Engineering Structures 87 (2015) 58–69

Contents lists available at ScienceDirect

Engineering Structures

journal homepage: www.elsevier.com/locate/engstruct

Review article

Finite element modeling of burst failure in unbonded flexible risers

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ARTICLE INFO

Article history:
Received 21 March 2014
Revised 13 January 2015
Accepted 14 January 2015
Available online 31 January 2015

Keywords:
Flexible riser
Burst failure analysis
Axisymmetric analysis
Finite element modeling

ABSTRACT

Flexible risers can be subjected to high levels of internal pressure caused by hydrocarbon pressure gradients coming from the reservoir. Therefore, flexible risers can failure by burst of the riser, caused by a pressure armor failure, which were designed to resist the internal pressure or radial loads. This paper presents a burst analysis methodology by performing analytical, 2D and 3D finite element modeling of a section of a flexible riser. In order to obtain reliable results, it was necessary to consider in the finite element modeling both, geometric and physical nonlinearities due to the contact conditions between the two different layers of the riser considered in burst analysis. The results were compared with various existing analytical models with good correlation, and also showed the importance to develop finite element models for a proper burst analysis of flexible risers. The methodology was applied to a flexible riser manufactured with pressure armor with a "*Z*" profile.

Contents

1. Introduction

Unbonded flexible risers are used to transport oil and gas from the submarine equipment to floating offshore installations, in shallow and deep waters, under large movement's conditions. Flexible risers are manufactured with several layers designed to resist specific loads. The number and layers type depend on the hydrocarbon conditions and the loads to which the riser may be exposed at the site of operation (tension, bending, twisting, etc.). A flexible riser is fabricated with pressure armor, which is designed to sustain the internal pressure. But in order to ensure structural integrity during the lifetime of the riser, it is necessary to analyze the possible failure mechanisms in each layer of the riser so thereby avoid production losses, environmental damage and economic losses. In this sense, burst is one of the most important failure mechanisms in flexible risers [1]. Bursting can occurs when the pipe is subjected to high levels of internal pressure in the bore of the pipeline caused from the oil coming from the well. Therefore, in case of flexible riser failure, it has to be replaced, which is no desirable in the exploitation of a hydrocarbon field, this shows the necessity to develop a methodology to predict burst in flexible risers.

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http://dx.doi.org/10.1016/j.engstruct.2015.01.021
0141-0296/

Nomenclature

σ_θ	hoop stress	N	total number of layers
p	internal pressure	σ_i	stress due to the load
r_m	average radius of the pressure armor	A_i	cross-sectional area of the geometry
t	wall thickness	α	winding angle
ψ	super/pitch	R_{int}, R_{ext}	inner and outer radius of the layer
I_{Gmin}	minimum moment of inertia of the cross section of the housing	P_{int}, P_{ext}	internal and external pressure of the layer
b	total length of the housing profile	R_i	average radius of the litter
Super	overlap distance between the profiles of the housing	w_i	length of each tendon
Pitch	winding pitch profile of the housing	θ	winding angle
n_i	number of layers	F_f	filling fraction
		ε_c	deformation in the axial direction

Production wells may have significant variations of the produced hydrocarbons, which can be transmitted to the riser, and the pressure gradients can be significant for burst to occurs, especially when the reservoir is characterized as high pressure well, pressures greater than 10,000 psi [2]. Variations of the internal pressure can also induce stresses in the tensile armors [3]. Hydrocarbon fluid passes through the carcass until the internal polymer sheath preventing its permeation to the annular region where the pressure and tensile armors are located. However, due to the characteristics of the material used in the manufacture of the internal polymer sheath, it do not provide structural strength against, hence the layer providing structural resistance to internal pressure is the armor pressure, the polymer sheath transmits the load of the internal pressure to the pressure armor. For the case of very high levels of pressure, burst can be caused and consequently tension reinforcement [4]. Because there have been a number of failure cases, some improvements have been made in the design of the risers [2]. It is noteworthy that despite the current state of the design of flexible risers considering the pressure vs inner diameter of 80,000 psi-in. [2], layers failure may be presented. In this sense some works have been developed related to high internal pressure loads (internal overpressure) for different types of risers [5–8]. Therefore, the pressure armor must be designed to withstand the loads of internal pressure and thus ensure that the pipeline will not failure when high pressure values are presented [9,10]. According to the above, this paper presents a comparison of existing analytical models in the literature with finite element modeling, the finite element models considered the following: (1) one axisymmetric study (with a 2D model) taking into account that the profile of the pressure armor has a pitch near 90° for its characteristic cylindrical shape, (2) a 3D model that considers the interrelation between the winding pitch of the internal pressure armor.

2. Models for burst analysis due to internal pressure

To the date, there have been published a few works focused in burst analysis of flexible risers, thus both numerical and analytical models are limited and almost exclusive of the use of riser's manufacturers. Bournazel and Feret [11] proposed an axisymmetric load analysis in order to determine the pressure loads acting on the flexible riser. The authors proposed an equilibrium equation between the tension and the external and internal pressures in the radial direction, which allows estimating the number of layers needed to prevent burst. Furthermore, Oliviera et al. [12] proposed an analytical model to evaluate the burst strength of flexible riser considering its axial and circumferential deformation. Another analytical approach used by Wellstream Corporation [13], which in order to describe the behavior of the pressure used the Barlow's formula for thin-walled cylinders. This method also considers an added factor so called "filling fraction" defined by the effective area of the pressure armor with a "Z" profile with the total square area of the profile. Neto et al. [9,10] proposed a linear and a non-linear analytical formulation, in the linear model assumed that the pressure armor can be represented as a thin-walled cylinder, considering a thickness equivalent of the "Z" section, with reasonably results. The nonlinear model employed by Neto et al. [9,10] considered non-linearities of both, the material and pressure armor geometry. The non-linearity material considered a bi-linear material represented by the Young's modulus and the tangent modulus model. To represent the geometric nonlinearity, the analytical model considers a radial displacement and thickness variation according to the internal pressure load. However, in previous analytical models the geometry section is represented in a simple form, and also it did not consider the pitch, contact and the winding pitch of the pressure armor, which is questionable. In this paper the linear analytical formulations are considered to compare with numerical results.

3. Analytical models

To perform the analysis, it was considered four different linear analytical approaches, which represented the riser as a model of thin-walled cylinder. Neto et al. [9,10] proposed a linear analytical formulation, assuming that the pressure armor has a relationship $\frac{D}{t} > 10$, where D and t represents the outer diameter and wall thickness respectively, behave exactly as a thin-walled cylinder. Therefore, the authors suggested that the hoop stress can be represented by:

$$\sigma_\theta = \frac{pr_m}{t} \tag{1}$$

where:

σ_θ = Hoop stress.
p = Internal pressure.
r_m = Average radius of the pressure armor.
t = Wall thickness.

Which was derived from Hooke's law and applied to thin-walled cylinders in plane stress state conditions, the radial displacements can be evaluated with the following expression:

$$u_r = \frac{pr_m^2}{tE} \tag{2}$$

where E is Young's modulus, because this model does not considers the geometry section and wrap angle of the pressure armor, for this reason the authors defined an equivalent thickness of the "Z"