



Department of Mechanical and Mechatronics Engineering

WKRPT 400
Numerical Analysis and Optimization



A report prepared for:
Altaeros Energies
Somerville, Massachusetts, United States

by:
Sebastien Blanchet
3B Mechanical Engineering

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Altaeros Energies
24 Park Street, Unit 10
Somerville, Massachusetts, United States, 02143

Professor Michael Collins
Associate Chair Undergraduate Studies, Mechanical Engineering
University of Waterloo
Waterloo, Ontario, Canada, N2L 3G1

Dear Professor Collins,

Prepared for my 3B work term at ***Altaeros Energies*** , my third and final work term report is entitled ***Numerical Analysis and Optimization***. The objective of this work term report was to perform an in depth structural analysis on the current aerostat's winch system.

Altaeros Energies has one simple mission, to deliver telecommunication in remote areas with increased reliability and more cost effectively. Another major selling point is fully autonomy led by state of the art controls.

As a systems engineering intern under the supervision of Ephraim Lanford, I was primarily responsible for conducting advanced analyses on the aerostat's tether management system.

This report was written entirely by me and has not received any previous academic credit at this or any other institution. I would like to thank Ephraim Lanford for all guidance. Also, John Umina for his expertise in the design of pressure vessels. No other sources of aid were used for this report.

Best Regards,

Sebastien Blanchet
ID:
3B Mechanical Engineering

Signature

Date

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Summary

The objective of this report is

Preliminary results found that the method was not good.

Results

Conclusions and recommendations

Section 1

Introduction

This is the introduction, why we are doing this analysis

1.1 Background

Altaeros Energies does this as per Figure 1.1 seen below.



Figure 1.1: Overview of the buoyant airborne technology

1.2 Purpose

The purpose of this report is to do this.

- Do analysis on drum
- Development numerical algorithm for solving

1.3 Scope

The scope of the report will be as follows.

Section 2

Preliminary Analysis

In this section, the methods used for the numerical approach are discussed in detail.

2.1 Basics: Deflection

The first equation that is presented is the main relation between the internal bending moment on a beam and its curvature by Equation 2.1 as per [1].

$$\frac{1}{\rho} = \frac{M}{EI} \quad (2.1)$$

Where $\frac{1}{\rho}$ from mathematics is the radius of curvature defined as 2.2

$$\frac{1}{\rho} = \frac{d^2y/dx^2}{(1 + (dy/dx)^2)^{3/2}} \approx \frac{d^2y}{dx^2} \quad (2.2)$$

2.1.1 Relations

With 2.2, relations based on deflection $y(x)$ for slope 2.3, moment 2.4, shearing force 2.5 and load intensity 2.6 as per [1].

$$\theta(x) = \frac{dy}{dx} \quad (2.3)$$

$$M(x) = EI \frac{d^2y}{dx^2} \quad (2.4)$$

$$V(x) = EI \frac{d^3 y}{dx^3} \quad (2.5)$$

$$q(x) = EI \frac{d^4 y}{dx^4} \quad (2.6)$$

2.1.2 Boundary Conditions

From these above equations, we may present boundary conditions as follows assuming that the boundary in question is located at x_0 .

Free Ends:

$$\begin{aligned} y(x_0) &= y_0 \\ \theta(x_0) &= \theta_0 \\ M(x_0) &= 0 \\ V(x_0) &= 0 \end{aligned} \quad (2.7)$$

End Supported:

$$\begin{aligned} y(x_0) &= y_0 \\ \theta(x_0) &= \theta_0 \\ M(x_0) &= M_0 \\ V(x_0) &= V_0 \end{aligned} \quad (2.8)$$

Fixed:

$$\begin{aligned} y(x_0) &= 0 \\ \theta(x_0) &= 0 \\ M(x_0) &= M_0 \\ V(x_0) &= V_0 \end{aligned} \quad (2.9)$$

In the following section, these preliminary relations will be used to further expand on more complex analysis approaches.

2.2 Theory of Cylindrical Shells

As per Timoshenko's book [2], this section will cover the method of approximating the cylinder as a long thin shell. The following coordinate system is presented as per Figure 2.1 below.

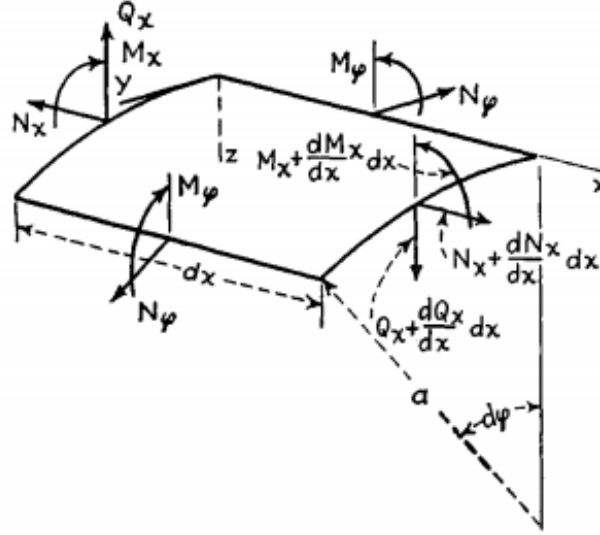


Figure 2.1: Coordinate system adopted for derivation of DEs.

Based on this, the following differential equations are presented as a pressure balance knowing that the differential area can be represented as Equation 2.10

$$dA = dS \, dx = a \, d\varphi \, dx \quad (2.10)$$

The equations of equilibrium may be written as a force projection about the x and z axis and moment balance about y as Equations 2.11, 2.12, 2.13 respectively.

$$\frac{dN_x}{dx} a \, d\varphi \, dx = 0 \quad (2.11)$$

$$\frac{dQ_x}{dx} a \, d\varphi \, dx + N_\varphi a \, d\varphi \, dx + Z a \, d\varphi \, dx = 0 \quad (2.12)$$

$$\frac{dM_x}{dx} a \, d\varphi \, dx - Q_x a \, d\varphi \, dx = 0 \quad (2.13)$$

First looking at 2.11, simplifying this equation and taking the integral with respect to x will leave 2.14.

$$N_x = C = 0 \quad (2.14)$$

This above equation simply states that the effects of bending due to the axial forces will be neglected.

Similarly with 2.12 and 2.13, simplifications will lead to 2.15 and 2.16 respectively.

$$\frac{dQ_x}{dx} + \frac{1}{a} N_\varphi = -Z \quad (2.15)$$

$$\frac{dM_x}{dx} - Q_x = 0 \quad (2.16)$$

With these equations and using strain relations from Hooke's law, the fine DE for displacement will be determined. In Equation 2.17, the relations between displacement and strain are presented.

$$\begin{aligned} \epsilon_x &= \frac{du}{dx} \\ \epsilon_\varphi &= -\frac{w}{a} \end{aligned} \quad (2.17)$$

From Hooke's law N_x may be also written as Equation 2.18. Substituting 2.17 will leave the final simplification.

$$N_x = \frac{Eh}{1 - \nu^2} (\epsilon_x + \nu\epsilon_\varphi) = \frac{Eh}{1 - \nu^2} \left(\frac{du}{dx} - \nu \frac{w}{a} \right) \quad (2.18)$$

Solving Equation 2.18 using 2.14 leaves 2.19.

$$\frac{du}{dx} = \nu \frac{w}{a} \quad (2.19)$$

Similarly, with N_φ , again applying 2.17 leaves 2.21.

$$N_\varphi = \frac{Eh}{1 - \nu^2} (\epsilon_\varphi + \nu\epsilon_x) = \frac{Eh}{1 - \nu^2} \left(-\frac{w}{a} + \nu \frac{du}{dx} \right) \quad (2.20)$$

$$N_\varphi = -\frac{Ehw}{a} \quad (2.21)$$

As a result of no change in curvature in the φ direction, we know that $\frac{dM_\varphi}{d\varphi} = 0$ hence no change in the circumferential moments. This relation is translated to axial moments M_x with 2.22.

$$\begin{aligned} M_\varphi &= \nu M_x \\ M_x &= -D \frac{d^2 w}{dx^2} \end{aligned} \quad (2.22)$$

Where D is the flexural rigidity of the shell 2.23.

$$D = \frac{Eh^3}{12(1 - \nu^2)} \quad (2.23)$$

Simplifying 2.15 to get $Q_x = \frac{dM_x}{dx}$, the following is put in 2.16 to get 2.24

$$\begin{aligned} \frac{d^2 M_x}{dx^2} + \frac{1}{a} N_\varphi &= -Z \\ D \frac{d^4 w}{dx^4} + \frac{Eh}{a^2} w &= Z \\ \frac{d^4 w}{dx^4} + \beta^4 w &= \frac{Z}{D} \end{aligned} \quad (2.24)$$

Where β^4 is some parameter defined as 2.25.

$$\beta^4 = \frac{Eh}{4a^2 D} = \frac{3(1 - \nu^2)}{a^2 h^2} \quad (2.25)$$

The solution to this common fourth order, linear, non-homogeneous, ordinary differential equation in 2.24 has is known to be Equation 2.26

$$w(x) = e^{\beta x} (C_1 \cos \beta x + C_2 \sin \beta x) + e^{-\beta x} (C_3 \cos \beta x + C_4 \sin \beta x) + f(x) \quad (2.26)$$

Where C_1, C_2, C_3, C_4 are integration constants to be solved based on BCs and $f(x)$ is the particular solution to the ODE.

In the following section, the solution to this differential equation will be explored to

Section 3

Discussion

Section 4

Conclusion

In conclusion, ODI must improve the DBM's existing MBA. The 100: 1 reduction Mijno MNT-115-100 gearbox (GB) and Allen Bradley (AB) N-4220-2-H00AA MBS are highly outdated.

Multiple high by 2. thick steel rule samples were bent 45 and from this current, speed and time data were collected. These data were converted to torques. Upon calculation of safety factors (SF) with maximum calculated nominal 280 and acceleration 345.2 torques both the MBS and GB failed SF the 1.25 SF requirement.

An ODI supplier for PLC control systems, Brock Solutions was consulted and the new MBS was selected to be an AB MPM-A1153F-MJ72AA, rated for 6. nominal, 19. acceleration torque. As no GB was suggested, the selection process was left to ODI. Constraints were mechanical fit into the DBM and gear ratios between 50 and 100. Existing, GB, GBC , GBM were studied for proper mechanical fit. A possible solution was to keep the same shaft size. Of twelve GBs, none passed the same SF requirements.

Another solution was to increase the shaft size to . The same procedure was followed and the WITTENSTIEN SP-140S-MF2-70-1E1 was selected upon passing aforementioned SF requirements. This 70:1 reduction GB is rated for 360. and 660. nominal and acceleration torques, respectively. To implement this solution, it is required to replace the GBM, bore and broach the GBC and replace the socket head cap screws. The proposed design was validated.

Section 5

Recommendations

Improving testing methods and data extrapolation could be highly beneficial as this could allow more accurate torque values. Due to the lack of data acquisition flexibility, there is possibly a large error in the analyzed data. For instance, the extrapolation methods with AutoCAD were approximated as best as humanly possible. Not to mention the error that is already included in the raw measurements themselves. Given that the extrapolated data is valid, there are other variables to be factored into the equation. Mechanical properties such as steel rule curvature, thickness are likely to be similar for the measured samples but could likely vary for other trials in the future. Hence, the observed torque used to calculate SFs do not represent the entire operation. These inconsistencies are normally a result of coiling the steel rule, which varies for all coils. Also, further research could be completed with regards to another possible gearbox solution which does not require any modification to the system. Possible ways to improve the retrofitting process could be to let a more experienced supplier (Brock Solutions) also deal with the selection of the gearbox, given ODI's data.

Section 6

Supplemental

References

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- [2] S. Timoshenko, S. Woinowsky-Krieger, Theory of plates and shells, Engineering societies monographs, McGraw-Hill, URL <https://books.google.com/books?id=rTQFAAAAMAAJ>, 1959.