A DESIGN STRATEGY ON THE TALBOT TYPE ARTICULATION OF A FREIGHT WAGON

EMIN SUNBULOGLU & ERGUN BOZDAG
Istanbul Technical University, Department of Mechanical Engineering, Istanbul, Turkey

ABSTRACT
The Talbot Type Articulation is a classical design for use in articulated freight wagons, where one bogie under the coupler serves to both sides coupled. Though being a traditional design, lighter weight wagons designed to transport bulky but lightweight materials return it to one of the alternatives to increase load-to-tare ratio. However, the design criteria of the Couplers are not clearly defined in TSI and recalled standards, but simply recalled to be stronger than other coupling means at ends. This fact requires other challenges of loading case reviews, since the coupling is acting both in tensile and compressive loads, and it is the means of generating the intermediate joint. To end up with a design, a way of generating the possible load cases and combining these into the scope of the DVS fatigue code for fatigue assessment is presented in this study, along with experimental verification of the model.
Keywords: fatigue assessment, DVS standard, railway design, wagon coupling, talbot articulation.

1 INTRODUCTION

The Talbot Type Articulation is one of the established methods for joining two wagons permanently [1]. It is used widely in Europe for freight transport wagons, which are designed mainly for container transport and combined transport. Lighter weight wagons designed to transport bulky but lightweight materials return it to one of the alternatives to increase load-to-tare ratio.

The Talbot Type Articulation is essentially a design by Waggonfabrik Talbot, which was a company founded in 1838, got its aforementioned name in 1866, and has been acquired in 1995 by Bombardier Transportation, Canada. The factory, to the Authors' knowledge, is known with its own patented designs (including the articulation type that is the main topic of this study), and the "Talent", one of the first intercity-type EMU/DMU trains.

The articulation is by itself enables reduction of weight of the wagon by eliminating one of the bogies from design. This way, however, the total weight transportable also decreases, but the overall system becomes feasible for bulky but lightweight materials. Considering the limitations of combined transport, where container weights are mostly restricted with the weight limitations of the road vehicles, the overall efficiency of the railway operation part of combined transport action increases.

The coupling itself located at the centre of the wagon this way is subject to more complicated loadings compared to standard drawgear-buffer-combination coupling based designs, since

1. It is under action of alternating tensile and compressive loads acting to ends of wagon, where the buffers only react to compression and the drawgear only to tension
2. The assembly also transfers vertical (payload) loads to the bogie centred underneath it, as a completely different concept.

However, the most critical part of the assembly, the pivoting part, is not subject to vertical loads. It only reacts to alternating tensile and compressive loads.

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This study is based on proposing a methodology of design control of this pivoting part of the Talbot Type Articulation, which is expected to cover the requirements of the European standards on new-built railway rolling stock.

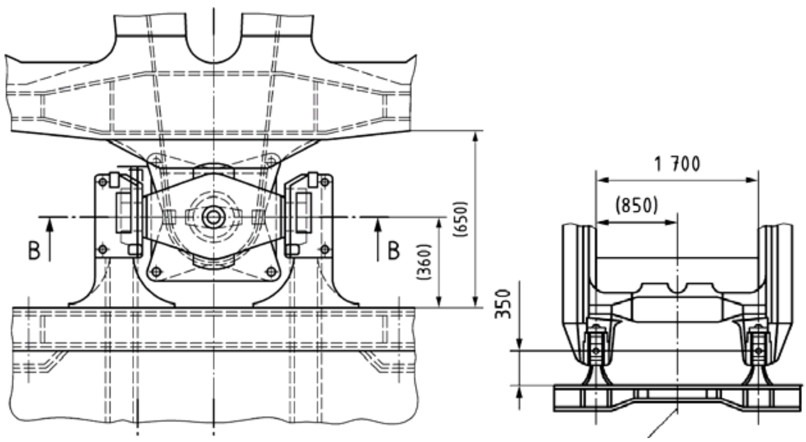


Figure 1: Standard views of Talbot Type Coupling [1].

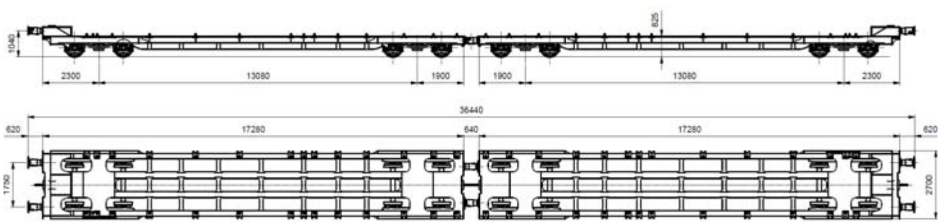


Figure 2: A permanently coupled wagon with conventional coupling having 4 bogies. (SFFGGMRRS, GOKRAIL, Turkey.)

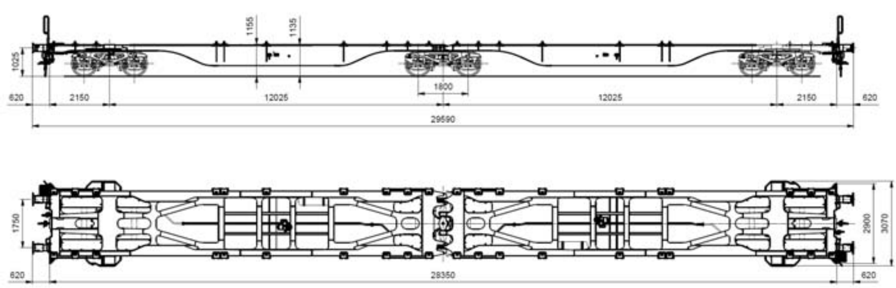


Figure 3: A wagon permanently coupled with Talbot Type coupling having 3 bogies. (SGGMRS, TUDEMSAS, GOKRAIL, Turkey.)

2 THE REQUIREMENTS

After many years of UIC (International Union of Railways / Union Internationale des Chemins de Fer) Regulations, the Continental Europe has adopted the TSI (Technical Specification for Interoperability) rules for newly built rolling stock, infrastructure, high speed trains and train electronics of all kinds. Throughout the years after introduction of the concept, the TSI concept is approaching a settlement and the application is getting more and more standardized and straightforward, which are also still (partially) based on the UIC rules and conventions.

The freight wagons are covered in "COMMISSION REGULATION (EU) No 321/2013 of 13 March 2013", which is the so-called "WAG-TSI" [2]. The interoperability constituents (IC) and other parts that are subject to European regulations are all recalled in the WAG-TSI and are under strict control.

For inner couplings like The Talbot Type Articulation for freight wagons are covered in Paragraph 4.2.2.1.2 of WAG-TSI [2] and must fulfil the requirement of *"The longitudinal strength of the inner coupling(s) shall be equal to or higher than the one of the end coupling(s) of the unit."* However, this precise bit of information about the requirement to be satisfied by inner couplings is not directly specifying the standards and engineering calculation methods of the design verification. Based on this fact, a set of standards have been investigated.

2.1 Related standards to inner couplings static safety

Going through the set of available EN standards [3-10] applicable to TSI [2] regulations, the basic applicable standards are predicted. The basic points applicable to Inner Couplings like the Talbot Type Articulation within these standards are provided in Table 1. A close investigation of the standards given in Table 1 yields the results that the category of axial load cases for SGGMRS Wagon where the Talbot Type Articulations to be utilized are based on F-II type according to WAG-TSI [2] Annex C §3. Thus, 1500kN Tensile load and 1200kN Compressive load maximum load scenarios apply for the central coupler. On the other hand, the loads on the coupler can also be calculated from the tables in EN 12663-1 (6.5.3 and Annex B) [3]. EN 12663-1 [3] also suggests superposition of vertical and axial load cases. However, for the central coupler of Talbot Type Articulation, the coupler does not bear vertical loads due to its design. Thus, no such scenario is applied.

UIC 572 [11] is a guide document (not recalled in TSI) that is directly aimed at defining load cases for conformity assessment of articulations of wagons. UIC 572 §3.1.3 [11] recalls, for central couplers, "the tensile strength of the central coupler should be " when the coupled wagons have certain concerns on brake systems for unrestricted safe utilization. Similarly, EN 16019:2014 [5] is related to automatic coupler design and verification, where under normal operating conditions, the static requirements are said to be satisfied under 1500kN Compressive load and 1000kN tensile load, which is higher than F-II category wagons for compression but lower than 1500kN drawgear requirement.

Since WAG-TSI [2] also recalls that in §4.2.2.1.2 that "The longitudinal strength of the inner coupling(s) shall be equal to or higher than the one of the end coupling(s) of the unit", thus for F-II type wagons of 1500kN draw-hook, a requirement of static strength of

- 1500kN Tensile load (due to Hook requirement [6,8])
- 1500kN Compressive load (due to EN 16019:2014 [5] requirement)

is applicable for the Talbot Type Articulation.

Table 1: Predicted standards applicable to inner coupling design according to WAG-TSI.

Standard ID	Applicable points	
	Paragraph	Content
EN 12663-1:2010 + A1:2014	6.5.3 6.7.4, 6.7.5	Proof load cases for articulations Fatigue load cases for articulations
EN 12663-1:2010 + A1:2014	5.4.2 ANNEX-A	Requirements for overall structural stress state of wagons
EN 12663-2:2010	General	Requirements for overall structural stress state of freight wagons Definition of axial loadings
EN 16235:2013	Annex L L.1,L.2	Overall main dimensions of Talbot Coupling
WAG-TSI	4.2.2.1.2	Inner coupling(s) should be strong(er) than outer couplings
WAG-TSI	Annex C-3	Wagon allowance F-II category
FKM GUIDELINE	Material Tables	Fatigue data of materials Dimensional effects
UIC 572	3.1.3	Inner coupling(s) proof load cases
EN 16019:2014	4.1 5.1.2	Automatic coupler proof test loads
EN 15566:2016	Overall	Static specs for DrawGear Fatigue specs for DrawGear
EN 15551:2017	Overall	Static specs for buffer Fatigue specs for buffer

2.2 Related standards to inner couplings fatigue safety

The fatigue evaluation conditions for intermediate couplers is also not well defined. As aforementioned, all standards related to inner couplings [1], [3]–[9] have static criteria (only [3] mentions somehow fatigue), but are somehow obsolete when compared to WAG-TSI [2] recalling that in §4.2.2.1.2 that *"The longitudinal strength of the inner coupling(s) shall be equal to or higher than the one of the end coupling(s) of the unit",* which also implies that it should be safer or at least at the same level of safety in terms of fatigue evaluation too.

The fatigue under longitudinal loads for coupling members is concerned in the standards EN 15566:2016 (Draw gear and screw coupling) [6] and EN 15551:2017 (Buffers) [8] which are mainly specific to the devices recalled thereof. EN 12663-1 [3] and EN 12663-2 [4] do not directly infer a longitudinal load standard to be applied ([3] has some longitudinal & lateral acceleration data valid for non-freight rolling stock), so in fact the buffer/coupler [6], [8] standards form a couple to provide a practical choice of referable load-case, since by simple logic, in can also be envisaged that the central coupler is a combination of a buffer and a drawgear as a single device, so the load cases can be derived from these. Though in WAG-TSI [2] the standards referred are EN 15566:2009+A1:2010 [7] and EN 15551: 2009+A1:2010 [9], more precise definitions of fatigue conditions with more exact loads are referenced in the more recent versions, so those versions are utilized throughout this study.

The drawgear/hook fatigue loads are well-defined as a dynamic loading of R≈0.1 and R≈0.03 in EN 15566-2016 [6], in Table A.2, Annex A. For the buffer, the conditions are a more complicated, according to the dynamic load cases are defined in EN 15551:2017 [8].

Table 2: Fatigue load cases for buffer and drawgears/hooks.

No	Origin	Type	Min Load	Max Load	Cycles	R (Min/Max)
1	**EN 15566:2016 Table A.2**	Tensile	30kN	330kN	1.500.000	0.091
2	**EN 15566:2016 Table A.2**	Tensile	30kN	1045kN	2.150	0.029
3	**EN 15551:2017 Annex B (B.2.7)**	Compressive housing	50kN	250kN	300.000	0.200
4	**EN 15551:2017 Annex F**	Compressive endurance	30kN	400kN	10.000	0.075
5	**EN 15551:2017 Annex G (C1)**	Compressive elastic part	00kN	≤1000kN	9.000	0.000
6	**EN 15551:2017 Annex G (C2)**	Compressive elastic part	00kN	≤1000kN	3.600	0.000
7	**EN 15551:2017 Annex G (C3)**	Compressive elastic part	00kN	1000kN (Test E3 limit)	600	0.000

There are two different considerations that are handled in separate cases of (i) Annex F – endurance testing under service load for elastic system and (ii) Annex G – Endurance testing under buffing load for life-cycle simulation [8]. For both of these cases, the load levels to be applied depend on the design category of the buffer, as well as different levels, which are pre-sequenced. All these load cases are summarized in Table 2 to generate the whole possible spectrum of loads that can be applied.

3 STATIC ASSESSMENT

The assessment of the Talbot Type Articulation was initiated by a static assessment. A Finite Element Analysis (FEA) Model has been set-up from 3D Model of the joint, which was then applied loads and boundary conditions. The FEA results then have been validated against static tests and strain-gauge applications.

The model of the wagon with articulator in between of the two parts of a SGGMRS wagon is shown in Fig. 4. The abstract model developed for the analysis of the Articulator is shown in Fig. 5.

3.1 Developing the FEA model

The CAD Model of Talbot Type Articulator has been converted into a FE model by disregarding structurally unimportant parts (such as screws, slide-plates, liners, etc…) of the full CAD model. Due to the structural style and methods of construction all the 3D members were retained as-is and 3D mesh was applied. The mesh of the model consisted of mainly Quadratic Tetrahedrons with Modified Formulation (C3D10M), and aiding elements such as Membrane Elements (M3D6), springs; as well as distributing coupling and kinematic coupling elements where appropriate.

The connections of Talbot Type Articulator to SGGMRS wagon is via a welded structure. Thus, the location, positioning and the surrounding area of welds are important in stress analysis. For the prepared model of the Talbot Type Articulator, an abstract part of the complete wagon has been considered. All of the weld areas are identified as perfectly joined solid bodies and have been modelled as one part. The model preparation and meshing processes were completed in Abaqus 6.13-1 CAE (Dassault Systémes) based on the CAD

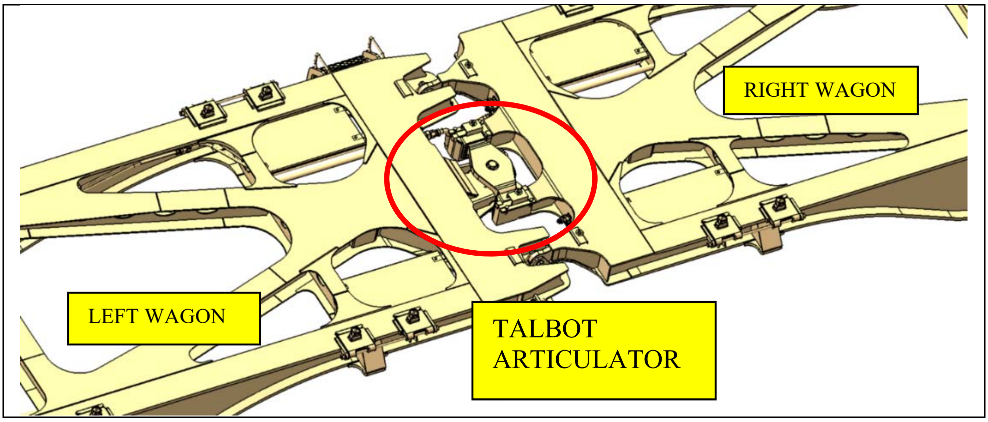


Figure 4: The wagon model with Talbot Articulated Joint.

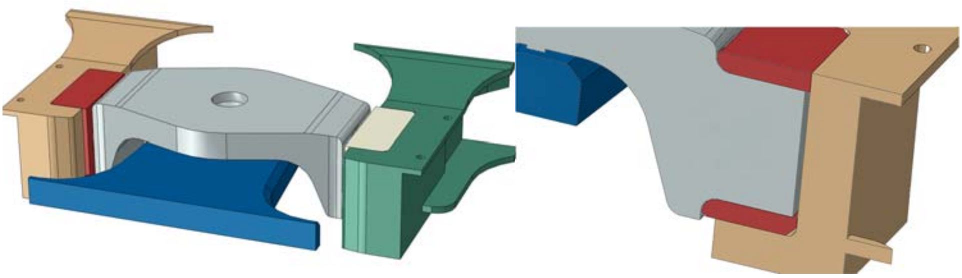


Figure 5: The abstract model of Talbot Articulated Joint developed.

model. After the mesh has been prepared, the numerical analysis solver process and the results post processing procedures were again completed in Abaqus 6.13-1 (Dassault Systémes) environment.

The load cases considered are given in Table 4 and material parameters used are given in Table 5, which are based on material used during the production. Material data for S355 is taken from EN 12663-2 [4]. Material data for 42CrMo4 steel is taken from FKM Guideline [11] and the material tests after heat-treatment.

The abstract model of the Talbot Type Articulation is fixed at the right wagon joints and is axially loaded at the left wagon interface. All the touching surfaces are interacted via contact, which resulted in a highly non-linear problem. The boundary conditions and mesh are shown in Fig. 6.

Table 3: Load cases applied for static analysis.

NO	Reason	ANALYSIS DEFINITION
1	WAG-TSI §4.2.2.1.2	Compression of 1500kN
2		Tension of 1500kN

Table 4: Material parameters used for analysis.

	E	ν	ρ	Yield Stress
S355	210 GPa	0.30	7850 kg/m^3	355MPa (base), 323 MPa (weld)
42CrMo4	210 GPa	0.30	7850 kg/m^3	750MPa (base), 675 MPa (weld)

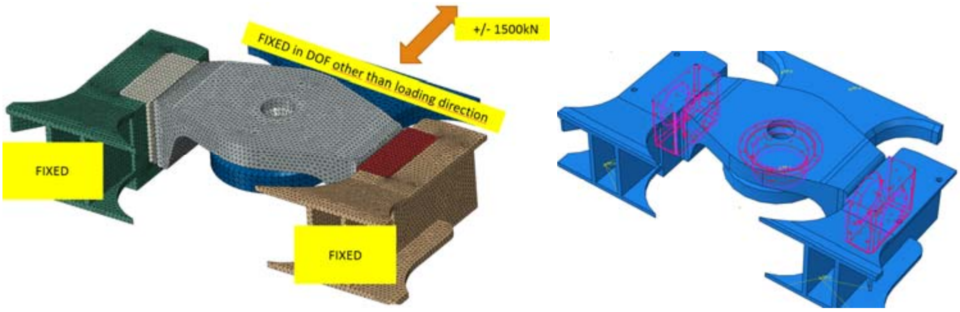


Figure 6: Boundary conditions and contact regions.

3.2 FEA results

The numerical stress outcomes have been obtained at specific regions of the model, as well as the contours of the Von Mises Stress results for aforementioned defined load scenarios. The contours of Von Mises Stress values are shown in Figure 7 for critical views and parts.

Highest stress values (except local mesh defects) is about 850MPa for the whole system (in the radii region of side-clamps) According to standards EN 15566-2016 [6], EN 15551:2017 [8], EN 16019:2014 [5], UIC 572 Ed. 4 (2011) [12] and EN 12663-1:2010 [4], the main criteria for articulations (and draw gears, hooks, etc. inter-wagon connective equipment) after the static tests is generally “not to have permanent deformations”. According to FEA outcomes, at very limited points the yield stress of 750MPa is exceeded. On the other hand, the exceedance is one-directional (i.e. not alternating) and it is observable that the design also satisfied EN 12663-1:2010 + A1:2014 §5.4.2, and §ANNEX-A [3] requirements for “local notch regions in very small regions”. Provided that the material requirements aforementioned are met, the design meets the relative standards’ and WAG-TSI [2] requirements.

3.3 Static tests – verification of the FEA model

After the FEM control of the coupler has been completed as reported in the previous sections, strain-gages have been distributed on the coupler according to the critical points observed in the analysis. Some additional points were also selected based on the previous experiences of the measurement team and site observations. Depending on the stress state (2D or uniaxial stress state) of the coupler’s specific critical points, single element type strain gauges were applied. In the evaluation of Static FEM Analysis studies prior to tests, the structure is found to be compliant to WAG-TSI [2] requirements.

The tests have been executed on the special hydraulic test bench which has been designed and manufactured according to requirements of related international standards at TÜLOMSAŞ plant in Eskisehir, Turkey. The coupler was installed on a SGGMRS Wagon

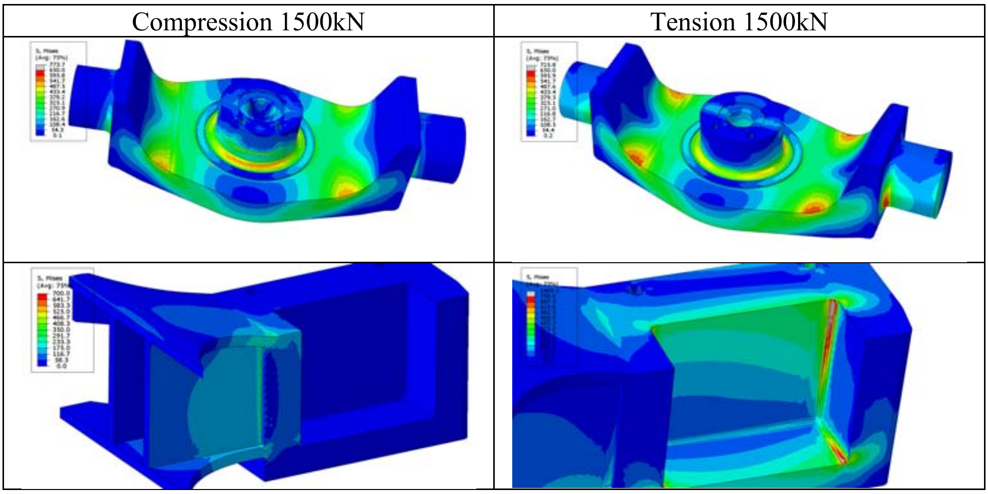


Figure 7: Sample stress contours via FE analysis.

prototype. Since SGGMRS wagon is F-II, the compression load has been applied in test up to 1200kN [2], [4]. Thus, the FEA comparison was also handled at this load.

The FEA data is compared to test results to validate the model. The following comparison table is obtained. Comparisons provided in Table 6. A plot of loading/unloading cycle for one gage is provided in Fig. 8 along with gage location example.

The results show that the stress distributions are mainly compatible. During tension and compression, different amounts of errors in calculations are observed, however they can be attributed to the fact that during the actual tests, the axial (lateral and vertical) positions of the system is not as fixed as the FEA analysis boundary conditions, and changes in the distances create unforeseen (bending) moments which are seen to affect the stress conditions. But as a final outcome, it can be stated that the FEA model is accurate enough for being considered as a valid model and further analysis can be carried on.

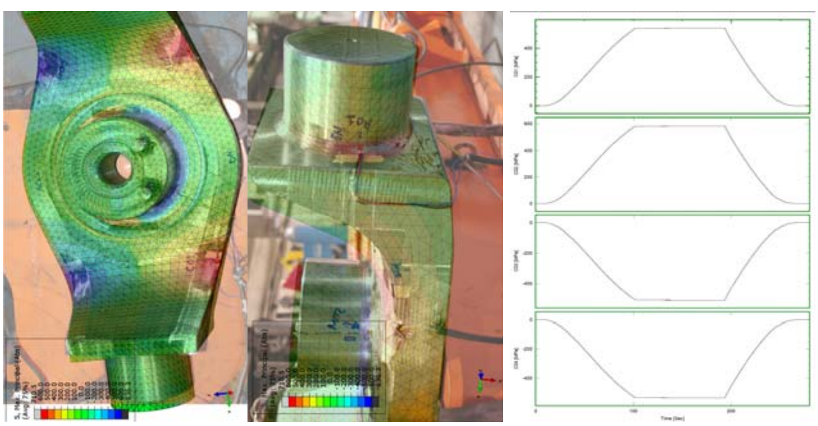


Figure 8: Representative gage locations and strain time history.

Table 5: Load cases applied for static analysis.

		Coupler 1500kN Tension			Coupler 1200kN Compression		
Gage ID	**Element ID**	**Test [MPa]**	**FEA [MPa]**	**Error**	**Test [MPa]**	**FEA [MPa]**	**Error**
C01	**209456**	540.9	554.4	2.4%	-402.9	-390.5	-3.1%
C02	**210414**	587.5	565.2	-3.9%	-399.6	-406.6	1.7%
C03	**209744**	-507.9	-512.9	1.0%	359.7	325.7	-9.5%
C04	**210702**	-543.0	-530.5	-2.4%	357.8	327.6	-8.4%
K01	**57782**	724.9	752.5	3.7%	2.1	2.1	N/A
K02	**266158**	560.6	709.1	20.9%	0.5	2.7	N/A
K03	**50218**	-235.9	-138.1	0.9%	-2.5	0.6	N/A
K04	**52413**	265.7	197.0	-34.9%	-70.0	-93.4	33.0%
A01	**203015**	687.3	631.4	-8.9%	-281.4	-281.7	0.1%
A02	**213193**	-611.1	-557.8	-9.6%	347.6	303.3	-12.8%
M01	**372773**	-598.8	-513.4	-16.6%	538.1	459.0	-14.7%
M02	**372720**	-599.3	-511.1	-17.2%	556.6	459.4	-17.5%

4 FATIGUE ASSESSMENT

As it has been stated before, the fatigue evaluation conditions for intermediate couplers is not well defined. However, following load cases provided in Table 2 above, some rationale can be derived. From a practical point of view, applying the many variations of loads is almost a must when there are non-linear members (such as the damping elastomers of buffers/hooks) contained in a system, but for purely-elastic (theoretically) systems, it is more advantageous to combine the load cases into a single spectrum.

4.1 Handling of various load cases

Taking care of the data, and regarding that the resilient part's tests are not directly applicable to the buffer itself, the load cases of the coupling devices can be reduced to ones that are summarized in Table 6.

A more conservative approach would be to combine all of above into a single spectrum. For achieving results based on this aim, the concept of EC3 [10] / IIW [13]/FKM [11] can be used for changing the stress range. According to these standards, the change of stress ranges will result in change of stress amplitude following eqn (1):

Table 6: Load cases applied for static analysis.

No	Origin	Type	Min load	Max load	Cycles	R
1	EN 15566-2016 Table A.2	Tensile	30kN	330kN	1.500.000	0.091
2	EN 15566-2016 Table A.2	Tensile	30kN	1045kN	2.150	0.029
3	EN 15551:2017 Annex B (B.2.7)	Compressive Housing	50kN	250kN	300.000	0.200
4	EN 15551:2017 Annex F	Compressive Endurance	30kN	400kN	10.000	0.075

$$\frac{\Delta\sigma_1}{\Delta\sigma_2} = \left(\frac{n_2}{n_1}\right)^{\frac{1}{m}} \tag{1}$$

where n_i is the cycles to fatigue failure under stress range $\Delta\sigma_i$. Using force values as singular input (one excitation – proportional loading and stress state) and considering all stresses are due to this force input ΔF, it is possible to scale the load ranges $\Delta F_1 = 200kN$

$$\frac{\Delta F_1}{\Delta F_2} = \frac{200}{\Delta F_2} = \left(\frac{n_2}{n_1}\right)^{\frac{1}{m}} = \left(\frac{15\cdot 10^5}{3\cdot 10^5}\right)^{\frac{1}{3}} \tag{2}$$

which yields $\Delta F_2 = 117kN$ where, holding R constant,

$$R = \frac{F_{\min}}{F_{\max}}\ ,\ F_{\min} = \frac{R}{1-R}\Delta F \tag{3}$$

so that $F_{\min} \approx 30kN$, $F_{\max} \approx 147kN$ can be obtained, using m=3 according to EC3/IIW/FKM [10], [11], [13] rules. This enables that combining load cases 1 and 3 into a single one with highly alternating loads will cover the required number of cycles. Moreover, taking the maximum loads of Table 6 Rows 2 and 4 to max loads of Table 6 Rows 1 and 3 above, additional cycles may be obtained. It can be seen that the limit of 1.5 million cycles (Table 6) is not greatly altered in this case. Thus, combining all these, a worst-case scenario of fatigue analysis for the articulator can be obtained as to the case given in Table 7.

The requirement of Table 7 can be safely obtained by applying rules of DVS 1612 [14], which is a standard accounting for 2.0 million cycles by default (which imposes 5% additional safety over 1.7 million cycles). It should be noted that the R value of the 1,700,000 cycle combined test is much more a strict requirement than the higher R-valued fatigue tests since materials suffer lower safety limits for reversing (alternating) loads.

4.2 Results of fatigue analysis

It can be seen that, under fatigue analysis conditions, the stress levels are expected to be lower (330/1500 = 22% for tensile, 147/1500 = 9.8% for compressive loads.) compared to values of the static tests via load scaling.

For the fatigue analysis LIMIT Software (Version 2016) from CAE Solutions (Austria) has been utilized. The analysis utilized the surface elements (membranes M3D6). The assumption is stress-based high-cycle fatigue (since stress levels are far below elastic yield) and DVS 1612 [14] standard is to be utilized. The articulator and the side clamps will be treated as base material, i.e. as "non-blasted, non-thermal influenced (i.e. thermal cut) material", thus have been chosen.

- notch case "A" for normal stresses and
- notch case "G+" for shear loads

Table 7: Combined load case for the coupler.

No	Origin	Type	Min Load	Max Load	Cycles	R
1	Combination	Alternating	-147kN	+330kN	1,700,000	-0.44

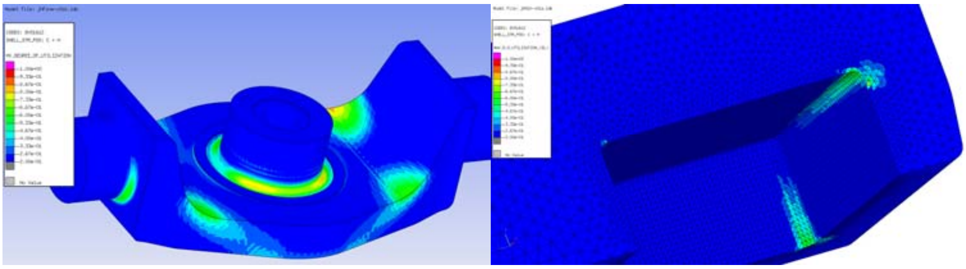


Figure 9: Material utilization contours of articulator region.

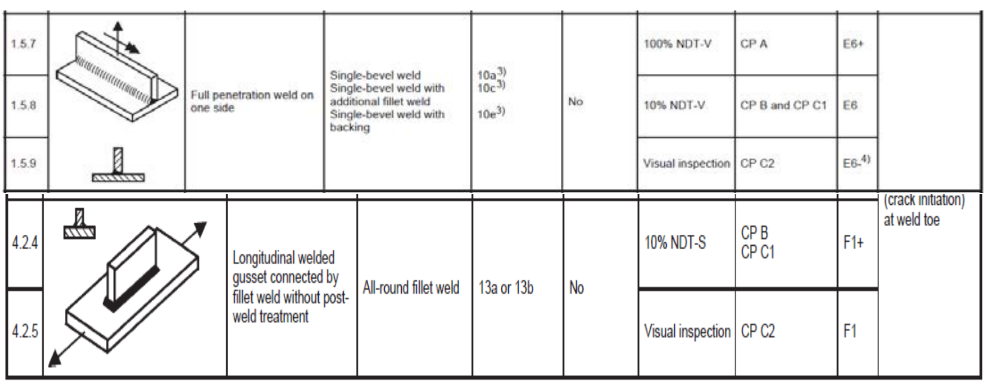
Additionally, static safety over material values (Ultimate) is taken as S=1.5 and yield safety is taken as S=1.1 without any residual stress. Loads have been taken as in Table 7. Due to Nonlinear contact conditions and highly changing stress state under tensile and compressive loadings, stress results from two separate analysis (tension and compression) have been used for the fatigue assessment to complete the load reversal cycle. The contour plots of the joint part are given in Fig. 10. It can be said that, based on the obtained numerical results, the overall material utilization for 42CrMo4 parts is about 0.85.

For the assessment of welded connections, the following cases apply: For connections to wagon by one-sided bevel weld, full penetration the notch cases are as in Table 8.

Checking stress contours around, in the following figures, the stress values are about 20MPa S11,20Mpa S22 and 30Mpa S12 max, thus, using simple calculations, we have about 50MPa of equivalent stress and about 1.2 safety (DoU approximately 0.8) considering elements (i.e. stress state) 1.5tx away from the weld point as to DVS [14] (excerpt in Fig. 10).

Table 8: Combined load case for the coupler.

Notch Case	**E6+**	**E6**	**E6-**	**F1+**	**F1**
Max Stress [MPa]	74	71	68	65	63



1.5.7		Full penetration weld on one side	Single-bevel weld Single-bevel weld with additional fillet weld Single-bevel weld with backing	10a 3) 10c 3) 10e 3)	No	100% NDT-V	CP A	E6+	
1.5.8						10% NDT-V	CP B and CP C1	E6	
1.5.9						Visual inspection	CP C2	E6- 4)	
4.2.4		Longitudinal welded gusset connected by fillet weld without post-weld treatment	All-round fillet weld	13a or 13b	No	10% NDT-S	CP B CP C1	F1+	(crack initiation) at weld toe
4.2.5						Visual inspection	CP C2	F1	

Figure 10: Notch cases for welded joints.

CONCLUSION

A strategy of mixed numerical and experimental method of a complex freight wagon is proposed and applied on a real case. The results of the analysis and the methodology has been seen to be satisfactory and providing an effective method for design assessment and validation.

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