Lloyd (GL) [2], there are generally two methods of mounting machinery to a ship's structural foundations: rigid mounting and resilient mounting.

The term 'rigid mounting' is used when a component of a propulsion plant or another item of equipment is attached

directly to the ship's structural foundation, or if rigid chocks are properly fitted between the foundation and the machine base. A resilient mounting is a connection between the plant components to the structural foundation of the ship, using resilient mounting components such as rubber pads or special vibrato-isolators.

The study presented in this paper considers only the rigid mountings of the plant components to the foundations (understood in the manner specified in the regulations [2, 9]). They involve the majority of the main propulsion engines and gearboxes, as well as most auxiliary equipment installed under and on the deck of sea-going ships. An example of a rigid mounting of a propulsion engine to a ship's structural foundation is shown in Fig. 1 [3].

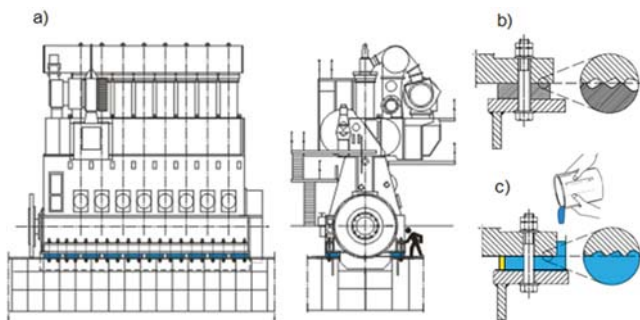


Fig. 1. (a) Schematic diagram of a rigid mounting of a propulsion engine to a ship's structural foundation using (b) chocks made of steel and (c) cast from resin

The use of chocks in the mounting of heavy machinery to the foundations arises from the difficulty of properly fitting the large bearing surfaces of the machine base and the foundation top plate with each other, and also from the necessity of properly setting and aligning the associated plant components, for example the main engine with the gearbox and the drive shaft of a ship.

According to the existing regulations [2, 9], the chocks used in this kind of rigid machinery mounting can be made of metal (steel or possibly iron) or cast in a resin compound specially developed for this purpose. The use of conventional steel chocks (Fig. 1b) has a long tradition. It also has many significant disadvantages, particularly in regard to the mounting of large propulsion engines on sea-going ships. This kind of mounting is characterised by high complexity, difficult and time-consuming installation, low technical quality and significant unreliability in operation [1, 3, 8]. Fastening systems of this type require continuous monitoring and frequent maintenance.

Various methods have been developed to improve the situation, for example using reusable adjustable metal chocks [4]. Although these are generally acceptable [9], they have not been widely used in the ship building industry. Significant progress was made in this area only after developing and applying special pourable resin compounds. Making chocks of these materials by pouring in place (Fig. 1c) greatly simplified the technology of machinery seating, shortened the time required and reduced the cost of implementation. As an unexpected result, this also significantly improved the technical quality of the seatings in comparison to that of conventional steel chocks.

In view of this, an important question arises as to why chocks cast in resin perform better in this kind of rigid mounting, i.e. guarantee better operation and higher stability, reliability and durability of the seated devices than conventional chocks made of steel, despite the fact that resin has much lower strength and stiffness than steel.

The answer to this question is not a simple or an easy one. Adequate knowledge is needed of the physical phenomena occurring in the real foundation bolted joints, in terms of their installation and operation. In order to obtain this knowledge, several experimental investigations were carried out. These investigations were performed on models of the foundation bolted joints with a traditionally made steel chock and with a modern chock cast in a resin compound designed for this purpose. A comparative analysis of their behaviour under similar loading conditions was then carried out.

The dynamic load acting on the foundation bolted joints of machines during their operation may be normal and tangential to the supporting surface. Taking these possibilities into account, separate studies were conducted using suitable models of foundation bolted joints under both normal and tangential loads. In earlier work [5, 6], a description and test results for models of foundation bolted joints subjected to normal loads were presented. The current paper presents a description and the results of experimental studies carried out on models of foundation bolted joints with a steel chock and a chock cast of resin, subjected to a constant normal load and time-varying tangential loads. The results obtained from these studies are presented in two parts, constituting two separate publications. This article forms Part I of this research.

THEORETICAL FUNDAMENTALS

The starting point for these investigations was the idea that the general division of structural connections into rigid and resilient joints, which is adopted widely in engineering theory and practice, makes sense only in a certain specific context. In fact, there are no perfectly rigid structural connections. The modelling of the foundation bolted joints of propulsion plants and auxiliary machinery mounted to the structural foundations of ships as a group of rigid connections (as in [2, 9]), is a considerable simplification. This suggests that their deformations are small and irrelevant, and that their description and analysis can be omitted, distorting our picture of the phenomena occurring within them and hindering a proper understanding of the role they play in seated objects.

The results of many studies and practical experience show [6] that fastening systems of many machines and devices that are made in the traditional manner using steel chocks are often the weakest links in the entire mechanical systems in which they are used, and do not ensure adequate integration; they therefore cause many problems to the users of these different machines and equipment.

It is worth noting that the number of chocks in the seatings of large machines and devices on foundations is usually much larger than three, and is often a dozen or even several dozen.

An example of a main propulsion engine of a sea-going ship is shown in Fig. 1a. In such cases, the fastening systems form multifold, statically indeterminate mechanical systems, which are not only very difficult to model and calculate but are also difficult to implement. This particularly applies to the appropriate individual fitting of each foundation chock; even a very small mismatch in their heights can cause large unpredictable mounting stresses and deformations in the whole mechanical system in which they are used.

Due to shape errors, i.e. the waviness and roughness of the mounting surfaces, a perfect fit between metal chocks over their entire nominal bearing surfaces and the foundation top plate and machine base is practically impossible. True contacts between two such surfaces develop only between the uneven tops of each surface (Fig. 1b). The true contacts are separate micro-areas where the uneven tops from one surface are squeezed against the uneven tops of the other surface. Depending on the conditions, the asperities may deform elastically or elastic-plastically.

The real area of contact in such chocks is only a very small fraction of their nominal contact area. This has very serious practical consequences, particularly under dynamic loads. However, an exact fit between contact surfaces is possible and easily achievable when using resin chocks that are poured in place (Fig. 1c).

As noted above, the foundation bolted joints of machines (with both steel and cast resin chocks) should not be regarded as rigid connections. In real foundation bolted joints, complex deformations and closely related physical phenomena (vibration, friction, structural damping, wear) occur during operation. These have a major impact not only on the behaviour of these joints, but very often also on the entire mechanical system of which they constitute a part. A thorough understanding of these phenomena is therefore necessary in order to evaluate the quality of these joints and to achieve a proper understanding of the role they play in machines and devices seated on foundations. In particular, it is required for a proper understanding and explanation of why chocks cast in resin in the seatings of machines perform better than the steel chocks traditionally used for this purpose.

TEST METHOD

Experimental studies were performed on specially designed and constructed bolted joints, as shown in Fig. 2. These were models of the foundation bolted joints of machines, and consisted of two interconnected steel members, a spacer member (quadratic, with a side of 80 mm and a thickness of 20 mm, with a hole of 24 mm in the centre) located between them and a fixing bolt M20x1.25 with a nut. The connected steel members represented some clippings of the structural foundation top plate and the base of the machine, respectively, and the spacer member between them, which was made of steel or cast of resin, represented the chock of the seated machine (Fig. 1). The bolt with glued strain gauges was properly calibrated and served as a dynamometer to achieve a proper mounting clamp for the connected elements and

control during the tests. A special spring washer and an axial ball bearing with a spherical washer (NSK Bearings 52204U) were placed under the nut in order to increase the elastic flexibility of the tensioning system and the uniformity of the pressure acting on the chock.

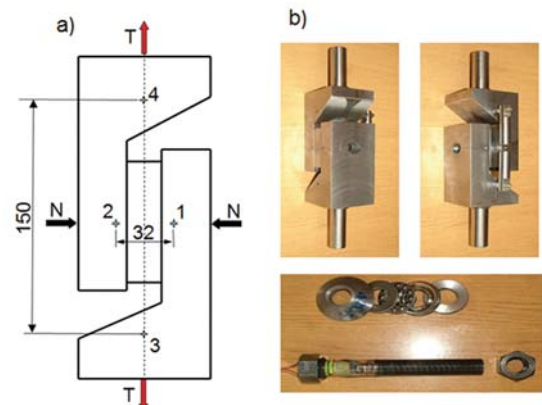


Fig. 2. Bolted joint used in a study of the characteristics of tangential displacements, friction and structural damping: (a) schematic with points marked for measurement of the relative displacements; (b) view of the joined elements and the bolt with glued strain gauges to measure the clamping force N

The joints were designed for a study of the characteristics of tangential displacements, frictional forces and structural damping, at a constant clamping force and with time-varying tangential loads. The tests were performed on a modern servo-hydraulic testing machine (INSTRON 8850). The heads of the machine were equipped with hydraulic clamp jaws (Fig. 3a). This enabled dynamic loading of the tested joint by applying tensile and compressive forces, with a continuous and smooth transition through zero.

Instron extensometers (Fig. 3) were applied to measure the relative displacements of the joined elements. The displacements were measured on the side surfaces at two locations, between points 1 and 2, and between 3 and 4, as shown in Fig. 2. Extensometer 1 (Fig. 3) measured the relative tangential displacements of the combined elements at points 1 and 2 (Fig. 2), while extensometer 2 measured the mutual change in the distance between points 3 and 4, which were at a considerable distance from each other ($l = 150$ mm), along the line of operation of the loading force T (Fig. 2a).

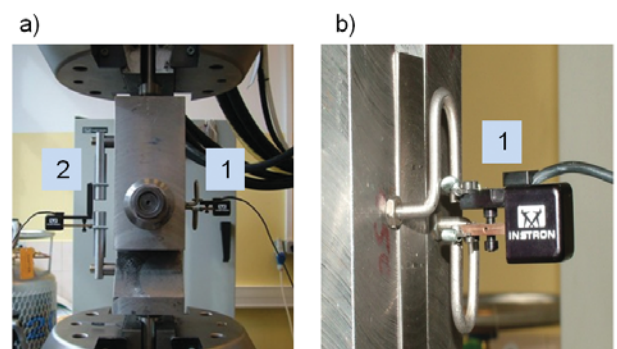


Fig. 3. Testing of a bolted joint with extensometers to measure the relative displacements of the connected elements: (a) general view; (b) fragment

We used Instron's Wave Matrix (v. 1.5.318), an intelligent software package designed for the dynamic testing of materials and components. It formed the basis for creating special programs for the dynamic loads or displacements to be applied to the system and for the computer-control of their implementation, with a very high accuracy and repeatability. Dynamic tests are generally characterised by the fact that all input and output quantities are treated as functions of time.

The studies were first performed for the bolted joint with a chock made of steel, as traditionally used for this purpose, and then (in Part II) with a chock cast in resin.

IMPLEMENTATION AND RESULTS OF THE STUDIES

The contact surfaces of the two joined elements were milled, and the measured roughness parameters of these surfaces had values $R_a = 2.47\text{--}2.51\ \mu\text{m}$, $R_z = 13.40\text{--}13.50\ \mu\text{m}$. The contact surfaces of the steel chock (thickness 20 mm), were turned and had roughness parameters $R_a = 2.29\text{--}3.60\ \mu\text{m}$, $R_z = 11.69\text{--}17.25\ \mu\text{m}$. The preloading force in the bolt was $N = 51700 \pm 200\ \text{N}$. This induced a bolt tensile stress of $\sigma = 199.7\ \text{MPa}$ and an average surface pressure on the chock of $p = 8.69\ \text{MPa}$. When determining the bolt mounting tension, the requirements of the classification societies in the maritime industry were taken into account. For the main engines and gears, the average allowable surface pressure for the material used was $5\ \text{N/mm}^2$ (PRS, GL, LRS, RMRS, BV, DNV), $15\ \text{N/mm}^2$ (ABS), $15\ \text{N/mm}^2$ (PRS) for mechanisms for which coaxiality is not required, $30\ \text{N/mm}^2$ (PRS) for anchor and mooring winches, and $< 60\ \text{N/mm}^2$ (PRS) for temporary loads.

The tests carried out using the testing machine are shown in Fig. 3a. Several experiments were carried out on the tested joints by applying various computer-controlled programs for loads or displacements at the input. Detailed descriptions and results of these experiments are contained in an earlier report [7]. Some of the results of these experiments are presented below.

EXPERIMENT 1

This experiment studied the relative tangential displacements of the connected elements, as measured at points 1 and 2. The system was loaded with an increasing force T , with some interruptions, and with several unloading (to zero) and re-loading cycles. The results of this experiment are shown in Fig. 4. The last unloading curve on the graph did not stop at zero, but passed the zero line in a continuous and smooth way, meaning that the load acting on the tested system changed from tension to compression. The compressive load was continued until the initial state of the tested system was reached, i.e. a zero value of the force T (shear stress τ) and tangential displacement δ_t (Fig. 4).

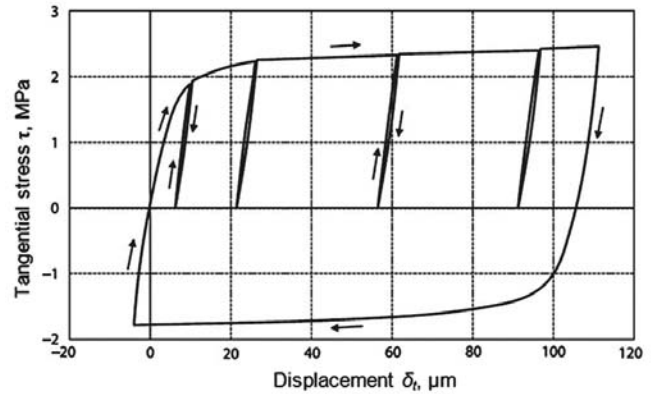


Fig. 4. Relationship between tangential stress τ (force T) and relative displacement δ_t of the connected elements for the bolted joint with the steel chock

During unloading and re-loading, the system behaves elastically until the re-loading force reaches the point on the graph at which the unloading process begins. After reaching this point, and with continued loading, further micro-slips occurred with a very low increase in the loading force. During unloading and re-loading, some slight elastic hysteresis loops were clearly visible, as shown in Fig. 4.

The mechanism of the relative tangential displacements δ_t at points 1 and 2, caused by the force T acting tangentially to the contact surface, is illustrated in Fig. 5. In this case, the measured relative displacements δ_t are the result of the elastic shearing strains γ of the material of the joined elements ($\delta_t' = \gamma h$), the elastic tangential contact deformations of the surface asperities and micro-slips occurring at the interfaces of the interacting surfaces.

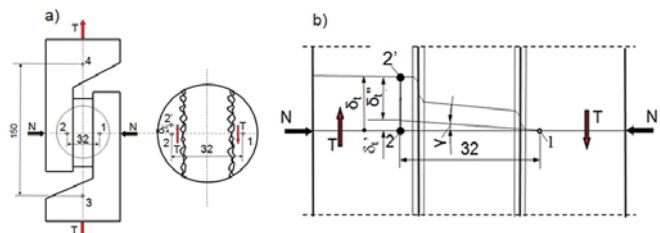


Fig. 5. Schematic diagrams of (a) loading and (b) deformation for the tested joint with the steel chock

The graph in Fig. 4 also shows the variation in the frictional force, which balances the given external load T . From these results, it follows that the development of the frictional force is associated with the tangential displacement dt of the connected elements, which only have an elastic nature to a small extent. The micro-slips play the dominant role here, and can achieve significant values before the contact is broken and macro-slipping occurs. The values of the elastic tangential contact deformations (the tangential contact flexibility) can easily be determined at any point by unloading and re-loading the joint (Fig. 4).

Note: In the above experiment, the tested joint was not loaded to the point of breaking of the contact and the occurrence macro-slipping, due to the possible damage to the costly extensometers.

EXPERIMENT 2

The purpose of this experiment was to investigate the behaviour of the system for a given program of forced oscillation of the relative displacements of the connected elements. A schematic of the test joint and the program of assumed relative displacements (sinusoidal oscillations), measured at points 1 and 2 are shown in Fig. 6.

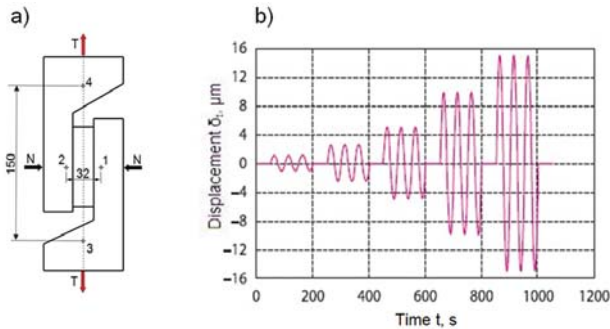


Fig. 6. (a) Schematic of the test joint and (b) the assumed program of sinusoidal oscillations; the amplitudes of these oscillations had values of $\delta_a = 1, 2.5, 5, 10$ and $15 \mu\text{m}$

The task of the testing machine equipped with the program for dynamic research was to provide an adequate time period for loading, which would ensure the implementation of the assumed oscillations shown in Fig. 6b. Figure 7 shows the results of this experiment.

These graphs show (a) the time evolution of the oscillations (the relative displacements $\delta_i(t)$ of the connected elements measured at points 1 and 2) performed by the testing machine; (b) the time evolution of the force $T(t)$ used to realise the assumed displacements; (c) the time evolution of the average shear stresses $\tau(t)$ acting on the chock; and (d) the relationships between the stresses τ and displacements δ_i and the hysteresis loops for the selected oscillation amplitudes shown in Fig. 7a. The high accuracy of the sinusoidal displacements (oscillations), with very small amplitudes (from 1 to $15 \mu\text{m}$), and the irregular complex time evolution of the force needed for their implementation should be noted. The hysteresis loops (Fig. 7d) show that the system behaves quasi-linearly only at sufficiently small displacement amplitudes (max. $5 \mu\text{m}$). At higher oscillation amplitudes in the tangential direction, substantial micro-slips and large hysteresis loops are clearly visible. This shows the loss of the elastic integrity of the tested joint and the dissipation of a large amount of vibration energy.

A positive effect of this phenomenon is good vibration damping, and a negative result is the intensive wear of the interacting surfaces, which leads to rapid destruction of the joint. In well-designed bolted joints, significant micro-slipping should not take place. One important issue is to establish a reasonable limit for micro-displacements at a contact that allows for safe long-term operation of the bolted joints loaded by variable tangential forces.

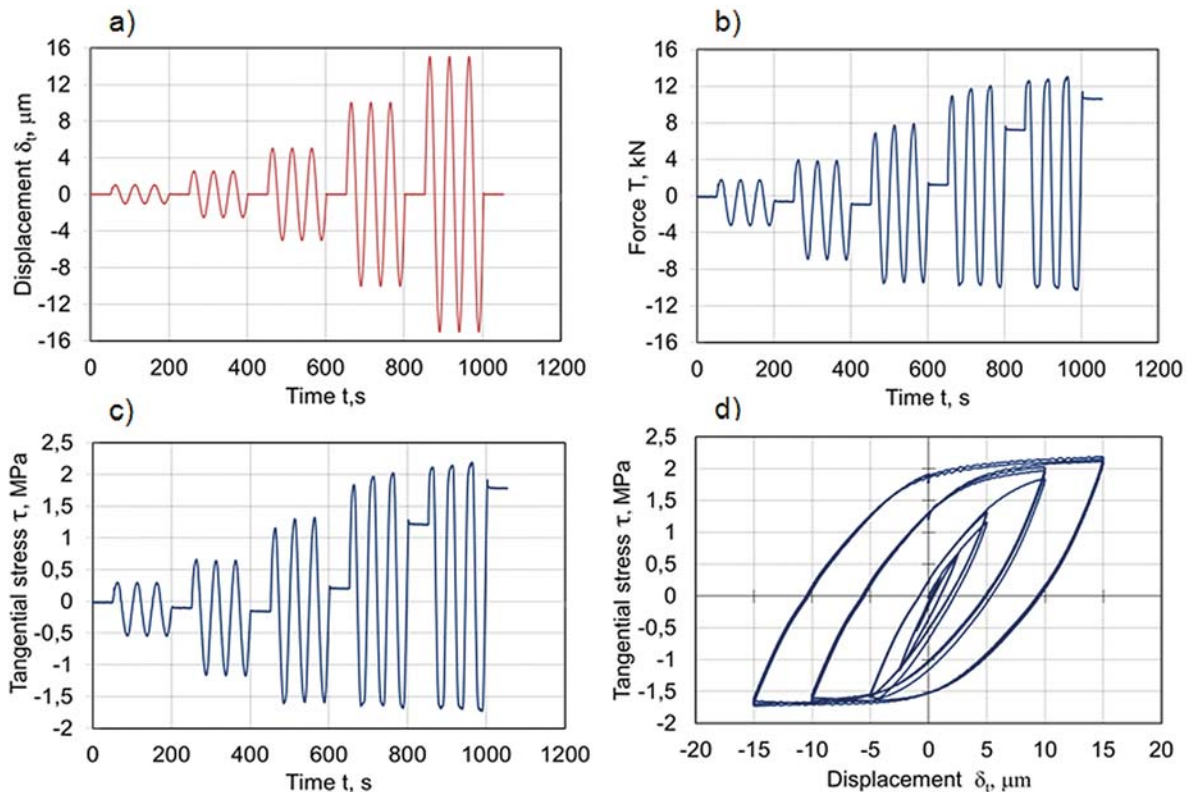


Fig. 7. Results of experiment 2: (a) time evolution of the implemented relative displacements of the connected elements using the testing machine; (b) time evolution of the force T used to realise the displacements; (c) time evolution of the average shear stresses acting on the chock; and (d) relationships between the stresses τ and displacements δ_i

EXPERIMENT 3

In experiment 3, the tested joint was loaded cyclically, with controlled displacements $s(t)$ of the head of the testing machine. The results of this experiment are shown in Fig. 8.

Figure 8a shows the given displacement program $s(t)$ (sinusoidal vibrations with amplitudes of 25, 50, 75, 100 and 125 μm) performed by the head of the testing machine. Figure 8b shows the time evolution of the relative displacements $\delta_t(t)$ of the connected elements (measured at points 1 and 2), and Fig. 8c shows the time evolution of the average shear stresses acting on the chock for the given kinematic excitations (shown in Fig. 8a). Figure 8d shows the relationships between the stresses τ and displacements δ_t and the hysteresis loops for the oscillation amplitudes shown in Fig. 8a.

The measurement results show (Fig. 8) that the tested system behaves in a linear elastic manner only at very small amplitudes of vibrations excited kinematically by the head of the machine (up to 0.05 mm). The system behaves nonlinearly at higher amplitudes of vibration. In such cases, complicated nonlinear relationships arise between the given harmonic excitation $s(t)$ and the response of the system. Significant micro-slipping occurs at the contact between the mating surfaces, which in conjunction with the friction forces form large hysteresis loops. Tangential forces carry out much of the work, and this in turn can cause high wearing of the interacting surfaces, and the quick loosening and destruction of this type of bolted joint.

CONCLUSIONS

This study has shown that foundation bolted joints with steel chocks, classified in the literature and in engineering practice as a group of rigid connections, are not perfectly rigid. During operation, in addition to small elastic deformations in the tangential direction, there can be significant micro-slips (plastic displacements) of the elements connected with each other. These cause abrasion of the asperity tops of the interacting surfaces, the occurrence of fretting corrosion, loosening of the foundation bolted joints and an increase in the intensity of vibrations. The result is not only rapid destruction of the mounting system, but often also intensive wear of many parts of the seated device and various failures. These phenomena and problems are well known in the engineering practice, as well as in shipbuilding. For comparison purposes, similar studies were performed for a bolted joint with a chock cast of resin. A description and the results of these studies and the conclusions that can be drawn are presented in Part II.

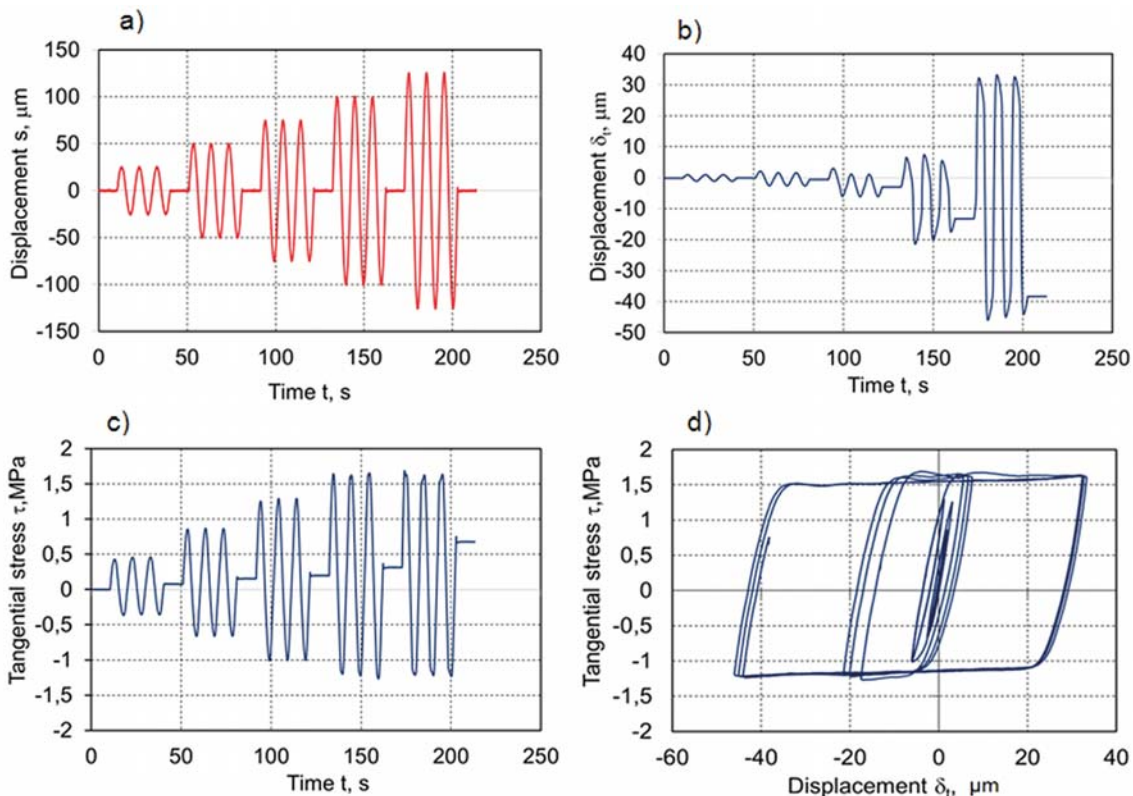


Fig. 8. Results of Experiment 3: a) time evolution of the assumed displacements $s(t)$ performed by the head of the testing machine; b) time evolution of the relative displacements $\delta_t(t)$ of the connected elements; c) time evolution of the average shear stresses acting on the chock; and d) relationships between the stresses τ and displacements δ_t .

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