



# QMSCERT®



## VALIDATION DOCUMENT

Report Nr.: 140.06013	Verification Date: 06/06/2013	Page: 1 of 5
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<b>Manufacturer</b>	<b>ITALIAN TOP GEARS S.R.L.</b> Via Giovanni Falcone 8/a 42020 Borzano di Albinia(RE) - Italy
<b>Subject of Verification</b>	<b>Calculation for main shaft of Gearless Lift Machine ITG 160</b>
<b>Year of fabrication</b>	<b>2013</b>
<b>Place of Verification</b>	<b>Korai 35 Kalamaria 55132 Thessaloniki Greece</b>
<b>Documentation / Attachments</b>	<b>DRAWINGS:</b> - Assembly drawing of the gearless Lift Machine ITG 160 -Detailed drawing of the main shaft. <b>CALCULATIONS</b> -Static verification of the main shaft. -Technical characteristics of bearings SKF 22214 E & SKF 22218 E -F.M.E. for the shell ( main body ).

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



Verified by

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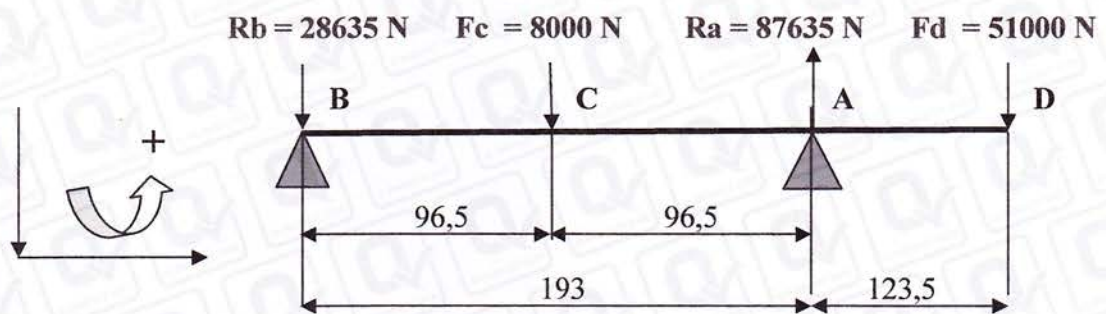


## VALIDATION DOCUMENT

Report Nr.: 140.06013	Verification Date: 06/06/2013	Page: 2 of 5
-----------------------	-------------------------------	--------------

### 1. VERIFICATION OF MAIN SHAFT FOR GEARLESS LIFT MACHINE ITG 160

According to the shaft drawing 02.160.002.00100 the most stressed diameter is 90 mm nearest to the bearing SKF 22218 E (90x160x40) of support beam A.



The shaft is bound to the bearings SKF 22218 E (90x160x40) at point (A) & SKF 22214 E (70x125x31) at point (B), and is stressed with the loads (according to the designer)

- 8000 N at the point C
- 51000 N at the point D

#### A. Equation of equilibrium to vertical translation

$$- R_a + R_b + F_c + F_d = 0 \quad (1)$$

#### 1. Equation of equilibrium of moments at point A

$$- R_a + R_b + 8000 + 51000 = 0 \quad \Rightarrow$$

$$59000 \text{ N} + R_b = R_a \quad (2)$$

$$R_b \times 193 \text{ mm} + F_c \times 96,5 \text{ mm} - F_d \times 90 \text{ mm} = 0 \quad \Rightarrow$$

$$R_b \times 193 \text{ mm} + 8000 \text{ N} \times 96,5 \text{ mm} - 51000 \text{ N} \times 123,5 \text{ mm} = 0 \quad \Rightarrow$$

$$R_b \times 193 \text{ mm} - 5526500 \text{ Nmm} = 0 \quad \Rightarrow \quad R_b = 5526500 / 193 = 28635 \text{ N}$$

From (1) & (2)  $R_a = 59000 \text{ N} + 28635 \text{ N} = 87635 \text{ N}$

Results:

- $R_a = 87635 \text{ N}$
- $R_b = 28635 \text{ N}$
- $F_d = 51000 \text{ N}$
- $F_c = 8000 \text{ N}$





## VALIDATION DOCUMENT

Report Nr.: 140.06013

Verification Date: 06/06/2013

Page: 3 of 5

### 2. STATIC VERIFICATION OF MAIN SHAFT

#### a. Solicitation of Bending ( bending stress )

The maximum bending moment resulted from:

$$M_f = F_d \times 123,5 \text{ mm} = 51000\text{N} \times 123,5 \text{ mm} = 6298500 \text{ Nmm} \Rightarrow M_f = 6298500 \text{ N mm}$$

The most stressed diameter of shaft is  $D = 90 \text{ mm}$

The solicitation of bending resulted from:  $\sigma_f = \frac{M_f}{W_f}$

$$\sigma_f = \frac{M_f}{W_f} = \frac{6298500 \text{ Nmm}}{\frac{\pi D^3}{32}} = \frac{6298500}{71569,4} = 88 \text{ Mpa} \Rightarrow \sigma_f = 88 \text{ Mpa}$$

#### b. Solicitation of torsion. ( torsion strength )

The maximum torsion moment offered by gearless lift machine according to the designer is:

$$M_t = 2700000 \text{ Nmm}$$

The solicitation of torsion resulted from:  $\tau = \frac{M_t}{W_t}$

$$\tau = \frac{M_t}{W_t} = \frac{2700000}{\frac{\pi D^3}{16}} = \frac{2700000}{143139} = 18,86 \text{ Mpa} \Rightarrow \tau = 18,86 \text{ Mpa}$$

#### c. Equivalent static solicitation.

The equivalent static solicitation is given by the formulate:

$$\sigma_{EQ} = \sqrt{(\sigma_f)^2 + 3(\tau^2)}$$

$$\sigma_{EQ} = \sqrt{(\sigma_f)^2 + 3(\tau^2)} = \sqrt{(88)^2 + 3(18,86)^2} \Rightarrow$$

$$\sigma_{EQ} = \sqrt{7744 + 1067} = \sqrt{8811} = 93,87 \text{ Mpa} \Rightarrow \sigma_{EQ} = 93,87 \text{ Mpa}$$





## VALIDATION DOCUMENT

Report Nr.: 140.06013	Verification Date: 06/06/2013	Page: 4 of 5
-----------------------	-------------------------------	--------------

### d. Safety factor

Technical characteristics of shaft material.( EN 10083 -3 )

- Steel: 42CrMo4
- Re min: 650Mpa
- R m: 900Mpa
- A min: 12%
- Z min: 50%
- KV(min): 35J

The safety factor for rupture according to the R<sub>m</sub> is:

$$\eta = \frac{R}{\sigma_{EQ}} \implies \eta = \frac{900}{93,87} = 9,6 \implies \eta = 9,6$$

Alternative verification:

$$\sigma_{AMM} = \frac{Re}{1,5} \implies \sigma_{AMM} = \frac{650}{1,5} = 433,3\text{Mpa} \implies \sigma_{AMM} = 433,3\text{Mpa}$$

where  $\sigma_{AMM}$  is the allowable stress

According to formulate:

$$\sigma_{EQ} < \sigma_{AMM} \implies 93,87 \text{ Mpa} < 433,3 \text{ Mpa}$$

### e. Fatigue verification (Gough – Pollard principle)

#### 1. Calculation of deflection limit effort $\sigma_{LIM}$

$$\sigma_{LIM} = \sigma_{FAF} \cdot b_1 \cdot b_2 / Kf$$

$$\sigma_{FAF} = \text{Solicitation of deflection limit} = \frac{Rm}{2} = 450\text{Mpa}$$

$$b_1 = \text{Coefficient referring to surface quality} = 0,89$$

$$b_2 = \text{Coefficient referring to size} = 0,78$$

Kf = Fatigue strength reduction factor ( Peterson equation )

$$Kf = 1 + q (Kt-1) \text{ where } q = 0,85 \text{ (Neuber)}, Kt = 1,15$$

$$Kf = 1,13$$

$$\sigma_{LIM} = 450 \cdot 0,78 \cdot 0,89 / 1,13 = 276,5 \text{ Mpa} \implies \sigma_{LIM} = 276,5 \text{ Mpa}$$





## VALIDATION DOCUMENT

Report Nr.: 140.06013	Verification Date: 06/06/2013	Page: 5 of 5
-----------------------	-------------------------------	--------------

### 2. Calculation of torsion limit effort $\tau_{LIM}$

$$\tau_{LIM} = 0,6 \times Re \text{ min}$$

$$Re \text{ min} = 650 \text{ Mpa}$$

$$\tau_{LIM} = 0,6 \times Re \text{ min} = 0,6 \cdot 650 = 390 \text{ Mpa} \Rightarrow \tau_{LIM} = 390 \text{ Mpa}$$

### 3. Gough – Pollard principle $\sigma_{GP}$

$$\sigma_{GP} = \sqrt{\sigma f^2 + \left(\frac{\sigma_{LIM}}{\tau_{LIM}}\right)^2 \cdot \tau^2 \text{ max}} \leq \frac{\sigma_{LIM}}{n} \Rightarrow \sigma_{GP} \leq \frac{\sigma_{LIM}}{n}$$

Where  $\eta = 1,5$

$$\sigma f = 88 \text{ Mpa}$$

$$\sigma_{LIM} = 276,5 \text{ Mpa}$$

$$\tau_{LIM} = 390 \text{ Mpa}$$

$$\tau_{MAX} = 18,86 \text{ Mpa}$$

$$\sigma_{GP} = \sqrt{(88)^2 + \left(\frac{276,5}{390}\right)^2 \cdot 18,86^2} \leq \frac{276,5}{1,5}$$

$$\sigma_{GP} = \sqrt{3778,5 + (0,5 \times 355,6)} = 74,54 \leq 184,3 \Rightarrow \text{Acceptable}$$



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