Recommendations for the estimation of the strength of the railway wheel set press fit joint

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Abstract: Although press fit joints are very important in railway engineering, few authors have investigated the strength of press fit joints in terms of their application in railway vehicles. This study analyses the tribological parameters that influence the strength of press fit joints, especially the contact pressure and the static friction coefficient. This research was targeted towards the control of the friction coefficient value to obtain the required strength of a press fit joint in conditions close to the minimum contact pressure that results in less prestressed press fit joints. To estimate press fit joint strength, this study examines the use of minimum and maximum values of the friction coefficient as recommended in the literature considering wide tribological conditions, as well as the experimentally determined friction coefficient values for specific tribological conditions. The study also points out the discrepancy between current railway standards and engineering practice considering the inspection of wheel set press fit joint strength.

Keywords: press fit joint strength, wheel set, static friction coefficient, tribological parameters, railway vehicle

1 INTRODUCTION

Press fit joints are widely used in engineering practice due to their simple form and assembly process. Because of their ability to carry massive loads, press fit joints are commonly used in the drive units of railway vehicles for the following assemblies: gear-shafts, bearing-shafts, wheel-axles, brake disc-axles, etc. One of the disadvantages of press fit joints is the high prestress level of parts around the contact area. Exploitation loads can produce even more complex stress conditions. Considering the stress condition of assembled parts, press fit joints can be divided into elastic, elasto-plastic, and plastic assemblies, with elastic press fit joints having the widest application.

Numerous authors have investigated press fit joints. Theoretical research has been mostly directed towards calculating procedures and load-transfer principles, while experimental research has considered the increase of press fit joint strength. The calculation procedures of press fit joints with elastic deformations of the contact area are very well explained in theory [**1–4**]. Gamer [**5, 6**] made a mathematical model of press fit joints with elasto-plastic deformations. For high dynamic loading, contact surfaces of press fit parts were coated with a phosphate, metallic, or sticky layer to prevent local sliding [7]. The influence of surface roughness on the strength of press fit joints was researched by Rammoorthy *et al.* [**8**], as well as by Kato *et al.* [**9**], who investigated metallic and ceramic materials' contact surfaces coatings. Research into the strength of press fit joints assembled from parts of different materials and exposed to high-temperature conditions has also been performed [**10**]. Experimental determination of contact pressure in press fit joints with the excessive tightening (occurrence of part surface damages or high levels of residual stresses) was performed

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by Lewis *et al.* [**11**] by the application of a method based on ultrasonic reflection.

Even though press fit joints are very important in railway engineering, only a few authors have investigated press fit joints regarding their application in railway vehicles. Benuzzi and Donzella [**12**] proposed a methodology for predicting the press fit curve in the assembly of a railway axle and wheel. Their methodology was based on friction measurements on samples taken directly from assembled components as the input data together with the wheel set geometry, design interference, and material characteristics. The authors varied the contact pressure and sliding speed so as to change the lubricant conditions of contact surfaces.

As far as high-speed railway system safety is considered, most of the research and experimental investigation has been aimed towards the determination of fatigue strength, especially fretting fatigue. In Europe, it is believed that the larger diameter of a press fit joint renders the fillet a critical part, while in Japan, the fatigue strength of a press fit joint is increased by the induction hardening method [**13**].

An analysis of the above-mentioned research shows that most of the authors, even those who have studied specific aspects of press fit joints regarding railway engineering, recognize press fit joint strength as being directly dependant on contact pressure (minimum and maximum), i.e. on the performed tightening. Moreover, the existing engineering calculations of press fit joint strength take recommended values given by various authors, lubricant manufacturers, etc., as the values of the friction coefficient between contact surfaces. However, values of the friction coefficient vary according to part materials, the manner of assembly, lubricants, etc.

Numerous authors have studied theoretical and experimental tribological issues, but not many of them have researched press fit joints as tribomechanical systems [**14, 15**]. Press fit joint strength is equal to the friction force, which is the main reason why tribological parameters should be the objective of press fit joint research.

In the production and maintenance of railway vehicles, engineers have problems with the inspection of press fit joint strength, because railway conventions are not compatible with industrial practice. Wheel set press fit joints are often rejected and reassembled because pressing-on forces are not in the permissible range as defined by railway standards. Although the values of geometrical and technological parameters are in the standardized range, in some cases, the maximum values of the pressing-on forces do not satisfy the recommendations of railway standards. To investigate the mentioned problem and to acquire more knowledge about tribological parameters and their influence on press fit joints, the Faculty of Mechanical Engineering, University of Niš, and the factory specialized in manufacturing locomotives and other railway vehicles, MIN Lokomotiva in Niš, Serbia, carried out this research on the press fit joints of railway vehicle drive units. The results of this research show a significant variation of the friction coefficient depending on tribological conditions. Therefore, press fit joint strength can be changed by the variation of applied lubricant, surface machining, assembly process, surface roughness, and so forth. This study proposes new recommendations for the estimation of press fit joint strength using minimum and maximum values of the experimentally determined friction coefficient for specific tribological conditions.

2 RAILWAY STANDARDS AND PRACTICAL EXPERIENCES CONCERNING PRESS FIT JOINTS

Leaflet 813 of the International Union of Railways, titled 'Technical specification for the supply of wheelsets for tractive and trailing stock – Tolerances and assembly' (Leaflet UIC 813) [**16**], defines the characteristics of press fit joints of solid-core wheels, wheel plates, tired wheels, and other component parts of axles. It also specifies assembly tolerances as well as inspection and delivery conditions. Furthermore, it identifies conditions for mounting solid-core wheels, wheel centres, tired wheels, axle-mounted brake discs, generator pulleys, gear wheels, chain wheels, and other components specified by the purchasing railway for wheelset assemblies onto wheelsets by pressed-on or shrunk-on fitting (under the effect of heat).

Figure 1 shows the electromotor-axle drive of the electrolocomotives series JŽ 441 and JŽ 461 of the national railway operator in Serbia, 'Serbian Railways' [**17**]. The torque is transmitted over eight press fit joints: wheel–flange of wheel (I – 2 units), wheel–drive axle (II – 2 units), gear hub–drive axle (III – 1 unit), gear-shaft (IV – 1 unit), coupler-shaft (V – 1 unit), and gear (of gear shaft) torsion shaft (VI – 1 unit).

Engineers can use recommendations from Leaflet UIC 813 to choose the tightening for press fit joints in the design phase. Figure 2 presents the tightening between seats and bores for different diameters and corresponding tolerance limits [**16**].

Joints like wheel-axle, gear-axle, and braking disk-axle are most often assembled as press fit joints. During the assembly of a press fit joint, it is necessary to record the force of the pressing-on process. Press fit joint strength is estimated on the basis of this record. According to Leaflet UIC 813, the assembly equipment should include a properly calibrated measuring device that is able to record the diagram of the pressing-on force as a function of the position of the wheel in relation to the seat on the axle during the pressing-on process. Figure 3 shows a force–motion diagram recorded at the factory MIN Lokomotiva in Niš, Serbia, for the electrolocomotive wheel-axle pressing-on process.

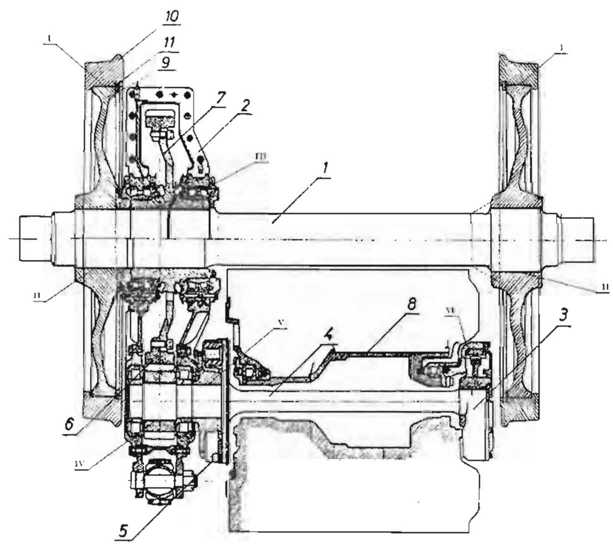


Fig. 1 The electromotor-axle drive: 1, axle; 2, housing; 3, gear coupler; 4, torsion shaft; 5, elastic coupler; 6, small gear; 7, large gear; 8, rotor shaft; 9, wheel; 10, wheel flange; and 11, ring for flange fixture

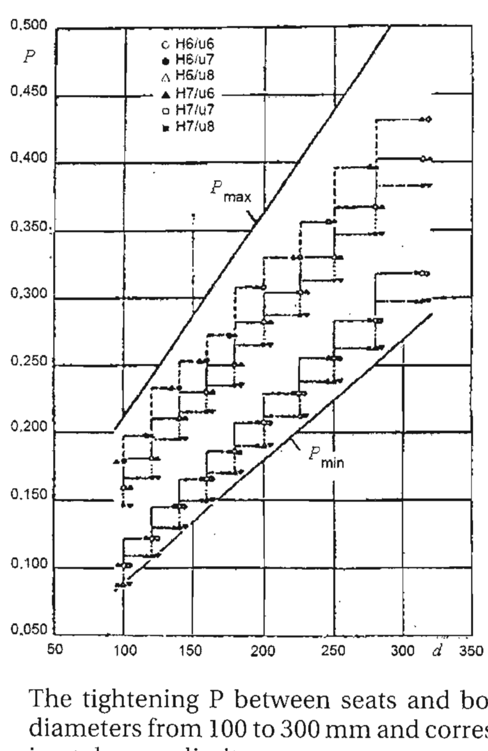


Fig. 2 The tightening P between seats and bores for diameters from 100 to 300 mm and corresponding tolerance limits

As stated in Leaflet UIC 813, the permissible value of the fitting-on pressure is calculated by [**16**]

$$P_F = a \cdot D \tag{1}$$

where D is the nominal diameter of the hub seat and a the coefficient of the type of wheel and lubricant. The values of the coefficient a are given in Leaflet UIC 813.

Table 1 contains the values of the calculated pressing-on forces according to the calculated procedure from DIN 7190 [**18**] and values obtained for the examined electrolocomotive wheel-axle assemblies. The given values are calculated using measured values of the tightening, surface roughness, and the mean value of the static friction coefficient $\mu = 0.1$ according to the recommendation of the manufacturer ($\mu = 0.09$–0.11) for the used lubricant LOCITE Wheel mount LT311. The obtained values are compared with the boundary values of pressing-on forces for wheel-axle assemblies given by the following standards:

(a) THYSSEN (manufacturer of wheel sets, Federal Republic of Germany) 898–1572 kN;
(b) UIC 813 687–1488 kN;
(c) SRPS P.F2.010 (National standard of the Republic of Serbia) 801–1374 kN.

The values of the obtained pressing-on forces (Table 1) marked as '–' are not satisfactory according to the standards and require a new assembly process. The values of the obtained pressing-on forces (Table 1) marked as '+' are satisfactory according to the standards. The values of the obtained pressing-on forces (Table 1) marked as '(+)' are not satisfactory according to the standards and require further testing of counter-pressure [**16**].

The obtained pressing-on forces (Table 1) for some of the press fit joints do not match the recommended values given in Leaflet UIC 813 or other standards, but that does not explicitly mean that these press fit joints do not have adequate exploitation characteristics. The range of obtained pressing-on forces in engineering practice is greater than the range defined by Leaflet UIC 813 and/or other standards, although the values of geometrical and technological parameters are the same. This can be explained by the dispersion of the static friction coefficient values caused by different tribological conditions during the assembling process, as it is not possible to provide identical properties of contact surfaces and the same assembling conditions. The stochastic character of real static friction coefficient value and the inability to

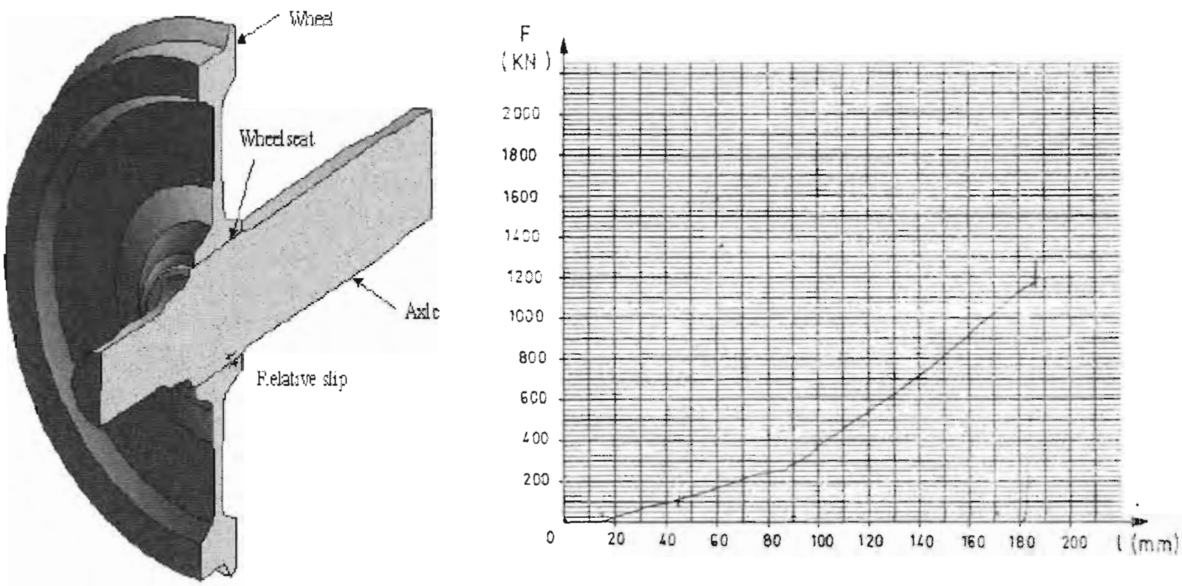


Fig. 3 The force–motion diagram for the electrolocomotive wheel-axle pressing-on process

precisely predict its value cause the stochastic deviation of obtained and calculated pressing-on forces.

3 FACTORS THAT INFLUENCE THE STRENGTH OF PRESS FIT JOINTS

The exploitation load, which causes tangential and/or axial forces on contact part surfaces in press fit joint, is used as the input information for the design of press fit joints. To transfer noted forces from one part to another, it is necessary to obtain a required value of friction force that resists sliding between the parts in contact. The calculation in the design phase of press fit joint results in a given estimation of the tightening value that ensures the required service conditions, i.e. nominal values and tolerances for parts in a press fit joint.

The press fit joint strength for an axial force load can be calculated as

$$F_{ag} = \pi d l p_{\min} \mu_a \tag{2}$$

while the press fit joint strength for a torque load can be calculated as

$$M_{og} = \frac{\pi d^2 l}{2} p_{\min} \mu_t \tag{3}$$

Table 1 Comparison of the calculated and obtained values of the pressing-on forces

No	Axle no	Wheel no	Pressing-on force Calculated (kN)	Obtained (kN)	UIC 813-V	THYSSEN	SRPS P.F2.010
1	2452A	2230	1605	1910	–	–	–
2	2452N	0263	1566	1860	–	–	–
3	300A	2201	1266	825	+	+	+
4	300N	8967	1630	1400	+	+	(+)
5	2847A	8929	1487	1730	–	(+)	–
6	2847N	8990	1600	1850	–	–	–
7	2490A	8921	1603	1325	+	+	+
8	2490N	2230	1615	1490	(+)	+	(+)
9	830A	2228	1566	1530	(+)	+	–
10	830N	2231	1455	1450	+	+	(+)
11	1198A	8977	1333	1640	–	(+)	–
12	1198N	2253	1632	1330	+	+	+
13	682A	2221	1499	1525	(+)	+	–
14	682N	8969	1614	1725	–	(+)	–
15	561A	2216	1422	1630	(+)	(+)	–
16	561N	0265	1547	1590	(+)	(+)	–
17	46A	2225	1434	1675	–	(+)	–
18	46N	8974	1548	1190	+	+	+
19	339A	2247	1256	1560	(+)	(+)	–
20	339N	8978	1622	1475	+	+	(+)

where μ_a is the friction coefficient in the axial direction and μ_t the friction coefficient in the tangential direction, while p_{min} the minimum contact pressure.

The safety factor against sliding is calculated as

$$S_\mu = \frac{F_{ag}}{F_a} \quad \text{or} \quad S_\mu = \frac{M_{og}}{M_a} \tag{4}$$

where F_a is the exploitation axial force load and M_o the exploitation torque load, while F_{ag} represents the press fit joint strength for an axial force load and M_{og} the press fit joint strength for a torque load.

The safety factor against plastic deformations is calculated as

$$S_R = \frac{R_{pi}}{\sigma_{iu}} \quad \text{and} \quad S_R = \frac{R_{po}}{\sigma_{io}} \quad \text{for the outer part} \tag{5}$$

where R_{pi} and R_{po} are yield stresses of the inner and the outer parts, while σ_{ii} and σ_{io} the stresses in the inner and the outer parts, which can be calculated as

$$\sigma_{ii} = \frac{2}{1-\psi_i^2} p_{max} \quad \text{and} \quad \sigma_{io} = \frac{1+\psi_0^2}{1-\psi_0^2} p_{max} \tag{6}$$

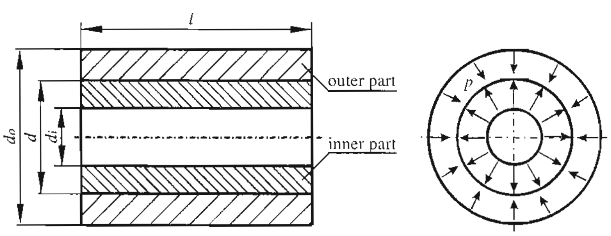


Fig. 4 Geometry of a press fit joint

where ψ_i and ψ_o are diameter ratios for the inner and the outer parts ($\psi_i = d_i/d$ and $\psi_o = d/d_o$), while p_{max} the maximum contact pressure.

As already noted, press fit joint strength is defined as the friction force at the contact surfaces of the assembled parts. It can be calculated using the nominal force F_N, or the contact pressure distributed over the contact surface area (A) as

$$F_\mu = \mu F_N = \mu p A = \mu p \pi d l \tag{7}$$

One can see from (7) that press fit joint strength depends on the joint diameter (d), the length of the press fit joint (l), the contact pressure (p), and the friction coefficient (μ).

The geometrical parameters (d, l) of a press fit joint (Fig. 4) are defined for specific design requirements. The contact pressure between the parts of a press fit joint generally depends on the geometrical parameters and material properties of the assembled parts. The friction coefficient has a large interval of values, and it is a function of numerous parameters such as the type of lubricant, surface roughness, surface hardness, presence of impurities, contact duration, etc. The Ishikawa diagram in Fig. 5 shows the main factors that influence press fit joint strength.

3.1 Contact pressure

In engineering practice, contact pressure is commonly calculated [1–4, 19] as

$$p = \frac{P_l}{d\left(\frac{K_i}{E_i} + \frac{K_o}{E_o}\right)} \tag{8}$$

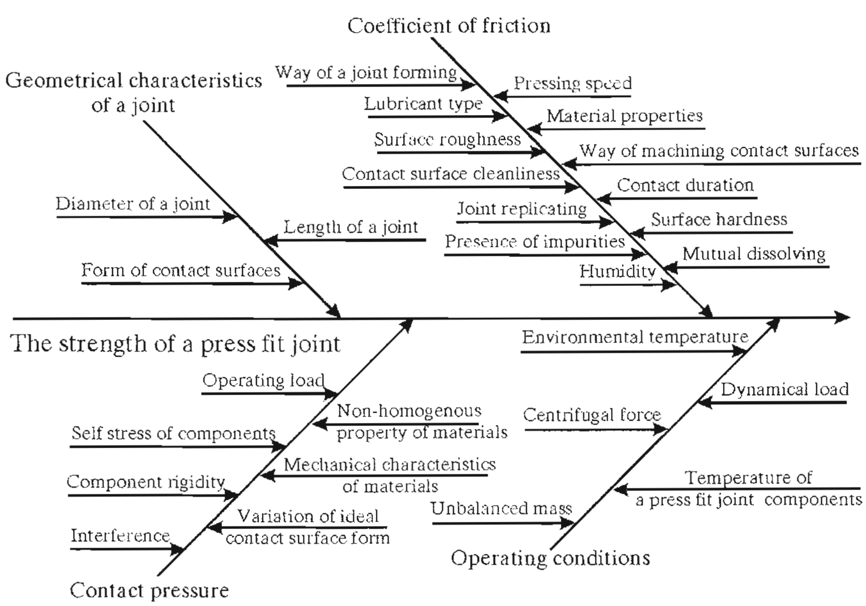


Fig. 5 Factors that influence press fit joint strength – the Ishikawa diagram

Factors K_i and K_o are elasticity coefficients and E_i and E_o the Young's moduli for the inner and the outer parts, respectively. The value P' represents the effective tightening of the assembled parts.

It is common in engineering practice to use the minimum (p_{min}) and maximum pressure (p_{max}). The minimum pressure is a necessary contact pressure that will provide the required strength of a joint, while the maximum pressure represents a maximum engineering value of the contact pressure that will not cause a joint damage, but still keeps the press fit joint functional.

Figure 6 [**11**] shows the measured contact pressure along the length $l = 90$ mm for different values of the tightening. While it is common to consider contact pressure as a constant, its value varies along the joint length.

The theory of elasticity and Lame's equations [**11**] consider parts in a press fit joint as thick pipes that result in a constant theoretical value of the contact pressure along the length of a press fit joint (l). The real distribution of the contact pressure varies along the length of a press fit joint because of the inhomogeneity of the material, the deflection of the assembled parts, prestresses, temperature, working conditions, etc.

The stress distribution of the joined parts is even more complex in the case of the non-symmetric parts. Based on the previous analysis, it can be concluded that the contact pressure of a press fit joint functionally depends on the geometrical parameters of parts, surface roughness, and the physical nature of joined parts.

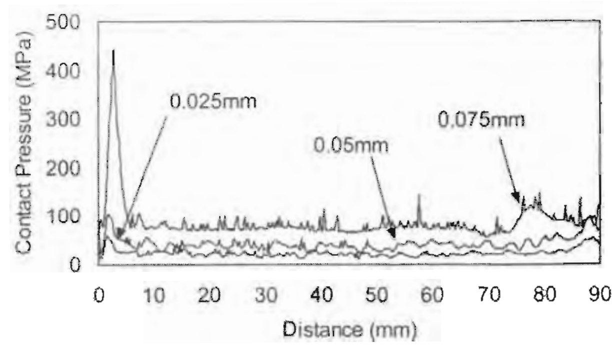


Fig. 6 Contact pressure for different tightening values: 0.025, 0.05, and 0.075 mm

3.2 Friction coefficient

The static friction coefficient value has the most influence on press fit joint strength. Moreover, it is of utmost importance for the prediction of the real value of press fit joint strength. Although there are some mathematical models for calculating static friction coefficients, they are not useful enough for engineering practice, so in majority of cases, the friction coefficients are experientially estimated.

Under permanent conditions, the coefficient of friction value is not a constant and may vary in a certain range even for the same material. The alteration

Table 2 An overview of different friction coefficient values for press fit joints

No	Author	Static friction coefficient		Material of parts	Note
		Force fit joint	Shrink fit joint		
1	Krause [**1**]	0.05–0.08	0.10–0.15	Steel/Steel	With mineral lubricant and in dry conditions
2	Vitas [**3**]	0.05–0.17	0.055–0.19	E295/E295	With machine lubricant
3	DIN 7190 [**18**]	0.08–0.11	0.12–0.20	Steel/Steel	With and without lubricants
4	Niemann [**4**]	$\mu_s = 0.08$–0.25 $\mu_k = 0.04$–0.17	$\mu_s = 0.13$–0.36 $\mu_k = 0.06$–0.14	Steel/Steel	With machine lubricant
5	Miltenović [**19**]	$\mu_s = 0.08$–0.25 $\mu_k = 0.05$–0.19	$\mu_s = 0.13$–0.24 $\mu_k = 0.08$–0.19	E295/E295	With mineral lubricant
6	Kragelskii [**14**]	$\mu_s = 0.15$–0.18	$\mu_s = 0.28$–0.37	Steel/Steel	Force fit joints lubricated by machine lubricants; internal part of shrink fit joints cooled
7	Haase [**22**]	$\mu_s = 0.12$–0.15 $\mu_k = 0.05$–0.09	$\mu_s = 0.28$–0.33 $\mu_k = 0.07$–0.09	Steel/Steel	With MoS2 lubricant
8	Beitz and Galle [**21**]		$\mu_s = 0.27$–0.35	Steel/Steel	Contact surfaces hardened and nitrated
9	Stamenković [**17**]	$\mu_s = 0.05$–0.26 $\mu_k = 0.04$–0.22		40Mn4/C45	Lubricated with tallow, Loctite, MoS2 and in dry conditions
10	Kragelskii [**14**]	$\mu_s = 0.43$–0.73	$\mu_s = 0.45$–0.69	steel/Steel	As in No.6 but the surfaces were covered with Zn, Cd, Ni, or Cr
11	Beitz and Galle [**21**]		$\mu_s = 0.38$–0.58	Steel/Steel	As in No.8 during dynamical bending and twisting
12	Romanos et al. [**23**]		$\mu_s = 0.41$–0.58 $\mu_k = 0.30$–0.48	42CrMo4/C45	With diamond grains on contact surfaces
13	Peeken et al. [**24**]		$\mu_s = 0.32$–0.62	42CrMo4/ 42CrMo4	With grains of SiC, B4C, and diamond during torque loading

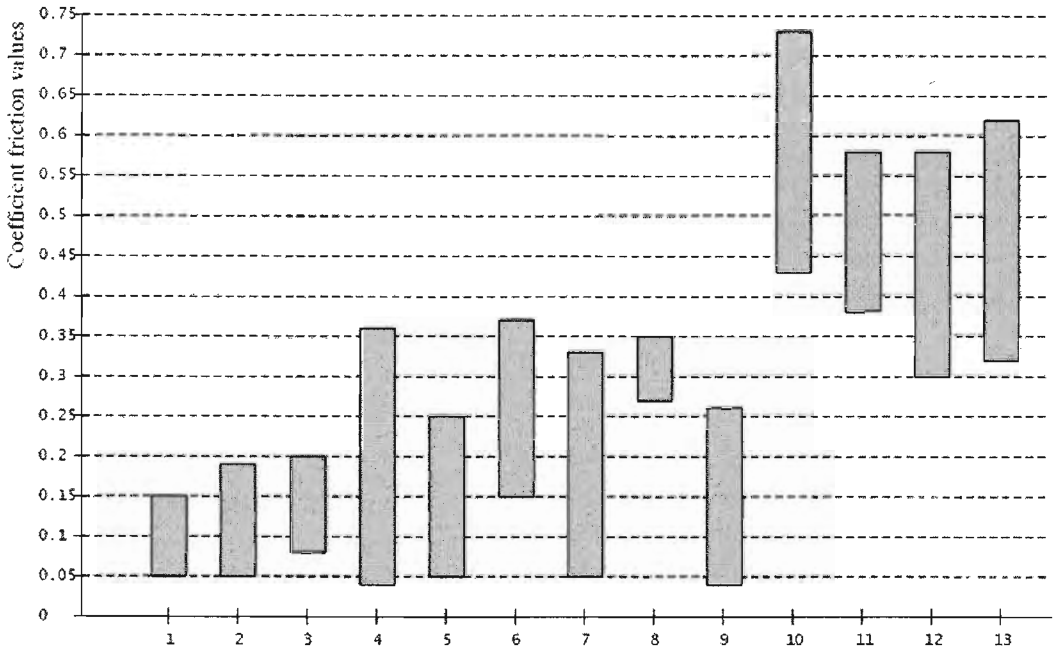


Fig. 7 Comparative graph of friction coefficient values according to different sources

of friction coefficient values is mostly stochastic, so one can only speak about the mean values of the friction coefficient.

The study of friction demands an interdisciplinary approach, since friction is the result of numerous interactive processes. Although it is simple to measure friction, it is much more complex to explain it [**20**].

Different recommendations are given in the literature for the friction coefficient values that engineers should use during design. Table 2 provides an overview of different friction coefficient values for press fit joints given by different authors [**1**, **3**, **4**, **14**, **17–19**, **21–24**], where μ_s is the static friction coefficient and μ_k the sliding friction coefficient. These empirical values are obtained for different materials of press fit joint parts, as well as for some lubricants.

It can be seen from Table 2 that the friction coefficient has a wide range of values, from 0.04 to 0.60 and even up to 0.70 in some special cases. Figure 7 shows the comparative graph of the coefficient values from Table 2.

Friction coefficient values depend on different parameters such as the nature and properties of used materials, contact pressure value, lubricant film properties, contact surfaces roughness, contact time, chemical interaction, presence of external bodies in the contact zone, manner of assembly, cleanness of contact surfaces, etc. [**15**, **25**, **26**].

Soft metals have higher friction values than hard metals, but the overall conclusion is that hardness has minor effects on the friction coefficient value [**20**].

Friction coefficient values can be summarized as ranging from 0.1 to 1.0 during the process of sliding without lubrication, 0.05 to 0.2 under lubrication, and 0.002 to 0.01 under hydrodynamic lubrication. Considering these large intervals of the friction coefficient value, it can be concluded that the presence of a lubricating film between contact surfaces is highly important [**27**].

The friction coefficient value can be increased using some technological treatments, but it must be limited for the purposes of assembling and disassembling press fit joint parts to avoid damage to the contact surfaces [**28**].

On the basis of the previous considerations, one can conclude that it is necessary to consider press fit joints as specific tribomechanical systems.

4 EXPERIMENTAL RESEARCH ON THE STRENGTH OF PRESS FIT JOINTS

The friction features of tribomechanical systems are estimated in laboratory research. Models for establishing friction features of sliding pairs take into account kinematic parameters and the way of forming contact between parts. Based on the above, a laboratory apparatus for estimating the parameters of friction and wear has to be made. Moreover, it is very important that experimental samples represent exploitation parts as much as possible.

In kinetic friction cases, experimental research can be done on a particular tribomechanical system to establish the friction coefficient value. The friction kinetic force can be measured without any difficulties, taking into account the fact that the measuring process is long in duration and that it is repeatable, so the mean value of the friction coefficient can be established with sufficient reliability.

In the case of static friction, however, the friction coefficient value should be established in a very short period of time at the moment of sliding start. The static friction force is a tangential resistant force that appears during the so-called 'relative boundary displacement'. The relative displacement develops into the visible, macrodisplacement. Thus, the static friction force can be measured only when the moment sliding begins, as after the start of sliding the kinetic friction force appears. This stochastic process is established under the conditions of simultaneous forming of new and breaking of old microconnections.

In the research that was carried out at the Faculty of Mechanical Engineering in Niš, Serbia, press fit joint strength was studied considering tribological parameters. All models, materials, and experimental conditions in the performed research were adapted and adjusted for the press fit joints of railway vehicle drive units.

The experiment was conducted on samples of press fit joints using different types of lubricants. The contact pressure and surface roughness of these samples were approximately identical. Press fit joint strength was estimated by measuring the longitudinal force up to the start of sliding.

Figure 8 shows the corresponding press fit joint with its inner (1) and outer part (2), as well as technical drawings.

This experimental research was performed on 30 press fit joint samples and about 100 pressing-on and pressing-out processes. Press fit joints were assembled as force fit joints, and a few days later, parts were disassembled by pressing-out. The friction force and sample movement were recorded during the assembly and disassembly processes.

The pressing-out processes of press fit joint assemblies lasted for about 40 s. However, sliding occurred in a very short period of time and it was possible to

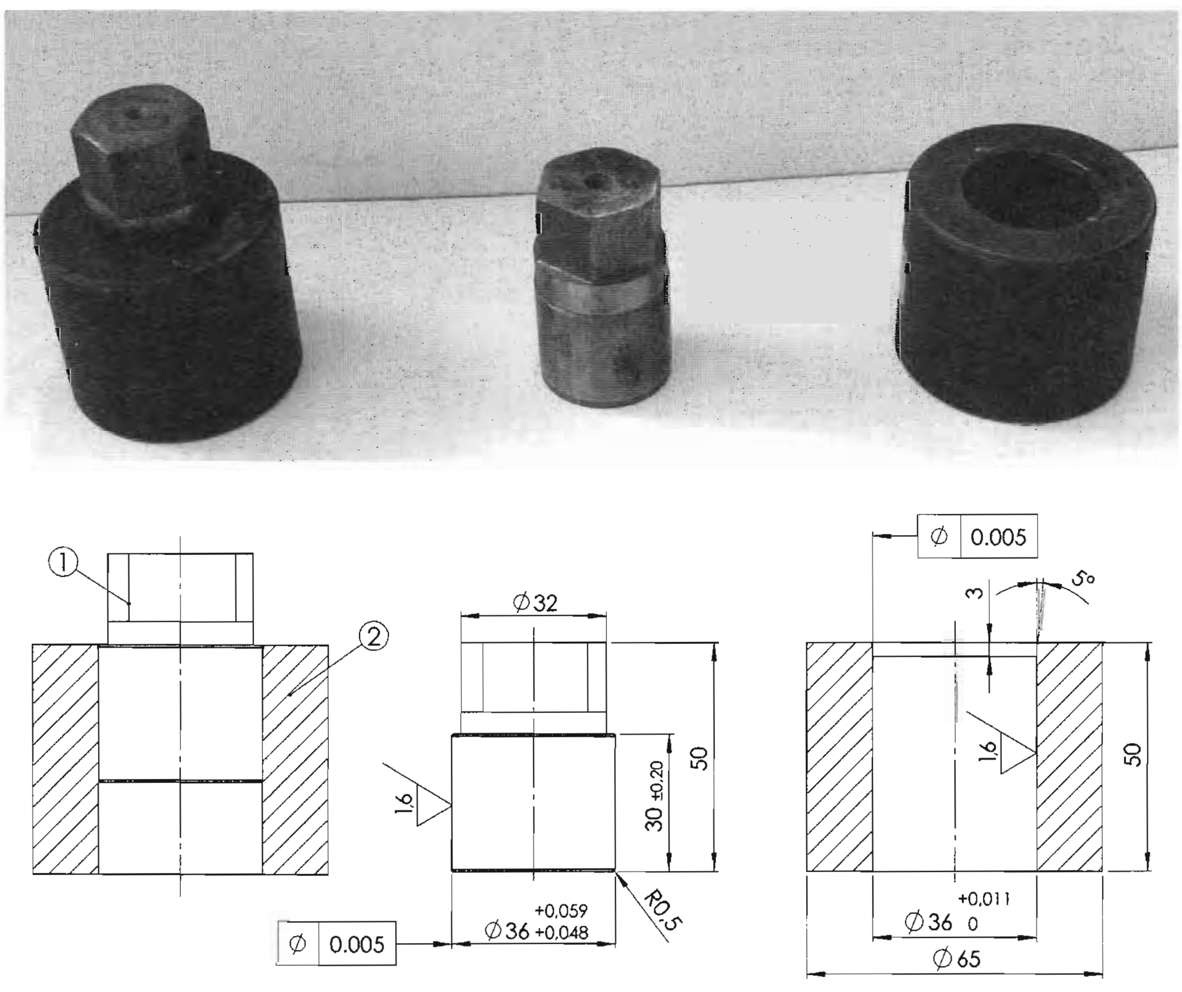


Fig. 8 The press fit joint sample

measure the static friction force only in that period. Figure 9 shows the typical force–movement diagram at the beginning of the disassembling process of the press fit joint (sample numbered A04).

During the experiment, the contact surfaces were lubricated with tallow, MoS_2, and LT311. The lubricants were chosen on the basis of Leaflet UIC 813 (Tallow, MoS_2) and Austrian and Serbian Railways (LT311) recommendations for the press fit joints of railway vehicle wheel sets. Some samples were assembled without lubrication, i.e. with dry surfaces. Table 3 shows typical graphs of pressing-on and pressing-out forces in the function of movement of parts with the use of different lubricants.

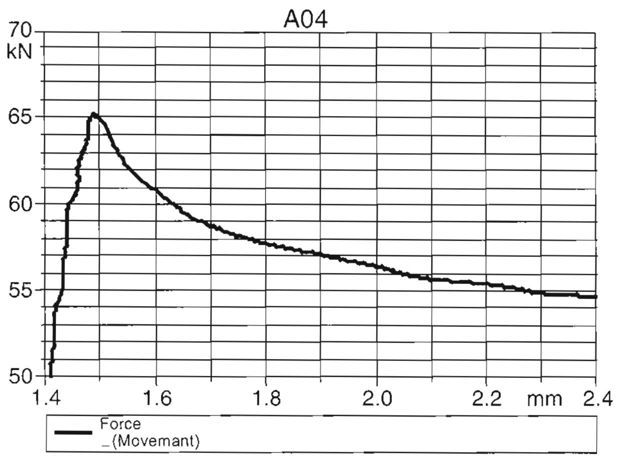
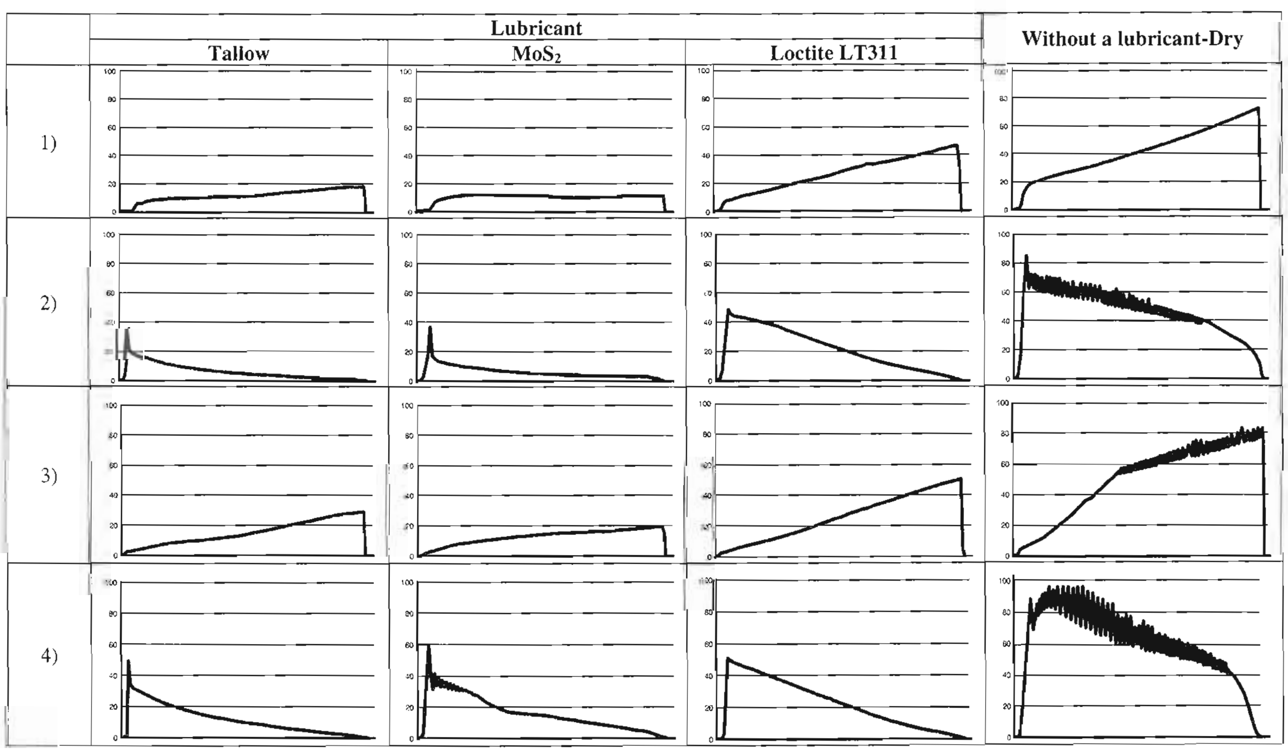


Fig. 9 Force–movement diagram at the beginning of the disassembling process

The examined press fit joints of railway vehicle drive units were assembled and disassembled a few times during their service. That was done within the framework of the technological process of maintenance/repair for the purposes of replacing damaged gears, bearings, and so forth. There are examples in the literature that point out that the strength of press fit joints decreases up to 25 percent after repeated assembly and disassembly processes [1]. In the performed experiment, results that departed from these examples were obtained. Specifically, the disassembly force presenting the strength of a press fit joint in the axial direction had a larger value after repeated assembly with respect to the disassembly force after the first assembly. This ratio was different for different lubricants. At repeatedly formed press fit joints lubricated with tallow, the disassembly forces were 11–39 percent larger than at the first assembly, with MoS_2 43–104 percent larger, LT311 3–9 percent

Table 3 Typical force-movement graphs in the processes of pressing-on and pressing-out with the use of different lubricants: (1) first pressing-on; (2) first pressing-out; (3) second pressing-on; and (4) second pressing-out



y-axis, pressing force (kN); x-axis, movement (mm) (the total movement $l = 30$ mm)

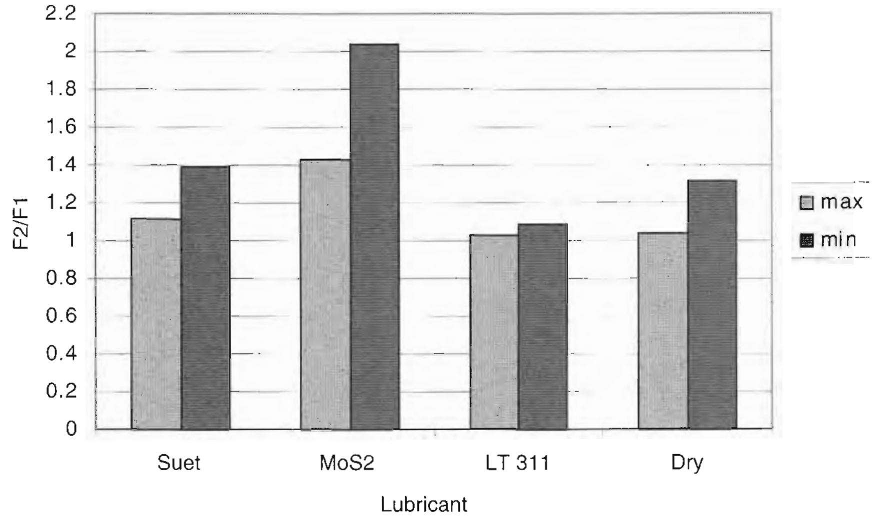


Fig. 10 Ratios of the press fit joint strength for repeated and first pressing-on processes

larger, and in cases with dry contact surfaces, the disassembling forces were 4 percent to 31 percent larger (Fig. 10).

According to formula (2), the static friction coefficient can be calculated as

$$\mu = \frac{F}{p\pi dl} \tag{9}$$

where F is the axial disassembling force that is measured at the moment sliding begins, and p the contact pressure that is calculated using formula (8) on the basis of the obtained tightening.

Table 4 gives the values of the obtained tightening for all examined press fit joints, used lubricants, measured surface roughness of shafts and flanges,

Table 4 Experimentally obtained values for the press fit joint samples

No.	Press fit joint no.	Lubricant	Tightening (μm) (calculated)	Roughness, R_a (μm) Shaft (measured)	Roughness, R_a (μm) Flange (measured)	Contact pressure (N/mm^2) (calculated)	Pressing-out force (kN) (measured)	Static friction coefficient (calculated)
1	10	Tallow	59	142	1.12	103.1195	38.3	**0.131**
2	14	Tallow	49	1.20	1.35	82.89995	21.3	**0.090**
3	25	Tallow	59	0.89	1.54	103.1195	22.4	**0.076**
4	03	Tallow	50	0.98	1.07	84.9219	11.41	**0.04**
5	17	Tallow	58	1.14	1.28	101.0975	35.62	**0.124**
6	28	Tallow	50	0.90	1.05	84.9219	27.46	**0.114**
7	20	Tallow	54	0.96	1.16	93.0097	24.4	**0.092**
8	06	Tallow	47	1.07	1.41	82.89995	17.1	**0.072**
9	11	MoS_2	57	1.10	1.31	99.07555	30.1	**0.107**
10	13	MoS_2	55	0.98	0.97	97.0536	47.2	**0.172**
11	26	MoS_2	45	1.11	1.09	80.878	30.02	**0.131**
12	04	MoS_2	50	1.04	1.56	84.9219	36.91	**0.153**
13	22	MoS_2	49	0.98	1.14	82.89995	13.6	**0.058**
14	18	MoS_2	58	1.28	1.35	101.0975	25.76	**0.090**
15	29	MoS_2	58	1.18	1.26	101.0975	38.05	**0.133**
16	07	MoS_2	48	0.95	1.34	80.878	16.99	**0.074**
17	15	LT 311	58	1.13	1.43	101.0975	36.02	**0.126**
18	27	LT 311	54	0.80	0.96	105.1414	32.08	**0.107**
19	12	LT 311	48	1.04	0.99	86.94385	28.07	**0.114**
20	05	LT 311	60	1.52	1.33	105.1414	33.65	**0.113**
21	19	LT 311	58	1.18	1.10	101.0975	39.44	**0.137**
22	23	LT 311	59	0.88	0.90	115.2512	44.63	**0.136**
23	30	LT 311	48	0.86	1.02	90.98775	21.23	**0.082**
24	08	LT 311	46	0.97	0.99	82.89995	25.66	**0.109**
25	02	Dry	59	1.03	0.94	115.2512	82.9	**0.254**
26	16	Dry	56	0.96	1.25	107.1634	55.82	**0.184**
27	01	Dry	42	1.00	1.09	76.8341	54.77	**0.252**
28	09	Dry	56	0.87	1.04	109.1853	70.46	**0.228**
29	21	Dry	48	1.22	1,18	80.878	59.28	**0.259**
30	24	Dry	59	1.46	1.52	103.1195	55.89	**0.191**

calculated contact pressure, and measured maximum forces in the first pressing-out process. The last column of Table 4 gives the static friction coefficient values calculated using formula (9).

The static friction coefficient values obtained in this experiment were:

(a) contact surfaces lubricated with tallow 0.047–0.131;
(b) contact surfaces lubricated with MoS_2 0.058–0.172;
(c) contact surfaces lubricated with LT311 0.082–0.138;
(d) contact surfaces without lubrications (dry surfaces) 0.184/0.259.

The samples lubricated with LT311 showed stable performance of press fit joints. There were no large variations between force values in the pressing-on and pressing-out processes. In that case, it was possible to very reliably predict the strength of the press fit joint (the static friction force in the pressing-out process) on the basis of the exerted pressing-on force.

5 DISCUSSION

The necessary prerequisite for the manufacture of press fit joints is the existence of tightening between assembled parts, as the contact pressure is a consequence of the elastic/plastic deformations of joined parts. The contact pressure is crucial for press fit joint strength, but has to be limited to values that do not cause damage to assembled parts. From the aspect of the stress of joined parts, the contact pressure has to be minimal.

If the dimensions and material properties of joint parts are given as a design prerequisite, it can be concluded that the contact pressure is directly in the function of tightening. Considering the recommendations of Leaflet UIC 813 for the choice of tightening, the contact pressure can be at most two times greater than the minimum value for the maximum value of tightening $p_{\max} \approx 2p_{\min}$.

The friction coefficient reached during parts assembly directly influences the value of the frictional force that resists the external load. When the various lubricants recommended by Leaflet UIC 813 and joining conditions are considered, the maximum value of the friction coefficient can be five times larger than the minimum value $\mu_{\max} \approx 5\,\mu_{\min}$.

Based on the previous analysis, the general conclusion is that the maximum strength of a press fit joint can be ten times greater than the minimum one

$$F_{\max} = \mu_{\max} \cdot p_{\max} \cdot A \approx 5\mu_{\min} \cdot 2p_{\min} \cdot A \approx 10F_{\min} \tag{10}$$

However, it is important to consider some supplements. Thus, in the press fit joint strength calculation, the value of the friction coefficient for the axial direction (2) should be minimal

$$F_{ag} = \pi dl p_{\min} \mu_{a\,\min} \tag{11}$$

and the friction coefficient for the tangential direction (3) should be minimal as well

$$M_{og} = \frac{\pi d^2 l}{2} p_{\min} \mu_{t\,\min} \tag{12}$$

Moreover, the maximum value of the friction coefficient has to be taken for the maximum pressing-on force

$$F_p = \pi dl p_{\max} \mu_{a\,\max} \tag{13}$$

Based on the recommendation to accept minimum and maximum values of the friction coefficient, the boundary values of the pressing-on forces established by the mentioned standards in section 2 can be extended. This was shown in the example of the examined wheel-axle assemblies of Serbian Railways electro locomotives. On the basis of the performed experiment for specific tribological conditions of locomotive wheel set press fit joints, the static friction coefficient value was estimated as 0.08 for the minimum value and 0.12 for the maximum value. With these minimum and maximum values of the friction coefficient and the contact pressure taken into account, the satisfactory pressing-on forces were estimated at the range 700–1650 kN. Figure 11 shows these new recommended boundaries together with the values of the calculated and obtained pressing-on forces from Table 1 and the boundary lines defined by Leaflet UIC 813.

According to Leaflet UIC 813, seven press fit joints were to be rejected (requiring the modification of the press fit joint parameters and new assembly processes), while six press fit joints demanded further testing regarding the counter pressure (Fig. 11). Although the geometrical and technological parameters were within the boundaries defined by Leaflet UIC 813, some press fit joints did not satisfy the recommendation regarding the maximum pressing-on force, and could have still been considered as reliable in the exploitation. It can be seen from Fig. 11 that most of the obtained pressing-on force values were in

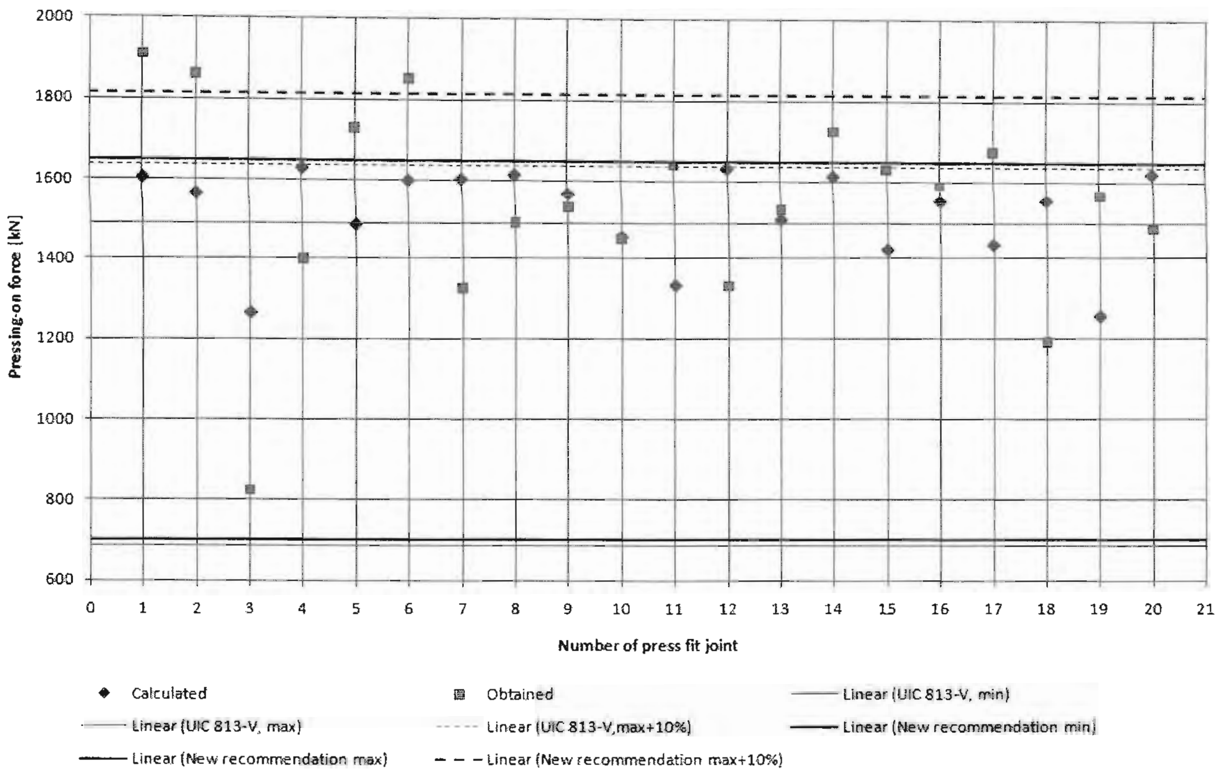


Fig. 11 Distribution of the calculated and obtained pressing-on forces

the boundaries of the new recommended range. According to the new recommendation, only three press fit joints were to be rejected, while three others required further testing in the same way as proposed by Leaflet UIC 813.

6 CONCLUSIONS

The performed research, experimental results, and acquired practical knowledge on press fit joints point to the following conclusions and recommendations.

1. Press fit joints must be treated as specific tribomechanical systems. It is very important to have knowledge of the tribological parameters that can influence the strength of a press fit joint and in this way ensures good load transmission.
2. The contact pressure due to tightening between assembled parts is indispensable for press fit joint strength, but its maximum value has to be limited to values that do not cause damage to assembled parts. From the aspect of the stress of the joined parts, the contact pressure has to be minimal.
3. The static friction coefficient is the most influential parameter of press fit joint strength.
4. If it is possible to control the friction coefficient value, then it is possible to obtain the required strength of the press fit joint in conditions close to the minimum contact pressure, which results in less prestressed press fit joint parts.
5. Press fit joint strength can be changed by varying the tribological parameters: the manner of assembly of press fit joint parts, surface roughness, contact surface machining, applied lubricant, surface hardness, and so forth.
6. The presence of a lubricant in the contact area and its characteristics highly define tribological conditions and thus have a significant influence on press fit joint strength.
7. It is recommended that design engineers use minimum and maximum values of the friction coefficient for the estimation of the strength of railway press fit joints. Minimum and maximum values of the friction coefficient can be taken from recommendations in the literature considering wide tribological conditions such as parts material, lubricant, and condition of the contact surfaces (shape, roughness, hardness, etc.).
8. For the significant press fit joints loaded with huge loads and/or produced in high series, it is better to perform an experiment to determine the friction coefficient value for those specific tribological conditions.

9. UIC recommendations should be altered to consider tribological conditions in the assembly process of wheel sets press fit joints, because some functional press fit joints are rejected in engineering practice according to current UIC recommendations.

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APPENDIX

Notation

a	coefficient of the type of the wheel and lubricant
A	contact surface area
d	joint diameter
d_i	diameter of the inner part
d_o	diameter of the outer part
D	nominal diameter of the hub seat
E_i	Young's modulus for the inner part
E_o	Young's modulus for the outer part
F	axial disassembling force
F_a	exploitation axial force load
F_{ag}	press fit joint strength for axial force load
F_{max}	maximal strength of the press fit joint
F_{min}	minimal strength of the press fit joint
F_N	normal force
F_p	maximal pressing-on force
$F\mu$	friction force
K_i	elasticity coefficient for the inner part
K_o	elasticity coefficient for the outer part
l	length of the press fit joint
M_o	exploitation torque load
M_{og}	press fit joint strength for torque load
p	contact pressure
p_{max}	maximal contact pressure
p_{min}	minimal contact pressure
P	fabrication tightening
P'	effective tightening of the joined parts
P_F	permissible value of the fitting-on pressure
R_a	surface roughness
R_{pi}	yield stress of the inner part
R_{po}	yield stress of the outer part
S_R	safety factor against plastic deformation
$S\mu$	safety factor against slipping
μ	friction coefficient
μ_a	friction coefficient in the axial direction
μ_k	sliding friction coefficient value
$\mu_{a\ max}$	maximal friction coefficient in the axial direction
$\mu_{a\ min}$	minimal friction coefficient in the axial direction
μ_{max}	maximal value of the frictional coefficient
μ_{min}	minimal value of the frictional coefficient
μ_s	static friction coefficient value
μ_t	friction coefficient in the tangential direction
$\mu_{t\ min}$	minimal friction coefficient in the tangential direction
σ_{ii}	stress in the inner part
σ_{io}	stress in the outer part
ψ_i	diameter ratio for the inner part
ψ_o	diameter ratio for the outer part

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