Calculation of lockbolt joints in mechanical engineering

Bemessung von Verbindungen mit Schließringbolzen im Maschinenbau

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Lockbolts become increasingly important for modern mechanical engineering. The advantageous notch effect in consequence of a softer groove geometry, the larger stress cross section for the same nominal diameter and a lower scattering of preload results in the potential for increasing the load capacity of lockbolt systems in mechanical joints in comparison to conventional bolted joints. Consequently costs can be reduced with structural adjustments in mechanical engineering. These include the reduction of the required nominal diameter and consequently the reduction of component dimensions. Furthermore savings in manufacturing and assembly as well as cost reduction for maintenance are possible. For the calculation of lockbolt systems in mechanical joints the Technical Bulletin DVS-EFB 3435–2 is available now for design and calculation engineers. The Technical Bulletin allows to calculate joints with lockbolt systems according to VDI 2230 – Part 1 [1, 2]. This paper presents joints with lockbolts in mechanical engineering. At first, the lockbolt technology and the assembly preload of the lockbolts will be introduced. After that the structural behavior of joints with lockbolts under static and alternating load will be presented. Finally, the modified calculation steps according to the VDI 2230 – Part 1 will be demonstrated [2].

Keywords: Lockbolts / mechanical joints / lockbolt joints / lockbolt assembly process / assembly preload / load bearing behavior

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1 Problem and approach

In Germany 68% of all detachable joints in the field of mechanical and vehicle engineering are dimensioned by the calculations according to VDI 2230 – Part 1 [2, 3]. For 40 years, VDI 2230 has been regarded worldwide as the standard guideline in calculations of bolted joints. It shows the general theoretical relations between loads, torques and deformations and derives the relevant equations. The stipulations of this standard apply to steel bolts with the dimensions from M4 to M39 with the strength grades of bolts of DIN EN ISO 898-1 in high-stressed and high-strength bolt joints [4, 5]. So far there are no appropriate fundamental equations for joints with lockbolts. In accordance with the characteristic of load bearing behavior and the dimensions of bolts in mechanical engineering the gap for the assessment of the joints with lockbolts under the influence of a concentric axial load \( F_A \) and / or a transverse load \( F_Q \) should be closed.

A bolted joint in mechanical and rail vehicle engineering is defined as follows: “A bolted joint has the task of joining components in such a way, that the appearance of slippage or widening the gap are prevented” [6]. As a calculation model for bolted joints the single-bolted joint and the joint diagram from will be used, Figure 1 [2]. Based on the previously known load conditions a loss of preload \( F_Z \) as a result of embedding during operation and a temperature difference from room temperature \( \Delta F_{\text{vh}} \) is taken into account. The operating transvers loads \( (F_Q) \) or in direction of the bolt axis \( (F_A) \) of the preloaded bolted joint will be applied by the preloaded components into the joint. The bolt is only stressed with an axial load by its preload and by an additional load \( (F_{S_A}) \), if \( F_A \) exists. The axial load \( (F_A > 0\), tensile load\) causes a loss of the assembly preload by the proportion of the relieving load from the clamped parts \( F_{PA} \) [7]. Furthermore, it is required that for the sealing function \( (F_{KP}) \), for preventing the one-sided opening at the interface \( (F_{KA}) \) or for transmitting a transverse load by friction grip \( (F_{KQ}) \), a minimum residual clamp load \( F_{KR} \) in the bolted joint has to be available. The required minimum clamp load \( F_{\text{Kerf}} \) results from the following equation:

\[
F_{\text{Kerf}} \geq \max(F_{KQ}; F_{KP} + F_{KA}) \tag{1}
\]

When determining the friction conditions and selecting the tightening processes, the maximum assembly preload \( F_{M_{\text{max}}} \) increased by the tightening factor \( \alpha_a \) is obtained by using the main dimensioning formula of the standard VDI 2230 – Part 1:
Because of the suitability for high strength pre-loading, the mode of action of lockbolts is quite similar to bolts in mechanical engineering. According to this circumstance the lockbolt technology shows a pleasant potential as an alternative fastener for mechanical joining due to its advantageous characteristics compared to bolts. In course of several completed public research activities at the Fraunhofer Research Institute for Large Structures in Production Engineering IGP the input variables for the calculation of concentric loaded joints with lockbolts have been developed and prepared for the calculation steps by the Technical Bulletin DVS-EBF 3435-2 \cite{1, 8, 9}.

### 2.1 Embodiments and fields of application

Lockbolt systems are divided and available in two different embodiments and also in different tempering conditions, Figure 2.

According to the Technical Bulletin DVS-EFB 3435-2 the following embodiments are dealt with:

- **Type A:** Lockbolt systems (FK 8.8 and 10.9) with plane-parallel locking-groove geometry
- **Type B:** Lockbolt systems (FK 10.9) with helical locking-groove geometry and with or without a breaking groove
- **Type C:** Lockbolt systems (FK 10.9) with helical locking-groove geometry without a breaking groove and with reduced pintail

Lockbolts made of steel are available in the strength grades 5.8, 8.8 and 10.9 according to DIN EN ISO 898-1 \cite{5}. Additionally, lockbolts and the related collar can also be offered as aluminium and corrosion resistant material. Currently lockbolts are available in metric and imperial dimensions. The

\[
F_{\text{Mmax}} = \alpha_A \cdot F_{\text{Mmin}} = \\
\alpha_A \cdot [F_{\text{Kerf}} + (1 - \Phi) \cdot F_A + F_Z + \Delta F_{\text{Vth}}] \tag{2}
\]

**Figure 1.** Main dimensions in the joint diagram (without thermal additional load $\Delta F_{\text{Vth}}$) according to VDI 2230 – Part 1 \cite{2}.

**Bild 1.** Hauptdimensionsgrößen im Verspannungsdiagramm (ohne thermische Zusatzkraft $\Delta F_{\text{Vth}}$) in Anlehnung an VDI 2230 – Blatt 1 \cite{2}.
nominal diameters range from 4.8 mm up to 36 mm.

The manufacturing of lockbolts and collars can be realized by machining as well as by hot or cold forming. Subsequent heat treatment provides with the required strengths of lockbolt systems. In the final production step the coatings and a wax film which serve on the one hand to protect against corrosion and on the other hand to ensure a defined friction during assembly were applied. In the context of this article the manufacturing processes will not be discussed in more detail.

A lockbolt system has several advantages in its application. The plastic deformation of the collar into the locking-grooves of the lockbolt creates a form fitting bolted joint that prevents the collar from automatically loosening. Another further advantage is the high stability of the assembly process in comparison to the torque-controlled bolt assembly process. In general, the scatter is less than 5 % during the application of assembly preload [11]. A disadvantage in joints with lockbolt systems is that currently only a few system manufacturers for strength grades of 8.8 and 10.9 (Arconic Fastening Systems and Rings Ltd., Avdel Deutschland GmbH, Gebr. Titgemeyer GmbH & Co. KG) are represented on the market, Figure 3.

**2.2 Lockbolt assembly process**

Joining with lockbolts is to be assigned to join by forming. The assembly process is divided in four steps. The following procedure is an example with a lockbolt system without a pintail, Figure 4.

1. The lockbolt gets inserted from one side into the hole of the component. On the opposite side the collar gets plugged or screwed on the lockbolt.

2. The installation tool which implements the tensile load pneumatically or hydraulically depending on the diameter of the lockbolt is applied on the side of the collar. When the installation tool is triggered, the assembly process starts in which the clamping jaws of the installation tool grasps the pintail of the lockbolt and create a tensile load on the lockbolt. During this step the nose assembly (load section with clamping jaws) moves with the deformation sleeve over the collar. As the tensile load increases, the components are pulled together and the lockbolt is pre-stretched by tightening the lockbolt pintail and counter-supporting on the collar.

3. In the further forming process the deformation sleeve continues moving in the direction of the collar flange and effects the final elongation of the lockbolt with simultaneous forming of the collar material into the lockbolt grooves. The collar is installed in a form and force fitting manner on the grooved lockbolt and the preload is applied. By forming the collar into the lockbolt grooves, recovery losses are scarcely observed, as they occur, for example, in pulling preloaded processes for bolted joints. The assembly process is completely free of torsion.

4. The removal of the installation tool marks the completion of the assembly process. Simultaneously, the lockbolt system offers the opportunity to inspect the successful assembly process (e.g. visual or gauge inspection).

The interaction between the individual parts (lockbolt, collar and installation tool) is significant. For the quality of the joint the influence of the ge-
The geometry part of the installation tool (called nose assembly) is elementary. The nose assembly forms the collar during the installation process and thus ensures the material-flow of the collar into the grooves of the lockbolt. The individual parts like lockbolt, collar and nose assembly influence each other. For this reason it is important to coordinate these parts for the installation of a perfect joint according to the quality aspects [12].

2.3 Securing effect with transverse loading

In frictionally engaged load-transmitting bolted joints, the maintenance of the clamp load $F_{Kerf}$ is of decisive importance for the required load-bearing effect. One of the causes of the loss of preload in alternating transverse loaded joints is the automatic loosening effect. This is the subject of numerous scientific investigations [13–16]. Based on these observations, a vibration test bench was developed.
with which the automatic loosening of bolted joints under alternating transverse load can be investigated, Figure 5 [13].

The vibration test bench and the test sequence are specified in the standard DIN 65151 [17]. The securing behavior of a joint is carried out appropriately on the basis of the preload referred to. In order to demonstrate and compare the securing effect of bolts and lockbolts, there is an example of experimentally determined relationships between the preload and the number of load cycles for the nominal diameter \( d = 16 \text{ mm} \) of respective fasteners, Figure 5. The curves show the loosening behavior of an unsecured mechanical engineering bolt ISO 4017-M16-8.8 with nut ISO 4032-M16 and washer ISO 7089, a mechanical engineering bolt joint secured with wedge-lock washers and a lockbolt system under comparable boundary conditions on the vibration test bench [18–20]. The tested bolts were degreased at the head bearing area and in the thread and then defined lubricated with Microgleit DF977. The evaluation of the experimental determined correlations between the related preload and the number of load cycles was carried out after 2,000 load cycles had been achieved in accordance with the standard EN DIN 25201-4 applied in rail vehicle construction, because the main loss of preload of the unsecured bolt joint occurs within this number of load cycles [21].

There is a significant advantage of the lockbolt in direct comparison to the bolted joint. The effect of automatic unsecuring is not effective with lockbolts. A lockbolt is a vibration proofed fastener. Accordingly, the lockbolt system does not require an additional unscrewing safety device. A facilitation is given with regard to feasibility. This is accompanied by an increase in economic efficiency due to the elimination of acquisition costs and installation times for separate securing elements, Figure 5.

3 Assembly preload for lockbolts

The lockbolt assembly process is basically different to the assembly process of classical mechanical engineering bolts. It is defined as a volume-controlled, forming and torsion-free joining process, also operator-independent and it is not going to be influenced by the installation procedure or the local friction conditions in the thread and in the head bearing area like for bolts. For bolts in mechanical engineering a tightening factor \( \alpha_A \) is determined according to the standard VDI 2230 – Part 1 depending on tightening procedures and local friction conditions [2]. The scattering for the assembly preload occurs from \( \pm 5 \% \) to \( 60 \% \). The lockbolt shows a much smaller scattering of the assembly preload in

![Figure 5. Comparison of the securing effect of joints with lockbolts and bolts, section vibration test bench (downright) [16, 22].](image-url)

**Bild 5. Vergleich der Sicherungswirkung von Schließringbolzen und Schraubenverbindungen, Ausschnitt Vibrationsprüfstand (unten rechts) [16, 22].**
Comparison to bolts. The range is ± 5% [22]. Out of this a tightening factor of $\alpha_A = 1.05$ has been defined in the Technical Bulletin DVS-EFB 3435-2 [1]. The preload is reached by lockbolts and collars with strength grades of 8.8 and 10.9.

During tightening the bolt with the tightening moment $M_A$ the bolt is stressed by a tensile load, which means it gets stretched by the value $f_{SM}$. Simultaneously, the components get compressed by the value $f_{PM}$. The result is the assembly preload $F_M$. The principle of equality of load applies. This relationship can be transferred to lockbolt joints. The joint diagram shows the load-deformation-ratio for the assembly state. The tightening process for lockbolt systems is presented by the assembly preload-time-diagram, Figure 6.

The assembly process of lockbolts can be described by the following steps:
1. The assembly process begins with the arrangement of the parts by inserting the lockbolt in the clearance hole. The collar gets plugged or screwed on the lockbolt on the opposite side.
2. The preloading starts with the release of the assembly process by the installation tool.
3. The assembly preload increases till reaching the maximum value $F_{\text{max}}$. This value marks the forming of the collar material into the lockbolt grooves.
4. The load after the removal of the installation tool is called the initiated lockbolt assembly preload $F_{M,SRB}$.

In addition to the assembly process of the lockbolt, a torque controlled tightening process can also be compared with a bolt, Figure 6. If, for example, ten preload-time-curves are compared to lockbolt systems with a nominal diameter of $d = 16$ mm, the extremely low scatter for the achieved assembly pretensioning load can be seen, Figure 7.

### 4 Load bearing behavior of lockbolt joints

The following section deals with the axial loaded lockbolt joints. In this context the structural behavior under static and alternating (dynamic) axial load

![Figure 6. Joint diagram for the assembly states of lockbolts (left side), characterization (description) of the tightening process of lockbolts and bolts (right side) [23].](Image)

**Bild 6. Verspannungsschaubild für den Montagezustand bei SRB (links), Charakterisierung des Anziehvorgangs bei SRB und Schraube (rechts) [23].**

![Figure 7. Preload-time-curve during the assembly process of a lockbolt system.](Image)

**Bild 7. Vorspannkraft-Zeit-Verhalten bei der Schließringbolzenmontage.**
will be presented. The aim of this illustration is the explanation of the general design principle of lockbolt systems.

4.1 Load bearing behavior under static load

The calculation of mechanical engineering bolts under static axial load is based on the general design principle of bolts. During a static overload the mechanical engineering bolt fails in the free loaded thread [7]. The conditions for this kind of failure are an effective length of thread engagement or rather a normal nut height \( ( = 1.0 \ d) \) as well as a matched strength of bolt and nut. Bolts have a high deformation capacity because of their sufficient material toughness. After reaching the 0.2 % proof stress \( R_{p0.2} \) of the bolt there is a typical high elastic plastic elongation till reaching the elongation at break and the failure of the bolt. The static bolt failure appears with an announcement e.g. due to noises generated by loosened components under alternating (dynamic) load [7].

The lockbolt behaves completely different under static tensile load. The failure mode for axial loaded lockbolt systems is characterized by stripping off the collar. The load-stroke-diagram during a static concentric axial load of a lockbolt system with a bolt diameter of \( d = 20 \text{ mm} \) in stroke controlled mode of the testing machine can be characterize. In the first part of the diagram the test load increases linear. By reaching the maximum load it comes to an audible failure (stripping off the collar) and thereby to a sudden loss of load capacity without an announcement, Figure 8.

After reaching the maximum load the test load decreases rapidly. Afterwards the test load increases again. The remaining collar grooves are then sheared off above the lockbolt grooves until the collar is completely stripped off. The stripping off the collar is the general design principle of lockbolt systems. The reason for stripping off the collar material is its lower strength compared to the lockbolt. The permitted axial substitution load \( F_{A,zul} \) is related with reaching the 0.2 % proof stress of the stress cross section \( A_{SRB} \) according to [22, 23]. Due to the fact that the stripping off the collar happens without an announcement an additional reduction of the load capacity about 10 % (safety margin \( S_F = 1.1 \) ) is applied in the Technical Bulletin DVS-EFB 3435-2 [1].

The comparison of the structural behavior of lockbolt joints during concentric and eccentric static axial load shows that the load-stroke-diagrams are quite similar. Even in the tensile test on eccentric loaded lockbolt systems the collar stripes also off while reaching the maximum load. Afterwards this leads to a sudden loss of the test load. However, the maximum load and thus the static load capacity for eccentric axial loaded lockbolt systems is almost about half less than the concentric loaded lockbolt systems, which can be explained by the additional bending stress.

**Figure 8.** Load-stroke-diagram and failure mode of lockbolt systems after static concentric (left side) and eccentric axial load (right side) [23, 28].

**Bild 8.** Kraft-Weg-Verlauf und Versagensbild von Schließbringbolzensystemen bei zügiger Längszugbelastung zentrisch (links), exzentrisch (rechts) [23,28].
4.2 Load bearing behavior under alternating (dynamic) load

In alternating (dynamic) loaded bolt joints in mechanical engineering the verification of fatigue strength or endurance limit for the bolt considering the influence of size of the bolt based on the nominal stress concept is necessary. Influence factors on the stress amplitude of the endurance limit $\sigma_A$ like the way of manufacturing the groove (bolts rolled before heat treatment or bolts rolled after heat treatment), the type of the groove (metric standard thread or fine thread) as well as the type of corrosion protection (e.g. hot-dip galvanized) are considered in the guideline VDI 2230 – Part 1 [2]. In alternating (dynamic) loaded lockbolt joints the verification must be carried out in accordance with the guideline VDI 2230 – Part 1 with the stress amplitude of the endurance limit of $\sigma_A = 33.5$ N/mm$^2$ (nominal-$\varnothing \leq 20$ mm) and $\sigma_A = 31.5$ N/mm$^2$ (nominal-$\varnothing > 20$ mm) [1, 2]. The determination of the endurance limit was carried out in accordance with the standard DIN 969 : 1997-12 [24]. Basically this standard defines the conditions for the procedure of fatigue tests under axial load on threaded fasteners. In addition requirements are also defined for the

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**Figure 9.** Fatigue resistance of lockbolts with nominal-$\varnothing \leq 20$ mm (left side), nominal-$\varnothing > 20$ mm (right side).

**Bild 9.** Schwingfestigkeit Schließringbolzen Nenn-$\varnothing \leq 20$ mm (links), Nenn-$\varnothing > 20$ mm (rechts).

**Figure 10.** Notch factor curve of a bolt-nut joint and a lockbolt joint [7].

**Bild 10.** Kerbfaktorverlauf an einer Schraube-Mutter-Verbindung und an einer Schließringbolzen-Verbindung [7].
test setup as well as the testing procedure. These standardized tests are tests under continuous alternating load on fasteners providing that the type of preloading of the fasteners causes the axial, sinusoidal operating load $F_A$ generated by the testing machine to act completely on the fastener [7]. The tests were only carried out in accordance to DIN 969:1997-12 to determine the endurance limit of the lockbolts [24]. The test procedure was developed and enables a fatigue test for lockbolt systems without preload [25]. To determine the fatigue strength the respective lockbolts were mounted into the test inserts with additional spacer plates in the tensile device of the test machine. In the course of a static preload above the gap, the spacer plates can be removed and the lockbolts can be tested free of preload with a constant mean load according to the preload level $F_{V,SRB}$ until fracture in the first load-bearing groove or until the number of alternating cycles during continuous loading $N_D = 2 \cdot 10^6$ (= run out) is reached. Selected results of fatigue tests on lockbolts type C for the nominal diameter of $d = 12$ mm; 20 mm; 25.4 mm and 36 mm are presented, Figure 9.

The fracture in the first load-bearing groove can be uniformly determined as the typical failure type. The slope of $m = 3$ for the fatigue strength ranges is proved to be suitable.

The stress distribution at a bolt-nut-joint and a lockbolt-joint can be compared. Due to their function-related design with the head, the shank and the thread, bolts have critical notches which result in high notch stresses $\sigma_{\text{max}}$. The notch factor curve shows that the highest notch stress occurs in the first load-bearing thread, resulting in the highest notch factor $\alpha_K$ at a bolt-nut joint, Figure 10 [7]. The notch factor $\alpha_K$ is the ratio of notch stress $\sigma_{\text{max}}$ and nominal stress $\sigma_{\text{Nenn}}$ and describes local stress concentrations. Measures are listed to improve the endurance limit of bolts in [7]. With the aid of these measures, the fatigue limit at the critical notches mentioned above has to be increased. With its corresponding groove geometry and the smaller transition radius from the head to the shank, the lockbolt fulfills these measures to increase the fatigue limit in its design. The advantageous groove geometry minimizes the local stress peaks which is advantageous for lockbolt joints compared to bolt-nut joints. This is reflected in the notch factor curve, Figure 10.

5 Calculation of lockbolt joints

The design and calculation of joints with lockbolt systems depends on the respective field of application. The Technical Bulletin DVS-EBF 3435-2 provides design basics for mechanical engineering as well as for steel constructions in accordance with the series of standards EN 1993 (Eurocode 3) [1].
The Technical Bulletin DVS-EFB 3435-2 is developed within the scope of the DVS-EFB working group “Mechanical joining AGMF3/V10.3 Blind rivets and lockbolts” and is taken into account a large number of research projects between 2008 and 2015 [8–9, 26–27]. In mechanical engineering, especially in vehicle and rail construction, the design rules are based on the standard VDI 2230 – Part 1 for the systematic calculation of highly stressed bolted joints under concentric clamped and concentric axial load [2]. Examples of calculations for the design and calculation engineers are provided in the appendix of the Technical Bulletin [2].

### 5.1 Specification for the application of the regulations according to VDI 2230 – Part 1

For joints with mechanical engineering bolts, a general distinction between single-bolt joints subjected to axial and / or transverse load is made. This is transferable to joints with lockbolts. Lockbolt joints can be calculated for different load types according to DIN 25201-2 and VDI 2230 – Part 1, Figure 11 [1, 2, 6].

The rules for the design of joints with lockbolts in mechanical engineering apply basically for high stressed cylindrical single bolt joints according to VDI 2230 – Part 1 [2]. The applicability of the design rules requires a realistic transmission of static and alternating operating loads (axial and transverse loads) and is valid for high-strength preloaded lockbolts made of steel with nominal diameters of $d = 5$ mm to $d = 36$ mm or rather of $d = 3/16"$ to $d = 1"$ in combination with the strength grades 8.8 and 10.9. The corresponding minimum values for 0.2 % proof stress $R_{p0.2}$ and the tensile strength $R_m$ can be summarized, Table 1.

Joints with lockbolts must be converted into predictable mechanical models. The loads and deformations have to be described by simple mechanical spring models according to VDI 2230 – Part 1 [2]. The lockbolt is regarded as a tension spring with the elastic resilience $\delta_{SRB}$ and the clamped components as a compressing spring with the elastic plate resilience $\delta_P$. To describe the axial resilience of a lockbolt it is assumed that the lockbolt is composed of individual elements characterized by cylindrical bodies, Figure 12.

The elastic linear deformation $f_i$ of the separate elements under an impacting concentric axial load $F_A$ results as follows:

$$f_i = \frac{l_i \cdot F}{E_{SRB} \cdot A_i}$$

### Table 1. Nominal values of the yield strength, tensile strength, shear strength ratio for lockbolts in dependence of the strength grade [1].

| Strength grade | 5.8 | 8.8 | 10.9 | A2-50 | A4-50 |
|----------------|-----|-----|------|-------|-------|
| $R_{p0.2}$ [N/mm$^2$] | 400 | 640 | 900   | 190   | 190   |
| $R_m$ [N/mm$^2$] | 500 | 800 | 1000  | 500   | 500   |
| Utilization factor $v$ of the yield point stress during tightening [-] | $-^1$ | 0.775$^2$ | 0.688$^2$ | $-^1$ | $-^1$ |
| Shearing strength ratio $\tau_p/R_m$ [-] | 0.55 | 0.55 | 0.55 | 0.50 | 0.50 |

$^1$ not suitable for preloading; $^2$ a higher degree of utilization factor shall be demonstrated by experimental investigations.

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**Figure 12.** Division of a lockbolt into individual cylindrical bodies according to VDI 2230 Part 1 [1].

**Bild 12.** Aufteilung eines Schließringbolzens in zylindrische Körper in Anlehnung an VDI 2230 Blatt 1 [1].
| Requirements Calculation step | Determination of Symbol |
|-------------------------------|-------------------------|
| R0                            | Minimum clamp load (considering the following requirement) |
|                               | Frictional locking for transmission of a transverse load $F_Q$: |
|                               | $F_{KQ} = \frac{F_{Q_{\text{max}}}}{\eta \cdot \mu_{\text{trans}}}$ |
|                               | Sealing against a medium: |
|                               | $F_{K_P} = A_D \cdot p_{i,\text{max}}$ |
|                               | Claim: |
|                               | $F_{\text{Kerf}} \geq \max(F_{KQ}, F_{K_P})$ |
| R1                            | Tightening factor $\alpha_A = 1.05$ |
| R2                            | Selection of the bolt diameter (preliminary sizing) |
|                               | Maximum assembly preload: |
|                               | $F_{\text{Mmax}} = \alpha_A \cdot F_{\text{Mmin}} = \alpha_A \cdot [F_{\text{Kerf}} + (1 - \phi) \cdot F_A + F_Z + \Delta F_{\text{Vth}}]$ |
|                               | Minimum assembly preload: |
|                               | $F_{\text{Mmin}} = A_{S,\text{vorh}} \cdot \nu \cdot R_{P0.2}$ |
|                               | Necessary preloadable stress cross section: |
|                               | $A_{S,\text{erf}} = \alpha_A \cdot \frac{F_{\text{Kerf}} + (1 - \phi) \cdot F_A + F_Z + \Delta F_{\text{Vth}}}{R_{P0.2}}$ |
|                               | Appreciated load factor $\phi = 0.2$; loss of preload by $F_Z$ and thermal stress $\Delta F_{\text{Vth}}$ will be appreciated at first |

Joint diagram

| R3                            | Distribution of working load/load factor |
|-------------------------------| Load factor during concentric preload and load: |
|                               | $\phi_n = n \cdot \frac{\delta_P}{\delta_{\text{SRB}} + \delta_P}$ |
|                               | Load factor as quotient of additional lockbolt load and axial working load: |
|                               | $\phi = \frac{F_{\text{PA}}}{F_n}$ |
|                               | Additional plate load: |
|                               | $F_{\text{PA}} = (1 - \phi) \cdot F_A$ |
|                               | For the determination of $\phi$ the elastic resilience of the lockbolt $\delta_{\text{SRB}}$ as well as the elastic resilience of the preloaded components $\delta_P$ are necessary. |
Table 2. continued

| Requirements Calculation step | Determination of Symbol |
|-------------------------------|-------------------------|
| R4 Preload changes            |                         |
| *Loss of preload as a result of embedding:* |                         |
| \( F_Z = \frac{f_z}{(\sigma_{u} + \delta_Y)} \) | \( F Z, \Delta F_{Vth} \) |
| *Loss of preload by different coefficients of thermal expansion:* |                         |
| \( \Delta F_{Vth} = \frac{\left(\sigma_{u} \Delta T_{z} - \alpha \sigma_{u} \Delta T_{e} + \alpha \sigma_{u} \Delta T_{e}\right)}{\sigma_{u} + \delta_Y} \) |                         |
| R5 Minimum assembly preload   |                         |
| \( F_{M_{min}} = F_{Kerf} + (1 - \Phi_n) \cdot F_{A_{max}} + F_Z + \Delta F_{Vth} \) | \( F_{M_{min}} \) |
| R6 Maximum assembly preload   |                         |
| *Claim:* \( F_{M_{zul}} \geq F_{M_{max}} \) with \( F_{M_{zul}} = A_{S,vcoh} \cdot \nu \cdot R_{p0.2} \) | \( F_{M_{max}} \) |

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| R7 Examination of the selected preload (assembly stress) | \( F_{M_{zul}} \) |
| R8 Working stress | \( \sigma_{z_{max}}, S_z \) |
| *Maximum load of lockbolt in working stress:* |                         |
| \( F_{S_{max}} = F_{M_{zul}} + \Phi_n \cdot F_{A_{max}} - \Delta F_{Vth} \) |                         |
| *Maximum working load in the stress cross section of bolt \( A_S \):* |                         |
| \( \sigma_{z_{max}} = \frac{F_{S_{max}}}{A_S} = \frac{1}{\alpha} \cdot \left( F_{M_{zul}} + F_{A_{max}} - \Delta F_{Vth} \right) \) |                         |
| Verification against the failure “slipping off the collar”: |                         |
| \( \sigma_{z_{max}} < \frac{R_{p0.2}}{1.1} \) |                         |
| Safety margin against “slipping off the collar”: |                         |
| \( S_F = \frac{R_{p0.2}}{1.1 \cdot \sigma_{z_{max}}} \geq 1.0 \) |                         |
| The safety has to be defined by the design and calculation engineers. |                         |
| R9 Alternating stress | \( \sigma_{a}, \sigma_{a}, \sigma_{A}, S_D \) |
| *Alternating stress as alternating tension for concentric stress:* |                         |
| \( \sigma_a = \frac{F_{a_{zul}}}{A_S} = \frac{F_{a_{zul}} - F_{a_{x_{zul}}}}{2 \cdot A_S} = \frac{F_{max} - F_{min}}{2 \cdot A_S} \) |                         |
| Stress amplitude of the endurance limit: |                         |
| \( \sigma_A = 31.5 \frac{N}{mm^2} \) (nominal \( \theta \leq 36 \, mm \), type B) |                         |
Table 2. continued

| Requirements Calculation step | Determination of Symbol for the numbers of alternating cycles during continuous loading \(N_D \geq 2 \cdot 10^6\) Verification of the dynamic strength within the fatigue strength range \((N_Z > 10^4)\) and slope of the fatigue strength curve \(m = 3\) [1]: |
|-----------------------------|---------------------------------------------------------------|
| \(\sigma_{AZ} = \sigma_A \cdot \sqrt{\left(\frac{N_D}{N_Z}\right)}\) Verification of the dynamic strength: \(\sigma_a \leq \sigma_A\) Safety margin against fatigue failure: \(\frac{S_D \sigma_a}{\sigma_A} \leq 1.0\) with \(S_D = 1.2\) |

R10 Surface pressure

| Surface pressure in assembled state: |
|--------------------------------------|
| \(P_{M\text{max}} = \frac{F_{\text{max}}}{A_{\text{max}}}\) with \(F_{\text{Amax}} = 0.95 \cdot R_{P0.2} \cdot A_s\) |

Surface pressure in working state:

| \(P_{B\text{max}} = \frac{(F_{\text{Vmax}} + F_{\text{max}} - \Delta F_{\text{Vth}})}{A_{\text{max}}}\) Verification against exceeding of the limiting surface pressure \(p_G\) (Tab. A9[2]): |
|--------------------------------|
| \(P_{M\text{max}} \leq p_G\) and \(P_{B\text{max}} \leq p_G\) |

R11 Slipping

| Minimum residual clamp load: |
|-----------------------------|
| \(F_{K\text{Rmin}} = \frac{F_{\text{max}}}{A_s} - (1 - \Phi_n) \cdot F_{\text{Amax}} - F_Z - \Delta F_{\text{Vth}}\) |

Required clamp load:

| \(F_{K\text{Qerf}} = \frac{F_{\text{QzulS}}}{r_{\text{PQUL}}^{\text{r}}r_{\text{PQUL}}^{\text{r}}\text{PQUL}}\) Verification against slipping: \(F_{K\text{Rmin}} > F_{K\text{Qerf}}\) Safety margin against slipping: \(S_G = \frac{F_{K\text{Rmin}}}{F_{K\text{Qerf}}} > 1.0\) with \(S_G = 1.20\) (static load) \(S_G = 1.80\) (alternating load) |

Shearing

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The elastic resilience $\delta_i$ of the cylindrical separate elements of a lockbolt results further to:

$$\delta_i = \frac{f_i}{F} = \frac{l_i}{E_{SRB} \cdot A_i}$$  (4)

If it is assumed that the separate elements like the head, shank, free loaded grooves as well as

| Requirements Calculation step | Determination of Symbol |
|-----------------------------|-------------------------|
| Permissible shearing capacity of the lockbolt: | $F_{QzulS} = \tau_a \cdot A_i = \left( \frac{d_i}{2} \right) \cdot R_m \cdot \frac{\pi}{2} \cdot d_i^2$ |
| Safety margin against shearing off the lockbolt: | $S_A = \frac{F_{QzulS}}{F_{Qmax}} > 1.25$ |
| Permissible bolt bearing capacity of the component: | $F_{QzulL} = \frac{h \cdot R_{pm}}{S_L}$ |
| Safety margin against bearing pressure of the bolt: | $S_L = \frac{F_{QzulL}}{F_{Qmax}} > 1.0$ |
| Increased permissible bearing capacity of the bolt for components made of steel and aluminum: | $F_{QzulL} = \frac{k_1 \cdot n_0 \cdot d_i \cdot R_{pm}}{S_L}$ with $S_L = 1.25$ |
| Alternating stress as an alternating shearing stress: | $\tau_a = \frac{\tau_{Qmax} - \tau_{Qmin}}{2 \cdot A_i}$ |
| Stress amplitude of the endurance limit ($N_t > 10^6$) under shear stress: | $\tau_A = 2.3 \frac{N}{mm^2}$ |
| Verification of dynamic strength within the fatigue strength range and slope of fatigue strength curve $m = 5$: | $\tau_{AZ} = \tau_A \cdot \sqrt{\frac{N_t}{N_t}}$ |
| Verification of the dynamic strength: | $\tau_a \leq \tau_A$ |
| Safety margin against fatigue failure: | $\frac{S_D}{\tau_a} \leq 1.0$ with $S_D = 1.2$ |
formed-in groove- and collar areas are arranged in series the total resilience of the lockbolt can be added up:

\[ \delta_{SRB} = \delta_{SK} + \delta_{Sch} + \delta_{Ril} + \delta_{SR} \] (5)

5.2 Calculation steps according to VDI 2230 – Part 1

The proof of the assembly and working stress of high strength preloaded lockbolt joints in mechanical engineering must be carried out using the calculation steps (R0 to R11). In regard to the characteristics of lockbolts the calculation steps existing for bolt joints had to be modified. For example, the tightening behavior (preload \( F_{V,SRB} \) and tightening factor \( \alpha_t \)), the load capacity and the load capacity during alternating stress are considered, Table 2.

6 Conclusion

The mode of action and applicability of the lockbolt is comparable to that of mechanical engineering bolts. Due to its strength properties, the lockbolt is suitable for high-strength preloading and therefore exhibits closely related load-bearing behavior with bolted joints. It is obvious in general to apply the model for single-bolt joints for mechanical engineering bolts also to the model of single-bolt joints for lockbolts in association with the joint diagram in the assembled and working states for the calculation algorithm according to VDI 2230 – Part 1 and the calculation steps connected. Here, advantageous properties of the lockbolt technology come into effect [2]. In this context, the operator-independent, torque free assembly process with low scatterings in the assembly preload (tightening factor \( \alpha_t = 1.05 \)) and the alternating stress resistance \( (\sigma_A = 33.5 \text{ N/mm}^2 \text{ for nominal diameter of } \leq 20 \text{ mm and } \sigma_A = 31.5 \text{ N/mm}^2 \text{ for nominal diameter of } > 20 \text{ mm}) \) should be mentioned. In addition to the advantages, the disadvantages of the lockbolt technology must also be pointed out. The purchase of special assembly tools is costly and uneconomical, especially for small-sized companies. However, the biggest disadvantage for the dimensioning of lockbolt joints in mechanical engineering could be solved with the publication of the Technical Bulletin DVS-EBF 3435-2 [1]. The modified calculation steps for the design of joints with lockbolts in mechanical engineering are presented in this article.

Since the calculation steps have been limited to concentric clamped and loaded joints with lockbolts, the author is confronted with the question how the load bearing behavior of lockbolts can be described under static and alternating load under the influence of an eccentric operating load \( F_A \) and the associated additional bending load. The authors already investigated this question in the AiF public research project (AiF 17824BR) in order to incorporate the findings gained here into the Technical Bulletin DVS-EBF 3435-2 [1, 28]. In addition, recommendations for the detection of multiple bolt connections using the Finite Element Method are generally planned for VDI 2230 – Blatt 2 [29].

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