International Journal of Engineering, Science and Mathematics (UGC Approved)
**Vol. 6 Issue 6, October 2017, ISSN: 2320-0294 Impact Factor: 6.765**
**Journal Homepage: http://www.ijesm.co.in, Email: ijesmj@gmail.com**
**Double-Blind Peer Reviewed Refereed Open Access International Journal - Included in the International Serial Directories Indexed & Listed at: Ulrich's Periodicals Directory ©, U.S.A., Open J-Gage as well as in Cabell's Directories of Publishing Opportunities, U.S.A**

# Review on Design Analysis and Modifications for EOT crane by using Design software, Numerical method and Mathematical Model

**Nishadevi N. Jadeja**

Assistant Professor,Mechanical Engineering Department,Government Engineering college
**Bhavnagar,Gujrat.**

Ghanshyam R. Patel, Vimalkumar G. Rathod, Nileshkumar V. Parmar

Mechanical Engineering Department,Government Engineering college Bhavnagar,Gujrat.

Abstact:

In industrial sector Hoisting mechanisms are play very important role. Also fast processing units are available in industries so for better work distribution and consider less time requirement are necessary parameters.Modify design for EOT crane by using different software and also numerical and theoretical methods which are available. So we can improve performance of different components of EOT crane. By using Mathematical model we can improve structural features of the crane and prevent damage of various components of EOT crane. A lot of work has been done in these field of design for EOT crane. Many researchers had been investigated , some of them are described here.

## Review of Literature

1. Xianwen Wu, Bo Chen, Dan Zhang, Jian Li

the general structure and hoisting mechanism of crane were optimized, the main technical parameters of crane were calculated. The key issues in the design were discussed. The references on the future design and calculation of similar cranes. were provided. The paper has some application value. According to application, structural features of the quenching crane, related technical documents, and specific national standards, main technical parameters of the quenching crane were designed and calculated. At present, the crane has been in the efficient operation in the heat treatment plant of an enterprise. This development of 320t quenching crane met the development needs of enterprise production. A guideline and reference for the future design of similar cranes were provided .The paper has some application value.

2. M.N.V Krishnaveni, M.Amareswari Reddy, M RajaRoy

Crane hooks are highly liable components that are typically used for industrial purposes. Failure of a crane hook mainly depends on three major factors i.e. dimension, material. In this paper load carrying capacity is studied by varying the cross sections. The selected sections are Circular, Trapezoidal and T-section. The area remains constant while changing the dimensions for the four different sections. The crane hook is modelled using SOLIDWORKS software. The stress analysis is done using ANSYS 14.0 workbench Educational version. It is found that T cross section yields minimum stresses at the given load of 6ton for constant cross section area among three cross sections. The stress distribution pattern is verified for its correctness on model of crane hook using Winkler-Bach theory for curved beams.Three types of crane hooks are designed according to curved beam concept and the induced stresses were determined by Winkler-Batch theory. Total three models are prepared in CAD software(SOLIDWORKS) and imported into ANSYS. Stresses and deformation values are noted from ANSYS software. Theoretical stresses and ANSYS values are compared and deviation is at minimum of 5%. For 6 Ton loading the stresses induced in Circular, Trapezoidal and T-section of Carbon steel and Grey cast iron are below the allowable stresses. The stresses induced in the T section are less than the other sections for same area of cross section and loading. It concludes that T section of Carbon steel offers good results for optimal crane hook design.

3. Omkar K. Sakurikar1, D.V. Kushare

The main aim of this paper is to study various components of electric overhead crane, types of overhead cranes, difference between single girder and double girder cranes on various parameters and to find the effect of increase in span on crane components. Planned and unplanned requirements may necessitate changes in span. The distance between two gantries may get modified during the shed building of factory or the crane in one bay of factory needs to be shifted from one bay to other which have different span. In such condition we have to modify the existing crane to suit the required parameters. Electric overhead crane having 10T capacity and 20m span is to be modified for 22m span. We will study effect of increase in span for various components like long travel wheel, long travel motor and long travel brake.In this work we have reviewed the work done on overhead cranes, types of overhead cranes and basic components of cranes. We have pointed out the components like wheel, motor, brake which will depend on the span of the crane. Using Indian standards, the components used for 20m span crane are safe for 22m span. The future work is to find the effect of increase in span on structural members like girder, end carriage and the gantry girders.

4. P. V. Joshi1, Dr S.K .Dhagat , M. D. Gnanavel

In this paper, the fatigue life of bridge girder of a working EOT crane is calculated based on Palmgren & Miner's rule for the important decision of replacing or maintaining the girder. The variable loadings coming on the bridge girder at the exact loading points of a working crane and stress spectrums by reservoir counting method are presented. A working crane of class of duty 4 as per BIS "M8" commissioned in 1983 is selected for this. The fatigue life considering probabilistic survival of 97.5% comes out to be 25.68 years. It is
suggested that the bridge girder has lived its fatigue life and needs replacement. The fatigue life of the crane girder under consideration is found to be 26.31 years. Considering probabilistic survival of 97.5%, the fatigue life comes to 25.68 years. It may be interesting to note here, that the subject crane was commissioned in the year of 1983, already it has rendered the service of 25 years, and hence the management has started thinking of replacing the entire crane/girders. EOT crane's bridge girder fatigue life calculation is made available in the form of MS-Excel program based on Palmgren-Miner rule considering the structural construction detail Tables of Euro code 3 and IS: 1024. It is possible to calculate the stress spectrum of the bridge girder for each and every point of loadings along with variable loadings, considering the effects of acceleration and deceleration of trolley movement. Using reservoir cycle counting method, stress spectrum of variable loadings is calculated. The program

developed can be useful to predict preventive maintenance of EOT cranes, and or in taking the larger decision of replacing parts or whole of the bridge girder and also to predict the fatigue life of the girder of a newly commissioned EOT crane.

5. V.V. Arun Sankar1, Deepak Vijayan.P2, Ibrahim Ashraf. MY

Overhead cranes are often subjected to heavy stresses on its structure as they carry heavy loads. Stress on its structure leads to unwanted vibrations that lead to structural damages and thereby reduced life span. In companies where the use of crane is of prime importance stresses acting on it plays a major role in its structural behaviour. One way of reducing its stresses and there by its life span is by optimization and mass reduction. The main aim here is to reduce the structural mass of a real-world double girder overhead crane, through the use of
modern computer modelling and simulation methods and applications. The structural mass reduction are designed and verified by structural static stress simulations.

6. Mr. A. Gopichand1, Ms. R.V.S.Lakshmi2, Mr. B. Maheshkrishna

Crane hooks are one of the important components which are used to transfer materials having heavy loads, mainly in industries. Crane hooks are liable components subjected to failure due to stress in accumulation of heavy loads. The design parameters for crane hook are area of cross section, material and radius of crane hook. In the present work optimization of design parameters is carried out using Taguchi method, total three parameters are considered with mixed levels and L16 orthogonal array is generated .The optimum combination of input parameters for minimum Vonmises stresses are determined. From the Taguchi analysis the optimum combination of input parameters for von misses stress were determined using SN ration plots, The optimum combination is TRIANGULAR cross section ,CAST IRON material and 200mmRADIUS .The first model simulation values are compared with analytical values and the error is 9.26%which is acceptable.

7. Muhammad Abid, Shahbaz Mahmood Khan

Overhead cranes for high capacity and large spans are extensively used in the industry for
material handling. In order to avoid overdesign, this paper presents parametric design optimization of the main girder (box type) of the overhead crane using 3D finite element analysis. As a reference, box girder of 120Ton capacity and 32m long span was considered. Keeping vertical and horizontal stiffeners constant, thickness of top, side and bottom plates, box height and mass was optimized for safe bending stress and deflection for a range of 50-100Ton capacities. This provided a generalized optimization procedure by standardizing different parametric relations. The study concluded a generalization methodology that related all the parameters with the capacity of the box girder. The generalization procedure developed can be used by knowing only the capacity as input and produces all the optimized dimensions as on output. As the capacity of the box girder is reduced the optimized deflection gradually increases i.e. 40.12 mm for 100 ton to 47.30 mm for 50 ton. Therefore, the camber may be accordingly increased as the capacity is reduced. During the analysis, on the other, hand no camber was considered. At least a 22% mass reduction was observed for each optimized girder. During the analysis, the box girders were loaded considering a load impact factor of 1.25, which means that a 100 ton box girder was

designed for a 125 ton load. Thus, if the impact factor is not needed, further mass may be reduced and therefore the results for 80 ton may suffice for 100 ton condition.

8. Leszek Sowa, Zbigniew Saternus, Marcin Kubiak

The mathematical model and numerical simulations of the mechanical phenomena in the gantry crane are presented in this paper. The design of modern mechanical constructions is a complex task that requires use of appropriate tools. Such a modern tool, that gives a broad opportunity to analyze strength parameters of designed structures, is numerical simulation which uses the various numerical methods. Here the problem was solved by the finite element method. The analysis of the stresses and strains of the structure of gantry crane was made using the advanced software with highly efficient possibilities of modeling. Motion of the crane trolley along the beam was also taken into account in the study. The research performed, allows the evaluation of the stress state, pointing out the critical areas and values which are made in order to increase the strength of the structure of the gantry crane. The results of numerical simulation are presented in the form of the fields of Huber-Misses stress for the selected location of the crane trolley. In this paper the numerical strength analysis of the gantry crane beam was made. The mathematical model and numerical simulations of mechanical phenomena in the gantry crane beam are presented. One takes into consideration in the numerical model the motion of the crane trolley along the beam length which cross-section was the I-beam.

9. Saptarshi Mandal a, Karmveer Singh a, R.K. Behera b, S.K. Sahu b, Navneet Raj a, J. Maiti

Human error identification and subsequent prioritization are the foremost tasks involved in HRA. In this study a methodology is developed for performing these tasks with an application to overhead crane operations. The application of the present methodology will help to understand how the risk associated with the human errors propagates through different hierarchy levels. The methodology provides a framework for quantifying the risk of different human errors using the experts' subjective opinions only. The incorporation of fuzzy VIKOR technique enables us develop a ranking mechanism for the failure modes where the individual constituent components are non-commensurable in nature. The developed ranking mechanism helps the decision makers in optimal allocation of safety critical resources, used for risk mitigation purposes.

10. C. R. CHAPLIN

The inevitable attrition of wire rope in service is discussed with an overview of the
consequences in terms of inspection and replacement criteria. Details are presented of specific gradation mechanisms observed in three different applications: a mine hoist rope operating on a drum winder, a mooring rope for an offshore structure and a spin-resistant single-fall offshore crane rope. In each case the mechanisms are analysed and steps outlined to alleviate the problems. It is concluded that generalisation is inappropriate: maintenance, inspection and discard policy must be determined in recognition of the degradation mechanisms that operate in different rope applications.General conclusions as to the causes and remedies of rope failure are inappropriate, inadequate or actually misleading. It is, however, of value to note that, for the majority of rope applications, the harsh duty cycle of working ropes means that degradation will inevitably occur over time.

It is the user's responsibility, as guided and regulated by the appropriate authority, to monitor degradation and remove rope before it becomes a safety risk. As indicated in the few examples described above, which may be compared with lift ropes that last for many years, and oil rig drilling lines, for which operating conditions can be so aggressive that they can be destroyed in a few hours, the spectrum of degradation mechanisms and deterioration rates is such that broad guidance can only be in the form of bland generalisations. Maintenance policy, inspection methods and frequency, and discard criteria must recognise the degradation mechanisms and deterioration rates applicable to the specific class of application.

11.. Pavel Peterka , Jozef Krešak , Stanislav Kropuch , Gabriel Fedorko , Vieroslav Molnar , Marek Vojtko

During a very short operation of a steel rope in a hoist system of a drilling rig it was significantly damaged. The wires of the upper layer of cable strands were damaged. This damage was not caused by fatigue of material; therefore it was necessary to determine the reason of the rope damage. The visual inspection of the damaged spots showed that during the operation there was a progressive release of the wires from the outer layer of the outer rope strands; subsequently released wires were rolled when passing the hoist system and they started to break.
In order to find out the reason of the damage the samples of the damaged rope were collected as well as the samples of the new rope from the reserve. According to the nature of the damage we decided to perform mechanical tests of the new (yet not used) rope and thereafter metallographic tests of wire ropes from the outer layer of the strand. The analysis of the mechanical tests results of the new rope revealed the fact that the rope had been manufactured in other rope grade – different from the class declared by the manufacturer. It was also found that the wires of the top layer reporting damage were made of wires of different rope grades.
Mixing wires having low and high strength in the upper layer of the rope strand caused different straining of wires in the specific (given) layer thus wires having lower strength begun to release from the strands and deform which led to the development of fractures at the weaken places. From the above results of mechanical and metallographic analyses is possible to state that the rope subjected to the tests is of unsatisfactory quality. The rope was manufactured in the1960 MPa rope grade and not in the rope grade 1770 MPa as declared by the manufacturer.
Evaluating the rope under the criteria and according to these in the declaration of conformity of the manufacturer, i.e. for the criteria for the rope grade 1770 MPa, the rope is insufficient because most of the wires were excluded because of the upper strength limit. Because of above mentioned facts and evaluation purposes the rope was upgraded to the rope grade in which it was actually manufactured (the rope grade 1960 MPa). Despite the upgrade the rope did not match the rope grade criteria and 33 wires were displaced. 24 displaces wires, substantial portion of them, is in the upper layer, the diameter is 1.85 mm. The mechanical tests showed that the wires used for manufacturing the rope and placed in the top layer of the strandbelong to two different rope grades – the rope grade 1770 MPa and even the rope grade 2160 MPa.Taking samples of each rope grade of the upper layer of the wire in the course of mechanical tests and metallographic tests and their subjection to metallographic tests such as magnification and observation under the light macro scope, and a light microscope and a scanning microscope confirmed that all observed wires have the same structure of the material.

It means that in the manufacturing process were not used wires having strength affected by the incorrect wire manufacturing process and even the wires were prevented from the additional thermal damage that might have caused possible subsequent change of the metal structure.
In conclusion, based on carried mechanical tests, it can be stated that in the upper layer of the rope were during manufacturing in woven wires of two different rope grades and it accelerated the damage to the rope in operation. Deformations and breakages of the wires were caused by the release of the wires of the lower rope grade of the rope strands as well as by strong dint and abrasion when released wires are passing the hoist system. Thus weakened wires then ruptured.

**Conclusions:**

- As a result of the computations carried out, the stresses and displacements of the structure of gantry crane were obtained.
- The influence of changing the loading force position on generate the equivalent stress in the crane beam was evaluated.
- After a thorough analysis of results, several trolley position were selected which we considered significant both for stresses because they generate critical areas within the strength structure of the gantry crane.
- Generally, the area of increased stresses is related to the current position of the trolley crane and moves with it. These stresses reach a maximum value in the middle of the beam length. In addition, a peak of stresses which is the main area of critical crane beam is located at the connection between the end of the crane beam and vertical support frame.
- However the Huber-Misses equivalent stresses in the all crane beam about I-beam cross section was less than the strength of beam material .
- Thus, the end of beam needs special attention in case we want to redesign the crane beam structure in order to improve the product and to eliminate any possible tension peak. It should be noted that results of numerical simulations are compatible with the results found in literature of the subject.
- Similarly, it was recommended reducing stresses near the end of the crane beam by improving structure gantry crane. Finite element analysis, not only help modifying the areas which are critical in terms of the durability of the structure, but also help the analyzer optimizing the over designed

**Referencese:**

1. Xianwen Wu, Bo Chen, Dan Zhang, Jian Li

2. M.N.V Krishnaveni, M.Amareswari Reddy, M RajaRoy

3. Omkar K. Sakurikar1, D.V. Kushare

4. P. V. Joshi1, Dr S.K .Dhagat , M. D. Gnanavel

5. V.V. Arun Sankar1, Deepak Vijayan.P2, Ibrahim Ashraf. MY

6. Mr. A. Gopichand1, Ms. R.V.S.Lakshmi2, Mr. B. Maheshkrishna

7. Muhammad Abid, Shahbaz Mahmood Khan

8. Leszek Sowa, Zbigniew Saternus, Marcin Kubiak

9. Saptarshi Mandal a, Karmveer Singh a, R.K. Behera b, S.K. Sahu b, Navneet Raj a, J. Maiti
10. C. R. CHAPLIN

11.. Pavel Peterka , Jozef Krešak , Stanislav Kropuch , Gabriel Fedorko , Vieroslav Molnar , Marek Vojtko