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Design and Optimization of the Rear Under-.Run Protection Device Using LS-DYNA

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ABSTRACT

Under-running of passenger vehicles is one of the important parameters to be considered during design and development of truck chassis. In India, the legal requirements of a RUPD (Rear Under-Run Protection Device) are fixed in regulation IS 14812-2005 which are derived from ECE R 58, which provides strict requirements in terms of device design and its behavior under loading that the device needs to fulfill for the approval of load carrying vehicles. The work focuses on optimization of RUPD Structure using Finite Element Analysis tool like LS-DYNA and HyperWorks Module and stress calculation for guard pipe has been performed. The regulation allows increasing the load bearing capacity of the RUPD.

***Keywords*–**IS (Indian Standards), ECE R-58(Economic Commission Europe Regulation-58)

I. INTRODUCTION

Many people get injured during underride accidents.Underride occurs when a small passenger vehicle goes beneath the heavy goods vehicle either from the front or rear or side. During such accidents the passenger compartment of the small vehicle strikes the chassis of the heavy vehicle causing severe injuries to passenger in the smaller vehicle. Underride accident are of three different types namely front, rear and side underrun accidents. To avoid such accidentsan underrun device has to be installed on the heavy good vehicle which would prevent the passenger of the small vehicle from getting fatal injuries. In this paper we are going to increase the load bearing capacity of the RUPD (Rear Under-Run Protection Device).

Without the installation of the RUPD the entire energy will be on the pillars of the car structure which in turn would not be able take such impact. Figure 1 shows damage caused to small passenger vehicle during an rear underride accident. The entire vehicle has gone underneath the truck and the entire structure of the car has got crumbled due to the sudden impact load.

Figure 1 Typical Rear Underrun Collision [1]

Table 1.1 shows the death involved in the underrun accidents in the USA till the year 2005. It shows that ninety seven percent (2771 deaths) of passenger vehicle occupants are killed in two-vehicle crashes involving a passenger vehicle and large truck and only 3% (72 deaths)

Occupant Type	Death	%
Passenger Vehicle Occupants	2771	97
Large Vehicle Occupants	72	3
All Occupant Deaths	2843	100

Table 1Accident in 2009 [2]

of large trucks occupants are dying. [2]

In the Figure 2 it is very much clear that in case of crash without the RUPD the impact of the truck is on the passenger compartment due to the underrunning of the car under the truck. The energy absorption is not there before the impact of truck will take place to the passenger compartment so due to this there will be high energy collision and as a result more fatalities will occur. But in the next case that is with RUPD the energy absorption is in the bonnet of the car before the impact will take place to the passenger compartment. Therefore, in this case the fatalities are less. It has been estimated that energy-absorbing front, rear and side under-run protection could reduce deaths in car to lorry impacts by about 12%. An EU requirement was introduced in 2000 based on ECE Regulation 93 requiring mandatory rigid front underrun protection defining a rigid front underrun protection system for trucks with a gross weight over 3.5 tones. Studies performed have shown that passenger cars can survive a frontal truck collision with a relative speed of 75 km/h if the truck is equipped with an energy absorbing underrun protection system. Furthermore, these systems could reduce about 1176 deaths and 23660 seriously injured car occupants in Europe per year. [3]

Figure 2 Rear Impact without RUPD [3]

In the year 2005 ECE Regulation 58 was introduced which made installation of the Rear Under-Run Protection Device compulsory to all trucks weighting more than 3.5 tones.

II. REAR UNDER-RUN PROTECTION DEVICE

The maximum distance between the RUPD and the chassis of the vehicle must be not more than 450 mm (Side View). The RUPD must have maximum ground clearance as 550 mm. It should have good load bearing capacity and must not come out of its fitment position during the time of the impact. The height of the transversal profile of the device should not be smaller than 100 mm. The side edges of this profile should not be curved back and should not have any sharp edges. [4]

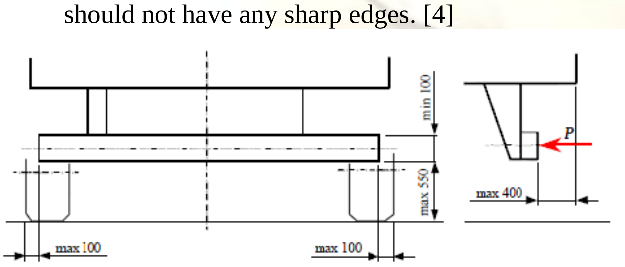


Figure 3 Design and Mountings of RUPD Model

III. RUPD MODEL

The modeling of the Rear Under-Run Protection Device has been done in CATIA V5 R17. The full assembly model of the rear under Guard and its different components are shown in following figures.

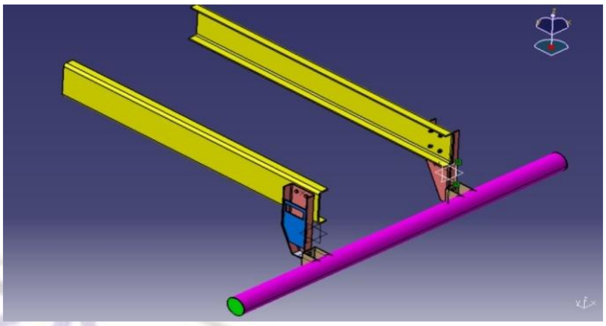


Figure 4 Design and Mountings of RUPD

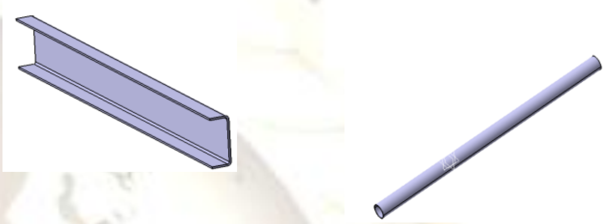


Figure 5 Chassis and Guard Pipe

The figure 5 shows the chassis and the Guard Pipe. The chassis is part on which whole body structure of the vehicle is mountedand the guard pipe comes in contact of the striking vehicle.

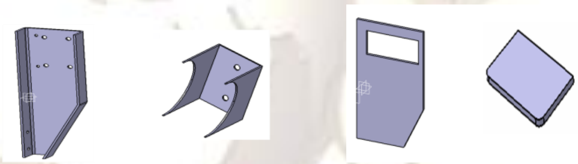


Figure 6 Support Bracket and Stiffener

The support bracket is the main connecting parts between the chassis and the guard pipe .These are the main part which take strength and energy absorption test.

IV. TEST PROCEDURE AND FEA MODEL OF RUPD

4.1 TEST PROCEDURE

The test procedure for Rear Under Run Protection device is mentioned below are derived from the ECE R- 58 and IS- 14812 2005 regulation. The order in which the forces are applied may be specified by the manufacturer. A Quasi Static analysis was conducted on the Rear Guard assembly and its load bearing capacity is tested. A Quasi test is a slow form of the dynamic test and is used when a dynamic code is used to produce static result.

A horizontal force of 100 KN or 50 per cent of the force generated by the maximum mass of the vehicle, whichever islesser, shall be applied consecutively to two pointssituated symmetrically about the center line of the device of the vehicle whichever is applicable at a minimum distance apart of 700 mm and a maximum of 1 m.

A horizontal force of 50 KN or 25 per cent of the force generated by the maximum mass of the vehicle, whichever is lesser, shall be applied consecutively to two points located 300 + 25 mm from the longitudinal planes tangential to the outer edges of the wheels on the rear axle and to a third point located on the line joining these two points, in the median vertical plane of the vehicle.

A horizontal force of 50 KN or 25 per cent of the force generated by the maximum mass of the vehicle for which the device is intended, whichever is lesser, shall be applied consecutively to two points located at the discretion of the manufacturer of the rear underrun protective device and to a third point located on the line joining these two points, in the median vertical plane of the device.

The vehicle mass rating on which RUPD is to be fitted is 12 tones. Therefore the load bearing capacity of the RUPD for each load case described above is given in below table.

Loadcase	P1	P2	P3
Required Load Bearing Capacity	34 KN	68 KN	34 KN

Table 2Load Bearing Capacity

4.2FEA MODEL SET UP

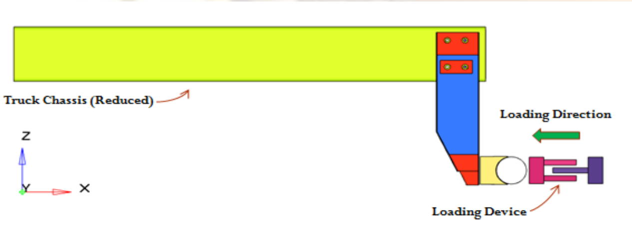


Figure 7 FEA Model of RUPD

The meshing was done in Hypermesh 9.0. The analysis of the Rear Under Run Protection device has beendone in LS-Dyna.

The Plastic Strain are drawn for each component to detect the stresses undergone by each component. The maximum bolt force required for clamping the model to the chassis is also known. The FE model consists of reduced truck model cut at around 2000mm location from the rear end of the chassis. The RUPD is attached to the chassis through bolt connections. The FE model is as shown in figure 7.The Loading Device and the direction of loading is also shown.

The loading device is constructed as per details given in the regulation. The construction of loading device is as shown in figure 8. The loading device consists of two blocks which are connected at center using revolute joint so that the device will be always in normal direction at every time during loading process. The loading device is modeled with LS Dyna Material Type 20 rigid material model.

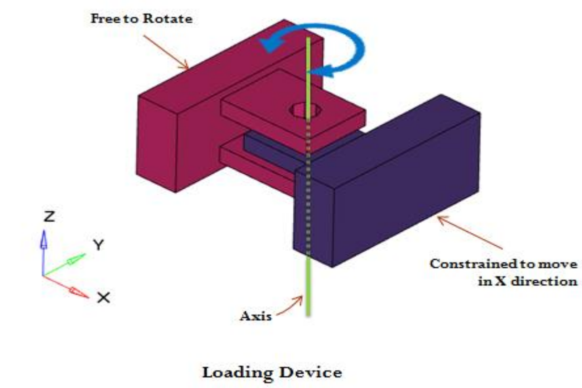


Figure 8 Loading Device Mechanisms

The boundary condition and the load applied are shown in figure 9

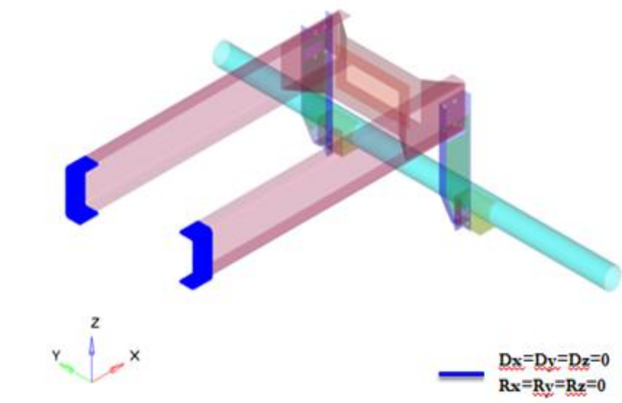


Figure 9 Boundary Conditions

4.3 MATERIAL DETAILS AND ELEMENT CRITRIA

The FE model consists of three materials namely, E38, FE 410 and FE 690. The Material type 24 Piecewise Linear Plasticity Material Model is used.*MAT_PIECEWISE_LINEAR_PLASTICITY (*MAT_024) is widely used material model for metals and in some cases plastics. Its popularity is widespread since it offers several plasticity models and can also be strain-rate dependent. One particle parameter, the Yield Stress, in the material card can appear in more than one place and can be sometimes confusing to know which value is used by LS-DYNA. Here is the hierarchy of the final value of the Yield Stress used in LS-DYNA.

Yield Stress Calculation

1. If LCSS is non-zero, the initial and evolving yield stress is always taken from either the Curve of Table that LCSS refers to.

2. If LCSS is zero AND EPS-ESS is defined, then the initial and evolving yield stress is determined by ESS
3. If LCSS is zero, EPS-ESS is zero, then the yield stress is obtained from SIGY parameter.
Strain-rate Dependency

<u>In *MAT_024, there are three ways to define strain-rate dependency. Its hierarchy is defined below.</u>

1. If LCSS refers to a table, then the strain-rate dependency is always computed from the table.
2. If LCSS is either a Curve or is zero and LCSR is nonzero, then LCSR is used
3. If LCSS is either a Curve or is zero and LCSR is zero and C & P is non-zero, then Cowper Symonds is used.

When using Cowper-Symonds method for strain-rate dependency and Viscoplasticity (VP) is turned on (equal to 1), SIGY, plays an important role in how the dynamic yield stress is determined. When VP=1, the strain-rate dependency is always based on SIGY which is then added to the static stress. However, when VP=0, the dynamic stress is based on the static stress curve which is now a function of the effective plastic strain.

<u>True Stress Vs True Strain</u>

The experimental data from a uniaxial tension test is expressed in terms of true stress vs. true strain, not engineering stress or strain. Be aware that experimental data always includes some degree of error and thus tends to be somewhat noisy or erratic. When using *MAT_24, one should input a smoothed stress-strain curve utilizing a minimal number of points. Input of noisy experimental data may cause spurious behavior, particularly in the case of the default, 3-iteration plane stress plasticity algorithm for shells.

True strain = ln(1 + engineering strain) where ln designates the natural log

True stress = (engineering stress) * exp(true strain) = (engineering stress) * (1 + engineering strain) where exp(true strain) is 2.71 raised to the power of (true strain).

Equation 1 Formulae for True Stress and True Strain

The effective plastic strain values input in defining a stress vs. effective plastic strain curve in a LS-DYNA plasticity model should be the residual true strains after unloading elastically. True stress is input directly for the stress values.

effective plastic strain (input value) = total true strain - true stress/E

Equation 2 Formula for calculation of effective plastic strain

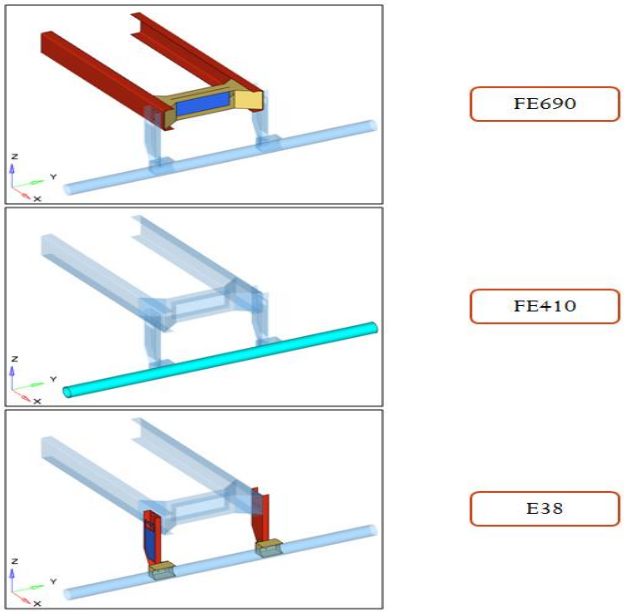


Figure 10 Material Details of FE Model

The table shows the FE Model Quality Criteria which is a baseline for meshing.

Sr. No.	Element Property	Permissible Value	Actual Value
1	Minimum Element Size	**2**	**2.32**
2	Warpage <	**15**	**3.08**
3	Aspect Ratio <	**5**	**2.81**
4	Skew Angle <	**60**	**49.37**
5	Jacobean Ratio >	**0.6**	**0.61**
6	Min Angle (Quad) >	**35**	**38.88**
7	Max Angle (Quad) <	**140**	**137.76**
8	Min Angle (Tria) >	**20**	**33.92**
9	Max Angle (Tria) <	**120**	**102.14**
10	% of Trias / Pentas <	**3%**	**0.3%**

Table 3 FEA Model Quality Criteria

V. BASELINE DESIGN RESULTS (P2 LOAD CASE)

The baseline model is been designed according to the ECE R-58 and AIS 14812-2005 Regulation but it fails to meet the load requirement. All other parts like the vehicle body and engine are not taken into consideration. The reduction in modeling takes less processing time and more accuracy is achieved.The objective is to increase the stiffness of the different elements.

We have taken FE 690 material for the

support bracket. The material is not able to withstand the impact load. The stress value of the material does not meet the regulation

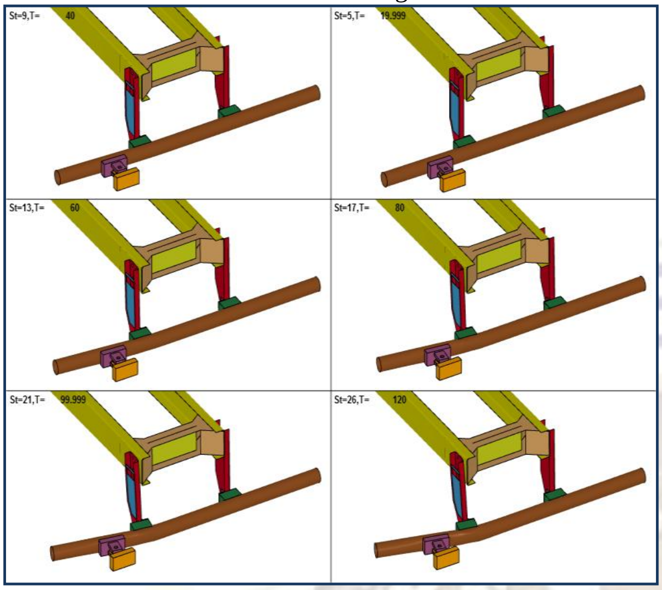


Figure 11 Animation Instances 01

The maximum stress and maximum strain are 0.60293 GPa and 0.653181 at 120 msec.

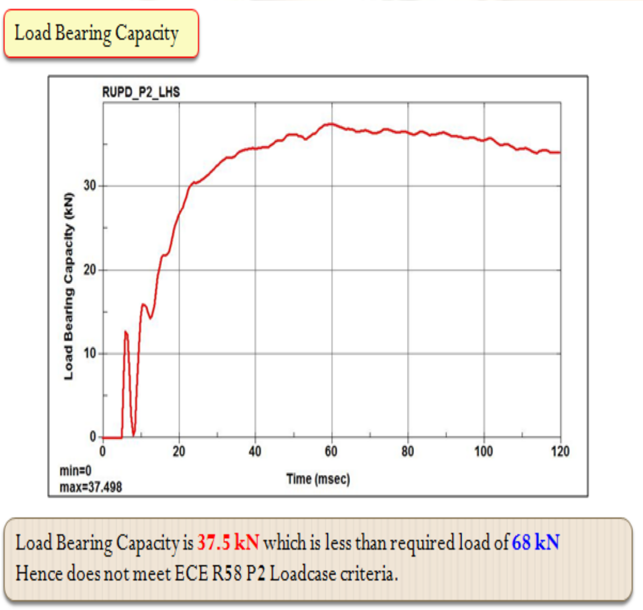


Figure 12 Load Bearing Capacity of RUPD Model for Iteration 01

The load bearing capacity is 37.5KN which is below 68 KN hence we need to go for next iteration. The figure 13 shows bolt force graph for the iteration 01. The maximum axial force is 21.04 KN and the max shear force is 103.56 KN.

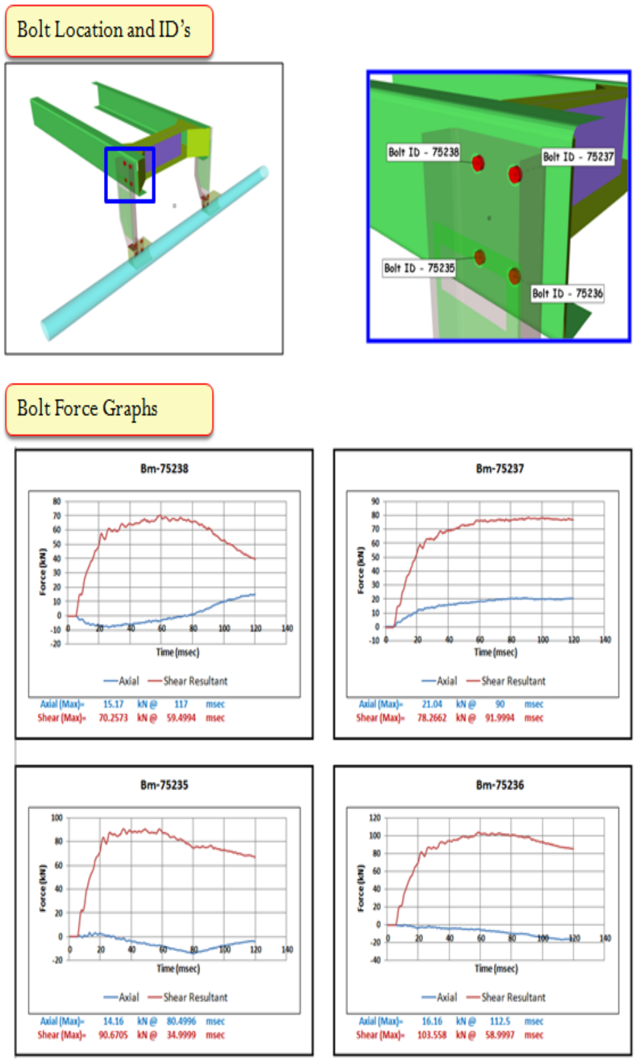


Figure 13 Axial and Bolt Force Graph for Baseline Design

VI. RESULTS FOR DIFFERENT RUPD MODELS

6.1 ITERATION 04 (P2 LOAD CASE)

A design modification is done on the support bracket which is a part of the RUPD Model. The shape of the support bracket has been changed. The material for the support bracket is changed from FE 690 to E 38 to check whether the model meets the design regulation.

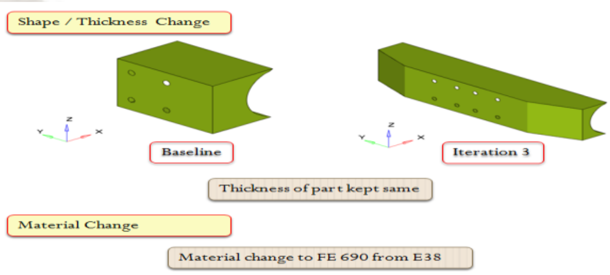


Figure 14 Design Changes for Iteration04

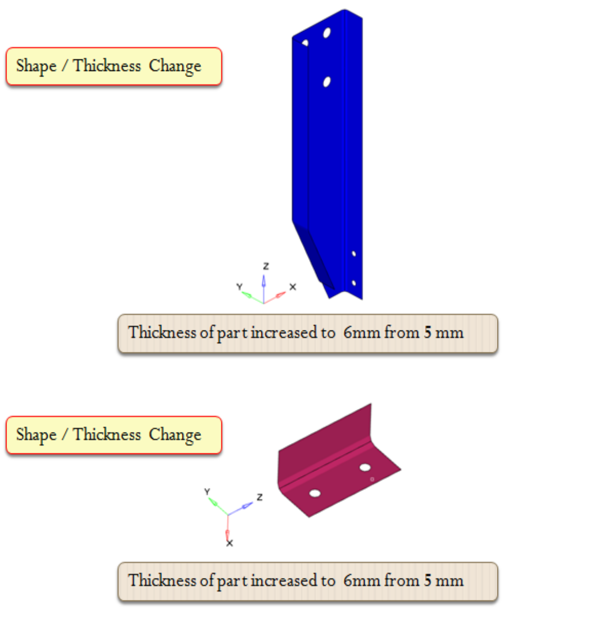


Figure 15 Design Changes for Iteration 04

The thickness of the other support bracket has been changed from 5 mm to 6 mm. The thickness of the stiffener has been increased by 1 mm.

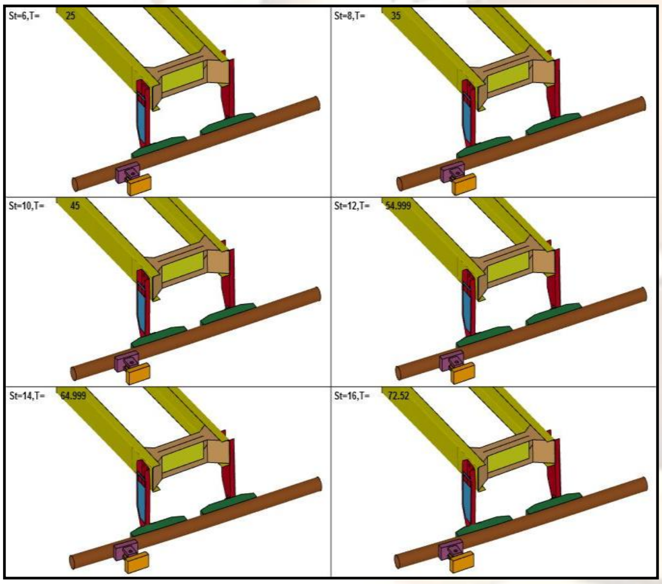


Figure 16 Animation Instance for Iteration 04

The lateral displacement of the RUPD Member is very small as compared to previous iteration.The figure 17 gives idea about Von Misses stress induced in the RUPD Model. For the Correspondence Von Misses stress 0.6911 GPa the plastic strain is 0.302406.

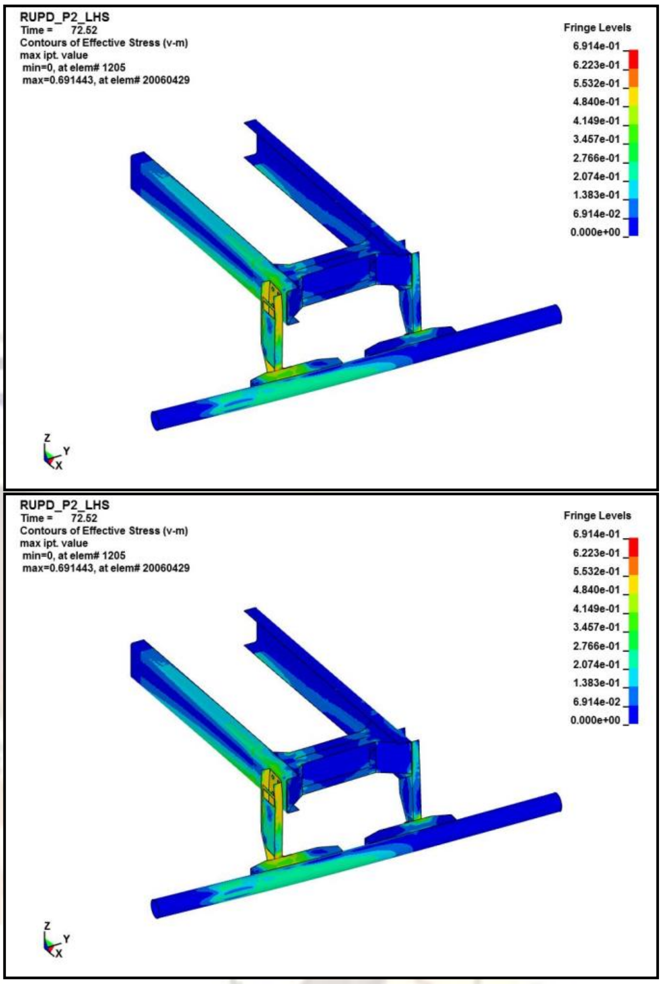


Figure 17 Von Misses Stress and Plastic Strains for Iteration 04

The load curve gradually meets the requirement but shows a certain dip between 50 – 60 msec which is not acceptable by the regulation. The load bearing capacity for the current model is 67.47 KN.It is marginally below the regulation so iteration is needed.

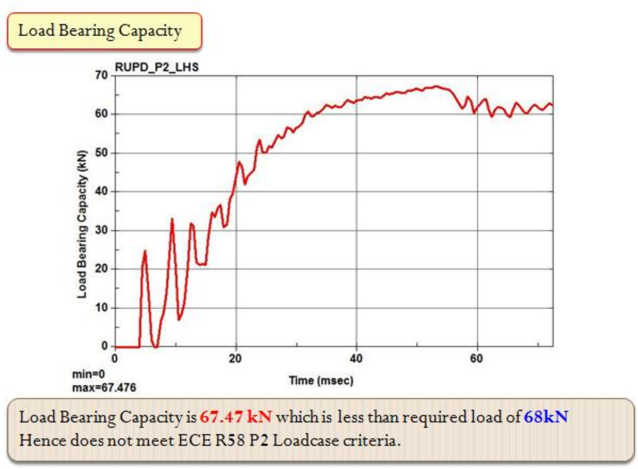


Figure 17 Load Bearing Capacity for Iteration 04 (P2 Load case)

6.2 ITERATION 05 (P2 LOAD CASE)

In Iteration 05 the material of the stiffener was changed from FE 690 to E-38 material.

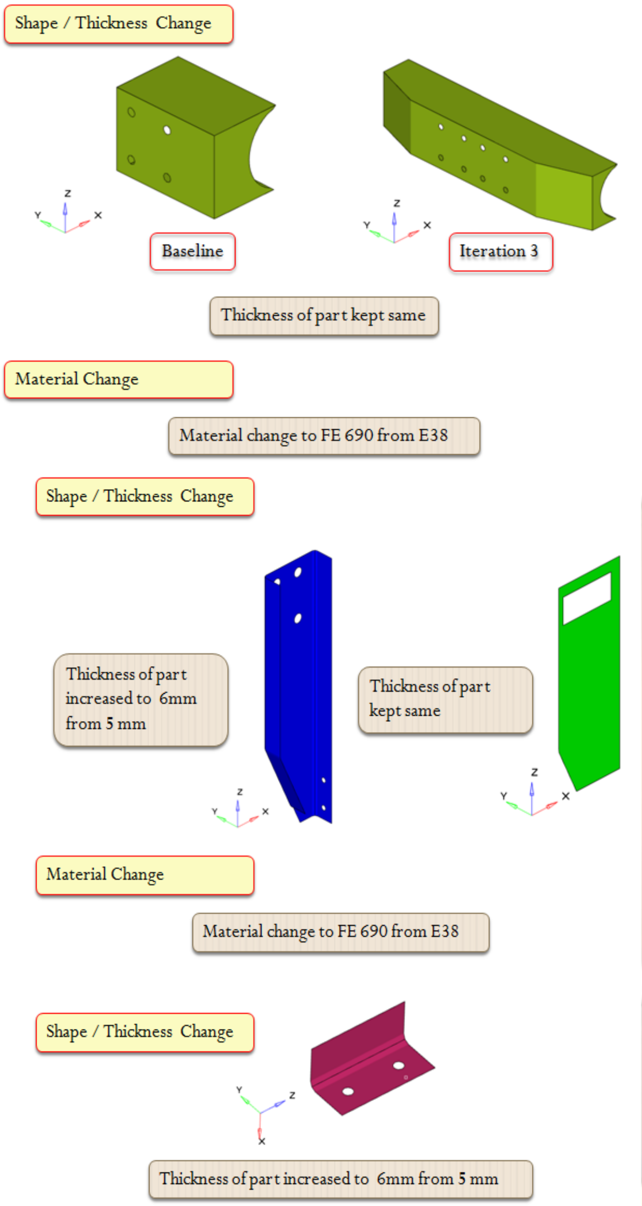


Figure 18 Design changes for Iteration 05

The animation instance shows the lateral displacement of the RUPD. The behavior of the RUPD Model at the different time instance is shown in the figure 19.

The Plastic strain value for the corresponding value of the Von Misses stress is shown in figure 20

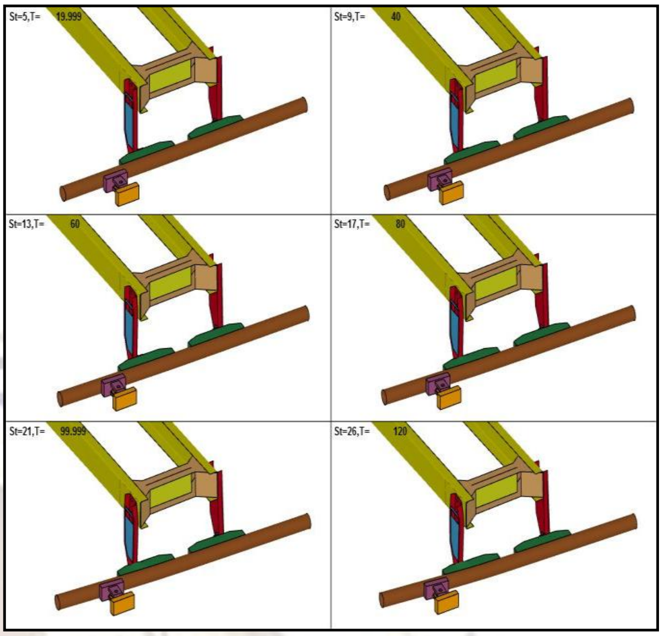


Figure 19 Animation Instance for Iteration 05

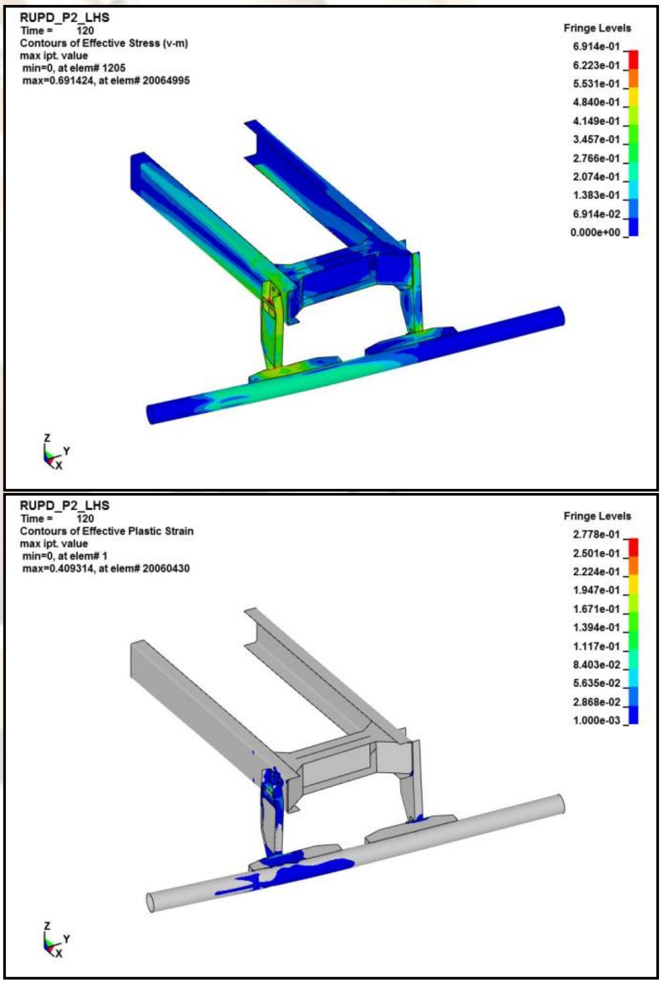


Figure 20 Von Misses and Plastic Strain Plot for Iteration 05

It is observed that the Load Bearing capacity for this design meets the regulatory requirement. But it is observed from the contact force graph that the trend of the load suddenly dips achieving 65.62KN force at 61 msec. which indicates buckling in RUPD. The load curve is not achieving full load gradually hence we need to modify the RUPD design.

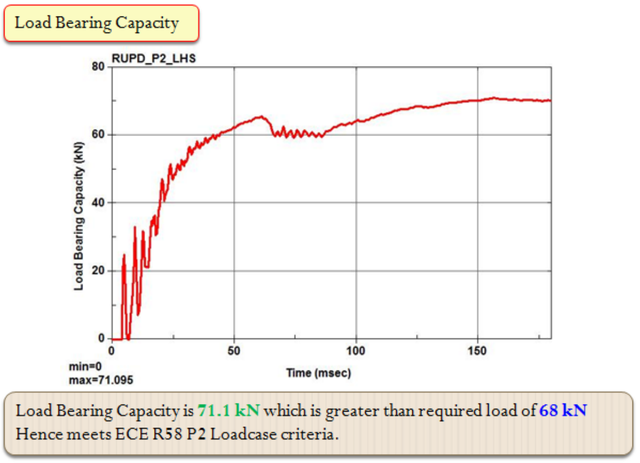


Figure 21 Load Bearing Capacity for Iteration 05 (P2 Load Case)

6.3 ITERATION 06 (P2 LOAD CASE)

The Shape of the stiffener has been changed to check whether this iteration passes the requirement or not.

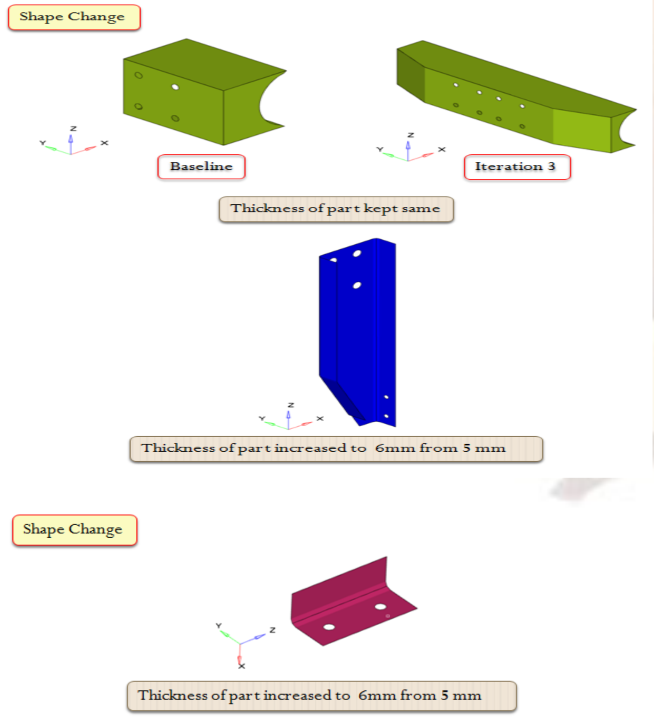


Figure 22Design changes for Iteration 06(P2 Load Case)

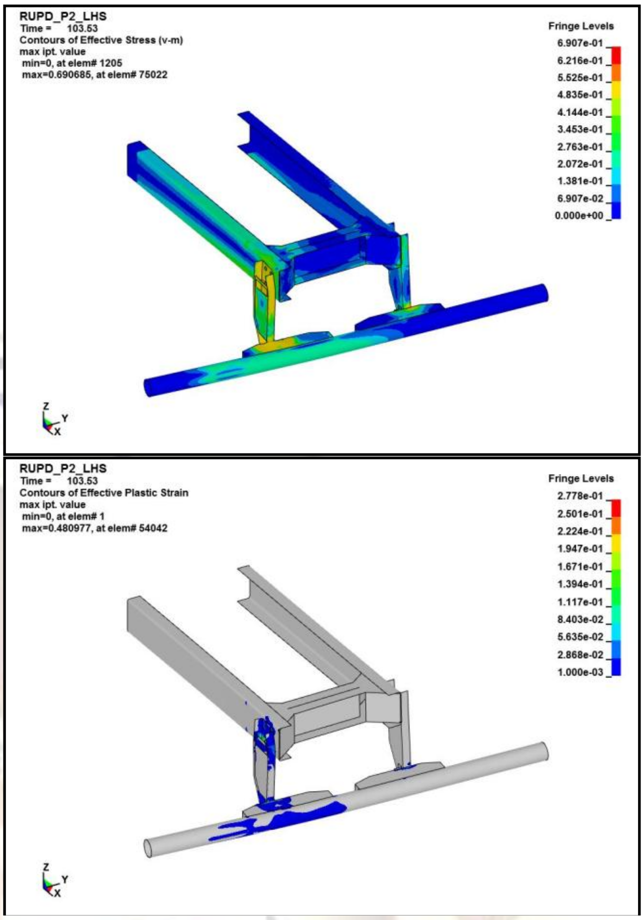


Figure 23 Von Misses and Plastic Strain Plot for Iteration 06 (P2 Load Case)

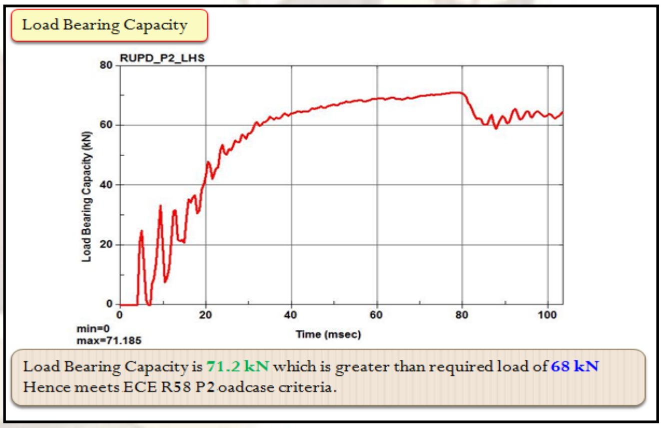


Figure 24 Load Bearing Capacity for Iteration 06 (P2 Load Case)

The maximum value Von Misses stress is 0.6907 GPa and corresponding Value of Strain is 0.4808977. The load curve is achieving full load gradually, hence iteration 06 suggested design changes meets the regulatory requirement of ECE R 58. The Load bearing Capacity is 71.2 KN.Now we have to check the design for P1 and P3 load case as it meets P2 load case requirement.

Testing of the P1 and P3 Load case is continued for the same iteration 06. The next load case is P1. The animation Instance for P1 load case shows minimum deformation in the shape. The maximum value of stress is 0.6286GPa at 111.03 msec and the maximum strain value is 0.4835.The Load bearing capacity is 44 KN which is greater than 34 KN and it meets ECE R-58 Regulation

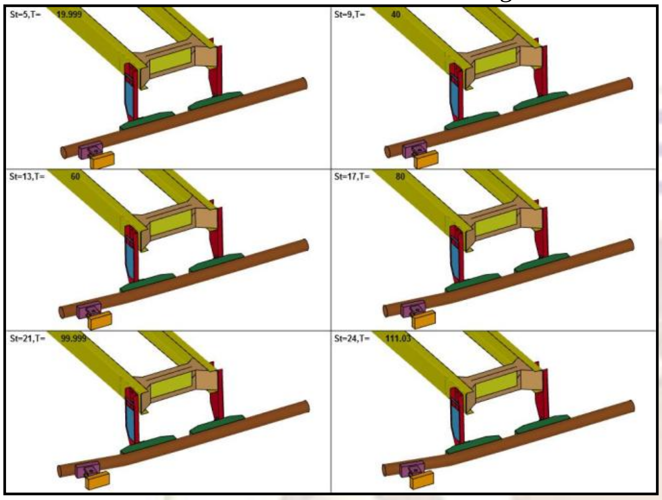


Fig 25Animation Instance for Iteration 06 (P1 Load case)

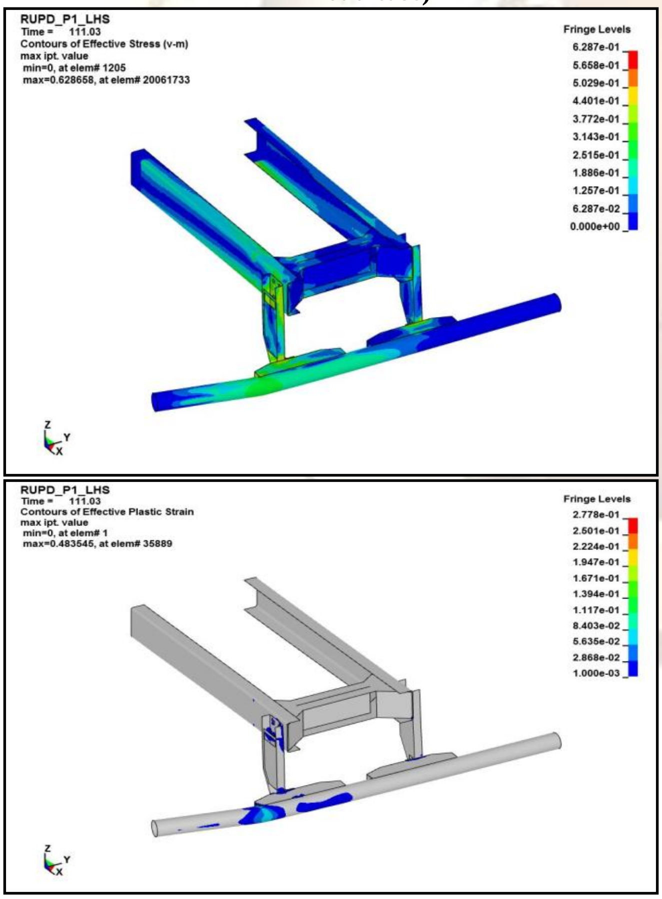


Figure 26 Von Misses and Plastic Strain Plots for P1 Load Case

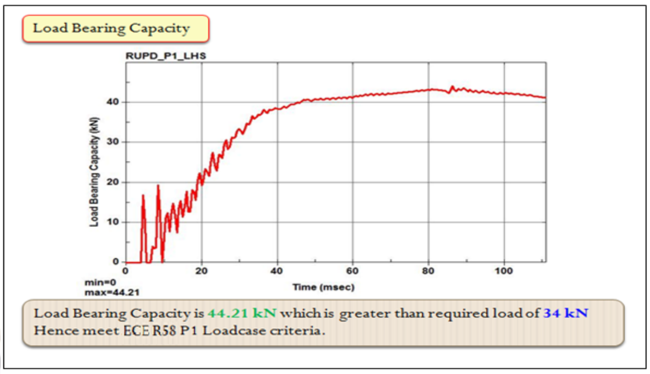


Figure 27 Load Bearing Capacity for Iteration 06 (P1 Load Case)

The animation instance for P3 load case is shown. The impact of the Loading device is exactly on the centre of the RUPD Model. The load bearing capacity is 84.3 KN which shows a near 50% improvement above the ECE R-58 Regulation. The Von Misses stress is 0.688 GPa at 131.09 msec and the maximum effective plastic strain 0.973

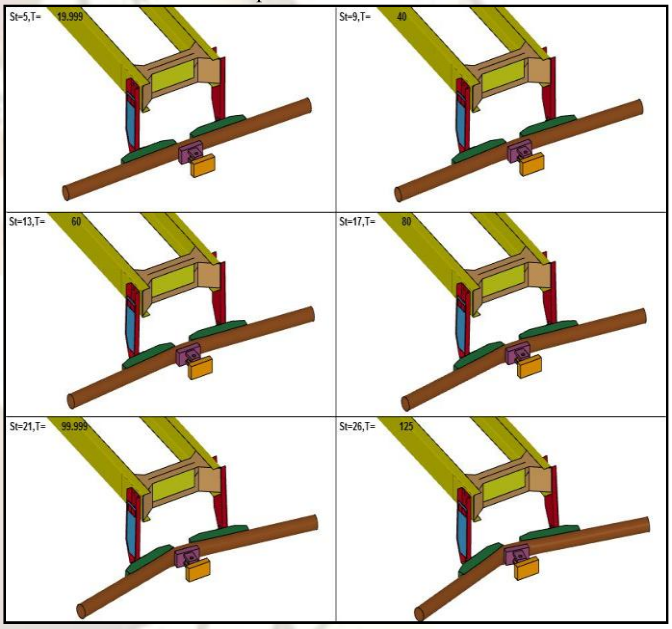
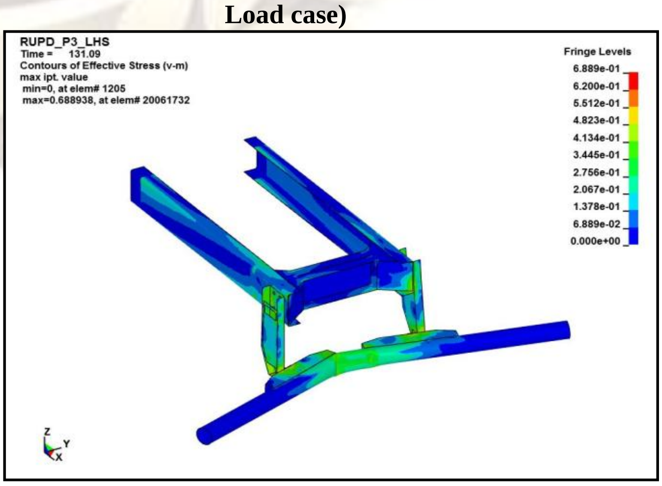


Fig 28Animation Instance for Iteration 06 (P3 Load case)



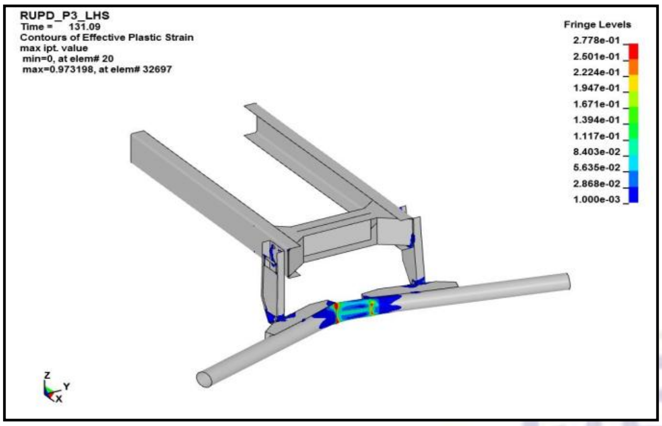


Figure 29 Von Misses and Plastic Strain Plots for(P3Load Case)

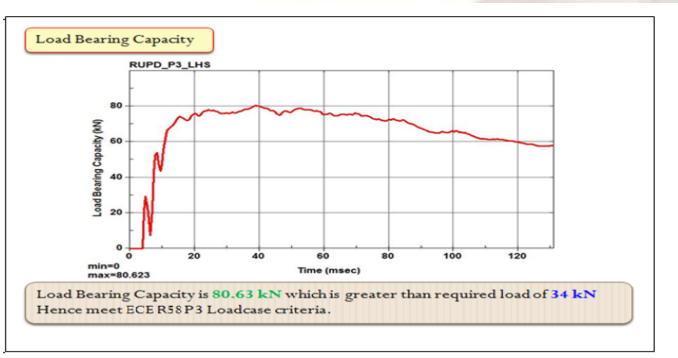


Figure 30 Load Bearing Capacity for Iteration 06 (P3 Load Case)

The maximum shear force and maximum axial force are 165.31 KN and24.91KN respectively. Bolts which withstand such high force must be used for mounting the RUPD Member.

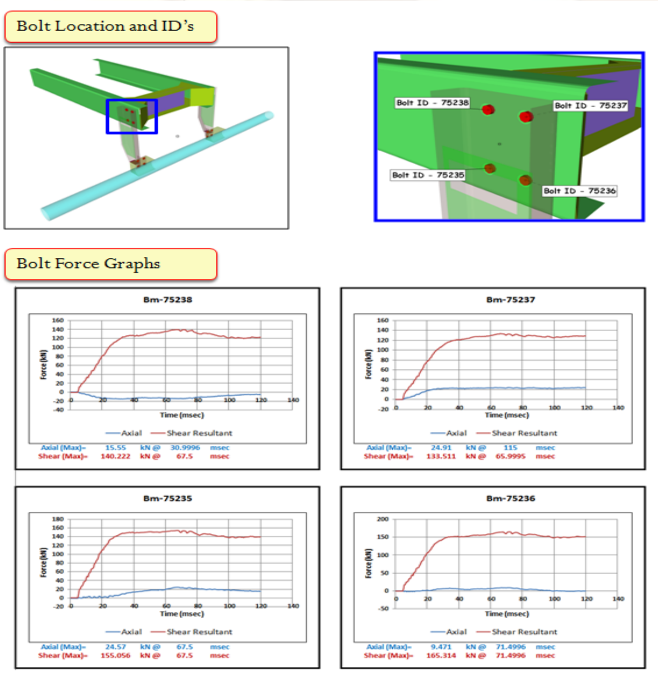


Figure 31 Bolt axial and shear force graph for Iteration 06 (P3 Load Case)

VII. RESULT SUMMARY

Result Summary

	Load Case	Acceptable Limit (with 15% overload)	Baseline	Iteration 1	Iteration 2	Iteration 3	Iteration 4	Iteration 5	Iteration 6
Load Bearing Capacity (kN)	P1	34	N.A.	N.A.	N.A.	N.A.	N.A.	N.A.	44.21
	P2	68	37.5	37.1	47.53	61	67.47	71.1	71.2
	P3	34	N.A.	N.A.	N.A.	N.A.	N.A.	N.A.	80.63

Table 4 Result summary For Baseline and all Iteration

- Baseline Design shows just 37 KN load bearing capacity which is well below the acceptable limit of regulatory requirement and hence does not meet the design criteria.
- Design Modification from Iteration 01 to Iteration 06 shows increasing trend of load bearing capacity.
- It is observed thatthe Load Bearing capacity for Iteration 04does not meet the regulatory requirement. But it is observed from the contact force graph that the trend of the load suddenly dips achieving 65.62kN force at 61 msec. which indicates buckling in RUPD. The load curve is not achieving full load gradually hence we need to modify the RUPD design.
- Strain value at the time of peak load condition are within the allowable range of 12% for iteration 06
- Iteration 06 design changes meets Load Bearing Capacity requirements for P1, P2 and P3 load case.

VIII. CONCLUSSIONAND RECOMMENDATIONS

CONLUSION

- To fulfill the objective of the study, one under ride protection device for a rear under ride accident was designed and its performance compared. A quasi static test was performed on guard to test the strength and energy absorption capacity by withstanding the applied loads. All the constrained and boundary condition used for the study worked well.
- Nearly six designs were studied and run simulation to study the effectiveness of each guard and results were plotted. Every Design Modification is done based on previous iterations results and finding, keeping an eye on available design space and constraints.

- Weight to strength factor and energy absorption was the key design principles used for developing Rear Underrun Protection Device.
- During FE modeling it has been assumed that bolts are elastic and safe. It is recommended to use suitable bolt grade as per the axial and shear forces experienced by respective bolts.
- The load bearing capacity of the Rear Under-Run Model was increased by a desired level. The Load Bearing capacity of the current RUPD increased from 68 KN to 71.2 KN as compared.

IX. RECOMMENDATIONS

- The RUPD is tested only for static condition. Dynamic condition will give us more insight on the designing of more accurate and promising design.
- Design and analysis of frontal under ride protection guard for the frontal scenario can also be studied.
- Design for the weight reduction can be done.
- Study can be done with actual moving and /or stationary truck, which is more realistic.
- Design methodology need to be generated based on RUPD designing for variable range of load carrying capacity.
- More energy absorbing models can be analyzed for the protection.
- Uses of composites can be a future area for development of efficient RUPD.
- The bolt forces observed are very high so it is recommended to study the number of bolt required to attach RUPD to chassis so as to minimize load coming on each bolt

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