Implementation of a Modelica Library for Simulation of High-Lift Drive Systems

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Abstract

The development and design of new high-lift drive systems is a complex and iterative process, which is often depending on experience. Especially results determined in the predevelopment phase and based on uncertain assumptions have decisive influence on the system specification and thus on the system design. In order to reduce development time and optimize the development process, a rapid generation and adaptation of simulation models for analysis of transient system behaviour is essential. This article presents an computer-integrated approach for further reduction of the high-lift development process. An interface to Modelia should enable an automated system model generation. A suitable component library is introduced and verified by simulation of the Airbus A380 flap actuation system.

The purpose of this article is to present the project of a computer-aided development process as well as an adequate component library for assembling simulation models of high-lift drive systems.

Keywords: high-lift system; power drive system; system development

1 Introduction

In order to reduce take-off and landing airspeed, modern transport aircraft are equipped with high-lift systems. The extension of slats and flaps at the wing's leading and trailing edges augments the effective wing area and also allows for higher angles of attack thus increasing the lift coefficient. Figure 1 depicts the high-lift surfaces, as well as the corresponding drive and actuation system.

A central power drive unit (PDU), mounted in the center fuselage, provides energy for driving a shaft transmission, which ensures mechanical synchronisation of the left and right actuation systems. The shaft transmission is routed across the wingspan by numerous bearings, while universal joints and gearboxes compensate changes in direction. Branch gears transmit the mechanical energy to rotary or ballscrew actuators which are coupled with the flap traverse mechanism. High actuator gear ratios reduce fast turning transmission inputs to slow panel movement.

As part of the secondary flight control, the high-lift drive system has to be fault-tolerant and fulfill high requirements regarding the reliability. While the power drive unit and the slat flap control computer are of redundant design, the shaft transmission system offers a single load path only. Sufficient mechanical strength of all elements in the actuation system is required for all possible system states. Peak loads occurring as a result of a system failure are often a design case for the mechanical components of the drive system. Thus, the analysis of transient system behaviour is of uttermost importance for the determination of strength requirements for the drive train's mechanical elements. As aerospace applications require certified components, no standard but custom-build components and assemblies have to be installed. In consequence, component parameters characterising their dynamical behaviour, e.g. the mass moment of inertia or the friction characteristics, are unknown in the early design phase. Thus, these parameters have to be estimated based on the knowledge of existing similar products.

Owing to numerous changes of the system architecture, requirements, constraints or parameters, the effort for installing and maintaining a complete simulation model in the early design and specification phase is not justified. For this reason, simplified models are used for a rough evaluation of peak loads, while adequate safety margins compensate uncertainties. However, increasing mechanical strength normally involves an increasing mass. Thus, considerable potentials in system weight reduction might be wasted.

In this report an integrated approach is presented that aims at an optimisation of the high-lift drive system, as well as its development process. Moreover, an au-

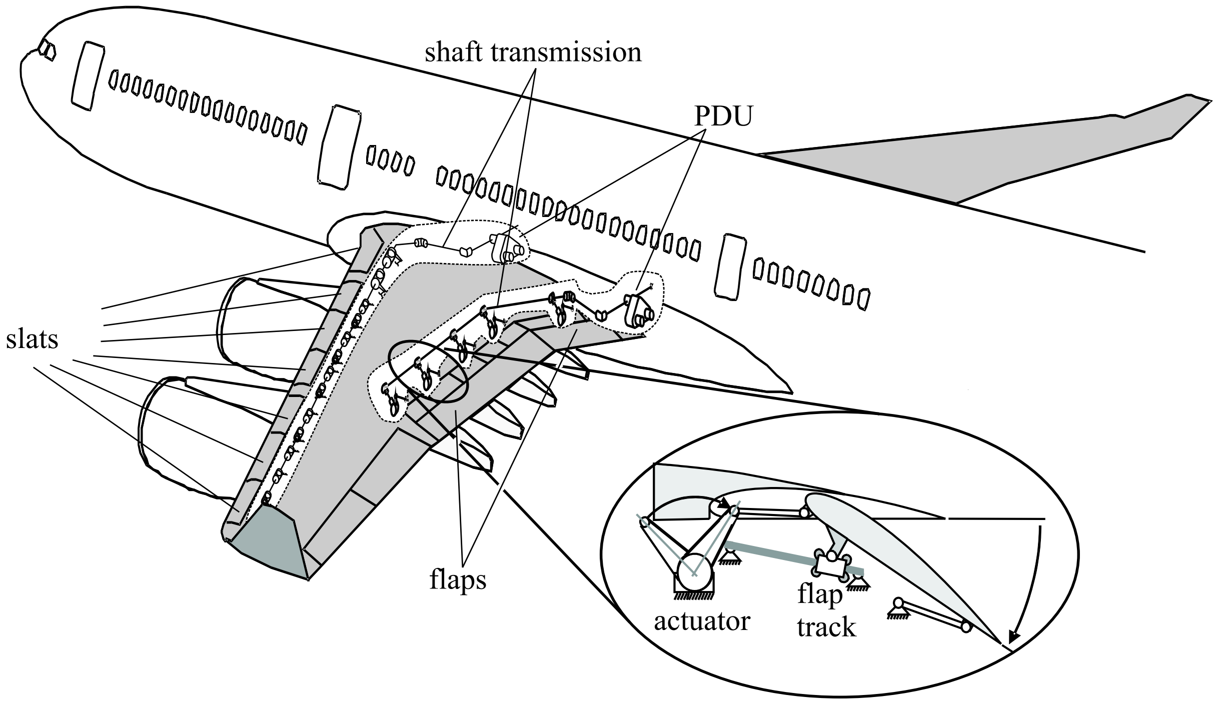


Figure 1: High-lift segments and power drive train at leading and trailing edge

tomated generation and easier maintenance of a complex simulation model for analysis of transient system behaviour should be realised in order to make simulation results available in the predevelopment phase. A software tool combining knowledge based methods for high-lift design and steady state calculations is to be extended to transfer available system information into a simulation model for analysis of transient behaviour. Modelica's characteristic of being object-oriented and providing a simple way to generate simulation models by combining library components makes it predestined for this task. In order to facilitate modelling a complex high-lift drive train, a library containing all required components has been created.

2 System description and modelling

The basic elements of a high-lift drive system, namely the power drive unit, the actuators and the shaft transmission connecting actuators and drive unit, were introduced in chapter 1. Besides gearboxes, shafts, joints and bearings that are essential for the shaft routing, there are further components required to react to mechanical failures. A shaft rupture leading to a separation of flap segments might result either in an asymmetric flap setting or even the complete transmission system might be decoupled from the PDU so that the aircraft looses its high-lift function in a critical situation. Furthermore, jamming in the flap tracks might cause an asymmetric flap setting as well as an overload in actuation system and wing structure. In order to avoid an unacceptable flap asymmetry that cannot be compensated by the rudders, safety brakes are installed at the spanwise ends of the shaft transmission. These wing tip brakes (WTB) are activated if the monitoring systems identifies a failure by comparing the position at the transmission ends, the drive units output angle and the commanded position.

Moreover, the installation of torque limiting devices reduces loads in the drive system and structure in case of jamming in one of the drive stations. High loads and load gradients result from rapid deceleration of the system by either jamming or brake activation.

The analysis of such transient behaviour requires a nonlinear model. Figure 2 exemplifies a flap drive system architecture and its elements. For the purpose of an acceptable simulation time, modelling each mechanical element separately is not practicable. Thus, adjacent parts are merged into a lumped model. The total inertia and torsional stiffness can easily be calculated from the elements connected in series. Other variables like friction coefficients or backlash can be determined accordingly.

While the system model in contrast to the real drive system posesses concentrated parameters, an appropriate discretisation must not change the dynamic behaviour of the system. Different approaches have

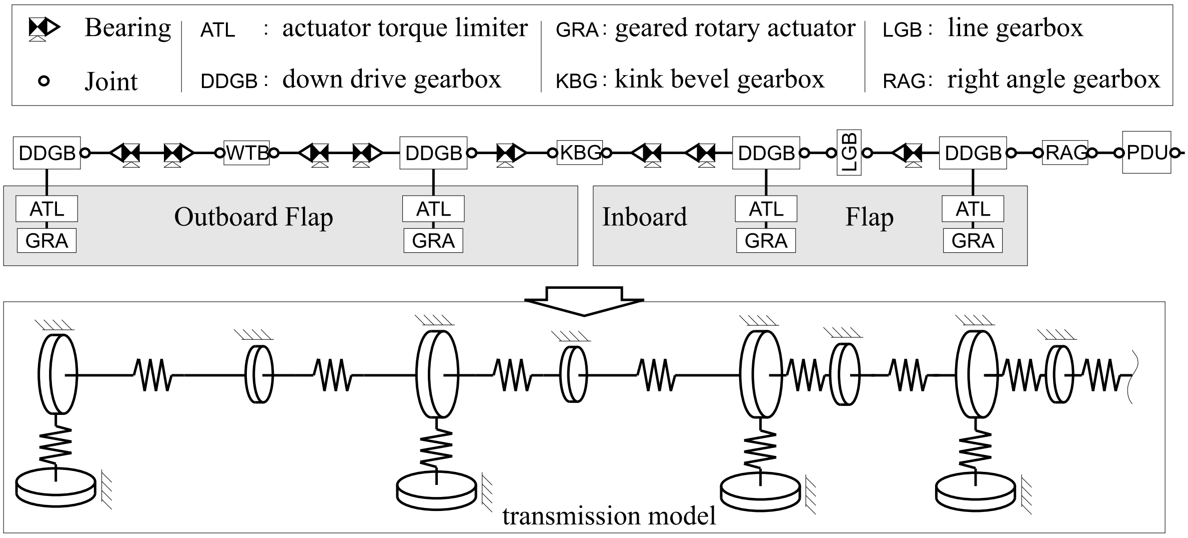


Figure 2: Schematic representation of a flap drive system and its model according to [4]

proven their value. In [2], Neumann proposes to substitute all transmission elements between the down-drives by at least two systems made up of spring, damper and mass. Adapting the distribution of the torsional stiffnesses, the relevant natural modes can be preserved.
A closer look at the distribution of drive system element masses and torsional stiffnesses reveals that installed gearboxes make up a major share of the total mass moment of inertia, while the torsional stiffness is mainly influenced by long shaft elements. Another approach, reducing the model order by summing up the mass moments of inertia around those areas that already show an accumulated inertia, like gearboxes, is presented in [4]. On the other hand, a single torsional spring represents the torsional stiffness of the elements between those inertias.
Both methods show a good correlation between simulation and measured data.

3 Development of high-lift actuation systems

The design and construction process of a new high-lift actuation system starts early in the overall development process of a new aircraft. Thus, only few and uncertain information is available at the beginning. Especially in the concept and preliminary design phase, the requirements, system constraints and component data often change. The data becomes more reliable and more detailed while the development process proceeds. However, mandatory inputs for the design process of the high-lift drive systems that have to be available from the start are:

- the number and type of actuators. The application of either a geared rotary actuator or a ballscrew actuator depends on the planned flap kinematics.
- effective airloads at the actuators.
- wing geometry and available installation space in the wing area.
- maximum travel of the actuators and required time for their adjustment.

The development and design of the mechanical transmission system, the hydraulic and/or electric power drive unit and the monitoring and failure detection system is complex and highly interdependent. The focus in this article is on the actuation system.
With the listed inputs, a first drive system architecture can be designed schematically. To guarantee a uniform motion of the actuators, gear ratios have to be determined accordingly. Mechanical properties of the components have to be estimated at first. Experience from the development of former drive systems is of tremendous value for this parameter estimation.
If gear ratios, characteristic friction coefficients and the architecture are known, steady state calculations, e.g. for determination of the drive torque required by the PDU and torque limiter settings, are possible.
For a rough evaluation of maximum loads resulting from transient changes in consequence of wing tip brake activation or torque limiter lock out, simple models seem practicable. As proposed in [4], the part of the transmission that is in focus of the analysis can

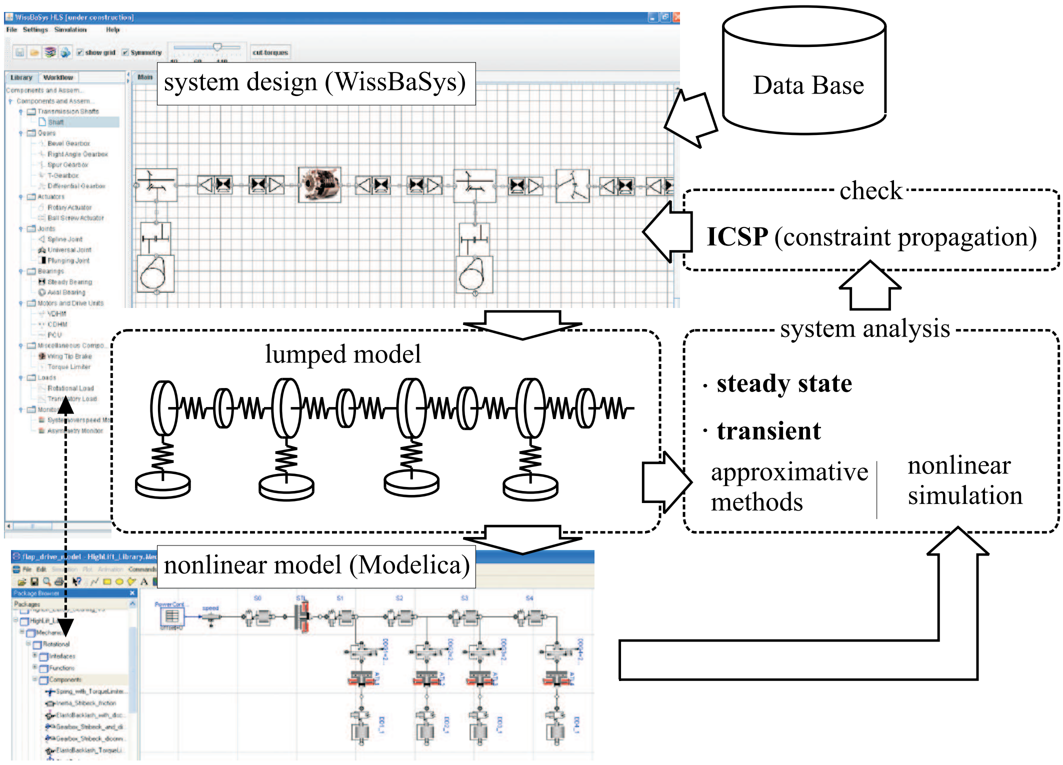


Figure 3: Concept for an computer-aided development process of high-lift drive systems

be transformed to a torsional oscillator with a single inertia J^*. For dynamic similarity, the torsional stiffness c^* of the vibrator is adjusted, such that the first eigen mode of the complete transmission system and the reduced model are identical. Presuming a sudden deadlock in the transmission and neglecting nonlinear influences, the kinetic energy E_{kin} of the transmission converts to potential energy E_{pot} in the spring, allowing the calculation of the peak load:

$$\begin{aligned} E_{kin} &= E_{pot} \\ \Rightarrow \frac{1}{2} \cdot J^* \cdot \omega^2 &= \frac{1}{2} \cdot c^* \cdot \Delta\varphi^2 = \frac{1}{2} \cdot \frac{\tau_s^2}{c^*} \\ \Rightarrow \tau_{s,max} &= \omega_{max} \cdot \sqrt{c^* \cdot J^*} \quad . \end{aligned} \tag{1}$$

Thus, the possibility to do rough system evaluations and trade-offs is provided. For example, the system dependency on the chosen gear ratio could be analysed.

Regarding equation (1), another problem seems obvious. The maximum transmission speed, especially after a mechanical disconnect, depends on nonlinear friction characteristics. While the effort of generating a complex simulation model and the time for running these simulations is not justified as long as most parameters are uncertain and many changes are necessary, the need for more detailed system analysis when the system specification reaches a mature level and reliable data are available is obvious.

In order to reduce development time, the Institute of Aircraft-Systems Technology at Hamburg University of Technology is working on a tool called WissBaSys to support the design process. Particularly, the efforts in early design and specification phases, that are in focus, could be reduced by numerous computer-aided features, which are introduced hereafter.

While the architecture of high-lift transmission systems may change, they generally consist of a relatively small number of different mechanical components. Thus, a library of generic, parameterised components has been created. A graphical user interface (GUI) offers the possibility to connect these generic elements to a complete transmission system. The resulting system layout can easily be changed by adding or removing components.

In order to support the difficult task of parameter estimation when reliable data are not available, not only default values are provided, but also functions describing an interdependence between variables are supported. Furthermore, the user has access to an external database containing extensive information about many existing aircraft components.

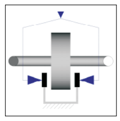
Another characteristic of the preliminary design phase is the handling of uncertain knowledge and checking

the system requirements after every change. For this reason, continuous domains are attached to all variables. This is the basis for an interval constraint satisfaction problem (ICSP). Constraint propagation as it is presented in [5] enables the evaluation of nondirectional equations and inequalities containing variables with interval domains.
Establishing an ICSP brings further useful advantages. Enabling nondirectional evaluation, trade-off studies are encouraged. Furthermore, violations of system requirements or constraints are detected automatically within the constraint propagation process.
The concept of a computer-aided development process is illustrated in figure 3. The system architecture is assembled utilising generic library components. System and component parameters are estimated with help of data base information, default values and empirical estimation functions. An automated generation of simplified models enables approximation of maximum load result from transient behaviour. The system analysis is completed by steady state calculations. The ICSP automatically checks all system requirements so that the basis for a system synthesis is available. While synthesis methods allow for an evaluation and optimisation of slat and flap traverse mechanisms [1] an all-including high-lift optimisation on aircraft level is not available up to now.
Containing all relevant component data, the transfer to a complex nonlinear simulation model would complete the development process. The way Modelica uses for modelling by combining generic library components offers ideal possibilities for an interaction in this context.
WissBaSys supports design studies in early development phases and generates lumped models of reduced order. An appropriate Modelica model has to be named for general concentrated transmission sections. Presuming the allocation of available and model parameters is existent, model instances corresponding to the concentrated parameters can be generated. With the knowledge that some parts execute special function, e.g. the wing tip brake, additional models have to be inserted. If an allocation of simulation models for the mechanical elements in the transmission system is existent, the generation of the complete simulation model can be realised.

4 HighLift library for drive systems

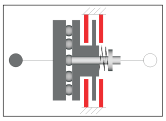
The high-lift drive system consists of the mechanical actuation system, hydraulic drive units, as well as a control and fault detection system. Here, the actuation system and the power drive unit are considered in more detail. For a determination of maximum transmission loads, the mechanical components of the drive train can be modelled as one-dimensional rotational elements. These are characterised by their mass moment of inertia, a torsional stiffness, structural damping, mechanical backlash, gear ratio and the friction characteristic. While the models *Inertia*, *ElastoBacklash* and *IdealGear* of Modelica's standard library cover most of these attributes a new friction model is needed and introduced in this chapter.
Besides models representing a nonlinear torsional oscillator, some components fulfill additional tasks that have to be taken into account. These components are the safety brakes and mechanical torque limiters. The HighLift library contains models for a shaft brake, an ideal torque limiter, the general mechanical rotational part and a geared rotary actuator. Moreover, hydraulic components necessary for modelling hydraulic power drive units are available.
The focus is on the mechanical drive train and its relevant models are discussed in the following. All models are designed such that they need only the information that is relevant for a specification.

Shaft brake



This model represents the brake function of the wing tip brake, which is mounted in the wing structure. If the brake is activated, the compression of friction packages causes a friction torque that stops the transmission. Essential parameters describing the brake behaviour are the maximum dynamic brake torque, maximum static brake torque and the time for reaching the maximum dynamic torque. Thus these are the only input variables of the model which extends the interfaces *Rigid* and *FrictionBase*.
In order to allow different approaches for describing the transient change of the friction torque when the brake is activated, the model's input `u` is the normalized maximum dynamic brake torque `tauB_max`. After reaching a halt, the static friction torque might increase up to the brake's maximum limit load `tauB_lim`. In contrast to the models available in the standard library, friction coefficients are no longer needed here.

Ideal torque limiter



A coupling consisting of balls embedded along the circumferece of two flanges guarantees a positive connection in normal operation mode. In case a torque limit is passed, the balls start to move along a ramp thus pushing one of the flanges against a friction device. The increasing relative angle between the flanges results in an increasing brake torque.
The torque limiting function has two characteristics. First of all, a brake torque depending on the relative angle of the flanges is induced. Moreover, the torsional stiffness changes within the lock out process. While the balls are in motion, the stiffness decreases significantly compared to the normal operation mode. When the balls reach their end stop, the device is grounded and the torsional stiffness changes again.

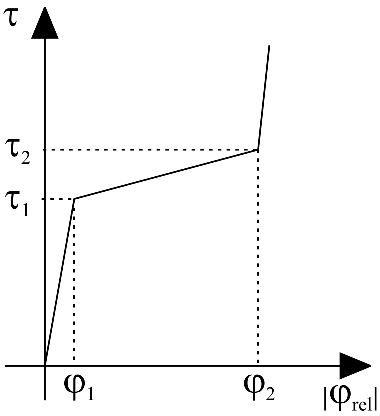


Figure 4: Nonlinear torsional stiffness characteristic of a mechanical torque limiter

For modelling these characteristics, a torsional spring with nonlinear stiffness, according to figure 4, is needed. Furthermore, the dynamic brake torque increases after lock out and reaches its maximum when the balls reach their end stop.
A new spring model has been created. Required inputs are the lock out torque τ_1 and the end stop torque τ_2 as well as the different torsional stiffnesses for all three states. Compared to the standard spring, this model has an additional output y describing a normalized brake torque:

$$y = \begin{cases} 0 & : \quad \varphi_{rel} < \varphi_1 \\ 1 & : \quad |\varphi_{rel}| \geq \varphi_2 \\ \dfrac{|\varphi_{rel}| - \varphi_1}{\varphi_2 - \varphi_1} & : \quad \varphi_1 \leq |\varphi_{rel}| < \varphi_2 \end{cases} \tag{2}$$

Combining the nonlinear spring with a shaft brake as figure 5 shows, an ideal mechanical torque limiter is modelled.

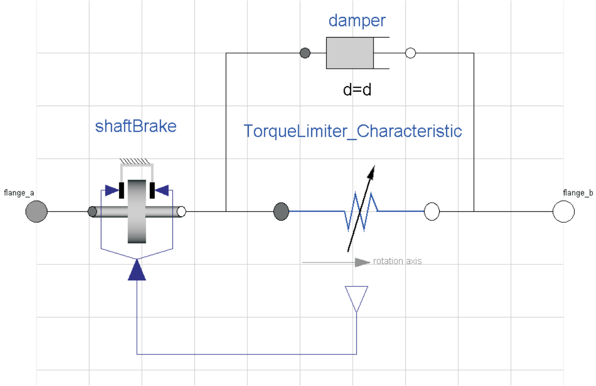
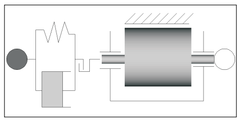


Figure 5: Ideal torque limiter model

General rotational element



As shown in its symbol the general rotational mechanical element consists of an *ElastoBacklash* model and a modified *Inertia* as well as of an *IdealGear*. The *LossyInerta* model takes friction losses into account. Most elements of the transmission system like bearings show friction behaviour corresponding to the Stribeck Friction Law:

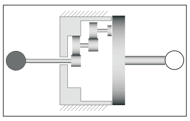
$$\tau_{fric,S} = \tau_{Coulomb} + d_{vis} \cdot \omega + \tau_{Stribeck} \cdot e^{-f_{exp} \cdot |\omega|} \tag{3}$$

However, the detailed analysis of single state gearboxes shows additional friction losses that highly depend on the transmitted loads [6]. This phenomenon is valid only when the unit is in motion and the breakout has occurred. Based on the results of sophisticated analyse of gearbox friction behaviour, a combined approach appears feasible. As discussed in [3] bearing losses and load dependent gear stage losses differ. For representation of a total drag torque, the friction torque is made up of the bearing friction according to the Stribeck law which is depending on ambient conditions and gearbox losses characterised by a gearbox efficiency η_{GE}:

$$\tau_{fric} = \tau_{fric,S} + (1 - \eta_{GE}) \cdot \tau_{load} \quad . \tag{4}$$

While η_{GE} varies between 0 and 1, it represents the dependence on the transmitted torque and is easily determined by measurement.

Geared rotary actuator



Most Airbus aircraft use planetary gears with high gear reduction for flap and slat actuation. Their dynamic behaviour has essential effects on the complete high-lift actuation system. Exact modelling of these components is of vital importance for the reliability of simulation results. Analysis of the friction behaviour of these actuator types also shows remarkable influence of the transmitted loads on the friction torque [2].
Furthermore, the load-dependent friction changes with the energy flow direction. Generally, driving against opposing load has better efficiency than in the case of aiding loads. The load-dependent friction does not occur stepwise as soon as the unit begins to move, but increases smoothly after a change in direction.

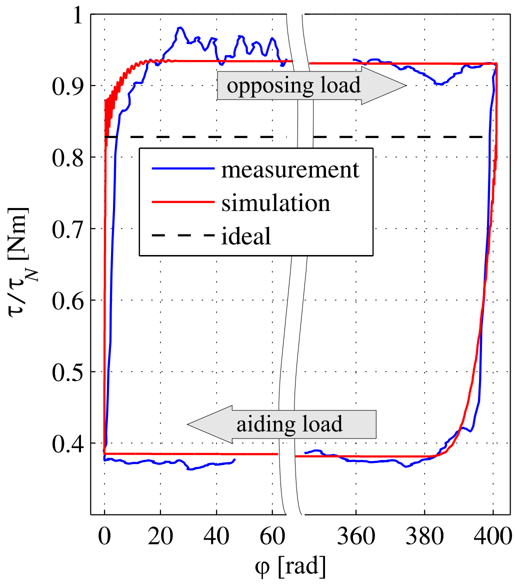


Figure 6: Normalised input torque of a geared rotary actuator with constant load

For validation a geared rotary actuator has been tested and its friction behaviour determined [2]. Simulation results using the model described above show good resemblance to test data as presented in figure 6. For validation of the actuation system measured data of the actuator loads as well as the power drive unit's speed is an inputs to the model. The contact of the gear wheel teeth is the reason for the load-dependent friction torque [3]. When a turnaround occurs, the wheels do not turn simultaneous but consecutively. Thus, the contact between the gear wheels establishes smoothly. Since the geared rotary actuator is modelled as a single stage gearbox in order to reduce the model order, this phenomenon can be represented by the gearbox efficiency η_{GE} as a function of the input angle φ_{in}. If the unit stops, η_{GE} increases linearly to 1 after a speed threshold is crossed. Consequently, load-dependent friction diminishes according to equation (4). When the unit starts to move again, η_{GE} is a function of φ_{in}, while its final value depends on the sign of the transmitted power. Figure 7 shows this characteristic.

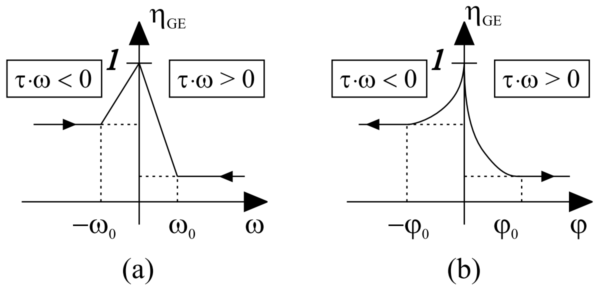


Figure 7: Gearbox efficiency for deceleration (a) and acceleration (b)

Further Models

The models presented in detail here are of vital importance for modelling a complete high-lift actuation system. Furthermore, the HighLift library contains models for inducing mechanical failures in the drive train. For this purpose, an element that can be used for a mechanical disconnection and another model that causes jamming at a specified time are included. Besides the transmission system, the power drive unit is of major interest. Hydraulic component models for turbulent resistances, servo valves, a differential cylinder as well as an example that uses these components for modelling a PDU's drive train with a variable displacement hydraulic motor (VDHM) are included.

5 Transient simulation of Airbus A380 flap actuation system

For a verification of the presented models the Airbus A380 flap actuation system is taken into account. The number of actuators and mechanical elements in total outnumbers that of all other flap actuation systems of Airbus aircraft. The actuation system utilises geared rotary actuators, a wing tip brake and a system torque limiter that is installed between the power drive unit and the first downdrive. A test rig replicating the A380 high-lift drive system of one wing only, has been installed at the Airbus facilities in Bremen in order to run certification tests. Utilising the models

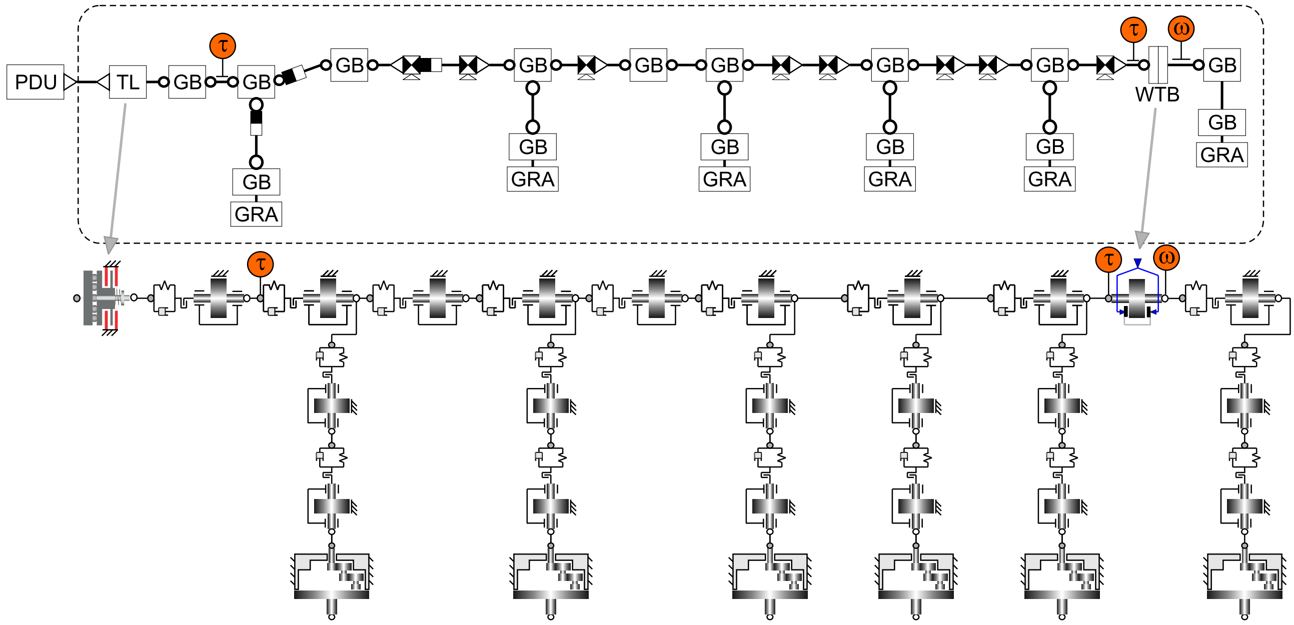


Figure 8: Airbus A380 flap actuation system and model

of the HighLift library the actuation system is modelled and verified by means of measured data. Figure 8 presents a schematic view of the transmission system and a lumped model of reduced order in Modelica. Furthermore, sensor positions are marked in figure 8. For modelling the approach presented in [4] and discussed in 2 is used.

For validation test data of the actuator loads and the PDU speed are used as input. The drive systems starts to operate against increasing opposing actuator loads. After an acceleration phase the system speed is almost constant until a position threshold is reached and the speed is reduced before the system stopps at its determined position.

Figure 9 shows that the speed within the shaft transmission system varies only slightly. Comparing simulation and test data for the input torque at the system torque limiter (STL) that depends on the exact modelling of the complete actuation system, the data show good conformability. While the simulated break out occurs 0.5 seconds earlier than in the test the simulation results are very accurate afterwards. Figure 10 compares simulation and test results for the specified sensor positions.

Now the introduced model is used to analyse a failure case scenario. At t_1 the disconnector model is used to simulate a shaft rupture between system torque limiter and first downdrive while the transmission system

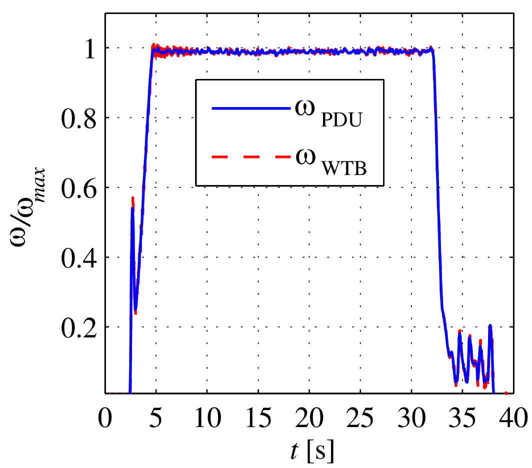


Figure 9: Transmission speed

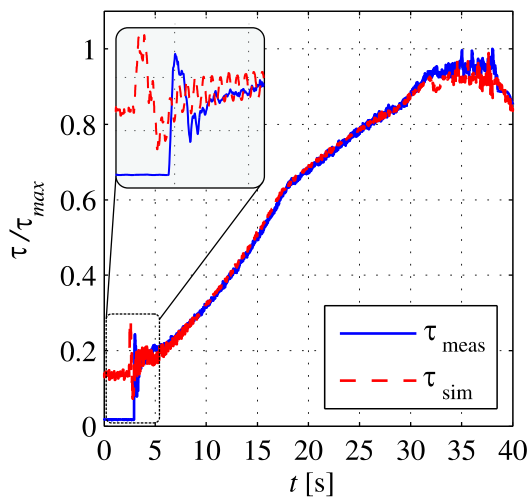


Figure 10: Simulated and measured actuation system input torque

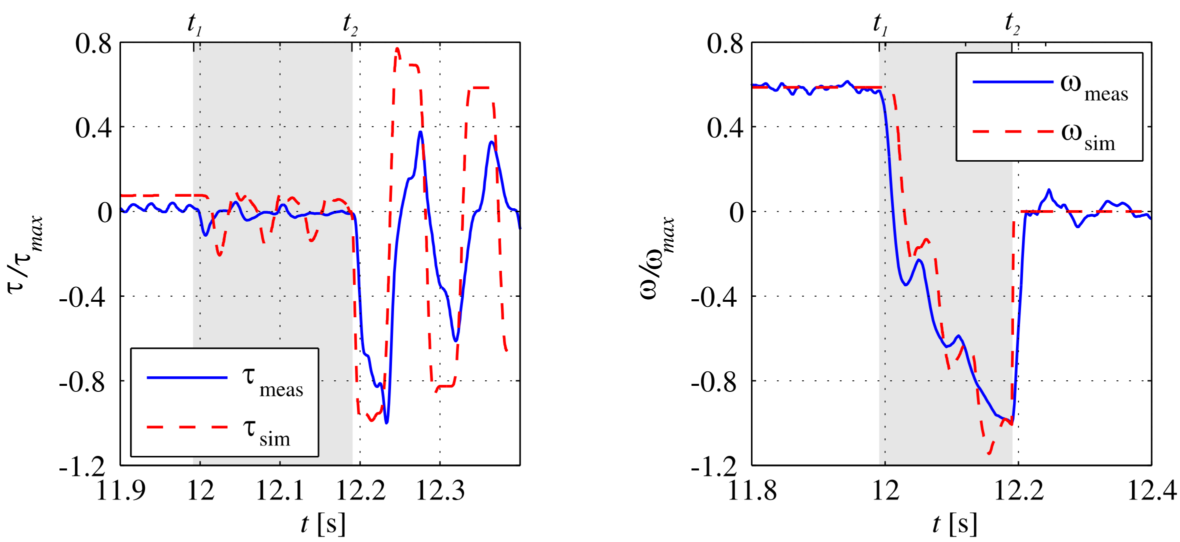


Figure 11: Transmission speed and torque at the wing tip brake after shaft rupture and brake activation

drives against opposing loads. After the mechanical disconnection the complete system is accelerated by the applied actuator loads. The failure is detected and the wing tip brakes are applied at t_2 and cause a system stop. In consequence of the rapid deceleration, load peaks occur within the shaft transmission. The maximum is to be found at the safety brake.

Figure 11 compares test rig data and simulation results for transmission speed and torque at the wing tip brake. Although simulated and measured speed have different gradients during the acceleration phase, their oscillatory behaviour is similar and their value at t_2 is almost identical. The simulated deceleration phase is shorter as it was in the test. Nonetheless, the maximum transmission loads differ only slightly.

6 Conclusion and future work

This article presents the development and design of high-lift actuation systems and its implied challenges. For further reduction of development time for new high-lift systems a computer-aided approached is aspired. In order to enable an automated generation of nonlinear models for simulation of the complete drive train, a library containing all essential elements of the described drive system is introduced. With the help of the modelled components the Airbus A380 flap actuation system has been modelled. Simulating a normal extension cycle, the simulation model provides results that are close to measured data. The verified model is used for analysis of maximum loads when the safety brakes are applied after a shaft rupture.

While the basis for an interface between the design tool WissBaSys and the Modelica environment has been established by the implementation of the presented HighLift library, its execution is still outstanding. Furthermore a simulation of the complete system including the power drive unit as well as the slat flap control computer is necessary.

Acknowledgment

The authors thank the Airbus Deutschland GmbH for sponsoring and supporting this project.

References

[1] Holert, B.: Eine Methode zum mehrkriteriellen Entwurf der Führungsmechanismen in Hochauftriebssystemen von Transportflugzeugen. Hamburg: Dissertation, Institut für Flugzeug-Systemtechnik, TU Hamburg-Harburg, 2005.

[2] Neumann, U.; Holert, B.; Carl, U.B.: Für eine sichere Landung - Simulation von Landeklappenantriebssystemen. Antriebstechnik, 4/2004.

[3] Pelchen, C.;Schweiter, C.; Otter, M.: Modeling and Simulating the Efficiency of Gearboxed and of Planetary Gearboxes. 2nd International Modelica Conference, Oberpfaffenhofen 2002.

[4] Rechter, H.; Richter, M.: Die Simulation als Hilfsmittel bei der Entwicklung und Integration der A330/340-Hochauftriebssysteme. DGLR-Jahrestagung, Band 1, Göttingen 1993.

[5] Runte, W.: YACS: Ein hybrides Framework für Constraint-Solver zur Unterstützung wissensbasierter Konfigurierung. Diplomarbeit, Fachbereich Mathematik / Informatik, Universität Bremen, 2006.

[6] Ruprecht, T.; Thielecke, F; Recksiek, M: SIVA - A testrig for the validation of high lift component models. AST Workshop on Aircraft System Technologies, Hamburg 2007.