Stress Analysis And Design Optimization Of Crankpin

S. M. Sorte, S. M. Sheikh

Abstract— the stress analysis and design optimization of a single cylinder crankpin of TVS Scooty Pep crankshaft assembly are 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 optimization and improvement of crankpin and engine design. [2]

Keywords— stress analysis; crankshaft; crankpin

I. 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 –

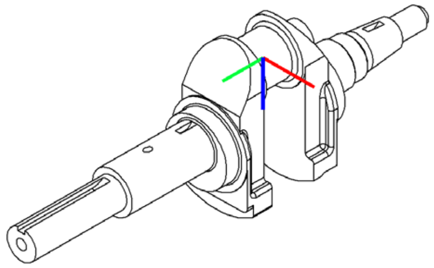
A) Shaft misalignment
B) Vibration cause by bearings application
C) Incorrect geometry(stress concentration)
D) Improper lubrication
E) High engine temperature
F) Overloading
G) Crankpin material & its chemical composition
H) Pressure acting on piston

II. 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]



Force acting on crankpin

Radial force, Tangential force, Inertia force STRESSES DUE TO FORCES ON CRANK PIN Bending stress and Shear stress

III. ENGINE SPECIFICATION

Table 1. Engine specification of TVS Pep

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

Fig.1 (20CrMo) High Precision Engine Crank Pin

Manuscript received on March, 2013.

Mr.sandip M.Sorte, M.Tech (CAD/CAM), RTM Nagpur/ G.H.Raisoni acadamy/Nagpur, India.

Prof.S.M.Sheikh, M.Tech (Design), University/ College/ Organization Name, City Name, Country Name., India.

Fig.2 Crankshaft of TVS pep

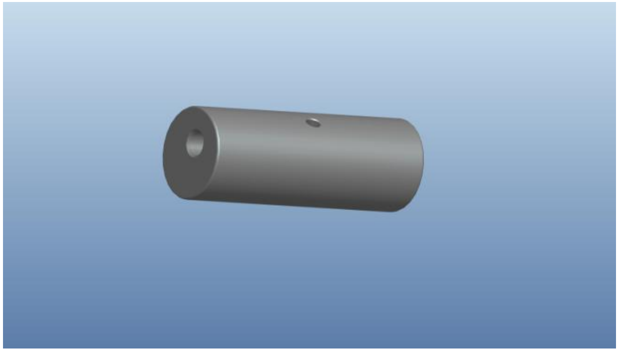


Fig.3 Model of crank pin

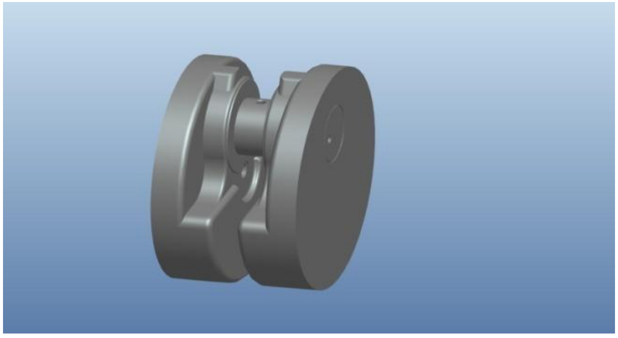


Fig.4 Model of crankshaft assembly

IV. 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

Table 2.Crank Material and design Stresses

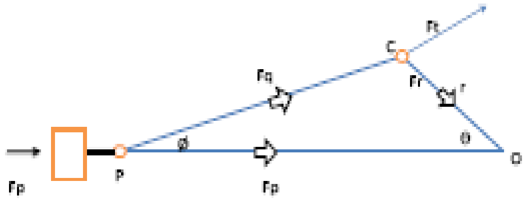
MATERIAL USED	ENDURANCE LIMIT (MPA)		ALLOWABLE SHEAR STRESS (MPA)	
STRESSES	BEN DING	SHEAR	BENDING	SHEAR
CROME NICKEL	525	290	130 TO175	72.5 TO 97
CARBON STEEL AND CAST STEEL	225	124	56 TO 75	31 TO 42
ALLOY CAST IRON	140	140	35 TO 47	35 TO 47

V. OBJECTIVE

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.

VI. METHODOLOGY

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 pressurei.e. (p_*A)



FI = Inertia force of reciprocating part
i.e. ($mR_*\omega^2{}_*r$ (cosϴ+cos2ϴ/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
$\sqrt{[(Mc^2) + (Tc^2)]}$
τ = max. Shear stress on pin
Te/ (π/16)*dc^3
Let max. Bearing pressure (Pb) – 12.5 N/mm2So, load on crank pin

(F_L) = dc*lc*Pb = (23*40*12.5) =11500N.

Max. Gas pressure on piston (P) =

(π/4 * D2)/ (F_L) =5.62 N/mm2

Net force acting on crank pin (Fp) = (F_L), neglecting inertia force.

Thrust on crank pin (Fq) = F_L /cosØ,

Where Ø = 28

F_q =13024.5 N

VII. STRESS CALCULATION RESULT

Table 3.Tangential force and Radial force

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

Table 4.Bearing Reactions

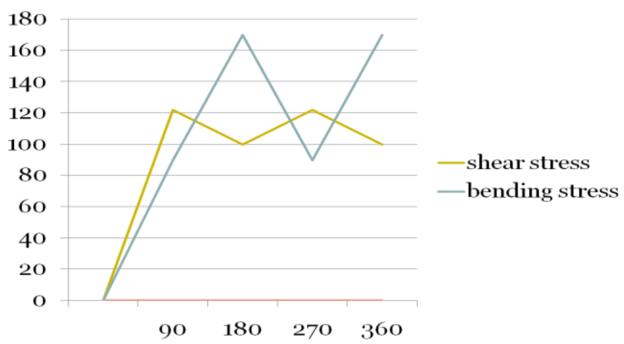
SN	Ø	Θ	HT1 = F_T *b /b_1	HR1 = F_R *b /b_1
1	28	90	5.7*10	-3.05*10
2	0	180	0	6.16*10
3	28	270	-5.7*10	3.05*10
4	0	360	0	6.16*10

Table 5.Bending Stress

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

Table 6.Shear Stress

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



Graph 1. Crank angle Vs Stresses

X- CRANK ANGLE, Y- STRESSES

VIII. 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 stress value 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.

ACKNOWLEDGMENT

I would like to express my gratitude to the many people who have assisted me during the course of this research. Special thanks must go to my supervisors Professor S.M.SHEIKH Warrior for their continued support, guidance and friendship. Thanks must also go to the works manager and whole team of A.K.Gandhi TVS group,Wardha who have both greatly assisted in my understanding of the subject.

REFERENCES

1. "A Generalized Methodology for the Analysis and Simulation of Four Stroke Diesel Engine Crankshaft" Vol.1,Issue.X/April 2012
2. Jian Meng "Finite Element Analysis of 4- Cylinder Diesel Crankshaft" Published Online August 2011 in MECS
3. Mohammad Reza Asadi "Analysis of Stress in the Nissan Z-24 Moulding Crankshaft "Australian Journal of Basic and Applied Sciences, 5(12): 798-803, 2011
4. H. D. Desai "Computer Aided Kinematic and Dynamic Analysis of a Horizontal Slider Crank Mechanism Used for Single-Cylinder Four Stroke Internal Combustion" , Paper published 2009 –
5. M.A.Alfares"Failure analysis of a vehicle engine crankshaft", Paper published 2007 in ASM international –
6. Farzin H. Montazersadgh and Ali Fatemi "Dynamic Load and Stress Analysis of a Crankshaft" Published 2007 SAE International
7. M. A. Alfares Æ A. H. Falah Æ A. H. Elkholy "Failure Analysis of a Vehicle Engine Crankshaft" Submitted: 24 June 2006 / in revised form: 1 December 2006 / Published online: 17 March 2007 ASM International 2007
8. Anthony P. Sime, B.Eng "Stress analysis of overlapped crankshafts" Thesis submitted to the University of Nottingham For the degree of Doctor of Philosophy October1998
9. EstimationV. Prakash and K. Aprameyan"An FEM Based Approach to Crankshaft Dynamics and Life" Bharat Earth Movers Ltd... SAEInternational Congress and Exposition Detroit, MichiganFebruary 23-26, 1998
10. " FEM based approach to Crankshaft dynamics and life estimation "Paper published 1998 SAE The variation of life of crankshafts due to changes in their material properties.
11. R.S.Khurmi, J.K.Gupta" Text book of Machine Design" page no.1125 -1200.

AUTHOPRS PROFILE

Mr.Sandip M.Sorte,Pursuing M.Tech (CAD/CAM) from G.H.Raisoni Acadamy,Nagpur ,field of Intrest is Automation and machine design.

Prof. S.M.Sheikh, M.Tech (Design) and Professor in G.H.Raisoni Acadamy,Nagpur ,field of Intrest is Automation and machine design.