Machine Elements Presentation

CONVEYOR BELT DESIGN ANLAYSIS BY Team-03

Contents

01

Part -1

Functions, Design requirements , kinematical- power -torque calculations

What is being Designed ?

Function:

A conveyor belt is designed to help with moving batches of packed potato chips from the packaging machine to the palate boxes at a constant speed of 5.4 km/h or 1.5 m/s

Design requirements :

The manufacturing cost has to be kept low, high reliability Factor is required along with easy repairability and user friendliness .The belt has work for 12 hrs everyday for an expected drive life service of 12 yrs

Kinematical Calculations

The power of the motor is found using eq 1.1

But it requires the Efficiency of the machine(η)
Which is calculated by muliplying all the chosen values from the efficiency table provided as per the drive elements in the machine.

The Required values were found to be
(η) = 0.841 and P_{motor} = 7 KW

$$P_{motor} = \frac{P_{WM.}}{\eta} \quad \text{......... (Eq 1.1)}$$

$P_{WM.}$ = Power required (Force x Velocity)
η = efficiency of the machine

	"closed"	"open"
Spur, helical gear coupling	0.96…0.98	0.93…0.95
Bevel gear coupling	0.95…9.97	0.92…0.94
Warm gear drive "self locking" Z1 = 1 Z1 = 2 Z1 = 3 Z1 = 4	 0.3…0.4 0.65…0.70 0.70…0.75 0.80…0.85 0.85…0.90	 0.2…0.3
Chain drive	0.95…0.97	0.90
Belt drive		0.95…0.96
Friction drive	0.90…0.96	0.70…0.80

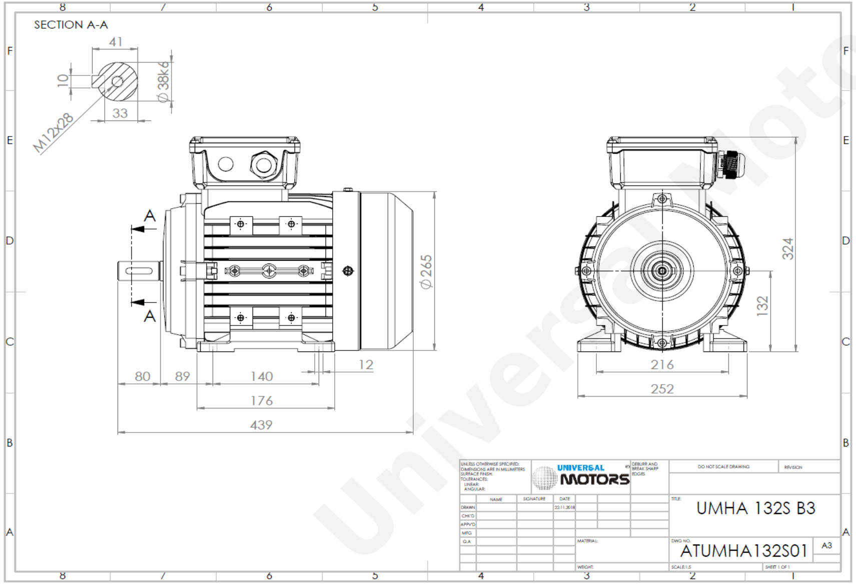
Pair of rolling bearings η = 0.99…0.995
Pair of sliding bearings η = 0.98…0.99

•Efficiency coefficients [1]

UNIVERSAL MOTORS
KENWORTH

TECHNICAL DATA

Three Phase Electric Motors - Aluminum Frame
UMHA 132SX-2 IE2
7.5 kW / 10 Cv
2915 RPM



Motor type:	UMHA 132SX-2 IE2
Output:	7.5 kW / 10 Cv
Speed (min^{-1}):	2915 RPM
Voltage:	400 V
Frequency:	50 Hz
Current:	13.8 A
Power factor (Cos Ø):	0.89
Poles:	2
Insulation class:	F
Temperature rise Class:	B
Efficiency 50%:	88.5 %
Efficiency 75%:	88.6 %
Efficiency 100%:	88.1 %
Torque:	24.6 N.m
Tstarting/Tnominal:	2.2
Istarting/Inominal:	7.8
Tmaximum/Tnominal:	2.3
Noise:	85 Db(A)
Weight:	52.0 Kg
Bearing DE:	6308 2Z C3
Bearing NDE:	6308 2Z C3

7.5 kw Kenworth UMHA 132SX-2 IE2 Three phase electric motor was chosen for this project.

Power, Torque and Angular velocities of the shafts:

The Power transmitted along the shafts are found by Multiplying the Power and Efficiency of the elements in that shaft

$$P_{shaft} \times \eta_{\textit{Drive elements}}$$

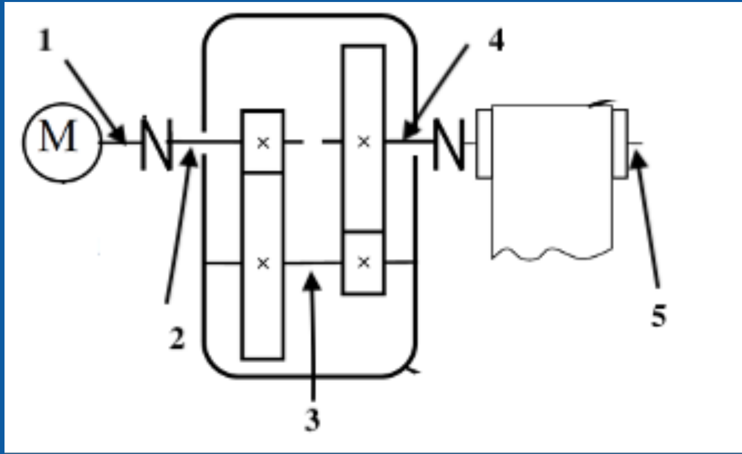
$P_{shaft1} = 7.5$ Kw

$P_{shaft2} = P_{shaft1} \times \eta_{bearing\ 2} = 7.425$ Kw

$P_{shaft3} = P_{shaft2} \times \eta_{gear2} \times \eta_{bearing\ 2} = 7.425 \times 0.96 \times 0.99 = 7.056$ kw

$P_{shaft4} = P_{shaft3} \times \eta_{gear2} \times \eta_{Bearing3} = 7.056 \times 0.96 \times 0.99 = 6.706$ Kw

$P_{shaft5} = P_{shaft4} \times \eta_{Belt} \times \eta_{Bearing4} = 6.706 \times 0.95 \times 0.99 = 6.306$ Kw



Machine and Shafts scheme

Power, Torque and Angular velocities of the shafts:

Transmission Ratio :

Since there are 2 sets of spur gears the transmission value is chosen from the table to be 3 and 4

The total transmission ratio (ι) is calculated by dividing the Rpm of the motor $(n\ e\text{ng})$ to the exit link $(n\ exi\text{t})$

$$i = \frac{n_{eng}}{n_{exit}} = \frac{2915}{358}$$

$= 8$

	Recommended	Max
Spur gear coupling	3…4	(10)
Helical gear coupling	3…5	(10)
Bevel gear coupling	2…3	(6)
Warm gear drive	10…40	(80) open till (120)
Belt drive		
- Flat	2…5	(6)
- Flat with tension roller	4…6	(8)
- V belt	2…5	(7)
Chain drive	2…6	(8)
Friction drive	2…8	(10)

Power, Torque and Angular velocities of the shafts:

The angular velocity of the shafts is the ratio between the angular velocity of the previous shaft to the chosen transmission value

$$\omega = \frac{\omega 1}{i1}$$

The angular velocity of shaft 2 = Shaft 1 in this machine as it is a direct connection from the motor .

Torque of the shafts is the ratio between the power transmitted by the shaft to the angular velocity of the shaft

$$\text{torque} = \frac{Power\ of\ shaft}{\omega\ shaft}$$

02

Part –2

Drives and Shaft diameters calculations, coupling and bearing selection

Shaft diameters calculations

Shafts play a significant role in almost all types of machinery. They revolve and transmit power . Since, the shafts are more susceptible to wear and high torque, the shafts are made from ANSI 1045 steel.

The diameters of the shafts are found using the formula below

$$d \geq \sqrt[3]{\frac{T}{0.2\,[\tau]}}$$

Where: d - diameter of the shaft, mm

T – Torque of the shafts, Nm

(τ) - Allowed shear stress (20 Mpa)

Shaft -2	19mm
Shaft -3	27mm
Shaft -4	41 mm
Shaft-5	41 mm

Calculated shaft diameters

Selection of the Elastic Coupling

Since the shaft diameters where found, the elastic couplings connecting them can be selected.

The nominal torque of the couplings is calculated below eq 1.2

$$\boxed{T_n = k\,T \qquad \text{(eq 1.2)}}$$

Where, T_n = Torque

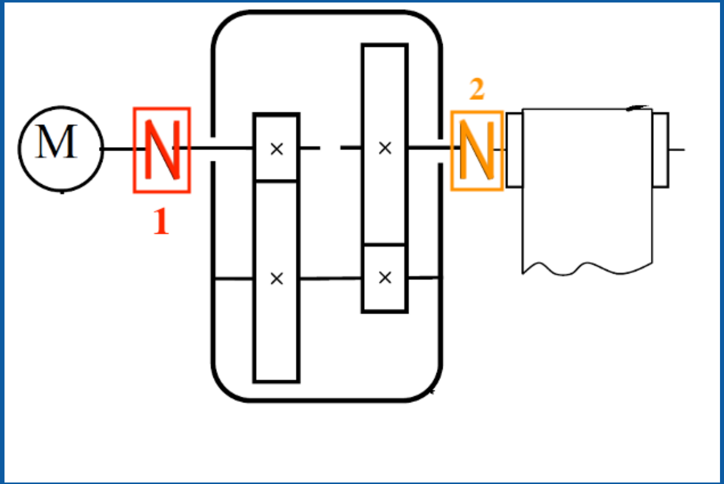
k = Dynamical coefficient (chosen to be 1 here)

T = Calculated Torque of the shafts

Coupling 1 = 1 x 25 = 25 Nm

Coupling 2 = 1 x 268 = 268 Nm

Based on these values couplings are chosen from manufacturers

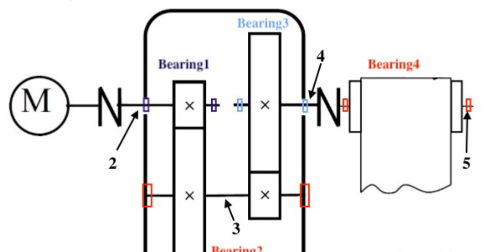


Elastic Coupling scheme

Bearing Type Selection

The required bearing type is chosen using the table below ,

Deep grove ball bearings are chosen for this machine. As they can handle high radial loads.



The bearing positions are marked with coloured rectangles

The matrix can only provide a rough guide so that in each individual case it is necessary to make a more qualified selection referring to the information given in the catalogue

Symbols
+++ excellent
++ good
+ fair
- poor
-- unsuitable
← single direction
←→ double direction

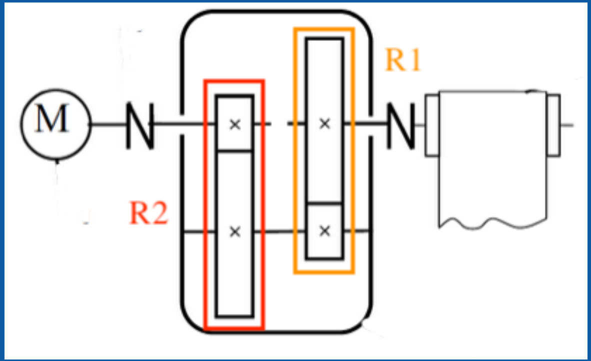
Design
1 Tapered bore
2 Shields or seals
3 Self-aligning
4 Non-separable
5 Separable

Characteristics
Suitability of bearings for
6 Purely radial load
7 Purely axial load
8 Combined load
9 Moment load
10 High speed
11 High running accuracy
12 High stiffness
13 Quiet running
14 Low friction
15 Compensation for misalignment in operation
16 Compensation for errors of alignment (initial)
17 Locating bearing arrangements
18 Non-locating bearing arrangement
19 Axial displacement possible in bearing

		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
Deep groove ball bearings	a, b		a				+	+ ←→	+ ←→	- b+	+++ b+	+++ b+	+	+++	+++	-	-	++ ←→	+	--
Angular contact ball bearings							+	+ ←	++ ←	-	++	+++	+	++	++	-	-	++ ←	--	--
	a, b, c		b		a, b	c	++	+ ←→	++ ←→	+	+	++	++	+	+	--	--	++ ←→	+	--
							-	++ ←→	+ ←→	+	++	+	+	+	+	--	--	++ ←→	-	--
Self-aligning ball bearings							+	-	-	--	+++	++	-	++	+++	+++	+++	+ ←→	+	--
Cylindrical roller bearings							++	--	--	--	++	++	++	++	++	-	-	--	+++	+++
	a, a, b, b						++	+ a← b←→	+ a← b←→	--	++	++	++	+	++	-	-	++ a← b←→	+ a←	+ a←
full complement	a, b				a	b	+++	-	+ ←	--	-	+	+++	-	-	-	-	+ ←	+	+
	a, a, b, c		a				+++	-	+ a←→ b←	+	-	+	+++	-	-	--	--	+ a←→ b←	+ c	+ b← c←→
Needle roller bearings	a, b, c		a	c			++	--	--	--	+	+ a++	a++ b++	+	-	--	-- c++	--	+++	+++
	a, b, c		b, c				++	--	--	--	+	+	++	+	-	--	--	--	+++	+++
	a, b, c		b, c				+	+ c++ ←	+ ←	-	+	+	++	+	-	--	--	+ ←	--	--
Tapered roller bearings							++	++ ←	+++ ←	-	+	+	++	+	+	-	-	+++ ←	--	--
	a, b						+++	++ ←→	+++ ←→	+	+	+	+++	+	+	-	--	+++ ←→	-	--
Spherical roller bearings							+++	+ ←→	+++ ←→	--	+	+	++	+	+	+++	+++	++ ←→	+	--
CARB bearings							+++	--	--	--	+	+	++	+	+	+++	+++	--	+++	+++
full complement							+++	--	--	--	-	+	+++	+	+	+++	+++	--	+++	+++
Thrust ball bearings	a, b						--	+ a← b←→	--	--	-	++ a	+	-	+	-	--	++ a← b←→	--	--
	a, b						--	+ a← b←→	--	--	-	+	+	-	+	-	++	++ a← b←→	--	--
Needle Cylindrical roller thrust bearings	a, b						--	++ ←	--	--	-	a+ b++	++	-	-	--	--	++ ←	--	--
Spherical roller thrust bearings							--	+++ ←	+ ←	--	-	+	++	-	+	+++	+++	+++ ←	--	--

Gear Design

Spur gears where chosen for this machine with a modulus value of 4 , the Z1 and Z2 values are found based on the speed reduced .The gear geometry is calculated in MATLAB Based on the formula in the table below



Gear scheme

Quantity	Formula	Pinion	Gear	Units
Module	m	4		mm
Number of teeth	z	20	60	-
Pressure angle	$inv(\alpha_w) = inv(\alpha_o) + 2tan\left(\alpha_o\left(\frac{x_{i1}+x_{i2}}{z_1+z_2}\right)\right)$	20		degrees
Operating angle	$\alpha_w = inverse(inv(\alpha_w))$	0.0149		radians
Profile shift coefficients	x_i	0	0	-
Radial Gap coefficient	c	0.25		-
Coefficient of height of tooth's head	h_a^*	1		-
Centre Distance	$A = m\left(\frac{z_1+z_2}{2} + a\right)$	160		mm
Diameter of initial circle	$D_w = mz\left(1 + \frac{2a}{z_1+z_2}\right)$	75	226	mm
Diameter of circles under division	$D = mz$	80	240	mm
Diameter of principle circle	$D_b = Dcos(\alpha_o)$	75	226	mm
Diameter of sub-root circle	$D_f = m(z - 2h_a^* - 2c + 2x_i)$	70	230	mm
Diameter of top circle	$D_a = m(z + 2h_a^* + 2x_i - 2\psi_n)$	69	229	mm
Thickness of teeth	$S = m(0.5\pi + 2x_i tan(\alpha_O))$	6.2	6.2	mm
Step on the circles under division	$t_{12} = \pi m$	13		mm
Height of teeth	$h_{12} = m(2h_a^* + c - \psi_n)$	10		mm

Gear Calculations for R2

Gear Design

Using Lewis Bending strength formula, the gear tooth strength is calculated.

W_t = [S x F x Y]/D_p

Where:

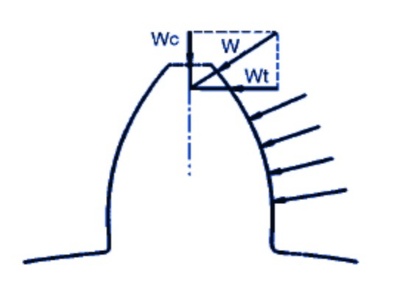
Wt = Maximum transmitted load (lbs, N)

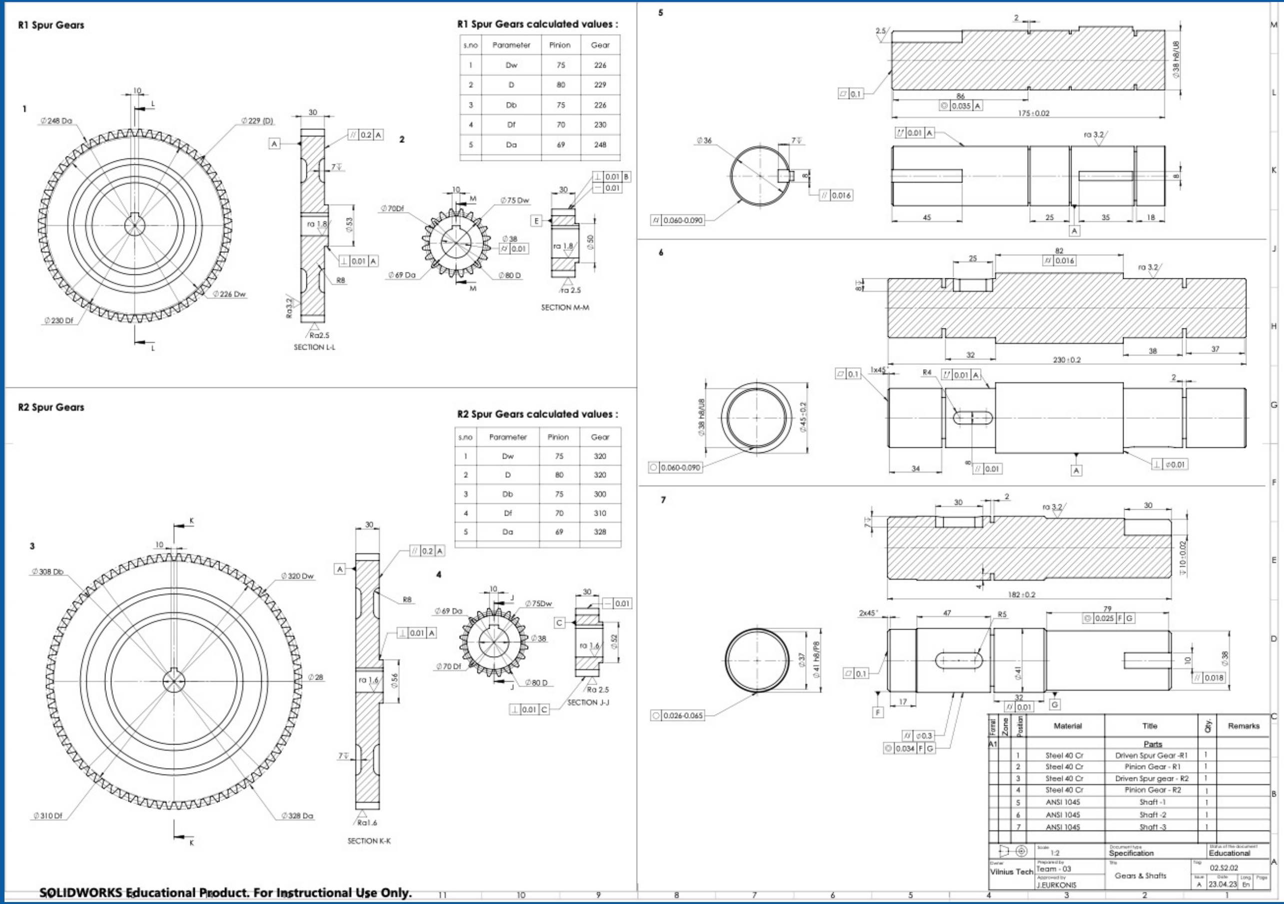
S = Maximum bending tooth stress taken as 1/3 of the tensile strength (psi, N/mm^2)

F = Face width of gear (in, mm)

D_P = Diametral I mm).

Y = Lewis Factor (See Lewis Factor for Gears) (no units)





Final Gear and Shaft diagrams

03

Part -3

Shaft - Bearing selection and Key calculations according to acting loads

Calculation of Shafts According to Acting Loads:

The forces exerted by the gears on the shaft is calculated using the following equations

Force tangential:

$$F_t = \frac{2000\,T}{d} \qquad ..(Eq3.1)$$

T - Torque of the shaft (Nm)

d - Diameter of the Gear (mm)

Radial Force:

$$F_r = F_t * \tan(20) \qquad ..(Eq\ 3.2)$$

F_t = force tangential (found using eq 3.1)

Calculation of Shafts According to Acting Loads

Using the found Radial - Tangential forces exerted by the gear the reaction forces are found at the supports and the graphs are drawn :

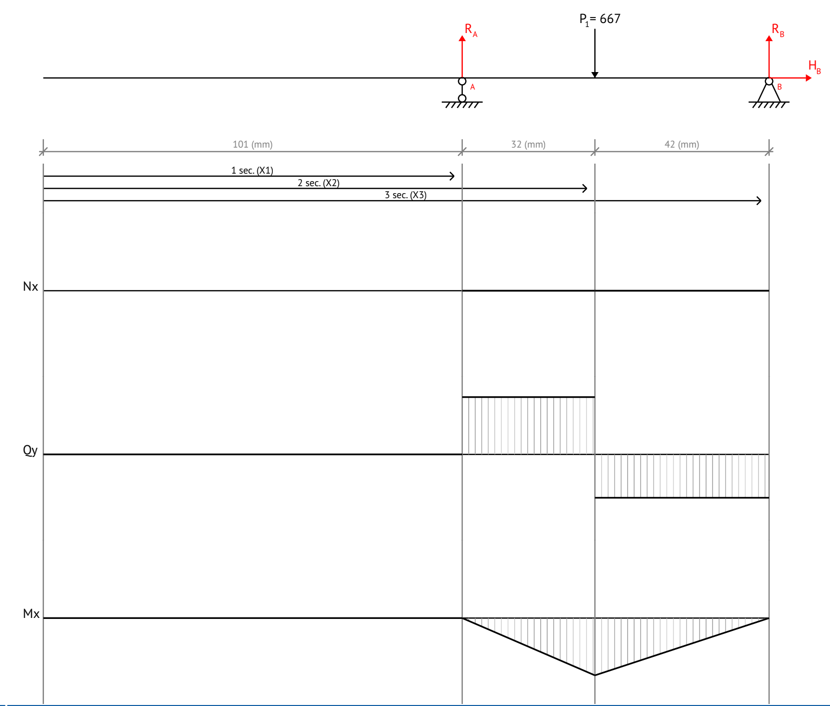


TABLE OF REACTION FORCES

	Radial forces Fr (N)	Tangential forces Ft (N)	Rotational speed (1/min)
Shaft -2	243	667	2915
Shaft - 3	879	2415	955
Shaft - 4	530	1456	238

Calculation of Shafts According to Acting Loads

Later, using the following formula the minimum diameters of the shaft are found .

$$D = \left[\frac{32 * N}{\pi}\sqrt{\left(\frac{K_t * M}{S_n'}\right)^2 + \frac{3}{4} * \left(\frac{T}{S_Y}\right)^2}\right]^{\frac{1}{3}}$$

$where$ Modified endurance strength : $S_n' = S_n * C_s * C_R$

Size Factor : $C_s = 0.8$, Reliability Factor : $C_R = 0.9$

$S_n = S_T * 0.45$, $S_n = 294.75 MPa$

$S_n' = 294.75 * 0.8 * 0.9$, $S_n' = 212.22 MPa$

$Where\ Safety\ Factor: N = 3,$

$Kt: Stress\ concentration\ Factor\ at\ shoulder, Kt = 1.5\ is\ chosen.$

The material chosen for the shafts is ANSI 1045 with the following properties ,

Property	Units	Value
Density ρ	Kg/m^3	7879
Ultimate Strength S_T	MPa	655
Yield strength S_Y	MPa	515

Selection of Bearings According to forces

The Actual Shaft load and Dynamic equivalent load are found using the following formula :

$$K = fw * Kc$$

K: Actual Shaft Load

f_w: Load Factor

K_c: Theoretically calculated Value

Dynamic equivalent load

$$P = V * X * F_r + Y * F_t$$

P: Equivalent load

V: Rotation Race Factor (1.2 for inner race of bearing rotating)

X: Radial load factor

Y: Axial Load Factor

F_r: Radial load

F_t: Axial load (tangential load)

* for single-row radial bearings, it is taken that X=1, and Y=0.

Calculated Values :

Units – (N)	K	P
Shaft-2	291.6	350.4
Shaft-3	1055	1266
Shaft-4	636	763

Bearing Life calculations

The basic rating life of a bearing is calculated using Eq 3.3

$$L_{10} = \left(\frac{C}{P}\right)^{P} \qquad \text{(eq3.3)}$$

Where,

L10 = basic rating life [100,000 m],

C = basic dynamic load rating [kN], (from the bearing catalogue)

P = equivalent dynamic bearing load [kN], ([3]weight of the gear given by the manufacturer)

p = exponent of the life equation (10/3 – for roller bearings)

The bearing life expressed in operating hours is calculated using following formula (eq 1.4)

Key dimensions calcuations

The Actual Shaft load and Dynamic equivalent load are found using the following formula :

The usual proportions of this key are calculated using the following formula,

Width of key, $w = \frac{d}{4}$

Where , d- diameter of shaf t (mm)

thickness of key, $t = \frac{2w}{3} = \frac{d}{6}$

	Key Height (mm)	Key Width (mm)
Shaft -2	8	10
Shaft -3	8	10
Shaft -4	8	12

Calculated values

04

Part –4

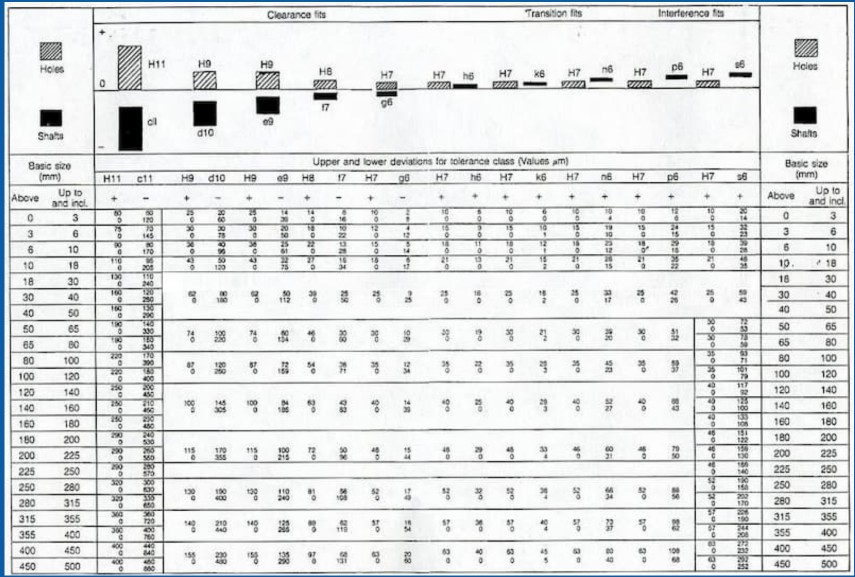
Tolerances-Roughness and Lubrication selection calculations

Selection of fits and tolerances:

The required fit is chosen as per the table 13.1

Type of Fit	Description	Hole Basis	Shaft Basis
Clearance Fits	Loose Running	H11/c11	C11/h11
	Free Running	H9/d9	D9/h9
	Close Running	H8/f8	F8/h8
	Sliding	H7/g6	G7/h6
	Locational Clearance	H7/h6	-
Transition Fits	Similar	H7/k6	K7/h6
	Fixed	H7/n6	N7/h6
Interference Fits	Press	H7/p6	P7/h6
	Driving	H7/s6	S7/h6
	Forced	H7/u6	U7/h6

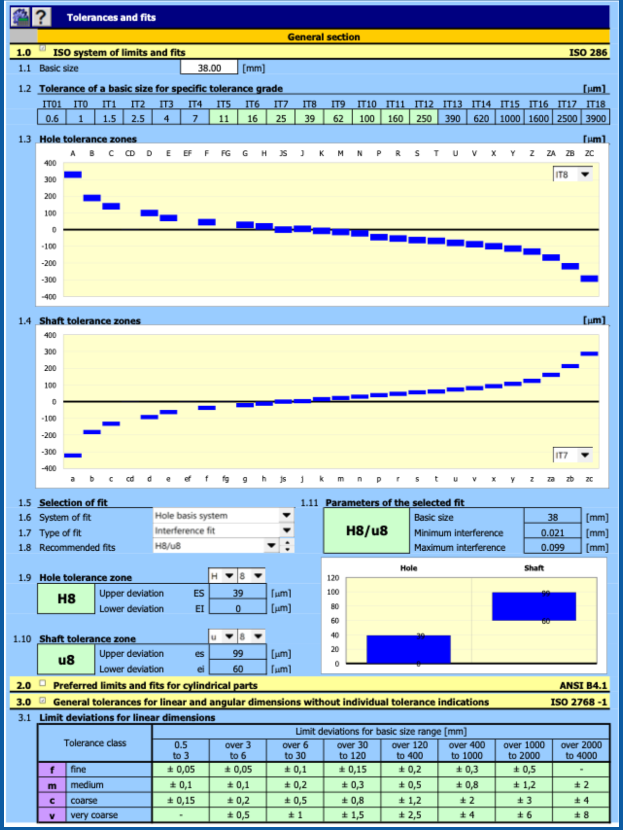
The tolerances are calculated based on the adding nominal diameter of the shaft to the selected grade from the ISO 286 TABLE :



To ease the processes MITCALC was used.

Selection of fits and tolerances:

Quick view of MITCALC



SELECTION OF LUBRICANTS

Synthetic lubricant oil with a ISO viscosity of <u>VG 320</u> was selected for the machine for its following properties

Resistance to the formation of residues and deposits at high temperatures

- Improves efficiency due to reduced tooth-related friction losses (low traction coefficients)
- Lower gearing losses due to reduced frictional losses (low traction coefficients)
- Reduced operating temperatures especially under fully loaded conditions

Final Composition of the machine

The conveyor belt was successfully designed

Thank You