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VALIDATION DOCUMENT

Report Nr.:	721.06018
Verification Date:	30/07/2018
Manufacturer	ITALIAN TOP GEARS S.R.L. Via Martiri della Romania 4/C
Subject of Verification:	Shaft of Geared Hoisting machine ITG125
Year of manufacture:	2017
Place of Verification:	Vlasiou Gavriilidi 28, 54655 Thessaloniki, Greece
Documentation / Attachments:	1. Shaft assembly drawing 2. Detailed drawing of the shaft 3. Verification calculations for the shaft 4. Technical characteristics of bearings SKF 6314 and SKF 6316

This Validation Document has been carried out to the best knowledge and ability, our responsibility is limited to the exercise of due care and the results concern only the item verified.

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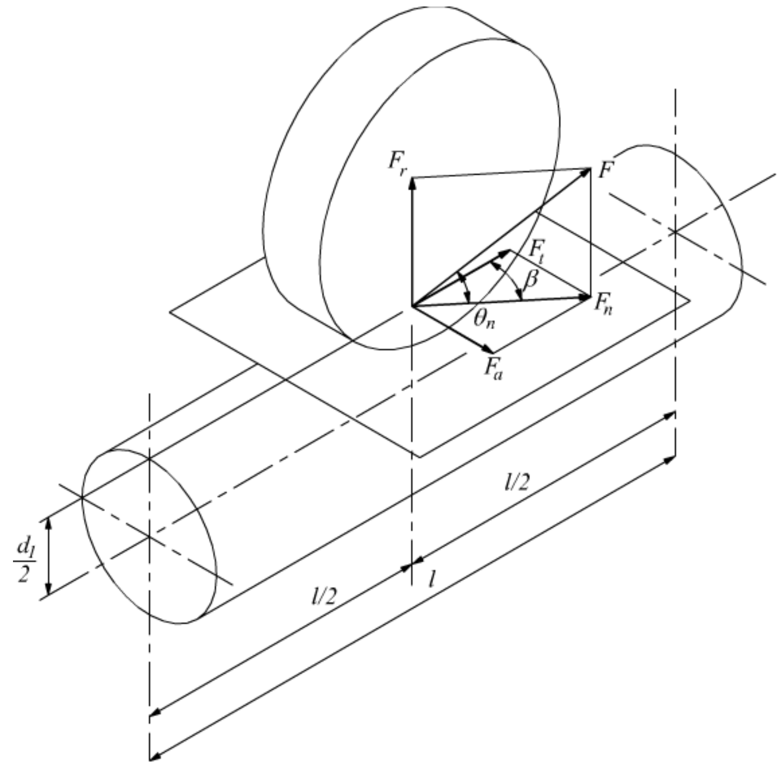
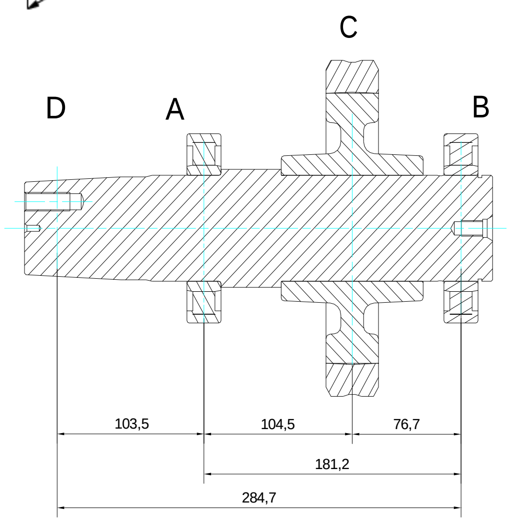
VERIFICATION OF SHAFT FOR GEARED HOISTING MACHINE ITG 125

The technical file on the geared Hoisting Machine ITG125 is composed of the following documents:

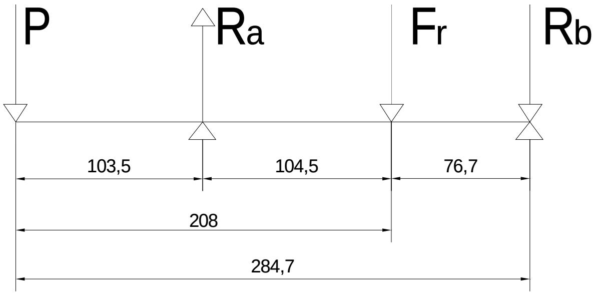
- **a.** Detailed drawing of the shaft;
- **b.** Verification calculations of the shaft;
- **c.** Technical characteristic of bearing SKF NU 214 EC





1. Verification of the shaft – Calculation of reaction forces on “xz” plane.

Plane “xz”



The shaft is mounted in the machine housing with the bearing SKF NU 214 EC at both points “A” and “B”. The shaft is stressed with the load P= 4500 kg (44145 N), located at one end at point “D” and with F_r, which is calculated as follows:

$$F_r = \frac{9549{,}2 \cdot P_u}{r_1 \cdot n}$$

Where:

$P_w = 5{,}5\ kW$, Power

$r_1 = 20\,mm$, Screw radius

$n = 1500\ r.p.m.$

$$F_r = \frac{9549{,}2 \cdot 5{,}5}{0{,}02 \cdot 1500} = \frac{52520{,}6}{30} = 1750{,}7\ N$$

Other data:

$D_p = 210\ mm$

$\vartheta = 20°$

$P = 3000\ kg$ (29430 N)

$\beta = 68{,}97°$

$$F_r = F \cdot \sin\theta = 1750{,}7 \cdot \sin 20 = 600\ N$$

$$F_n = F \cdot \cos\theta = 1750{,}7 \cdot \cos 20 = 1645{,}2\ N$$

$$F_t = F_n \cdot \sin\beta = 1645{,}2 \cdot \sin 68{,}97 = 1535{,}6\ N$$

$$F_a = F_n \cdot \cos\beta = 1645{,}2 \cdot \cos 68{,}97 = 590{,}6\ N$$

2. Reaction forces at the bearing

Equilibrium equation in y direction gives:

$$-P + R_a - F_r - R_b = 0 \quad (1)$$

3. Summing moments about point “A”

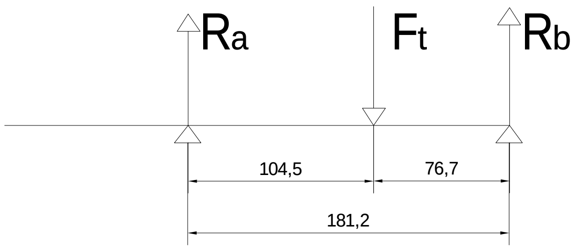
$$-P\cdot 103{,}5+F_r\cdot 104{,}5+R_b\cdot 181{,}2=0$$
$$R_b\cdot 181{,}2=29430\cdot 103{,}5-600\cdot 104{,}5$$
$$R_b\cdot 181{,}2=3046005-62700$$
$$R_b=\frac{2983305}{181{,}2}=16464{,}1\ N \qquad R_b=16464{,}1\ N$$

Replacing in (1) gives:

$$R_a=\ P+\ F_r+\ R_b$$
$$R_a=\ 29430+\ 600-16464{,}1=46494{,}1\ N \qquad R_a=46494{,}1\ N$$

4. Verification of the shaft – Calculation of reaction forces on “xy” plane.

Plane “xy”



5. Reaction forces at the bearing

Equilibrium equation in y direction gives:

$$R_a-F_t+R_b=0 \quad (2)$$

6. Summing moments about point “A”.

$$F_t\cdot 104{,}5-R_b\cdot 181{,}2=0$$
$$F_t\cdot 104{,}5=R_b\cdot 181{,}2$$
$$R_b\cdot 181{,}2=1535{,}6\cdot 104{,}5$$
$$R_b=\frac{160470{,}6}{181{,}2}=885{,}6\ N \qquad R_b=885{,}6\ N$$

Replacing in (2) gives:

$$R_a-F_t+R_b=0$$
$$R_a=1535{,}6-885{,}6=650N$$

7. The bending moment at point "A"on "xz" plane, is given by

$$M_{fA_{XZ}} = P \cdot 103{,}5 = 29490 \cdot 103{,}5 = 3046005 \;\; Nmm \qquad M_{fA} = 3046005 \; Nmm$$

8. The bending moment at point "A" on "xy" plane, is given by

$$M_{fA_{XY}} = R_a \cdot 104{,}5 = 600 \cdot 104{,}5 = 67925 \; Nmm \qquad M_{fA_{XY}} = 67925 \; Nmm$$

9. Equivalent reactions forces at the bearing "A"

$$R_{EQ_A} = \sqrt{R^2_{A_{XZ}} + R^2_{A_{XY}}} = \sqrt{46494{,}1^2 + 650^2} = \sqrt{2162123835} = 46498{,}6 \; N$$

10. Equivalent reactions forces at the bearing "B"

$$R_{EQ_B} = \sqrt{R^2_{B_{XZ}} + R^2_{B_{XY}}} = \sqrt{16464{,}1^2 + 885{,}6^2} = \sqrt{271850876{,}2} = 16487{,}9 \, N$$

11. Static verification of the shaft – Calculation of equivalent stress and static safety factor

11.1. Bending stress

The maximum bending stress moment M_{fB} is equal to the bending moment at point "A" on "XZ" plane:

$$M_{fA_{XZ}} = P \cdot 103{,}5 = 29490 \cdot 103{,}5 = 3046005 \; Nmm$$

The diameter of the shaft is equal to 70 *mm*, as specified in the drawing.

The bending stress σ_f is equal to: $\sigma_f = \frac{M_{fA}}{W_f} = \frac{32 \cdot M_{fA}}{\pi \cdot D^3} = \frac{32 \cdot 3046005}{\pi \cdot 70^3} = 90{,}5 \quad \frac{N}{mm^2}$

$$\sigma_f = 90{,}5 \, \frac{N}{mm^2}$$

11.2. Torsion stress

The maximum torque M_t transmittable from this mating, as declared by designer, is equal to:

$$P = M_t \cdot \omega \qquad M_t = \frac{P}{\omega} = \frac{5500}{157{,}1} = 35 \; Nm$$

Where

$$\omega = \frac{2\pi \cdot n}{60} = \frac{2\pi \cdot 1500}{60} = 157{,}1 \; \frac{rad}{\sec}$$

Taking into account a gear ratio of 1:55, without friction, the maximum torque, on shaft, is:

$$M_{t2} = M_t \cdot 55 = 35000 \cdot 55 = 1925000 \ Nmm$$

$$\tau = \frac{16 \cdot M_{t2}}{\pi \cdot D^3} = \frac{16 \cdot 1925000}{\pi \cdot 70^3} = \frac{30800000}{1077566{,}3} = 28{,}5 \ \frac{N}{mm^2} \qquad \tau = 28{,}5 \ \frac{N}{mm^2}$$

11.3. Static equivalent stress (Von Mises Criterion)

The equivalent static stress σ_{Eq} is:

$$\sigma_{Eq} = \sqrt{\sigma_f^2 + 3 \cdot \tau^2} = \sqrt{90{,}5^2 + 3 \cdot 28{,}6^2} = \sqrt{8190{,}25 + 2453{,}9} = \sqrt{10644{,}2} = 103{,}2 \ \frac{N}{mm^2}$$

$$\sigma_{Eq} = 103{,}2 \ \frac{N}{mm^2}$$

11.4. Safety factor

The shaft is made of 42CrMo4 steel.
This is alloy steel, quenched and tempered, with the following characteristics, as shown in the UNI EN 10083-3 Standard.

Steel designation: 42CrMo4
$R_m = 900 \ MPa$
$KV_{\min} = 35J$
$A_{\min} = 12\%$
$R_{e_{Min}} = 650 \ MPa$

The realized safety factor is given by the ratio between R_m and the equivalent stress σ_{Eq}, as calculated above:

$$n = \frac{R}{\sigma_{Eq}} = \frac{900}{103{,}2} = 8{,}7$$

For alternative (completely alternative cyclic) stress, the stress concentration effects must be considered. It should be verified that the equivalent stress is less than the fatigue limit as follows:.

$$\sigma_{Eq} < \sigma_{\lim}$$

From Goodman-Smith diagram, we have that $\sigma_{inv} = 440 \ \frac{N}{mm^2}$

The fatigue limit is given by $\sigma_{\lim} = \frac{b_1 \cdot b_2 \cdot \sigma_{inv}}{K_f}$ where:

$b_1 = 0{,}89$ Coefficient linked to the surface finish

$b_2 = 0{,}78$ Dimensional coefficient

$K_f = 1 + q \cdot (K_t - 1)$ Fatigue notch factor by Peterson

$q = 0{,}9$ Neuber Formula

$K_t = 1{,}9$ Shape factor, from table.

$K_f = 1 + q \cdot (K_t - 1) = 1 + 0{,}9 \cdot (1{,}9 - 1) = 1{,}81$

Therefore: $\sigma_{\lim} = \frac{b_1 \cdot b_2 \cdot \sigma_{inv}}{K_f} = \frac{0{,}89 \cdot 0{,}78 \cdot 440}{1{,}81} = 168{,}76 \frac{N}{mm^2}$

$$\sigma_{Eq} = 103{,}2 < \sigma_{\lim} = 168{,}76 \frac{N}{mm^2}$$

$$n = \frac{\sigma_{\lim}}{\sigma_{Eq}} = \frac{168{,}76}{103{,}2} = 1{,}63$$

12. Shaft fatigue failure: Gough – Pollard Criterion

12.1. Calculation of the torsional fatigue limit $\tau_{\lim}$

$$\tau_{\lim} = 0{,}6 \cdot \mathrm{Re}_{\min} = 0{,}6 \cdot 650 = 390 \frac{N}{mm^2}$$

12.2. According to Gough-Pollard criterion should be verified

$$\sigma_{GP} = \sqrt{\sigma_f^2 + \left(\frac{\sigma_{Lim}}{\tau_{Lim}}\right)^2 \cdot \tau_{Max}^2} < \frac{\sigma_{Lim}}{n}$$

Where:

n = 1,5

$\sigma_f = 90{,}5 \frac{N}{mm^2}$

$\sigma_{Lim} = 168{,}76 \frac{N}{mm^2}$

$\tau_{Lim} = 390 \frac{N}{mm^2}$

$$\tau_{Max} = 28{,}5 \quad \frac{N}{mm^2}$$

$$\sigma_{GP} = \sqrt{\sigma_f^2 + \left(\frac{\sigma_{Lim}}{\tau_{Lim}}\right)^2 \cdot \tau_{Max}^2} < \frac{\sigma_{Lim}}{n}$$

$$\sigma_{GP} = \sqrt{90{,}5^2 + \left(\frac{168{,}76}{390}\right)^2 \cdot 28{,}5^2} < \frac{168{,}76}{1{,}5}$$

$$\sigma_{GP} = \sqrt{8190{,}25 + (0{,}187) \cdot 817{,}96} = \sqrt{8343{,}2} = 91{,}3 < 112{,}5 \frac{N}{mm^2}$$

Therefore $\sigma_{GP} = 91{,}3 < \sigma_{GP} = 112{,}5 \frac{N}{mm^2}$

Which is valid.