Dynamic Analysis of Crankshaft

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Abstract—The dynamic and inertial loading characteristics of the slider crank mechanism are studied and the necessary equations for the same are deduced. The torque and the loads acting on the crankpin are analytically determined. The numerical values required are determined using MATLAB.

Keywords—Crankshaft; Dynamic Analysis; Inertia Load; Combustion Load; Matlab.

I. INTRODUCTION

The crankshaft experiences complex loading due to the motion of the connecting rod, which transforms into two sources of loading to the crankshaft. The loading on the crankpin consists of bending and torsion. The significance of torsion during a cycle and its maximum compared to the total magnitude of loading should be investigated to see if it is essential to consider torsion during loading or not.

The objective of this paper is to determine the magnitude and direction of the loads that act on the crankpin and the crankshaft torque. An analytical approach will be used on the basis of a single degree of freedom slider crank mechanism. MATLAB programming was used to solve the resulting equations.

II. LITERATURE REVIEW

Work done by various researchers in the areas of defined problem is focused as below.

H. D. Desai [4] explained that the reciprocating engine mechanism is often analysed, since it serves all the demands required for the convenient utilization of natural sources of energy, such as steam, gaseous and liquid fuels, for generation of power.

Momin Muhammad Zia Muhammad [5] presented that the crankshaft is an important component of an engine. This paper presents results of strength analysis done on crankshaft of a single cylinder two stroke petrol engine, using PRO/E and ANSYS software. The three dimensional model of crankshaft was developed in PRO/E and imported to ANSYS for strength analysis. This work includes, in analysis, torsion stress which is generally ignored. A calculation method is used to validate the model. The paper also proposes a design modification in the crankshaft to reduce its mass. The analysis of modified design is also done.

Amit Solanki et.al [6] explained that the performance of any automobile largely depends on its size and working in dynamic conditions. The design of the crankshaft considers the dynamic loading and the optimization can lead to a shaft diameter satisfying the requirements of automobile specifications with cost and size effectiveness. The review of existing literature on crankshaft design and optimization is presented.

Farzin H. Montazersadgh and Ali Fatemi [1] [7] presented that a dynamic simulation was conducted on a crankshaft from a single cylinder four stroke engine. Finite element analysis was performed to obtain the variation of stress magnitude at critical locations. The pressure-volume diagram was used to calculate the load boundary condition in dynamic simulation model, and other simulation inputs were taken from the engine specification chart. The analysis was done for different engine speeds and as a result critical engine speed and critical region on the crankshaft were obtained. Stress variation over the engine cycle and the effect of torsional load in the analysis were investigated.

In a study carried by D B Sadaphale, and J R Chaudhari, Mahesh L Raotole [8] a dynamic simulation is conducted on forged steel crankshaft from single cylinder four stroke engine. Finite element analysis is performed to obtain the variation of stress magnitude at critical locations. The dynamic force analysis is carried out analytically using MATLAB program.

III. DYNAMIC LOAD ANALYSIS OF CRANKSHAFT

The main objective here is to determine the magnitude and direction of the loads that act on the bearing between connecting rod and crankshaft.
A. Analytical approach to determine dynamic loads

![Diagram](image)

**Inertia torque:**

As shown in the figure, the inertia force due to the mass at A has no moment arm about O2 and therefore produces no torque. Consequently, we need consider only the inertia force due to the reciprocating part of the mass. From the force polygon, the inertia torque [2] exerted by the engine on the crankshaft is:

\[ T_{21} = \frac{m_2}{2} r^2 \omega^2 \left( \frac{T_{stroke}}{21} \sin \theta - \sin \omega t - \frac{3\pi}{21} \sin \theta \right) \]

(1)

Equation (1) gives the inertia torque exerted by the engine on the shaft in the positive direction.

**Crankshaft torque:**

The torque delivered by the crankshaft to the load is called the crankshaft torque and it is negative of the moment of the couple formed by the forces \( F_{41} \) and \( F_{52} \). Therefore, it is obtained from the equation:

\[ T_{21} = -F_{52} x \omega = \left[ (m_{3g} + m_4) x + P \right] \omega \sin \theta \omega t \]

(2)

The analytical approach was solved for a general slider crank mechanism which results in equations that could be used for any crank radius, connecting rod geometry, connecting rod mass, connecting rod inertia, engine speed, engine acceleration, piston diameter, piston and pin mass, pressure inside cylinder diagram, and any other variables of the engine.

The results of the MATLAB code include linear velocity and acceleration of piston assembly, various forces between different joints in the mechanism and the crankshaft torque. In this analysis, it was assumed that the crankshaft rotates at a constant angular velocity, which means the angular acceleration was not included in the analysis. However, in a comparison of forces with or without considering acceleration, the difference is less than 3%.

B. Combustion pressure variation

The pressure versus crank angle of this specific engine was not available, so the pressure versus volume (thermodynamic engine cycle) diagram of a similar engine was considered. This diagram was scaled between the minimum and maximum of pressure and volume of the engine. The four link mechanism was then solved by MATLAB programming to obtain the volume of the cylinder as a function of the crank angle.

![Graph](image)

**Fig 1. Variation of combustion pressure over operating cycle**

Pressure versus crankshaft angle data is used as the applied force on the piston during the dynamic analysis. It should be noted that the pressure versus volume of the cylinder graph changes as a function of engine speed, but the changes are not significant and the maximum pressure which is the critical loading situation does not change. Therefore, the same diagram was used for different engine speeds in this study.

As the dynamic loading on the component is a function of engine speed, the same analysis was performed for different engine speeds which were in the range of operating speed for this engine (with the minimum engine speed of 2000 rpm). The variation of forces at various engine speeds are plotted. Comparison of magnitude of maximum torsional load and bending load at different engine speeds was shown. As the engine speed increases, the maximum bending load decreases.

The reason for this situation could be explained as follows. There are two load sources in the engine: combustion and inertia. The maximum pressure in the cylinder does not change as the engine speed changes, therefore the load applied to the crankshaft at the moment of maximum pressure due to combustion does not change. This is a bending load since it passes through the center of the crank radius. On the other hand, the load caused by inertia varies as a function of engine speed. As the engine speed increases, this force increases too. The load produced by combustion is greater than the load caused by inertia and is in the opposite direction, which means the sum of these two forces results in the bending force at the time of...
combustion. So as the engine speed increases the magnitude of the inertia force increases and this amount is deducted from the greater force which is caused by combustion, resulting in a decrease in total load magnitude.

IV. RESULTS AND DISCUSSION

The dynamic characteristics of the engine mechanism are studied in detail with the help of slider rank mechanism. The variation of piston velocity, piston acceleration with respect to crankshaft angle are obtained in graphical format and are shown below. The variation has been studied for different engine speeds like 2000 rpm, 2400 rpm, 2800 rpm, 3200 rpm, 3600 rpm respectively. The results are tabulated. The variation of load on the crankpin and torque on the crankpin are studied at different engine speeds. From the Matlab data it is clear that the net load and torque decrease with increasing engine speed and that the lower rpm range i.e, 2000 rpm is the critical engine speed. The reason is that below 2000 rpm speeds are transient in nature and only after attaining 2000 rpm can the engine loading and torque be steady. The graphs and comparisons are as follows

Fig 2. Variation of piston velocity at 2000 rpm

Fig 3. Variation of piston acceleration at 2000 rpm

Fig 4. Variation of crankpin load at 2000 rpm

Fig 5. Variation of crankshaft torque at 2000 rpm

Fig 6. Variation of piston velocity at 3600 rpm

Fig 7. Variation of piston acceleration at 3600 rpm
Fig 8. Variation of crankpin load at 3600 rpm

Fig 9. Variation of crankshaft torque at 3600 rpm

Net load on crankpin (N)

At 2000 rpm At 3600 rpm

Net torque (Nm)

At 2000 rpm At 3600 rpm

Fig 10. Comparison of maximum load on the crankpin at different engine speed.

Fig 11. Comparison of maximum crankshaft torque at different engine speed.

Table 1. Tabulation of max crankpin load and max crankshaft torque at different speeds.

<table>
<thead>
<tr>
<th>Speed</th>
<th>Max crankpin load at combustion point</th>
<th>Max crankshaft torque at combustion point</th>
</tr>
</thead>
<tbody>
<tr>
<td>2000</td>
<td>2.2148e4</td>
<td>438.7397</td>
</tr>
<tr>
<td>2400</td>
<td>2.1602e4</td>
<td>430.6314</td>
</tr>
<tr>
<td>2800</td>
<td>2.0958e4</td>
<td>421.0511</td>
</tr>
<tr>
<td>3200</td>
<td>2.0214e4</td>
<td>409.9991</td>
</tr>
<tr>
<td>3600</td>
<td>1.9373e4</td>
<td>397.4786</td>
</tr>
</tbody>
</table>

Fig. 12 – variation of max crankshaft torque with speed range

Fig. 13 – Distribution of load on crankpin. Fx is the bending load, Fy is the torsional load and Fz is the axial load.

C. Effect of torsional load

In this specific engine with its dynamic loading, the torsional load has negligible effect on the stresses.
induced. The main reason for torsional load not having much effect on the stress range is that the maximum of bending and torsional loading happen at different times during the engine cycle. In addition, when the main peak of the bending takes place the magnitude of torsional load is zero.

V. CONCLUSION
Crankshaft is a very important component in an engine which helps in the conversion of reciprocating motion of piston to final rotary output. Dynamic Analysis of the crankshaft has been proven to be very helpful in analysis of load and torque on the crankshaft. These data are used further in FEM platforms like Ansys to determine the stresses induced in the crankshaft. The FEM analysis results are used in the design optimization of crankshaft.

Numerical analysis platform Matlab has been proven to be very useful in the analysis of slider crank mechanism as complex equations are easily solved in a short time and the accuracy of solutions is also very good.

VI. REFERENCES

VII. APPENDIX
Matlab code for dynamic analysis:
...clear all
clc
load theta_p2.mat
r= 0.037; %crank length
l= 0.12078; %length of connecting rod
m2 = 3.7191; %mass of crank
m3 = 0.283; %mass of connecting rod
R=0.03698; %crank throw
I2 = 0.663e-3; %moment of inertia of crankrod
la = 0.0286; %location of center of gravity from the crank pin end
lb = l-la;
rg=r; %center og gravity of crank displaced outward along crank from the axis of rotation
m4 = 417.63e-3; %mass of the piston
dp = 0.089; %diameter of the piston
vc=0.0035; %clearance volume
N=2000; %speed in rpm
omega1= 2*pi*N/60; %angular velocity
alpha1= 0; %constant angular velocity
theta_rad = theta*pi/180;
P=(p*10^5)*pi/4*dp^2; %converting pressure from bar to N/mm^2 by multiplying p with 10^5
%location of piston rp wrt origin
rpx=l-
r^2/(4*l)+r*(cos(theta_rad)+(r/(4*l))*cos(2*theta_rad));
%velocity of piston
vpx= -1*r*omega1*(sin(theta_rad)+(r/(2*l))*sin(2*theta_rad));
%acceleration of piston
apx= -r*alpha1*(sin(theta_rad)+(r/(2*l))sin(2*theta_rad))
r*omega1^2*[cos(theta_rad)+r/l*cos(2*theta_rad)]

%force of cylinder wall acting against the piston
tanphi=

r/l*sin(theta_rad).*(1+r^2/(2*l^2)*(sin(theta_rad)).^2);

phi=atan(tanphi);

f41_1= P.*tanphi;
f41_1=-f41_1;

%torque delivered to the crankshaft by the gas force
t21_1= f41_1.*rpx;

%f21_1=(P*r).*sin(theta_rad).*(1+(r/l)*cos(theta_rad));

%inertia forces of rotating parts
fx_rot=

ma*r*(alpha1*sin(theta_rad)+omega1^2*cos(theta_rad));

fy_rot= ma*r*(-alpha1*cos(theta_rad)+omega1^2*sin(theta_rad));

%inertia forces of reciprocating parts
fx_rec=

mb*r*alpha1*(sin(theta_rad)+r/(2*l)*sin(2*theta_rad))+mb*r*omega1^2*(cos(theta_rad)+r/l*cos(2*theta_rad));

%total inertia forces
fx= fx_rot+fx_rec;
fy= fy_rot;

%torque delivered to the crankshaft by the inertia force
t21_2= mb*apx.*tanphi.*rpx;

%or expanding the above equation, we get

%t21_2= mb/2*r^2*omega1^2*(r/(2*l))*sin(2*theta_rad)-sin(2*theta_rad)-r/(2*l)*sin(3*theta_rad));

%net crankshaft torque
t21= t21_1+t21_2;

%total work done
wd= trapz(theta_rad,t21);

%power transmitted
power= t_mean.*2*pi*N/60;

%bearing loads
f41_2= -m4*apx.*tanphi; %f41_2 = f41_y_2
f34x_2= m4*apx;
f34y_2= -m4*apx.*tanphi;
f32x_2= -f34x_2;
f32y_2= -f34y_2;

f12y_2= f34x_2;
f34 = f12x_2;

f32 = sqrt(f32x.^2+f32y.^2);

%resultant bearing loads
f41= f41_1 + f41_2 + f41_3;

%inertia of crankpin
Irec= 0.5*mb*R^2;

%Inertia of crankpin
Irot= 0.5*ma*R^2;

I= Irec*Irot;