KISSsoft
Calculation programs for machine design

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KISSsys application:

Efficiency of a worm gear flap actuator as function of temperature

1 Task

The SABA Flap-Actuator, a worm gear driven ball screw-spindle system, is analyzed by two methods:

- Programmable test bench including variable loads representing actual loads on the wing and programmable temperature from –50°C to +80°C
- Modeling by KISSsys of the total system identifying all major loss-carrying factors

The results show clearly that especially load independent factors correlate strongly with temperature and are non-linear at lower temperatures. KISSsys allows to introduce actual test results into the formulas to create an expert-system; in the present case an expert-system for actuators.
Based on the measurements done in the tests, the basic value for the friction μot – which, according to the DIN3996 is independent of the viscosity of the lubricant – can be expressed as a function of the viscosity of the lubricant. It is the objective of the companies involved, to estimate the mechanical behavior of future actuators more precisely using the simulation tool described here.

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2 Solution

The task to design a flap actuator for a commuter aeroplane in an enlarged temperature range from –50 to +80°C was a challenge for the Sauter Bachmann Company [4]. The SABA flab actuator (for Pilatus PC12 aircraft, see photograph on first page) is essentially a worm gear driven ball screw – spindle system, featuring a self-retaining worm gear design and an input torque absorber which limits the output force of the ball screw.

A fully digitized test rig allows for extensive testing including a temperature range from –50°C to +80°C. The theory of small worm gear sets performs only relatively non-precise estimations of efficiency. The present paper shows that, with precise measuring of input torque at variable loads, the characteristics of a worm gear actuator can be extensively analyzed.

2.1 Description of the actuator, design and properties

The worm gear assembly consists of a hollow worm shaft with hexagonal recesses at both ends, to allow the connection of the actuator to the flex shafts, respectively to a rotation transmitter. The worm wheel is designed as a hollow gear, driving the ball nut unit. The ball nut then transforms the rotation from the gear into an axial movement via the ball screw. A torque absorber is incorporated on the worm shaft and consists of multiple stationary and rotating discs which act as a brake when the torque exceeds the admitted spindle load on the ball screw. Under normal flap-operation loads the torque absorber has no effect.

The gear ratio of the worm gear is 40:1, which, applying the nominal value of 3280 input RPM, relates to a worm gear speed of 82 RPM and therefore to a ball screw axial movement of 5.5 mm/sec. The essential data of the worm gear set are: number of teeth on the worm z1=1, number of teeth on the wheel z=40, normal module m_n=1.1, lead angle of 4.1°. The material of the worm is Cronidur X30, hardened at 58 ± 2HRC, mating with the worm wheel bronze CuSn12Ni. Critical is the wear resistance of the worm gear assembly. Suitable material combination for low temperatures is necessary, as well as, the selection of non-corrosion resistant material for parts being exposed to environmental conditions like the worm shaft. Further, an important feature is the sealing between worm shaft, support bearing and torque absorber to avoid contamination of the worm gear, introducing on the other hand additional friction through the seals on the worm shaft to be taken in account for the efficiency analysis.

2.2 Experimental setup

The test bench functionality is based on the actual use on the aeroplane. The force on the spindle can be programmed depending on the actual position and movement of the spindle. The input torque on the worm shaft is measured in Nm. Load cycles with forces in extending and retracting direction are programmed and directly compared to the resulting input. Both axes (load in N and input torque in Nm) are shown in an x/y diagram to visualize and store the complete duty cycle. Besides using the test bench for acceptance test procedures, two identical test benches can be used as a master-slave system driven by the central power drive unit, using different loads for the inboard and outboard actuator to simulate the actual loads on the wing. Further, both test benches are equipped with nitrogen cooling units programmable down to – 55°C to simulate high altitudes. Life cycle tests are therefore being carried out under extreme cold soak conditions or as an even more realistic scenario, cold and warm cycles can be mixed to come as close to reality as possible. Manual utilization of the test bench is also an option introducing the input torque manually and measuring "static" load points over the complete load cycle.

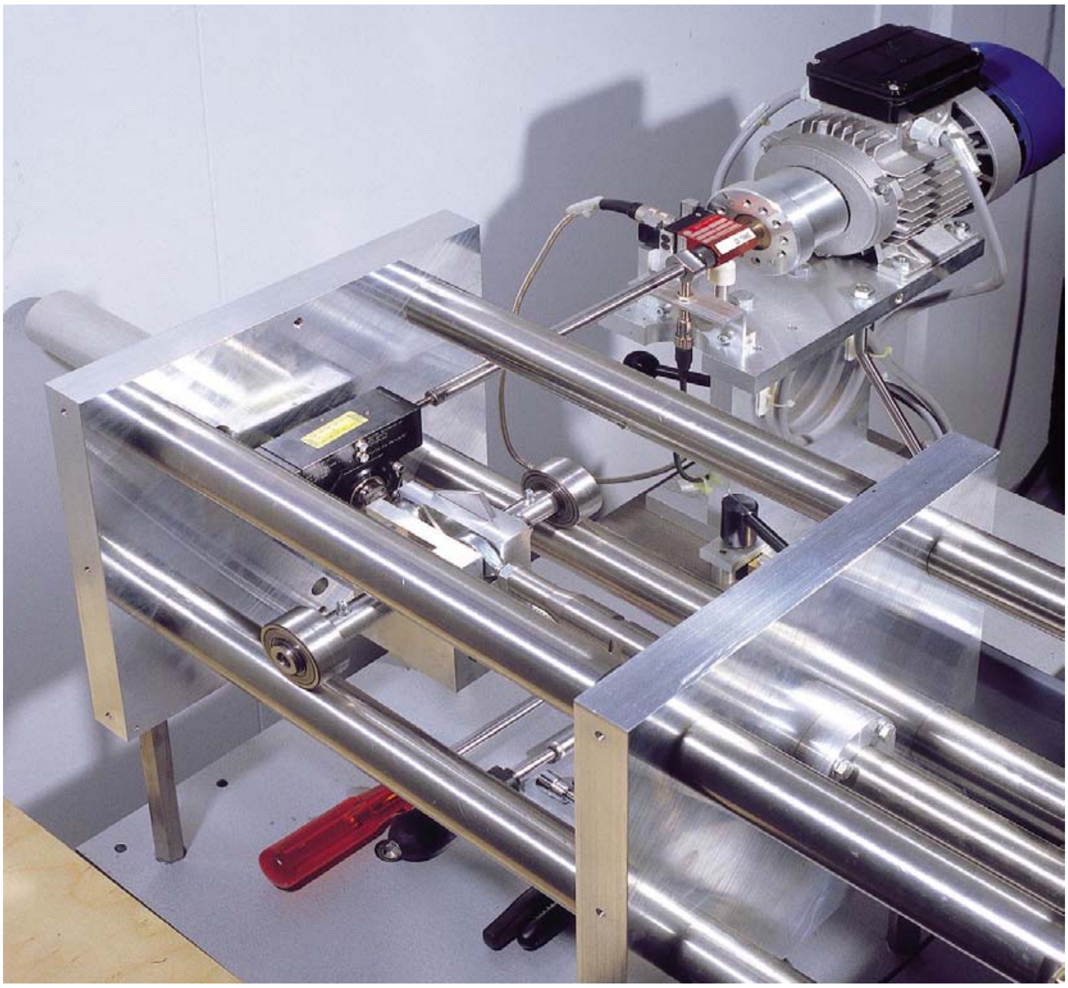

Figure 2.2-1 The flap actuator mounted on the test bench

3 The KISSsys model

3.1 KISSsys: Software for calculation of powertrains and gearboxes

Concept: KISSsys [3] is a program to manage calculations for power transmission systems. It models the total view of a system of machine elements and is capable to perform stress analysis and life time calculations for the machine elements of the system, using the proven KISSsoft calculation program [3]. KISSsys therefore increases the efficiency and security for new designs or re-calculations of old designs. Variations in modeling parameters can be administered within the same framework. KISSsys is highly flexible through the possibility of in-built programming options.

Administration of calculations: The main task of KISSsys is the administration of results from calculations therefore a number of system data (input speed, torque, load, temperatures, etc.) are defined in KISSsys as variables to calculate the stress situation of singular elements. KISSsys defines load data and limiting factors. The kinematics and power flow of a system is administered by KISSsys in a consistent way. A total system can be represented in three-dimensional pictograms.

Administration of variants: Within many designs there are design variations and options which exist in the form of design families. Also the investigation of alternative design solutions is desirable. KISSsys takes in account variations and options of all input data. Therefore, not only design families or different designs can be modeled but also variable loads and constraining factors can be introduced. It is easy to analyze individual variants and optimize those or calculate all variants at the same time; as an example, all ratios of a gear box.

Adaptations through programming: KISSsys uses an adapted programming language. Therefore, the definition of complex applications, for example, definition of gear trains is possible. Furthermore, through special control gates it is possible to analyze whether a design can be fitted into an existing housing or not. Gear design specialists can within their firm define expert systems which in turn can be used by the firm's customer engineers.

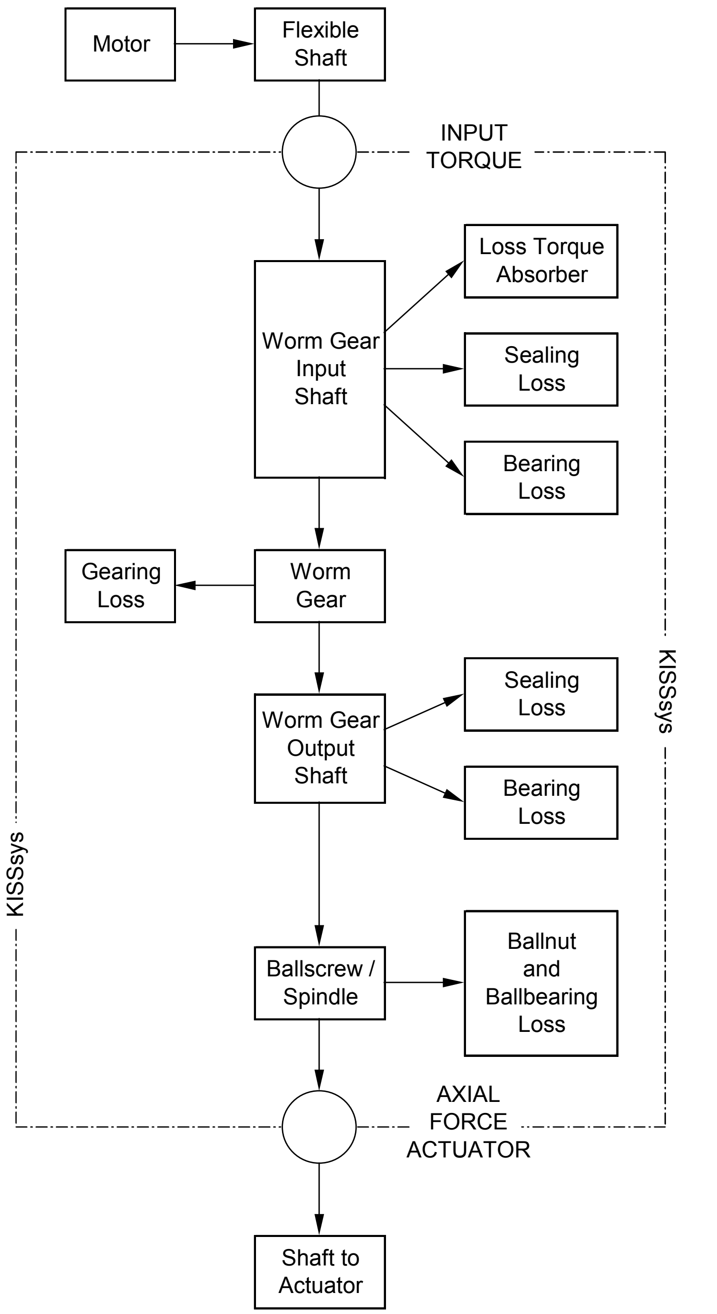

Figure 3.1-1 Analytical model with torque flow and power losses

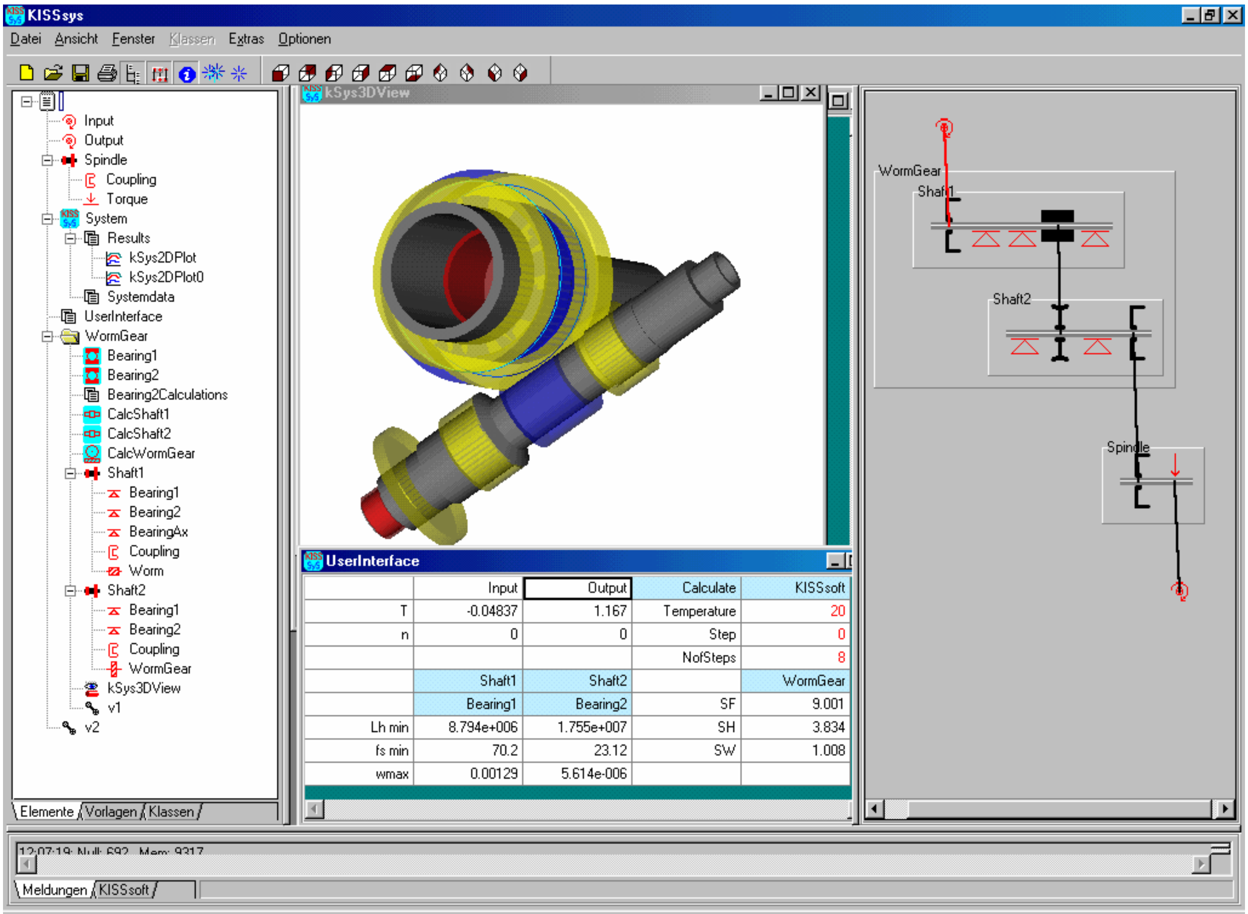

Figure 3.1-2 Modell of the flap actuator, left: tree structure, centre: 3D view, right: scheme of power flow.

3.2 Formulas for calculation of strength, lifetime and efficiency

Input data for the calculation are given through the actual load cycles measured on the plane. The load cycle (Figure 3.2-1) shows the time-load curve on the ball screw. This is used to calculate the necessary torque on the worm. The calculation model has two tasks:

- It calculates the efficiency of all components of the system at temperatures between –50 and +80°C.
- It verifies the resistance over time of the individual elements (worm gear set, bearings, shafts, ball screw).

Using the approach of a given actual force curve over time and working-path, the input torque can be determined. The calculation of the worm gear set (worm/worm gear) is based on DIN 3996 [1]. This is a new norm which is based on the most detailed known method for worms. The bearings are calculated according to the formulas of the manufacturers whereby an enlarged life cycle is used including load collective integrals. Bearing efficiency or bearing losses are calculated with simple formulas used by the manufacturers although ISO/TR 14179 [2] offers a more differentiated calculation method. The linear ball screw calculation for efficiency and load resistance is calculated to a formula of the manufacturer; no useable basis has been found in the literature. Data for the applied grease ISOFLEX TOPAS NCA 51 was provided by Klüber Lubrification.

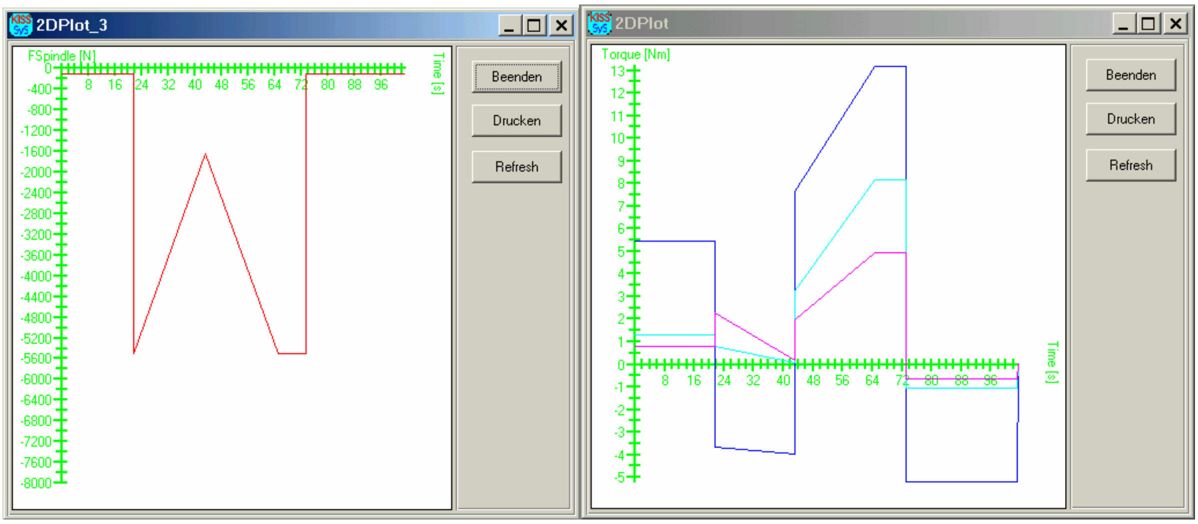

Figure 3.2-1 **Left: Load function on the ball screw (axial force on actuator during start and landing)**

Right: Torque on the ball screw (red), worm gear (light blue) and input shaft (multiplied by i=40) (dark blue) during the load cycle

3.3 Calculation of characteristic parameters for the simulation on the test bench

The test bench uses simplified load cycle, increasing the axial load during a time of 30 seconds from 1650N to 6800N and then, for a further 30 seconds a reduction back to 1650N. The measuring system plots the values, whereby the X-axis shows the actual force on the ball screw and the Y-axis the input torque on the worm.

KISSsys can model the axial force as a function over time. The calculation is done automatically for pre-defined time slots. KISSsys diagram-use can be configured in such a way that identical views as on the test bench are seen. Therefore, the calculations of the model show exactly the measured load cycles on the test bench.

In the User-Interface-Table (Figure 3.1-2) by KISSsys, the most important input and output values are presented in the calculation model. In our case as follows: the bearing life cycles, worm gear set security factor and the different loss portions which can be identified on the test bench through measurements directly. Further, temperature can be introduced as a function with parameters. This is important, because in the basic formulas the influence by kinematic viscosity of the lubrication (at the actual temperature) is not represented.

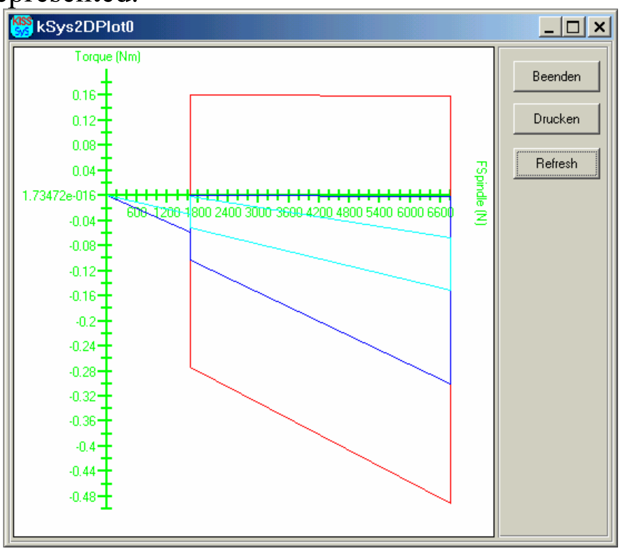

Figure 3.3-1 Calculation of the duty cycle on the test rig with KISSsys at 20°C. Represented is the result in the same configuration as drawn by the test rig plotter (axial force on the horizontal axis, input torque on vertical axis).

Red line:	**Input torque**
Dark blue line:	**Torque on the worm gear (divided by worm gear ratio of 40 to represent the torques within the same diagram-frame)**
Light blue line:	**Torque on the spindle (divided by 40)**

The torque-difference between the different lines shows the losses.

3.4 Comparison between test bench measurements and modeling calculation

Based on the set-up of the test bench the influence of some elements resulting in the total efficiency can be measured individually. Included are:

- loss by the torque absorber
- loss through the seals on the worm shaft
- loss through the seals on the worm gear
- total loss unloaded

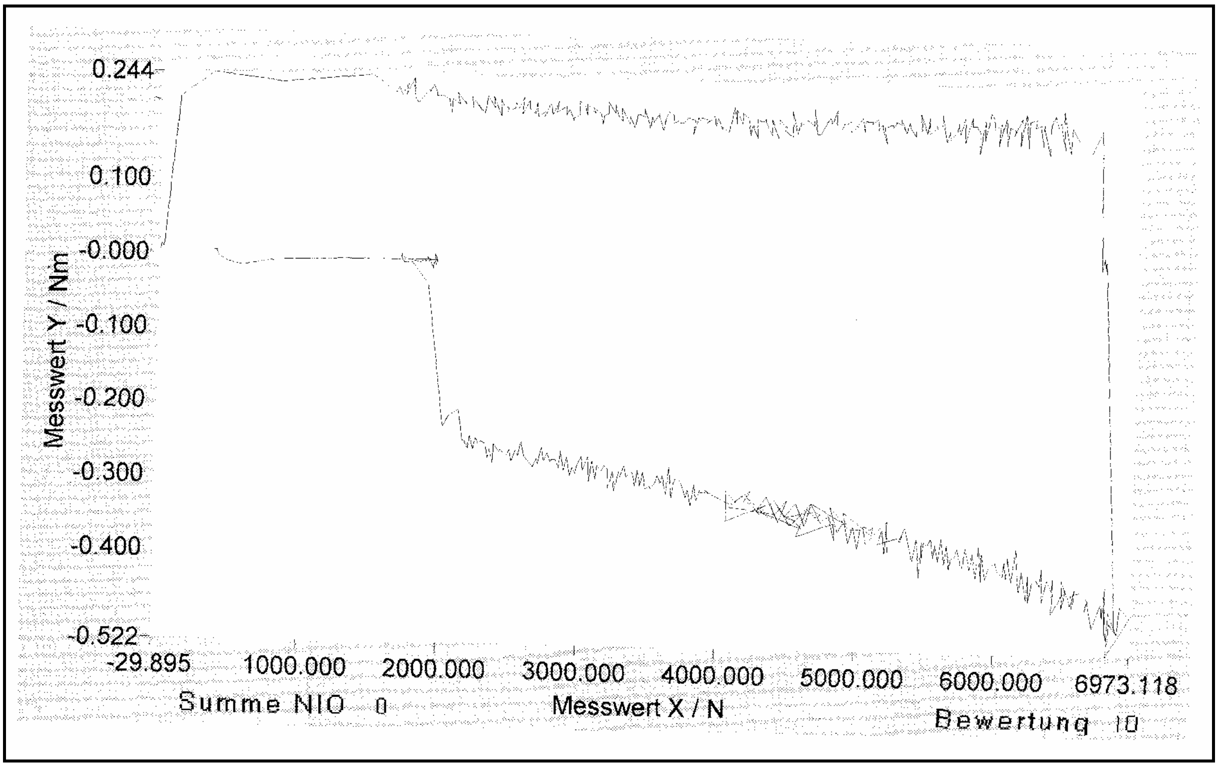

Figure 3.4-1 Measurements at 20°C

These losses, as they are not very well documented in literature, can be introduced directly into the calculation model. Figure 3.4-1 shows test bench results at ambient temperature (20°C). The measurements and test bench results correlate with above shown calculations and show good correspondence for absolute values as well as good quality of load curve correlation. Test bench results compared with the calculation (including the Viscosity Factor as discussed below) in the temperature range between –50° and 70°C showed a maximum discrepancy of 11%, which is a surprisingly good result!

3.5 The influence of the temperature on the efficiency of worm gears

The norm DIN 3996 [1] defines the calculation of gear efficiency and a number of loss factors (bearings, seals, no-load losses). Calculations, using identical lubrication at different temperatures (e.g. 20°C or 70°C), show identical results. Therefore the norm does not show efficiency differences due to temperature. Also, oil viscosity, including big differences in kinematic viscosity (e.g. 32 or 680 mm²/s), show no difference. This is surprising and shows that the viscosity influence due to temperature change has been neglected so far probably due to lack of systematic research.
To measure the efficiency of the worm in the most direct way, test measurements are executed without seals and without torque absorber. Therefore besides the worm, the only remaining loss-carrying elements are the bearings and the ball screw.

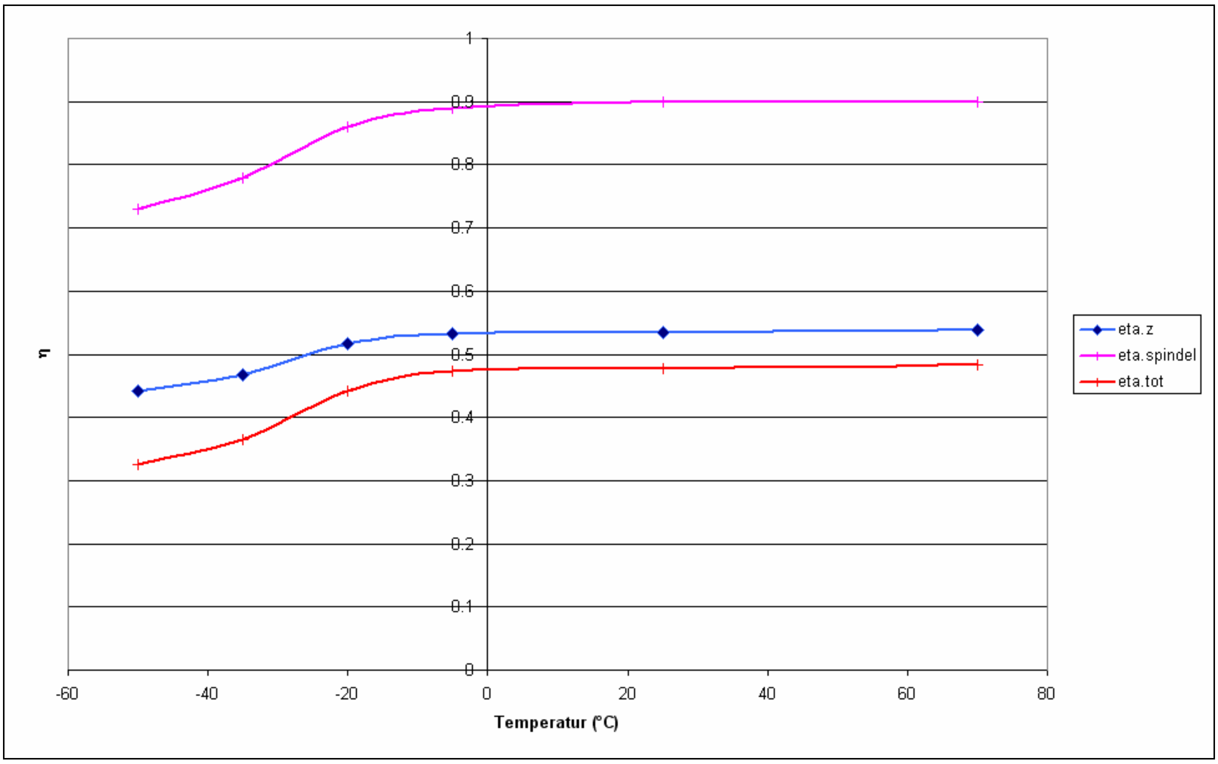

Figure 3.5-1 Efficiency (without seals): Total: ηtot (measured); Spindle: ηspindle (from data sheet); Worm Gear: ηz (calculated)

The figure above shows the efficiency of the actuator depending on temperature. On the test rig it is not possible to measure separately the torque loss on the linear spindle and the worm gear. So the curve ηtot shows the efficiency measured of the combination spindle plus worm gear. The spindle manufacturer supplied values for the efficiency of the spindle. Therefore it is possible to deduce the worm gear efficiency ηz.

The calculation of the gear efficiency ηz is based on the known formula [1]:

$$\eta z = (\tan \gamma m) / (\tan (\gamma m + \text{arc tan } \mu zm))$$

The averaged friction coefficient μzm according to DIN3996 [1] is calculated from the base friction μot :

$$\mu zm = \mu ot * Y_S * Y_G * Y_W * Y_R$$

The factors Y_S (size factor), Y_G (geometry factor), Y_W (material factor) and Y_R (surface factor) are dependent on geometry, materials and surface quality but without connection to the lubrication. For the actual investigations these factors remain fixed. The variation of the total efficiency as a function of the temperature (through the different lubrication viscosity) is referred to μot. Based on the test results the gear efficiency is determined. Therefore the gear friction µzm and the base friction µot is determined (table 1). To compare: µot = 0.058 following DIN 3996, independent of temperature or viscosity.

Temperature (°C)	-50	-35	-20	-5	25	70
ηtot	0,325	0,365	0,442	0,473	0,477	0,484
ηspindle	0,730	0,780	0,806	0,890	0,900	0,900
ηz	0,441	0,468	0,516	0,532	0,534	0,538
µzm	0,090	0,081	0,067	0,063	0,062	0,061
µot	0,069	0,062	0,051	0,048	0,047	0,047
ν (mm²/s)	30000	5000	1020	310	57	11
Y_{VISC}	1,46	1,31	1,08	1,01	1,00	1,00

Table 3.5-1 Deduction of the base friction µot at different temperatures. Proposed viscosity factor YVISC depending on the kinematic viscosity of the lubricant.

It is not surprising that the base friction coefficient depends on the viscosity of the lubricant. This coefficient increases if the kinematical viscosity is higher than 500 mm²/s. To consider this phenomena, a Viscosity Factor Y_{VISC} should be introduced in the formula for µzm.

$$\mu zm = \mu ot * Y_S * Y_G * Y_W * Y_R * Y_{VISC}$$

Figure 3.5-2shows this factor, which is proposed for worm gear efficiency calculations in low temperature range.

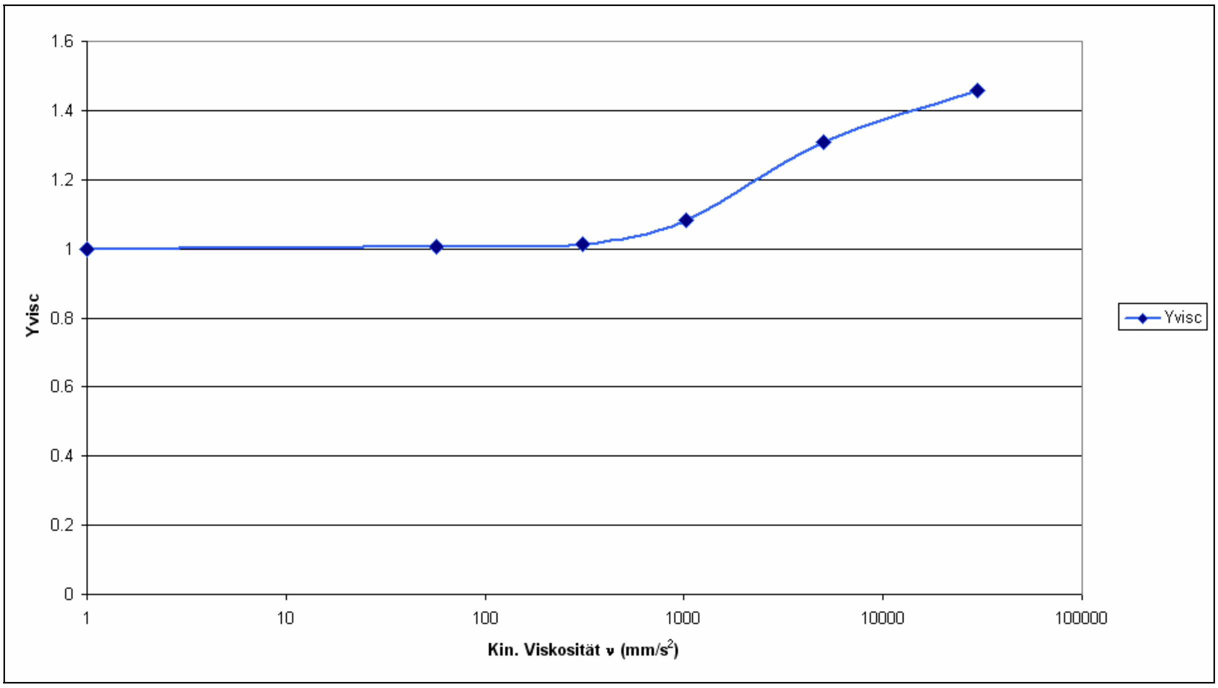

Figure 3.5-2 Viscosity factor Y_{VISC}

4 Conclusion

The modeling of the SABA flap actuator within KISSsys on the basis of formulas used in norms and manufacturer indications are compared to measurements on the test bench. A deeper analysis of the theoretical basis shows that the dependence on temperature of loss components are not enough represented. As to be expected, the comparison of measurements and calculations show that especially non-load depending factors of losses correlate stronger with temperature within the window of –50°C to +80°C and the influences of temperature are stronger and especially non-linear in lower temperature fields.

Based on the measurements on the test bench the base friction µot as a function of viscosity (to DIN3996) can be shown and integrated into the calculation formulas.
The calculation model allows those factors of losses which can be directly determined on the test bench to be introduced as a function of temperature. This leads closer to the aim of a model which shows the actual behavior of a system and especially allows to predict security factors of a system in extreme temperature ranges. The model therefore can be used to predict behavior of similar actuators for other load and temperature profiles and it can also be used to determine and analyze design changes to improve functional behavior.

5 Bibliograph

[1] DIN 3996, Calculation of load capacity of cylindrical Worm Gear Pairs, 1998.
[2] ISO/TR 14179-2, Gears - Thermal load-carrying capacity, 2001.
[3] Additional information on www.KISSsoft.ch
[4] Additional information on www.sauterbachmann.ch