Implementation of a bolted joint model in Modelica

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Abstract

The basic mechanics of a bolted joint are well-known and have been studied for a long time. The dominating principle is to represent the parts in a joint as a series connection of linear compression and tension springs. However, traditional models often neglect the tightening dynamics and their interrelation with, for instance the friction or embedment. To study these phenomena further and determine their impact on the tightening process and dynamics, and for developing new tightening control strategies, it is necessary to model a threaded fastener and implement it in a suitable simulation environment.
Existing models and experimental data have been studied to find equations that fit the observed behavior. Novel models were combined with standard Modelica components to form a threaded fastener model. The simulation results were compared with tightening data from experiments. This work proposes new models for the first three tightening phases, embedment, and threaded fastener friction. These models are implemented in the modeling language Modelica. The results show that it is possible to resemble a typical threaded fastener tightening with power tools. The friction and tightening phases show the expected behavior, while the embedment model needs further experimental verification. During modeling, the model is susceptible to the chosen parameters. Parameters for the joint stiffness, obtained via the VDI guidelines, needed to be reduced by 30% to resemble the joint in a dynamic simulation.

1 Introduction

Threaded fasteners are often referred to as the most common machine element, and therefore, the importance of threaded fastener joint reliability is apparent. The generated clamp force is difficult to measure but of great importance for the functionality. Therefore, it is of great interest to study the behavior of threaded fasteners under dynamic tightening conditions, and the approach of modeling the fastener is a first step for validating theories, e.g., the impact of friction behavior.

2 System Overview

A threaded fastener joins or holds together two or more components or materials. This is done via the clamp force.
A threaded fastener assembly typically involves an operator, a power tool, and a threaded fastener. This work focuses on threaded fastener behavior.
In its simplest form, a joint is composed of a bolt, a nut, and at least two clamped parts. Additional parts like washers or gaskets are common. In a tightened joint, the bolt is under tension between the bolt head and the engaged thread, while the clamped parts are under compression. Friction occurs during tightening under the fastener head and the adjacent surface and between the bolt threads and the nut. Usually, up to 90% of the applied torque in a tightening goes to overcoming the friction. This highlights the importance of understanding friction during tightening. A difference from many other systems subjected to friction is that the normal load and friction torque are constantly increasing. This leads to very high friction torques and explains the large share of friction losses on the total energy put in the system.
Power tools used for tightening are typically composed of an electrical or pneumatic motor, gears, drive shafts, and housing.

2.1 Tightening Mechanics

The tightening mechanics can be separated into two domains: the rotational domain with driving and load torques and the translational domain with the preload and clamp force.

2.1.1 Rotational Domain

The Kellerman-Klein equation adapted by (VDI - Verein Deutscher Ingenieure, 2015) and originating from (Kellermann & Klein, 1956) describes the rotational domain (see Equation 1). The formula is derived from a special case of Newton's second law where the angular acceleration is zero. M_t is the driving torque, and F_c is the clamp force. The origin of that equation can better be understood by studying the free body diagram of the bolt thread; this can be found in the chapter Torque and Tension in Fasteners in (Oberg et al., 2004).

$$M_t = F_c \left(\frac{P}{2\pi} + 0.58 d_2 \mu_{th} + \frac{D_b}{2} \mu_h \right) \quad (1)$$

The under-head friction torque is the product of the under-head friction radius $\frac{D_b}{2}$, the clamp force, and the underhead friction coefficient μ_h. The pitch torque is the product of the clamp force and the thread pitch P. The thread friction torque is the product of the thread mean radius $\frac{d_2}{2}$, the thread friction coefficient μ_{th}, and lumped geometric parameters.
A model for that domain is found in (Japing et al., 2015).
The transformation from pitch torque into a linear force is done via the thread.

2.1.2 Translational Domain

In the translational domain, two forces act: the preload and the clamp force. The force balance is often visualized with joint diagrams. Such a joint diagram can be seen in more detail in (Shoberg, 2000). The bolt elongation is usually larger than the joint compression, but due to different resilience, the forces are equal. A typically used analogy for the interaction between the bolt and the clamped parts is a two-spring model where a tension and a compression force are coupled in series. During tightening, the fastener constantly rotates relative to the nut. Every angular change $\Delta\varphi$ causes the nut to move upwards on the bolt thread, called Δs. Equation 2 describes the upward movement.

$$\Delta s = \Delta\varphi \cdot P \quad (2)$$

The resulting force increment is described via Equation 3, where s is the length and c is the stiffness. As seen, the total force equals the force in the bolt and joint.

$$\Delta F_c = \Delta s \cdot c_{\text{tot}} = \Delta s_{\text{bolt}} \cdot c_{\text{bolt}} = \Delta s_{\text{joint}} \cdot c_{\text{joint}} \quad (3)$$

The mechanics in more detail can be found in literature (Toth, 2006).

2.2 Tightening Phases

A threaded fastener tightening is usually divided into four different tightening phases. Run Down, Alignment, Linear Elastic Clamping, and Yield. The classification is done via torque traces in the angle or time domain, as in Figure 1
The phases are characterized by the following.
Run Down: The nut is not yet in touch with the clamped parts during run down. The resistance to overcome is friction in the thread due to interference and the acceleration of the rotating masses. The torque

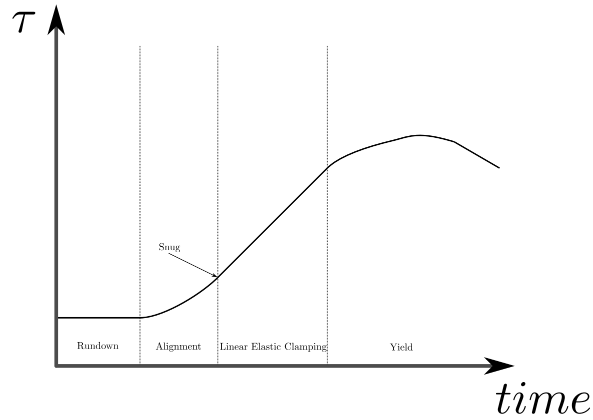

Figure 1. Torque evolution during the tightening phases of a constant speed tightening

during run-down is assumed to be constant. The rundown ends when the torque increases from that constant level. The clamp force is zero during run down.
Alignment: The alignment phase begins when the nut and clamped parts come into contact. The tightening torque increases at a larger non-constant rate. Reasons for the non-linearity, material imperfections, and the initial alignment of the joint components. Alignment ends at snug when the torque increment transitions to a constant rate. The clamp force build-up rate is not constant in this phase.
Linear elastic clamping: In the linear elastic clamping phase, the torque and preload increase happen at a constant rate. Many standard tightening methods, based on torque or angle measurements, end the tightening in that phase.
Yield: Yield is the last phase of the tightening. It starts when the material behavior of the fastener shaft changes from linear elastic to plastic deformation. In the yield phase of a tightening, a linear relationship between the tightening torque and the tightening angle ends.

2.3 Embedment

Embedment during and after tightening leads to a clamp force loss over time. Most prominent is the clamp force loss due to embedment that can be observed after a finished tightening or at short resting times or pauses during tightening. Embedment occurs due to high local stresses on rough surfaces. These high stresses lead to local plastic deformation in the contact regions. As a result, the total length of a component is shortened, which leads to a clamp force loss. Experimental data shows that the clamp force loss rate decays over time. To account for the clamp force loss, fixed losses are assumed based on the surface roughness of the joint components. The exact dynamics of embedment are not analytically described. It can be concluded that a longer tightening duration results in less post-target embedment. Another insight is that more embedment happens at higher loads.

3 Modelica Implementation

The previously described system is the basic structure combining a rotational oscillator and a translational oscillator. The connection between these two oscillators is via the kinematics in the thread, which converts between the rotational and translational domains. This system is driven by an input torque and is overdamped due to friction. Implementing such a complex system with a component-based modeling language like Modelica allows rapid model development and offers a convenient way of evaluating different submodels.
The entire model is built mainly with components from the Modelica standard library. Components from the mechanics rotational and translational libraries are used. The components developed explicitly for the threaded fastener model are: *HeadFriction*, *ThreadFriction*, *IdealThread*, *ThreePhaseBolt*, and *Embedment*.
The composed model can be seen in Figure 2.
The order of the models in the rotational domain is the following from left to right: *Control block*, *Torque*, *TorqueSensor*, *BearingFriction*, *Inertia*, *Spring* and *Damper*, *HeadFriction*, *Inertia*, *Spring* and *Damper*, *ThreadFriction*, *BearingFriction*, *Inertia*.
Connected via the *IdealThread* follows the translational domain, the models are from right to left: *ForceSensor*, *ThreePhaseBolt*, *Mass*, *Embedment*, *SpringDamper*, *Fixed*.
The translational domain has one difference between the theoretical model and the implementation. The joint and the bolt are modeled as compression springs, contrasting the tension and compression spring serial connection found in the literature.

3.1 Friction Models

The implementations of the friction models can be seen in Appendix A.1 for the head friction model. Apart from the two parameters, the thread friction model is identical and therefore not shown. Both are based on the rotational Brake model from the Modelica standard library.
The friction implementation is a state machine with the states: *backward*, *forward*, *free*, or *stuck*. The transition to the stuck state is made when the velocity is zero. The acceleration of the component is set to zero in the stuck state. The condition for a forward and backward movement is that the sum of the external torques is larger than the defined friction torque at zero velocity, multiplied by the given peak factor. This behavior is essential for the friction component to behave like friction in threaded fasteners. Otherwise, the fastener would unwind after the input torque is removed.
The implementation is coulomb friction combined with speed-dependent dynamic friction. For the fastener model, a friction coefficient that increases with speed is essential to avoid oscillations of the inertia and masses.
The remaining adaptations are to align the parameters with the friction radii as in Equation 1. For the thread friction model, d_2 is directly given as a parameter. For the head friction model, the friction radius $\frac{D_b}{2}$ is calculated from the plane head bearing diameter of the bolt d_W and the plane head bearing areas inside diameter D_{ki}.

3.2 Embedment Model

The embedment model is based on the rod model from the Modelica standard library. The difference is that the component has a variable length that becomes negative under pressure. The model can be found in Appendix A.2.
The length change is described by Equation 4. The authors invented this equation to model the clamp force loss due to embedment.

$$\dot{L} = -\frac{1}{\tau}\left(L - L_{\max}\left(\frac{f}{f_{\max}}\right)\right) \tag{4}$$

Here, L is the length of the embedment surface, $L_{\max}$ the maximum possible embedment, f the force applied, $f_{\max}$ the maximum possible force, and τ the time constant defining how fast the process happens.

3.3 Thread Model

The thread model is based on the *IdealGearR2T* model from the Modelica standard library. The functionality is identical, and only the input parameters have been changed to align with the terminology from threaded fasteners. The model can be seen in Appendix A.3.
The model parameter is the pitch per revolution P instead. The torque (τ) and force (f) relationship is given by Equation 5. The rotation (φ) displacement (s) relationship is given by Equation 6.

$$\varphi\frac{P}{2\pi} = s \tag{5}$$

$$\tau = f\frac{P}{2\pi} \tag{6}$$

3.4 Tightening Phases

The first three tightening phases are distinguished by what happens with the joint and bolt.
During run down, the nut and the joint are not yet in contact, so neither the bolt is stretched nor the joint compressed. Since the rotation is applied to the bolt,

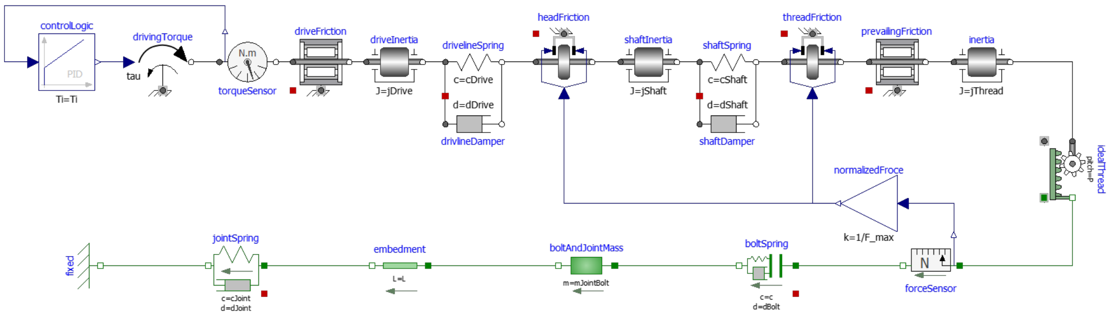

Figure 2. Threaded fastener model, implemented with Open Modelica

the bolt can be seen as a rotating inertia at a free end. During the alignment phase, when the nut and the joint come in contact, the bolt is stretched while the joint is compressed. The displacement force relationship, and thereby even the displacement-torque relationship, is not linear. The rotation of the bolt is opposed by a resistance that is ultimately caused by the bolt stretch and joint compression. Therefore, the rotating bolt can no longer be considered inertia on a free end.
In the linear-elastic clamping phase, the displacement-force relationship changes from non-linear to linear. These effects are lumped into one component. The choice was made to lump the free-end and non-linear behavior into the bolt model, creating the three-phase bolt. Further experiments would be needed to identify what component contributes how much to the nonlinearities, and due to the lack of insight, these effects are combined into the bolt with the reasoning that the bolt is earlier in the drive line and that this arrangement will minimize the risk for unintended oscillations.

The model for the three-phase bolt is based on the *ElastoGap* component from the Modelica standard library. The spring force for the three-phase bolt is modeled according to Equation 7.

$$f_c = \begin{cases} 0 & \text{if} \quad s_{\text{rel}} < s_{\text{rel0}} \\ c_{\text{quad}} |s_{\text{rel}} - s_{\text{rel0}}|^2 & \text{if} \quad s_{\text{rel0}} \geq s_{\text{rel}} < s_{\text{rel2}} \\ c_{\text{lin}} |s_{\text{rel}} - s_{\text{rel1}}| & \text{else} \end{cases} \tag{7}$$

For such a model, c_{lin}, s_{rel0} $s_{\text{rel}p2}$ are the model parameters for the linear spring constant, the rundown displacement, and the relative displacement for the second phase, namely the alignment. The remaining parameters c_{quad}, s_{rel1} are the quadratic spring constant and the hypothetical crossing of the linear spring phase with the zero force line.

The parameters c_{quad}, s_{rel1} can then be derived through the fact that there is a smooth transition between the phases, which means that the derivative is equal in the transition points. That yields Equations 8, 9, and 10 for the parameters.

$$s_{\text{rel}_1} = s_{\text{rel2}} - \frac{s_{\text{rel2}} - s_{\text{rel0}}}{2} \tag{8}$$

$$s_{\text{rel}_2} = s_{\text{rel0}} + s_{\text{rel}p2} \tag{9}$$

$$c_{\text{quad}} = \frac{c_{\text{lin}}}{2(s_{\text{rel2}} - s_{\text{rel0}})} \tag{10}$$

The model for the implementation is given in Appendix A.4

4 Verification

The objective of the fastener model is to approximate a real tightening trace from tightening experiments. For that, the two-step tightening from (Persson et al., 2021) is taken as a reference. The speed profile, the clamp force trace, and the torque trace can be seen in Figures 3, 4, and 5, respectively. Here, the data from the experiments is marked as recorded data in blue. The joint is an M10×70 hexagonal flange head fastener type of strength class 8.8 with Zn-Fe coating + wax. The clamp length of the joint is 56 mm, and the coefficient of friction has been determined to be 0.147 ± 0.016 (±3σ) according to ISO16047 at a tightening speed of 20 rpm. The set target torque was 43 Nm.
An initial guess for the simulation parameters is made with the given data. They are based on the VDI guidelines ((VDI - Verein Deutscher Ingenieure, 2015)), taken from data sheets of the used equipment, read from the given plots, or are empirical values based on modeling experience.
From the initial guess to the tuned parameters, the Bolt and Joint Stiffness were reduced by 30%. The initial guess has a rundown and alignment angle of 70° and 52°. The remaining parameters remained unchanged.
The tightening parameters are set to 100 rpm until 21 N m are reached. The pause step is 50 ms, and the final step is 20 rpm until 43 N m are reached.
The tuned parameters can be seen in Table 1. The change in % refers to how much the parameter was altered from the initial calculation to the final parameter. Further geometric parameters are d_2 7.19 mm ,

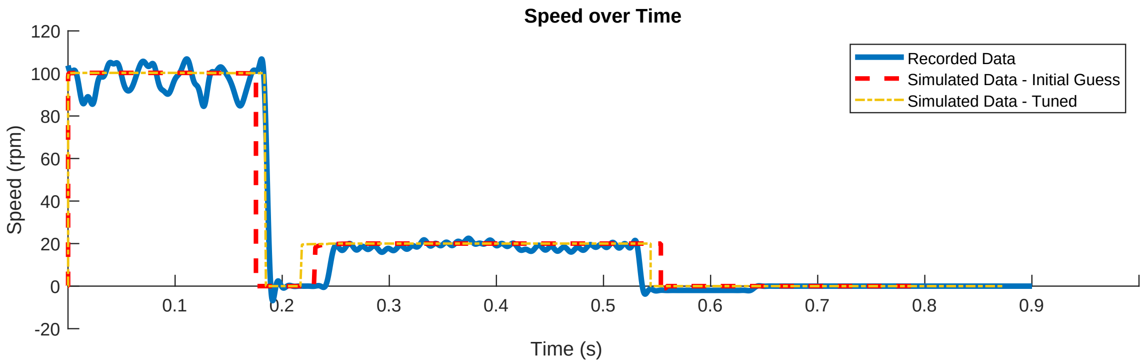

Figure 3. The speed over time trace of the recorded tightening, simulated tightening with initial guess parameters, and the simulated tightening with tuned parameters

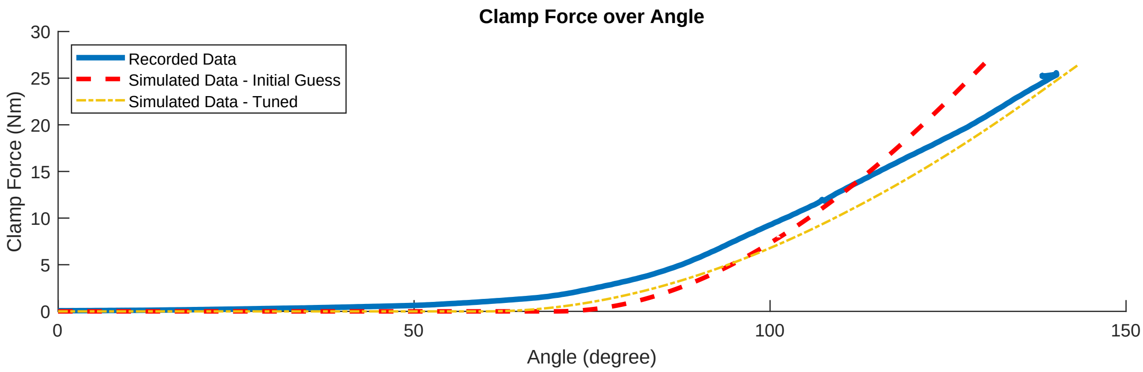

Figure 4. The clamp force over angle trace of the recorded tightening, simulated tightening with initial guess parameters, and the simulated tightening with tuned parameters

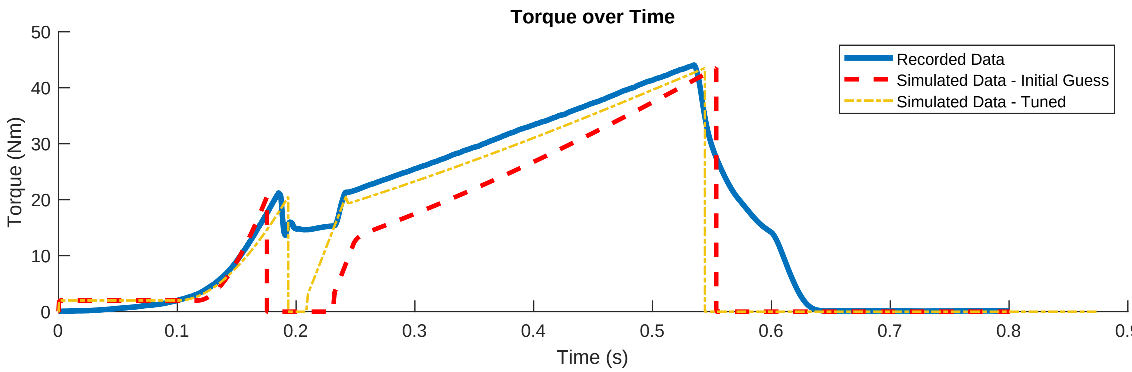

Figure 5. The torque over time trace of the recorded tightening, simulated tightening with initial guess parameters, and the simulated tightening with tuned parameters

Table 1. Tuned parameters used for the simulation

Parameter	Tuned Value	Change in %
Bolt and Joint Mass (kg)	0.3356	0
Bolt Stiffness (N/m)	$1.5758 \cdot 10^8$	30
Bolt Damping (N s/m)	$2.4106 \cdot 10^2$	0
Shaft Stiffness (N m/rad)	$1.3685 \cdot 10^3$	0
Shaft Inertia (kg m^2)	$2.8846 \cdot 10^{-7}$	0
Shaft Damping (N s/m)	$2.4106 \cdot 10^2$	0
Thread Inertia (kg m^2)	$1.7572 \cdot 10^{-7}$	0
Joint Stiffness ((N/m)	$8.6478 \cdot 10^8$	30
Joint Damping (N s/m)	$1.5279 \cdot 10^3$	0
Driveline Inertia (kg m^2)	$3.45 \cdot 10^{-6}$	0
Driveline Stiffness (N m/rad)	265.74	0
Driveline Damping (N m s/rad)	$1.63 \cdot 10^{-6}$	0
CoF Thread 0/20 rpm (-)	0.145 / 0.147	-20.8 / 0
CoF Bolt Head 0/20 rpm (-)	0.145 / 0.147	-20.8 / 0
Prevailing Torque(Nm) 0/100 rpm	0.7 / 1.3	0 / 0
Driveline Torque(N m) 0/100 rpm	0.2 / 0.7	0 / 0
Rundown / AlignmentAngle (°)	60 / 70	-20 / -66
Embedment total (m)	$8 \cdot 10^{-8}$	0
Embedment time constant (s)	0.5	0
Preload at yield (N)	32000	0

D_{Ki} 9 mm , P 1.5 mm, and d_W 11.63 mm. The ratio between static and dynamic friction is 1.5 for all friction components. The control parameters are kp = 100, Ti = 0.005, maximum Torque yMax = 50 N m, rundown speed = 100 rpm, final speed = 20 rpm, rundown torque = 20.5 N m, final torque = 43.5 N m, break = 0.05 s.

As a result of the tuning, a reduction of the bolt and joint stiffness by 30% was made. The length of the alignment angle was increased by 8 degrees, while the rundown angle was shortened by 10 degrees.

The speed profile of the recorded data is only followed to a certain degree, as seen in Figure 3. In the model case, an optimal torque source with no delays. Moreover, a relatively fast controller is used. Due to that follows the modeled result the reference value better. It can nevertheless be seen that the duration of different speeds deviate from the reference.

Overall, the model resembles the tightening data well, even if there is no exact match between the recorded data and the modeled tightening. A better fit can be obtained with further tuning or optimization of the parameters. Regardless of that, it can be seen that the characteristic elements of the tightening are accurately represented with the simulation model.

5 Results and Discussion

5.1 Modelling Process

The modeling process was iterative. The involved components were tested in isolation, in combination with other components, and how different parameter ranges affect the behavior of the components in composed systems. When working with OpenModelica, comparing the simulated results with variations of models and parameter combinations could be more convenient. Tracking and relating the tested parameters to simulation results gets more complicated with a growing system complexity.

A suitable method for that modeling work has been to do the modeling work and system composition within OMEdit and experiment with initial parameters in fast iterations. Then, the model can be loaded into an OMNotebook. There, it is of greater convenience to study how different parameters impact the modeling result while at the same time keeping track of the parameter changes. That is of increased importance for threaded faster modeling due to the interrelation of the parameters, which often prohibits the change of just a single parameter. An example of coupled parameters is the dependency of the mass, stiffness, and damping on the geometry of the component, so a change in length would impact all of them, while a change of one of the mentioned parameters would require the others to be changed to be consistent.

5.2 Model Alignment

The initial parameters obtained via analytical calculations following the VDI2230 guidelines showed discrepancies between the recorded data and the model. This was expected since the calculations are simplifications based on the static case. It can be seen in Figure 4 that the clamp force increases too much with advancement in angle. Similarly, the initial torque trace is rising too fast, as seen in Figure 5. A more similar torque rate was achieved by reducing the bolt and joint stiffness by 30%. The exact reasons for the discrepancy are not further studied but are left for future work. Still, it can be taken as a result that the analytical stiffness for threaded fastener joints overestimates the clamp force rate if applied to the given model.

As seen in Figures 3, 4, and 5, the initial agreement could be more optimal. The model fit can be improved with parameter tuning. Beyond the first shown attempt, this is left to future work.

Overall, the system has a good agreement with the recorded data. Several experiments have been done to verify the model, but only the data from one experiment has been used for tuning. Therefore, the data from further experiments is not included in the results. Non-optimized model parameters can explain the discrepancies and can be minimized further. Hence, the model agrees well with a real threaded fastener joint. One important conclusion is that the data obtained in alignment with the VDI guidelines is either unsuitable for dynamic modeling in general or for the specific implementation of the model.

5.3 Model Components

The modeling work resulted in three new component models and is otherwise composed of standard components from the Modelica library. The models fulfilled the purpose in the system model and contributed to the overall model agreement with the recorded data.

5.3.1 Friction Components

The purpose of the modified friction components is to resemble the friction under the fastener head and in the thread, respectively. This behavior was already implemented via the brake component from the Modelica standard library. These models are simplified so that most of the effects, such as lubrication and surface profiles, are described through the coefficient of friction. All that was necessary was a redefinition of the model parameters to match bolt nomenclature. Both components work well in the complete system, which can be observed while studying the brake modes. The brake modes indicate in which state the friction component is, which could be either moving backward or forward, stuck or free. The behavior for transitioning from stuck to any other state is essential for the model to resemble the behavior of an actual fastener. This is implemented via the peak ratio. Without that behavior, a threaded fastener would unwind again once the applied torque is taken away or set to zero.
During the development of the model, the friction behavior caused the biggest challenges. It could be observed that the friction components did not transition to a locked state, even though, based on empirical experience from threaded fasteners, they should have. From a simulation perspective, that occurred since the condition to transition to a locked state - a relative velocity of 0 - was never met. That happened especially when the adjacent components had relatively small masses or inertias. One observation is that this behavior does not occur in the overall system with the given parameters. Further studies must be done to determine if a particular parameter combination is causing that behavior. For now, it can only be concluded that in terms of stability, it is more beneficial to handle the entire system than extra single components and test them in separate test scenarios.

5.3.2 Three-phase bolt

The three-phase bolt is a key component in resembling the different tightening phases. As seen in the shown results, the chosen approach works, and the threaded fastener behaves accordingly in the different stages, which are determined via the tightening angle. The difference, especially during the early phases of the tightening, is instead a debate about the presence of a run-down in the recorded data. It can be addressed by shortening the rundown phase and extending the alignment phase in exchange. That could come at the cost of a worse fit and lead to the conclusion that a second-degree polynomial function for the spring force is unsuitable. Further studies have to be done to determine that.
The transition between alignment and the linear elastic phase is smooth and working as in the recorded data; therefore, is the chosen approach sufficient for the modeling purpose.
The recorded data has a relatively short run-down but a much longer alignment phase, as seen in Figure 4. Further studies are needed to confirm that a second-degree polynomial function for the spring force during alignment is a good fit for the case of such a long alignment phase.

5.3.3 Embeddment

The effects of the embedded can be read from the model where a clamp force loss due to a shortened spring length can be detected. In the tightening trace, that can not be detected or distinguished from a tightening with a different spring stiffness. The embedment effect is only vaguely present in the recorded data. A more suitable tightening scenario must be chosen to further investigate the embedment model's alignment. This could be a joint with rougher surfaces to increase the embedment effects and a shorter tightening duration by higher tightening speeds to relocate more of the embedment effect after the final shutdown. For the current model, it can hence only be concluded that it is possible to model embedment in the proposed way, that the effects are present, and that an embedment component in the spring chain does not negatively impact the other components, such as the friction components, as long as the entire threaded fastener model is kept as whole. In test scenarios where the embedment component was tested with only one friction component, one spring, and one inertia, it kept the friction component from locking until the maximum embedment was reached.
Embedment is not very distinct in the recorded data. Therefore, no quantitative comparison is possible. It can be concluded that the chosen embedment model works in principle but that the effects must be studied further.

6 Conclusions

It can be concluded that the proposed model does resemble a threaded fastener during tightening and a simplified driveline. Some differences could be observed. Most of the differences were due to differences in the control input for the system. The remaining deviations can be minimized or reduced by tuning model parameters. Hence, the overall model is a valid representation of a threaded fastener. The main insights generated by the modeling work are:

- The model is very sensitive in terms of parameter combinations

- The parameters obtained by the VDI guidelines overestimate the spring systems stiffness

The combination of these factors impedes the development of fastener models, while the actual parameters are difficult to obtain. At the same time, the model does not work accurately if the wrong model parameters are chosen. Therefore, a good test case with well-known parameters is beneficial when developing models of that type. A comparison with parameters obtained by commercial FEM software and to what model fit they lead could indicate if more suitable stiffness parameters could be obtained for future model development.

Acknowledgment

We thank Erik Persson and his Tightening Technique team from Atlas Copco Industrial Technique for the support and discussions leading to the model's development. Atlas Copco fully finances the work, and therefore, we would like to thank Daniel Brandt for sponsoring the research work. The work was conducted as part of the Linköping Center for Sensor Informatics and Control. Therefore, we would like to thank Svante Gunnarsson and the Ph.D. students within the center for their support.

References

Japing, A., Seibel, A., & Schlattmann, J. (2015). Modellentwicklung zur beschreibungvon reibschwingungen bei der schraubenmontage. *Reibung, Schmierungund Verschleiß: Forschung und praktische Anwendungen. 52. Tribologie-Fachtagung*.

Kellermann, R., & Klein, H.-C. (1956). Berücksichtigung des reibungszustandes bei der bemessung hochwertiger schraubenverbindungen. *Konstruktion*, *8*, 236.

Oberg, Jones, Horton, & Ryffel. (2004). *Machinerys handbook* (27th ed.). New York, NY: Industrial Press.

Persson, E. V., Kumar, M., Friberg, C., & Dressler, N. (2021). *Clamp Force Accuracy in Threaded Fastener Joints Using Different Torque Control Tightening Strategies* (Tech. Rep.). Warrendale, PA. doi: 10.4271/2021-01-5073

Shoberg, R. S. (2000). Engineering fundamentals of threaded fastener design and analysis. i. *Fastening*, *6*(2), 26–29.

Toth, G. (2006). *Torque and angle controlled tightening of bolted joints* (Unpublished doctoral dissertation). Göteborg.

VDI - Verein Deutscher Ingenieure. (2015). *Systematische berechnunghochbeanspruchter schraubenverbindungenzylindrische einschraubenverbindungen* (techreport No. 1). Beuth Verlag GmbH.

A Appendix

A.1 Head Friction Model

```
model HeadFriction
  extends
      PartialEl..TwoFlangesAndSupport2;
  parameter Real mu_pos[:, 2]=[0, 0.5];
  parameter Real peak(final min=1) = 1;
  parameter Real dW(final min=0);
  parameter Real Dki(final min=0);
  parameter SI.Force fn_max(final min
      =0, start=1);
  extends
      Rot.Interfaces.PartialFriction;
protected
  parameter Real frad(final min=0) = (
      dW+Dki)/4;
equation
  mu0 = interpolate(mu_pos[:,1], mu_pos
      [:,2], 0, 1);
  w_event = w_relfric > 0;
  phi = flange_a.phi - phi_support;
  flange_b.phi = flange_a.phi;
  w = der(phi);
  a = der(w);
  w_relfric = w;
  a_relfric = a;
  flange_a.tau+flange_b.tau-tau = 0;
  fn = fn_max*f_normalized;
  tau0 = mu0*frad*fn;
  tau0_max = peak*tau0;
  free = fn <= 0;
  tau = if locked then sa*unitTorque
      else if free then 0 else frad*fn*(
    if startForward then
      interpolate(mu_pos[:,1], mu_pos
          [:,2], w, 1)
    else if startBackward then
      (-interpolate(mu_pos[:,1], mu_pos
          [:,2], -w, 1))
    else if pre(mode) == Forward then
      interpolate(mu_pos[:,1], mu_pos
          [:,2], w, 1)
    else (-interpolate(mu_pos[:,1],
        mu_pos[:,2], -w, 1)));
end HeadFriction
```

A.2 Embedment Model

```
model Embedment
  ...
  parameter SI.Distance L_max = 0.01;
  parameter SI.Time tauT = 0.5;
  parameter SI.Force fmax = 10;
  parameter SI.Force fmin = 10;
equation
  flange_a.s = s - L / 2;
```

```
  flange_b.s = s + L / 2;
  0 = flange_a.f + flange_b.f;
  der(L) = if flange_b.f < -fmin then
      -1 / tauT * (L - L_max * (
      flange_b.f / fmax)) else 0;
end Embedment;
```

A.3 Thread Model

```
model IdealThread
  extends PartialElementaryRotational..
  ToTranslational;
  parameter Real pitch(final unit="m",
      start=1);
equation
  (flangeR.phi - internalSupportR.phi)*
      pitch/(2*pi) = (flangeT.s -
      internalSupportT.s);
  0=flangeR.tau+flangeT.f*pitch/(2*pi);
end IdealThread;
```

A.4 Three Phase Bolt

```
model ThreePhaseBolt
  extends ..Interfaces..
  PartialCompliantWithRelativeStates;
  parameter
      SI.TranslationalSpringConstant
      c_lin(final min = 0, start = 1);
  parameter
      SI.TranslationalDampingConstant d(
      final min = 0, start = 1);
  parameter SI.Position s_rel0 = 0;
  parameter SI.Position s_rel_phase2;
  ...
algorithm
  s_rel2:= s_rel0 - s_rel_phase2;
  c_qua:= c_lin/(2*(s_rel0-s_rel2));
  s_rel1:= s_rel2-((s_rel2-s_rel0)/2);
equation
  contact =s_rel < s_rel0;
  linear =s_rel < s_rel2;
  f_c_lin =-c_lin*abs(s_rel - s_rel1);
  f_c_qua =-c_qua*abs(s_rel-s_rel0)^2;
  f_c =smooth(1, noEvent(if contact
      then f_c2 else 0));
  f_c2 =smooth(1, noEvent(if linear
      then f_c_lin else f_c_qua));
  f_d2 =if contact then d*v_rel else 0;
  f_d =smooth(0, noEvent(if contact
      then (if f_d2 < f_c then f_c else
      if f_d2 > -f_c then -f_c else f_d2
      ) else 0));
  f =f_c + f_d;
  lossPower = f_d*v_rel;
end ThreePhaseBolt;
```