Finite element analysis of marine diesel engine components

M. Morsy El-Gohary
Naval Architecture and Marine Engg. Dept., Faculty of Engg., Alexandria University, Alexandria, Egypt

The objective of this paper is to design a better marine power plant for the multipurpose vessel Abu 'Aegela and replace the existing MAN B and W two stroke engine with another engine that better suits operating requirements. Engine thermodynamics equations were solved using Engineering Equation Solver software (EES) which helped in reducing solving time and showed the effect of changing variables on other engine parameters. The dimensions of the main engine components were then calculated using machine design equations and the components were modeled with the previous dimensions using AutoDesk Inventor 3D software package which gives great flexibility to the design process and allows for easy changes in all design stages. Finite element analysis of the components was carried out in ANSYS program which highlighted areas of high stress concentration and showed failure probabilities. The results showed the influence of the engine loads on the different components and helped in correcting the weak points of the design which could cause severe problems in the unforgiving marine environment.

يهدف هذا البحث إلى تصميم محطة توليد قوى أفضل للسفينة متعددة الأغراض (أبوعجيلة) واستبدال المحرك الثنائي الأشواط الحالي (MAN B and W) بآخر أكثر ملائمة لمتطلبات التشغيل. تم حل معادلات الديناميكا الحرارية للمحرك باستخدام برنامج (EES) الذي ساهم في تقليل الوقت اللازم لحل المعادلات وأظهر تأثير تغيير قيم المتغيرات على عوامل المحرك الأخرى. تم بعد ذلك حساب الأبعاد الأساسية للمحرك باستخدام معادلات التصميم الميكانيكي وتم تمثيل الأجزاء المختلفة للمحرك باستخدام (AutoDesk Inventor 3D) و الذي ساعد في زيادة المرونة في عملية التصميم بمختلف مراحلها. التحليل باستخدام طريقة العناصر المحددة تم باستخدام برنامج (ANSYS) والذي بين مناطق تركيز الاجهادات واحتمالات الانهيار. بينت النتائج تأثير أحمال المحرك على الأجزاء المختلفة وساعدت في تحديد نقاط الضعف في التصميم والتي قد تسبب مشاكل خطيرة في البيئة البحرية الصعبة.

Keywords: ANSYS, CAD, Diesel engine, Finite element method

1. Introduction

The four stroke Diesel engines are probably the most prevalent engines in the world. They can be found producing power for oil rigs in the North Atlantic or driving Volkswagen's new Beetle DTI. The medium speed four stroke trunk piston engine can be found on most medium to large merchant vessels even if the main engine is either a steam turbine or a 2 stroke crosshead engine [1]. In these cases it will often be found that the electrical power is supplied by alternators driven by medium speed 4 stroke engines. They are the favored method of propulsion on ships where head room is a minimum, as in case of ferries and passenger vessels, and where, as is the current trend for these ships, diesel electric propulsion is utilized. Diesel electric propulsion allows the engines to be placed wherever is most suitable, as they no longer have to be aligned with reduction gearing and shafting as is the case with conventional installations [2] fig. 1.

Four stroke engines are more complicated than the two stroke and "pack" less horsepower per pound. The two stroke actually produces more power, around 1.8:1. But the four stroke consumes less oil and can operate safely at high speeds [3]. Many companies build four stroke Diesel engines. Caterpillar, Cummins, Detroit Diesel engines are the most popular engine manufacturer for highway trucks and smaller applications. Others like MAN (the company that first developed Rudolph Diesel’s engine), MAK, MTU, Wartsila, Deutz - just to name a few, offer a wide range of power plants often found in marine applications. Four stroke engines have a power range of 2kW to 25,000kW. These can be found in many V style and inline configurations: V8, V12, V16, V20; the inline 6 being the most popular one [4 - 6].

The practical maximum output of a marine diesel engine may be said to have been reached when one or more of the following factors operate:
1. The maximum percentage of fuel possible is being burned effectively in the cylinder volume available.
2. The stresses in the component parts of the engine generally, for mechanical and thermal conditions prevailing, have attained the highest safe level for continuous working.
3. The mean piston speed and thus Revolutions Per Minute (RPM) cannot safely be increased [3].

2. Engine design

In the process of designing any engine the brake horsepower and RPM are related by the following formula

$$BHP = \frac{P_e \, . \, V_s \, . \, n \, . \, Z}{c} \, . \tag{1}$$

Mean effective pressure (P_e) can be calculated from indicated pressure P_i, by assuming mechanical efficiency η_m (around 0.85) [7].

As shown in the previous equation, if P_e is determined, the engine bore, stroke and number of cylinders can be calculated. Before the widespread use of computers in the process of engine design, the only way to calculate P_e was from using equations derived from the (p-v) curve. Nowadays, computers make it a lot easier for designers, and the equations that consumed days and days of calculations are now solved in few seconds [8].

A computer code was developed using EES software (Engineering Equation Solver) to determine the bore and stroke of the engine required to propel the vessel. The following are the steps and equations used for developing this code:
1. The air to fuel ratio (AF_{ratio}) required for complete combustion was found from the combustion equations of diesel fuel $C_{12}H_{26}$ and equals 15.03.
2. The heat added, Q_{add} per kg by the air/fuel mixture is calculated using the calorific value (CV) of the fuel (=10,000 kCal/kg), air to fuel ratio and excess air factor (λ=1.5).

$$Q_{add} = \frac{CV}{AF_{ratio} \, . \, \lambda} \, . \tag{2}$$

3. By assuming the cooling water losses percentage (CW) to be 25% and subtracting from the total heat added, effective heat (Q_{eff}) can be found which is the heat added to the cycle, where a part of it is converted into mechanical energy and the rest to exhaust heat and other losses.

$$Q_{eff} = (1 - CW) . \, Q_{add} \, . \tag{3}$$

4. The initial conditions are specified as: pressure = 4 bar (turbocharger outlet pressure) and temperature = 85 ^{o}C (air temperature directly before compression stroke).
5. Assuming a compression ratio fig. 2 (r) of 15, volume at point 2 can be calculated. Pressure and temperature at point 2 can be calculated if the compression is assumed to be isentropic, fig. 3.
6. The effective heat previously calculated is then added to the cycle at constant pressure to obtain point 3.
7. Assuming isentropic expansion, point 4 can be obtained.
8. The heat rejected at constant volume from point 4 to point 1 is found from:

$$Q_{rej} = U_4 - U_1 \, . \tag{4}$$

Also exhaust losses can be found from:

$$EXH_{LOSS} = Q_{rej} \, / \, Q_{add} \, . \tag{5}$$

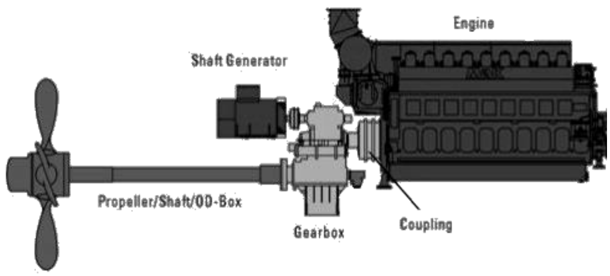

Fig. 1. Diesel propulsion main components.

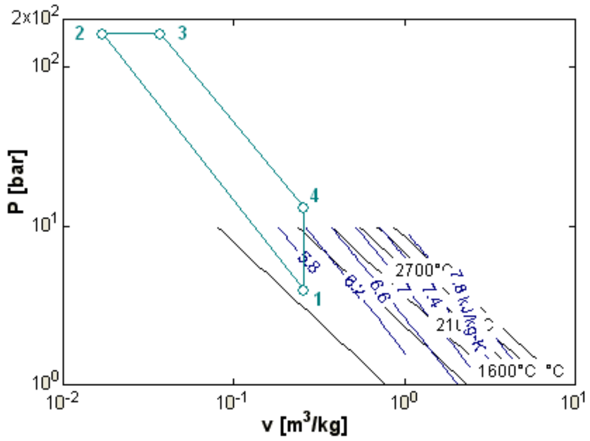

Fig. 2. Cycle P-V chart.

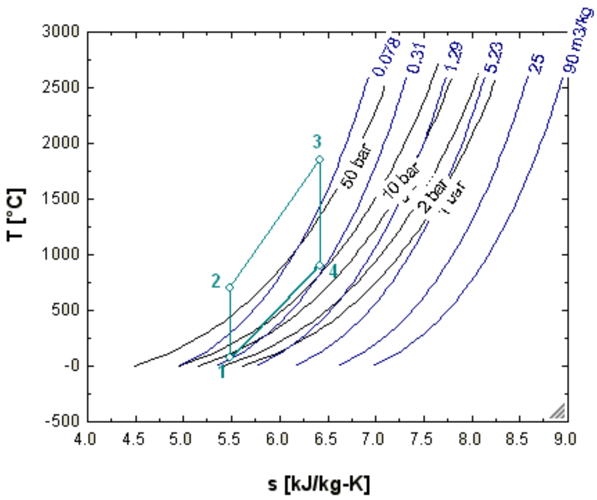

Fig. 3. Cycle T-S chart.

9. Indicated heat, (Q_i), is calculated from the difference between effective heat and rejected heat.
10. Indicated pressure, P_i, and indicated thermal efficiency, η_{ith}, can then be calculated from the following eqs. [9]:

$$P_i = c_1\left[\frac{Q_i}{V_1 - V_2}\right]. \tag{6}$$

$$\eta_{ith} = \frac{Q_i}{Q_{add}}. \tag{7}$$

11. Taking mechanical efficiency equals to 0.85, brake mean effective pressure and brake thermal efficiency can be found.
12. By substituting the calculated value of the mean effective pressure in the following equation:

$$D = \sqrt{\frac{c_2\ .\ BHP}{Pe\ .\ \pi\ .\ L\ .\ n\ .\ Z}}\ , \tag{8}$$

and taking BHP = 6718 hp which is the brake engine power to propel the ship, n = 700 rpm, L/D = 1.25, Z = 6 cylinders, the cylinder dimensions can be found table 1.

Solving eqs. (2 through 8) using EES, engine design parameters are determined as described in the following sections.

3. Mechanical design of main engine components

The next step after the bore and stroke are calculated is to design the internal components of the engine. Machine design equations are used to design the cylinder liners, pistons, connecting rods and crankshaft [10]. Finite element analysis is performed using ANSYS to check the results of the equations.

3.1. Cylinder liners

The following formula could be used to calculate maximum liner thickness taking into account mechanical stresses only.

$$t_{\max} = \frac{D}{2} \times \left[\sqrt{\frac{\sigma + P_{\max}}{\sigma - P_{\max}}}\right] + t_a\ . \tag{9}$$

Table 1
Design results

Cooling losses	0.25
Exhaust losses	0.3529
Brake thermal efficiency	0.3375
bsfc	187.2 gm/kwh
Bore	38.01 cm
Stroke	47.51 cm
bmep	26.7 bar
Maximum cylinder pressure	164.6 bar

Where:
t_{max} is the maximum liner thickness, mms,
D is the Cylinder bore, mms,
σ is the allowable stress, bar,
P_{max} is the max cylinder gas pressure, bar, and
t_a is the allowance for reboring or regrinding, mms.

Solving eq. (8), t_{max} = 37.334 mm

3.2. Piston design

As a general practice, calculations are made for the following parts of the piston: skirt (or body), crown, piston pins and piston rings.

3.2.1. Piston crown

Assuming that the piston crown is subjected to a uniformly distributed load of the maximum gas pressure P_{max}, Faust equation can be used to calculate the stresses resulting from the pressure of the combustion gases on the piston crown, the thermal stresses are included in the constant of the equation as a factor of the mechanical load.

$$\frac{\sigma_{max}}{P_{max}} = c(\frac{r}{h})^2 . \tag{10}$$

Where:
c is the 1.25,
h is the crown thickness, cms,
r is the piston radius, cms,
P_{max} is the maximum gas pressure, bar, and
σ_{max} is the high allowable stress for steel.

Solving eq. (10), h = 7.037 cm

3.2.2. Piston rings

Piston rings are made from special cast iron having high elasticity.

Let
b is the length of the ring which presses the cylinder, and
a is the breadth

$$b = D\sqrt{\frac{3P}{f_b}} . \tag{11}$$

$$a = \frac{D}{25} . \tag{12}$$

Where:
P is the pressing force (ring on cylinder), and
f_b is the high allowable stress of ring material (centrifugal cast iron).

Solving (11 and 12), b=1.61cm, a=1.52 cm

3.2.3. Results verification using finite element analysis

1. The piston crown, skirt, piston rings and piston pin with the previous dimensions where modeled using Autodesk Inventor fig.4.
2. A uniform pressure of 164.6 bar was applied on the surface of the piston crown and two cylindrical supports where applied at the pin bosses fig.5.
3. Solving for equivalent (Von-Mises) stresses the following results were obtained, fig. 6.

3.4. Piston pins

Piston pins are made from nitrogen hardened steel or case hardened steel [11]. Their tubular construction gives them maximum strength with minimum weight. The stresses acting upon pins are bearing stresses (at the connecting rod small end) and bending moments (assuming that the pin is a beam simply supported with two supports at the center of the bosses).

Pin diameter can be written as:

$$d = c \times D. \tag{13}$$

Where:
d is the pin diameter, and
c is the 0.3~0.425.

Solving eq. (13), d = 0.425 × 38 = 16.15 cm

3.4.1. Bearing stresses

$$\sigma_{br} = \frac{Load}{\text{projected area}} . \tag{14}$$

σ_{br} is the high allowable bearing stress.

Solving eq. (14), l = 14.449 cm.

3.5. Results verification using FEA

1. The pin was modeled using AutoDesk Inventor fig. 7.
2. Maximum cylinder force ($P_{max} \times A$) was applied at the center of the wrist pin, and two cylindrical supports were applied at the bosses fig. 8.
3. Solving for equivalent (Von Mises) stresses, the obtained results were presented in fig.9:

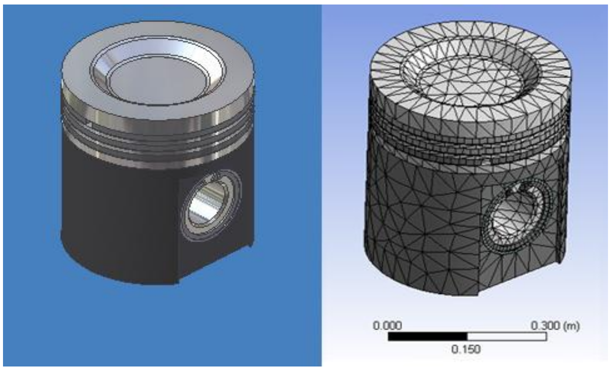

Fig. 4. Modeled piston and its finite element simulation.

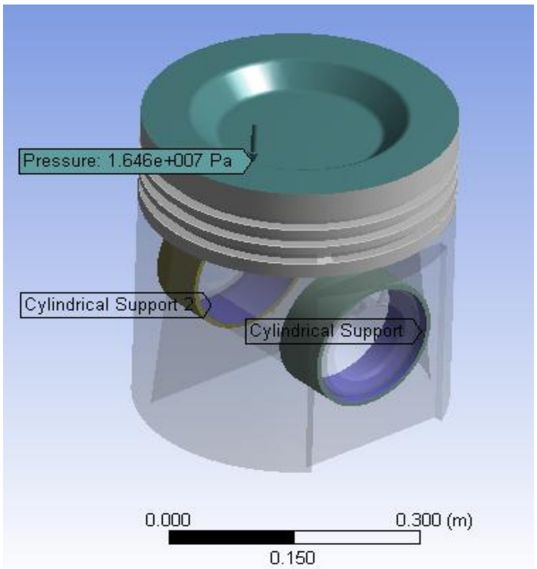

Fig. 5. Applied boundary conditions to the piston.

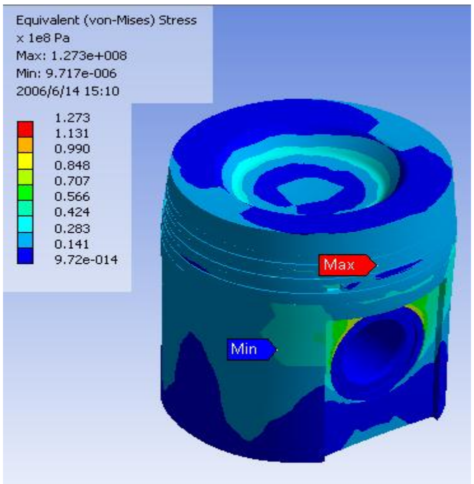

Fig. 6. Simulation result for piston.

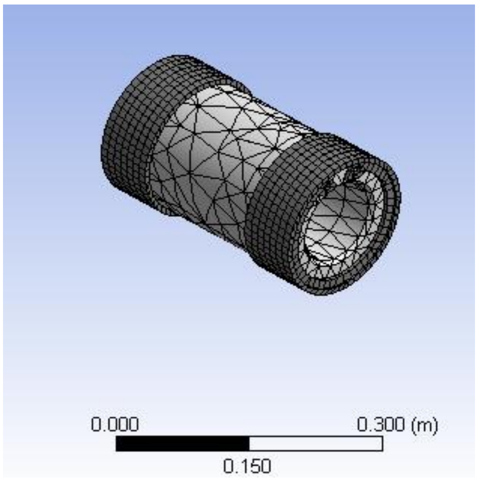

Fig. 7. Finite element model for pin.

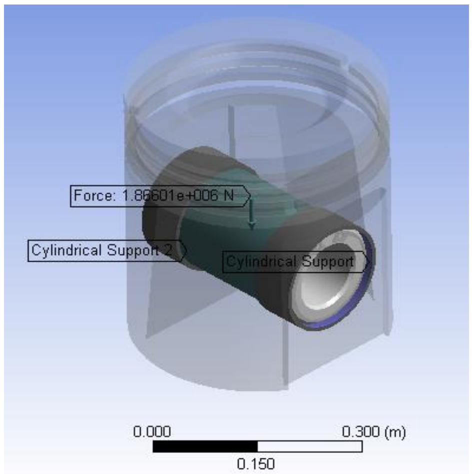

Fig. 8. Applied boundary conditions for pin and bossings.

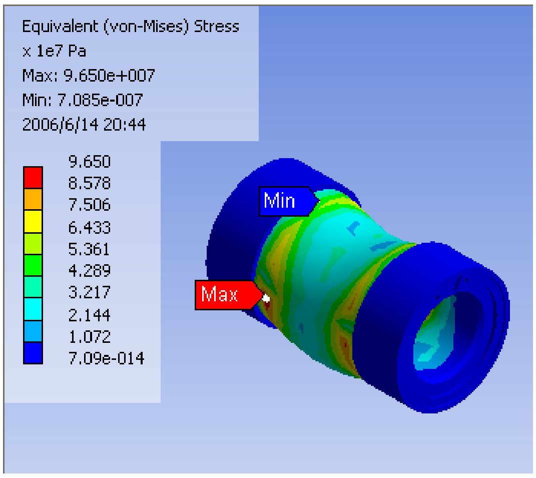

Fig. 9. FEA result for pin and bossings.

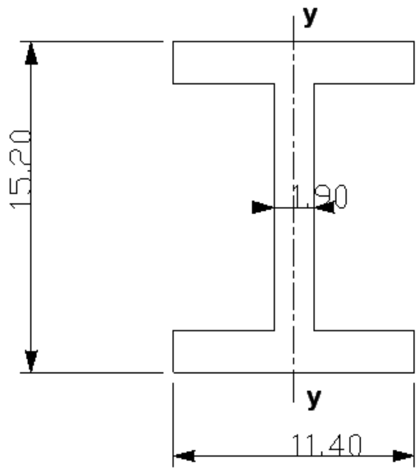

Fig. 10. Connecting rod cross section.

3.6. Connecting rod design

Calculations the connecting rod is made of forged steel and is checked for both compression stresses and buckling, fig. 10.

Let:
l is the connecting rod length, and
r is the crank radius.

$$r = \frac{1}{2} . \qquad (15)$$

Taking $l \cong 4r$ and solving eq. (15), l = 95 cm.

To check the connecting rod for buckling, Euler's formula is used.

$$Safe\ load = S.\,l\,. = \frac{\pi^2 .E\,.I}{l^2} . \qquad (16)$$

Where:
l is the length of connecting rod,
r is the radius of crank shaft,
I is the least moment of inertia, and
E is the modulus of Elasticity (forged steel).

Solving eq. (16), Safe load = 1.04 × 10^6 kg. Note that the safe load calculated from Euler's formula must be five times greater than the maximum force acting on the connecting rod [7].

$$S.\,l\,. > 9.333 \times 10^5$$

In addition to buckling, the connecting rod should also be checked for compression stress

$$\sigma_c = \frac{\frac{\pi}{4} D^2 P_{max}}{A_{section}} . \qquad (17)$$

From eq. (17), σ_c = 2872.812 bar.

Bending stresses resulting from the rotation effect of the big end also act on the connecting rod, and can be calculated from:

$$\sigma_b = \sigma_{bending} = \frac{{}^2\!/_3\, F_{max} .\, l}{Z_s} . \qquad (18)$$

Where
Z_s is the section modulus and F_{max} can be found from:

$$F_{max} = \frac{\gamma .A_{section} .l}{2\,g}\,\omega^2 .\,r \quad . \qquad (19)$$

Where
g is the steel density,
ω is the angular velocity of the crankshaft,
$A_{section}$ is the sectional area of the connecting rod, and
r is the crank radius = L/2.

From eq. (19), F_{max} = 3215.542 kg. Solving (18), σ_b = 2440.395 bar.

The resulting stresses equal:

$\sigma_{total} = \sigma_c + \sigma_b$
$\sigma_{total} = 2872.812 + 2440.395$
$= 5313.207$ bar

Which is less than the allowable stress of forged steel, σ_a =5500 bar.

3.7. Results verification using FEA

1. The connecting rod was modeled using Autodesk Inventor, see fig. 11.
2. Maximum cylinder force was applied at the small end while fixing the big end, see fig. 12.
3. The obtained results are shown in fig. 13.
4. By fixing the small end and applying maximum cylinder force at the big end, the results shown in fig. 14 were obtained. Fig. 15 shows the results of the second case where the big end is fixed and F_{max} is applied at the small end.

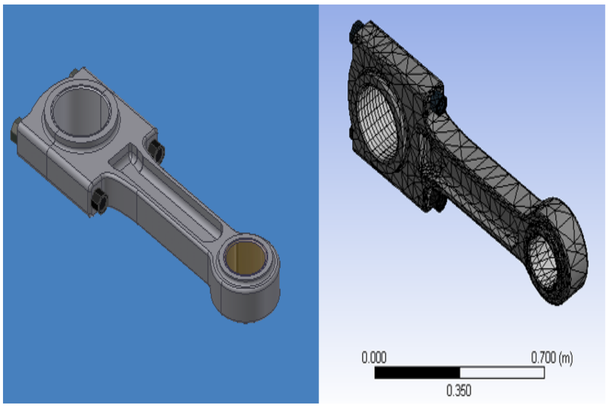

Fig. 11. Modeled rod and its finite element model.

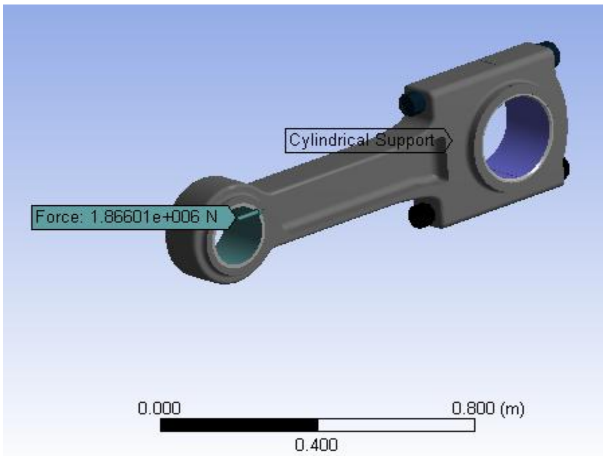

Fig. 12. Boundary condition for 1st case of rod analysis.

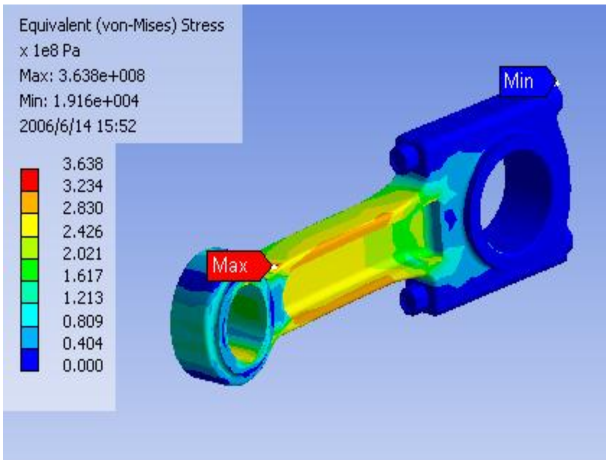

Fig. 13. FEA results for 1stcase of rod.

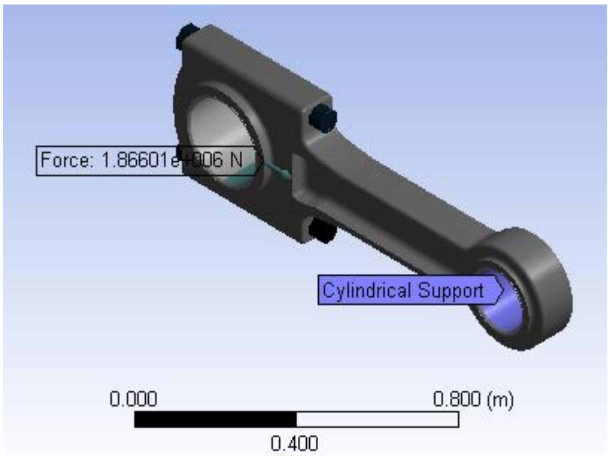

Fig. 14. Boundary conditions for the 2nd case.

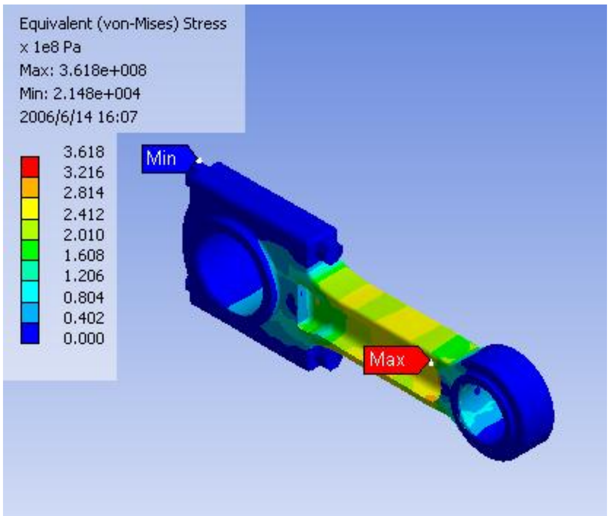

Fig. 15. Second case results.

3.8. Crankshaft design

Calculations the crankshaft diameter can be calculated from the following formula:

$$d = \sqrt[3]{D^2(\alpha\ L + \beta\ X)}. \qquad (20)$$

Constants	Multi cylinder
α	0.131
β	0.050

From eq. (20), the crankshaft diameter is calculated, d = 22.8 cm

The firing sequence used was 1-5-3-6-2-4 with angle between explosions 120° as shown in fig. 16. [12].

The complete crankshaft modeled using Autodesk Inventor is shown in fig. 17.

3.9. Results verification using FEA

1. A section of the crankshaft was separated for detailed analysis, as shown in fig. 18.

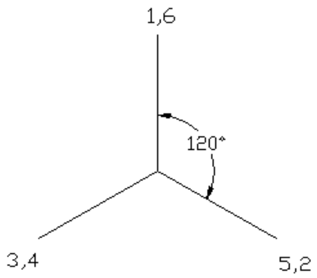

Fig. 16. Angles between cranks.

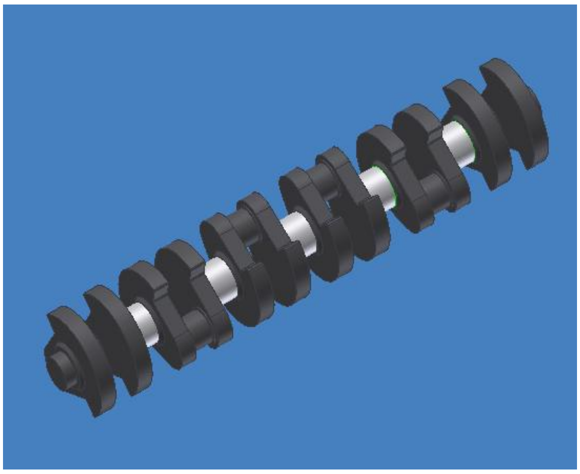

Fig. 17. Complete crankshaft model.

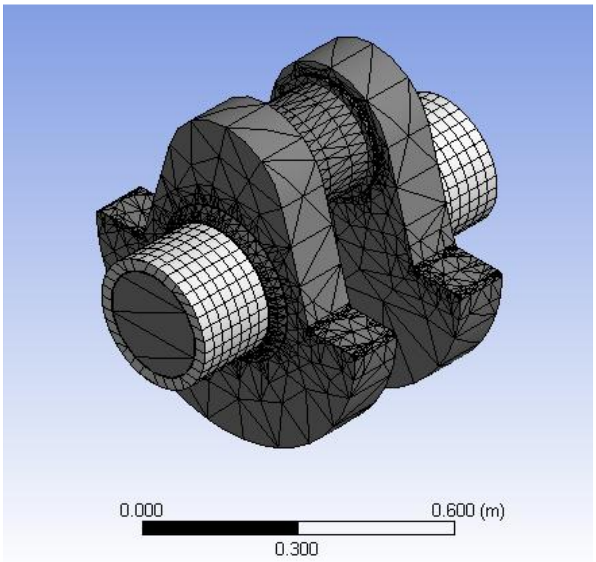

Fig. 18. Finite element model for one crank.

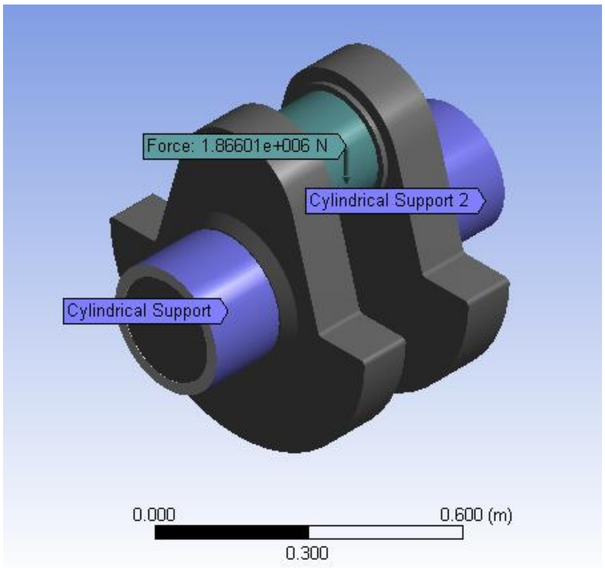

Fig. 19. Crank with applied boundary conditions.

2. Maximum cylinder force was applied at the center of the crankshaft, and two cylindrical supports were applied at the crankshaft bearings, as shown in fig. 19.
3. Solving for equivalent (Von-Mises) stresses the results shown in fig. 20 were obtained.

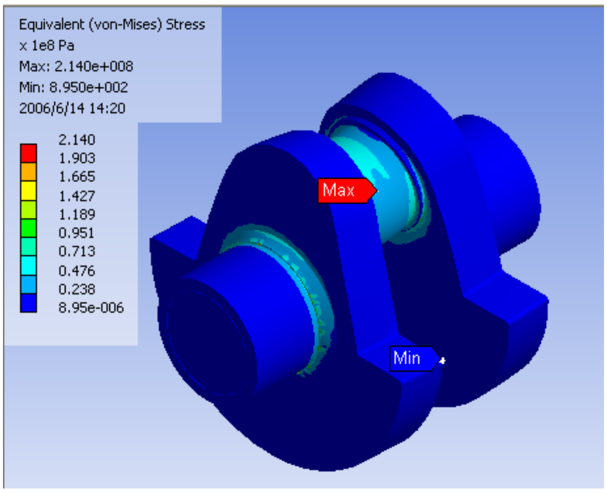

Fig. 20. FEA result for one crank.

4. Conclusions

Design process was dramatically speeded up by using EES program which reduced calculation times and showed the effect of changing variables on other engine parameters. Modeling the parts using specialized CAD software like Autodesk Inventor is necessary to obtain accurate results later in the simulation stage, it also gives flexibility to the design process allowing for changes in parts shape and dimensions at any time. FEA for the different engine components is essential before manufacturing prototypes as it shows areas subjected to high stresses, avoiding a lot of problems and saving money that will be spent if the problem was not dealt with early. ANSYS is a great tool for FEA and can be used by engineers of different expertise levels and has high connectivity with other CAD programs.

All this leads in the end to fast and accurate engine design, allowing for making design improvements easier, these improvements may appear to be of a great importance especially with the continuous reduction in emissions levels issued in IMO (International Maritime Organization) regulations. Also, a need for changing fuel types and making the engine capable to burn gaseous fuels like natural gas and hydrogen to deal with the increasing oil based fuels prices, may be one of the challenging issues that have to be well understood and accurately applied to help the move towards a 'greener' future.

Nomenclature

L	is the stroke,
D	is the bore,
BHP	is the brake horsepower,
$bsfc$	is the brake specific fuel consumption,
Z	is the number of cylinders,
n	is the engine speed,
P_{max}	is the maximum cylinder pressure,
F_{max}	is the maximum cylinder Force,
Z_s	is the section modulus,
g	is the steel density,
$S.l.$	is the safe buckling load,
σ_c	is the compression stress,
σ_b	is the bending stress,
σ_a	is the allowable stress,
E	is the modulus of elasticity,
I	is the least moment of inertia,
P_e	is the brake mean effective pressure,
P_i	is the indicated mean effective pressure,
c, c_1	is the unit conversion factors,
CV	is the fuel calorific value,
AF_{ratio}	is the air/fuel ratio,
λ	is the excess air factor,
η_{ith}	is the indicated thermal efficiency,
V_s	is the stroke volume, and
U_i	is the internal energy at point i.

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