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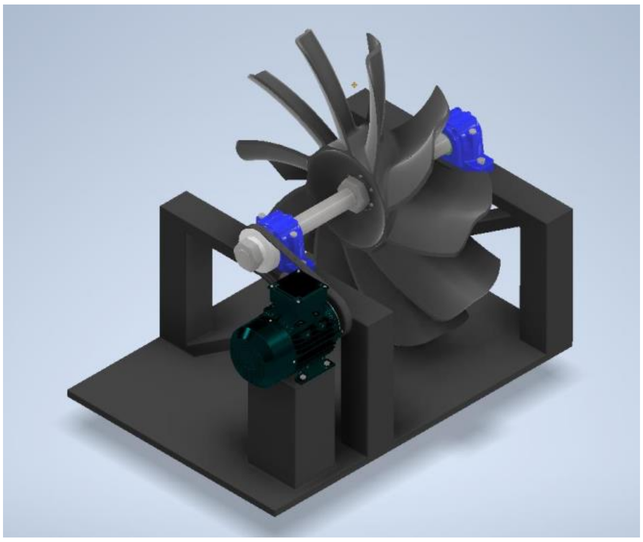
Designing an industrial ventilation fan drive system



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The goal of this design task is to design a belt drive system for an industrial extraction fan that meets the specifications listed below. The design needs to account for the stresses and mechanical fatigue on the load bearing components posed by the continuous operation of the fan, while components need to be selected to cope with such demands.

Specifications:

Type = V1 (internal impeller)

Power P_{jk} = 3 kW

Operating speed n_{jk} = 1400 RPM

Thrust (axial) load on the rotor = F_{ax} = 470 N

Radial load on the rotor F_R = 985 N

Mass of the rotor m_{jk} = 44 kg

Geometrical data:

Distance of the bearings from the rotor a= 300mm

Shaft diameter under the rotor d = 48 mm

Data of the belt drive mechanism:

Shaft distance estimated in advance (from outer non-locating bearing to pulley a_{centre}) = 660mm

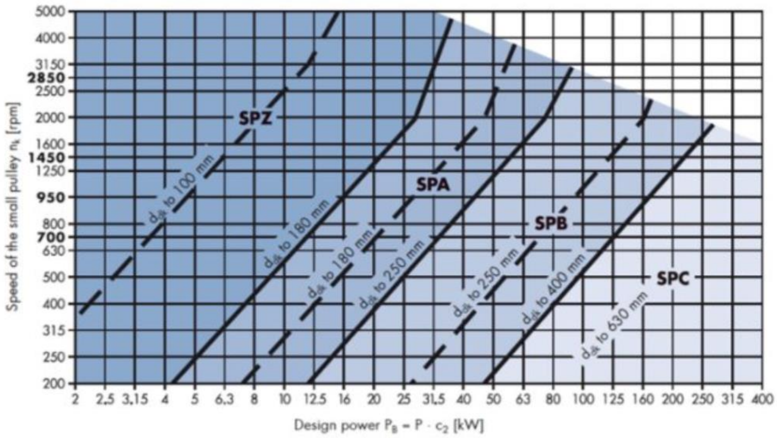
Direction of the belt pulling force α = 10°

An industrial extractor fan should be able to operate for extended periods of time, allowing a workshop to continue operations throughout multiple staff shifts. This requires a daily operating time of up to and over 18 hours. Though fans typically do not need to start up under heavy load, an ability to do so is still useful in case of dirty or blocked fan components causing excess friction.

This gives our fan a load factor of c_2 = 1.3.

Finding the design power $P_B = c_2 \times P_{jk} = 3.9\ kW$

Based on this, we can determine the belt profile to be used, using the graph below:



Our parameters are within the limits of the SPZ-type belt profile, with a minimum small pulley diameter d_{dk} of 100mm.

The simplest solution for a motor is a 1 : 1 drive ratio for the belt drive. This means i = 1. The motor I chose is the 6SH 100LB-4 model, with a power output of 4KW and a maximum rotational speed of 1430 RPM.

Choosing the pulley and belt parameters:

Wedge belt profile DIN 7753 Part 1 and ISO 4184	–	–	–	SPZ	SPA	SPB	–	S
Datum diameter d_d	20.0							
	22.0							
	25.0							
	28.0	28.0						
	31.5	31.5						
	35.5	35.5						
	40.0	40.0	40	40				
	45.0	45.0	45	45				
	50.0	50.0	50	50				
	56.0	56.0	56	56				
	63.0	63.0	63	63	63			
				67	67			
	71.0	71.0	71	71	71			
				75	75			
	80.0	80.0	80	80	80			
				85	85			
		90.0	90	90	90	90		
		100.0	100	100	100	100		
		112.0	112	112	112	112		
				118	118	118		

$$d_{d2} = d_{d1} = 100mm$$

$$n_1 = n_2 = 1400RPM$$

Finding the limits of the centre distance:

$$a_{min} = 0.7(d_{d1} + d_{d2}) = 0.7(100 + 100) = 140mm$$

$$a_{max} = 2(d_{d1} + d_{d2}) = 2(100 + 100) = 400mm$$

Assuming a = 300mm, we can choose a belt length:

$$L_{dth} = 2a + 1.57(d_{d1} + d_{d2}) + \frac{(d_{d1} + d_{d2})^2}{4a} = 2 \times 300 + 1.57(100 + 100) + \frac{(100 - 100)^2}{4 \times 300} = 914mm$$

From ISO 4184:1992, we can choose a suitable belt:

L_d	Distribution according to the sections			
	SPZ	SPA	SPB	SPC
630	+			
710	+			
800	+	+		
900	+	+		
1 000	+	+		
1 120	+	+		

Therefore, L_d = 1000mm.

Finding the centre distance:

$$a_{nom} = \frac{L_{Belt\ length} - \frac{\pi}{2}(d_{d1} + d_{d2})}{4} + \sqrt{\left[\frac{L_{Belt\ length} - \frac{\pi}{2}(d_{d1} + d_{d2})}{4}\right]^2 - \frac{(d_{d1} + d_{d2})^2}{8}}$$

$$= \frac{1000 - \frac{\pi}{2}(100 + 100)}{4} + \sqrt{\left[\frac{1000 - \frac{\pi}{2}(100 + 100)}{4}\right]^2 - \frac{(100 + 100)^2}{8}} = 327.66mm$$

Therefore, the value of a is 327.66mm, which is suitable, as it is between the values of a_{min} and a_{max}. As seen from the table, the minimum allowance for the easy fitting of the belt is 15mm.

Datum length [mm]	Minimum allowance x [mm] – for tensioning	Minimum allowance y [mm] – for easy fitting			
		SPZ, XPZ	SPA, XPA	SPB, XPB	SPC, XPC
487 ≤ 670	10	10	10	–	–
> 670 ≤ 1000	15	15	15	–	–
> 1000 ≤ 1250	20	15	15	–	–
> 1250 ≤ 1800	25	20	20	20	–

Finding the belt speed:

$$v_{belt} = \frac{d_{d1}\pi n_1}{60} = \frac{0.1 \times \pi \times 1400}{60} = 7.33ms^{-1} \leq 55ms^{-1}$$

The belt frequency:

$$f = \frac{2 \times 1000 v_{belt}}{L_{belt}} = \frac{2 \times 1000 \times 7.33}{1000} = 14.66s^{-1} \leq 100s^{-1}$$

The length and angle factors are obtainable from the tables below, based on a parameter β given by:

$$cos\frac{\beta}{2} = \frac{d_{d1} - d_{d2}}{2a} = \frac{100 - 100}{2 \times 327.66} = 0$$

This means that β=180˚.

$\frac{d_{dg} - d_{dk}}{a_{nom}}$	$\beta \approx$	c_1
0	180°	1.00
0.05	177°	1.00
0.10	174°	1.00
0.15	171°	1.00
0.20	168°	0.99

Datum length [mm]	c_3
630	0.83
670	0.84
710	0.85
750	0.86
800	0.87
850	0.88
900	0.89
950	0.90
1000	0.91
1060	0.92

From the table, we can observe that the angle factor c_1 is 1 and the length factor c_3 is 0.91.

Finding the nominal power rating P_N – Optibelt catalogue, for profiles spz:

Pulleys	v [m/s]	n_k [min⁻¹]	Datum diameter of small pulley d_{dk} [mm] 63	71	80	85	90	95	100	112	125	132	140	150	160	180	200	Additional power [kW] per belt for speed ratio i 1.01 to 1.05	1.06 to 1.26	1.27 to 1.57	> 1.57
		700	0.50	0.68	0.88	1.00	1.11	1.22	1.33	1.60	1.88	2.03	2.20	2.42	2.63	3.05	3.47	0.01	0.06	0.09	0.11
		950	0.63	0.87	1.14	1.29	1.44	1.59	1.74	2.08	2.46	2.66	2.89	3.17	3.45	4.00	4.54	0.01	0.09	0.12	0.15
		1450	0.87	1.23	1.62	1.84	2.06	2.27	2.49	3.00	3.54	3.83	4.16	4.56	4.96	5.75	6.51	0.02	0.13	0.19	0.23
		2850	1.38	2.03	2.74	3.13	3.52	3.90	4.27	5.15	6.07	6.55	7.08	7.72	8.34	9.50	10.55	0.04	0.26	0.37	0.46
		100	0.10	0.13	0.16	0.18	0.20	0.22	0.24	0.28	0.33	0.35	0.38	0.42	0.45	0.52	0.59	0.00	0.01	0.01	0.02
		200	0.18	0.24	0.30	0.34	0.37	0.41	0.44	0.52	0.61	0.66	0.71	0.78	0.85	0.98	1.12	0.00	0.02	0.03	0.03
		300	0.25	0.33	0.43	0.48	0.53	0.58	0.63	0.75	0.88	0.95	1.03	1.13	1.23	1.42	1.62	0.00	0.03	0.04	0.05
		400	0.32	0.43	0.55	0.62	0.68	0.75	0.81	0.97	1.14	1.23	1.34	1.47	1.59	1.85	2.10	0.01	0.04	0.05	0.06
		500	0.38	0.51	0.66	0.75	0.83	0.91	0.99	1.19	1.39	1.51	1.63	1.79	1.95	2.26	2.57	0.01	0.05	0.07	0.08
		600	0.44	0.60	0.78	0.87	0.97	1.07	1.16	1.39	1.64	1.77	1.92	2.11	2.29	2.66	3.02	0.01	0.06	0.08	0.10
		700	0.50	0.68	0.88	1.00	1.11	1.22	1.33	1.60	1.88	2.03	2.20	2.42	2.63	3.05	3.47	0.01	0.06	0.09	0.11
		800	0.55	0.76	0.99	1.12	1.24	1.37	1.50	1.79	2.12	2.29	2.48	2.72	2.96	3.44	3.91	0.01	0.07	0.11	0.13
		900	0.61	0.84	1.09	1.24	1.38	1.52	1.66	1.99	2.35	2.54	2.75	3.02	3.29	3.81	4.33	0.01	0.08	0.12	0.15
		1000	0.66	0.91	1.19	1.35	1.51	1.66	1.81	2.18	2.57	2.78	3.02	3.31	3.61	4.18	4.75	0.01	0.09	0.13	0.16
		1100	0.71	0.98	1.29	1.46	1.63	1.80	1.97	2.37	2.79	3.02	3.28	3.60	3.92	4.54	5.16	0.02	0.10	0.14	0.18
		1200	0.76	1.06	1.39	1.57	1.76	1.94	2.12	2.55	3.01	3.26	3.54	3.88	4.22	4.90	5.56	0.02	0.11	0.16	0.19
		1300	0.80	1.12	1.48	1.68	1.88	2.07	2.27	2.73	3.23	3.49	3.79	4.16	4.52	5.24	5.95	0.02	0.12	0.17	0.21
		1400	0.85	1.19	1.58	1.79	2.00	2.21	2.42	2.91	3.44	3.72	4.04	4.43	4.82	5.58	6.32	0.02	0.13	0.18	0.23
		1500	0.89	1.26	1.67	1.89	2.12	2.34	2.56	3.08	3.64	3.94	4.28	4.69	5.11	5.91	6.69	0.02	0.14	0.20	0.24
	5	1600	0.93	1.32	1.76	2.00	2.23	2.47	2.70	3.26	3.85	4.16	4.52	4.95	5.39	6.23	7.05	0.02	0.15	0.21	0.26
		1700	0.98	1.39	1.85	2.10	2.35	2.59	2.84	3.42	4.05	4.38	4.75	5.21	5.66	6.55	7.40	0.02	0.16	0.22	0.27
		1800	1.02	1.45	1.93	2.20	2.46	2.72	2.98	3.59	4.24	4.59	4.98	5.46	5.93	6.85	7.74	0.03	0.17	0.24	0.29
		1900	1.06	1.51	2.02	2.29	2.57	2.84	3.11	3.75	4.43	4.80	5.20	5.70	6.19	7.15	8.07	0.03	0.18	0.25	0.31
		2000	1.10	1.57	2.10	2.39	2.68	2.96	3.24	3.91	4.62	5.00	5.42	5.94	6.45	7.44	8.38	0.03	0.19	0.26	0.32
		2100	1.13	1.63	2.18	2.48	2.78	3.08	3.37	4.07	4.81	5.20	5.64	6.17	6.70	7.72	8.68	0.03	0.19	0.28	0.34
		2200	1.17	1.69	2.26	2.58	2.89	3.20	3.50	4.22	4.99	5.39	5.84	6.40	6.94	7.99	8.98	0.03	0.20	0.29	0.35
		2300	1.20	1.74	2.34	2.67	2.99	3.31	3.63	4.38	5.17	5.58	6.05	6.62	7.18	8.25	9.26	0.03	0.21	0.30	0.37
		2400	1.24	1.80	2.42	2.75	3.09	3.42	3.75	4.52	5.34	5.77	6.25	6.84	7.41	8.50	9.52	0.03	0.22	0.32	0.39
		2500	1.27	1.85	2.49	2.84	3.19	3.53	3.87	4.67	5.51	5.95	6.44	7.04	7.63	8.74	9.77	0.04	0.23	0.33	0.40

Therefore, P_N = 2.42kW, with 0.02kW of additional power.

The required number of belt loops is given by:

$$z = \frac{Pc_2}{P_N c_1 c_3} = \frac{3 \times 1.3}{2.42 \times 1 \times 0.91} = 1.77 \approx 2 \; to \; the \; nearest \; greater \; integer$$

Therefore, 2 belt loops will be used.

To find the static pulling force we use:

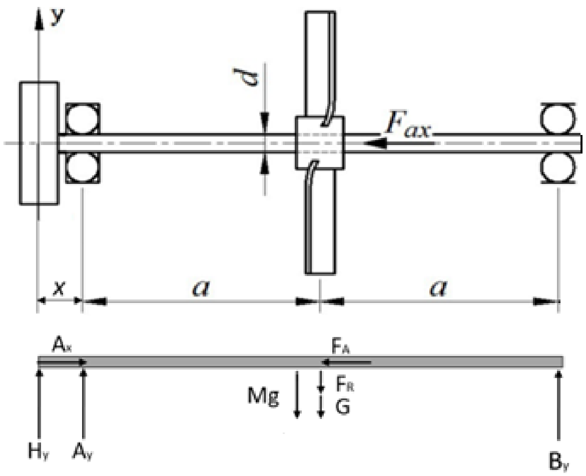
$$T = \frac{500 \times (2.04 - c_1) \times P_B}{c_1 \times z \times v} = \frac{500 \times (2.04 - 1) \times 3}{1 \times 2 \times 7.33} + 0.07 \times 7.33^2 = 110.17N$$

Calculating the belt pulling force:

$$H \approx 2T \times sin\frac{\beta}{2}z = 2 \times 110.17 \times sin\left(\frac{180}{2}\right) \times 2 = 440.69N$$

<u>Load and stress analysis on the shaft:</u>

xy plane:



Parameters:

a = 300mm

x = 60mm

Belt pulling force H_y = H sin(10) = 76.53 N

Axial force F_A = 470N

Radial load on rotor F_R = 985N

Weight of the impeller G = 44*9.81 = 431.64N

Estimating the shaft weight Mg: We will assume the shaft to be perfectly circular in profile. Since the exact material has not been selected yet but will likely be some kind of steel, we will assume its density to be 8000kgm^{-3}, which is slightly above the density of most steel alloys.

The volume using the cross-sectional area:

$$A = \frac{\pi d^2}{4} = \frac{\pi \times 0.048^2}{4} = 1.809 \times 10^{-3} m^2$$

$$V = A(2a + x) = 1.809 \times 10^{-3} \times (2 \times 0.3 + 0.06) = 1.1994 \times 10^{-3} m^3$$

Finding the mass and weight:

$$M = V\rho = 1.1994 \times 10^{-3} \times 8000 = 9.552 kg$$

$$Mg = 9.552 \times 9.81 = 93.70 N$$

Writing out the equilibrium equations to solve for the reaction forces A_x, A_y and B_y:

$$(1) \ \sum F_x = A_x - F_A = 0$$

$$A_x = F_A = 470N$$

$$(2) \ \sum F_y = H_y + A_y + B_y - F_R - G - Mg = 0$$

$$(3) \ \sum M_{Rotor} = aB_y - aA_y - (a + x)H_y + 0.03\text{Mg} = 0$$

(Note: Since Mg is assumed to act on the shaft's midpoint, it is 30mm from the impeller's centre)

Combining equations (2) and (3) and rearranging to find B_y results in:

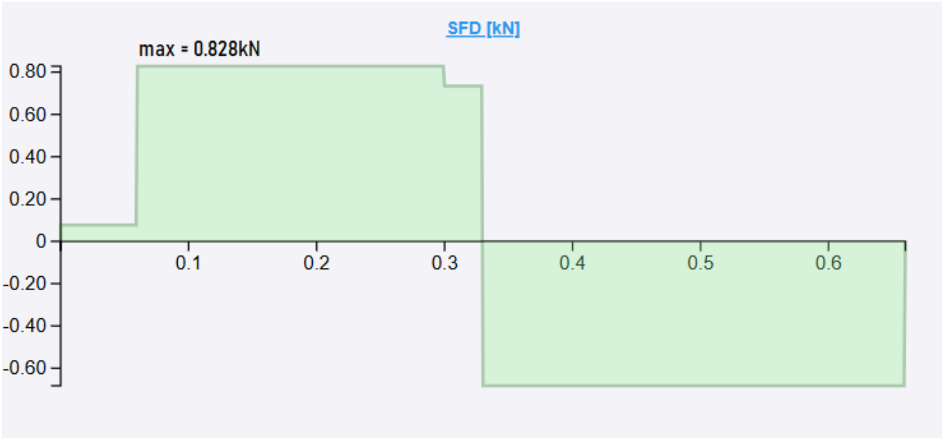
$$B_y = \frac{F_R + G + \left(1 - \frac{0.03}{a}\right) Mg + \frac{x}{a} H_y}{2} = \frac{985 + 431.64 + \left(1 - \frac{0.03}{0.3}\right) \times 93.70 - \frac{60}{300} \times 76.53}{2} = 742.8N$$

Substituting B_y back into equation (2) to solve for A_y:

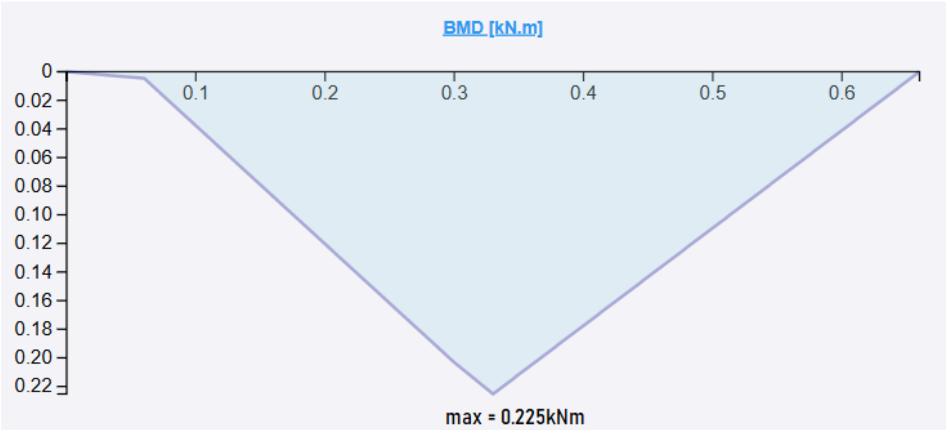
$$A_y = F_R + G + Mg - B_y - H_y = 690.1N$$

Following this, we can build stress resultant functions for shear forces and bending moments:

Shear force as a function of distance along the beam:

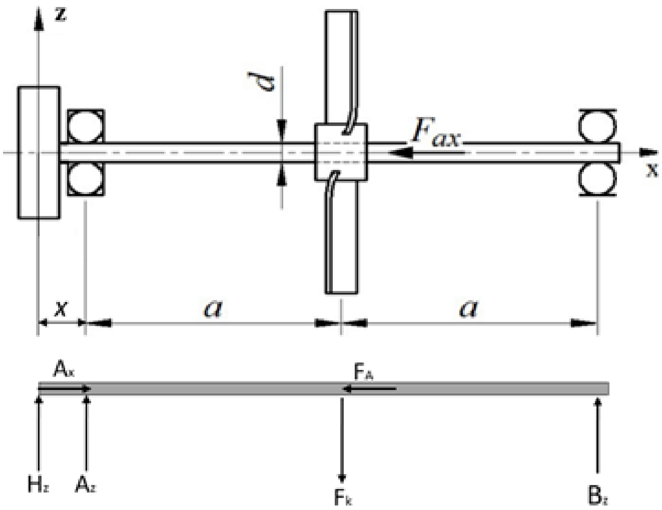


Bending moment as a function of distance along the beam:



The maximum shear force and bending moment is 828N and 225kNm, respectively.

xz plane:



Parameters:

Belt pulling force H_z = H cos(10) = 433.99N

The tangential force F_K can be estimated by dividing the torque by the fan's radius. The torque M_{fan}, obtained by dividing the power by its angular velocity 26.6Nm.

Therefore, assuming a fan radius of 1.5m, F_k = 26.6/1.5 = 17.73N

Finding the reaction forces:

$$(1)\ \sum F_z = H_z + A_z - F_k + B_z = 0$$

$$(2)\ \sum M_{Rotor} = -aB_z + aA_z + (a+x)H_z = 0$$

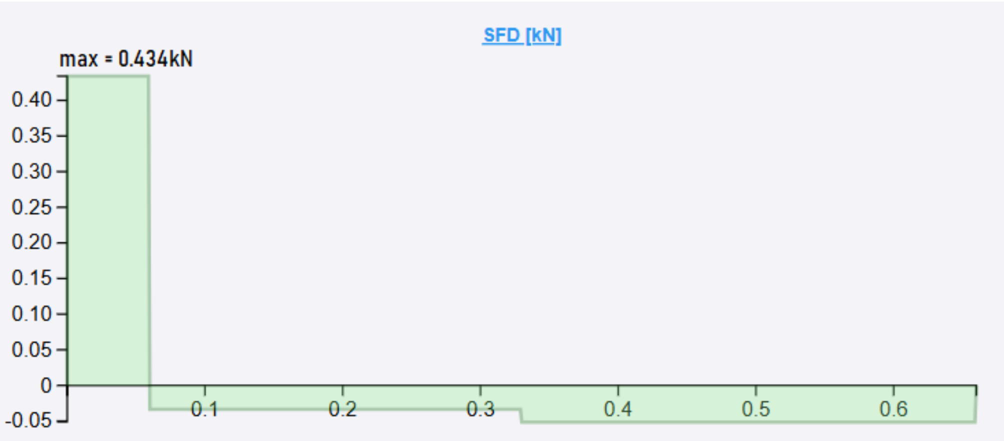
Combining the two equations gives B_z:

$$B_z = \frac{F_k + \frac{x}{a}H_z}{2} = \frac{17.73 + \frac{60}{300} \times 433.99}{2} = 52.25N$$

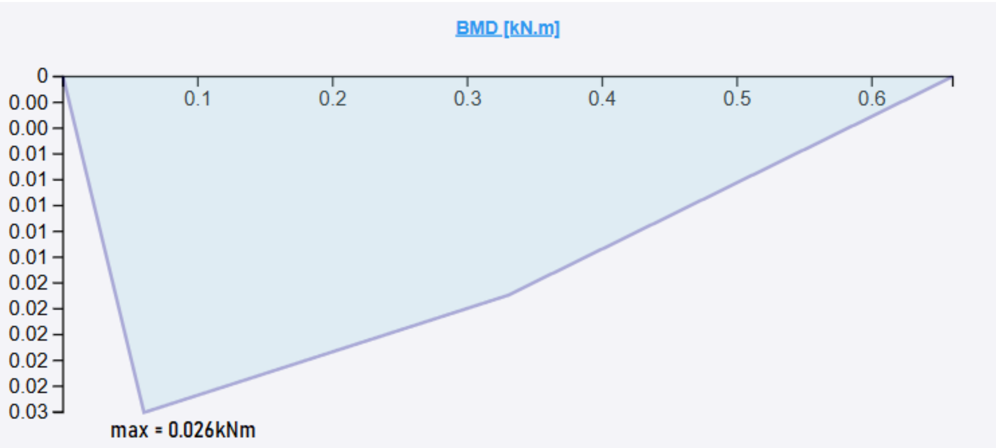
This, when substituted back into the first equation gives A_z:

$$A_z = -H_z + F_k - B_z = -433.99 + 17.73 - 52.25 = -468.51N$$

Shear force as a function of distance along the beam:



Bending moment as a function of distance along the beam:



The maximum shear force and bending moment is 434N and 26Nm, respectively.

Resultant loads and stresses:

The resultant maximum bending moment from the two stresses:

$$M_b = \sqrt{{M_{bxy}}^2 + {M_{bxz}}^2} = \sqrt{225^2 + 26^2} = 227.49Nm$$

Determining the normal and shear stress arising in the most dangerous cross-section of the shaft:

$$\sigma = \frac{M_b}{\frac{d^4 * \pi}{64}} * \frac{d}{2} = \frac{227.49}{\frac{0.048^4 * \pi}{64}} * \frac{0.048}{2} = 20952698 = 20.95MPa$$

$$\tau = \frac{M_{fan}}{\frac{d^4 * \pi}{64}} * \frac{d}{2} = \frac{26.6}{\frac{0.048^4 * \pi}{64}} * \frac{0.048}{2} = 2449954Pa = 2.450MPa$$

The overall reduced stress on the shaft material (assuming the material is absorbent) is thus:

$$\sigma_{red} = \sqrt{\sigma^2 + 3\tau^2} = \sqrt{20.95^2 + 3 \times 2.450^2} = 21.378MPa$$

Based on this, we can choose an appropriate steel alloy for the shaft. This will be DIN C45E, due to its relatively low cost. The safety factor for this is:

$$Z = \frac{R_{eH}}{\sigma} = \frac{400}{21.378} = 18.71$$

With a minimum yield strength of 400MPa for the annealed version of this steel, it will still provide a very large margin of safety of 18.71, which is far higher than the usual industry-wide expected safety factors of 1.5-4. This makes using stronger alloys a needless expense.

Selecting the bearings:

The shaft fan will have two bearings, each locating and non-locating. The locating bearing will be the SKF NUP 210 ECP while the non-locating one will be the SKF NU 210 ECP. Both are cylindrical bearings able to handle the radial and axial loads expected and are rated for speeds exceeding 1400RPM. The locating bearing will be at support A (the support closer to the belt) while the non-locating bearing will be at support B.

Checking the radial force on bearing A:

$$F_{ra,A} = \sqrt{{A_y}^2 + {A_z}^2} = \sqrt{690.1^2 + (-468.51)^2} = 834.11N$$

Axial load on bearing A:

$$F_A = A_x = 470N$$

Equivalent dynamic load: The bearing has an equivalent limiting factor e of 0.2, which means that equivalent dynamic load calculations are required if the ratio of axial to radial forces exceed this number.

$$\frac{A_x}{F_{ra,A}} = \frac{470}{834.11} = 0.563 > e = 0.2$$

The axial load factor Y is 0.6 and the radial load factor X is 1.

$$P = YA_x + XF_{ra,A} = 0.6 \times 470 + 1 \times 834.11 = 1116.11\ N$$

The basic dynamic load rating C of the bearing is 73.5kN, according to the manufacturer. Since we are dealing with a roller bearing, the lifetime power number p is 10/3. The lifetime of the bearing in million revolutions is given by:

$$\mathrm{L} = \left(\frac{\mathrm{C}}{\mathrm{P}}\right)^{\mathrm{p}} = \left(\frac{73000}{1116.11}\right)^{\frac{10}{3}} = 1127334 \text{ million cycles}$$

From this, we can obtain an estimate for the lifetime in operating hours:

$$\mathrm{L_h} = \frac{10^6 \mathrm{L}}{60\mathrm{n}} = \frac{10^6 \times 1127334}{60 \times 1400} = 13420651 \text{ hours}$$

The static load limit of the bearing is 69.9kN. Our values are far below this, meaning the bearing is suitable.

Checking the radial force on bearing B:

$$F_{ra,B} = \sqrt{{B_y}^2 + {B_z}^2} = \sqrt{742.8^2 + 52.25^2} = 744.64N$$

Axial load on bearing B:

$$B_x = 0N$$

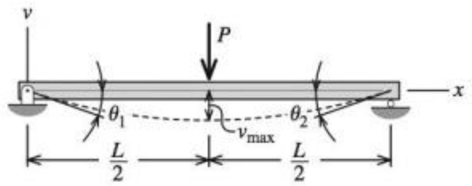
Equivalent dynamic load: This bearing bearing has an equivalent limiting factor e of 0.2.

$$\frac{B_x}{F_{ra,B}} = \frac{0}{742.8} = 0 < e = 0.2$$

This means equivalent dynamic load calculations are unneeded and bearing B does not undergo fatigue.

Finding the fundamental harmonic critical speed:

For the impeller side:

	Beam	Slope	Deflection
1		$\theta_1 = -\theta_2 = -\frac{PL^2}{16EI}$	$v_{max} = -\frac{PL^3}{48EI}$

Finding the maximum deflection where the Young's modulus E of C45E steel is 210GPa:

$$f_1 = \frac{P\left(\frac{L}{2}\right)^2 \times 64}{48\pi d^4 E} = \frac{1116.11 \times \left(\frac{0.6}{2}\right)^2 \times 64}{48\pi \times 0.048^4 \times 2.1 \times 10^9} = 3.824 \times 10^{-5}m = 0.03824mm$$

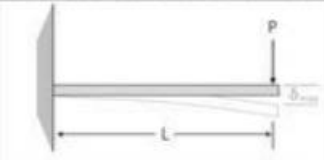
Shaft stiffness:

$$S_1 = \frac{P}{f_1} = \frac{1116.11}{3.824 \times 10^{-5}} = 29186976\frac{N}{m}$$

The natural frequency is given by:

$$\omega_{w1} = \sqrt{\frac{S_1}{M_1}} = \sqrt{\frac{29186976}{10.14 \times \frac{0.6}{0.66}}} = 1779.39\frac{rad}{s} = 16991RPM$$

For the pulley side:

Beam and load cases	Maximum Beam Deflection
	$\delta_{max} = \dfrac{PL^3}{3EI}$

The pulley is 360mm from the point P, therefore L=0.33m

$$f_2 = \frac{H_y L^2 \times 64}{48\pi d^4 E} = \frac{76.53 \times 0.36^2 \times 64}{48\pi \times 0.048^4 \times 2.1 \times 10^9} = 3.173 \times 10^{-4} m = 0.3173mm$$

$$S_2 = \frac{H_y}{f_2} = \frac{76.53}{3.173 \times 10^{-4}} = 241191 \frac{N}{m}$$

$$\omega_{w2} = \sqrt{\frac{S_2}{M_2}} = \sqrt{\frac{241191}{10.14 \times \frac{0.06}{0.66}}} = 511.51 \frac{rad}{s} = 4884.6RPM$$

Finding the critical speed:

$$n_{crit} = \sqrt{\frac{\omega_{w1}^2 \times \omega_{w2}^2}{\omega_{w1}^2 + \omega_{w2}^2}} = \sqrt{\frac{18479^2 \times 757963^2}{18479^2 + 757963^2}} = 4694RPM$$

This means that harmonic oscillation-based failure is not a concern, since the critical speed is well out of the operating speed range of the fan.

Checking for shaft fatigue:

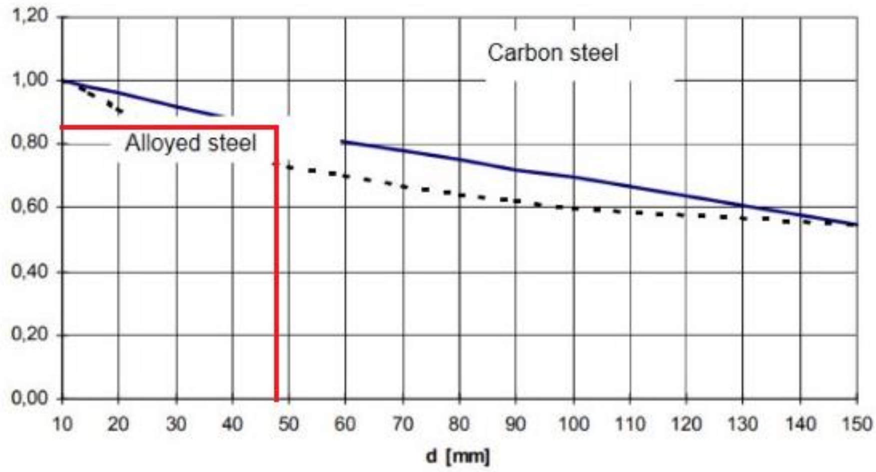
Fatigue characteristics of steels

Assumed fatigue characteristics of some standard steels (having medium thickness from 40 mm to 150 mm)

Material symbol		σ_{ult}	Tension σ_y	Tension σ_v	Bending σ_y	Bending σ_v	Shear τ_y	Shear τ_v
E 295		500	270	220	400	260	200	140
E 335	*	600	300	250	440	300	220	160
E 360		700	340	280	530	330	270	200
C 35 E		550	330	220	470	260	240	140
C 45 E	**	630	380	250	540	300	270	160
C 55 E		710	430	270	600	330	310	180
C 60 E		750	460	290	640	350	330	190

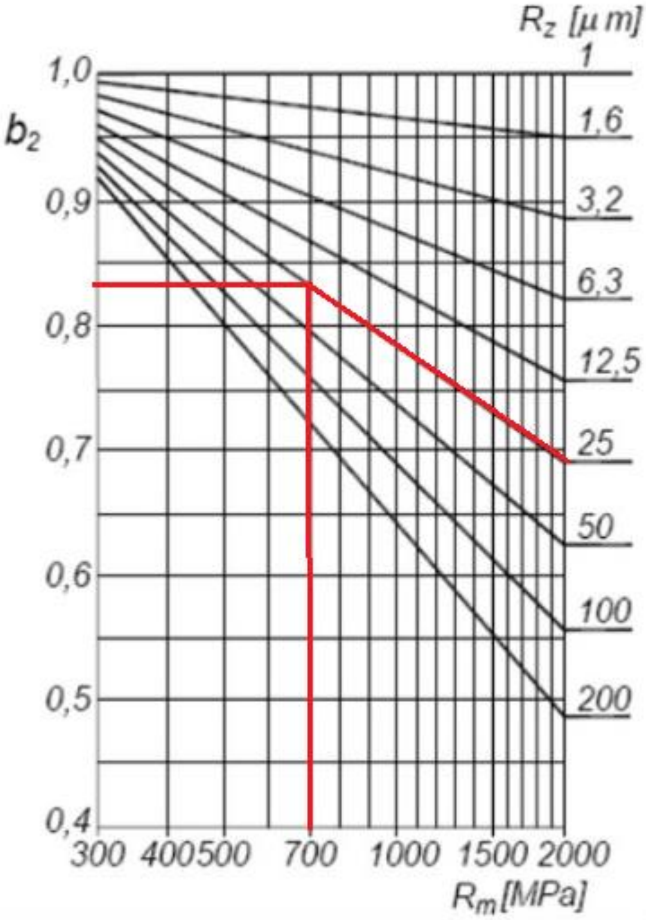
* Hot rolled, not alloyed steels (MSzE 10025)
** Hardenable carbon steels, tempered (MSzEN 10083)

The steel alloy used is C45E. The table below shows the size factor b_1 as a function of the shaft diameter. For us, $b_1 = 0.85$.



The table below allows us to obtain the surface roughness factor b_2 based on the surface roughness R_z and tensile strength R_m of C45E steel.

For C45E: R_z = 20 – 30μm and R_m = 600 - 800MPa. The exact values depend on the heat treatment and other strengthening techniques used; therefore, the midpoint of these values are used. The table below gives us a value of b_2 = 0.83.

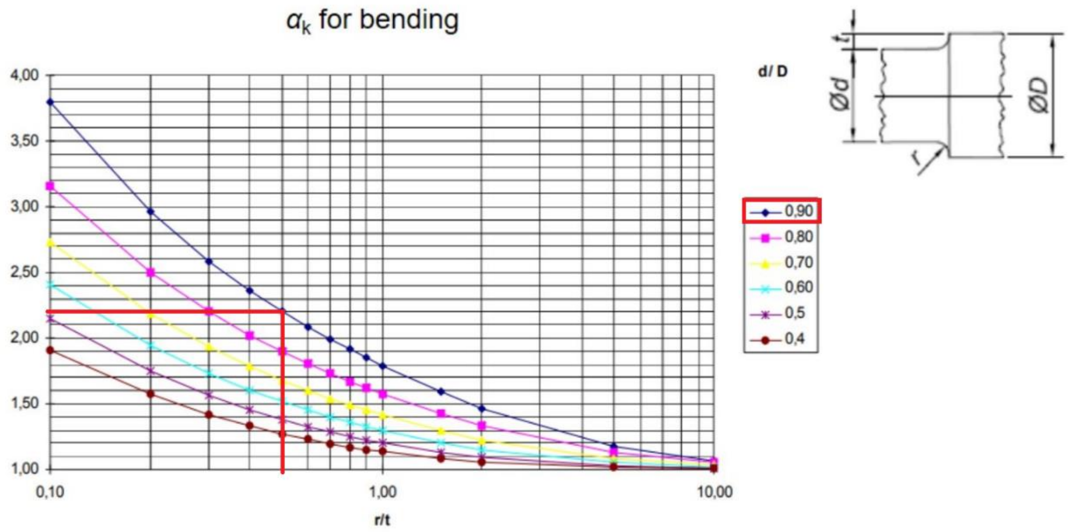


<u>Finding the modified stress concentration factor resulting from a narrowing of the shaft:</u>

The shaft will have a section with a diameter of D = 52 mm to allow the impeller to be secured. This creates a distance of t = 2mm between the perimeters of the two profiles. The sudden transition from the usual 48mm shaft diameter to this can create stress concentrations that can lead to cracking. This Adding a chamfer radius of r=0.5mm will ease some of these stresses. The stress concentration factor α_k for bending fatigue can be determined from the table below. Since torsional shear force is relatively small, torsional fatigue can be considered negligible. If fatigue failure is possible, it will occur due to normal stress far before it does due to shear. For C45E steel, the sensitivity factor η_k is approximately 0.6.

$$\frac{d}{D} = \frac{48}{52} = 0.92 \approx 0.9$$

$$\frac{r}{t} = \frac{1}{2} = 0.5$$



Therefore, α_k = 2.2.

The modified stress concentration factor is given by:

$$\beta_k \approx 1 + \eta_k(\alpha_k - 1) = 1 + 0.6 \times (2.2 - 1) = 1.72$$

The fatigue limit of C45E is 190MPa. Since the size, surface roughness and stress concentration factor change the risk of fatigue failure occurring, it is important to modify the fatigue limit by factoring appropriately to each of these characteristics. The modified fatigue limit of the material is given by:

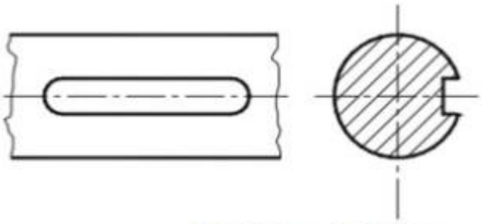
$$\sigma'_{v1} = \frac{b_1 b_2}{\beta_k}\sigma_v = \frac{0.85 * 0.83}{1.72} * 190\ MPa = 75.12\ MPa$$

Since the maximum normal stress on the material is 20.95MPa, the shaft is not expected to undergo fatigue from bending loads.

Note: the shaft contains another narrowing from 48 to 45mm, but since the ratio of diameters is less pronounced than the narrowing from 48 to 52mm, it will have a higher fatigue limit.

<u>Finding the modified stress concentration factor resulting from a keyway in the shaft:</u>

The shaft assembly will feature two key joints as a means of torque transmission from the pulley to the shaft and from the shaft to the impeller. The keyway will be produced using an end mill. For a keyway produced by an end mill, the following stress concentration factors apply:



	Bending	Torsion
R_m [MPa]	β_{hajl}	β_{csav}
400	1,6	1,4
500	1,7	1,5
600	1,9	1,6
800	2,2	1,7
1000	2,4	1,8
1300	2,6	1,9

Since R_m was taken as 700MPa, we need to interpolate between he two bending values to obtain the midpoint of the values corresponding to 600 and 800MPa. The midpoint value is β_{kb} = 2.05.

$$\sigma'_{v2} = \frac{b_1 b_2}{\beta_{\mathrm{kb}}} \sigma_v = \frac{0.85 * 0.83}{2.05} * 190\ MPa = 65.38\ MPa$$

This lowers the fatigue limit of the shaft, but it is still above any load encountered by it.

Final parts list:

The components of the fan drive assembly, as well as their standards, dimensions and materials are shown in the table below:

ITEM	QTY	PART	SIZE	STANDARD	MATERIAL
1	1	Retaining ring	ID1 13/16", OD 0.280", x0.062"	Acron Ring 1400-181	Copper, Alloy
2	1	Feather key	14 x 9 x 40	DIN 6885	Steel, Carbon
3	1	Shaft	D48 x 765	Custom	Steel, C45E
4	1	Feather key	14 x 9 x 28	DIN 6885	Steel, Carbon
5	1	Tapered bushing	ID49, OD100	SKF Q1 Series	Iron, Ductile
6	1	Non-locating roller bearing	ID50, OD90, x10	SKF NU 210 ECP	Steel
7	1	Locating roller bearing	ID50, OD90, x10	SKF NUP 210 ECP	Steel
8	2	Bearing housing	90 x 200 x 110	SKF SNL 510-608	Iron, Cast
9	4	Rubber seal	ID48, OD70.5, x11	SKF TXL 510	Rubber
10	2	Tapered adapter sleeve	ID48, OD51, x36	SKF H 210	Steel, Carbon
11	1	Hex nut	2-8 UN	ANSI B18.2.2	Steel, Mild
12	1	Industrial fan	D76	Horton G Clark FD033003	Plastic, Opaque Black
13	6	M5 hex bolt	M5 x 50	ANSI B18.2.3.2M	Steel, Mild
14	2	Bearing locating spacer	ID70, OD90, x5	SKF GS series	Steel, Galvanized
15	1	Assembly fixing frame	700 x 1000 x 400	Custom	Steel, Paint Finish, Dark Gray, Matte
16	4	M12 hex bolt	M12 x 70	ANSI B18.2.3.2M	Steel, Mild
17	4	M12 hex nut	M12	ANSI B18.2.4.5M	Steel, Mild
18	1	3-phase motor	ID45, OD100, x36	SMEM 6SH 100LB-4	*Varies*
19	1	Hex nut	1 3/4-5 UNC	ANSI B18.2.2	Steel, Mild
20	2	Pulley	ID45, OD100	SPZ 100X2	Aluminium 6061
21	4	M8 hex bolt	M8 x 40	ANSI B18.2.3.2M	Steel, Mild
22	2	V-belt	L1000	SPZ-type	Rubber
23	2	Sleeve lock washer	ID50, OD70	SKF H 210	Steel, Galvanized
24	2	Sleeve locknut	M50	SKF H 210	Steel, Galvanized

References:

1) ***Manual Of Assignment by Dr. Baka Zsolt***
2) ***Motor selection:*** Three-phase electric motor from stock immediately - Theiss Drive Technology
3) ***Belt and pulley selection selection:*** Technical Manual V-Belt Drives (optibelt.com)
4) ***Bearing House:*** SNL 211 - Bearing housings | SKF
5) ***Seal Selection:*** TSN 611 L - Bearing housing accessories | SKF
6) ***Bearing selection:*** 2211 ETN9 - Self-aligning ball bearings | SKF
7) ***Shaft selection:*** *https://www.skf.com/binaries/pub12/Images/0901d196803969af- 13186_1-EN-SKF-bearing-housings-and-roller-bearing-units_tcm_12- 315185.pdf*
8) ***Base selection(mounting): https://bmeedu-my.sharepoint.com/personal/ernozsoltbaka_edu_bme_hu/_layouts/15/onedrive.aspx?id=%2Fpersonal%2Fernozsoltbaka%5Fedu%5Fbme%5Fhu%2FDocuments%2FSUBJECTS%2FMachine%20Elements2%2FSeminar%2FFanPedia%2DThe%2DFan%2DBlower%2DEncyclopedia%2Dby%2DAerovent%2Epdf&parent=%2Fpersonal%2Fernozsoltbaka%5Fedu%5Fbme%5Fhu%2FDocuments%2FSUBJECTS%2FMachine% 20Elements2%2FSeminar&ga=1***