Design and Optimization of Rail Wheel of Shuttle Head Assembly using Finite Element Analysis

[1]Aniket Bindu, P.G.Student, *Dr D Y Patil Institute of Engg. & Technology, Pimpri, Pune 411018*
[2]Dr. Vikrama Singh, Professor, *Dr D Y Patil Institute of Engg. & Technology, Pimpri, Pune 411018*

Abstract—A movable shuttle head is required to transfer material from one conveyor to different discharge points. Rail wheel is a part of movable shuttle head. Design & 3D modeling of rail wheels of two types viz. flanged and unflanged are done and subsequently weight/cost optimization of the wheels is achieved by Finite Element Analysis (FEA) of the assembly of Wheel-Rail interface for contact stresses under heavy loads. FEA is used as a means for modeling contact mechanics, it's assessment, and simulation of the shuttle head Wheel-Rail contact through improving the traditional design. The wheel designs are validated analytically by using Hertzians contact stress theory.

Keywords: Shuttle Head, Rail Wheel, Wheel/Rail Interface, Contact Stresses, Hertzian Theory

I. INTRODUCTION

When it comes to bulk material handling in ports, a wide variety of minerals like iron ore, limestone, dolomite, coal etc. are being handled. These materials are stored at dedicated stockpiles away from the port area. When bulk material is brought to port, it is unloaded from the ship by a ship unloader and routed through various streams of conveyors to reach respective stockpile for storage. To transfer material from one conveyor to different discharge points a movable shuttle head is required. Benefits of using movable shuttle head are: a single conveyor can feed to multiple receiving conveyors, height of transfer tower is reduced, power consumption of feed conveyor and infrastructure cost is reduced. The shuttle head travels on square bar rails, with 4 rail wheels, positions itself on one of the three locations, as per the command by the central control room operator. The linear speed of the proposed shuttle head is 0.2 m/s and the distance travelled is 15 m in forward and reverse directions. The traverse movement is accomplished with the help of rack and pinion and a drive motor. Two wheels on one rail of track have

Dr. Vikrama Singh, Professor,

flanged wheels and two wheels on another rail have no flanges. This will take some mis-alignment in rails. The flanged wheels will constrain across rail movement of shuttle head and the another two wheels without flange will allow little misalignment in track. This combination of wheels will not create any across rail stress on the equipment. This paper presents the design of the rail wheel of the shuttle head assembly and optimization of the wheel by performing Finite Element Analysis, for various load cases. A wheel-rail contact formulation to analyze the contact stresses between wheel and rail is also explained. The contact stresses obtained by using Finite Element Analysis are also validated analytically by using Hertzian's contact stress theory.

II. CHARACTERISTICS REQUIRED FOR RAIL WHEELS

Essential Characteristics of Wheels:

(1) Weight: Wheels are unsprung masses; therefore these should be light in weight to avoid jerks and vibrations.
(2) Fatigue strength: Wheel web should have sufficient fatigue strength to withstand cyclic stress due to weight of the shuttle body.
(3) Rolling contact fatigue strength: tread should have strength to withstand the Hertz stress between the wheel tread and rail.
(4) Change in the Stress caused by thermal attack: Rim expands because of heat input caused by tread braking, thermal stress is developed at the web and rim areas. This excess heat input changes the normal stress distribution into an abnormal situation.
(5) Resistance to thermal crack and fracture: Thermal cracks are generated on a rim due to frictional heat generated by tread brake. In worst case, wheel fracture takes place.
(6) Resistance to Wear: Abrasion or wear is developed on a tread when it is in contact with a

rail. It is developed by the friction between brake shoes and the tread which affects the wheel life. Non uniform wear is a major problem rather than the wear amount .

(7) Operational Performance: Stability on a straight track and curving track performance are to be evaluated. The tread profile is one of those factors.

(8) Acoustics : Wheel running noise is required to be reduced for environmental demand by oil lubrication on rails. For wheels, noise dampening wheels are fitted with sound absorbers.

(9) Wheel Vibration: Is caused by damages on a tread and by imbalance of a wheel. The first one is influenced by the rolling contact fatigue strength and the wear resistance of the tread. The second , depends on the machining accuracies at manufacturer and/or maintenance. This is important for high-speed wheels.

(10) Gripping force for axle: For fixing a wheel onto an axle firmly by interference and press-fitting.

III. LITERATURE REVIEW

As per Nippon Steel & Sumitomo Metal Technical Report [1], A solid wheel as shown in Fig.1 consists of three parts, a hub, wherein an axle is inserted, a rim that contacts the rail, and a web that unites the two parts. Tread is the name for the outer circumferential surface of the rim, in contact with the rail, and the projected part is flange. The web portion should have sufficient mechanical strength to withstand the vehicle weight. It’s configuration should be designed considering thermal stress distribution. For the rim, steel having good anti wear and anti thermal characteristics should be chosen having proper carbon content..

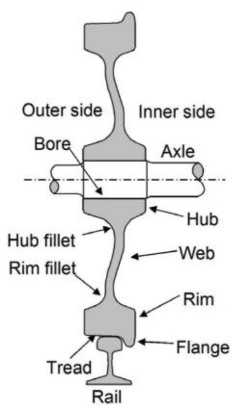

Fig. 1 Designations of each part of a solid wheel

V. C. Sathish Gandhi, et. al.[2] performed contact analysis by considering the material properties of an elastic–plastic frictional contact. It was observed that the maximum stress and strain developed at a point near by the contact edge for material having low E/Y value and move along the center of a straight line in the contact region between wheel and rail as the E/Y value increases.

Joao Pombo, Jorge Ambrósio, Manuel Pereira, et. al.[3] developed a computational tool to predict the evolution of wheel wear in different operating conditions . Nicolae Faur et. al. [5], analyzed the stress-strain condition which occurred in the wheel-railtrack assembly, by using FEA model. The results of FEA model were validated with strain gauges mounted on the track for measuring strains at different speeds. The high level of wear is responsible for vibrations. F. Cocheteux, J. Benabes,T. Palin-Luc, N. Saintier, F. Bumbieler,[6], have analyzed the designs of railway wheels which had holes at the core of their plate for fitting the brake discs assembly, a sound-proofing system or any other element. Currently the failure risk evaluation, related to the use of such wheels, is only possible by the experience feedback. Pramod Murali Mohan [7], has depicted wheel behavior under varying loading conditions. It was observed that heavy braking of wheel leads to thermal overloading resulting in fatigue, crack propagation leading to fracture and wear. Q. Y. Xiong, S. T. Yu, and J. S. Ju [8], demonstrated FEM model of whole wheel system, including wheel, axle, and track, created by using ABAQUS. Stress distribution under self-weight condition was studied when contact effect was taken into consideration. S L Grassie [9], showed that Irregularities on wheels and rails are responsible for noise, ground-borne vibration and general dynamic loading, which increases damage of components of both wheel and track. Brandon J. Van , et. al., [10], in their paper identified several design factors and evaluated their effectiveness based on wheel loads using several existing and new evaluative metrics. NS Vyas & AK Gupta [11], developed FEA model of wheel-flat and rail interaction dynamics. Typical wheel flat loading patterns were generated and response was computed for various cases of rigid and flexible rail supports and damping. Radu Popovici [12], experimentally found out the friction between wheel and rail by measurement and validated the results with FEA model. The contact between wheel and rail was considered to be elliptical, which was shown to be accurate for the described purpose. N.F. Doyle [13], reviewed the design methodology and examined in detail current track design practices. The report contains detailed information relating to the engineering design of rails, sleepers and ballast, presents alternate design formulae and assumptions.

R. Lewis1, R.S. Dwyer-Joyce1, S. Bruni, A. Ekberg, M. Cavalletti, K. Bel Knani [14], developed models to predict the wear and rolling contact fatigue (RCF) of railway wheels. It was assumed in the Wear model that wear is related to T, γ /A, where T is tractive force, γ is slip at the wheel/rail interface and A is contact area. Twin disc testing of rail and wheel materials was carried out to generate wear coefficients for use in the model. The RCF model gives fatigue impact by three fatigue indices, The time evolution of the indices may be employed to assess the likelihood of RCF occurring due to surface fatigue; subsurface fatigue and fatigue initiated at deep defects. Endurance strength design approach for wheels and axles in presented in the Project report[15] on wheelset integrated design and effective maintenance. In the book titled "Modern Railway Track" [16], the theory of railway track and vehicle track interaction has been substantially enhanced and much more attention has been given to dynamics.

IV. FEA SIMULATION OF UNFLANGED TYPE RAIL WHEEL OF SHUTTLE HEAD - PREREQUISITES

Shuttle Head is a terminal part of conveyor bulk material handling system from ship to port and is located at the discharge end of conveyor. Rail wheel is an integral part of the shuttle head.

All necessary engineering computations such as the load acting on rail wheel, allowable /actual pressure between the rail and wheel are computed based on analytical and numerical analysis. The necessary mappings were plotted such that the design can be optimized. Based on the analysis the design was optimized and the original design was upgraded and using optimal values, design drawings were drawn in CATIA. FEA analysis of the rail wheels, for various load conditions was done using ANSYS 16.0 software. wheel diameter was fixed at 600 mm as this was a standard wheel which ThyssenKrupp India Ltd. had used for similar applications. Since it was their standard design there was no need to prepare new pattern for wheel casting, which saved pattern cost. GS 42 Cr Mo 4 High Grade Steel is proposed by the company as wheel material owing to the heavy loading conditions. Square bar rail as per standard **IS 2062** is selected as recommended by sponsors.

Table 1 Input Data For ANSYS (Wheel)

Material For Wheel	GS 42 Cr Mo 4 High Grade Steel	IS 2707 GR3 Steel
Yield Strength	750 Mpa	370 Mpa
Ultimate Tensile Strength	1000 Mpa	700 Mpa
Poissons Ratio of the both material	0.29	0.29
Material Density	7850 Kg/cu.m	7850 Kg/cu.m
Elastic Modulus	210 GPa	210 GPa
Maximum wheel load - (on single wheel under Static Loading)	154 kN	154 kN
Maximum wheel load - (on single wheel under Dynamic Loading)	125 kN	125 kN
Velocity of Shuttle Head for Dynamic Loading Condition	0.203m/s	0.203m/s

Table 2 Input Data For ANSYS (Rail)

Sr. No.	Material For Rail	(IS 2062) -+ Hot rolled steel
1	Yield Strength	250 MPa
2	Ultimate Tensile Strength	410 MPa
3	Poisson's Ratio of the both material	0.29
4	Material Density	7850 Kg/cu.m
5	Elastic Modulus	210 Gpa
6	Rail width	63 mm

V. RESULTS & DISCUSSION

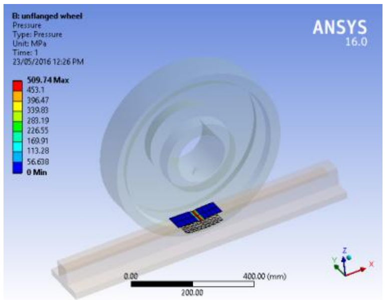

Fig. 2 Contact Pressure GS 42 Cr Mo 4 Solid Wheel

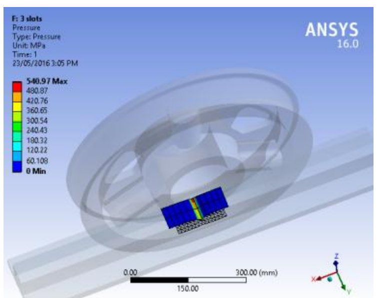

Fig. 3 Contact Pressure GS 42 Cr Mo 4, 3 slotted Wheel

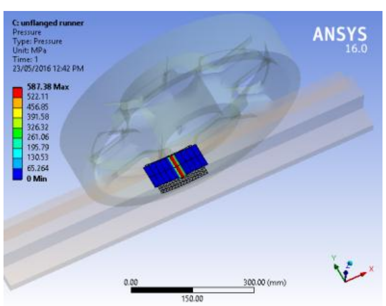

Fig. 4 Contact Pressure IS 2707 GR 3, 6 slotted Wheel

Table 3 Result Table

Sr No.	Wheel Material & Configuration	Contact Pressure (MPa)	Yield Strength of Wheel Material (MPa)
1	GS 42Cr Mo 4 Solid Wheel	509.74	750
2	GS 42Cr Mo 4 Wheel with 3 slots	540.97	750
3	GS 42Cr Mo 4 Wheel with 6 slots	587.38	750
4	IS 2707 GR 3 Wheel with 6 slots	587.38	370

As seen from Result Table 3, contact pressure for solid wheel in contact with rail is 509.74 MPa, which is much less than yield strength of the material 750 MPa. Therefore wheel geometry was modified to reduce the weight of the wheel. A wheel with 3 slots was modeled and analyzed and contact pressure was found to be 540.97 MPa which is much less than 750 MPa. So wheel with 6 slots was modeled and analyzed and contact pressure was found to be 587.38 MPa which is again much less than 750 MPa.

It was therefore decided to propose an alternate cheaper material for wheel. IS 2707 GR 3 steel was selected as a cheaper wheel material and this material was applied to 6 slot wheel ANSYS Model, resulting in contact pressure of 587.38 MPa. However since the Yield Strength of IS 2707 GR 3 steel is 370 MPa only, contact pressure was exceeding the material elasticity limit. Therefore decision was taken to do the surface hardening of the wheel to a case depth of 5 mm assuring 45 to 55 HRC. Equivalent Yield Strength of case hardened IS 2707 GR 3 steel, as found out from the standards IS 4258-1982 (Indian Standard Hardness conversion tables for metallic materials) is 1665 MPa (at 49.8 HRC) ***Therefore this wheel design was safe, lighter and cheaper and hence accepted by the company.***

VI. ECONOMIC ANALYSIS

1) IS 2707 GR3 Steel cost = Rs 65/Kg
2) GS 42 Cr Mo4 Steel cost= Rs 160/Kg
3) Weight of Wheel = 138 Kg
4) One wheel of IS 2707 GR3 Steel costs = 65*138= Rs. 8970/-
5) One wheel of GS 42 Cr Mo4 Steel costs = 160*138 = Rs. 22080/-
6) Surface hardening cost for the wheel of IS 2707 GR3 Steel = Rs. 6000/wheel
7) Total cost of IS 2707 GR3 Steel Wheel = Rs 8970+ 6000= Rs. 14970/-
8) Cost saved /wheel due to the proposed alternate Wheel Material = (Rs. 22080- Rs. 14970) = Rs. 7110/-
9) ThyssenKrupp India Ltd. is manufacturing and supplying 11 shuttle heads.
10) No. of Rail Wheels required = 11 *4 = 44
11) ***Total Cost saved = Rs. 7110 * 44***

= Rs. 3,12,840/-

VII. VALIDATION OF FEA RESULTS BY HERTZIAN CONTACT STRESSES THEORY

Contact Stresses are developed when two objects with curved surfaces are in contact under a force. The point or line contact between the two objects , under a force, changes to small area of contact, and therefore 3 dimensional stresses are developed which are called are contact stresses. A knowledge of contact stresses is important in calculating strength of bearings, gear and worm drives, ball and cylindrical rollers, and cam mechanisms. Failures due to contact stresses are seen as cracks, pits, or flaking in the surface material.

A contact stress calculator is used to calculate contact pressure and contact stress for spherical and cylindrical contact. Maximum shear stresses are drawn with respect to the depth from contact surface for Object-1 and Object-2.

Table 4 Hertzian Contact Stresses Calculations

INPUT PARAMETERS				
Parameter	Symbol	Object-1	Object-2	Unit
Object shape		Cylinder	Plane	
Poisson's ratio	v_1, v_2	0.29	0.29	
Elastic modulus	E_1, E_2	210	210	GPa
Diameter of object	d_1, d_2	600		mm
Force	F	154000		N
Line contact length	l	63		mm

RESULTS				
Parameter	Symbol	Object-1	Object-2	Unit
Maximum Hertzian contact pressure	p_{max}	545.3		MPa
Max shear stress	τ_{max}	163.7	163.7	MPa
Depth of max shear stress	z	2.244	2.244	mm
Rectangular contact area width	2b	5.708		mm

VIII. CONCLUSION

3D modeling and FEA simulation of the wheel-rail interface was done as a means to optimize the rail wheel design of the proposed shuttle head for an offshore bulk material handling system at Bhambra Port Orissa. The weight of the solid wheel of material GS 42CrMo4, which was proposed by the company, was 182 Kg. The optimization was carried out by designing and analyzing new wheel geometries: wheel with 3 slots and wheel with 6 slots. Based on the outputs of the FEA simulation, an alternate cheaper wheel material IS 2707 GR 3 was proposed but , it required surface hardening of the wheel upto 50 HRC to improve it's yield strength. Also Optimized wheel has reduced weight (138 Kg). This resulted in appreciable cost savings to the company.

IX. SCOPE FOR FUTURE WORK

The shuttle head is designed to have two unflanged type of wheels and two flanged type of wheels so that this combination of wheels will not create any across rail stress on the equipment. Modeling and FEA simulation for flanged type rail wheel can be undertaken similarly with a view to optimize the flanged wheel design.

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