Design and Development of Buffer by Analyzing Collision Force to Resist Jerk Impact on Crane Operator in EOT Crane With Cabin

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***Abstract*— The design of crane buffers for the Electric Overhead Travelling Crane with cabin were studied during this investigation. This leads to analyze the buffer system as a decoupled system for ease of operator working in the crane cabin.**

The maximum end buffer impact force was determined for a chosen level of reliability based on the responses from the sensitivity analysis using the Lagrange Multiplier method. These maximum end buffer impact forces are then compared with the forces prescribed by the codes. Like SABS 0160, SANS 10160 the end buffer impact force obtained from the constraint optimization technique for a aim to achieve level of reliability of β =3.

It is observed from analysis that the buffer collision force for hydraulic buffers is within limit which is suitable for crane with cabin.

Keywords— Buffer, Impact forces, Collision energy, Hydraulic buffers.

I. INTRODUCTION

Electric Overhead Travelling Cranes are predominantly used in manufacturing buildings to handle the material from one position to another. EOT Cranes are used to increase the mobility of the operational process, hence improving production and ultimately reducing the production costs of the manufactured item. EOT Cranes provides very high efficiency so that many industries would not be immobilized. It has very high range of the hoist loads range from insignificant to several hundred tons, sometimes under very demanding conditions, such as in steel manufacturing environments.

The crane buffer system is important integral part of the EOT Crane system. Also for the EOT Crane to transverse smoothly, the crane supporting structure must be adequately designed. The members of the crane and the crane supporting structure must be designed to have sufficient strength and stiffness to prevent failure at the Ultimate Limit State and excessive deflection and vibrations under the Serviceability Limit State.

Designer designs the EOT Crane depends on a unique set of criteria to which it will be exposed during its lifetime. The factors which influence the design of the EOT Crane which is with operator cabin is the buffer system, but are not limited to; the maximum hoist load, the span of the crane, the type of lifting equipment, the height restriction, the speed of the EOT crane and trolley, the location of the crane, the environmental conditions, etc. Figure 1 shows a picture of the EOT Crane. This study is limited to overhead travelling cranes of which are with operator cabin and also both rails are at the same level on top of the crane girder. Portal cranes and semi-portal cranes, as well as under-slung cranes are excluded from this study. The in-depth study of the various types of cranes is outside the scope of this investigation. The reader is referred to Roswell [1] for more information on the subject.

Fig.1 EOT crane with cabin

Industrial Buffers are designed to protect the crane structures from impact forces. Circular and rectangular types and easily fitted. Reduction of transmitted shock loads enables equipment to be designed more economically and the rising stiffness properties enable vehicle suspension characteristics to be optimized.

There are three types of buffer used in industrial crane operations.
-Rubber Buffer
-Polyurethane Buffer
-Hydraulic Buffer

II. LITERATURE REVIEW

Kohlhaas[7], conducted experimental investigations into the end buffer impact response when the crane including payload, the crane supporting structure and end buffers are considered as a coupled system when the crane collides with the end stops. The experimental investigations were conducted on the 5-ton EOHTC at the Department of Civil Engineering at Stellenbosch University. The investigation provided valuable information and insight into the behavior of the crane and the crane supporting structure during impact, especially with the payload attached. He conducted a series of 5 different impact test experiments by including the payload and by varying the position of the payload during the collision. These tests were repeated changing the buffer position slightly. The results obtained revealed some interesting phenomena. Figure 2.3.4.1 shows the end buffer impact force vs. time response when the crane collides onto the end stops when the condition of payload not considered. Interestingly, the 1st peak is followed by 2 consecutive peaks which occur approximately 1.0s and 1.8s after the 1st peak.

The additional peaks are due to the stepdown torque in the drive motors for the longitudinal motion of the crane after the operator releases the acceleration button. The torque present in the drive motors for the longitudinal motion of the cranes after the 1st impact is sufficient to drive the crane back into the end stops for a second and even a third time. The crane manufacturer did not provide information on the magnitude of the residual torque in the drive motors for the longitudinal motion of the crane.

Lobov [16 to 20] conducted various authors have investigated actions induced by cranes onto the supporting structure mainly through theoretical work. Lobov [16 to 20], published several refereed journal articles on the actions induced by cranes on the crane supporting structures, based on analytical methodologies. A selection of his published work is reviewed below. Lobov [16] analytically investigated the dynamic effects of an electric overhead traveling crane during its movement. The author proposes formulae to determine the horizontal lateral dynamic load when the crab transverses on the crane bridge.

Lobov [17] analytically investigated the additional loads applied to the crane rails as a result of the transverse and rotatory motion of the crane bridge and end carriages. The author proposes formulae for the calculation of the above loads. Lobov [18] analytically investigated how skewing of the crane occurs and the resulting horizontal lateral forces imposed by the wheels onto the crane rails. The author proposes formulae for determining the horizontal lateral wheel forces under various conditions. Lobov [19] analytically investigated whether a crane can travel in a straight line with a constant skew angle of the crane. The author proposes formulae for determining the horizontal lateral wheel forces under various conditions

Karmakar [20] et al, investigated the dynamics of electric overhead travelling cranes using the bond method to simulate the hoisting of the load, braking of the crane as the load is at down position or at ground level with crane travelling on different type of rail connections. The authors conclude that the bond graph method is suitable for simulating crane dynamics due to its efficiency, ease of modifications during the design phase and can include effects such as the motor hoisting the payload.

Grigor'ev [21] et al, investigated the effect which tapered wheels have on the rotational stability of overhead travelling cranes. The authors show that driven tapered wheels assists in the self-alignment of the crane when skewing occurs.

SABS 0160-1989 (As Amended 1989), "South African Standard: Code of Practice For: The General Procedure And Loadings To Be Applied In The Design Of Buildings" Clause 5.7.6 of SABS 0160: 1989, [10], provides two methods with different approaches to estimate the maximum end buffer impact force (horizontal longitudinal force) when the crane collides with the end stops. These two methods are presented and investigated in detail. To obtain the estimated maximum end buffer impact forces clause 5.7.6 of SABS 0160-1989, states; "Take the horizontal force imposed on each end stop by a crane in the direction of travel to be the lesser of the following:
(a) A force equal to the combined weight of the crane bridge (crane) and the crab;
(b) A force calculated on the assumption that the crane strikes the end stop while travelling at its full rated speed, taking into account the resilience of the end stops and crane buffers.

In method (a), the estimated end buffer impact force is determined by taking the product of the mass of the crane with the crab and gravity, i.e. estimated end buffer impact force = (mass of crane with crab) × 9.81m/s2. Method (a) of clause 5.7.6 of SABS 0160-1989, does not explicitly account for the contributions of the following factors at the moment of impact:
(i) The impact speed of the crane
(ii) Mass of the payload
(iii) Vertical position of the payload below the crane bridge
(iv) Horizontal longitudinal position of the hoist load with respect to the crane bridge
(v) Elastic characteristics of the crane buffers
(vi) Damping characteristics of the crane buffers
(vii) Resilience of the crane supporting structure
(viii) Dynamic effects
(ix) Longitudinal misalignment of the end stops or crane at the moment of impact
(x) "Power-off" / "Power-on" (Torque present from the moment of impact)
In method (b), the impact force is a function of the weight of the crane with the crab, the maximum impact speed, the resilience of the end stops and the resilience of the crane buffers. Since the crane supporting structure and the end stops are not expected to significantly displace longitudinally during impact, its resilience is assumed to be zero. Thus, it is assumed that only the crane's buffers is flexible and can deform significantly during impact.

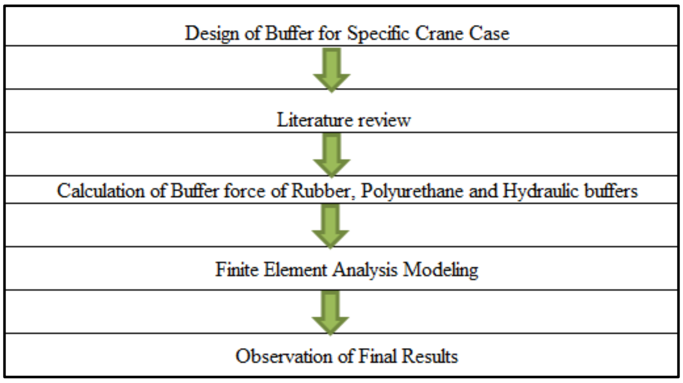
III. PROBLEM DEFINATION

Buffer of the EOT crane when specifying the loads which the crane exerts on the end stops of the crane support structure, the interaction of the various components of the crane like crane cabin, buffer and the crane support structure is ignored, which results in an incorrect assessment of the forces computed in the crane structures. This leads to greater collision forces in the crane cabin members and applies jerk impact on crane operator. To obtain accurate member forces in the crane members under the various loading conditions to resist the jerk impact, it is imperative to study the different types of buffers for specific crane with minimum collision energy while stopping at end stop.

IV. OBJECTIVES

1. To calculate and analyses the collision energy of different types of buffer for specific EOT crane with cabin.
2. FEA of the most suitable buffer for EOT crane with Cabin.

V. METHODOLOGY



VI. CASE STUDY

Considered the case of scrap crane of 25ton with fixed operator cabin located under drive side girder.

Table 1 Crane Specifications:

SPAN	102'-6"
CRANE CAPACITY	25TON
LIFTING HEIGHT	43'-3 5/8"
TRAVELLING SPEED	400ft/min Stepless
WEIGHT OF CRANE	165900lbs
WEIGHT OF BRIDGE	144300lbs
LENGTH OF RUNWAY	400'-0"
CRANE CLASSIFICATION	CMAA Class E
MAIN VOLTAGE	460V -3PH -60Hz
CONTROL VOLTAGE	115V -1PH -60Hz
AMBIENT TEMPERATURE	-10^0F………$+104^0$F
SEISMIC SPECIFICATIONS	PER ASCE 7-2010 SITE CLASS = C SDC = B $S_{S=0.19}$ $S_{I=0.06}$ IMP. FACTOR=1.0
ELECTRIC NORM	NEC/IEC
ALARM	HORN
BRIDGE COLOR	RAL1028
PAINT	KCL SYSTEM 3

25t top running crane equipped with operators cabin under the drive side girder and access to the operator cabin is provided from the drive side platforms through the stairs.

Crane Loads

Successful design of the EOT crane Buffer and associated supporting structure relies on the interactions between the moving crane and the stationary runway. Three principal types of loads (forces) induce a complex pattern of stresses in the upper part of the girder and the structural framing of the building. We will discuss the various loads (forces) below:

Vertical Loads:

Vertical crane loads are termed as wheel loads. The maximum wheel load (MWL) is the sum of:

- The weight of the trolley (carriage) and lifted load, plus,
- The weight of the crane bridge, plus,
- The self-weight of the crane girder and rail.

MWL occurs when the crane is lifting its rated capacity load, and the trolley is positioned at the extreme end of the bridge directly adjacent to the girder. In addition to the shear and bending stresses in the girder cross-section, the wheel loads result in localized stresses under the wheel.

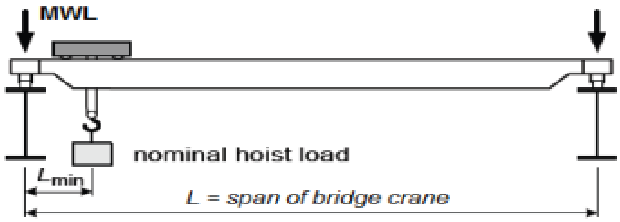


Fig.2 Vertical loads

Lateral Loads (side thrust):

Lateral crane loads are perpendicular to the crane runway and applied at the top of the rails. Lateral loads are caused by:

- Acceleration and deceleration of the trolley and loads
- Non-vertical lifting
- Unbalanced drive mechanisms
- Oblique or skewed travel of the bridge

The magnitude of the lateral load due to trolley movement and no vertical lifting is limited by the coefficient of friction between the end truck wheels and rails.

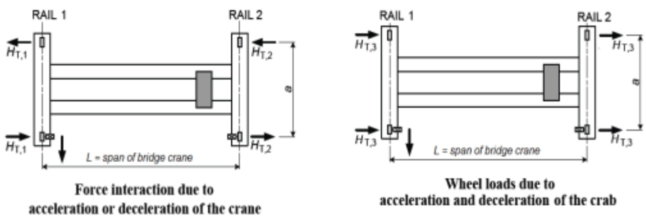


Fig.3 Lateral Loads

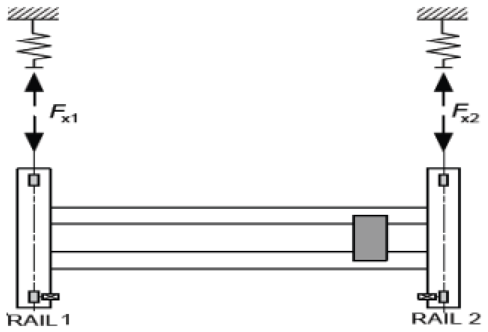
HT1 and HT2 are the horizontal lateral or transverse forces at the wheels, which act as a couple as a result of the force moment. HT1 and HT2 are influenced by the wheel spacing (a) and the dynamic behavior of the crane during acceleration and deceleration. Provided that the payload is free to swing, the horizontal load HT3 represents the horizontal transverse wheel force related to the movement of the crab. The wheel forces can also be in an opposite direction. If the drive mechanism is not balanced, acceleration and deceleration of the bridge crane results in skewing of the bridge relative to the runways. The skewing imparts lateral loads onto the crane girder. Oblique travel refers to the fact that bridge cranes cannot travel in a perfectly straight line down the center of runway. It may be thought of as similar to the motion of an automobile with one inflated tire. The AISC specification and most model building codes set the magnitude of lateral loads at 20% of the sum of the weights of the trolley and the lifted load.

Longitudinal Forces (traction load and bumper impact loads):

Longitudinal crane forces are due to either acceleration or deceleration of the bridge crane or the crane impacting the bumper.

- Tractive forces - are limited by the coefficient of friction of the steel wheel on the rails.
- Impact load - is the longitudinal force exerted on the crane runway by a moving crane striking the end stop. The impact force is a function of the length of the stroke of the bumper and the velocity of the crane upon impact with the crane stop.

The longitudinal forces are normally provided by the crane manufacturer. If this information is not available, the AISE Guide (1996) provides equations that can be used for determining the bumper forces. If the number of driven wheels is unknown, take the tractive force as 10% of the total wheel loads. The figure below indicates the longitudinal impact forces and the relation of these forces to the deformation of the buffers.



Force configuration during buffer impact

Fig.4 Buffer Impact Loads

In an experimental procedure gearbox is allowed to run at its rated power and speed by applying different load conditions on rope brake dynamometer is used. For vibration measurements magnetic base accelerometer is placed on the top just below the location of bearing in axial and radial direction of gearbox.

By making all above arrangements readings are taken for healthy gear and good lubrication condition. This data is stored in FFT analyzer for further analysis.

Vibration spectrums are taken for gears having various faults and the data is stored in computer for further analysis. For different condition of faults & different load conditions data is collected.

VII. RESULTS AND DISCUSSION

Selection of buffer for selected case:

As we seen longitudinal forces are very important while selection the buffers, for current case considering the parameters to select buffer for all three types of buffers

Selection of rubber buffers

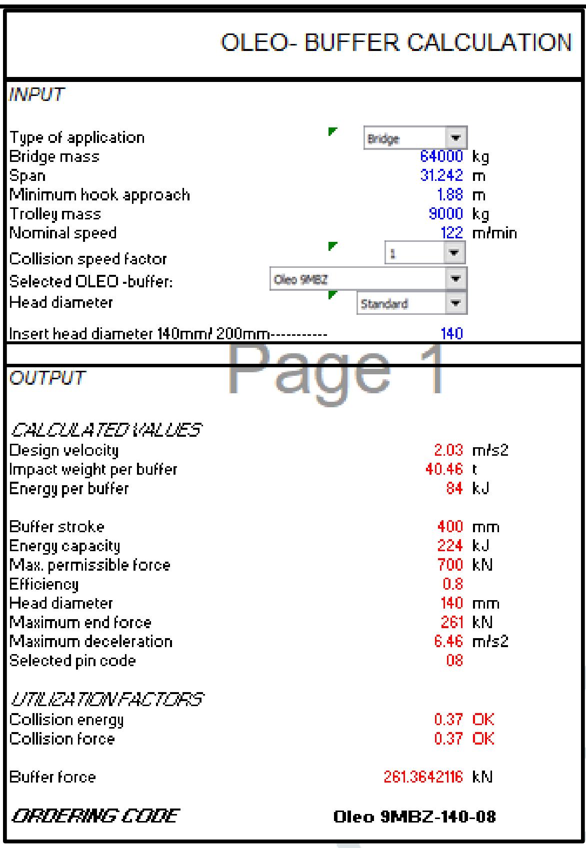
INPUT		
Type of application	Bridge	
Bridge mass	64000	kg
Span	31242	mm
Minimum hook approach	1880	mm
Trolley mass	9000	kg
Nominal speed	122	m/min
Collision speed factor	1	
OUTPUT		
CALCULATED VALUES		
Mass per buffer	40.46	t
Bridge calculated speed	122	m/min
Collision energy	83637	J
SMALLEST ACCECTABLE BUFFER ACCORDING TO COM 12-02		
Type	#N/A	
Diameter	#N/A	
Buffer force	#N/A	kN

Fig.5 Considering Rubber Buffer

Selection of polyurethane (PUR) buffers

INPUT		
Type of application	Bridge	
Bridge mass	64000	kg
Span	31242	mm
Minimum hook approach	1880	mm
Trolley mass	9000	kg
Nominal speed	122	m/min
Collision speed factor	1	
Fastening	Corner holes	
OUTPUT		
CALCULATED VALUES		
Mass per buffer	40.46	t
Bridge calculated speed	122	m/min
Collision energy	83637	J
SMALLEST ACCEPTABLE BUFFER ACCORDING TO COM 12-04 (PROGRAM SELECTS)		
Type	400-400	
Diameter	400	mm
Length	400	mm
Buffer force	946.53	kN
Maximum deceleration	23.4	m/s^2
Deceleration too big for cranes with cabin!!		

Fig.6 Considering PUR Buffer



OLEO- BUFFER CALCULATION

INPUT		
Type of application	Bridge	
Bridge mass	64000	kg
Span	31.242	m
Minimum hook approach	1.88	m
Trolley mass	9000	kg
Nominal speed	122	m/min
Collision speed factor	1	
Selected OLEO -buffer:	Oleo 9MBZ	
Head diameter	Standard	
Insert head diameter 140mm/ 200mm----------	140	
OUTPUT		
CALCULATED VALUES		
Design velocity	2.03	m/s2
Impact weight per buffer	40.46	t
Energy per buffer	84	kJ
Buffer stroke	400	mm
Energy capacity	224	kJ
Max. permissible force	700	kN
Efficiency	0.8	
Head diameter	140	mm
Maximum end force	261	kN
Maximum deceleration	6.46	m/s2
Selected pin code	08	
UTILIZATION FACTORS		
Collision energy	0.37	OK
Collision force	0.37	OK
Buffer force	261.3642116	kN
ORDERING CODE	Oleo 9MBZ-140-08	

Fig.7 Considering Hydraulic Buffer

After calculating for current case above three types of buffers, **the collision energy by considering hydraulic buffer is very low (84KJ)** as compared to rubber or polyurethane buffer.

FEA results of Hydraulic Buffer:

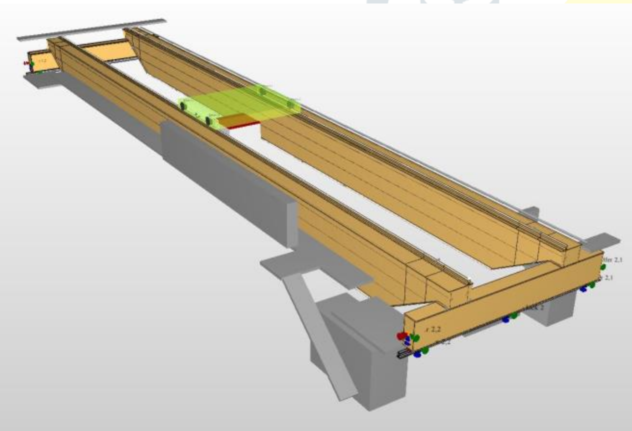


Fig.8 Crane model in FEA

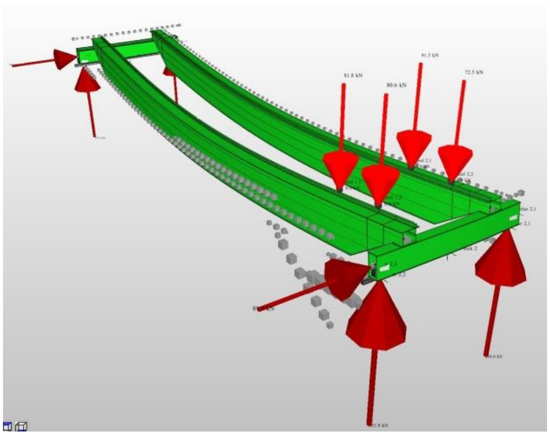


Fig.9 Load case considering trolley at right end

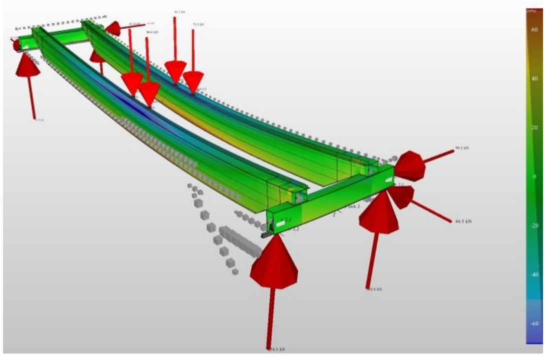


Fig.10 Load case considering trolley at middle

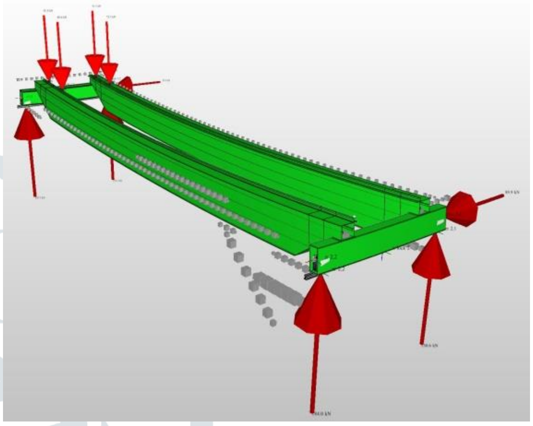


Fig.11 Load case considering trolley at left end

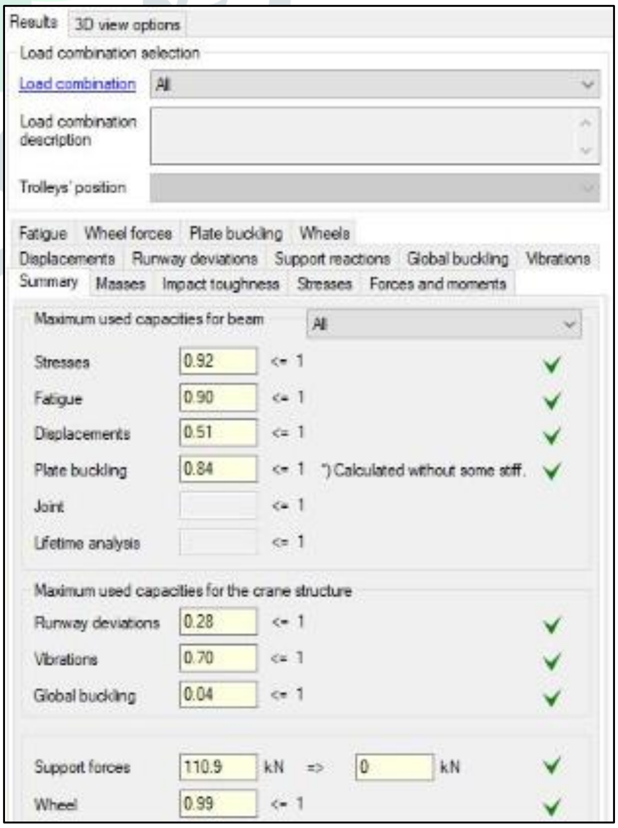


Fig.12 Overall Results of crane

VIII. CONCLUSION

1) To obtain accurate member forces in the crane members under the various loading conditions to resist the jerk impact, it is imperative to study the EOT crane with the fixed operator cabin system.
2) For the Safety of operator in the EOT crane cabin and to resist the jerk impact on crane operator, considering the different buffer type like Rubber, PUR and Hydraulic buffers for numerical calculations and by FEA to find out the best suited buffer for minimal collision energy to resist the impact of collision of buffer with the end stops.
3) It has been seen from the results obtained from the calculation and FEA that the hydraulic buffers are very much suitable for the application of EOT crane with cabin.

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