A nonlinear stability analysis of the support structure of a particle detector

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Abstract

This paper describes an investigation into the stability performance of a novel type of aluminium support structure for a high-energy physics experiment by utilising a series of elastic nonlinear buckling analyses, and the results are presented. Values for the maximum allowable asymmetric loading and assembly imperfection are also presented. A unified proceedure for evaluating similar structures has been developed.

Keywords: Nonlinear analysis; Stability; Buckling; Aluminium; Large deformations; Assembly imperfections

1. Introduction

A fast and reliable procedure was required for the assessment of the buckling vulnerability of a set of four 24 m diameter, wheel-like aluminium support structures at a stage close to the end of their design. Local failures or total collapses of such structures represent not only immediate concern for loss of life and repair costs, but also major economic impacts due to loss of service. Preliminary studies, based on an early conceptual design, had been performed in the past from Kouzmine et al. [1], but recent advances in the project required more detailed studies. In particular the various installation, loading and service stages that are foreseen during the structure's lifetime need to be assessed.

2. Description of the wheel structure

The structure under study is the first thin-gap gas chamber (TGC1) wheel supporting 216 muon chambers for the ATLAS particle detector. This detector is currently under construction at CERN, close to Geneva, Switzerland, in a cavern located 100 m underground. This wheel and three others create the muon forward regions, which are two vertical structures at the ends of the detector, with outer diameters exceeding 24 m and a total weight of 200 T.

The four wheels will be mechanically connected but, throughout the installation phase and later during their operational life, frequent disconnections will be made for servicing purposes, including a 4 m translation of a single wheel. Therefore it was decided that each of them should be analysed separately. The wheels have similar dimensions and configurations although the TGC1 wheel, having a thickness of only 250 mm and total mass of 49 T, exhibits the highest risk of buckling. Furthermore, it is the wheel closest to the ATLAS magnet and for this reason all electronic racks and corresponding services, with a mass of 5 T, are located on one of its sides creating a notable non-symmetric load. For these reasons it was decided that a fully nonlinear straightforward stability analysis should be employed to investigate the buckling behaviour of this wheel.

The dodecagonal shape of the TGC1, shown in Figs 1a and 1b, is influenced by the layout of the detector's chambers and governed by the physics needs. It is assembled by connecting 12 identical sectors, each supporting 18 muon chambers.

The main structural elements are the outer frame, made from welded square profiles, the suspension frame, connected to the outer frame and used to support the wheel, and the wheel's spokes, made from I-profiles. The centre-hole of the wheel will accommodate the beam pipe containing the two counter-rotating proton beams. The size of the hole is determined by the outer diameter of the shielding surrounding the beam pipe at this location.

Aluminium is virtually the only material used in the structure. This material was chosen for its paramagnetic

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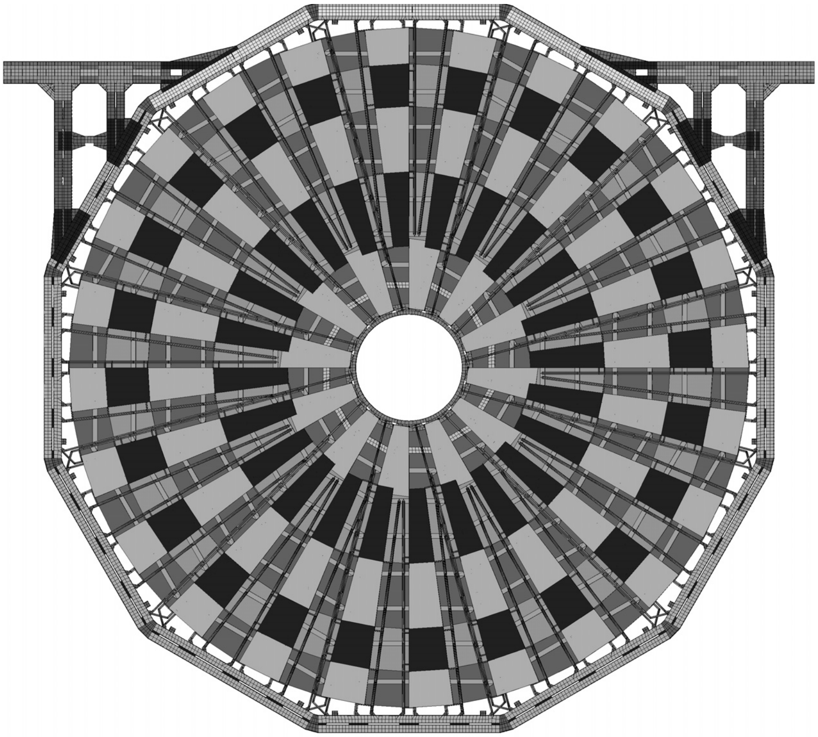

Fig. 1a. FE model of the wheel including the muon chambers.

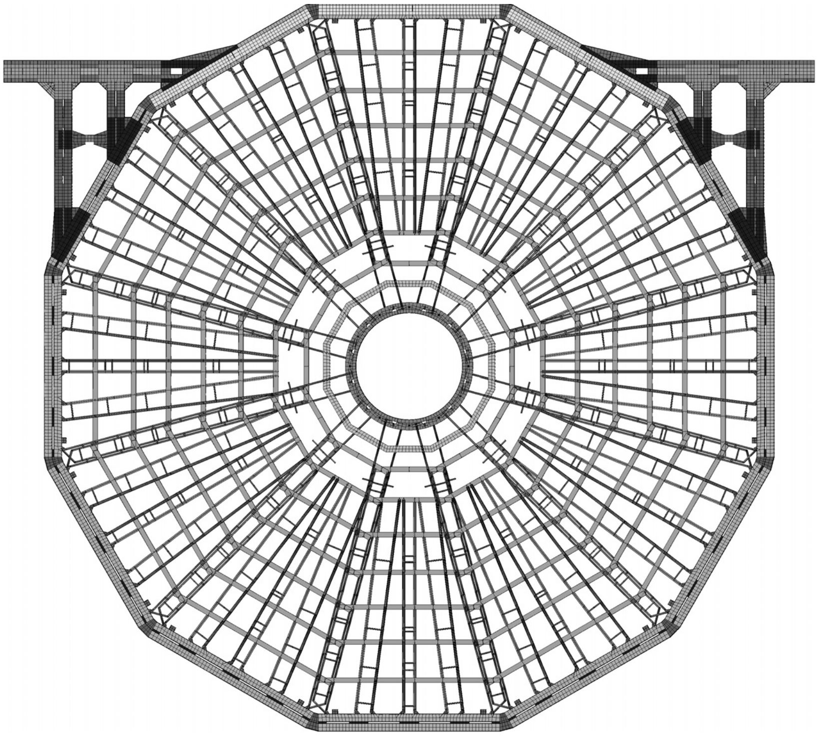

Fig. 1b. FE model of the wheel without the muon chambers.

properties so that the magnetic field induced by the ATLAS superconducting magnet, and needed for the particle momentum measurements, is not distorted.

The wheel is suspended on a chariot, travelling on a rail system and supported by five steel brackets connected to the cavern's concrete walls.

3. Finite element model

Because of the complexities of the geometry and the difficulties associated with predicting the stability behaviour of such structure, a detailed three-dimensional finite element model was employed. During the modelling stage special care was taken to create a model which was as representative as possible and with good quality elements, especially in the zones where buckling was anticipated.

The model is made of 23 000 4-node, linear shell elements with large displacement capabilities and 5000 uniaxial beam elements. For simplicity, certain members, not participating in the structural stiffness, have been represented by point-mass elements. Each muon chamber has also been represented by a single shell element with the corresponding thickness and mass but with decreased stiffness, due to their isostatic support to the wheel.

The material used is aluminium alloy 6082 with modulus of elasticity $E = 70\,000$ MPa, Poisson ratio $\nu = 0.33$ and mass density $\rho = 2710\,\mathrm{kg/m^{-3}}$. Gravity is considered as the only load on the structure. The total mass of the wheel is 49 T of which 29 T are due to the support structure and 20 T are due to the muon chambers.

Preliminary analyses had shown that the stiffness of the wheel support system is much greater than the wheel stiffness, therefore for the stability analysis it was decided that, instead of modelling it, vertical constraints would be implemented at each end of the suspension frame.

To study the validity of the model, analysis verification studies have been performed and the results examined for any irregularities. During the analysis phase the results show that the structural stresses are always below the elastic limit of the material and no over-distorted elements have been detected. Based on the evaluation of these results, one can be confident that the model will predict the buckling behaviour of the structure with a significant degree of accuracy.

4. Stability analysis technique

Even though there is no specific need to trace the load-deflection curve through regions of 'snap-through' and 'snap-back' response for this specific structure, the analysis technique deployed here is based on the arc-length method for a large deformation analysis. This is due to the fact that this method is suitable for nonlinear static equilibrium solutions of unstable problems [2].

Preliminary studies had shown that the application of a small out-of-plane perturbation force, usually required for this type of analysis to start to show a buckling response, is not needed here because, as noted earlier, the wheel is not symmetrically loaded.

Merely establishing that a structure is stable at a given load level is generally insufficient for most design practices. Therefore, one should carry the stability analysis through to the point of identifying the critical load required in order to calculate the structure's safety factor with respect to buckling.

Here the nominal load was scaled up four times and applied in small increments starting from zero, while the deformations at the middle of the suspension frame were continuously monitored. This technique was repeated for different configurations of the wheel geometry and restraints.

5. Results

The results presented in Fig. 2 show surprisingly low stability performance of the wheel. This could be explained by the fact that the suspension frame is subjected to a compressional force of 100 kN at nominal loading, and by taking into account the fact that it is a square aluminium profile of 240 mm × 680 mm × 12 mm and is 25 600 mm long making the entire wheel very unstable.

It is also worth noting that a bifurcation point should not occur before the nominal load is scaled up at least three times, which is a condition not satisfied by the original design of the structure. The most significant buckling mode is shown in Fig. 3.

The impact of the asymmetrical loading of the structure on its stability behaviour was further investigated. The results showed that the maximum allowable mass is 4 T with an offset no greater than 75 mm.

The model also appears to be very delicate to small restraint variations, and their effect on the stability behaviour has been further studied in order to guarantee the most realistic representation in it.

Finally, the sensitivity of the suspension frame assembly tolerances was investigated. From Fig. 2 it can be noted that any misalignment of more than 20 mm jeopardises the stability of the wheel. This fact imposes very stringent requirements and gives a clear idea of the precision to be achieved during the assembly stage of the structure.

To overcome these issues it has been decided to

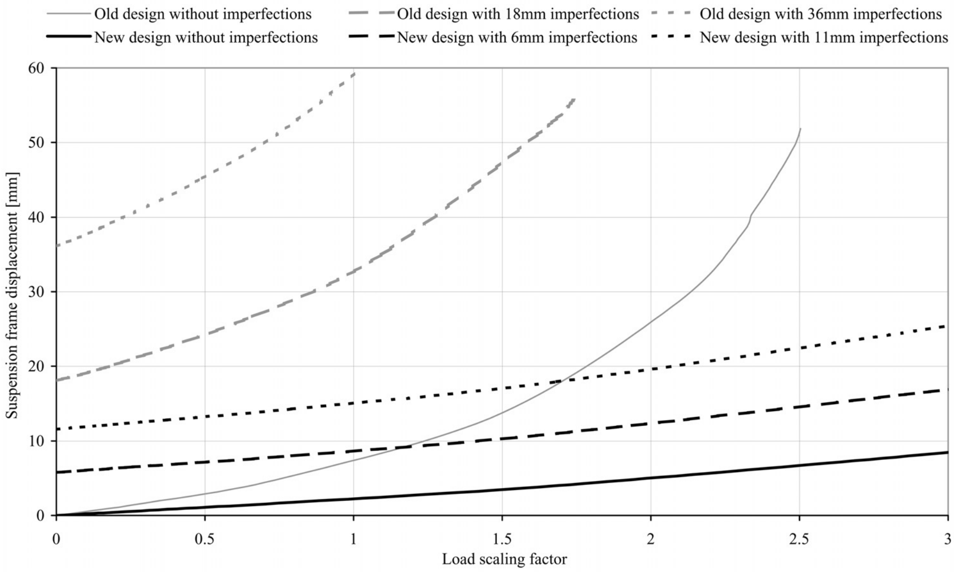

Fig. 2. Load/displacement stability curves.

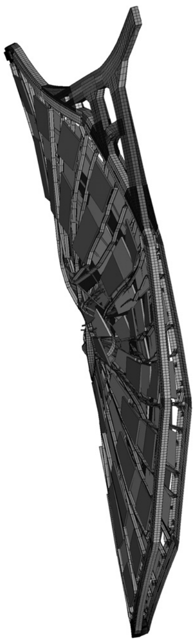

Fig. 3. The buckling mode.

increase the initial lateral inertia of the suspension frame by means of additional permanent and temporary profiles. Also, certain elements close to the wheel centre have been connected together and additional stiffening has been implemented on the suspension frame corners.

One can also deduce from Fig. 2 that these measures have proven to be sufficient to increase the global stability of the structure when the buckling curves before and after the design changes are compared.

6. Conclusions

In this study a series of nonlinear stability analyses were carried out on an aluminium support structure with an unusual configuration by using a detailed, purpose-built, three-dimensional, finite element model.

Stability vulnerability has been discovered in the structure when it is disconnected and moved for servicing. Design changes have been implemented and verified. Values for the maximum allowable asymmetric loading and assembly imperfections are also presented.

The finite element model was later used to develop a fast and reliable, unified procedure for stability assessment of similar structures.

References

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