Design and stress simulation of crankshaft for slider crank-drive Stirling engine

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Abstract. In this present work, the design and simulation of crankshaft for multi-cylinder Stirling engine is studied based on finite element analysis. The proposed crankshaft design is based on the typical crosshead slider-crank mechanism that is being used with the consideration of design needs for multi-cylinder Stirling engine. The study focused on the piston-crankshaft assembly that is subjected to compression load in Stirling cycle. Based on the simulation results, the maximum von Mises stress for crankshaft model varies from 0.82 MPa at 1 bar charge pressure to 1.65 MPa at 20 bar charge pressure. Minimum factor of safety is founded to be 33 with maximum deformation under maximum charge pressure. For piston-crankshaft assembly load, minimum factor safety of 2 was observed with maximum compression pressure for minimum charge pressure. The results indicate no yielding and structural failure under compression load case, can be satisfied.

Introduction

Crankshaft plays an important role as a main moving engine component. The crankshaft transfers the rotary power produced by the engine which will be an input to many devices such as generators, pumps or compressor. In reciprocating engine type, the crankshaft converts the reciprocating displacement of the piston to a rotary motion [1]. For many years, the slider crank-drive has been used in reciprocating engines because it is extremely reliable, with a wealth operating experience.

Stirling engine designs are usually known as either kinematic or free-piston engine [2]. For kinematic Stirling engine, the power piston is mechanically connected to a rotating output shaft (crankshaft) with a connecting rod. This type of mechanism is typically used in slider crank-drive configurations Stirling engine. The slider crank-drive is being used extensively in double acting Stirling engine because of the reliability and ease of construction, although having the disadvantage of being almost impossible to balance [3]. Moreover, this mechanism is widely used in multi-cylinder versions of the basic Stirling engine.

For the present work, the objective is to design a crankshaft for displacer-piston Stirling engine having a slider crank-drive as a driving mechanism.The study was conducted within designing a twin cylinder gamma-type Stirling engine that could potentially being used as energy conversion devices in parabolic Dish/Stirling technologies for Concentrating Solar Thermal (CST) application.

Crankshaft Design

Taking into consideration the design needs for multi-cylinder configuration, the typical crosshead slider-crank mechanism of own design has been used as the driving mechanism. In order to obtain necessary relation of movement of the displacer against the power piston, both crankshafts are interconnected by means of coupling, stated by the constant phase angle, θ. In addition, a counterweight balance has been used to minimize the vibrations. The counterweight provides for the possibility of balancing the reciprocating and rotating forces, since the masses of the displacer assembly and power piston assembly are quite different. The overall proposed design of the crankshaft is shown in Fig. 1. The design was expected to be as simple as possible for ease of construction, minimizing the machining difficulties and avoiding complex construction processes. Besides, the material selection also has been considered in order to produce lightweight crankshaft and strength enough that could take large compression load during engine operation. For this kind of purpose, an aluminium alloy has been selected for the crankshaft's material and will be used within the stress simulation for the crankshaft. Table 1 and Table 2 indicate the design parameters for crankshaft and material properties for crankshaft construction, respectively.

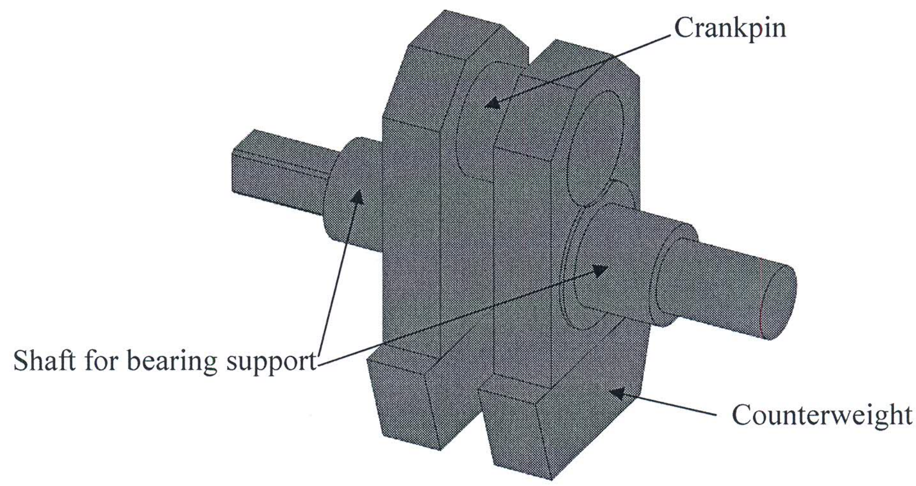


Fig. 1.A 3D model of proposed crankshaft design.

Table 1. Design parameters for crankshaft.

Parameters	Value
Crankpin diameter	28 mm
Crankpin offset	29 mm
Shaft diameter	25 mm
Web thickness	15 mm
Web width	44 mm
Theoretical Mass	0.562 kg

Table 2.Material properties for Aluminium Alloy.

Properties	Value
Elastic Modulus	6.9×10^{10} N/m^2
Poisson's Ratio	0.33
Shear Modulus	2.6×10^{10} N/m^2
Tensile Strength	1.24×10^{8} N/m^2
Yield Strength	55.15×10^{6} N/m^2
Density	2700 kg/m^3

Finite Element Analysis

Finite element analysis could be a regular part of design process in reducing the need for costly prototypes, eliminating rework and delays, and saving time and development costs. The FEA is based on the idea of building a complicated object with simple blocks or dividing a complicated object into a small and manageable piece, which is known as element with nodes [4].Before the simulation began, several assumptions were made with regards to the modelling the crankshaft. The simulation can be performed to power piston's crankshaft assembly since these components were subjected to compression stroke pressure. The displacer's crankshaft assembly was removed from the complete assembly in order to reduce the calculated time and simple boundary conditions could be enforced. For stress analysis, the crankshaft was modelled with finer mesh in order to get accurate stress distribution. Hence, a model of the crankshaft with 72645 solid elements and 107762 nodes were produced as shown in Fig. 2 (a). A model of piston-crankshaft assembly with 81157 solid elements and 124062 nodes were produced as shown in Fig. 2 (b).

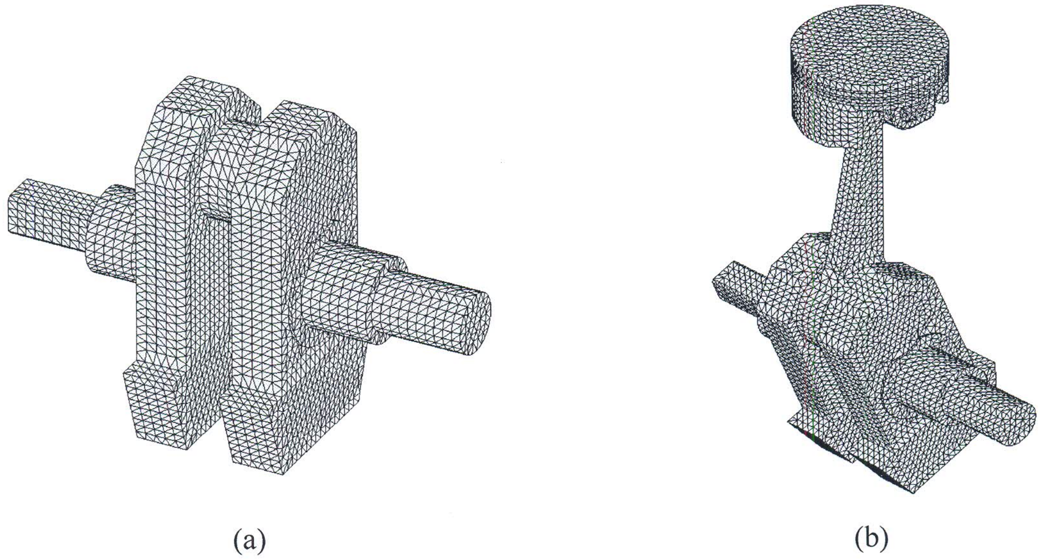


Fig. 2. A 3D solid element meshes (a) crankshaft model (b) piston-crankshaft assembly model

Boundary Conditions

In order to decrease the complexity of the boundary conditions, the interaction between the cylinder block, cylinder head, ball bearings and bearing housing were not be modelled. The crankshaft is fixed to the engine block by two ball bearings. Constraints were set on the shaft surface to be fixed surface in all degree of freedom. This indicates that the surface could not move in either direction and could not rotate. Besides, the pressure from the compression stroke within the Stirling cycle will be used as a load for the analysis. The calculation of maximum pressure will be based on the theoretical cycle pressure, following the method provided by Thombare and Verma [3]. Hence, the crankshaft model and piston-crankshaft assembly model will be simulating to determine the stresses and displacement, and to ensure their value were not exceed the strength and deformation limit of the material strength.For crankshaft model stress simulation, the load is applied to the crankpin surface, facing downward in parallel with the connecting rod angle, φ from the vertical axis of engine stroke, as shown in Fig. 3.According to Brahmbhatt and Choubey [1], the force acting on the piston, F_p can be determined by rearranging this equation,

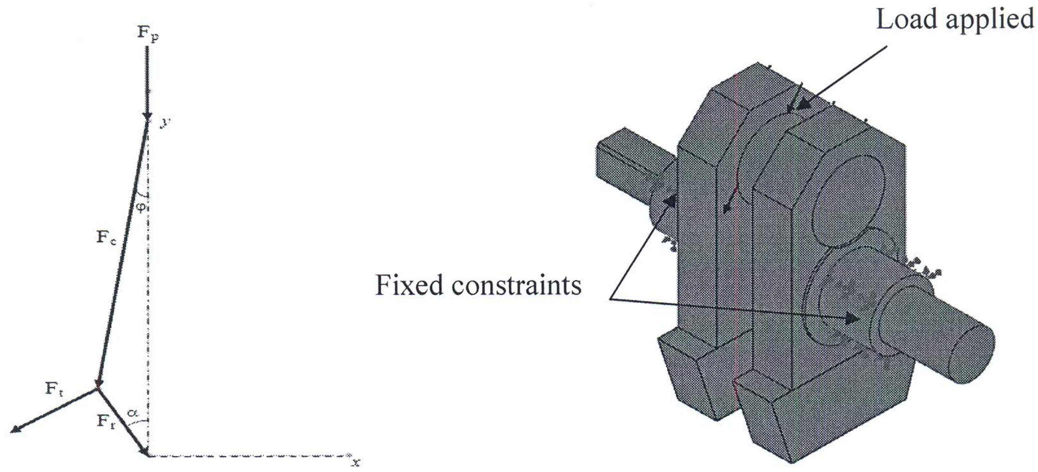
$$\text{Pressure} = \frac{\text{Force}}{\text{Area}} \tag{1}$$

In order to find the force acting on the connecting rod (F_c), and the angle of inclination of the connecting rod with respect to the line of stroke (angle, φ), sine law is applied based on this equation [5]:

$$\sin\varphi = \frac{b \sin\alpha}{L} \tag{2}$$

where b is the crank offset radius, L is the connecting rod length. Thus, the force acting on the connecting rod,

$$F_c = \frac{F_p}{\cos\alpha} \quad (3)$$



(a) (b)

Fig. 3. (a) Load distribution for stress simulation (b) Boundary condition setting

Results and Discussions

Finite element analysis predictions of stresses and deformation distribution for crankshaft model are shown in Fig. 4 and Fig. 5, respectively. The maximum stress for the crankshaft design varies from 0.82 MPa at 1 bar charge pressure to 1.65 MPa at 20 bar charge pressure as an increasing in compression cycle pressure. Meanwhile, the moderate von Mises stress is located at the areas between the step for bearing sit and lower area of the crankpin location due to compressive loads from radial force. The crankshaft model shows the largest deformation mostly on the top surface of crankpin and crosshead under gas pressure load varies from 1.6 x 10^{-8} m to 3.20 x 10^{-7} m. Fig. 6 illustrates the minimum factor of safety of 33 was produced under maximum charge pressure, although the maximum deformation is observed. These indicate that the crankshaft is able to withstand high pressure loading without exceeding the stress and deformation limit of material strength.

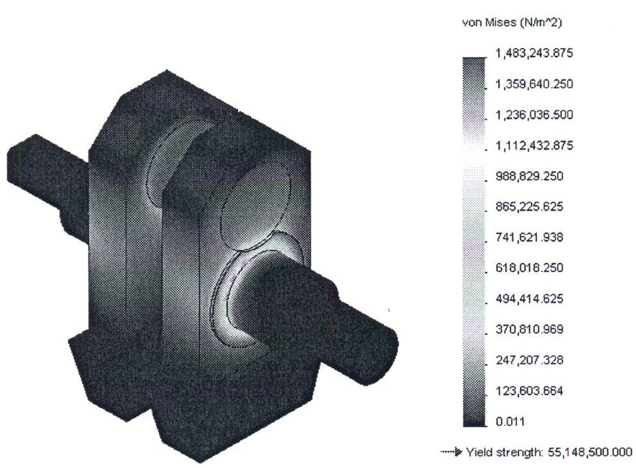


Fig. 4. Stress distribution at maximum charge pressure.

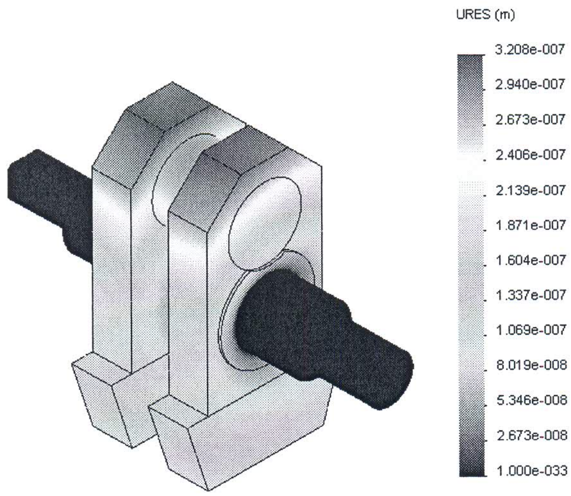


Fig. 5. Deformation on the crankshaft model at maximum charge pressure.

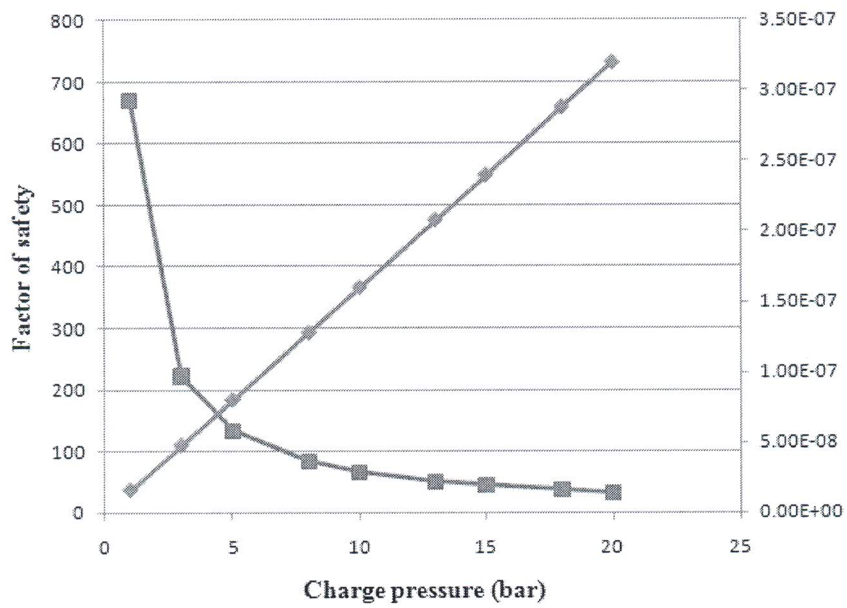
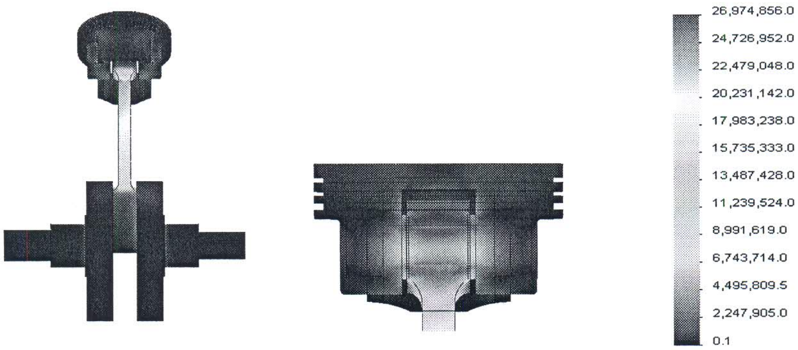
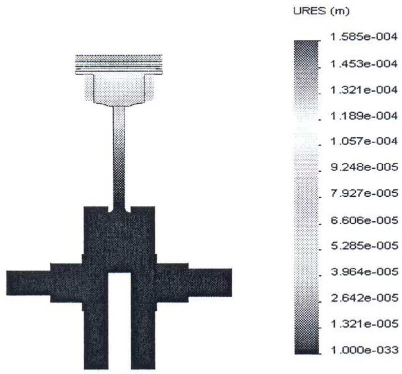


Fig. 6. Minimum factor of safety and maximum deformation with different charge pressure.

Under assembly loads on power piston-crankshaft assembly, the maximum von Mises stress value of 26.97 MPa at minimum charge pressure is located at the area between small-end connecting rod and its body because of bending moment due to compressive load, as shown in Fig. 7 (a). The stress distribution seems to be concentrated at the connecting rod is because of the force exerted on the piston head is directed to the connecting rod body. The pressure load on the piston head is equal but, in the cross sectional area of the connecting rod is minimum, the pressure rise is larger and so as the force. Moderate stress distribution is observed at the connection between the piston and the wrist pin, as shown in Fig. 7 (a). The result of the stress simulation shows that the maximum deformation occur at the top of piston head due to compressive load from the gas pressure, as shown in Fig. (b). Under this assembly load, minimum factor safety of 2 was observed with maximum compression pressure for minimum charge pressure. Meanwhile, the stress distribution seems to be lower at the crankshaft and is predicted to be safe for engine operation without exceeding the stress and deformation limit.



(a)



(b)

Fig. 7. (a) Stress distribution (b) Deformation plot for minimum charge pressure.

Conclusions

In this paper, the structural of proposed crankshaft design under compression pressure for varies charge pressure were carried out using FEA. Maximum stress distribution in the crankshaft is found not to exceed the allowable yield strength and the minimum deformation limit is observed under varies compression load. Thus, the basic design criteria with no yielding and structural failure under various charge pressure can be satisfied.

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