Analysis and Optimization of Clamping Jaws & Pins for SPM by using FEA

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Abstract— This is About Anylising Failure of SPM Which is Used in Tata Cummins Ltd., Phaltan. This SPM is used for Lifting Engine for Valve Adjusting. We are going to Analyze Breakdown occurring Due to Failure of Pins Fixed on Clamping Jaws When Engine is lifted. So We Need to Analyze and will also Optimize Material for Designing of Pin.

Keywords: Pin, Clamping Jaws, Analysis, Optimized

I. INTRODUCTION

Gati Automation is a firm which designs & manufactures special purpose machine as per customer requirement. This SPM is designed for Tata Cummins LTD Phaltan. SPM is used for lifting the component for valve tightening operation. In this project we need to diagnose and resolve customer's problem.

Engine is lifted from pallet with the help of clamping jaws. Pins are fixed on clamping jaws which get inserted in engine holes. Pneumatic cylinder's are used to operate clamping jaws. Clamping jaws are fixed on a carriage plate which moves up and down. Carriage plate is connected with a ball screw and ball screw runs with the help of a geared motor, which is mounted on top of the machine. Another motor is fixed on one side of clamping jaw, which is also mounted on carriage plate through a bracket which moves along with clamping jaw. This motor is used to rotate crankshaft which helps in rotating camshaft. As the camshaft is rotated, it also helps in tightening and loosening of valves mounted on the camshaft.

In material handling system mostly this type of SPM is used for lifting components. As the engine is to be lifted for valve tightening process, it is found that there is misalignment of engine, which could fall on the pallet and lead to a major breakdown. So we are running on this project.

SPM helps to achieve production target with high quality. Present SPM is used for work holding and lifting the automobile component .The work holding device has two important components i.e. clamping jaws and locating pins, which plays a vital role in the operation of SPM as shown in fig.(1). Work holding refers to any device that is used to a secure a work piece and hold and lift from pallet which moves on conveyor.

There is problem of deformation in jaws and failure of pins which affects the production. This work is undertaken to do stress analysis and optimization of clamping jaws and pins for SPM using FEA for solving above problem

II. PROBLEM DEFINITION

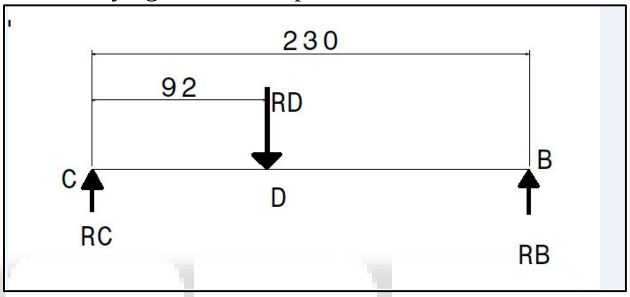
When the engine is lifted for valve tightening there was as a change in angle of engine position, which was visible. So when engine was unloaded and inspected we found damaged pins on LH clamping jaw

A. *Objective of the Study*

1) To analyze the causes of failure.
2) To carry out design calculation analytically.
3) To minimize the deflection /stress analysis & FEM
4) Static analysis.

III. CALCULATION

Design of Pin:-Four pins carry component load which are fixed on clamping jaws. RH jaw has one pin and LH has three pins. Load on pins acted vertical so only two pins is subjected to load carrying out of three pins.



At Point 'C' –

$-R_B \times 230 + R_D \times 92 = 0$

$-R_B \times 230 + 5150.25 \times 92 = 0$

$-230\, R_B + 473.823 \times 10^3 = 0$

$R_B = 2060.1N$

At Point 'D' –

$R_C \times 92 - R_B \times 138 = 0$

$92\, R_C = 138 R_B$

$R_C = 3090.15N$

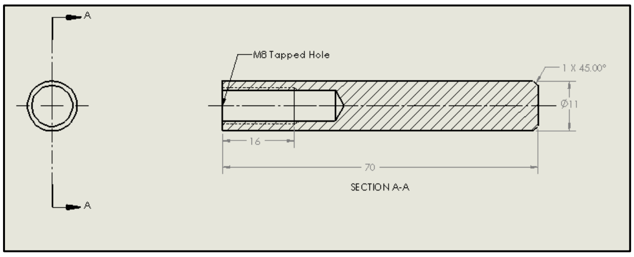


Fig. 1: Previous Pin

Maximum Deflection of Pin: Ys

$= (P \times L^3) / (3 \times E \times I)$

E = Modulus of Elasticity

$= 2.1 \times 10^5$ N/mm^2 (From Design Data Book, P.S.Gill)

Moment Of Inertia (I) = $(\pi/64) \times d^4$

= d = 11mm

I = 718.68 mm^4

$= (2060.1 \times 70^3) / (3 \times 2.1 \times 10^5 \times 718.68)$

Ys = 1.560 mm

Maximum allowable deflection = 0.1% of Total span

Span is 70mm (So design is fail)

Step II: Bending Stress

σ_b = Bending Stress

$\sigma_b = (M_b \times Y) / (I)$

$M_b = P \times e$
$= 2060.1 \times 70$
$= 144207$ N-mm
$\sigma_b =$ ((144207 x 5.5) / (718.68))
$\sigma_b =$ 1103.60 N/mm^2
Step III: Maximum Shear Stress
τ_{max} = Maximum Shear Stress
$\tau_{max} = \sqrt{(\sigma_{b/2})^2 + (\tau)^2}$
$\tau = P/A$
$= 2060.1 / ((\pi/4) \times d^2)$
$= 2060.1 / ((\pi/4) \times 11^2)$
$\tau = 21.67$ N/mm^2
$\tau_{max} = \sqrt{(1103.60/2)^2 + (21.67)^2}$
$\tau_{max} = 552.22$ N/mm^2
$\tau_{max(Std)} = 0.5(Syt)/F.S.$
Syt = 420 N/mm^2
$\tau_{max(Std)} = (0.5 \times 420)/1.5$
$\tau_{max(Std)} = 140$ N/mm^2
$\tau_{max} > \tau_{max(Std)}$
Maximum stress is greater than allowable Shear Stress so this design is unsafe or failed.

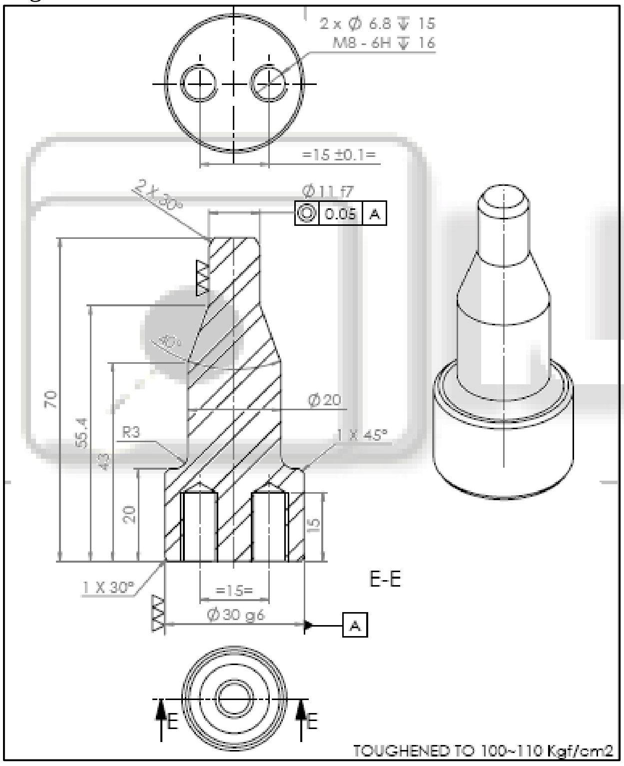


Fig. 2: (Pin – 1)

Moment Of Inertia (I) = $(\pi/64) \times d^4$
d = 11mm
I = 718.68 mm^2
P = 2060.1N
$I = (\pi/64) \times d^4$
$= (\pi/64) \times 11^4$
I = 718.68 mm^4
Maximum Deflection of Pin: $(Ys)_1$
$= (P \times L^3) / (3xE \times I)$
E = Modulus of Elasticity
$= 2.1 \times 10^5$ N/mm^2 (From Design Data Book, P.S.Gill)
$= (2060.1 \times 14.6^3) / (3 \times 2.1 \times 10^5 \times 718.68)$
$(Ys)_1 = 0.0141$ mm
Maximum allowable deflection = 0.1% of Total span
So that Design is safe (Span = 14.6mm)
Maximum Deflection of Pin: $(Ys)_2$
$= (P \times L^3) / (3 \times E \times I)$
E = Modulus of Elasticity
$= 2.1 \times 10^5$ N/mm^2 (From Design Data Book, P.S.Gill)
$= (2060.1 \times 6.2^3) / (3 \times 2.1 \times 10^5 \times 2833.326)$
$(Ys)_2 = 0.000275$ mm
Maximum allowable deflection = 0.1% of Total span
So that Design is safe (Span = 6.2mm)
Maximum Deflection of Pin: $(Ys)_3$
$= (P \times L^3) / (3 \times E \times I)$
E = Modulus of Elasticity
$= 2.1 \times 10^5$ N/mm^2 (From Design Data Book, P.S.Gill)
$= (2060.1 \times 23^3) / (3 \times 2.1 \times 10^5 \times 7853.98)$
$(Ys)_3 = 0.00506$ mm
Maximum allowable deflection = 0.1% of Total span
So that Design is safe (Span = 23mm)
Maximum Deflection of Pin: $(Ys)_4$
$= (P \times L^3) / (3 \times E \times I)$
E = Modulus of Elasticity
$= 2.1 \times 10^5$ N/mm^2 (From Design Data Book, P.S.Gill)
$= (2060.1 \times 17^3) / (3 \times 2.1 \times 10^5 \times 636172.51)$
$(Ys)_4 = 2.52 \times 10^{-5}$ mm
Maximum allowable deflection = 0.1% of Total span
So that Design is safe (Span = 17mm)
Note: Maximum deflection consider for design of pin, Span 14.6 is max deflection points. ($(Ys)_1 = 0.0118$mm)
Step II: Bending Stress
σ_b = Bending Stress
$\sigma_b = (M_b \times Y) / (I)$
$M_b = P \times e$
$= 2060.1 \times 14.6$
$= 30.08 \times 10^3$ N-mm
$\sigma_b = ((30.08 \times 10^3 \times 5.5)/(718.68))$
$\sigma_b = 230.198$ N/mm^2
Step III: Maximum Shear Stress
τ_{max} = Maximum Shear Stress
$\tau_{max} = \sqrt{(\sigma_{b/2})^2 + (\tau)^2}$
$\tau = P/A$
$= 2060.1 / ((\pi/4) \times d^2)$
$= 2060.1 / ((\pi/4) \times 11^2)$
$\tau = 21.67$ N/mm^2
$\tau_{max} = \sqrt{(230.198/2)^2 + (21.67)^2}$
$\tau_{max} = 117.12$ N/mm^2
$\tau_{max(Std)} = 0.5(Syt)/F.S.$
Syt = 420 N/mm^2
$\tau_{max(Std)} = (0.5 \times 420)/1.5$
$\tau_{max(Std)} = 140$ N/mm^2
$\tau_{max} < \tau_{max(Std)}$

Maximum shear stress is less than allowable Shear Stress so this design is safe of modified pin.

IV. ANSYS ANALYSIS

Static analysis of critical part of pins i.e. static analysis of pin is done by using FEA. Analysis is done for the material En19 in order to check Shear stresses and its corresponding deformations induced in Pins and Clamping jaws.

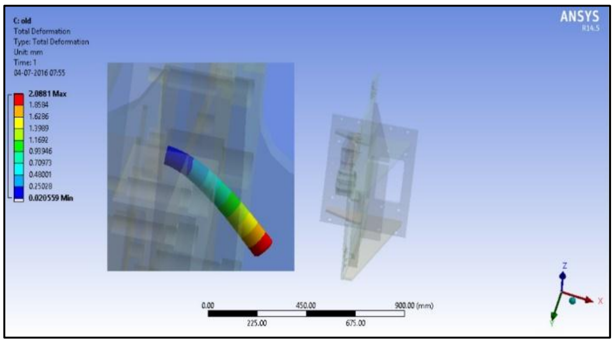


Fig. 3: Total Deformation of old Pin (L.H.Side), mm

Old pin static analysis (failure of pin due to excessive deformations-2.0681mm)

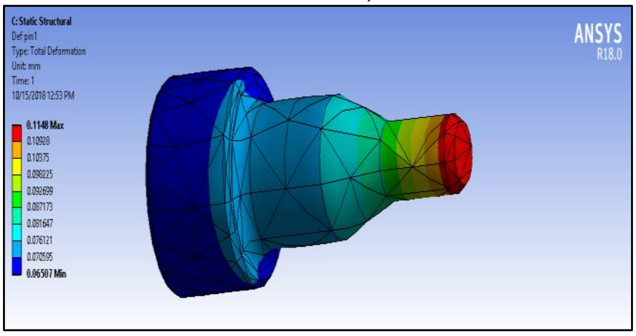


Fig. 4: Total Deformation of Pin-1(L.H.Side), mm

This pin is modified after failure of existing pin so we designed the new pin as per above and took the static analysis and to find the actual deformation of pin during the lift of component is 0.1148mm.

Design of pins is safe and analysis of bending stress of all pins as shown below.

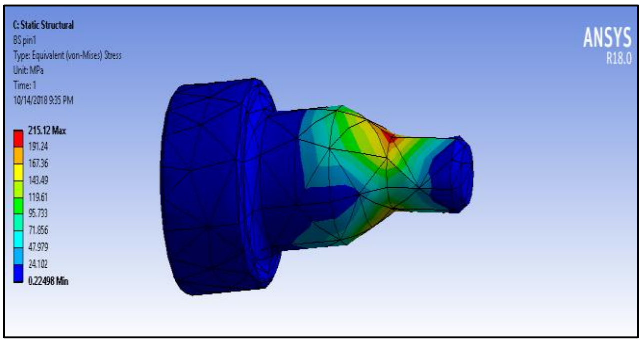


Fig. 5: Bending Stress of Pin-1(L.H.Side), mm

Bending stress of this pin is shown in above image. We get bending stress by ansys. This pin is located in left hand side clamping jaws. Bending stress is 16.529 N/mm^2

A. Analysis of Shear Stress of Pins as Shown Bellow

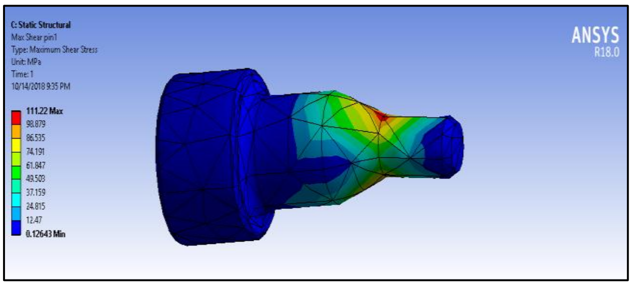


Fig. 6: Shear Stress of Pin 1 (L.H.Side), mm

Maximum shear stress is 111.21 N/mm^2 by using ansys etc. red color indicates the highly maximum stress point as shown in figure.

V. CONCLUSION

Two Parallel Clamping Jaws with pins should easily hold component and carry weight of component i.e. 700kg. Pins to be designed using material optimization, considering further factor of safety.

VI. RESULT

	Analytical Deformation (Ys) , mm	Deformation by Ansys mm	Bending Stress (σ_b) N/mm^2	Bending Stress by Ansys	Max Shear Stress (τ_{max}) N/mm^2	Max Shear Stress by Ansys
Pin - 1	0.0141	0.114	230	215.12	117.12	111.22
Pin – 2	0.0016	0.240	17.21	16.52	9.50	9.17
Pin – 3	0.000584	0.010128	2.13	1.96	1.066	1.1033

VII. FUTURE SCOPE

1) SPM can be used for visual inspection with minor modifications.
2) This type of pins to be used in encapsulated gripper with modifications.

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