International Journal of Research in Advent Technology, Vol.2, No.2, February 2014
E-ISSN: 2321-9637

Static and Dynamic Analysis for Stress Calculations on Crank Pin

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Abstract- The stress analysis of a single cylinder crankpin of TVS Scooty Pep crankshaft assembly is discussed using stress analysis in this paper. Three-dimension models of crankshaft and crankpin forces were created using Pro/ENGINEER software and software ANSYS was used to analyze the stress status on the crankpin. The maximum deformation, maximum stress point and dangerous areas are found by the stress analysis. The relationship between the crank rotation and load acting on crank pin would provide a valuable theoretical foundation for the stress calculation and improvement of crankpin and engine design. [2]

Index Terms- — stress analysis; crankshaft; crankpin

1 INTRODUCTION

Crankshaft of Internal Combustion Engine is a well known phenomenon. The problem of their premature failure has attracted several investigators for over a century. The extensive studies have been made to identify the cause of failure and several have been listed.

Forces acting on the crankpin are complex in nature. The piston and the connecting rod transmit gas pressure from the cylinder to the crankpin. It also exerts forces on the crankpin, which is time varying. In this project one crank model of TVS Scooty pep will used to calculate the effect of stresses.

Crankshaft consists of the parts which revolve in the main bearings, the crankpin to which the big ends of the connecting rod is connected, the crank arms or webs (also called cheeks) which connect the crankpins and the shaft parts. The crankpin is like a build in beam with a distributed load along its length that varies with crank position. [1]

Reasons for Failure of crankshaft assembly and crankpin may be –

A) Shaft misalignment
B) Vibration cause by bearings application
C) Incorrect geometry(stress concentration)
D) Improper lubrication
E) High engine temperature
F) Overloading
G) Crankpin material & its chemical composition
H) Pressure acting on piston

2 OVERVIEW

In the process of converting the linear reciprocating motion of the pistons into a rotational output, the crankshaft undergoes both bending and torsion. [3]As these forces are transmitted through the crankshaft, it becomes highly stressed, particularly so at the crankpin/web and the journal/web intersections of the cranked shaft. As a consequence, fillet radii are used in these areas to reduce the stresses, but if the shaft is not carefully designed, these stresses can still reach unacceptably high levels with regard to material strength and fatigue life. [5]

Engine specifications		
1	Cylinder bore	51mm
2	Stroke	43mm
3	Piston displacement	87.8cc
4	Compression ratio	10.1:1
5	Maximum power in KW	3.68@6500rpm
6	Maximum Torque in Nm	5.80@4000rpm
7	Max. Speed	60km/hr
8	Bearing pressure	7-12.5 N/mm^2

Tabl 1. Engine specification of TVS Pep

Fig.1 (20CrMo) Precision Crank Pin

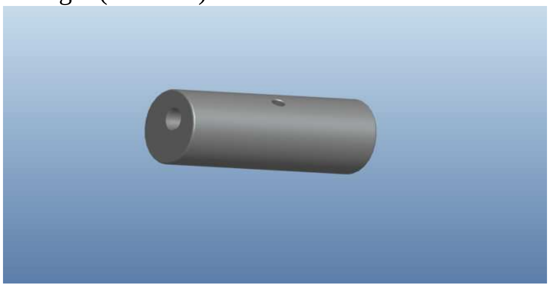


Fig.2 Model of crank pin

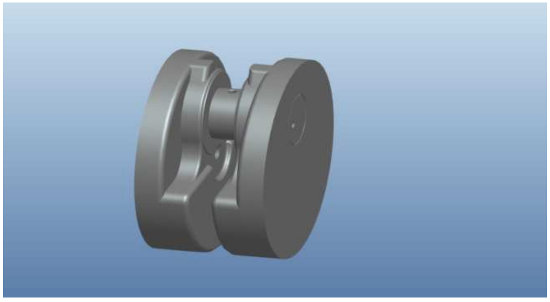


Fig.3 Model of crankshaft assembly

2.1 *Design specification*

Crankpin material –20CrMo

(Chromium nickel Alloy steel)

Crankpin diameter – 23mm

Crankpin axial length – 40mm

Crank web thickness – 15mm

Crank web Height – 85mm

Length of connecting rod – 90mm

Material used	Endurance Limit (MPa)		Allowable strength (MPa)	
Stresses	Bending	Shear	Bending	Shear
Crome Nickel	525	290	130 to175	72.5 to 97
Carbon steel and cast steel	225	124	56 to 75	31 to 42
Alloy cast iron	140	140	35 to 47	35 to 47

Table 2.Crank Material and design Stresses

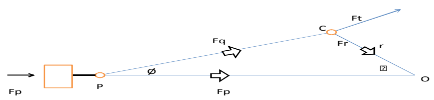
3 OBJECTIVES

Analyze the stresses acting on crank pin due to the gas force. Analyze the maximum deformation, maximum stress point and dangerous areas of failure. Optimize the design to reduce the rate of failure and improve the life of crank shaft and engine also if possible.

3.1 Methodology and design calculations

Let

P = max. Pressure of gas
D =Diameter of piston
mR= mass of reciprocating parts
ω= angular speed of crank
ϴ= angle of inclination of crank from top dead center
Ø= angle of inclination of connecting rod with the line of stroke
r= radius of crank
l= length of connecting rod
n= ratio of (l/r)
dc= diameter of crank pin
lc= length of crank pin
FL = Force on piston due to gas pressure
i.e. ($p_* A$)



FI = Inertia force of reciprocating part
i.e. ($mR_* \omega^2{}_* r$ ($\cos\Theta + \cos 2\Theta / n$))
Fp=net force on crank pin
$F_L \pm F_I$
Fc = force on connecting rod

Fc=Fp/cosØ
F_I= inertia force on crank pin
m*r*ω2
Pc = load on crank pin
dc*lc*pbc
F_T= tangential force on crank pin
F_Q sin (ϴ+Ø)

F_R= radial force on crank pin

F_Q cos (ϴ+Ø)
H_{T1} and H^{T2} =reaction at bearing due to F_T
F^T*b_1/b and F_T*b_2/b
H_{R1} and H_{R2} = reaction at bearing due to FR
F_R*b_1/b and F_R*b_2/b
Mc = max.bending moment on crank pin
$H_{R1}*b_2$
Tc = max.twisting moment on crank pin
$H_{T1}*r$

Te = equivalent twisting moment on crank pin
√ [$(Mc^2) + (Tc^2)$]
τ = max. Shear stress on pin
Te/ (π/16)*dc^3

Let max. Bearing pressure (Pb) – 12.5 N/mm2
So, load on crank pin (FL) = dc*lc*Pb = (23*40*12.5) =11500N.
Max. Gas pressure on piston (P) = (π/4 * D2)/ (FL) =5.62 N/mm2
Net force acting on crank pin (Fp) = (FL), neglecting inertia force.

Thrust on crank pin (Fq) = FL/cosØ,
Where Ø = 28
Fq =13024.5 N

SN	**Ø**	**ϴ**	**F_T(N) = F_q sin (ϴ+Ø)**	**F_R(N)= F_q cos (ϴ+Ø)**
1	28	90	$11.4*10^3$	$-6.1*10^3$
2	0	180	0	$-12.3*10^3$
3	28	270	$-11.4*10^3$	$6.1*10^3$
4	0	360	0	$12.3*10^3$

Table 3.Tangential force and Radial force

SN	**Ø**	**ϴ**	**HT1 = F_T*b_1/b**	**HR1 = F_R*b_1/b**
1	28	90	$5.7*10^3$	$-3.05*10^3$
2	0	180	0	$6.16*10^3$
3	28	270	$-5.7*10^3$	$3.05*10^3$
4	0	360	0	$6.16*10^3$

Table 4.Bearing Reactions

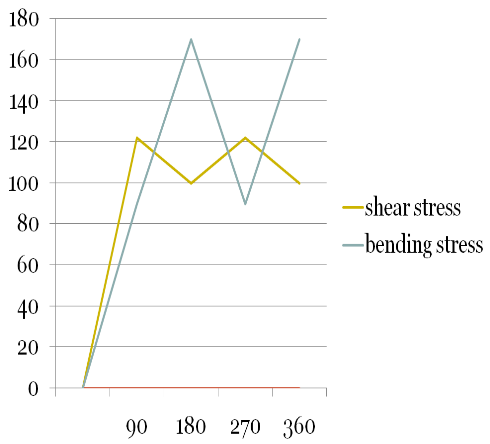
SN	**Ø**	**ϴ**	**Mc= $H_{R1}*b_2$**	**σb = Mc/ [π/32*dc^3] N/mm2**
1	28	90	$-106.7*10^3$	90
2	0	180	$215.6*10^3$	180
3	28	270	$106.7*10^3$	90
4	0	360	$-215.6*10^3$	180

Table 5.Bending Stresses

E-ISSN: 2321-9637

S N	Ø	ϴ	Te = √($(Mc)^2$)+($(Tc)^2$)]	τ = [Te/(π/16)*dc^3] N/mm2
1	28	90	$267.3*10^3$	112
2	0	180	$215.6*10^3$	90
3	28	270	$267.3*10^3$	112
4	0	360	$215.6*10^3$	90

Table6.shear Stresses



Graph 1.. Crank angle Vs Stresses
X- CRANK ANGLE, Y- STRESSES

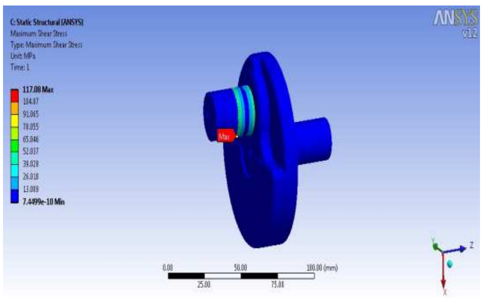


Fig.4 Model of crankpin failure(ANSYS)

4. DYNAMIC ANALYSIS APPROACH
To identify the shear stresses on crank pin of crankshaft assembly

Let,
P = gas force on piston
S = wrist pin effort
Q = thrust on connecting rod.
Fr = radial force on crank pin
T = tangential force on crank pin
r = radius of crank

Now,
D =diameter of piston = 0.051m.
Dc = diameter of crank pin = 0.023m.
Pa = atm. Pressure of air = 0.1032 N/mm2
Pd = driving pressure = 6.90 N/mm2
Pb = backpressure = 6.70N/mm2
Pd –Pb = 0.20 N/mm2
R = weight of reciprocating parts i.e., (mass of piston assy. + 1/3 weight of connecting rod)
= (90 + 77)gm. = 0.167 kg
N = 1500 rpm.
ω = 2πN/60 =157 rad/sec.
Ø = sin-1(rsinϴ/l)

Where, length of connecting rod = 0.090m.
Now,
X = (r(1-cosϴ)+l(1-cosØ))
Where, X = displacement of piston at various phase of crank rotation.
Fp = P *(πD2)/4
Now, Fi = (R/g)ω2.r[cosϴ+(cos2.ϴ)/n
Where n=l/r = 2.09

Now, piston Effort(PE)
PE = Fp±Fi
When piston move from TDC to BDC (Fp-Fi) & BDC to TDC (Fp+Fi)
Now, thrust on crankpin
Q = PE/cosØ
Tangential force on crank pin
T = Qsin(ϴ+Ø)
Now, Turning moment on crankshaft due to tangential force.
Tm = T*r
Now, max. shear stress on crank pin due to tangential force (T)
τ = Tm/((π/16)*dc3)

S. N.	**Te = √(Mc^2)+(Tc^2)] (N-mm)**	**τ = [Te/(π/16)*dc^3] N/mm2**	**ANSYS RESULTS (Max. Shear stress) N/mm2**
1	**$267.3*10^3$**	**112**	**117.08**
2	**$215.6*10^3$**	**90**	**97.56**

Table7.Result Comparison

5. CONCLUSION

Analytically it is to be concluded that the value of Max. Shear stress is 112N/mm^2 and Max. Bending stress is 180 N/mm^2 which is more than the value of allowable strength of material without taking any factor of safety. The relationship between the crank rotation and force acting on crank pin would provide a valuable theoretical foundation for the material selection and design optimization of crankpin and engine design.

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