

# **CALCULATION ON PISTON, ROD, CRANKSHAFT ASSEMBLIES**

## **ADVANCED ENGINE DESIGN**

### **SUPERVISION**

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MEng Motorcycle Engineering

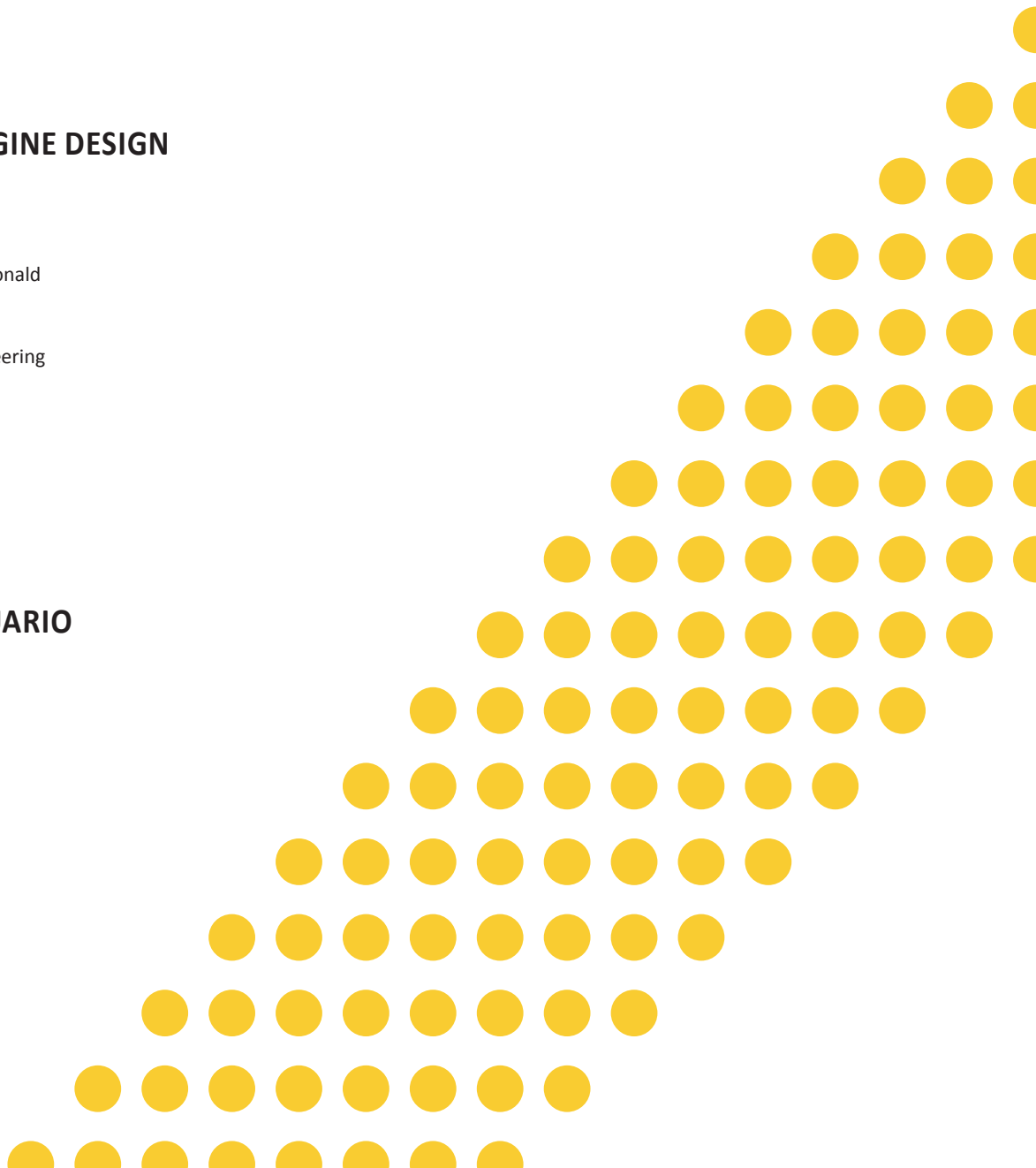
Level 6

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# 1 INTRODUCTION

The following assessment was developed for the module of Advanced Engine Design of the third year – level 6 of the Motorcycle Engineering Degree and represents the second evaluation component of the module.

The aim of the presented work is to analyse and provide fatigue safety factors for infinite life of the major structural components of the standard engine, through several theoretical and practical methods lectured in classes and sourced from other authors.

With the results obtained from the previous assignment, several calculations will be developed to evaluate the gudgeon pin, conrod, conrod bolts and crankshaft performance and the fatigue safety factors of those, using Goodman diagrams in the critical sections.

Success will be measured by the reasonability and accuracy of the results achieved, allowing the identification of fatigue safety factors used in the design of several critical engine parts through the use of a reverse engineering process.

## 2 GUDGEON PIN – ANALYSIS

In this Chapter the author is proposed to plot a Goodman diagram to identify the baseline gudgeon pin fatigue safety factor used by the manufacturer. To do so, the maximum net compressive force and maximum net tensile force determined in the previous assignment were used, in order to calculate the mean stress and the range of stresses for the gudgeon pin that will then be used to plot what was previously proposed.

### 2.1 METHODOLOGY

This methodology will present, by order, all the calculations performed to achieve the Goodman diagram and the gudgeon pin fatigue safety factor. Similar methodology will be implemented in further Chapters when needed to calculate the fatigue safety factor for other parts such as the conrod, conrod bolts or crankshaft.

**Forces acting in one side of the gudgeon pin (N):**

$$F_{Net} = F_{Gas} - F_{Inertia}$$

- because it will only be analyse one side of the gudgeon pin, the force used is  $\frac{F_{Net}}{2}$  (N);

$$F_{Inertia\ Pin} = Mass_{Pin} \cdot acc$$

Where,

- $Mass_{Pin}$  is the mass of the gudgeon pin sticking out (kg);

$$Mass_{Pin} = Length \cdot \pi \cdot (R_2 - R_1) \cdot \rho$$

- $acc$  is the gudgeon pin acceleration ( $m/s^2$ );

$$acc = R \cdot \omega^2 \cdot \cos\theta + \frac{R}{L} \cdot R \cdot \omega^2 \cdot \cos 2\theta$$

**Stresses on gudgeon pin ( $N/mm^2$ ):**

$$\sigma_{Bending} = \frac{M \cdot y}{I}$$

Where,

- $\sigma_{Bending}$  is the bending stress at a certain point in the gudgeon pin ( $N/mm^2$ );
- $M$  is the bending moment at a certain point in the gudgeon pin ( $Nmm$ );

$$M = \frac{F_{Net}}{2} \cdot L_1 - F_{Inertia Pin} \cdot L_2$$

- $L_1$  is the distance between where the force from the piston is being applied in the gudgeon pin to the conrod surface (mm);
- $L_2$  is the distance between the cg of gudgeon pin sticking out to the conrod surface (mm);
- $y$  is the distance from the Neutral Axis to the outside surface of the gudgeon pin (mm);
- $I$  is the Second Moment of Area of the gudgeon pin ( $mm^4$ );
  - $I = \frac{\pi}{64} \cdot (D_1^4 - D_2^4)$

$$\tau = \frac{\frac{F_{Net}}{2} - F_{Inertia Pin}}{A}$$

Where,

- $\tau$  is the Shear stress applied in the gudgeon pin ( $N/mm^2$ );
- $A$  is the cross-sectional area ( $mm^2$ );
  - $A = \frac{\pi}{4} \cdot (D_1^2 - D_2^2)$

Combining the previously calculated stresses, it will end up as following ( $N/mm^2$ ):

$$\sigma_{Resultant} = \sqrt{\sigma_{Bending}^2 + (2 \cdot \tau)^2}$$

To calculate the fatigue at a certain point in the gudgeon pin, two moments where the maximum stress values vary - Alternating Stress - needed to be defined. For the scope of this assignment it was defined that these moments were when the piston was at TDC during the Power (firing) and Exhaust stroke.

It can then be calculated the following:

$$Range\ of\ Stress = abs(\sigma_{Max}) + abs(\sigma_{Min})$$

$$\sigma_{Mean} = \frac{(\sigma_{Max}) + (\sigma_{Min})}{2}$$

$$Allowable\ Range = 2 \cdot EL$$

Where,

- EL is the endurance limit of the part's material being analysed (steel);

$$EL = UTS \cdot 0,45$$

$$Allowable\ Range\ at\ Mean = 2 \cdot EL \cdot \left(1 - \frac{\sigma_{Mean}}{UTS}\right)$$

$$FSF = \frac{Allowable\ Range\ at\ Mean}{Range\ of\ Stress}$$

Where,

- FSF is the Fatigue Safety Factor, where it is pretended to be a value superior than 1, indicating that the part will have an infinite life during its working cycles;

## 2.2 RESULTS

Table 1. Gudgeon Pin Specifications to calculate FSF

| Gudgeon Pin Specifications    |        | Units             |
|-------------------------------|--------|-------------------|
| D1 - Outer Diameter           | 20,8   | mm                |
| D2 - Inner Diameter           | 11,8   | mm                |
| Gudgeon Pin Length            | 45,8   | mm                |
| I - 2nd Moment of Area        | 8236,4 | mm <sup>4</sup>   |
| y - Neutral axis              | 10,4   | mm                |
| Cross Sectional Area          | 230,4  | mm <sup>2</sup>   |
| Piston gap for conrod         | 22,5   | mm                |
| Conrod width                  | 12,0   | mm                |
|                               | 18,8   | mm                |
| L1<br>(Piston force - conrod) | 11,1   | mm                |
|                               | 7,7    | mm                |
| L2<br>(Gudgeon cg - conrod)   | 7,7    | mm                |
|                               | 7,7    | mm                |
| Mass Sticking Out             | 25,0   | grams             |
| UTS                           | 1000,0 | N/mm <sup>2</sup> |
| Endurance Limit               | 450    | N/mm <sup>2</sup> |

The table above, Table 1, presents all the real values used to feed the formulas from the previous chapter to obtain the results below.

In order to create a maximum scenario for the gudgeon pin to perform, the maximum value of Net Compressive Force of the engine was picked, which occurs at 5700 RPM and the maximum value of Net Tensile Tensile Force that occurs at 6000 RPM.

Table 2. Maximum Stress values on Gudgeon Pin

|                                      | MAX VALUES FOR GUDGEON PIN |         |                   |
|--------------------------------------|----------------------------|---------|-------------------|
|                                      | Firing                     | Exhaust | Units             |
| NET FORCE                            | -38 537                    | 8 724   | N                 |
| INERTIA FORCE                        | 473                        | 524     | N                 |
| STRESS                               |                            |         |                   |
| BENDING STRESS (N/mm <sup>2</sup> )  | 264,9                      | -66,1   | N/mm <sup>2</sup> |
| SHEAR STRESS (N/mm <sup>2</sup> )    | 81,6                       | -21,2   | N/mm <sup>2</sup> |
| COMBINED STRESS (N/mm <sup>2</sup> ) | 311,1                      | -78,5   | N/mm <sup>2</sup> |
|                                      |                            |         |                   |
| RANGE OF STRESSES                    | 389,6                      |         | N/mm <sup>2</sup> |
| MEAN STRESS                          | 116,3                      |         | N/mm <sup>2</sup> |

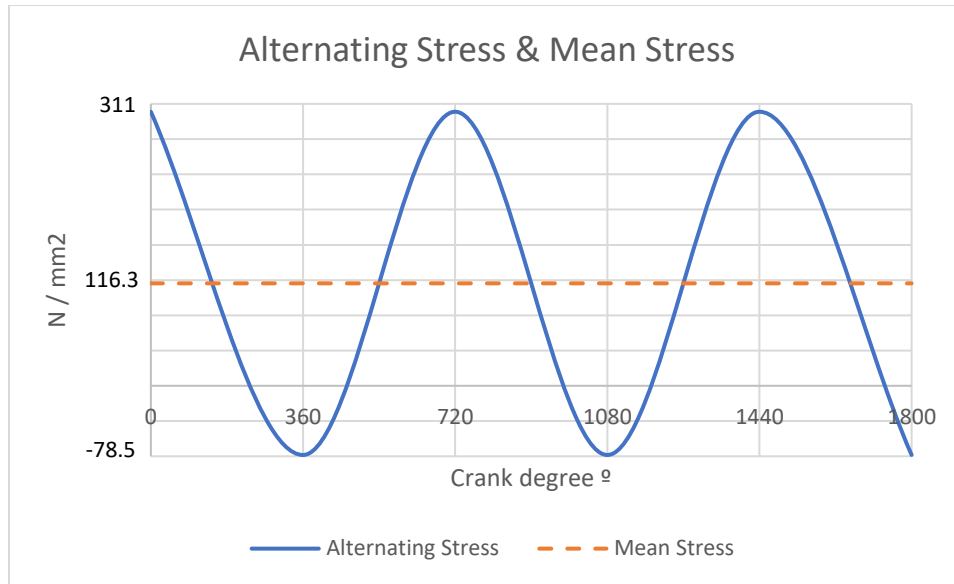


Figure 1. Range of Stress and Mean Stress on Gudgeon Pin

In the figure above, Figure 1, it can be observed how the stress varies in the gudgeon pin when the engine is performing at its maximum capabilities.

To calculate the Fatigue Safety Factor, the fatigue test uses the MEAN STRESS as zero reference and the Endurance Limit (EL) as the maximum amplitude of stress that the gudgeon pin can go to last forever – Figure 2.

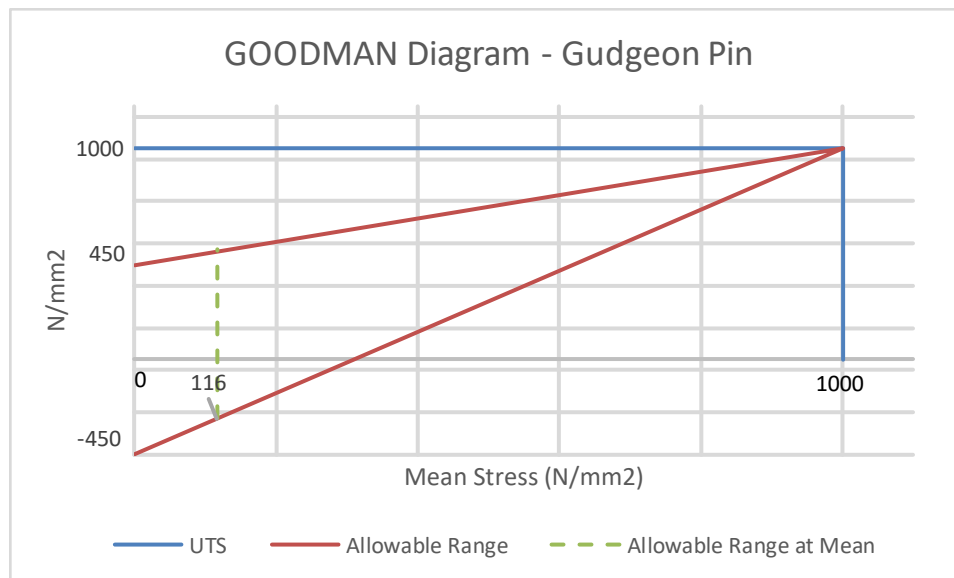


Figure 2. Goodman Diagram for Gudgeon Pin

| GUDGEON PIN - STRESS ANALYSIS |       |                   |
|-------------------------------|-------|-------------------|
|                               |       |                   |
| RANGE OF STRESSES             | 389,6 | N/mm <sup>2</sup> |
| MEAN STRESS                   | 116,3 | N/mm <sup>2</sup> |
|                               |       |                   |
| ENDURANCE LIMIT               | 450,0 | N/mm <sup>2</sup> |
| ALLOWABLE RANGE AT MEAN       | 795,3 | N/mm <sup>2</sup> |
|                               |       |                   |
| FATIGUE SAFETY FACTOR         | 2,04  |                   |

*Table 3. Gudgeon Pin – FSF*

As it can be observed in the figures above, Figure 2 and Table 3, a Goodman Diagram to determine the baseline gudgeon pin fatigue safety factor was plotted. The value for the fatigue safety factor obtained was 2.04, which means that the gudgeon pin was safely/over engineered by the manufacturer, once any FSF over 1 means infinite life for the part.

### 3 CONROD – ANALYSIS

In this Chapter the author is proposed to plot a Goodman diagram for each three rod sections shown in the brief and determine the standard “unmodified” fatigue safety factors. Once more, the maximum net compressive force and maximum net tensile force determined in the previous assignment were used. To make it easier to measure some values, such as the cross-sectional area and mass of each individual section being analysed, the conrod was perfectly copied into a CAD software.

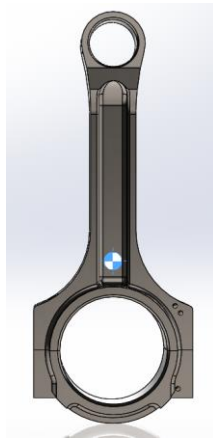


*Figure 3. CAD of conrod being analysed*

### 3.1 METHODOLOGY

The methodology to determine the Fatigue Safety Factor for Section A is similar in most aspects to the one already explained on Chapter 2.1.

Therefore, only the differences will be highlighted in this Chapter.

- Cross sectional areas being analysed were calculated from the CAD software used;
- $EL = \frac{UTS \cdot 0.45}{K_t}$ , where  $K_t$  is the stress concentration factor and comes from (Pilkey, 1997):

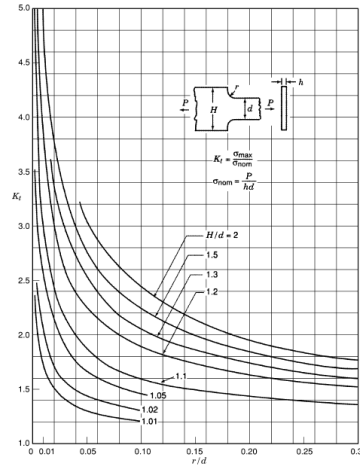


Figure 4. Stress concentration factor

- For Section C, in the conrod cap, below the conrod cg, the Inertia forces are equal to:

$$F_{Inertia} = Mass_{Rec} \cdot (R \cdot \omega^2 \cdot \cos\theta + \frac{R}{L} \cdot R \cdot \omega^2 \cdot \cos 2\theta) + Mass_{Rot} \cdot R \cdot \omega^2$$

Where,

- $Mass_{Rec}$  is the total reciprocating (Kg);

$$Mass_{Rec} = Mass_{Piston} + Mass_{Gudgeon\ pin} + 1/3 Mass_{Conrod}$$

- $Mass_{Rot}$  is the total rotational mass experiencing centrifugal force (Kg);

$$Mass_{Rot} = 2/3 Mass_{Conrod}$$

## 3.2 RESULTS

Table 4. Conrod specifications to calculate FSF

| CONROD SPECIFICATIONS |       | Units             |
|-----------------------|-------|-------------------|
| UTS                   | 750,0 | N/mm <sup>2</sup> |
| Endurance Limit       | 337,5 | N/mm <sup>2</sup> |
| A2 sectional area     |       |                   |
| Area                  | 158,2 | mm <sup>2</sup>   |
| Kt                    | 1,60  |                   |
| B sectional area      |       |                   |
| Area                  | 274,0 | mm <sup>2</sup>   |
| Kt                    | 1,75  |                   |
| C sectional area      |       |                   |
| Area                  | 157,2 | mm <sup>2</sup>   |
| Kt                    | 1,65  |                   |
| Mass                  |       |                   |
| Conrod                | 406,4 | grams             |
| Piston Assembly       | 344,2 | grams             |

### 3.2.1 SECTION A

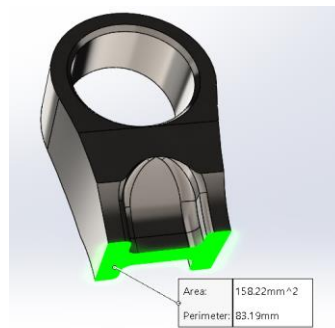


Figure 4. Conrod - Section A

Table 5. Section A - FSF

|               | SECTION A |         |                   |
|---------------|-----------|---------|-------------------|
|               | FIRING    | EXHAUST | Units             |
| Gas Force     | -46 863   | -491    | N                 |
| Inertia Force | 10 048    | 10 048  | mm <sup>2</sup>   |
| Stress        | -232,7    | 60,4    | N/mm <sup>2</sup> |

| STRESS ANALYSIS         |                         |
|-------------------------|-------------------------|
| RANGE OF STRESSES       | 293,1 N/mm <sup>2</sup> |
| MEAN STRESS             | -86,1 N/mm <sup>2</sup> |
| FINAL ENDURANCE LIMIT   | 210,9 N/mm <sup>2</sup> |
| ALLOWABLE RANGE AT MEAN | 470,3 N/mm <sup>2</sup> |
| FATIGUE SAFETY FACTOR   | 1,60                    |

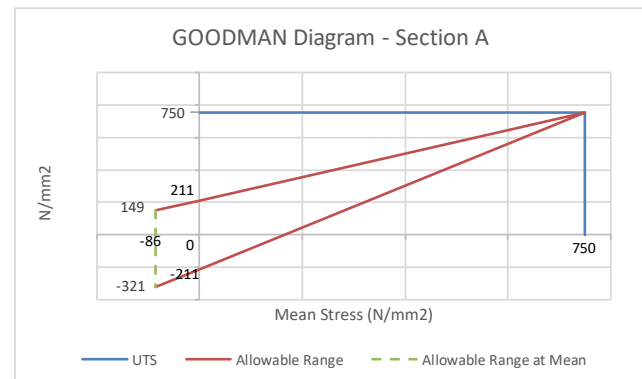


Figure 5. Goodman diagram - Section A

As it can be observed in the table and figure above, table 5 and figure 5, a Goodman Diagram to determine the baseline for the conrod section A fatigue safety factor was plotted. The value for the fatigue safety factor obtained was 1,6, which means that in that specific section the conrod was safely engineered by the manufacturer, once any FSF over 1 means infinite life for the part.

### 3.2.2 SECTION B

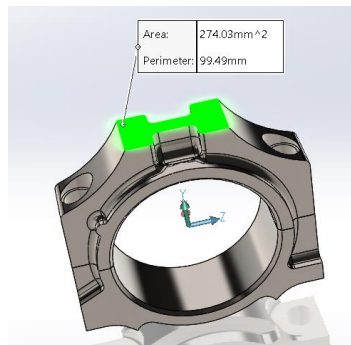


Figure 6. Conrod - Section B

Table 6. Section B - FSF

|               | SECTION B |         | Units |
|---------------|-----------|---------|-------|
|               | FIRING    | EXHAUST |       |
| Gas Force     | -46 863   | -491    | N     |
| Inertia Force | 10 048    | 10 048  | mm2   |
| Stress        | -232,7    | 60,4    | N/mm2 |

| STRESS ANALYSIS         |             |
|-------------------------|-------------|
| RANGE OF STRESSES       | 293,1 N/mm2 |
| MEAN STRESS             | -86,1 N/mm2 |
| FINAL ENDURANCE LIMIT   | 192,9 N/mm2 |
| ALLOWABLE RANGE AT MEAN | 430,0 N/mm2 |
| FATIGUE SAFETY FACTOR   | 1,47        |

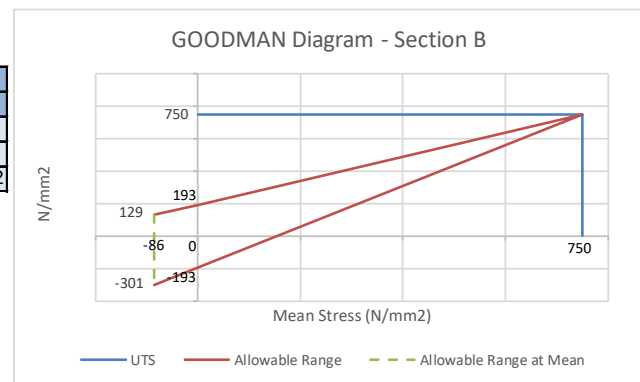


Figure 7. Goodman diagram – Section B

Once more the value for the fatigue safety factor obtained was superior to 1, meaning infinite life for the part. The FSF value for section B, 1.47, was inferior to the one obtained in section A due to the bigger difference in the step observed in the conrod.

### 3.2.3 SECTION C

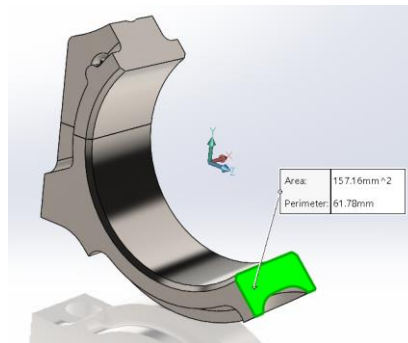


Figure 8. Conrod - Section C

Table 7. Section B - FSF

|               | SECTION C |         |       |
|---------------|-----------|---------|-------|
|               | FIRING    | EXHAUST | Units |
| Gas Force     | -23 431   | -245    | N     |
| Inertia Force | 6 443     | 6 443   | mm2   |
| Stress        | -108,1    | 39,4    | N/mm2 |

| STRESS ANALYSIS         |             |
|-------------------------|-------------|
| RANGE OF STRESSES       | 147,5 N/mm2 |
| MEAN STRESS             | -34,3 N/mm2 |
| FINAL ENDURANCE LIMIT   | 205 N/mm2   |
| ALLOWABLE RANGE AT MEAN | 427,8 N/mm2 |
| FATIGUE SAFETY FACTOR   | 2,90        |

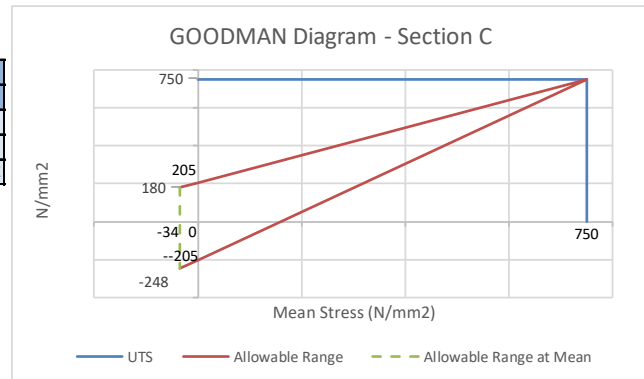


Figure 9. Goodman diagram - Section C

As it can be observed in the table and figure above, table 7 and figure 9, a Goodman Diagram to determine the baseline for the conrod section C fatigue safety factor was plotted. The value for the fatigue safety factor obtained was 2,90, which means that in that specific section the conrod was over engineered by the manufacturer, once any FSF over 1 means infinite life for the part.

## 4 CONROD CAP & CONROD BOLT – ANALYSIS

### 4.1 RATIO OF THE BOLT STIFFNESS TO THE CAP-ROD STIFFNESS

$$K_b = \frac{E_b \cdot A_b}{L}$$

$$K_c = \frac{E_c \cdot A_c}{L}$$

Where,

- $K_b$  is the bolt stiffness;
  - $E_b$  is the bolt's material Young's Modulus;
  - $A_b$  is the bolt average area;
- $K_c$  is the conrod cap stiffness;
  - $E_c$  is the conrod's material Young's Modulus;
  - $A_c$  is the rod cap interface area;

So, the ratio of cap-rod stiffness to the bolt stiffness is:

$$Ratio = \frac{K_c}{K_b} = \frac{E_c \cdot A_c}{E_b \cdot A_b} = \frac{170000 \cdot 154,18}{210000 \cdot 41,28} = 3,02$$

### 4.2 BOLT STIFFNESS

$$K_b = \frac{E_b \cdot A_b}{L}$$

Where,

- $K_b$  is the bolt stiffness;
  - $E_b$  is the bolt's material Young's Modulus;
  - $A_b$  is the bolt average area;
  - $L$  is the distance from the head of the bolt to the second engaged thread;

$$K_b = \frac{210000 \cdot 41,28}{24} = 361\,200 \text{ N/mm}$$

### 4.3 MAXIMUM VALUE OF ALTERNATING LOAD ON THE BOLT

$$Load_{Max} = Load_{Min} + Load_{Extra}$$

Where,

- $Load_{Min} = 0,9.Yield . A_{Bolt}$

$$Load_{Min} = 810 . 41,3 = 33\,439\, N$$

- $Load_{Extra} = \frac{1}{\left(1 + \frac{K_c}{K_b}\right)} \cdot F_{Total\, Inertia\, on\, bolt}$

- $F_{Total\, Inertia\, on\, bolt} = Mass_{Rec} \cdot (R \cdot \omega^2 \cdot \cos\theta + \frac{R}{L} \cdot R \cdot \omega^2 \cdot \cos 2\theta) + Mass_{Rot} \cdot R \cdot \omega^2$

$$Load_{Extra} = \frac{1}{(1 + 3,02)} \cdot 12\,733 = 3\,165\, N$$

Because there are two bolts in the conrod cap:

$$Load_{Max} = 33\,439 + \frac{3\,165}{2} = 35\,021\, N$$

#### 4.4 BOLT TIGHTENING TORQUE

For reusable bolts it is needed to tighten to 90% yield and as it was seen in the previous chapter:

$$Load_{Min} = 0,9.Yield . A_{Bolt} = 810 . 41,3 = 33\,439\, N$$

When a bolt is tightened, there is a combination of axial and torsional stress. Normally when tightened to 90% of Yield, the axial stress is approximately 60% of Yield.

$$Force_{Clamping\,Bolt} = 0,6.Yield . A_{Bolt} = 20\,063\, N$$

Because in a bolt there is two areas of friction:

$$Torque_{Bolt} = Torque_{Threads} + Torque_{Head}$$

$$Torque_{Bolt} = Force_{Clamping\,Bolt} (\mu_{Threads} . r_{Threads} + \mu_{Head} . r_{Head})$$

$$Torque_{Bolt} = 20\,063 . \left( 0,18 . \frac{7,25}{2} + 0,12 . \frac{\frac{13+8}{2}}{2} \right) = 25,7\, Nm$$

#### 4.5 MEAN STRESS AND RANGE OF STRESS OF ROD BOLT

$$\sigma_{Max} = \sigma_{Min} + \sigma_{Extra}$$

Where,

- $\sigma_{Min} = 0,9.Yield = 810 \text{ N/mm}^2$
- $\sigma_{Extra} = \frac{Load_{Extra}}{A_{Bolt}} = \frac{\frac{3\ 165}{2}}{41,3} = 38,3 \text{ N/mm}^2$
- $\sigma_{Max} = 810 + 38,3 = 848,3 \text{ N/mm}^2$

So,

$$Range\ of\ Stress = (\sigma_{Max}) - (\sigma_{Min}) = 848,3 - 810 = 38,3 \text{ N/mm}^2$$

$$\sigma_{Mean} = \frac{(\sigma_{Max}) + (\sigma_{Min})}{2} = 829,2 \text{ N/mm}^2$$

## 4.6 GOODMAN DIAGRAM AND FSF FOR STANDARD BOLT

Some considerations prior the results:

$EL = \frac{UTS \cdot 0,45}{K_t}$ , where  $K_t$  is the stress concentration factor and comes from (Pilkey, 1997):

$$EL = \frac{1000 \cdot 0,45}{2} = 225 \text{ N/mm}^2$$

$$\text{Allowable Range at Mean} = 2 \cdot EL \cdot \left(1 - \frac{\sigma_{\text{Mean}}}{UTS}\right) = 76,9 \text{ N/mm}^2$$

$$FSF = \frac{\text{Allowable Range at Mean}}{\text{Range of Stress}}$$

Results:

Table 8. Conrod Bolt - FSF

| STRESS ANALYSIS         |       |                   |
|-------------------------|-------|-------------------|
|                         |       |                   |
| RANGE OF STRESSES       | 38,3  | N/mm <sup>2</sup> |
| MEAN STRESS             | 829,2 | N/mm <sup>2</sup> |
| FINAL ENDURANCE LIMIT   | 225   | N/mm <sup>2</sup> |
| ALLOWABLE RANGE AT MEAN | 76,9  | N/mm <sup>2</sup> |
| FATIGUE SAFETY FACTOR   | 2,01  |                   |

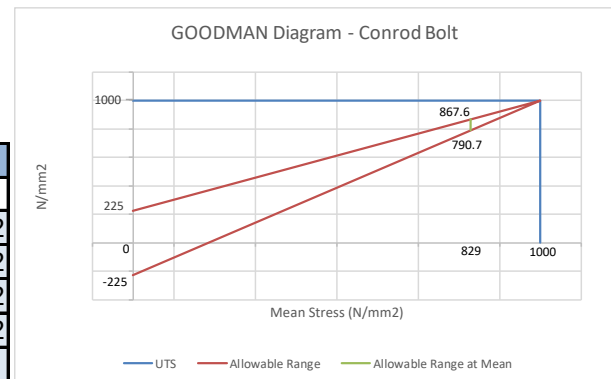


Figure 10. Goodman diagram of Conrod Bolt

As it can be observed from the Goodman diagram and the fatigue safety factor above, with a value of 2.01, the conrod bolt used in the standard engine was safely engineered to perform during an infinite period of time at maximum power regimes.

## 4.7 NEW BOLT DESIGN FOR UPGRADED ENGINE

In the new upgraded engine, introduced in the previous assignment, the engine is capable of higher RPM (from 6000 to 8000 RPM) and higher power figures.

Results from the engine upgrade:

Table 9. Comparison of standard engine (6000RPM) vs upgraded engine (8000RPM)

| STANDARD ENGINE VS UPGRADED ENGINE |        |        | Units |
|------------------------------------|--------|--------|-------|
| NET FORCE                          | 38 537 | 46 280 | N     |
| INERTIA FORCE                      | 10 048 | 17 862 | N     |
| CENTRIFUGAL FORCE                  | 2 685  | 4 774  | N     |
| TOTAL INERTIA FORCE                | 12 733 | 22 636 | N     |

With the upgraded engine the new axial forces going through the bolt are:

$$\text{Maximum Loads on bolt} = \frac{\text{Net Force}}{2} + \frac{\text{Total Inertia Force}}{2} = 25\,953\,N$$

Which are superior than the required clamping forces for the analysed bolt:

$$\text{Force}_{\text{Clamping Bolt}} = 0,6 \cdot \text{Yield} \cdot A_{\text{Bolt}} = 20\,063\,N$$

Thus, without changing the conrod, the only solution will be reducing the bolt stiffness to reduce the load acting upon the bolt, because:

$$\text{Load}_{\text{Extra}} = \frac{1}{\left(1 + \frac{K_c}{K_b}\right)} \cdot F_{\text{Total Inertia on bolt}}$$

So, to keep the same axial stress % of UTS, it will be necessary to increase the UTS of the bolt so that until the same clamping force it is not exceeded 90% of Yield of the new bolt in the shank.

The grade of the bolt was increased to 12.9, meaning a bigger UTS value of 1200 N/mm<sup>2</sup>:

$$D_{\text{average standard bolt}}^2 \cdot \text{UTS}_{\text{standard bolt}} = D_{\text{average upgraded bolt}}^2 \cdot \text{UTS}_{\text{upgraded bolt}}$$

$$D_{\text{average upgraded bolt}} = \sqrt{\frac{7,25^2 \cdot 1000}{1200}} = 6,61\,mm$$

#### 4.8 SKETCH OF STANDARD VS UPGRADED BOLT

As it can be observed in Figure 11, the bolt shank was reduced from 6.5mm to 5.22mm, to obtain the average diameter of 6.61 that was calculated in the previous Chapter, *4.7 New bolt design for upgraded engine*. The grade of the bolt was also increased to 12.9, allowing a smaller shank to hold the same clamping force, the UTS increased from 1000 to 1200 N/mm<sup>2</sup>.

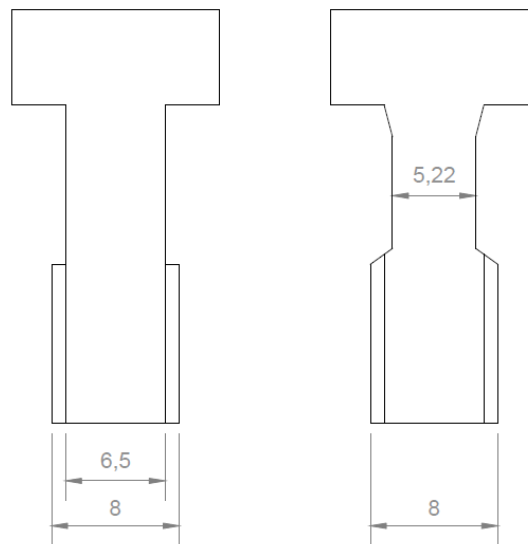


Figure 11. Standard VS Upgraded Bolt design

## 5 CRANKSHAFT GOODMAN DIAGRAM AND FATIGUE SAFETY FACTOR

In this Chapter the methodology used for the calculations will not be discussed, because of its similarities with the fully descriptive methodology already demonstrated on Chapter 2.1, having as the only difference the addition of rotational mass as contribution for the final Inertia Force (once again, already explained on Chapter 3.1).

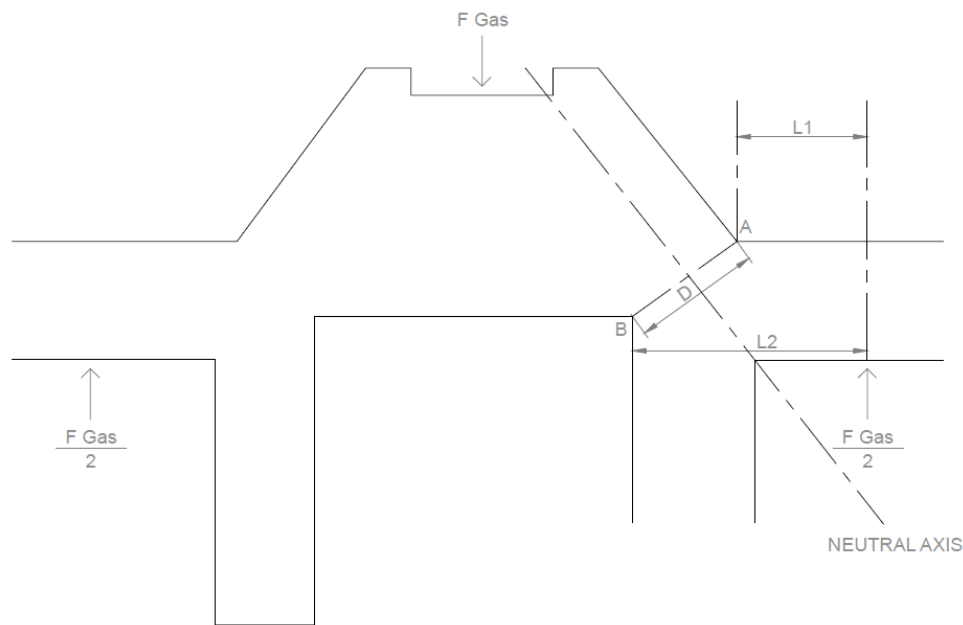


Figure 12. Crankshaft Diagram

The relevant dimensions of the crankshaft are presented in the table below, table 10, as well as the relevant calculations for the analysis of the crankshaft performance, plot of the Goodman diagram and consequently the fatigue safety factor.

Table 10. Crankshaft specs and stress calculations

| CRANKSHAFT SPECIFICATIONS | Units                   |
|---------------------------|-------------------------|
| UTS                       | 750,0 N/mm <sup>2</sup> |
| Kt                        | 1,85                    |
| Endurance Limit           | 182,4 N/mm <sup>2</sup> |
| D                         | 20 mm                   |
| y = D / 2                 | 10 mm                   |
| b                         | 42,5 mm                 |
| I - 2nd moment of area    | 28 333 mm <sup>4</sup>  |
| L1                        | 12 mm                   |
| L2                        | 40 mm                   |
| Effective Beam Area       | 1257 mm <sup>2</sup>    |
| Mass                      |                         |
| Reciprocating             | 479,6 grams             |
| Rotational                | 270,9 grams             |

|                                           | CRANKSHAFT ANALYSIS |         |                   |
|-------------------------------------------|---------------------|---------|-------------------|
|                                           | Firing              | Exhaust | Units             |
| <b>GAS FORCE</b>                          | 46 863              | 491     | N                 |
| <b>INERTIA FORCE</b>                      | -12 733             | -12 733 | N                 |
| <b>STRESS</b>                             |                     |         |                   |
| <b>BENDING STRESS (N/mm<sup>2</sup>)</b>  | 240,9               | -86,4   | N/mm <sup>2</sup> |
| <b>SHEAR STRESS (N/mm<sup>2</sup>)</b>    | 13,6                | -4,9    | N/mm <sup>2</sup> |
| <b>COMBINED STRESS (N/mm<sup>2</sup>)</b> | 242,4               | -87,0   | N/mm <sup>2</sup> |
| <b>RANGE OF STRESSES</b>                  |                     |         |                   |
| <b>MEAN STRESS</b>                        | 329,4               | 77,7    | N/mm <sup>2</sup> |

Finally, as it can be observed by the Goodman Diagram below and the analysis of table 11, the introduction of plasma nitriding introduces in the surface and fillets of the crankshaft residual compressive stress that results in a superior allowable range. As observed, increasing the actual range, results in a bigger safety factor.

For this specific case, an introduction of a residual stress of 800N/mm<sup>2</sup> increases the fatigue safety factor for double.

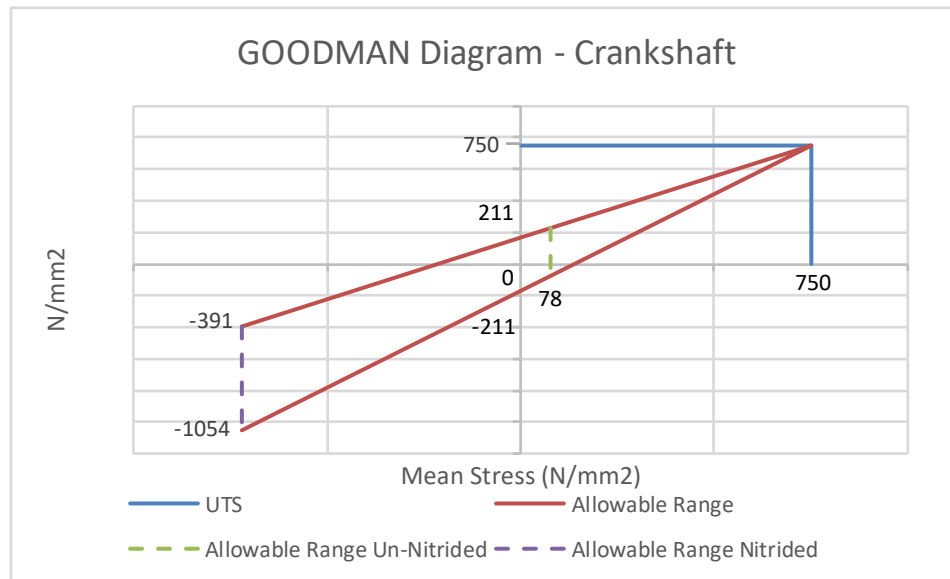


Figure 13. Goodman diagram for standard Crankshaft

Table 11. FSF of Un-Nitrided VS Nitrided Crankshaft

|                                | Un-Nitrided | Plasma Nitrided | Units             |
|--------------------------------|-------------|-----------------|-------------------|
| <b>RANGE OF STRESSES</b>       | 329,4       | 329,4           | N/mm <sup>2</sup> |
| <b>MEAN STRESS</b>             | 77,7        | -722,3          | N/mm <sup>2</sup> |
|                                |             |                 |                   |
| <b>ENDURANCE LIMIT</b>         | 182,4       | 182,4           | N/mm <sup>2</sup> |
| <b>ALLOWABLE RANGE AT MEAN</b> | 327,0       | 716,2           | N/mm <sup>2</sup> |
|                                |             |                 |                   |
| <b>FATIGUE SAFETY FACTOR</b>   | 0,99        | 2,17            |                   |

## 6 CONCLUSION

This project allowed the author to work out the safety factors of several vital engine components of the “standard” project engine, through analytical calculations and models developed by other authors.

For the gudgeon pin, it was found that reducing the offset between the piston and the conrod would result in less bending stress. Reducing the stress, for the same FSF, allows the use of a lighter gudgeon pin that will only benefit the engine’s performance by reducing the reciprocating mass and consequently the inertia force.

In the conrod cap and conrod bolt region it was stated the importance of this region to be as stiff as possible in order to reduce stretch at higher RPM values. Reducing stretch will reduce the extra force in the conrod bolt.

From all the parts’ results and making use of the analysis of the Goodman diagrams, it was concluded that increasing the allowable range would result in a bigger fatigue safety factor. Such thing could be possible by introducing residual compressive stress into the parts’ surface and fillets.

The reverse engineering process developed represents a powerful tool when upgrading the standard engine or design new engine components. This process allows the author to start with valuable references when starting to design vital engine components, representing quicker progress and consequently cost savings.

## 7 BIBLIOGRAPHY

- Pilkey, W. D. (1997). *PETERSON'S STRESS CONCENTRATION FACTORS*. JOHN WILEY & SONS, INC.
- Heywood, J. B. (1988). *Internal Combustion Engine Fundamentals*. McGraw-Hill, Inc.
- Blair, G. P. (1999). *Design and Simulation of Four-Stroke Engines*. Warrendale, Pa: Society of Automotic Engineers, Inc.