IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)
e-ISSN: 2278-1684,p-ISSN: 2320-334X
PP. 78-
www.iosrjournals.org

Analysis for Failure in Welding in Welding of Pivot Pin in WAP 4 Locomotive

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Abstract: *The project "Analysis for failure in welding of pivot pin in WAP 4 locomotive" contains weld failure of one part in locomotive that carries whole load of locomotive body. Analytical result and FEA result of static loading condition gives result of failure. Loading condition causes the failure of pivot pin weld. So, modification is essential to eliminate this failure. FEA analysis of weld gives stress values for existing pivot pin which can eliminate by proposed design of pin.*
Keywords: *FEA Analysis, Pivot pin, WAP 4.*

I. Introduction

WAP 4 is widely used passenger electric locomotive used in India. WAP means Wide Gauge Alternating current traction Passenger locomotive. It is a capable of hauling 24 – 26 coaches at a speed of 140 km/hr. Indian railway is the busiest rail network in the world. The railway routes cover a total length of 63140 km (39,233 miles). Electricals are traditional DC loco type tap changers, driving 6 traction motors arranged in Co-Co fashion. This locomotive has proved to be highly successful, with over 750 units in service.

The project on "Analysis for Failure in Welding of Pivot Pin in WAP 4 Locomotive". Total Pivot pin is an element which carries whole load of super structure negotiates the turning movement of locomotive and attachment of bogies with superstructure. During the progression (acceleration) of locomotive, running, cross over and rail joints pivot pin subjected to various loads so it should be enough to sustain this load. This part actually attached to superstructure by welding but due to above reason weld failure takes place. It can cause separation of pin from superstructure and further major accident on railway. Metal inert gas (MIG) is an arc welding process that are used for joining pivot pin and mounting plate.

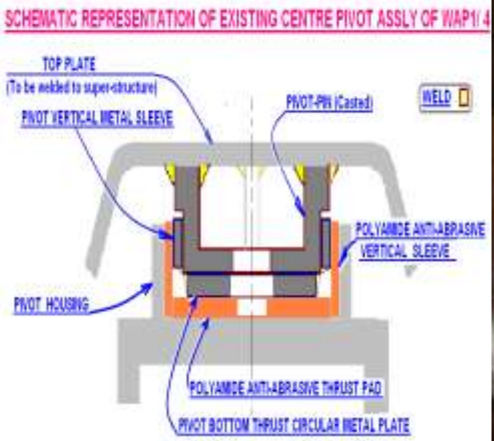


Fig. 1 schematic of pivot pin welding   Fig. 2pivot pin welding with mounting plate

II. Analytical Stress Analysis Of Spring

1. Eccentrically Loaded Welded Joint

An eccentric load may be imposed on welded joints in many ways. The stresses induced on the joint be of different nature of the same nature. The induced stresses are combined depending upon the nature of stresses. When the shear and bending stresses are simultaneously present in a joint, then maximum stresses are as follows.

Maximum normal stress,

$$\sigma_{t(max)} = \frac{\sigma b}{2} + \frac{1}{2}\sqrt{(\sigma b)^2 + 4\tau^2}$$

Maximum shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma b)^2 + 4\tau^2}$$

Where,

σ_b = Bending stress

τ = Shear stress

Consider a T-joint fixed at one end and subjected to an eccentric load P at a distance e as shown in fig. 3

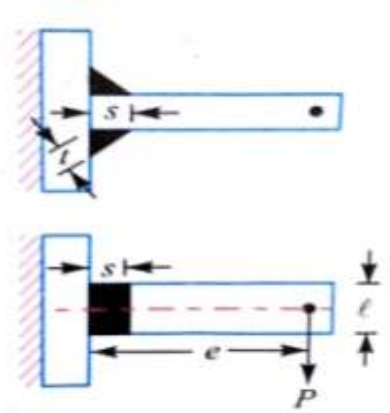


Fig. 3 Eccentrically Loaded Welded Joints

Let,

s = size of weld

l = length of weld

t = throat thickness

The joint will be subjected to the following two types of stresses

1. Direct shear stress due to the shear force P acting at the welds.
2. Bending stress due to the bending moment P*e.

We know that, area at the throat A

A = throat thickness x length

=t*l*2 …….(for double fillet weld)

=2*0.707 s * l

=1.414 s* l …….t=s cos45=0.707s

So,

Shear stress in the weld (assuming uniformly distributed),

$$\tau = \frac{P}{A} = \frac{P}{1.414 * s * l}$$

Section modulus of the weld metal through throat,

$Z = t*l^2/6*2$ ……..(for both sides weld)

$= 0.707*s*l^2/6*2$

$=s*l^{2*}/4.242$

Bending moment,

M = P*e

So,

Bending stress,

$$\sigma_b = \frac{M}{Z} = \frac{P*e*4.242}{s*l2} = \frac{4.242*P*e}{s*l2}$$

We know that,
Maximum normal stress,

$$\sigma_{t(max)} = \frac{\sigma b}{2} + \frac{1}{2}\sqrt{(\sigma b)^2 + 4\tau^2}$$

Maximum shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma b)^2 + 4\tau^2}$$

2. Eccentric Loading on Pivot Pin Weld

Outer diameter of pin = D = 460mm
Inner diameter of pin = d = 377mm

Stresses present

1. Maximum shear stress
2. Maximum normal stress

Limit of shear stress for weld is given = 70 N/mm^2
D = 460 mm
Thickness of plate = 10 mm

The joint subjected to direct shear stress and bending stress
We know that,
The throat area for circular fillet weld t,

t = 0.707*h*2 = 0.707*10*2 = 14.14mm

Now,

Fillet weld area = t*πD = 14.14* π*460 = 20345mm^2

Where,

Primary shear stress (direct shear stress)

$$\tau = \frac{P}{A} = \frac{32.4\ T*1000*9.81}{20345} = 15.622 \text{ N/mm}^2$$

Now,

M = P*e = 317844*144 = 45769536 Nmm

For circular cross section (hollow),

Sectional modulus,

$$Z = \pi \text{ t } (D^2 - d^2)/4 = 3.14*14.14*(460^2 - 377^2)/4 = 751123 \text{ mm}^3$$

So,
Bending stress,

$$\sigma_b = \frac{M}{Z} = \frac{45769536}{751123} = 60.94 \text{ N/mm}^2$$

Maximum normal stress,

$$\sigma_{t(max)} = \frac{\sigma b}{2} + \frac{1}{2}\sqrt{(\sigma b)^2 + 4\tau^2}$$

$$\sigma_{t(max)} = \frac{60.94}{2} + \frac{1}{2}\sqrt{(60.94)^2 + 4*(15.622)^2}$$

$$\sigma_{t(max)} = 64.711 \text{ N/mm}^2$$

Maximum shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma b)^2 + 4\tau^2}$$

$$\tau_{max} = \frac{1}{2}\sqrt{(60.94)^2 + 4*(15.622)^2}$$

$$\tau_{max} = 34.241 \text{ N/mm}^2$$

So,
Resultant stresses produced,

$$\tau = \sqrt{(\sigma tmax)^2 + (\tau max)^2}$$

$$\tau = \sqrt{(64.711)^2 + (32.241)^2}$$

$$\underline{\tau = 73.211 \text{ N/mm}^2 > 70 \text{ N/mm}^2}$$

So,
<u>It is unsafe design for weld.</u>

III. Solution

For safe design we increase the pivot pin upper area.

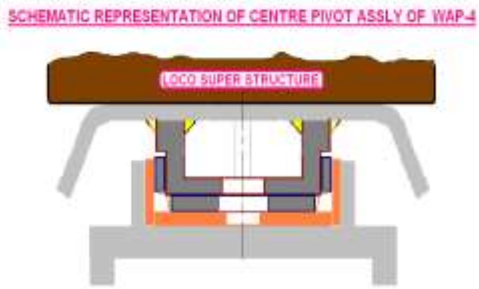


Fig. 4 Schematic representation of center pivot assembly of WAP 4

Take,

D = 560mm

d = 370mm

$A = \pi D t$

$= \pi * D * 0.707 * h * 2$

$= \pi * 560 * 0.707 * 10 * 2$

$= 24876 \text{ mm}^2$

Direct shear stress = P/A = 32.4 T *1000*9.81/24876 = 12.8 N/mm^2

Bending moment = M = P*e = 46405224 Nmm

Sectional modulus,

$Z = \pi t (D^2 - d^2)/4 = 3.14*14.14*(560^2 - 370^2)/4 = 1962347 \text{ mm}^3$

Now bending stress,

$\sigma_b = \frac{M}{Z} = \frac{46405224}{1962347} = 23.65 \text{ N/mm}^2$

Now,

Maximum normal stress,

$\sigma_{t(max)} = \frac{\sigma b}{2} + \frac{1}{2}\sqrt{(\sigma b)^2 + 4\tau^2}$

$\sigma_{t(max)} = \frac{60.94}{2} + \frac{1}{2}\sqrt{(23.65)^2 + 4 * (12.8)^2}$

$\sigma_{t(max)} = 29.25 \text{ N/mm}^2$

Maximum shear stress

$\tau_{max} = \frac{1}{2}\sqrt{(\sigma b)^2 + 4\tau^2}$

$\tau_{max} = \frac{1}{2}\sqrt{(23.65)^2 + 4 * (12.8)^2}$

$\tau_{max} = 17.426 \text{ N/mm}^2$

So,

Resultant stresses produced,

$\tau = \sqrt{(\sigma tmax)^2 + (\tau max)^2}$

$\tau = \sqrt{(29.25)^2 + (17.426)^2}$

$\underline{\tau = 34.047 \text{ N/mm}^2 < 70 \text{ N/mm}^2}$

So,

<u>It is safe design for weld.</u>

Due to the safe design of given solution, it can be applied in Indian railway locomotive.

IV. Finite Element Analysis Of Pivot Pin

The finite element method (FEM), sometimes they referred to as a finite element analysis (FEA), is a computational technique used to obtain approximate solution of boundary value problems in engineering. simply stated, a boundary value problem is a mathematical problem in which one or more dependent variables must be satisfy differential equation everywhere within known domain of independent variable and satisfy specify conditions on the boundary of the domain. Boundary value problems are also sometimes called field problem, the field is the domain of interest and most often represent physical structure.

The procedure as follows

1. Modeling of Pivot Pin
2. Force applied on the shaft
3. Shear stresses on the shaft
4. Equivalent (von-mises) stresses

V. Pivot Pin Without Collar (Existing Pin)

Tractive effort 32.4 T (Considering eccentric loading)

Value for max shear stress = 27.27 N/mm^{2}Value = for Equivalent Stress 49.833 N/mm^2

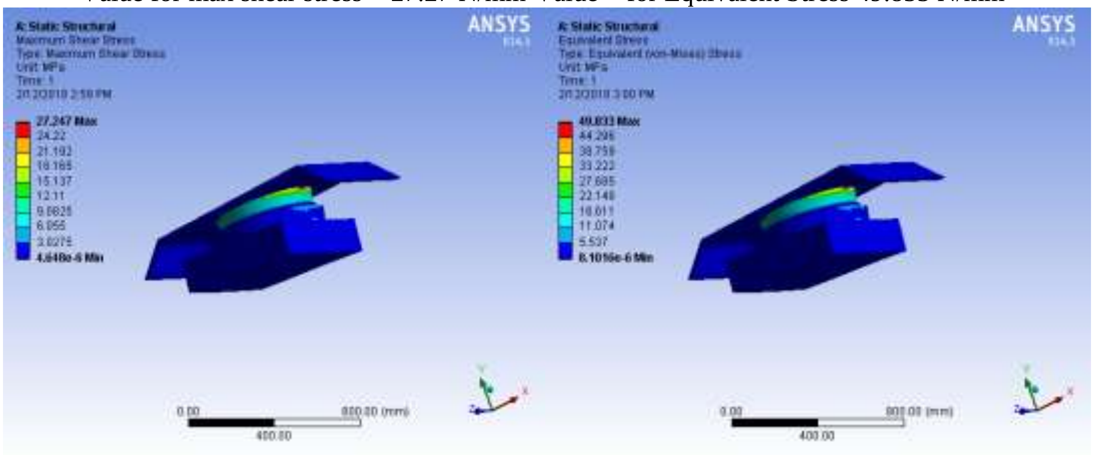
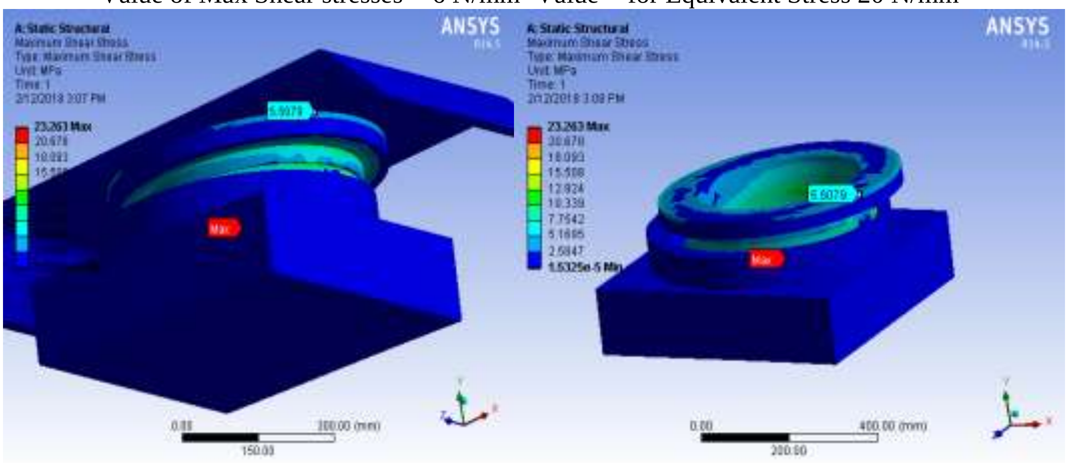


Fig. 5 FEA analysis for existing pin

VI. Pivot Pin With Collar

Value of Max Shear stresses = 6 N/mm^2 Value = for Equivalent Stress 20 N/mm^2



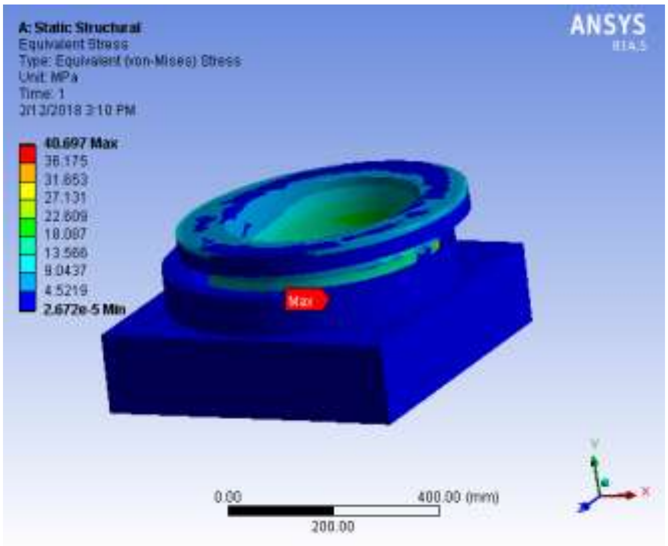


Fig. FEA Analysis for Pivot Pin with collar

From FEA analysis we can say that, FEA values is near about Analytical values and increasing the weld area reduce the stresses developed. Its implementation is necessary to avoid failure.

VII. Results And Discussion

Based on the above study following conclusions are made by analytical and FEA

1. Tractive force is only responsible for crack generation
2. Back thumping force causes cracks to propagate to the mounting plate.
3. Providing the collar to the pivot pin which reduces the stresses.

VIII. Conclusion

The static analysis has been carried out which reveals the shear strength of weldof pivot pin is not enough which responsible for failure weld along with mounting plate.Where, FEAanalysis also gives same stress analysis. Pin weld are failed due to eccentric loading condition which cause the bending and shear stresses in the weld.pin weld are failed only due to design of weld and there is no material lack as the normal stresses are reaches to the limiting condition of shear loads and crosses the limit of shear strength. Final result is that by increasing area of the pin towards mounting plate lowers the value of stresses which reduce weld failure.

IX. References

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