HCM University of Technology

Machine Elements

ME2007

Lab Report

Submitted To:

Le Thanh Long

PhD

Machine Design Dept.

Submitted By:

Nguyen Quy Khoi

Phan Quang Minh

Vuong Tuan Quan

Nguyen Hoang Duc Minh

Nguyen Bao Phien

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Chapter 1

Slip coefficient and Slip curve of Belt drive

1.1 Nomenclature

a	center distance, mm	n	rotational speed, rpm
f	coefficient of friction	Q	load, $kg \cdot F$
d	diameter, mm	α	wrap angle, $^\circ$
F_0	initial tension, N	β	slack angle due to load Q, $^\circ$
F_{ms}	friction force, N	Δh	difference between h_i and h_f, mm
F_t	tangential force, N	ϕ	drag coefficient
g	gravitational acceleration at sea level, m/s^2	ϕ_{crit}	critical drag coefficient
		$\bar{\xi}$	average slip coefficient
h_f	distance between outer sides of the belt after applying load Q, mm	$\tilde{\xi}$	relative slip coefficient
		ξ	slip coefficient
h_i	distance between outer sides of the belt before applying load Q, mm	$_1$	subscript for driving pulley
		$_2$	subscript for driven pulley

1.2 Aim

1. Investigating on the slip phenomenon of belt drive.

2. $\tilde{\xi}$ and ξ determination.

3. Determining F_0.

4. Drawing slip curve - load diagram.

1.3 Technical rules on safety

Students must comply with the technical rules on safety in the laboratory.

1.4 Conducting and dealing with experimental results

1.4.1 Determine the parameters for the experimental model

- Diameters of the pulley:

  $d_1 = 67.8\,\text{mm}$

  $d_2 = 165\,\text{mm}$

- Belt type: flat belt.

- Rotational speed: see table (1.1).

- Contact angles:

  $\alpha_1 = 180 - 57 \times \dfrac{d_2 - d_1}{a} \approx 162.3°$

  $\alpha_2 = 180 + 57 \times \dfrac{d_2 - d_1}{a} \approx 197.6°$

- Initial tension:

  $h_i = 124\,(\text{mm})$, $h_f = 94\,(\text{mm})$, $Q = 4.1\,(\text{kg} \cdot \text{F})$

  $\Delta_h = |h_f - f_i| = 30\,(\text{mm})$, $\beta = \arctan \dfrac{2\Delta_h}{a} \approx 10.78°$

$$F_0 = \frac{Qg}{2\sin\beta} \approx 107.48\ (\mathrm{N})$$

- Tangential force: see table (1.1).

1.4.2 Measure and deal with the measured results of F_0

1.4.3 Measure and deal with the measured results in order to determine the $\bar{\xi}$ and ϕ.

Filling out the measured results in table 1.1 and calculate the coefficients. Using the formulas $\xi = 1 - \dfrac{d_2 n_2}{d_1 n_1}$ and $\phi = \dfrac{F_t}{2F_0}$, we obtain the following table:

No.	F_0 (N)	n_1 (rpm)	n_2 (rpm)	ξ	F_t (N)	ϕ
1	107.48	283.62	114.04	0.018	3.1	0.014
2	107.48	330.47	133.35	0.018	8.8	0.041
3	107.48	273.83	110.27	0.02	14.4	0.067
4	107.48	307.52	123.71	0.021	20.2	0.094
5	107.48	354.42	142.43	0.022	22.1	0.103

Table 1.1: Experimental results of the slip coefficient

Averaging the values of ξ yields $\bar{\xi} \approx 0.0198$. From the data above, we can approximate the best fitted line through the data points (assuming linearity since ϕ does not reach critical value)

1.5 Discussion and conclusions

In summary:

- Slip coefficient from experiment is in allowable range $(0.01 \div 0.02)$.

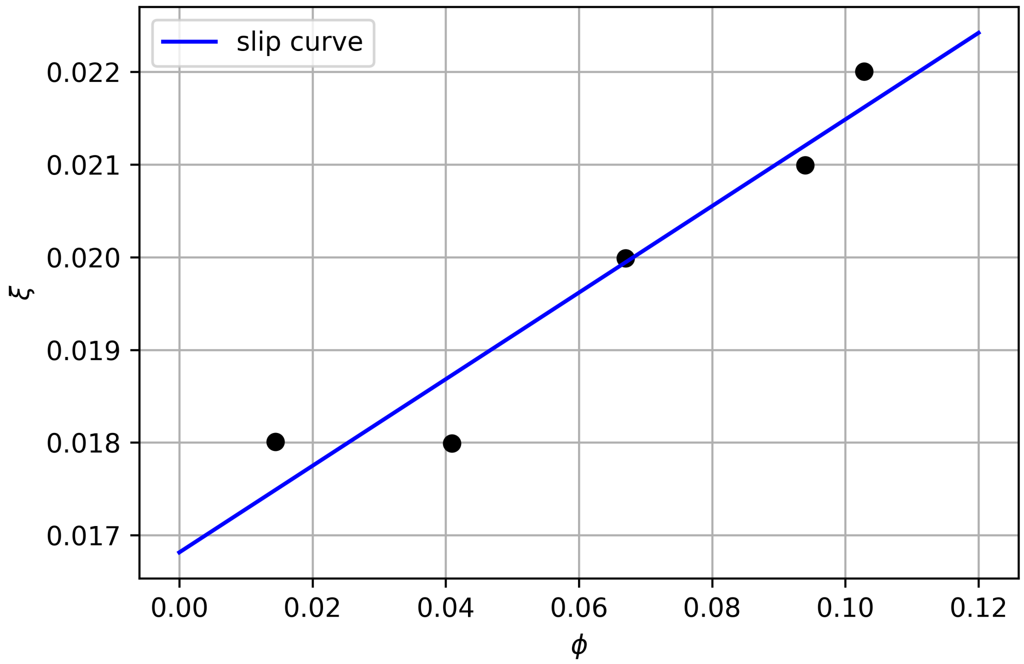

Figure 1.1: The slip curve is established by the experimental data.

- The slip curve is in agreement with theory (error is smaller than 5%). Since ϕ does not exceed critical value (the motor is frequency-controlled), we can safely assume linearity for the curve.
- Possible errors:
  - manually measure dimensions in the kit.
  - rounding.
  - incorrect reading of rotational speeds.
- The slip coefficient and slip curve is considerably accurate due to reliable instrument

1.6 Review questions

1. *Define the types of slip in the belt drive.*

   There are 3 types of slip:

   - **Geometry slip**, which depends on the geometrical profile of the cross section of belts. For flat belts, geometry slip is nonexistent while V-belt does. However, this type of slip does not have much effect on impeding the movement of belts, hence is often disregarded in calculations.
   - **Creep**, whose cause is the elasticity of belt materials. Its value is often small (around 0.01 to 0.03), which is negligible. As a result, the velocity ratio of belt drive depends on the transmission power of the system.
   - **Slip** happens due to overload. This type of slip has a negative effect on the belt drive that inhibits velocity transmission from the driving pulley to the driven pulley. Thus, we have to take into consideration slip while designing a belt drive. In addition, elasticity of belt material also contributes to slip, though the amount is insignificant (around 0.01 to 0.03).

2. *The method determining the slip coefficient in the belt drive.*

   Given the diameters d_1 and d_2 along with the rotational speeds n_1 and n_2, we calculate the slip coefficient using equation:

$$\xi = 1 - \frac{d_2 n_2}{d_1 n_1}$$

3. *The relationship between F_t and F_0.*

From theory we have learned,

$$\begin{cases} F_0 &= F_1 - \frac{F_t}{2} \\ F_0 &= F_2 + \frac{F_t}{2} \end{cases}$$

Where F_1 is the tension on the tight side, N

F_2 is the tension on the slack side, N

4. *Present the formula determining ϕ and the ϕ_{crit} of the kinds of belt drives.*

Chapter 2

Determination of Tensile Force of Bolts

2.1 Nomenclature

$[F_{cb}]$	tension force at failure of common bolt, N
$[F_{sb}]$	tension force at failure of steel bolt, N
$[\sigma_{cb}]$	tension at failure of common bolt, MPa
$[\sigma_{sb}]$	tension at failure of steel bolt, MPa
d	nominal diameter of M8 bolt, mm
F_c	tension force of hydraulic cylinder
F_{cb}	tension force at failure of common bolt, N
F_{sb}	tension force at failure of steel bolt, N

2.2 Aim

1. Help students understand more clearly about tensile force of some types of steels, the relation between central M_k and localized strain of material.
2. Help students approach methods, measuring devices in determining tensile force.

2.3 Safety Procedures

1. Safeguard is compulsory when straining the bolt.
2. Close the machine gate when operating.

2.4 Experimental Report

No.	Experiment with d = 8 mm	
	F_{sb}	F_{cb}
1	33898	37377
2	33574	37053
3	34211	36426
4	33727	37053
5	34211	36426
Avg	33323.4	36867

Table 2.1: Tension force at failure of common and steel bolts

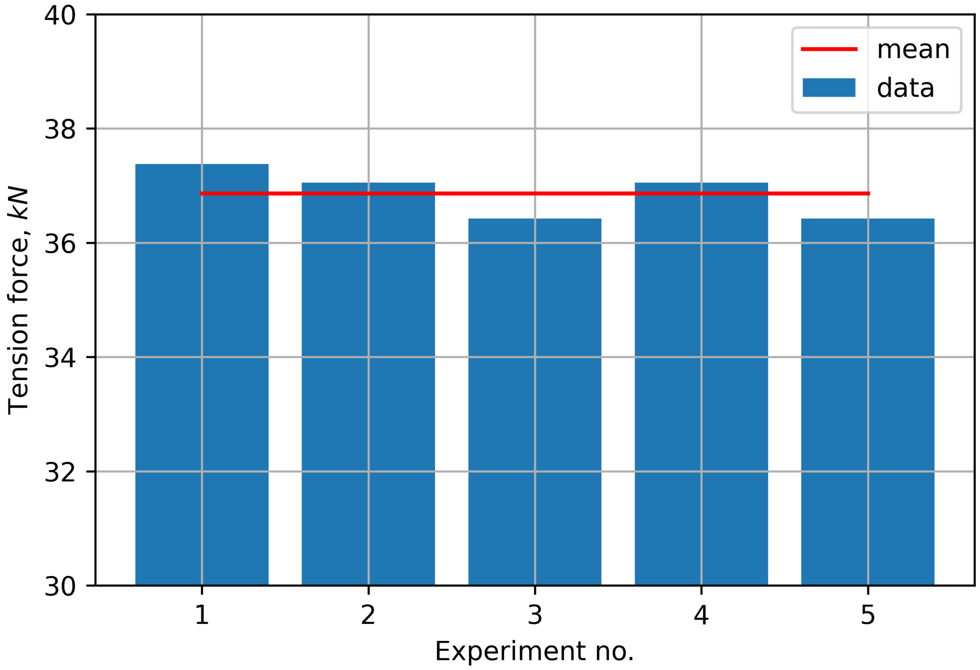

Figure 2.1: Tension force at failure of common bolt

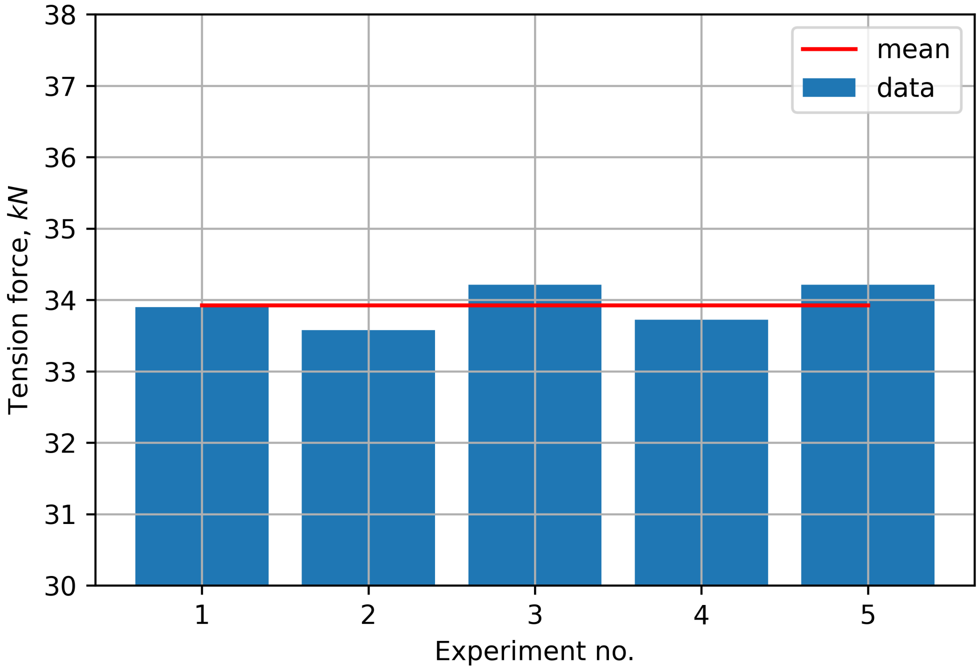

Figure 2.2: Tension force at failure of steel bolt

2.5 Discussion and conclusions

Based on the data and graphs, it can be concluded that steel bolts are more durable than common bolts.

Through 5 experiments conducted on each type of bolt, we saw that in spite of being the same type, each experiment gave different results, which could be explained from minor dissimilarities in shape and internal structure as a consequence of:

- Errors during the designing and machining.
- Environmental effects from transportation, preservation, climate, etc.
- Difference in microscopic level due to manufacturing: defects, varying temperature, mold structure, cooling rate, etc.
- Unreliable testing.

In summary, we can see that the fracture point of each individual bolt is different but not too much. This feat is achieved by modern and increasingly accurate manufacturing techniques.

Chapter 3

Determination of Tightening Coefficient of Threaded Joints

3.1 Nomenclature

D_0	outer diameter of the nut contacted with bolt, mm
d	nominal diameter of the bolt, mm
d_0	diameter of the bolt hole, mm
d_2	thread diameter, mm
F_v	tightening force, N
f	friction coefficient between nut and assemble part
K	tightening coefficient
K_{exp}	tightening coefficient through experimentation
K_{theo}	theoretical tightening coefficient
p	thread height, mm
T_r	force moment acting on thread, $\mathrm{N} \cdot \mathrm{m}$
T_v	tightening moment, $\mathrm{N} \cdot \mathrm{m}$
δ_K	error between K_{exp} and K_{theo}
γ	back rake angle of thread, °
ρ'	the friction angle on the thread surface, °

3.2 Aim

1. Understand deeply the theory of screw coupling.
2. Use the wrench to determine T_v.
3. Understand the principle and use load cell to measure the F_v on the bolt;
4. Determine $K_{exp} = \frac{1000T_r}{F_v d} = \frac{1}{F_v d}\left[1000T_v - \frac{1}{2}\left(\frac{D_0 + d_0}{2}\right)F_v f\right]$ and understand the relationship between T_v and F_v, as well as the factors of assembling condition of the joint.

3.3 Technical rules on safety

Students must comply with the technical rules on safety in the laboratory.

3.4 Experimental report

3.4.1 Determine the parameters of threaded joint and measuring tools

- $d_0 = 13.8$ (mm)
- $D_0 = 18.8$ (mm)
- $p = 1.49$ (mm)
- $d_2 = \frac{p}{\pi \tan \gamma} = 10.86$ (mm)
- $\gamma = 2.5°$
- $\rho' = 10°$
- $f = 0.25$

3.4.2 Experimental results

No.	Nominal diameter $d = 12$ (mm)		
	T_v (N · mm)	F_v (N)	K_{exp}
1	16.5	2875	0.31
2	17.6	3243	0.28
3	19.1	3938	0.23
4	22.4	4872	0.21
5	27.9	5229	0.27
Average value			0.26

Table 3.1: The experimental results

3.4.3 Compare theoretical results with experimental results

The theoretical result K_{theo} is determined by the formula

$$K_{theo} = \frac{1000T_r}{F_v d} = 0.5\left(\frac{d_2}{d}\right)\left[\left(\frac{d_0 + D_0}{2d_2}\right) f + \tan(\gamma + \rho')\right] = 0.27$$

Comparing K_{exp} and K_{theo} yields:

$$\delta_K = \frac{|K_{exp} - K_{theo}|}{K_{theo}} = 2.75\%$$

3.5 Discussion and conclusions

- From the graph, the tightening coefficient and tightening moment have a linear relation.
- The error of K in the experiment compares to its theoretical counterpart is

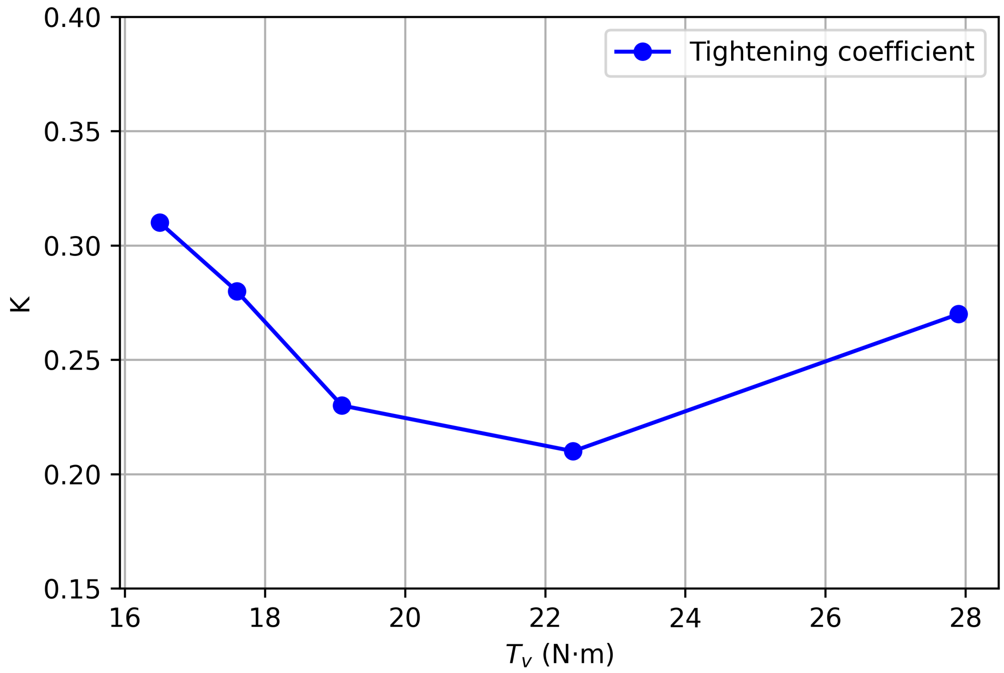

Figure 3.1: Relation between T_v and K_{exp} through experimentation

negligible (±2.75%)

- Possible causes of error:
  - The bolt is not tightened enough, reducing the value of measured tightening moment.
  - Worn condition of the bolt affects the friction coefficient and thread angle.
  - Error of the lab instrument (instability in displaying results), thus the measured values are only relative.
  - Rounding errors through calculations.

3.6 Review questions

1. *Show the role and importance of finding F_v and T_v in reality.*
   Typically, an under torqued bolt will deform and be unable to provide as much clamping force as needed. An over torqued bolt will break.

2. *Explain the meaning of K.*
   K is the synthesis of all the factors that affect the relationship between tightening moment and tightening force in reality, including friction, bending, elastic deformation of thread and many other factors that we may or may not already know. Therefore, it is difficult to determine accurately the tightening coefficient. It can only be determined relatively be experiments for each specific application. Typically, in a specific application, a value range of K is usually determined to predict the maximum and minimum values of the tightening force. Then, the initial tightening moment value is determined

3. *Explain the principles of ultrasonic operation and force measuring tool.*

   - Ultrasonic sensors emit short, high-frequency sound pulses at regular intervals. These propagate in the air at the velocity of sound. If they strike an object, then they are reflected as echo signals to the sensor, which itself computes the distance to the target based on the timespan between emitting the signal and receiving the echo.
   - As the distance to an object is determined by measuring the time of flight and not by the intensity of the sound, ultrasonic sensors are excellent at suppressing background interference.
   - Virtually all materials which reflect sound can be detected, regardless of their color. Even transparent materials or thin foils represent no

problem for an ultrasonic sensor.

4. *Determine K according to the theory of screw coupling.*

   The tightening factor K is the experimental coefficient. In comparison with the theory of screw coupling, we have the following relationship:

$$K_{theo} = \frac{1000T_r}{F_v d} = 0.5\left(\frac{d_2}{d}\right)\left[\left(\frac{d_0 + D_0}{2d_2}\right) f + \tan(\gamma + \rho')\right]$$

   The parameters are defined in nomenclature.

5. *Compare K in cases of with and without lubrication to the joints. Then draw your conclusions.*

   In this experiment, three types of oil, grease, and solid film lubricants are investigated for their effect on the friction and torque-tension relationship in threaded fastener applications. The nut factor, the coefficients of thread and under-head friction were obtained from the experiments. The effect of the number of tightening and loosening cycles, the tightening speed and the lubricants on friction and nut factor were investigated. It was found that lubrication had a significant effect on the friction and the torque-tension relationship in threaded fasteners.

   Under the dry-and-cleaned condition, the nut factor for coarse threads is 0.181 and 0.1827 for fine threads during the first tightening. Comparing the three types of lubricants investigated, it can be seen that the solid film lubricants have the smallest thread friction and under-head bearing friction. Therefore, the same amount of input torque will generate a higher clamping force, which means lower nut factors. The greases and the oils have very similar friction behavior. However, for the low tightening speed case, the greases produce lower friction than the oils do.

Chapter 4

Determination of External Force Coefficient of External Threaded Joints

4.1 Nomenclature

A	area of the raw section, mm^2
F	applied force, N
F_H	horizontal component of F, N
F_V	vertical component of F, N
$J_{x'x'}$	moment of inertia along XX-axis, $kg \cdot mm^2$
k	safety factor
V	tightening force, N
V_{max}	maximum tightening force, N
W	bending moment of the raw section, $N \cdot mm$
y_{max}	maximum distance, mm
z	a coefficient
α	contact angle, $^\circ$
χ	external force coefficient
$\bar{\chi}$	average value of χ
ΔF	force difference between each experiment, N

4.2 Aim

1. Help students understand clearly about the method of determination of the external force coefficient by theory.
2. Help students calculate the tightening force in the case of force acting in any direction.
3. Help students approach to the methods, instruments and determine the tightening force, deal with the experimental results to determine the external force coefficient.

4.3 Technical rules on safety

Students must comply with the technical rules on safety in the laboratory.

4.4 Experimental report

Each group conducts the experiment with the given α and F

$\alpha = -5^\circ$

$F = 303.6$ (N)

$\Delta F = 15$ (N)

4.4.1 Theoretical calculation of the external force coefficient

Measuring the size of bolts and assembled details to determine the external force coefficient by using theory.

4.4.2 Calculating V

$$V = \frac{k}{z}\left(F_V + \frac{MA}{W}\right) = \frac{k}{z}\left(F_V + \frac{MA_{yc}}{J}\right)(1-\chi) \tag{4.1}$$

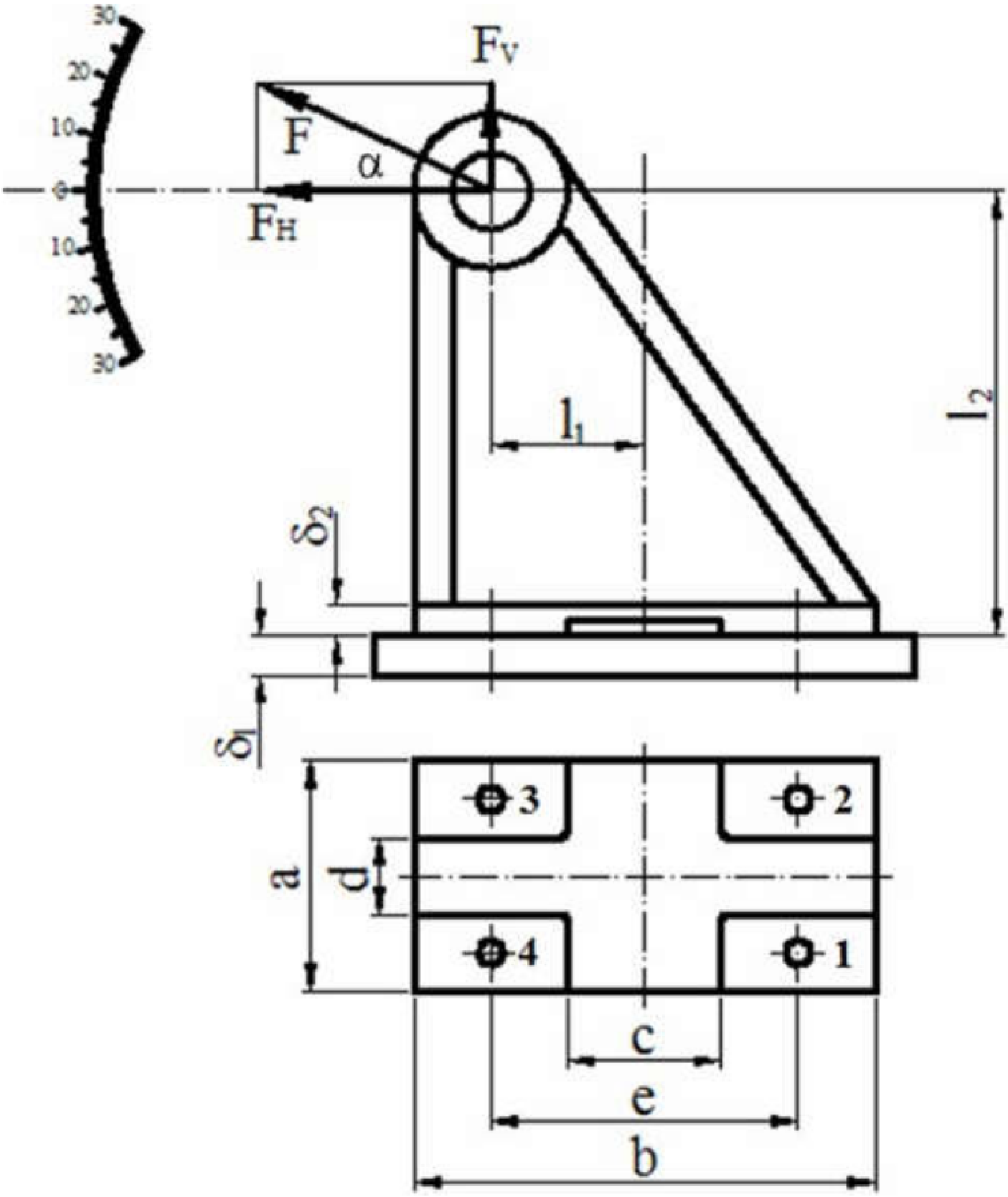

Figure 4.1: Model of experimental calculation

$$V = \frac{kF_H + (1-\chi) f F_V}{f_z} \tag{4.2}$$

According to formulas (4.1) and (4.2), selecting V max for these two values.
Note: The tightening force for surfaces not separated is defined by the formula:

$$V = \frac{k}{z}\left(\frac{(F_V l_1 \pm F_H l_2) A y_{max}}{J_{x'x'}} - F_V\right)(1-\chi)$$

where $J_{x'x'} = \dfrac{(a-b)(b^3-c^3)}{12}$

$A = (a-d)(b-c)$

$y_{max} = \dfrac{b}{2}$

Tightening the bolt with the tightening force $V = V_{max}$ (using equations 4.1 and 4.2) and checking by measuring device.

4.4.3 Measured results and process

We increase the load by hydraulic cylinder 5 to reach values: $F_1, F_2, ..., F_N$. They occur on the display screen (these values are less than F) and fill in column 2 of Table 4.1. The values $F_i = F - i\Delta F$.
Write down the value of tightening moment, tightening force V_{tni} by load cell method and put them into columns 3, 4 of Tables 4.1.

4.4.4 Calculate the external force

Calculate the following values:

1. $F_{Vi} = F_i \sin\alpha$

2. $F_{Hi} = F_i \cos\alpha$

3. $M_i = F_{Hi} l_1 \pm F_{Vi} l_2$

and put these values into column 5, 6 of table 4.1.

In this experiment, $l_2 = 0$ and $Y_i = \frac{e}{2}$, therefore $M_i = F_{Hi}l_1$

$$V_{tni} = V_0 + \chi\left(\frac{F_{Vi}}{z} + \frac{M_1Y_1}{\sum z_iY_i^2}\right) = V_0 + \chi\left(\frac{F_{Vi}}{z} + \frac{F_{Hi}l_1 \pm F_{Vi}l_2}{2e}\right)$$

where $i = 1, 2, ..., N$

Here, χ is determined by the formula:

$$\chi_i = \frac{V_{tni} - V_{tn1}}{\frac{F_{Vi} - F_{V1}}{z} + \frac{(F_{Hi} - F_{H1})l_1 + (F_{Vi} - F_{H1})l_2}{2e}}$$

According to the experimental model, $z = 4$, $e = 200$ (mm), $l_1 = 300$ (mm), $l_2 = 100$ (mm) and then write down the results into column 7 of table 4.1.

Therefore, the average value of external force coefficient through N measuring times 18305.42 (Pa · s)

$$\bar{\chi} = \frac{\chi_1 + \chi_2 + ... + \chi_{N-1}}{N - 1} \tag{4.3}$$

From the experimental results, plot the curve illustrated the relationship between χ_i and F_i.

4.5 Discussion and conclusions

Comparing the theoretical results with experimental results and then drawing conclusions. Possible causes of error:

- Worn condition of the bolt affects the friction coefficient and thread angle.
- Error of the lab instrument (instability in displaying results), thus the

No.	F_i (N)	V_{tni} (N)	F_{Vi} (N)	F_{Hi} (N)	χ_i
1	303.2	3025	-26.43	302.05	0
2	287.8	3097	-25.08	286.71	0.12
3	271.4	3221	-23.65	270.37	0.23
4	257.8	3304	-22.47	256.82	0.16
5	242.8	3426	-21.16	241.88	0.25
6	227.8	3525	-19.85	226.93	0.22

Table 4.1: The experimental results

measured values are only relative.

- Rounding errors through calculations.

4.6 Review questions

1. *The role and importance of determining the tightening force and the tightening moment in reality.*
   As mentioned, an under torqued bolt will deform and be unable to provide as much clamping force as needed. An over torqued bolt will break.

2. *The meaning of χ and determining this coefficient by the fundamental theory.*
   When the joint is subjected to the external force in the non-sealed surface limitation of assemble plates, the bolt is elongated about Δl in length, the compression deformation of the assemble plates also reduces the same length. This means that only a part of external force $F(\chi)$ acting on a bolt causes the extension. The other part of external force $(1 - \chi)F$ causes the decrease of the compression deformation of the assemble plates. χ is

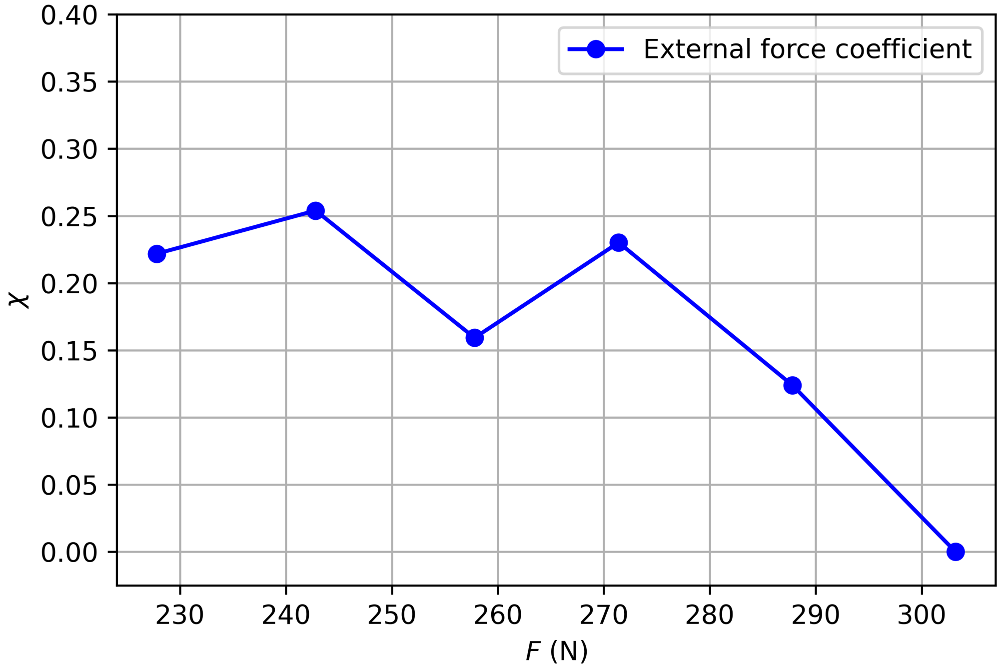

Figure 4.2: Relation between F and χ

called the external force coefficient . Therefore:

$$\Delta l = \chi F \lambda_b = (1-\chi) F \lambda_m$$

Where λ_b, λ_m are the flexibility of bolt and assembled parts, respectively. As a result:

$$\chi = \frac{\lambda_m}{\lambda_b + \lambda_m}$$

3. *Determining the required tightening force of the bolt to avoid segregation and slip.*

For assembled machine element avoid segregation and no slip, the bolt needs to be tightened with the tightening force V:

$$V = \frac{k}{z}\left(F_V + \frac{MA}{W}\right) = \frac{k}{z}\left(F_V + \frac{MA_{yc}}{J}\right)(1-\chi)$$

4. *Comparing the tightening coefficient in the cases of the joint with and without lubrication, and then drawing conclusions.*

   With lubrication, the tightening coefficient will decrease and vice versa. Therefore, it can be concluded that the amount of lubrication and the tightening coefficient has an inverse relationship.

Chapter 5

Applying Computer Software in Calculating Machine Elements

5.1 Aim

Helping students understand the method, how to use the design software to select and test the general machine elements.

5.2 Technical rules on safety

Students must comply with the technical rules on safety in the laboratory.

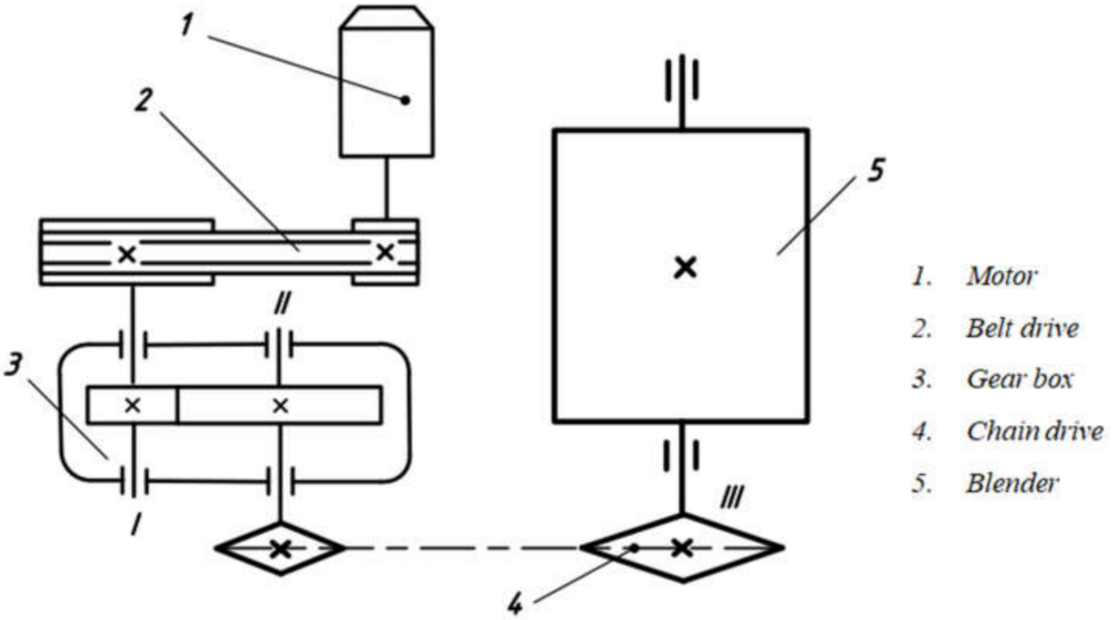

Figure 5.1: Transmission system for the blender

5.3 Experimental report

5.3.1 Problem

Given the transmission system as shown in below figure.

Initial data

- Capacity of blender: $P_3 = 4$ (kW)
- Number of revolution of blender: n_3 (rpm)
- Service life cycle: $L_h = 8$ (years)
- 1-way rotation, work 2 shifts, static load (300 working days per year, 8 hours per 1 working shift)
- Number of revolution of motor: $n_{dc} = 1420$ (rpm).
- Efficiency:
  - Belt drive $\eta_d = 0.95$.

  - Spur gear drive $\eta_{br} = 0.96$.
  - Bearings $\eta_{ol} = 0.99$.
  - Chain drive $\eta_x = 0.95$.

- The gears are calculated by ISO standard, the material is ENC60, the coefficients including $K_A = 1$, $K_{Hv} = 1.2$, $K_{H\beta} = 1.2$, $K_{H\alpha} = 1$ are input into Autodesk Inventor.
- The belt drive is calculated by DIN 2215 standard. Choose $d_1 = 180$ (mm), center distance $a = d_2$, the belt length, DIN belt type. The coefficient such as $P_{RB} = 3.8$ (kW), $k_1 = 1.2$.
- The chain is selected by ISO 606:2004 (EU) standard.

Pr	Parameters				Pr	Parameters				Pr	Parameters			
	P_3	u_1	u_2	u_3		P_3	u_1	u_2	u_3		P_3	u_1	u_2	u_3
1	2.5	2	2	3	31	6.5	4	2	3	61	12	3	2	4
2	2.5	2	2.24	4	32	7	3	2.24	3	62	12	4	2.24	2
3	2.5	4	2.5	4	33	7	4	2.5	4	63	12.5	3	2.5	3
4	3	2	3.15	3	34	7	3	3.15	4	64	12.5	3	3.15	4
5	3	2	3.55	3	35	7	2	3.55	3	65	12.5	4	3.55	2
6	3	4	4	2	36	7	3	4	4	66	12.5	3	4	4
7	3	2	2	3	37	7.5	4	2	2	67	12.5	4	2	3
8	3	4	2.24	3	38	7.5	4	2.24	3	68	12.5	4	2.24	2
9	3.5	3	2.5	2	39	7.5	4	2.5	3	69	13	4	2.5	4
10	3.5	4	3.15	2	40	8	4	3.15	3	70	13	3	3.15	4
11	3.5	3	3.55	2	41	8	4	3.55	2	71	13	2	3.55	2
12	3.5	3	4	4	42	8	4	4	3	72	13	3	4	2
13	4	4	2	3	43	8	2	2	3	73	13	4	2	3

	P_3	u_1	u_2	u_3		P_3	u_1	u_2	u_3		P_3	u_1	u_2	u_3
14	4	3	2.24	3	44	8.5	2	2.24	2	74	13	3	2.24	3
15	4	3	2.5	4	45	8.5	2	2.5	3	75	13	2	2.5	4
16	4	4	3.15	2	46	8.5	3	3.15	2	76	13	4	3.15	4
17	4	3	3.55	3	47	8.5	2	3.55	2	77	13.5	4	3.55	4
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22	4.5	3	3.15	3	52	10	4	3.15	3	82	14.5	4	3.15	4
23	5.5	4	3.55	4	53	10.5	4	3.55	4	83	14.5	3	3.55	4
24	5.5	2	4	3	54	10.5	2	4	4	84	15	3	4	2
25	6	3	2	4	55	10.5	2	2	4	85	15	2	2	3
26	6	2	2.24	4	56	10.5	2	2.24	2	86	15	4	2.24	3
27	6	4	2.5	4	57	10.5	2	2.5	2	87	15.5	3	2.5	3
28	6	4	3.15	3	58	11	4	3.15	3	88	15.5	3	3.15	4
29	6	4	3.55	3	59	11	3	3.55	3	89	16	4	3.55	2
30	6.5	2	4	3	60	11.5	2	4	4	90	16	4	4	3

Table 5.1: Projects

5.3.2 Results

The distribution table of transmission ratio

Parameters	Shaft			
	Motor	I	II	III
P (kW)	4.758	4.475	4.253	4
u	$1 \to 1$	$1 \to 3$	$3 \to 2.5$	$2.5 \to 4$
n (rpm)	1420	473.33	189.33	47.33
T (N · mm)	31999	90288	214526	807099

Table 5.2: Table of transmission ratio

Table of gear drive parameters

No.	Parameter	Result
1	Selected material	EN C60
2	Center distance	$a = 160$ (mm)
3	Module	$m = 4$ (mm)
4	Number of teeth	$z_1 = 23$
5	Number of teeth	$z_2 = 57$
6	Pitch circle diameter	$d_1 = 92$ (mm)
7	Pitch circle diameter	$d_2 = 228$ (mm)
8	Face width	$b_1 = 64$ (mm)
9	Face width	$b_2 = 60$ (mm)
10	Radial force	$F_r = 714.346$ (N)
11	Tangential force	$F_t = 1962.649$ (N)
12	Gear speed	$v = 2.28$ (m/s)

Table 5.3: Design specifications of the gear drive

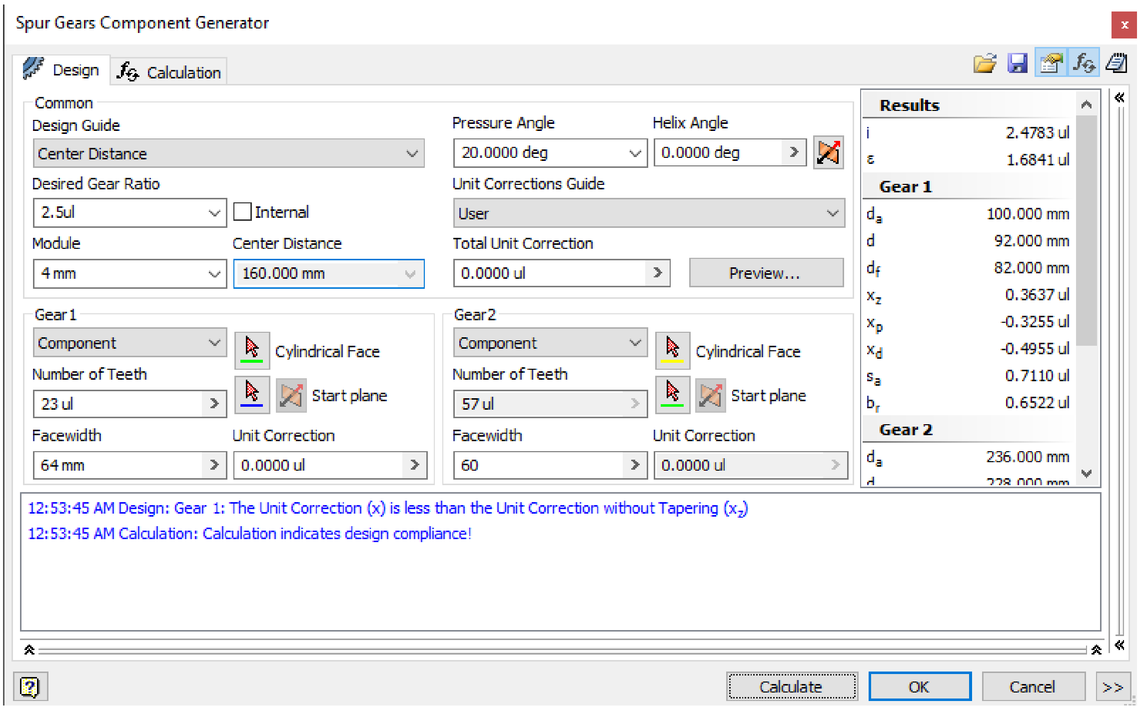

Figure 5.2: Gear design tab

The belt drive parameters table

No.	Parameter	Result
1	Belt type	V-Belt DIN 2215 (17x2240)
2	Number of belt	$z = 3$
3	Belt speed	$v = 13.383$ (m/s)
4	Initial tensile force	$F_v = 700.197$ (N)
5	Tensile force on each side	$F_t = 125.165$ (N)
6	Tensile force on the tight side	$F_1 = 553.254$ (N)
7	Tensile force on the slack side	$F_2 = 197.734$ (N)
8	Tangential force	$F_p = 355.521$ (N)
9	Radial force	$F_r = 711.895$ (N)
10	Wrap angle	$\alpha_1 = 137.61°, \alpha_2 = 222.39°$
11	Belt length	$L_d = 2283$ (mm)

No.	Parameter	Result
12	Pulley width	$B = 63$ (mm)
13	Center distance	$C = 572.572$ (mm)

Table 5.4: Design specifications of the belt drive

The chain drive parameters table

No.	Parameter	Result
1	Chain type	Roller Chain 16B-3-106
2	Number of chains	$k = 3$
3	Number of chain links	$X = 106$
4	Tangential force	$F_p = 2517.405$ (N)
5	Tensile force on the tight side	$F_1 = 2540.238$ (N)
6	Tensile force on the slack side	$F_2 = 22.834$ (N)
7	Radial force	$F_r = 2555.607$ (N)
8	Contact angle	$\alpha_1 = 132.11°, \alpha_2 = 227.89°$
9	Center distance	$C = 626.966$ (mm)
10	Drivsing sprocket diameter	$D_p = 170.421$ (mm)
11	Driven sprocket diameter	$D_p = 679.304$ (mm)

Table 5.5: Design specifications of the chain drive

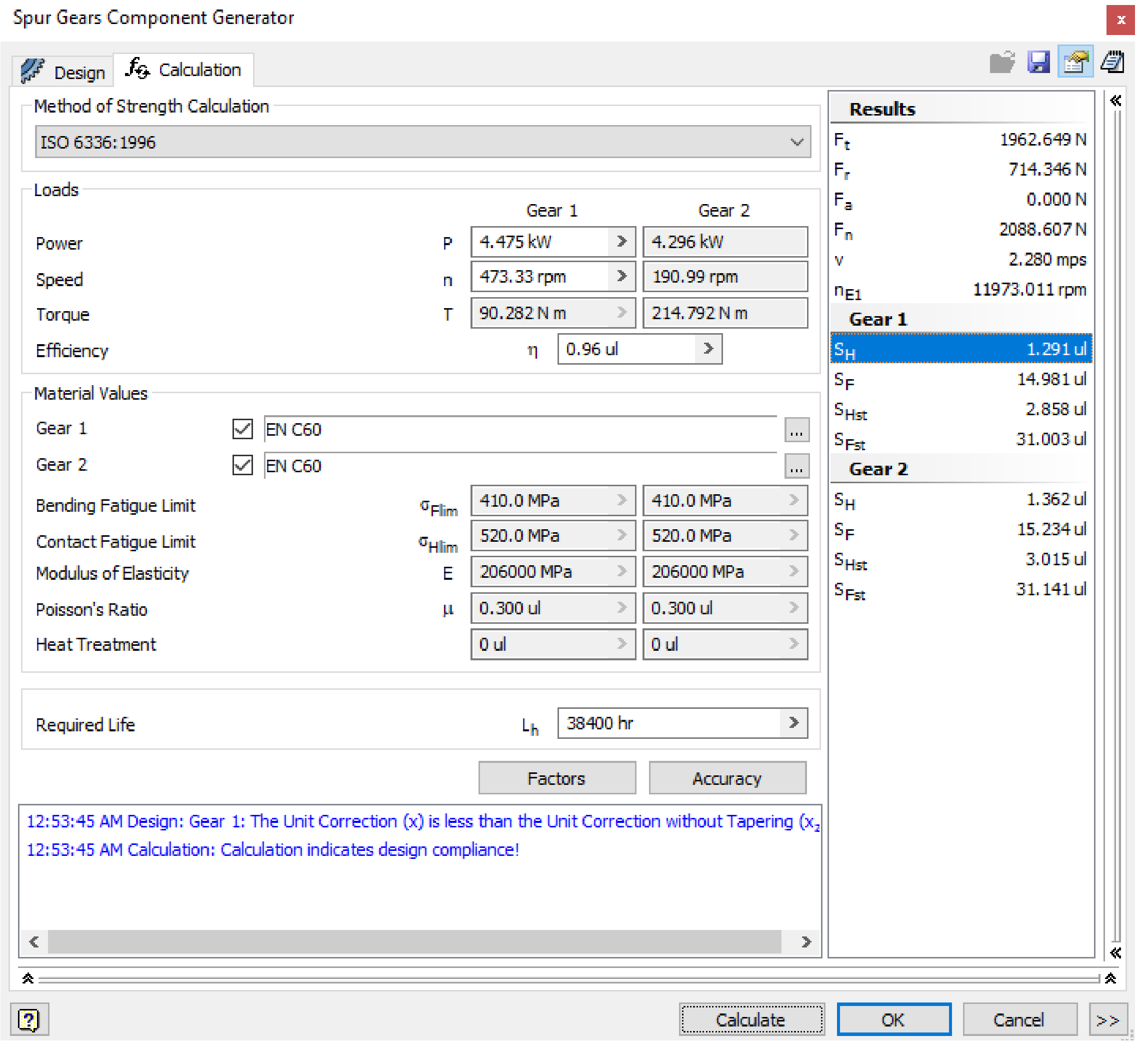

Figure 5.3: Gear calculation tab

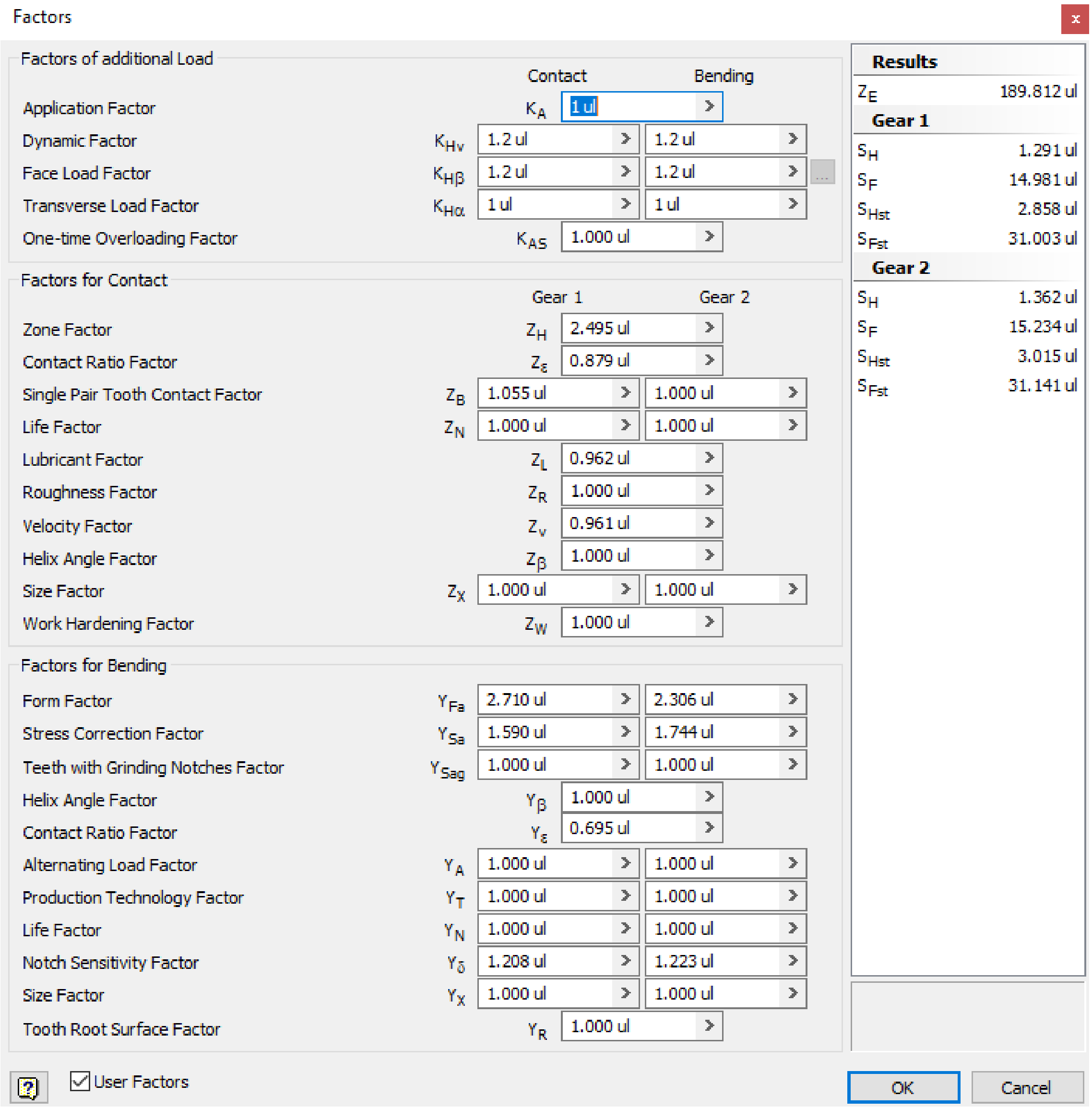

Figure 5.4: Gear factors tab

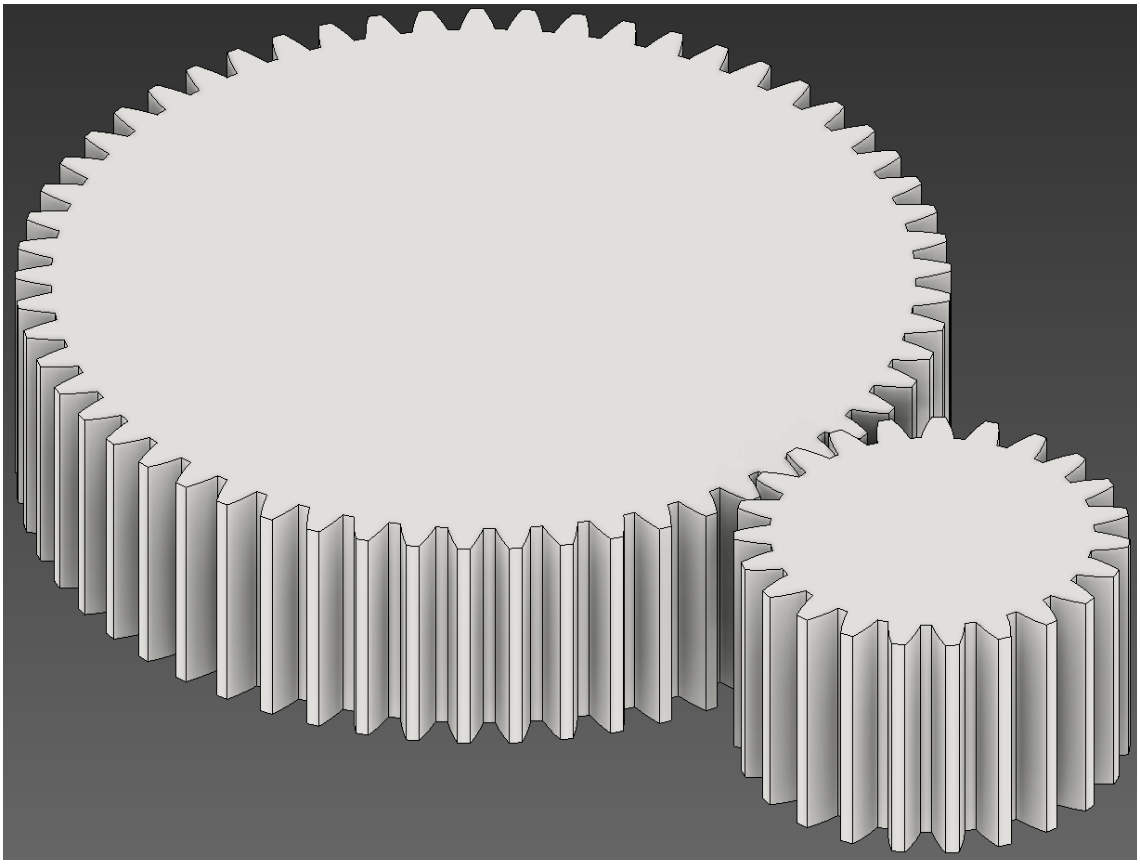

Figure 5.5: Gear drive model

5.4 Discussion and conclusions

Students compare the computational results in Inventor software with the theoretical results. Then draw conclusions when we use Inventor software.
Calculation results from Autodesk Inventor are mostly similar to theoretical calculation. The software is a convenient tool for calculating and designing machine parts, which also provides a complete machine detail modules. The users can easily input and adjust the parameters and use the built-in libraries from the software.

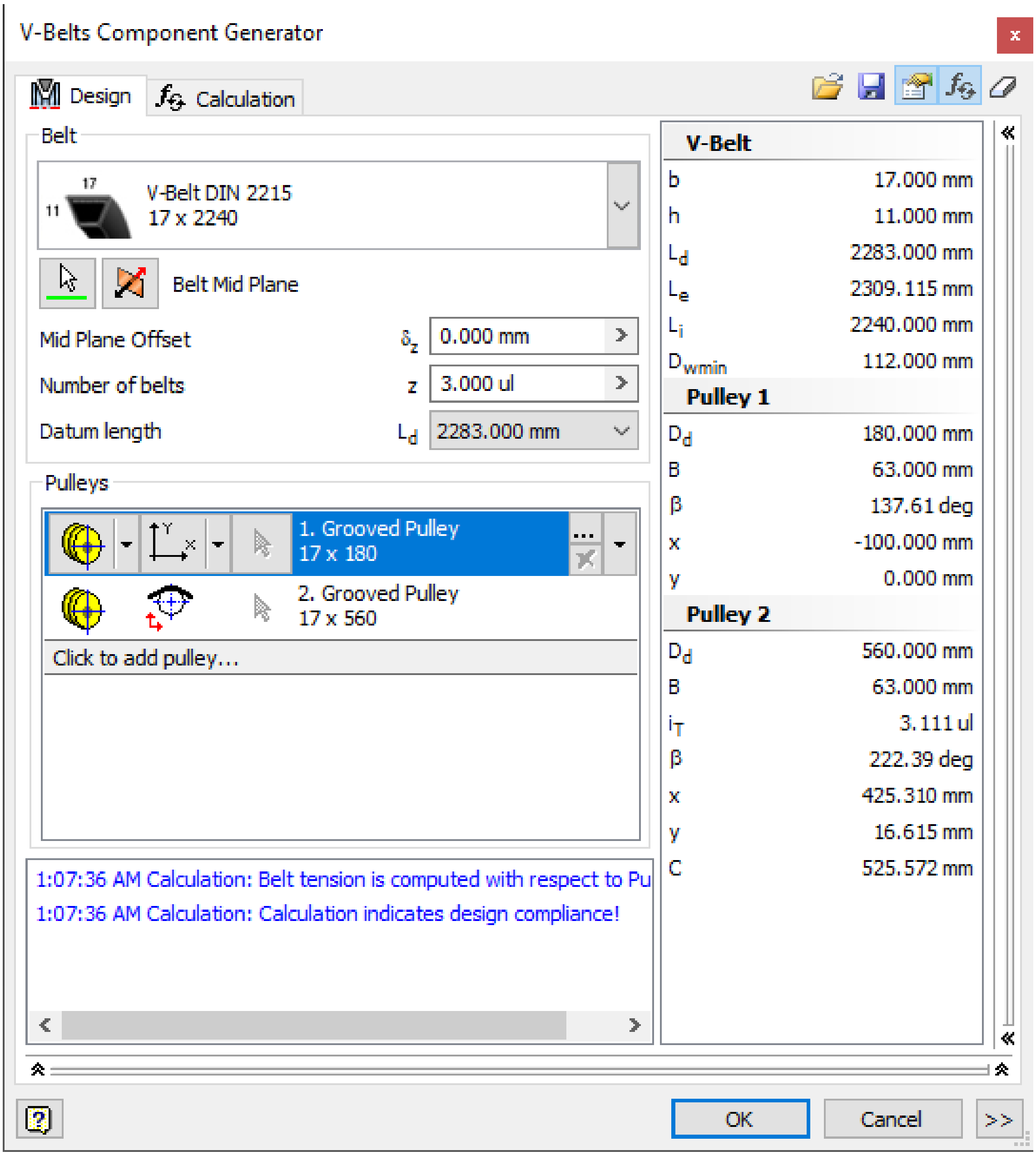

Figure 5.6: Belt design tab

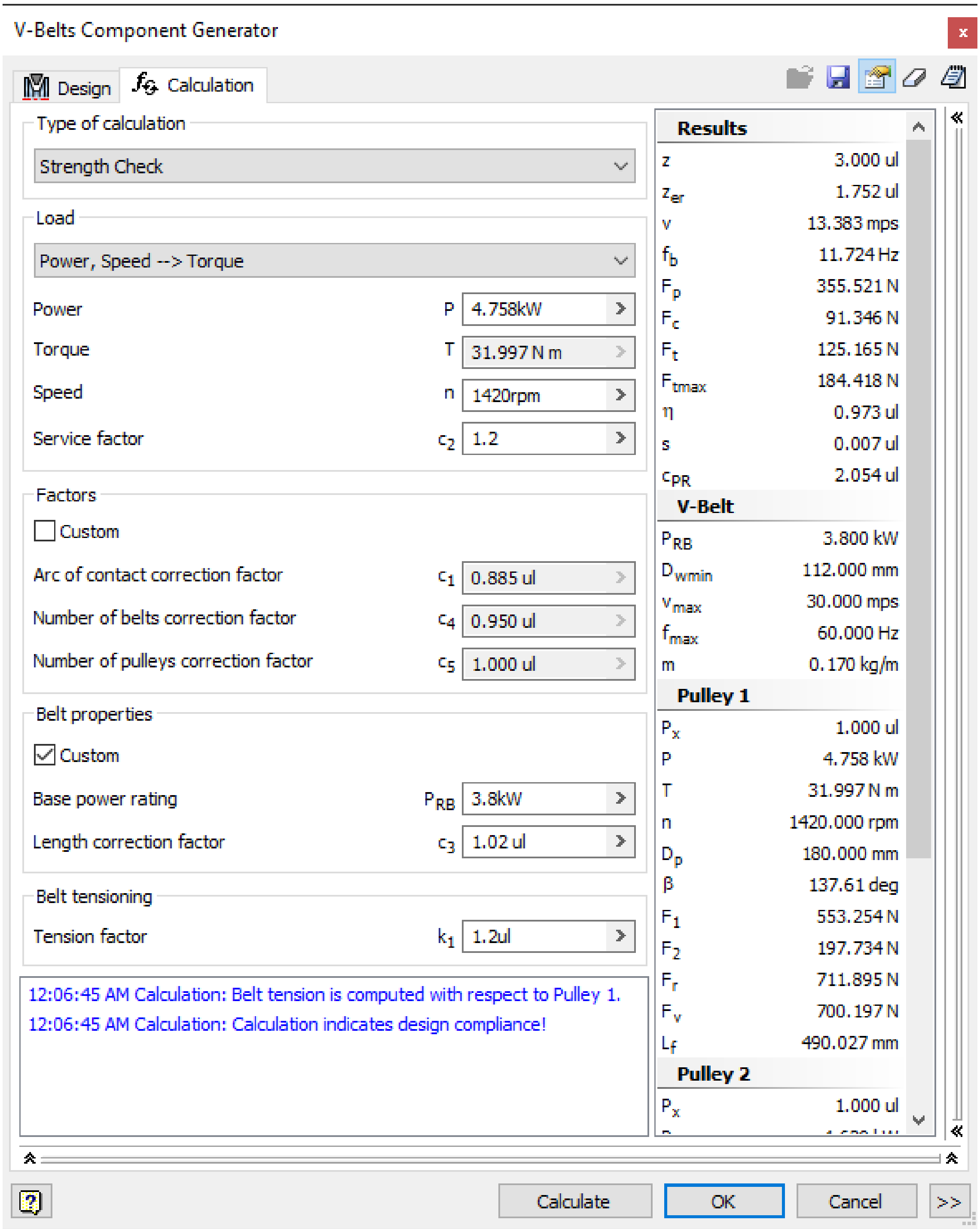

Figure 5.7: Belt calculation tab

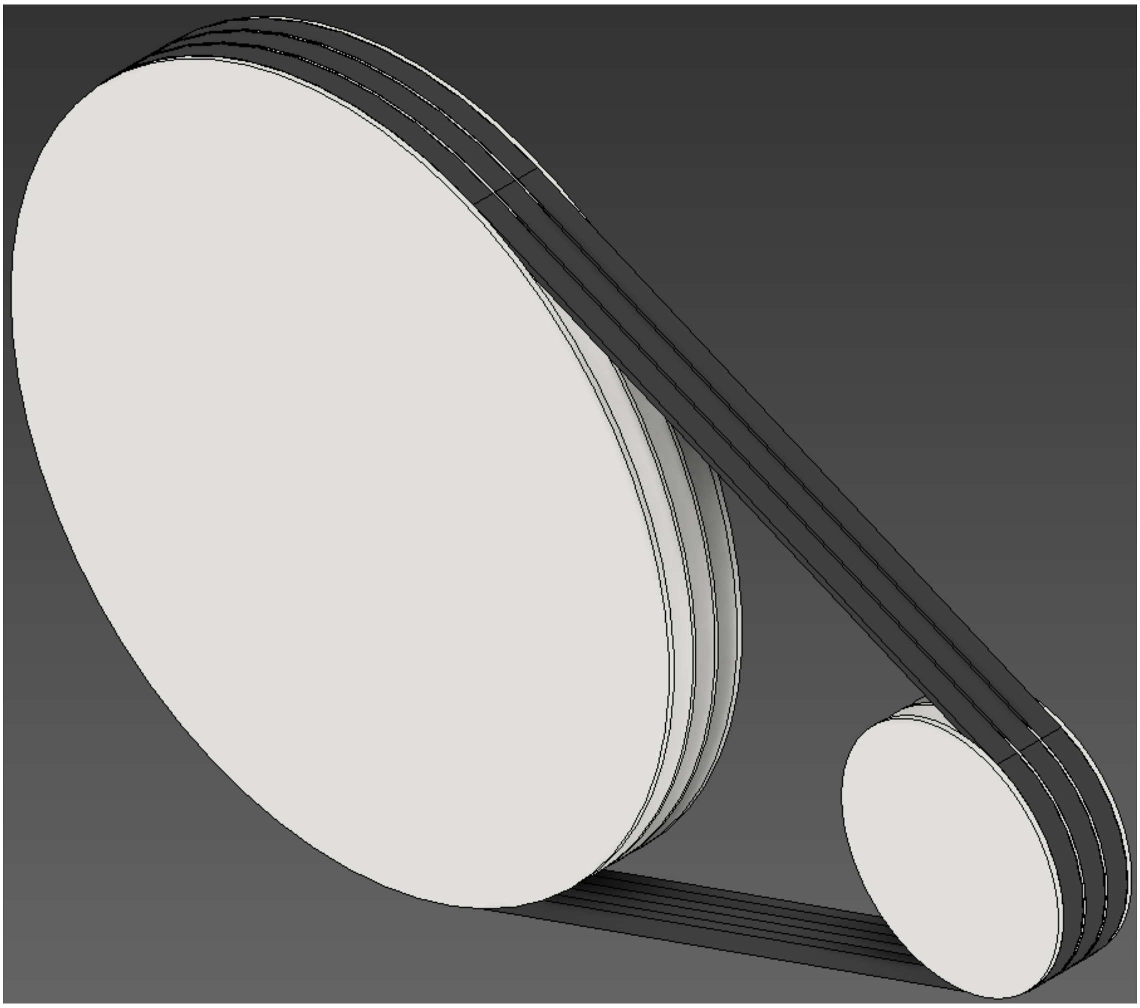

Figure 5.8: Belt drive model

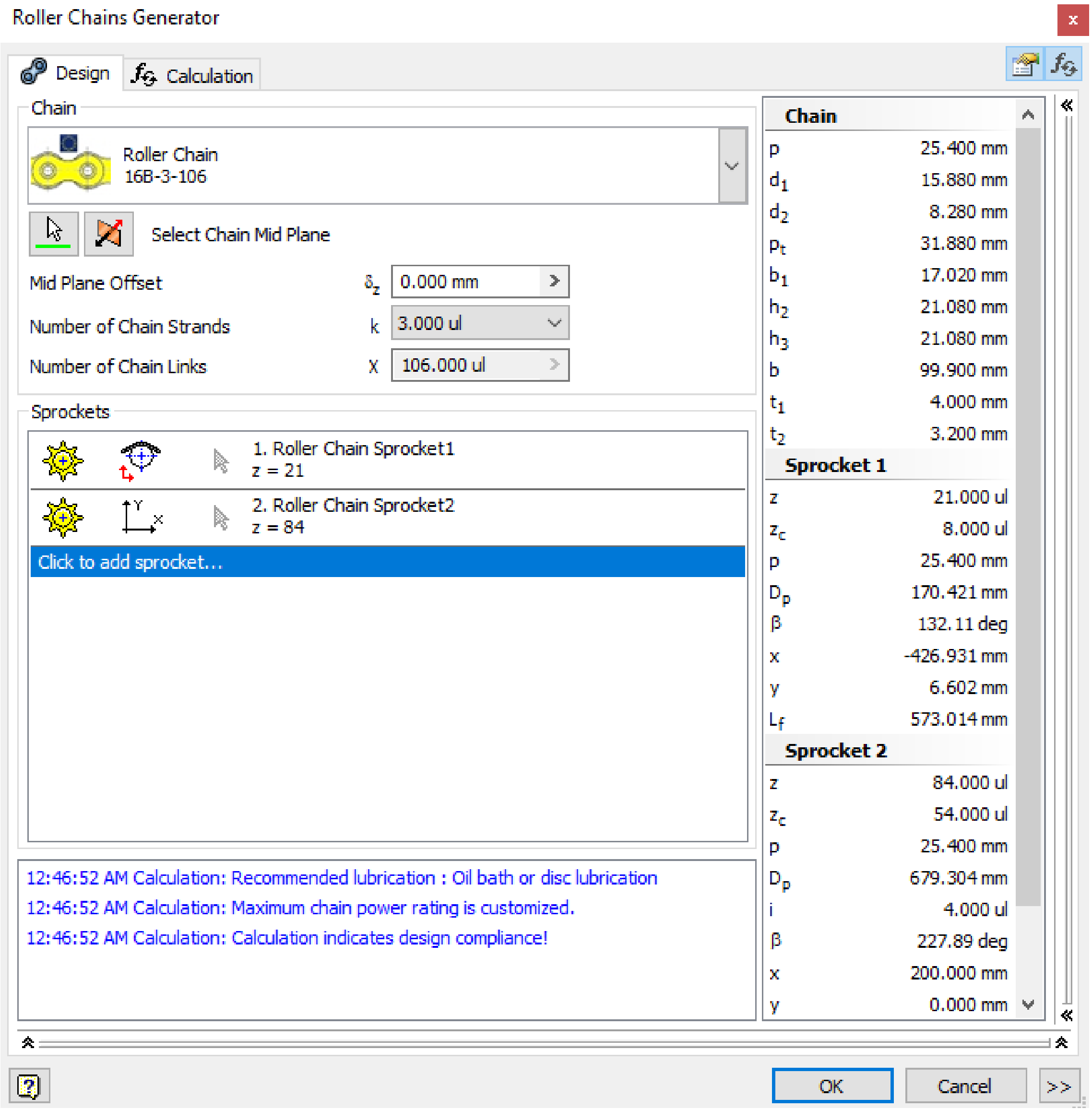

Figure 5.9: Chain design tab

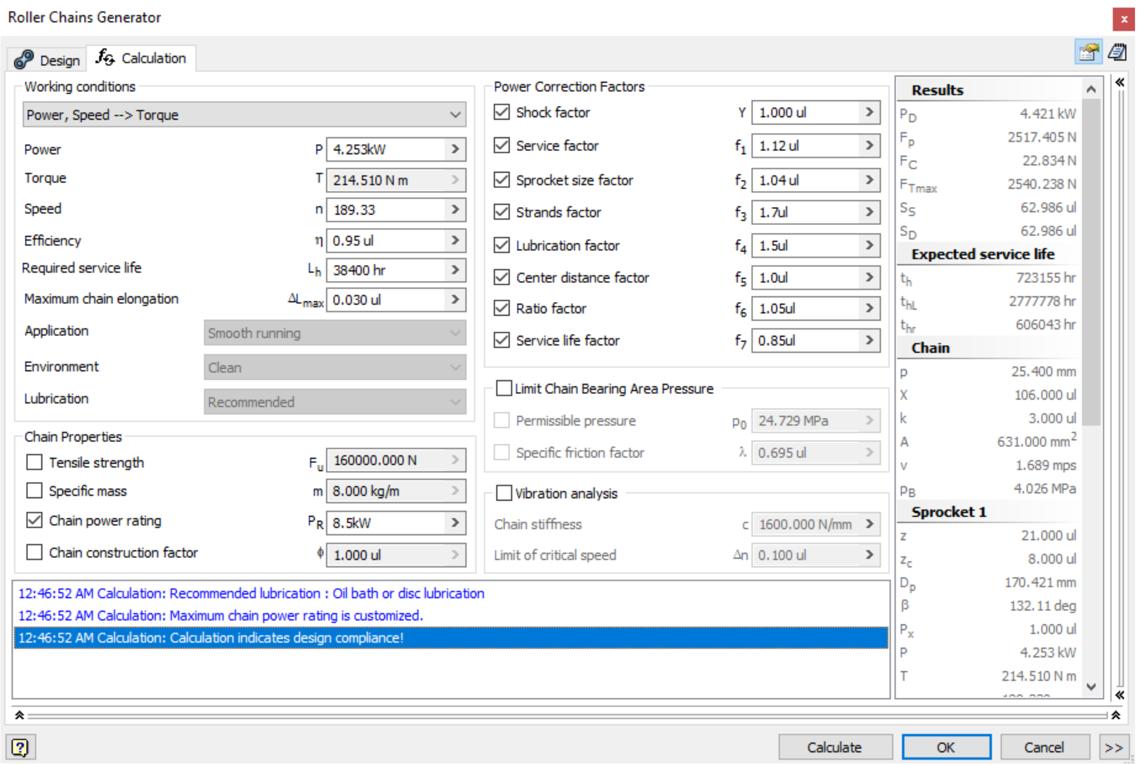

Figure 5.10: Chain calculation tab

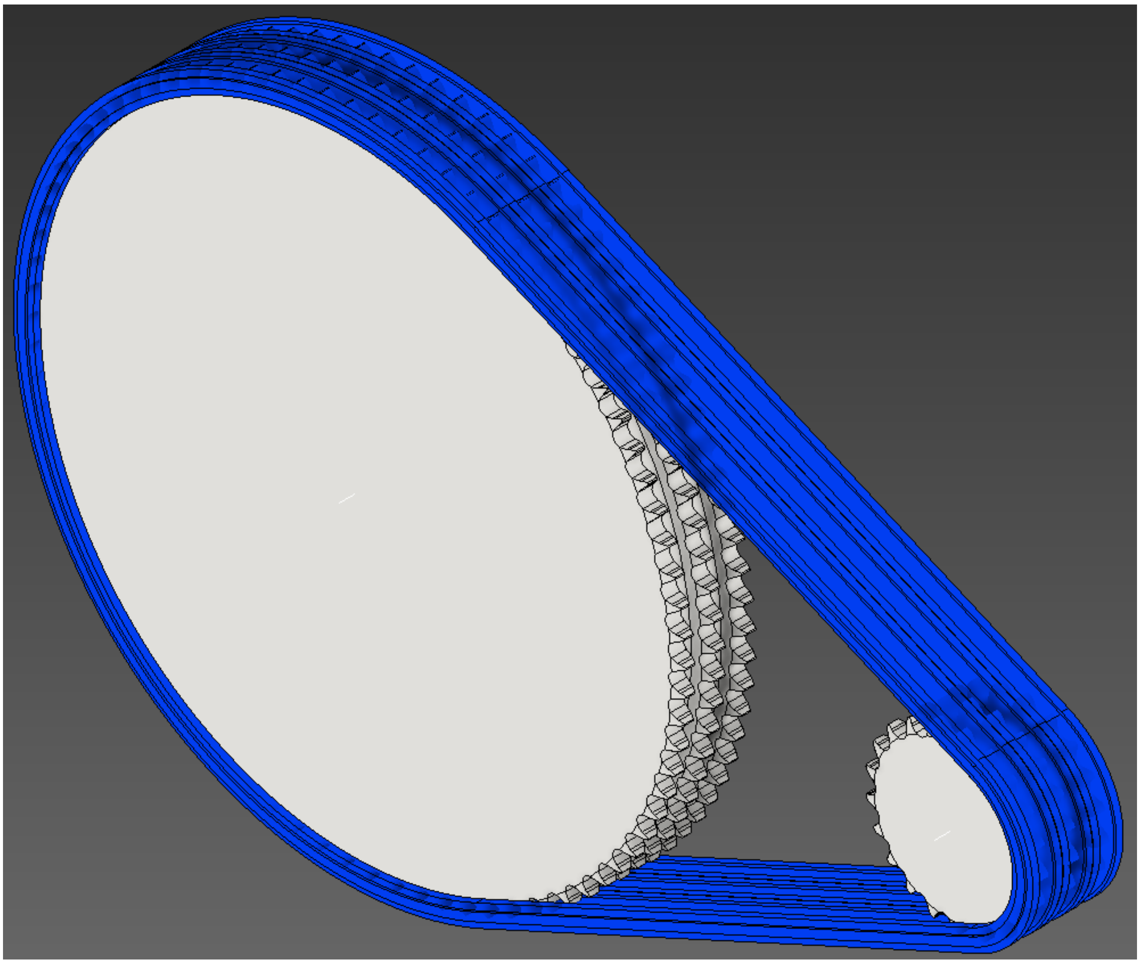

Figure 5.11: Chain drive model