Twin Crank Shaft Engine Arrangement

Uzaldin S. Abdulhussain[1], Taj Elssir Hassan[2] and Maisara Mohy Eldin Gasim[3]

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مستخلص

الاحتكاك أسوأ عدو للمكائن. الاحتكاك في محرّكات الاحتراق الداخلي على الأجزاءِ المتحركة يسبب تأكل الوصلاتِ والأسطوانة مسبباً تقصير عمرَ المحرّكاتِ. في البحث الحالي، بُنِى نموذج للمرة الأولى في جامعةِ السودان للعلوم والتكنولوجيا / كليَّة الهندسة، لدراسة خصائصِ التآكل في محرك ثنائي عمود المرفق الذي يعتبرَ أحد الحلولَ لتَخفيض الاحتكاك في أسطوانات المحرّكِ. النموذج صُنِعَ معتمداً نفس معادلاتَ التصميم المستعملةْ للمحرّكاتِ التقليديةِ. الاختبارات الأولية أثبتت أن النموذج يمكن استخدامه لدِراسَة خصائصِ التآكل في محرّكاتِ الاحتراق الداخلي.

ABSTRACT

Friction is the worst enemy of machinery. Combustion engine friction and wear on moving parts, flaking of bearings and cylinder scuff cause a reduced life of engines. In the present investigation a model has been built, for the first time at Sudan University of Science and Technology / College of Engineering, to measure wear characteristics in twin crankshaft engine which considered as one of the solutions to reduce friction in engine cylinders. The model was manufactured considering the same design equations used for conventional engines. Preliminarily tests showed that the engine is capable to carry out tests to study wear characteristics in internal combustion engines.

Keywords: internal combustion engine, liner wear.

NOTATION

a	piston acceleration, m/s^2
A_{bc}	rod's small end loading area ,m^2
A_{N-N}	shank l cross-section area,m^2
A_p	cylinder cross-section area, m^2
C	static load, N/m^2
C_b	belt center distance, m
c_F	surface condition factor
c_p	the elastic coefficient
D	cylinder diameter, m
D_{bp}	outer diameter of the main pin, m
D_{ip}	inner diameter of big end pin, m
Di	inner diameter of the main pin, m
D_{is}	rod's big end inner diameter, m
d_{op}	rod's big end outer diameter ,m
d_{os}	rod's small end outer diameter, m
d_p	pitch diameter, m
D_{PL}	outer diameter of the big end pin
F	net face width, m
F_g	gas force, N/m^2
f_i	inertia force N
F_s	piston force, N
f_s	safety factor
h	horsepower
H	life time for the bearing, hours
K_B	rim-thickness factor
K_m	load distribution factor
K_o	overload factor
K_s	shape factor
K_v	dynamic factor
L	the length between the centers of the big and small pins, m
L_{PL}	the length of the big end pin, m
L_s	Rod's small end length, m
L_p	Rod's big end length, m
m	mass of the piston and piston's pin, kg
N	speed, rpm
P	gas pressure, N/m^2
P_b	dynamic load applying on bearings
P_d	diametric pitch, (1/in)
r	radius of crankshaft, m
R_p	radius of the big pulley m
r_p	radius of the small pulley, m
t_s	Rod's small end thickness, m
v	Velocity, m/s
θ	angle of crank arm, deg
σ_b	load stress,N/m^2
σ_c	compression stress, N/m^2
σ_c	contact stress for the gear, N/m^2
ω	angular velocity, rad/s

[1] Mechanical Engineering Department, Sudan University for Science and Technology
[2] Mechanical Engineerong Department, Omdrman Islamic University
[3] Mechanical Engineerong Department, Elimam Elmahdi University

1. INTRODUCTION

Combustion engine friction and wear on moving parts, flaking of bearings and cylinder scuff cause a reduced life of engines.

Friction is the force that resists the motion of one object along the surface of another. Friction depends partly upon how much surface an object has friction and also depends upon the type of surface an object has. Ordinarily, all objects have a certain amount of friction because nothing is perfectly smooth.

Friction is the worst enemy of machinery. It wears out metal, wastes power and generates heat [1]. IC engines have been around for more than a century. Due to inherent high efficiency and low cost, IC engines continue to dominate many commercial markets, from passenger cars to ocean going vessels to on-site power generation. Because of its very high efficiency, the I C engine is industry's leading prime mover, and will likely remain so for the foreseeable future [2].

In conventional internal combustion engines, each piston drives a single crankshaft through a single connecting rod extending between a wrist pin centrally located in the piston and a crankshaft pin. This arrangement has problems with balance, noise and sidewall thrust on the piston resulting in undesirable friction. About 10 % of the potentially useful power is lost to mechanical friction and this represent 50% of the total engine friction loss [3]. Even though improvements in friction performance have been made via design optimizations and lubricant improvement, no significant engine redesign has been attempted in order to re-capture friction energy.

When the reduction of friction loss is deemed worthwhile, relatively simple design changes make it possible. Rubbing of the piston rings against the cylinder walls is an example of friction loss. This particular source of loss is most severe during the combustion stroke when the hot gasses push down on the piston, which is the aim of the present investigation.

The connecting rod that links the piston to the crankshaft makes an angle with the axis of the piston so that as the piston pushes down it is itself pushed to one side, hard against the cylinder wall. By simply moving the crankshaft a bit to one side, the connecting rod can be made more upright during the combustion stroke so that the force of the piston against the cylinder wall is smaller and hence friction is reduced [3,4].

The engine under investigation is an attempt to overcome the problems associated with the conventional internal combustion engines, where in general relates to system using two crank shafts, contra-rotating, geared together, connected to a piston through two connecting rods each driven by one of the crankshafts. The model was tested and showed that the engine is capable to carry out tests to study wear characteristics in internal combustion engines cylinders.

2 ENGINE MODEL COMPONENTS DESIGN [4,5]

In order to design and manufacture a small and simple model, to carry out experimental work, the same calculations procedures used for the design of the conventional internal combustion engines are used to design the model under investigation.

2.1 Connecting rod

Force acting on connecting rod (Figure 1) is the resultant of two forces, the pressure of gas and inertia forces.

$$F_s = F_g + f_i \quad (1)$$

$$F_g = P\ A_p = P\ \pi\ r^2 \quad (2)$$

The force component (Figure 1) which acting on the connecting rod is:

$$F_c = F_s/\cos\varphi \quad (3)$$

$$f_i = m*a \quad (4)$$

There are three parts in the connecting rod, Figure 2, the small end, the big end and the connecting rod shank

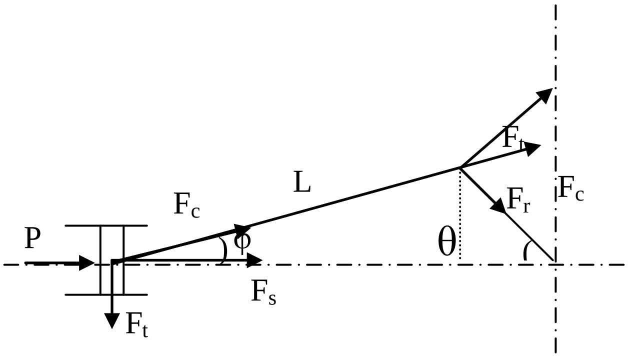

Figure 1: Force components diagram

2.1.1 Small end and big end:

The standard dimensions for small and big ends, Figures 2, 3, 4, are based on cylinder diameter (D), and given by:

$$\left.\begin{aligned} d_{os} &= (125 \text{ to } 165) * d_{is} \\ d_{is} &= (022 \text{ to } 028) * D \\ L_s &= (028 \text{ to } 032) * D \\ t_s &= (016 \text{ to } 027) * d_{is} \end{aligned}\right\} \qquad (5)$$

The stress load applying on the small end could be calculated from the equation:

$$\left.\begin{aligned} \boldsymbol{\sigma_b} &= \mathbf{F_{net} / A_b} \\ F_{net} &= F_g - f_i \\ A_{bc} &= L_s \;\; d_{is} \end{aligned}\right\} \qquad (6)$$

Low carbon steel was selected with allowable shear of100 MPa, and allowable tensile stress of 165 MPa.

The big end design standard dimensions are:

$$\left.\begin{aligned} d_{op} &= (056 \text{ to } 175) * D \\ L_s &= (045 \text{ to } 095) * d_{op} \end{aligned}\right\} \qquad (7)$$

The max loading stress is:

$$\left.\begin{aligned} \boldsymbol{\sigma_b} &= \mathbf{F_{net} / A_b} \\ F_{net} &= F_s - f_i \\ A_b &= L_p \;\; d_{op} \end{aligned}\right\} \qquad (8)$$

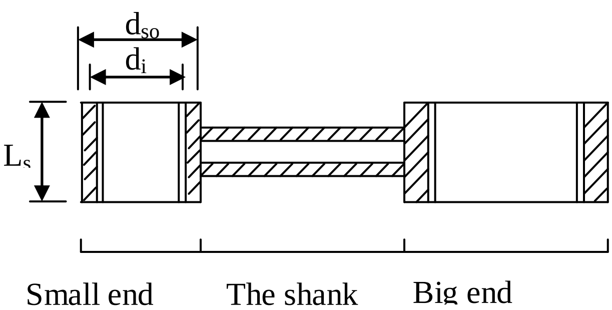

Figure 2: Connecting rod assembly

2.1.2 Shank:

Connecting rod shank is working under compression and buckling stresses which are calculated using the following equations.

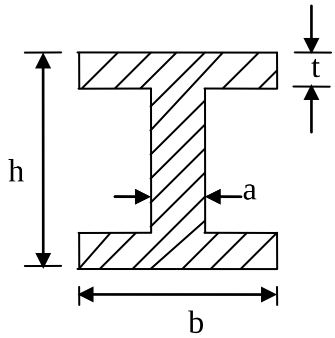

Figure 3: Shank cross-section

The compression stress is given by the following equation:

$$\left.\begin{aligned} \sigma c &= fsFnet / (AN\text{-}N) \\ Fnet &= Fg - fi \\ AN\text{-}N &= 2bxt + (hmin - 2t) * a \\ hmin &= (\, 05 \text{ to } 055) * dos \\ b &= (05 \text{ to } 06) * Ls \end{aligned}\right\} \qquad (9)$$

The Buckling stress:

$$\left.\begin{aligned} \sigma b &= fs * Ksfnet / AB\text{-}B \\ AB\text{-}B &= 2b*t + (h - 2t) * a \end{aligned}\right\} \qquad (10)$$

The two connecting rods overlap during a portion of their motion, so one of the two connecting rods use a forked configuration, with a single big end connected to the crankshaft and a double small end connected to the wrist pin. The other connecting rod has a single flattened, blade-like configuration to pass between the two spaced second ends of the forked connecting rod. Figure 4 shows the tow connecting rods.

2.2 Crank Shaft

2.2.1 Crank big end pin:

$$\left.\begin{aligned} D_{PL} &= (055 \text{ to } 07) * D \\ L_{PL} &= (045 \text{ to } 065) * D_{PL} \\ D_{ip} &= (06 \text{ to } 08) * D_{PL} \\ L &= (11 \text{ to } 125) * D \end{aligned}\right\} \qquad (11)$$

$$\left.\begin{aligned} &\text{The torque } T_e = \sqrt{(M^2+T^2)} \\ &\text{Maximum shear stress } \tau_{max}\text{: -} \\ &\boldsymbol{\tau_{max}} = Te*(d_{PL}/2)/J \\ &J = (\pi/32) * D_{PL}^{\,4} \,\{1-(D_i / D_{PL})^{4}\} \end{aligned}\right\} \qquad (12)$$

222 Crank bearing pin

$$\left.\begin{array}{l} D_{bp} = (06 \text{ to } 08)^{*}\, D \\ Di = (06 \text{ to } 08) * Dbp \\ \boldsymbol{\tau_{max} = Te^{*}(d_{bp}/2)/J} \\ J = (\pi/32) * D_{bL}^{4}\, \{1-(D_i/ D_{bp})^4\} \end{array}\right\} \quad (13)$$

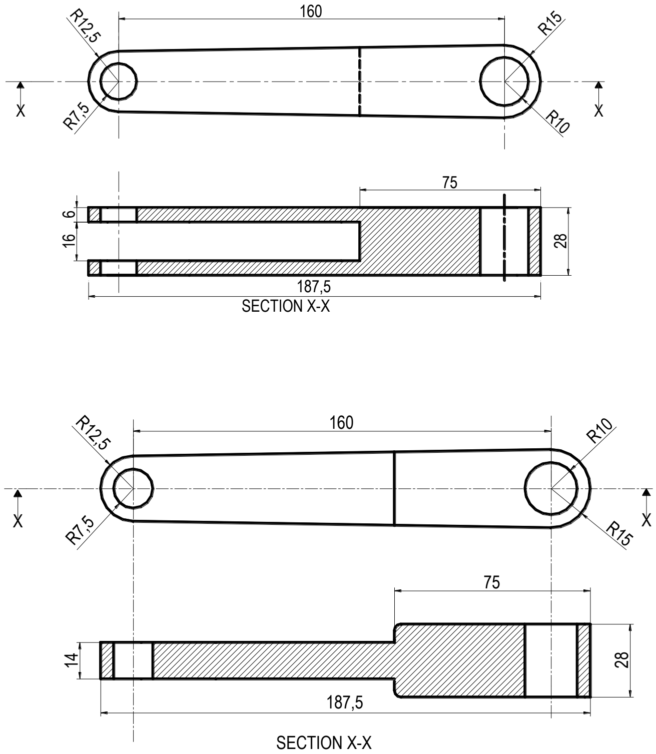

Figure 4: The two connecting rods dimensions

2.2.3 Crankshaft flange:

A flange was used instead of the crank arm in this model with dimensions to fit the arrangement shown in the model assembling, Figures 7,8,9.

2.3 Gears

For simplicity of manufacturing a spur gear type, Figure 5, was selected to gear the two crankshafts together The main parameters in gear design are the wear and bending stresses These stresses are given by the following equations.

Wear stress:

$$\sigma_c = c_p{*}(W^t{*}K_o{*}K_v{*}K_s{*}\,(K_m/d_p){*}(c_F/l) \quad (14)$$

Bending stress

$$\sigma = W^t * K_o * K_v * K_s\, (P_d/F){*}(K_m K_B/J \quad (15)$$

24 Liner

An aluminum liner was used with dimension according to design calculations, selection of aluminum is to show a significant wear compared to alloy cast iron liner used in conventional engines

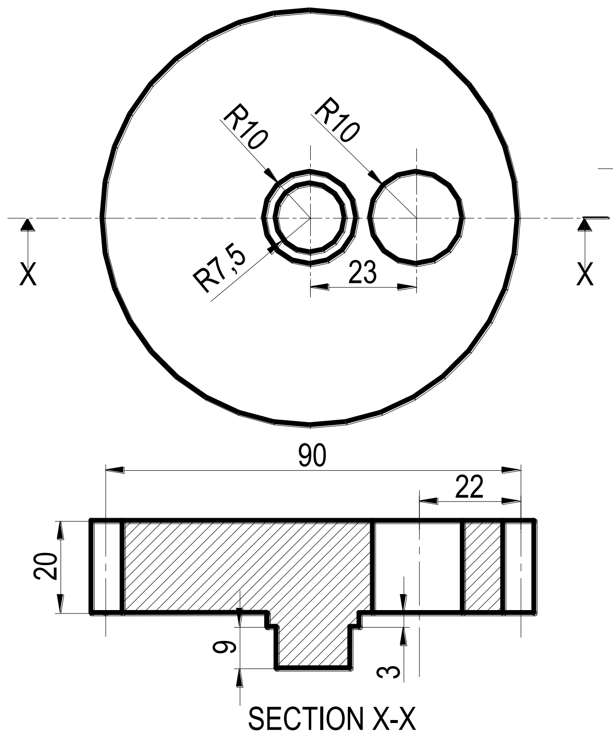

Figure 5: Gear dimensions

2.5 Piston

In this model a Honda generator engines piston used, and it has the following dimensions:

Piston diameter = 67 mm

Wrist pin diameter = 15 mm

2.6 Pulley

Different speed of piston is one of the operating variables; this was met by using two pulleys instead of using a gearbox to achieve the variable speed required.

2.7 Bearing selection

The main parameters playing a major part in the selection of engines bearing are:

Load, Life time and shaft diameter which are calculated by using the following equations:

Life time,

$$H = \{1000000*p_b^3\}/(60*N*C_b^3) \qquad (16)$$

Bearing **No6202** was selected to meet the calculated values.

2.8 Motor Selection

In order to motivate the model single vase motor of one horsepower was adapted with two variables speeds.

2.9 Belt Selection

To find the length of the belt:

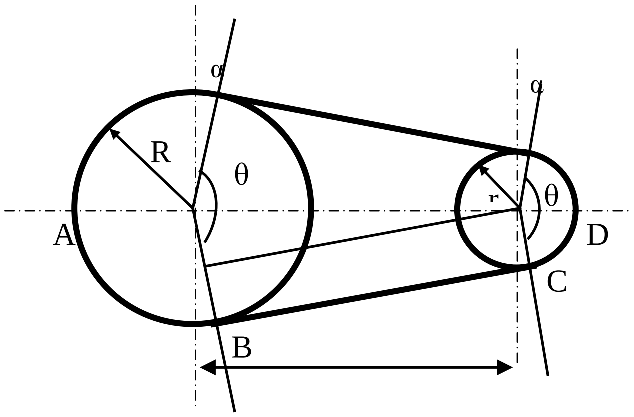

Figure 6: Belt arrangement

The length of the belt, Figure 6:

$$L = 2(AB + BC + CD)$$

$$L = 2 \{[(\pi/2) +\alpha] R + \sqrt{(c^2- (R-r)^2)} + [(\pi /2)- \alpha] r\} \qquad (17)$$

$$\alpha = \sin^{-1} (R-r)/c$$

3 DESCRIPTION OF THE ENGINE MODEL

The engine, Figure 7, basically comprises two crankshafts that are connected together in a contra-rotating relationship, crankshafts connected to the piston wristpin (10) with first (4) and second (11) connecting rods, which cross between corresponding crankshaft and wristpin. Any suitable connecting means may connect the two crankshafts to provide the desired rotation in opposite directions, such as gears, timing belts, chains or the like. Preferably, the crankshafts are geared together to assure positive equal contra-rotation. The gears (1) may be in the form of meshing single axial gears on each crankshaft The connecting rod means may have a single first connecting rod and a single second connecting rod spaced with the connecting rods spaced longitudinally along the length of the crankshafts to prevent interference. However, for best results, the first crankshaft means will consist of a pair of spaced connecting rods and the second connecting means will consist of a single connecting rod extending between the two first connecting rods in the required crossed relationship. This provides superior balance and piston stability. The strokes in twin crankshaft arrangement are not equal, because of the offset crankshaft, possessing expansion and compression strokes of differing length. The twin crankshaft configuration may typically have an expansion stroke length grater than the compression stroke length, Figure 8, This characteristic has benefits in providing improved cylinder filling with and air-fuel charge and an extended power stroke for a longer, more complete and cleaner burn.

4 CONCLUSIONS

In the present investigation a model has been built to measure wear characteristics in twin crankshaft engine. The model was manufactured considering the same design equations used for conventional engines. Preliminarily tests showed that the engine is capable to carry on tests to study wear characteristics in internal combustion engines

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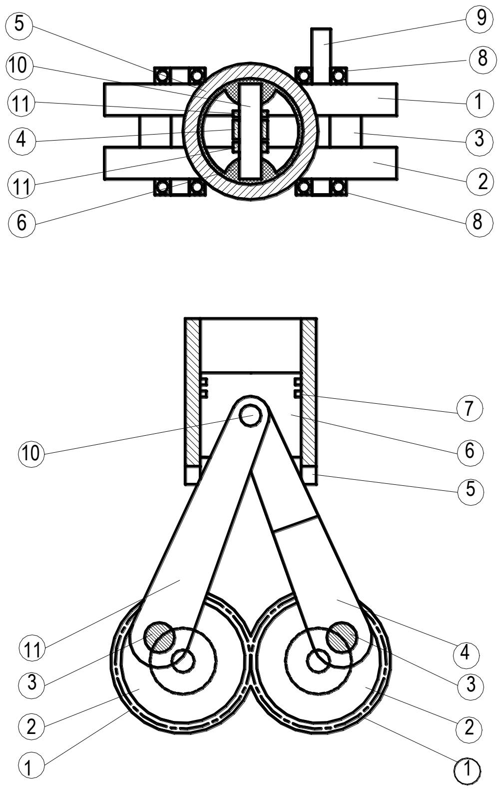

Drawing description:

1	Gear	7	Rings
2	Flywheel	8	Bearing
3	Crank pin	9	Driver shaft
4	Connecting rod 1	10	Wrist pin
5	Cylinder	11	Connecting rod 2
6	Piston		

Figure 7: Engine assembly

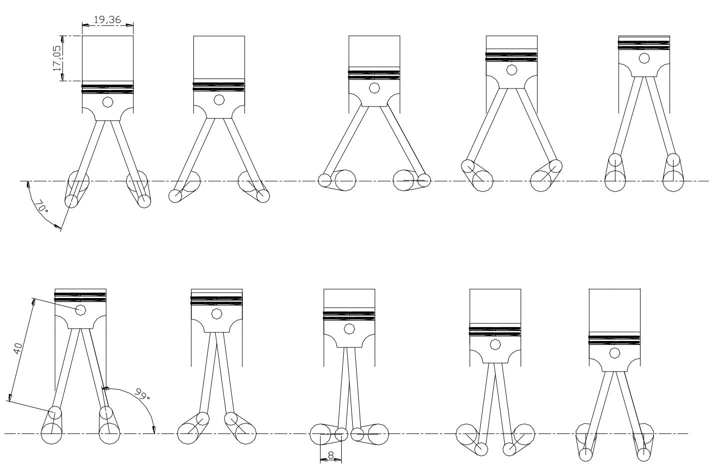

Figure (8) Piston displacement.