ESTUDIO DE LA VIDA FRENTE A FATIGA DE SEMIRREMOLQUES. BANCO DE ENSAYO Y SIMULACIÓN POR ORDENADOR

Marco Carrera-Alegre[1], Luis Castejón-Herrer[1], Hugo Malón-Litago[1], David Valladares-Hernando[1], Carlos Martín-Lafuente[2], Javier García-deJalón[3], Pablo Luque[4], Daniel Álvarez-Mantaras[4], Enrique Alcalá[3] y Ángel Martín[3]

[1] Universidad de Zaragoza. c/ María de Luna, s/n - 50018 Zaragoza. Tfno: +34 976 762556.
[2] Lecitrailer S.A. Camino de los Huertos, s/n - 50620 Zaragoza.
[3] Universidad Politécnica de Madrid. INSIA. Campus Sur UPM, Carretera de Valencia (A-3), km7 - 28031 Madrid. Tfno: +34 913365300.
[4] Universidad de Oviedo. Escuela Politécnica de Ingeniería de Gijón. Campus de Gijón, s/n - 33203 Gijón. Tfno: +34 985 182059

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FATIGUE LIFE ANALYSIS OF SEMITRAILERS. FATIGUE TESTING BENCH AND NUMERICAL SIMULATION

ABSTRACT:
Freight transport by road currently is one of the most important mean of transportation in the worldwide logistics sector, due to a number of advantages compared to other means operating within the same sector. This road transportation is largely developed through the use of semi-trailers. In recent years there has been a tendency to optimize the weight of semi-trailers in order to lighten them and to reduce fuel consumption and emissions. Of course, this optimization has to be done without compromising vehicle safety. In this regard, special interest requires analysis of fatigue resistance, since the breakage in such vehicles appears in welded joints areas by the application of cyclic loading. Until recently, design methodologies applied were based primarily on experience and experimental analysis, remaining in the background optimization processes. However, in this article the research carried out by the participating universities and Lecitrailer SA company is summarized. This optimization work is focused to achieve a lightweight semitrailers with optimal behavior against fatigue. The research is based on the development and implementation of a fatigue testing bench for semitrailers, combined with numerical simulation tools for estimating fatigue behavior of these vehicles, and specifically in the areas of welding.

Keywords: Semitrailers, fatigue life, testing bench, FEM, welding, optimization.

RESUMEN:
El transporte de mercancías por carretera supone en la actualidad uno de los medios de mayor importancia en el sector logístico a nivel mundial, debido a una serie de características y ventajas que presenta respecto a otros medios que operan dentro del mismo sector. Este transporte por carretera se realiza en gran medida gracias a la utilización de semirremolques. Durante los últimos años ha existido una tendencia a optimizar el peso de los semirremolques con objeto de aligerarlos de cara a reducir el consumo y las emisiones contaminantes y aumentar la capacidad de carga. Por supuesto, esta optimización tiene que realizarse sin perjudicar la seguridad del vehículo. En este sentido, un especial interés requiere el análisis de la resistencia frente a fatiga, puesto que las posibles roturas en este tipo de vehículos aparecen por la aplicación de cargas cíclicas en zonas próximas a las uniones soldadas. Hasta fechas recientes, las metodologías de diseño aplicadas estaban basadas fundamentalmente en la experiencia y en el análisis experimental, quedando en un segundo plano las operaciones de optimización. Sin embargo, en el presente artículo se resume el trabajo realizado por las Universidades participantes y la empresa Lecitrailer S.A. para lograr semirremolques aligerados con un comportamiento óptimo frente a fatiga. Esta labor de optimización se basa en el desarrollo y aplicación de una bancada de ensayo frente a fatiga de semirremolques, en combinación con herramientas numéricas de simulación del comportamiento frente a fatiga de estos vehículos, y específicamente en las zonas de soldadura.

Palabras Clave: Semirremolques, vida a fatiga, banco de ensayo, MEF, soldadura, optimización.

1. INTRODUCTION

This article summarises the research work carried out by a group of Spanish universities and the company Lecitrailer SA to develop lightweight semi-trailers with optimised fatigue performance. This optimisation work is based on the development and implementation of a fatigue test bench for semi-trailers which is unique in the world, combined with the use of numerical simulation tools for the analysis of fatigue performance of these vehicles, and more specifically of the welded parts.

There is a growing interest in developing lightweight semi-trailers[1], either via design improvements or via the incorporation of new materials, such as aluminium alloys, high-strength steels amongst others. This leads to a series of advantages such as reduced emissions of pollutants, fuel savings, better dynamic performance, enhanced vehicle stability and reduced load tipping risk, increased payload capacity and reduced maintenance costs.

Vehicle design criteria are based on stiffness values and resistance to static and dynamic loads. Often these requirements are not specified in relevant Regulations and thus each vehicle manufacturer applies his own set of design criteria[2]. There is therefore a need, in the vehicle manufacturing industry, for the determination of the different static and dynamic stresses acting on a vehicle under various possible situations[3]. Of all possible load cases, fatigue stresses at or near welded joints are especially critical to the life of the vehicle, since this is where fatigue stresses and fractures tend to occur.

When developing a new semi-trailer vehicle, the final stage is the production of a prototype and its testing under various static, dynamic and fatigue loads. Highly reliable numerical simulation techniques have already been developed, which provide an accurate representation of the actual conditions of each one of the possible events which were previously determined. As a result, the need to manufacture prototypes during the design phase and the subsequent tests is minimised, with obvious cost savings. Only the prototypes for the optimal configuration [4] have to be produced and tested.

For this reason a unique, word-first, fatigue test bench was developed in collaboration with the company Lecitrailer SA, for testing critical manoeuvres. However, in order to reduce the number of prototypes to be produced, it should allow for a reliable prediction of the number of cycles to failure associated to the most critical manoeuvre for a semi-trailer, which is the minimum radius manoeuvre. Moreover, given the flexibility that a typical chassis of a semi-trailer has, the critical points where fatigue failure may occur are those corresponding to the weld beads, in particular those areas affected by the heat generated during the welding process.

One of the most commonly used tools for the design of vehicle structures in general, and of semi-trailers in particular, is the simulation of structural behaviour using numerical techniques based on the finite element method (FEM)[5][6]. This tool provides an approximation to the real solution of variables of interest, both in terms of rigidity (deformed structure, displacements, etc.) and resistance (stress, plastic deformation, etc.). However, it must be kept in mind that numerical models do not provide an exact solution. Consequently, this methodology should be complemented with experimental measures which can validate the theoretical models considered.

The application of experimental techniques on real prototypes is a solution to this problem. On the one hand, it provides real values at the measuring points, and on the other it helps validate the numerical models discussed above. The test methods traditionally used on this type of vehicle (field tests) have a number of important limitations (reproducibility of test, execution times, high cost, performance under uncontrolled conditions, etc.), which have led to the development of specific tests for the experimental analysis of semi-trailer structures.

The test bench developed for the fatigue analysis of large vehicles[7], shown in Figure 1, is prepared to reproduce the manoeuvres considered critical in this type of vehicle and is suitable for testing under extreme operating conditions, providing results for variables such as, for example, stiffness, strength, dynamic behaviour, performance under fatigue loads, vibrations, and other influencing variables.

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Figure 1. Fatigue test bench for semi-trailers developed for this study

2. DEFINITION OF CRITICAL MANOEUVRABILITY CONDITIONS

When carrying out numerical simulations or tests on a semi-trailer, it is of great importance to know in detail the set of loads applicable to the vehicle[7][8]. Figure 2 shows the most significant ones.

2.1. REST POSITION/ STRAIGHT MOTION / POT-HOLED

In the case of semitrailers, the "rest position" corresponds to a case where the vehicle is stationary and loaded to its maximum capacity, or where it is moving at a constant speed while coupled to the tractor through its 5th wheel, which is a hitch plate-shaped horseshoe located in a central rear position that holds the semi-trailer by anchoring its king pin.

2.2. BRAKING

This scenario illustrates the state reached when a loaded vehicle of the characteristics described in section 2.1 is subjected to an abrupt braking manoeuvre.

2.3. MINIMUM TURNING RADIUS MANOEUVRE (MTRM)

This scenario applies when the tractor pulls the loaded semi-trailer at an angle of approximately 90 ° (all performed at low speed).

3. DEVELOPMENT OF AN INNOVATIVE EXPERIMENTAL TECHNIQUE: DESIGN OF A FATIGUE TEST BENCH FOR THE ANALYSIS OF SEMI-TRAILERS.

3.1. LIMITATIONS IN CURRENT TECHNIQUES

The field tests currently used are very effective, but have a number of limitations such as: the need for a driver and a test track; reduced reproducibility ; long test times; complex testing and vehicle manipulation requirements; influence of environmental conditions, etc. Apart from avoiding the aforementioned limitations, one of the advantages of using a laboratory test facility, is the possibility of establishing a purely laboratory controlled approval testing protocol for semi-trailers in the future, as is currently the case with other types of vehicles.

3.2. NEW TESTING TECHNIQUE FOR SEMI-TRAILERS

First of all, the least favourable combination of manoeuvres for the vehicle chassis has been analysed in detail. It was concluded that the two most demanding manoeuvres in terms of mechanical requirements for a semi-trailer are:

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- **Minimum turning radius manoeuvre + Kerb step**. This combination of operations could occur, for example when the tractor cab – semi-trailer assembly enters a building.
- **Pot-holed road**. In this case the vehicle is performing "normal" manoeuvres except for a number of variable forces which appear due to uneven road surfaces (presence of pot-holes).

After identifying such manoeuvres, the operating conditions of the vertical and horizontal hydraulic actuators of a test bench for semitrailers were deducted. These are shown in Table 1. The values of maximum displacement and force required in the actuators were obtained through numerical simulation, using models for semi-trailers. These are the displacement and force values necessary in order to safely apply on the test bench the load case for the minimum turning radius manoeuvre. The values of speed and flow of the actuators depend on the characteristics of the hydraulic group used and must have the capability to adequately reproduce the effect of pot-holes.

The dimensions of the test bench are determined by the maximum size of the vehicle (14 x 2.5 m^2), which should be supported by seven points in the base, namely the king pin, which is the bolt connecting the semi-trailer with the tractor unit, and the wheels of the three rear axles, with specific actuator-wheel couplings. Furthermore, the test bench has been designed with a certain flexibility to allow for the testing of vehicles of various dimensions and with different wheelbases. This is possible thanks to a guideway system that enables the movement of the uprights supporting the wheels.

4. DEVELOPMENT OF THE TEST BENCH

The proposed test bench consists of six vertical actuators which apply forces on each of the wheels of the semi-trailer. There is also a horizontal actuator that applies a force on the king-pin, equivalent to that exerted by the tractor truck. This set up is shown in Figure 3.

The mechanical system resulting from this configuration is formed by a suspended mass (corresponding to the weight of the vehicle and of the load), an unsprung mass (corresponding to the weight of the semi-trailer axles) and the mass of the rod placed in each of the wheels. This model was analysed using the FEM based on the conditions discussed above.

All components of the test bench were calculated using the FEM in order to reach an optimal solution in terms of strength, stiffness and fatigue and were designed to be as light as possible. Once the designs of the various components and systems that make up the test bench had been completed, the next step was to develop and manufacture them for their subsequent integration[9].

In parallel to the development of the test bench components, the civil works concerning the structure itself that serves as bases to accommodate other systems integrated into the test bench were carried out.

During the development of the civil engineering works, the set of supports was constructed as per applicable standard design specifications.

In general terms, the hydraulic system comprises the following main elements: actuators, control valves, a hydraulic unit and auxiliary elements[10]. All of these elements were incorporated into the test bench once the supporting structure had been assembled.

The aforementioned elements were assembled on the test bench in addition to others such as the control system, auxiliary connecting elements, the heat exchanger to protect the hydraulic unit, etc.[11].

The detailed process for construction of the test bench is shown in figure 4.

5. APPLICATION EXAMPLE: STRUCTURAL PERFORMANCE TEST OF A SEMITRAILER BY USING THE TEST BENCH

5.1. PREPARATION OF THE VEHICLE

First of all, the vehicle must be placed on the bench where it can be loaded with a series of calibrated blocks.

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Once the vehicle is on the bench, the next step is to fit strain gauges at certain points in order to obtain a number of measurements. For example, the first measuring point is located in one of the cross beams in the structure of the king pin's bridge (see Figure 7).

The next step is to load the vehicle using calibrated blocks. In this case, it has been decided to load the vehicle with a total of seven blocks, i.e., a total charge of 31.5 T.

5.2. MANOEUVRE DEFINITION

Given the range of options offered by the test bench, it has been considered appropriate to include in the test a load case corresponding to a combination of the minimum turning radius manoeuvre with the kerb step and pot holed manoeuvre or road profile (description of these manoeuvres can be found in section 2) . The displacement-time curve for this case scenario has been extracted from real data obtained from direct measurement of road profiles[12].

In each manoeuvre to be simulated, the magnitude, direction and sense of the loads on each wheel and / or the 5th wheel king pin must be known exactly. This set of forces will be induced by the system's actuators, by fixing their position and motion and taking into account their capacity. In the "minimum turning radius" manoeuvre, a high force causing bending and torsion stresses on the chassis is induced. Depending on the position of the kerb step tested, this twisting can be strengthened or attenuated, being very negative to the structure in the first case, due to the high mechanical stresses appearing in certain components. In addition, new tracking profiles for the various hydraulic cylinders of the test bench have been developed, as shown in Figure 5 which, unlike the profiles developed initially, have been aimed at the testing of the semi-trailer's structure when subjected to critical fatigue conditions and will be used to determine the fatigue life of the vehicle [8].

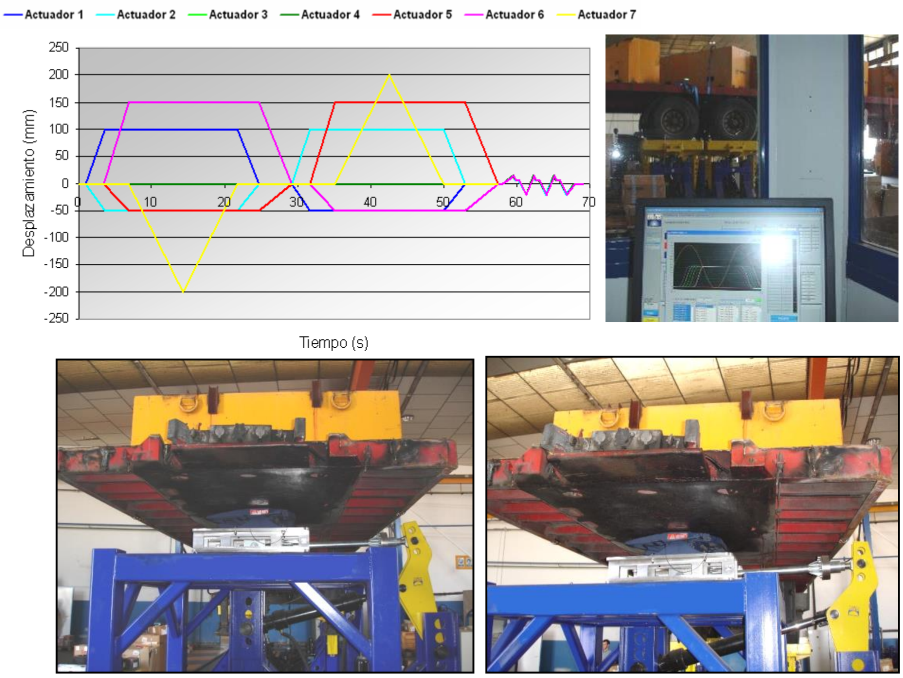

Figure 5. Profile developed for semitrailers fatigue test and displacement in the king pin during the test

It is convenient to monitor continuously the testing conditions to ensure that consistent profiles are reproduced correctly by each of the actuators. The control software of the machine[13] displays all significant variables during the test run. With this, the stresses are calculated from the measurements taken of material stiffness and deformation via carefully placed gauges and rosettes .

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6. NUMERICAL SIMULATION OF THE TESTED STRUCTURE. COMPARISON OF THEORETICAL-EXPERIMENTAL RESULTS

The first step here is the development of the numerical model of the structure . In addition to the semi-trailer itself, the corresponding load blocks used in the test have also been included in the simulation as shown in Figure 6.

The model developed consists of 167,063 nodes and 134,900 elements. Most of the elements used in the model are sheet-like elements with 3 and 4 nodes and reduced integration. As for loads and boundary conditions, the king pin is subjected to a displacement of 300 mm (the maximum allowable according to the graph above) and the bases of all the springs that simulate the suspension system are fixed.

In terms of strength, the stress level reached was calculated both for the full model and for the specific areas corresponding to the measuring points, i.e., at those points in the model where the strain gauges and rosettes were placed in the vehicle during the test.

For example, Figure 7 below shows the stresses obtained in measuring point 1. In this case, the measuring direction is transversal to the vehicle and the following stress distribution is obtained:

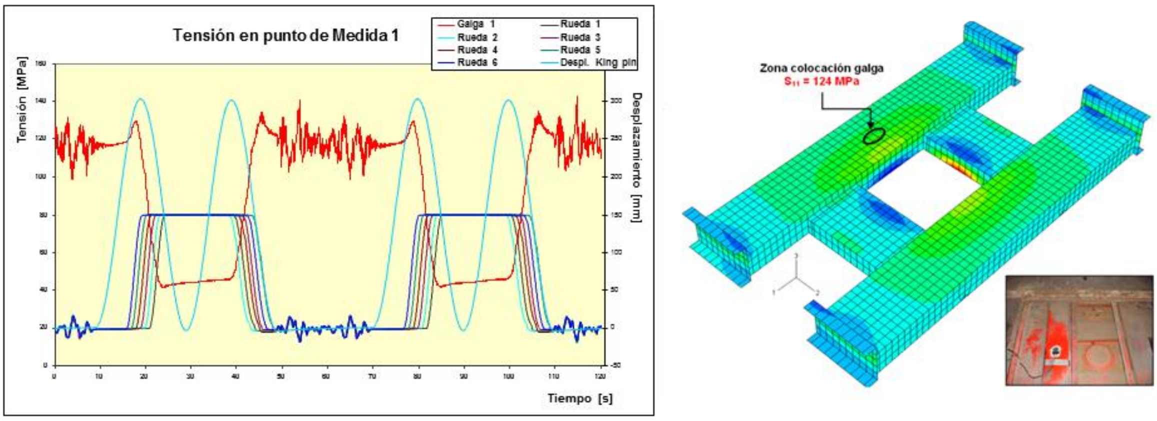

Figure 7. Time evolution for stresses and stress diagram (S11, in MPa) with the value obtained at measuring point 1

At measuring point 1, the stress reached in the direction of the gauge is 124 MPa.

Table 2 below compares the stress results obtained in the simulation with those obtained experimentally in the test bench at selected measuring points.

Measurement point	Stress	Numerical result [MPa]	Experimental result [MPa]	Variation [%]
P1: **Gauge at king pin's bridge**	**S11**	**124**	**126**	**1.6**
P2: **Gauge at longitudinal beam's section**	**S22**	**88**	**85**	**3.4**
P3: **Gauge at quadrant**	**S22**	**7.2**	**6.5**	**9.7**
P4: **Gauge at king pin's bridge**	**S22**	**-49**	**-46.5**	**5.1**
P5: **Rosette at support**	$\mathbf{S_{Mises}}$	**23**	**21.5**	**6.5**

Table 2. Comparative theoretical-experimental results at the measuring points selected

It is concluded that the correlation of the numerical model results with the experimental ones is satisfactory. Therefore, it can be said that the model developed is valid and can be used in subsequent design processes.

7. SIMULATION OF THE BEHAVIOUR UNDER FATIGUE LOADS OF WELDED JOINTS IN SEMI-TRAILERS

In parallel with the development of the fatigue test bench for semi-trailers and the overall simulation of semi-trailers using numerical techniques, a new line of research was developed. The objective was to analyse the behaviour of structural components and welded joints of semi-trailers under fatigue loads in a single simulation. This analysis allows to study the behaviour of the critical areas of semi-trailers, such as the welded joints of axle support structures or the section change of the longitudinal beams. These areas are critical depending on the materials used in the construction of these components and the critical manoeuvres of semi-trailers.

Numerical results of fatigue life for the base material of the structural components are obtained by methods of fatigue life prediction based on the Basquin-Manson-Coffin equation. The correlation of numerical and experimental results obtained from the base material is shown in Figure 8.

The load cycles in semi-trailers are of the “repeated stress cycle” type, and not of the “reversed stress cycle” type. Therefore, criteria considered in the study were modified in order to account for the effect of mean stress in the fatigue life predictions. This correction is included in the analysis by means of the mean stress correction developed by Morrow [14][15].

Morrow developed a correction method that is applicable to any of the criteria used for predicting fatigue life. This method allows the fatigue life of any specimen under study to be obtained by taking into account the mean stress. The correction method of Morrow consists of the modification of the fatigue life obtained from any fatigue life criteria using equation 1

$$N^{*} = N_{f} * \left(1 - \frac{\sigma_{m}}{\sigma'_{f}}\right)^{\frac{1}{b}}$$

(eq. 1)

In Equation 1, N_f is the fatigue life obtained without considering the mean stress, σm is the mean stress and σ´f y b are the fatigue strength coefficient and the fatigue strength exponent respectively. Applying the correction method developed by Morrow to the Basquin-Manson-Coffin equation, the latter is modified as follows:

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma'_{f}}{E} * \left(1 - \frac{\sigma_{m}}{\sigma'_{f}}\right) * (2N)^{b} + \varepsilon'_{f} * \left(1 - \frac{\sigma_{m}}{\sigma'_{f}}\right)^{\frac{c}{b}} * (2N)^{c}$$

(eq. 2)

As a result of the comparative analysis of different criteria for predicting fatigue life is obtained that the "Modified Universal Slopes" method has the highest correlation of numerical and experimental results for the base material of all analysed steels.

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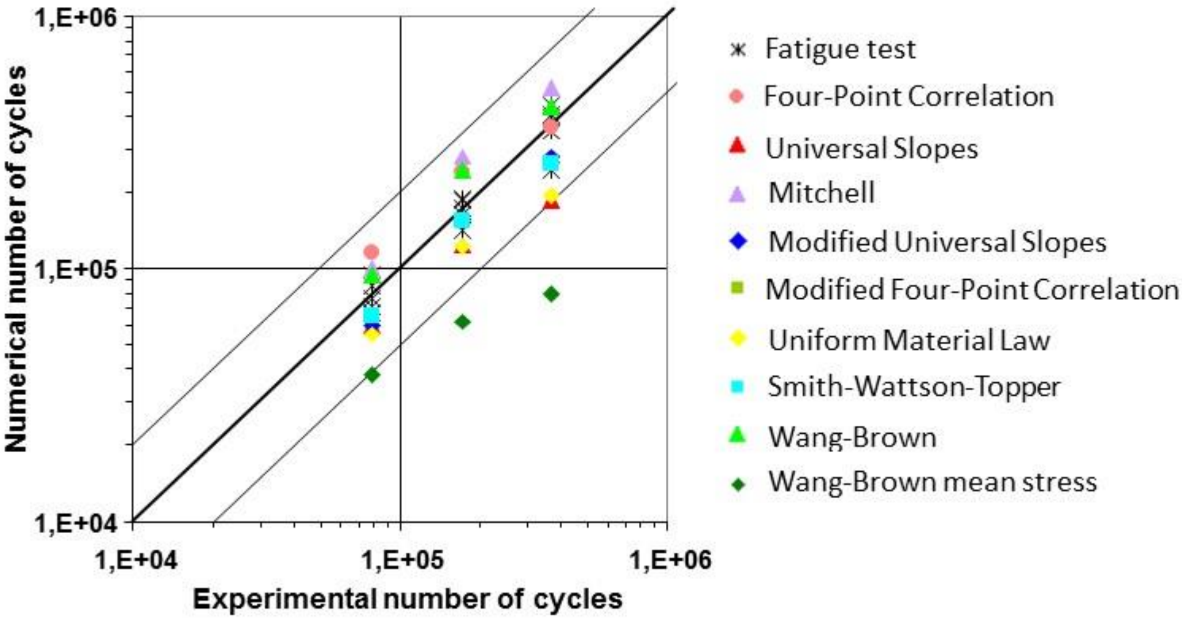


Figure 8. Comparison of fatigue life obtained by numerical and experimental methods of specimens of Domex 500 steel (Yield strength 500 MPa, Tensile strength 550-700 MPa, Elongation on failure 14-18%)

The criterion for predicting fatigue life proposed by the International Institute of Welding (IIW) [16], which is based on the fatigue analysis detailed in Eurocode 3 [17], is applied to the study of welded joints. Results of fatigue life for welded joints obtained by the IIW criterion were compared with results obtained by other criteria of uniaxial and multiaxial fatigue

In order to carry out the numerical analysis of welded joints, six numerical models were considered. Four of these models correspond to the models proposed by the International Institute of Welding (for example model 4 in Table 3) and the remaining two are additional models specifically developed for this study (models 5 and 6 in Table 3). The main characteristic of the latter two is the discretisation of the heat-affected zones by the welding process as well as the weld bead. These zones have different mechanical properties, which are obtained by means of traction tests and hardness tests carried out on base material specimens and welded specimens.

The curves provided by the IIW, specifically class FAT 63, have been used in the fatigue life analysis for welded overlap specimens. Mechanical properties are not equal in all zones of the numerical model due to the welding effect. Von Mises stresses of each integration point of the numerical model have been used to carry out the fatigue life analysis. Hugo Malón[8] in his PhD thesis demonstrated the validity of this procedure in numerical models applied at specimen level, by applying Von Mises stresses to the IIW curves. Results obtained showed a better numerical-experimental correlation in terms of fatigue life in all methods studied. In addition, these models provide the advantage of identifying critical areas in such unions

The results of fatigue life prediction obtained through the application a post-process subroutine allow the first three models proposed by the IIW to be discarded. Table 3 shows an example of the results obtained in the three remaining models for a particular load case.

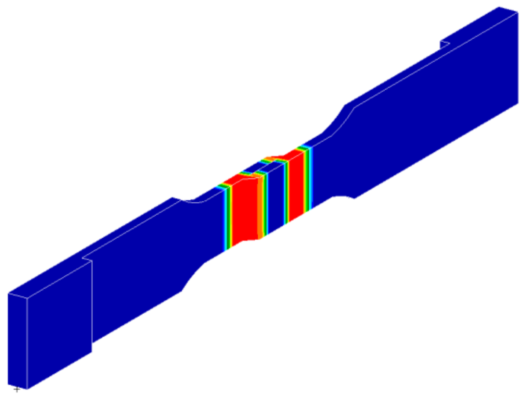
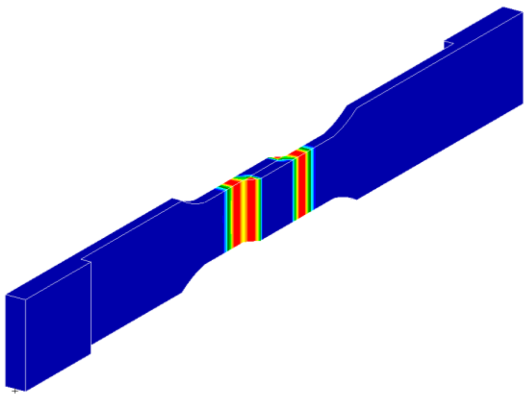
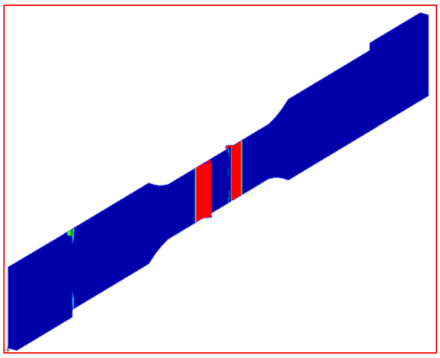
	
Model 4 : IIW	Model 5 : UNIZAR
 Minimum value: 13.000 cycles	
Model 6 : UNIZAR	

Table 3. Fatigue life results in welded specimens obtained by means of the post-process subroutine

The correlations between the numerical-experimental results obtained from the numerical model recommended by the IIW and the two numerical models developed by the University of Zaragoza are shown in figure 9. In this figure, fatigue life obtained by the application of the subroutine to the results obtained in previous numerical simulations is compared to experimental results of fatigue life.

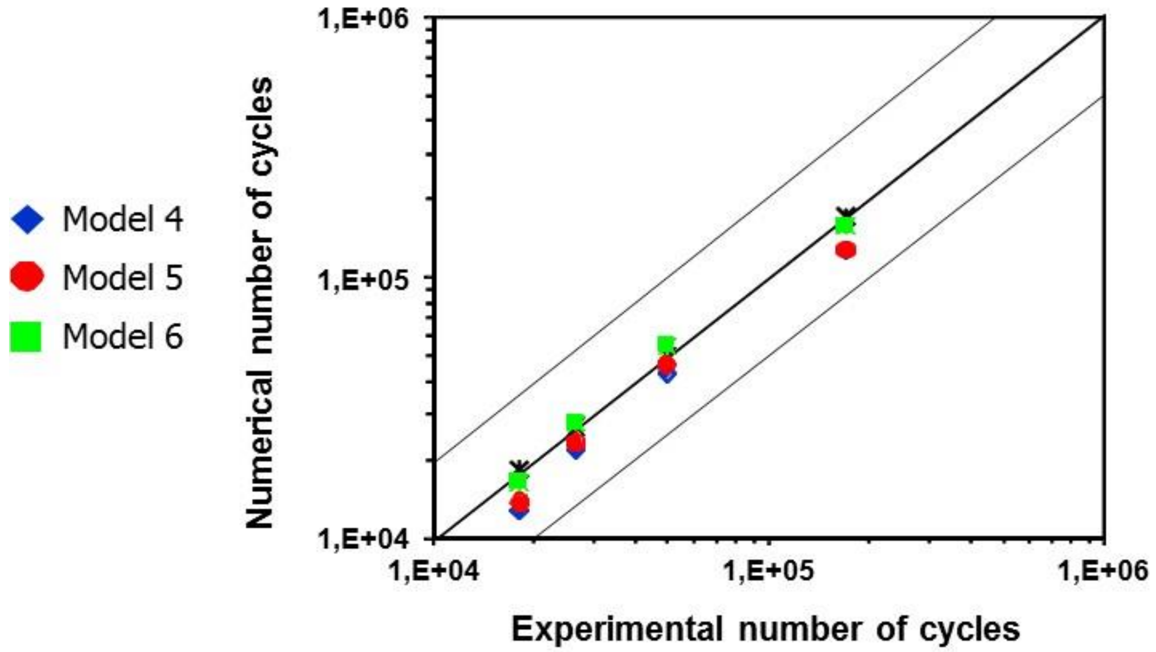


Figure 9. Correlation of numerical-experimental results of welded overlap specimens

After development and validation, the numerical technique for specimen level fatigue life prediction in welded joints and structural components was applied for predicting fatigue life of a structural part[18] of a semi-trailer. Specifically, an axle support structure of a semitrailer was analysed, because it is a critical area in this type of vehicle against loads generated by the minimum turning radius manoeuvre according to fatigue life criteria. In this case, numerical and experimental results also showed a high correlation, which permitted the study to proceed to the next level, namely the analysis of a complete semi-trailer.

In order to validate the experimental technique for fatigue life prediction, a fatigue experimental test was carried out on a semi-trailer prototype. This test is shown in Figure 5. The duration of the fatigue test was two months. During the experimental test, the semi-trailer was subjected to repeated cycles of fatigue, which consisted of a combination of horizontal displacements in the King-pin and vertical displacements in the wheels. Vertical displacements were introduced in the first and third axle, which generate a high torsion in the semi-trailer structure. The objective of this manoeuvre is to force the premature fatigue failure in welded joints of the axle support structures and the section change of the longitudinal beams. Figure 5 shows the vertical displacements in the wheels and the horizontal displacement in the King-pin.

In collaboration with the engineers of Lecitrailer, the research group have considered that 20.000 cycles of this fatigue test corresponds to the typical lifespan ofa semi-trailer, which is six years or 1.5 million kilometres. The tested prototype exceeded 20,000 cycles without failure.

A numerical model of the tested semi-trailer was developed and simulated by means of the FEM. The objective of this phase was to analyze the numerical model of semi-trailer within the same boundaries and loads of the experimental test. In this way, the obtained numerical results can be compared with the experimental results in terms of stress, strain and fatigue life.

In a numerical model of a complete semi-trailer is not viable the discretization of weld beads and thermally affected areas by the welding process. In order to analyze in detail the critical areas of the numerical model, sub-models were created in which weld beads and thermally affected areas were modelled. Figure 10 shows the complete model of semi-

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trailer and the sub-model of the section change of the longitudinal beams, as well as the fatigue life prediction obtained from these models.

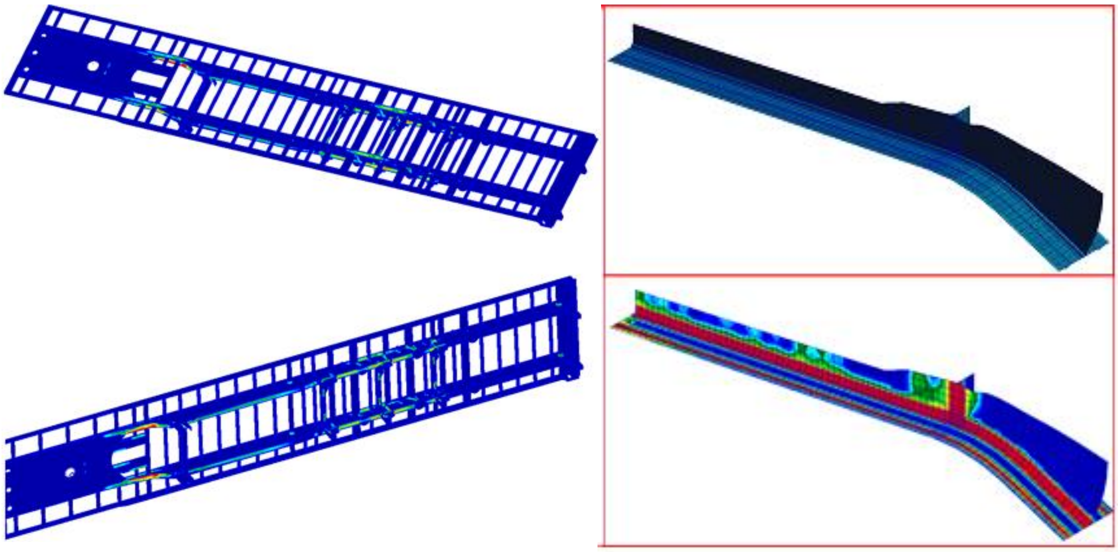

Figure 10. Fatigue life prediction obtained from the complete semi-trailer model (30,000 cycles) and from the sub-model of the section change of the longitudinal beam (21,360 cycles).

The results obtained from the detailed models (sub-models) showed a fatigue life for the semi-trailer of 21,360 cycles. Numerical results showed that the most critical area according to fatigue criteria was the section change of the longitudinal beams. The numerical result of fatigue life is consistent with the experimental results of fatigue life. The experimental test was stopped after passing 20000 cycles without fatigue failure

8. CONCLUSIONS

This article summarises the result of the collaboration between the University of Zaragoza, the Polytechnic University of Madrid and the University of Oviedo with the company Lecitrailer S.A.
The main result of this collaboration is the development of a new optimization procedure for semi-trailers, which permits the design of lighter yet sufficiently rigid and resistant vehicles against static and dynamic loads. Furthermore, the developed research has preferably focused on optimising the resistance of vehicles against fatigue loads. For this reason areas affected by the welding process have been studied in detail, because they are the most stressed areas and therefore the areas in which the fatigue crack occurs

The method developed fordesign optimisation of semi-trailers consists in the combination of numerical simulation tools based on FEM with the application of experimental techniques in prototypes. In order to improve the experimental part of the method, a fatigue test bench for semi-trailers has been developed. This test bench, a world-first, allows the testing of different road profiles and critical manoeuvres for semi-trailers under perfectly controlled and reproducible laboratory conditions.

In order to ascertain the fatigue resistance of semi-trailers, different types and models of semi-trailer have been tested using critical manoeuvres, such as minimum turning radius and critical road profile. Furthermore, in order to establish the degree of correlation between the experimental results obtained in the bench test with those obtained numerically, numerical models were developed and calculated. These calculations were carried out with the same loading conditions as those applied in the experimental tests. The comparison between both types of analysis, numerical and experimental, provided a strong correlation of results, which permits the validation of the developed numerical-experimental technique.

In the study, the research group has worked in the development of different critical cycles for fatigue tests for semi-trailers which can generate the fatigue failure , and therefore are of great interest to the sector.

Reliability and accuracy of existing scientific methods for fatigue life prediction have been analysed for the case of the base material and welded joints. As result, a subroutine for fatigue life prediction has been implemented to be applied in semi-trailers. This subroutine is based on existing methods for low cycle fatigue life. The methods applied for prediction are based on numerical-experimental correlations of results obtained from laboratory specimens and full-scale vehicles. The numerical results obtained with the subroutine developed have achieved a high correlation with the experimental results of fatigue life in welded joints in semi-trailers.

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