Numerical and Experimental Random Spectral Analysis of a Condensing Unit Discharge Tube

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ABSTRACT

Transportation is a factor of paramount importance on the life cycle of manufactured goods. Components are in constant movement in the production line, the product is transported to one or more distributors and then to the final customer. During transport, all components are subjected to vibration. Such vibrations depend on several factors, like transportation method (Truck, Train, Air, …), vehicle suspension type (Spring, Air, ...), road and rails quality. Due to the unknown nature of all factors involved, these vibrations are usually assumed to depict random characteristics, resulting in a somewhat random stress in the components. Most methods to evaluate fatigue are based on the load history, making them unsuitable for the product damage estimation during transportation. In this paper a frequency domain approach is used to estimate the fatigue damage in these situations. The random nature of the load is transformed to the frequency domain using power spectrum density (PSD) and a probability density function (PDF) for the resulting stress. Afterwards, with a common fatigue curve and the stress PDF the component damage is calculated. The methodology has been used to evaluate the transportation effects on condensing unit tubes based on fatigue damage.

1. INTRODUCTION

Transportation is an intrinsic part of a product life cycle. Truck transportation represents a significant portion of all products transport. There are several reasons to assume a random behavior to model truck movement: road quality, truck suspension, tire pressure, etc. A traditional fatigue analysis, based on harmonic load is not possible due to the randomness of the load.

A rainflow counting is a solution, but it depends on the stress time series on the components weak spot. Knowing the weakest spot is not trivial due to the dynamic characteristic of most products, besides, instrumentation in this specific spots may lead to unreasonable difficulties or unreliable data. Even if it is possible to easily collect experimental data on the weakest spot, evaluate different geometries leads to another difficulties, like time to fabricate the prototypes and new instrumentation for each geometry (BOSCO, 2007).

All mentioned experimental difficulties are eliminated with a numeric model. If a truck movement randomness is captured in a long acceleration time series it could be used as input to the numeric model. This approach solves all difficulties at the cost of elevated computational time. To capture the truck movement randomness minutes, maybe hours, of time data needs to be used as simulation input, resulting in unfeasible time to solve the models. The last problem mentioned, computational time, is solved when all evaluations are made in the frequency domain (BISHOP, 1999). To do so, the acceleration time series is converted to the frequency domain in the power spectrum density (PSD) form and the numeric model is converted to the frequency domain with a modal superposition. The only drawback of this method is the modal superposition characteristics, limitating the numerical model to linear behavior.

For a condensing unit, besides component movement in production line, the compressor is not always assembled in the production line as the condensing unit, adding another complexity to the evaluation. Failure can happen in any moment before the product is operating in its final destiny, but usually the most significant is the transport between manufacturer and final destiny. To prevent these failures, vibratory tables are used to reproduce the truck movement randomness and products are submitted to these loads during development tests. If this test results in any component failure a more detailed analysis, using the numeric model mentioned, is performed to ensure that the product reliability is guaranteed. Subsequent chapters present the evaluation of a copper tube in a condensing unit that failed in the vibratory table test and the steps used to solve the problem.

2. NUMERICAL MODEL

Numerical model was designed with a random vibration analysis as objective. In Ansys, the random vibration module demands a modal evaluation, because all results are based on a modal superposition. In a modal analysis it is not possible to consider nonlinearities, so all material were assumed to have isotropic linear behavior and all contacts have linear characteristics. As a consequence, accelerations peaks due to impacts are not considered in this model.

Boundary conditions were defined based on the product transportation characteristics. The product is designed with several locations for bolting the condensing unit on the refrigeration system. These bolting regions were used as fixed boundary conditions, and subsequently as input region for the transport vibration. Fixation points are presented in Figure 1.

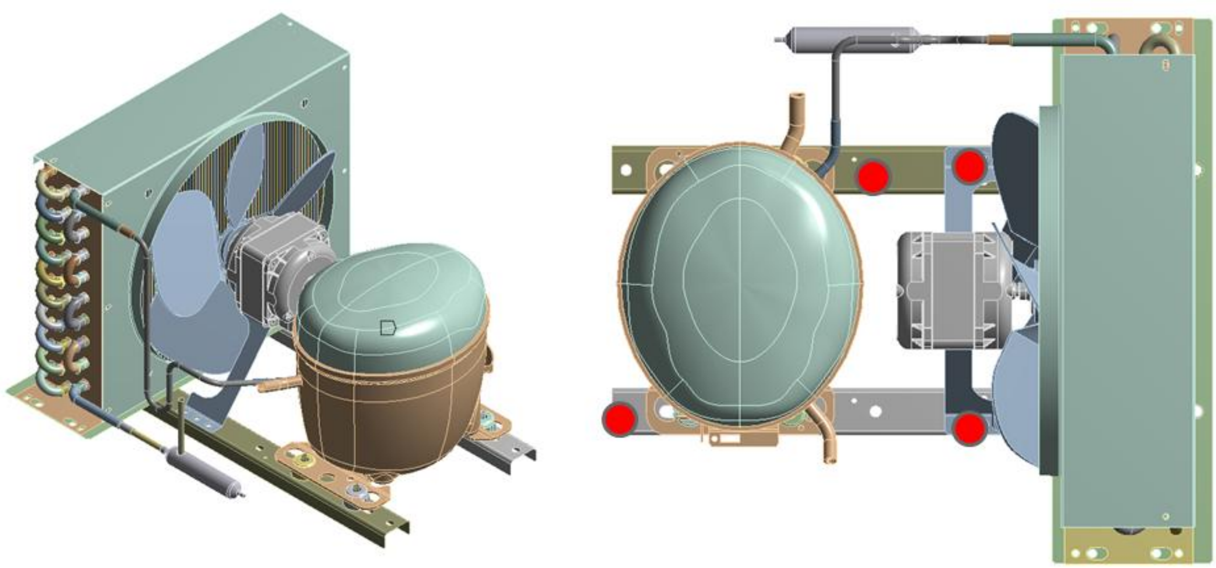


Figure 1: Components used in numeric model on left. Fixation point used as boundary condition on right.

A mesh study was performed, elements type and mesh refinements were evaluated. The most efficient model was modeled with quadratic and linear quadrilateral shell elements and rigid bodies to represent some condensing unit parts. These simplifications reduced the computational time from hours to minutes without compromising the result quality. Optimized mesh is presented in Figure 2

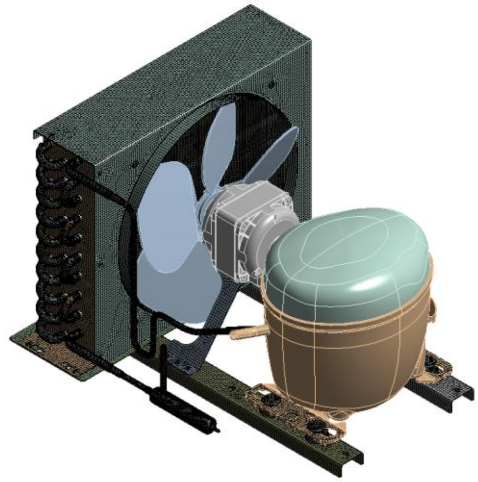
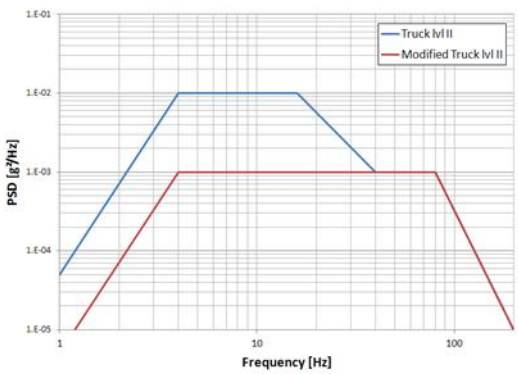


Figure 2: Detail optimized mesh. Shell elements used to represent most elements.

2.1 Random vibration stress results

Four simulations were performed. The differences between models were the excitation spectrum and direction. In the first model, Truck lvl II spectrum from ASTM D4169-09 was used as excitation, on vertical direction (Y axis). Next two models were simulated with a modified acceleration spectrum. This modified spectrum was based on ASTM D4169-09 recommendations and was applied in lateral direction (X and Z axis) one per simulation. The fourth model had all excitations simultaneously. Both acceleration spectrum used are presented in Figure 3.



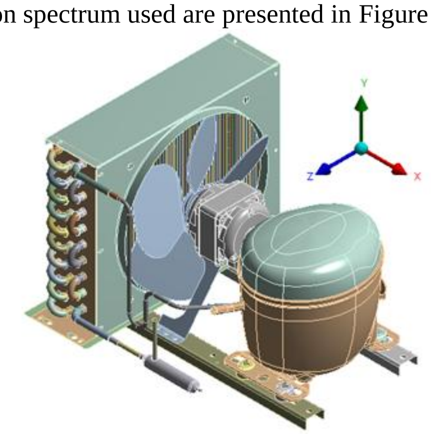


Figure 3: On left both acceleration spectrum used in the numeric models. On right the simulation reference system is presented.

The combined acceleration model is the most representative when compared with the product transportation, equivalent stress values from this model were used to normalize all others models. Equivalent stress was used to evaluate this component, in Figure 4 is presented images from equivalent stress from all four models.

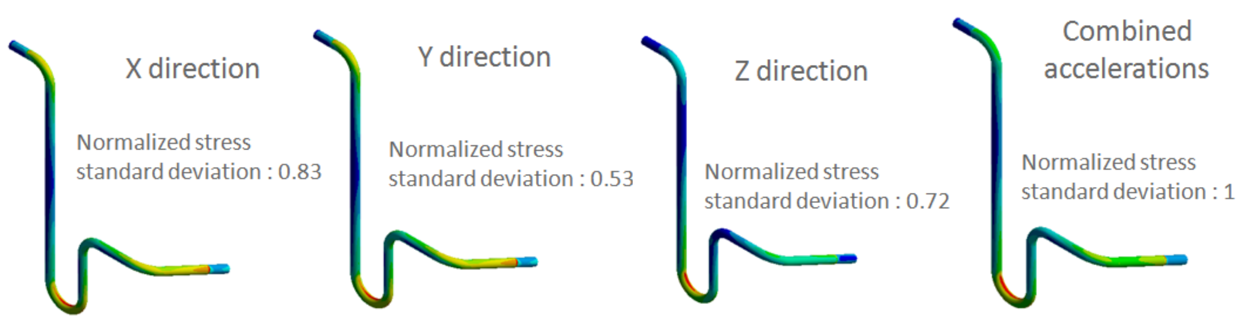


Figure 4: Equivalent stress comparison between different accelerations spectrum and directions.

2.2 Modal analysis results

Simulated stress results showed a significant stress on connecting tubes when acceleration is in X direction, even with lower acceleration spectrum when compared with Y direction. Analyzing the modal results, there is a mode in lower frequency that justifies this behavior. Component stress spectrum also shown a great influence of this particular mode. Besides the preferential displacements direction in the same direction as the base excitation, this mode has a frequency of 21Hz, with is close to the maximum excitation on the Y direction.

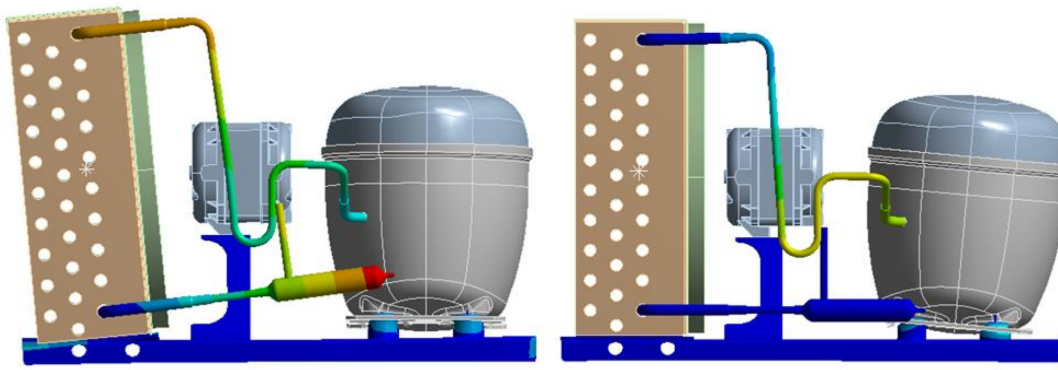


Figure 5: Two most relevant modes. On left the mode related to the condensing unit fixation, on right the mode related to the compressor fixation.

3. EXPERIMENTAL SETUP

An experimental setup was designed to validate the simulation results. A shaker was used to generate the random vibration, fixations were made to duplicate the designed transportation and simulated conditions. Two accelerations spectra were used as input for test, the same ones used in the simulations. In Figure 6 is presented the experimental setup used to validate the simulations.

Figure 6: Experimental setup. On left a test rig with accelerations on Z direction. On right attest rig with accelerations on Y direction

3.1 Numerical and experimental comparison

A strain gage was used to validate the stress results output from the simulation. Experimental output is a stress time series, as shown in Figure 7, the easiest method to compare is using the measurement standard deviation and a direct comparison with simulation results. A comparison between three different simulated model and experimental results is presented in Figure 7.

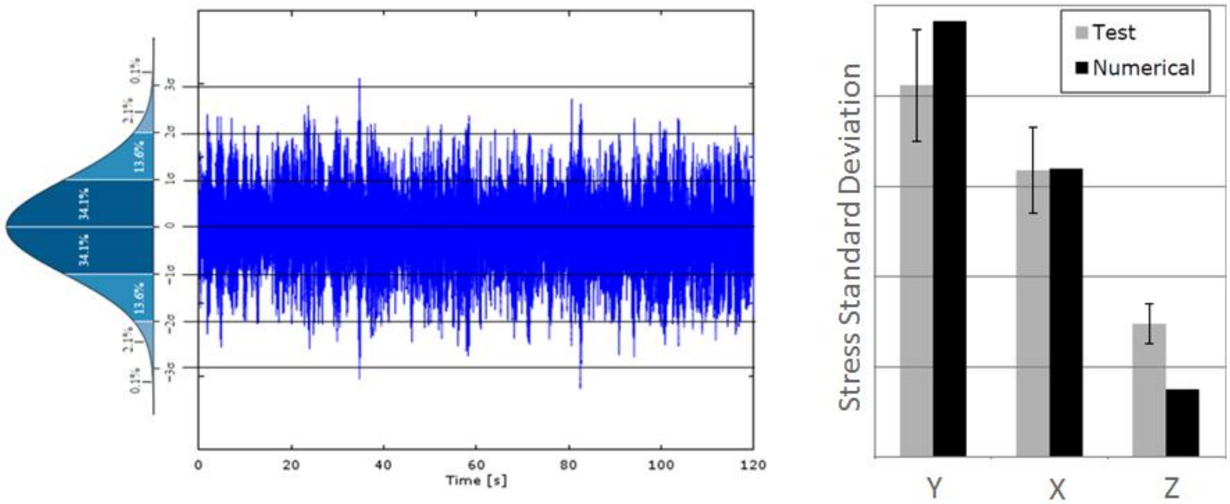


Figure 7: Example of experimental test data on left. Comparison between numeric and experimental stress for different excitations spectrum and amplitudes.

When converting the stress time series to the frequency domain, another effect presented in the simulations was also seen in the test results, the influence of a low frequency mode in the stress. Stress PSD is presented in Figure 8.

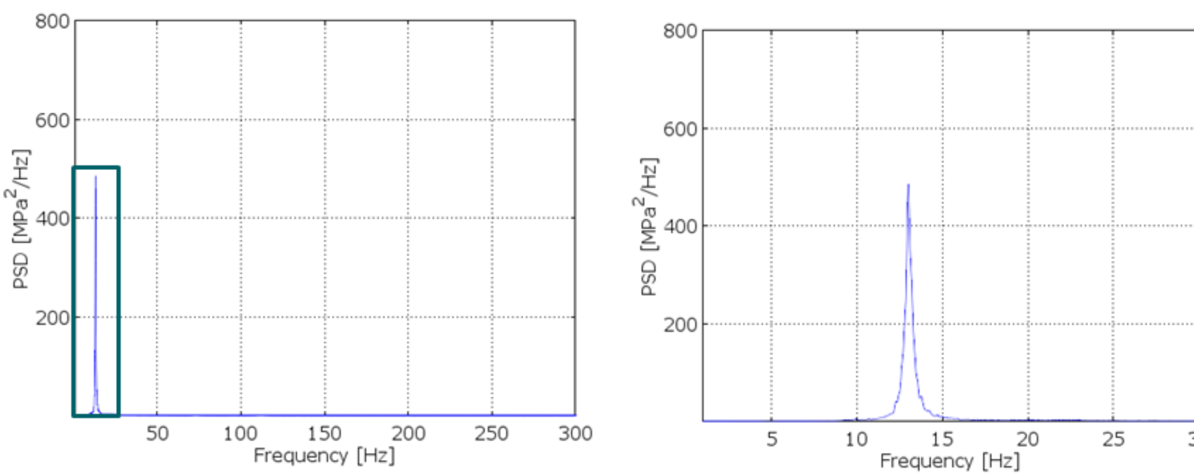


Figure 8: Strain gage time series power spectrum density. Detail on lower frequency on left image.

4. GEOMETRY MODIFICATION

After analyzing all simulated conditions and validating the numeric model with experimental tests, a geometry change was suggested to minimize the stress on the copper connecting tube. The geometry change objective is to change the natural frequency of the mode with frequency close to the peak stress on experimental data. Vibrational mode presented in Figure 5 is the closest to the experimental data, and this mode is related to the condenser base deformation. In proposed geometry ribs were added to the condenser base plate to increase it stiffness. In resulting geometry the same vibrational mode was above 100 Hz. The geometry change in presented in Figure 9.

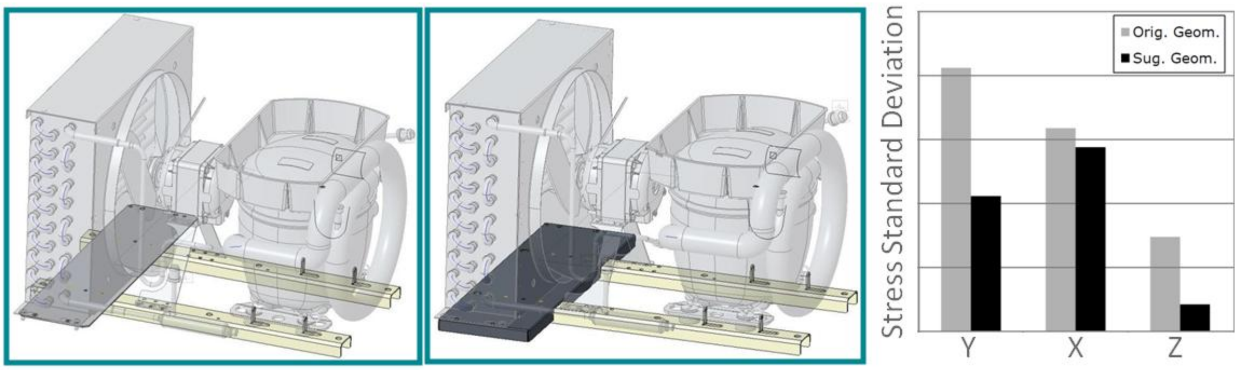


Figure 9: Original geometry on left. Proposed geometry on center and stress reduction on right.

Suggested geometry equivalent stress is presented in Figure 10.

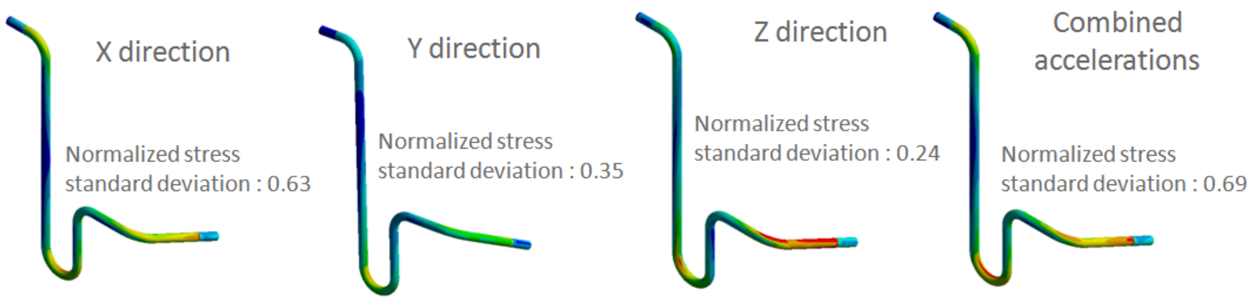


Figure 10 – Suggested geometry equivalent stress

4.1 Fatigue Analysis

Although there is a significant reduction in stress when both geometries are compared, a more detailed analysis is needed to ensure that this reduction is enough to eliminate the component failure risk. To do so, the most common procedure is to evaluate the component cumulated damage. Considering miner's rule for the cumulative damage and Goodman for the mean stress effect, the cumulated damage can be estimated with Equation (1) (Wirsching, 1995) (SOBCZYK and SPENCER 1992):

$$D = \frac{N}{K} \int_{\sigma_m} \int_{\sigma_a} p(\sigma_m, \sigma_a) \left(\frac{\sigma_a}{1 - \frac{\sigma_m}{\sigma_{ut}}} \right)^m d\sigma_m \, d\sigma_a \tag{1}$$

Where D is the cumulated damage. N is the expected number of cycles and it is estimated based on modal analysis and experimental PSD. σ_m is the mean stress. σ_a is the alternating stress. K and m are material properties from the fatigue curve. $p(\sigma_m, \sigma_a)$ is the mean and alternated stress probability density function (PDF).

For this analysis, the mean stress is considered null, this is a reasonable assumption due to the excitation random nature. The PDF can be estimated with a rainflow counting from the time series data, other alternative is using Dirlik's equations to estimate the alternated stress PDF. A simpler method is to approximate the stress PDF to a normal distribution, this is the chosen method due to the simulation results characteristics. For alternating stress there is no difference from positive to negative values, so, a simplified unilateral normal distribution was considered. The simplified PDF is presented in Figure 11. A fatigue curve for this material was obtained by experimental tests and a statistical confidence interval was used to ensure product reliability, the resulting curves are presented in Figure 12. These simplifications were proposed by Steinberg (2000) and are widely used due to it's simplicity.

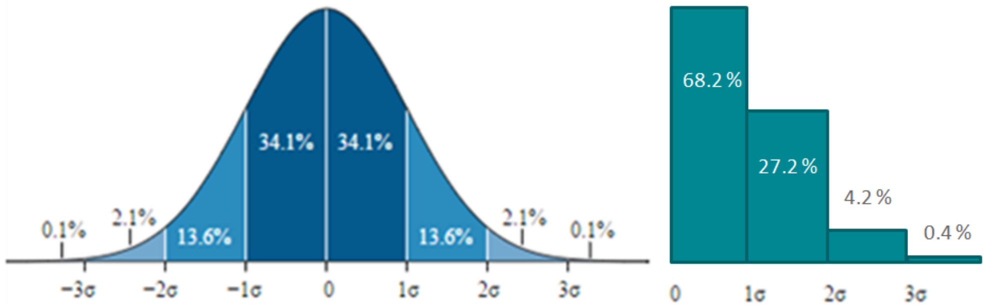


Figure 11: Simplified stress probability density function

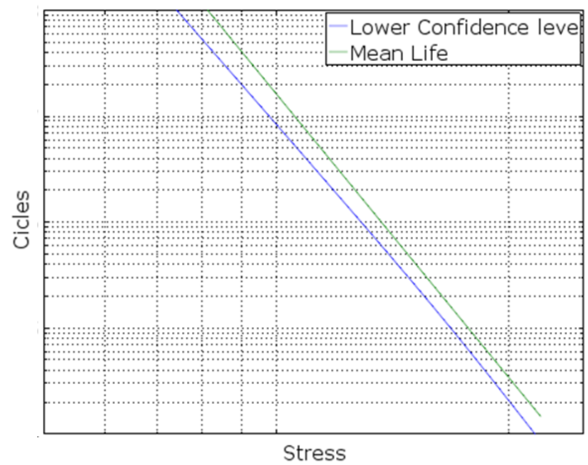


Figure 12: Fatigue curve resulting from experimental tests

With all the information the cumulated damage is calculated to verify if the stress reduction is enough to eliminate the risk. In Table 1 the calculated cumulated damage from both geometries is presented.

Table 1 – Cumulated damage comparison between different geometries.

Stress	Original geometry	Suggested Geometry
1σ	0.1 %	0.0 %
2σ	9.4 %	0.4 %
3σ	52.9 %	1.6 %
4σ	74.3 %	1.9 %
Total Damage	**136.6 %**	**3.9 %**

4.2 Solution Validation

Simulations shown a great increment in components life, so a prototype was manufactured and same experimental procedure presented in Chapter 3 was performed with the new geometry. The objective is to ensure that none of the model's simplification was the reason for the stress reduction. Stress results are presented in Figure 13

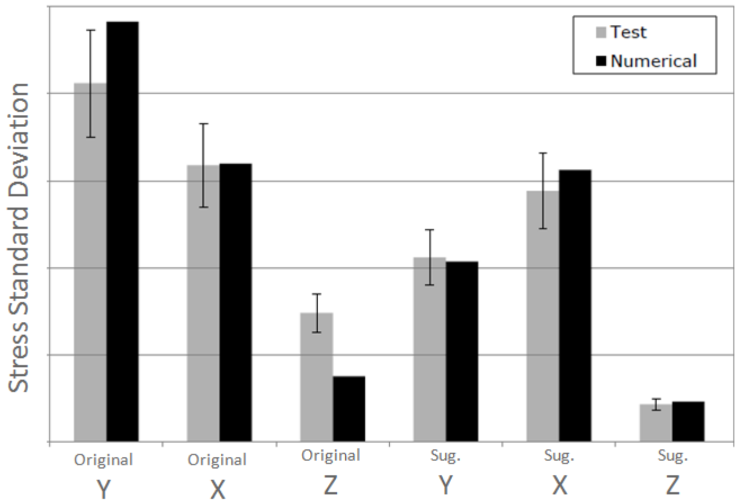


Figure 13: Stress standard deviation results. Comparison between suggests and original geometry, numerical and experimental results.

6. CONCLUSION

Numerical evaluations were essential to increase the product reliability. After the weak spot identification through routine approval tests, a simply modification was able to dramatically increase product reliability. At the end, a product with high safety factor was fast developed and without cost compromising.

The combination of numerical models and experimental data is the best way to ensure reliability without testing hundreds of samples. Numerical model allows to perform experimental instrumentation on any component point, without being limited to the weak spot. This advantage is critical for a good experimentation and ensure reliable experimental data.

In the presented case experimental data correlation with numerical model ensures that no relevant physics was ignored and the calculated safety factor is properly evaluated.

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