FEM-AIDED MATERIALS SELECTION OPTIMIZATION

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During the design process materials selection is usually carried out unsystematically, selection is mostly based on previous experience. Professor M. Ashby developed a methodology of materials selection optimization based on material indexes. Our aim is to link this methodology with optimization interface of software ANSYS. This link allows solving complicated multi-criteria problems that cannot be solved analytically. In this paper it was shown that it is possible to create connection between material selection methodology based on material indices and numerical finite element method analyses conducted in ANSYS. New material indices were determined for buckling of a cylinder. The developed methodology is also applicable for more complicated problems which is the next step in this research.

KEY WORDS
materials selection optimization; material indices; ANSYS; FEM; CES EduPack; DOE, MATLAB

1. INTRODUCTION

Materials selection is an essential part of the product design process as materials influence most of the product's properties. Currently design engineers are able to choose from up to 160 000 engineering materials [ASHBY 2011]. Professor M. Ashby developed a methodology of materials selection optimization which is driven by engineering design process. CES (Cambridge Engineering Selector) EduPack materials database include the methodology and by using this database it is possible to optimize the selection process. Our aim is to link this methodology with optimization interface of software ANSYS. This link allows to solve complicated multi-criteria problems that cannot be solved analytically. In chapter 3 we deal with the verification of the possibilities of linking these two systems. In chapter 4 our method has been applied to specific cases.

2. ENGINEERING DESIGN-DRIVEN MATERIALS SELECTION

Materials selection is based on and driven by engineering design process of a given product. The requirements on the product are the inputs for this process. The requirements can be divided into three categories: *function, constraints, objectives and free variables*. The *function* is defined by the purpose of the part, e.g. to support load. The *constraints* are conditions that must be met, e.g. maximum deflection or maximum dimensions. During the design process there are some objectives to be achieved. We might want the product to be as light or as cheap as possible. Some parameters might be adjusted to maximize the fulfilment of objectives. These are *free variables*.

In Fig.1 engineering design-driven materials selection scheme is shown. Design requirements are translated into product specification suitable for materials selection. This can be done by deriving of material indices. The constraints set out limit values of certain properties. The objectives define the material indices for which we seek extreme values. If an objective is not bound with a constraint the material index becomes a simple material property. Otherwise the index becomes a group of properties.

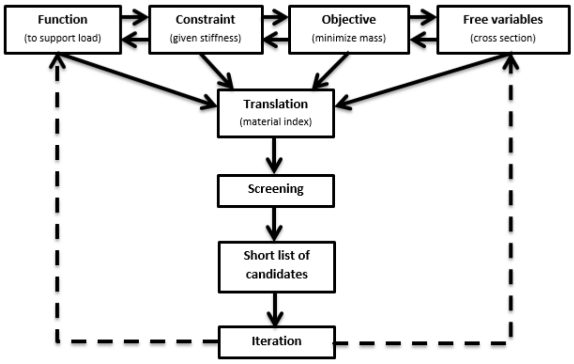

Figure 1. Engineering design-driven materials selection

The performance of a part is given by three parameters: functional requirements (F), geometrical dimensions (G) and material properties (M). It can by expressed by the equation below:

$$P = f_1(F) \cdot f_2(G) \cdot f_3(M) \tag{1}$$

The performance for is maximized when we maximize $f_3(M)$ which is called material index. Each combination of function, objective and constrain leads to a material index which makes this method general an applicable for a wide range of problems [ASHBY 2011].

3. VERIFICATION OF LINKING CES AND ANSYS

Fig. 2 shows the algorithm of linking CES and FEA. Firstly, a parametric CAD model is created and imported into ANSYS. Design Points are prepared taking into account an appropriate range of parameter values. Then the actual FEA is performed followed by data fitting procedure in MATLAB. And finally the material index is derived.

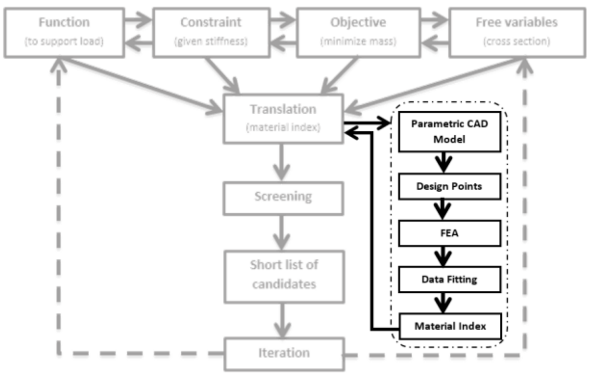

Figure 2. FEM-aided materials selection optimization

3.1 TIE ROD

Basic constrains are length *L* which is specified (geometric constraint) and the fact that rod must support axial pressure load *F* without buckling (functional constrain). The objective is minimizing the mass *m* of the tie. Cross-section area *A* and type of material are free variables. We have 3 free variables, density ρ, Young's modulus E and radius of round rod r. To obtain the material index we use the equation for mass

$$m = \pi \cdot r^2 \cdot L \cdot \rho \tag{2}$$

where we substitute r from equation for F_{crit}

$$F_{crit} = \frac{\pi^2 \cdot E \cdot I}{L^2} = \frac{\pi^3 \cdot E \cdot r^4}{4 \cdot L^2} \tag{3}$$

After that operation we get this final equation

$$m \geq \left(\frac{4F}{\pi}\right)^{\frac{1}{2}} \cdot (L)^2 \cdot \left(\frac{\rho}{E^{\frac{1}{2}}}\right) \tag{4}$$

Material index is taken from the last part of the equation

$$M = \frac{\rho}{E^{\frac{1}{2}}} \tag{5}$$

A similar problem was set up in ANSYS to prove the material index above. As can be seen in Fig. 3 [MAZINOVA 2016] the resultant surface has the same equation therefore the material index is verified [ASHBY 2011].

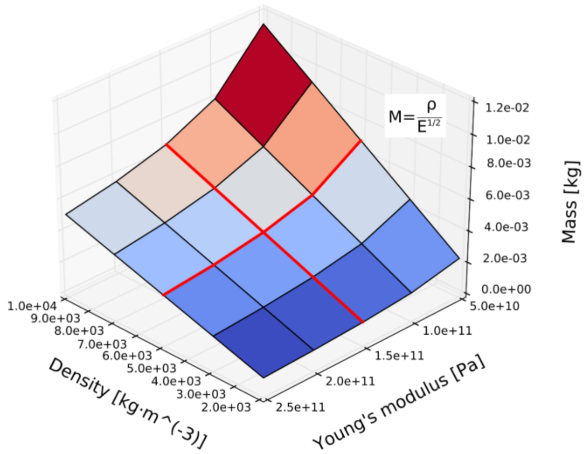

Figure 3. Material index for buckling of a tie rod

3.2 STIFF PANEL

A panel is a flat slab. The panel has a 3 geometric dimensions - length, width and thickness. Geometric constrains are length **L** and width **b**. Thickness **h** is a free variable as well as material. A panel is loaded in bending by a central load **F**. In this case is not a functional constrain force **F**, but bending stiffness **S***. The objective is minimizing the mass **m** of the panel. To obtain the material index we use the equation for mass

$$m = b \cdot h \cdot L \cdot \rho \tag{6}$$

and equation for bending stiffness which must be at least **S***

$$S = \frac{C \cdot E \cdot I}{L^3} \geq S^* \tag{7}$$

where **C** is only a constant which depends on the distribution of the loads. The second moment of area is equal to

$$I = \frac{b \cdot h^3}{12} \tag{8}$$

Similarly as in the previous case we substitute the free variable. Here it is height **h**. After substitution we get the following equation

$$m = \left(\frac{12 \cdot S^*}{C_1 \cdot b}\right)^{\frac{1}{3}} \cdot (b \cdot L^2) \cdot \left(\frac{\rho}{E^{\frac{1}{3}}}\right) \tag{9}$$

Material index is taken from the last part of the equation again

$$M = \frac{\rho}{E^{\frac{1}{3}}} \tag{10}$$

A similar problem was set up in ANSYS to prove the material index above. As can be seen in Fig. 4 [MAZINOVA 2016] the resultant surface has the same equation therefore the material index is verified [ASHBY 2011].

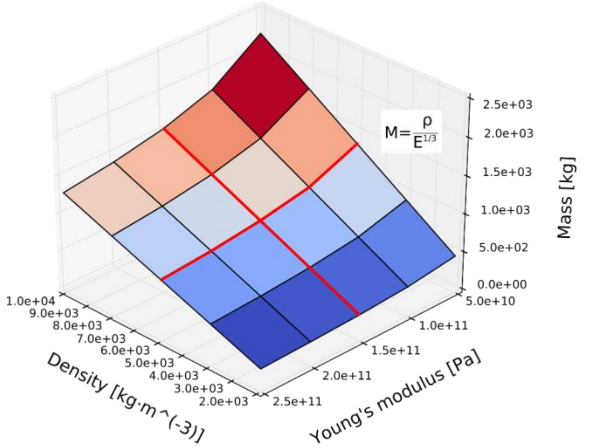

Figure 4. Material index for bending of a panel

3.3 ROUND BEAM

Beams come in many shapes: solid rectangles, cylindrical tubes, I-beams, and more. In our case we have round rod with a defined length **L** and an undefined radius **r**. The beam must support bending load **F** without deflecting too much, meaning that bending stiffness **S** is specified as **S*** (functional constraint). To obtain the material index we use the equation for mass

$$m = A \cdot L \cdot \rho = b^2 \cdot L \cdot \rho$$
(11)

and equation for bending stiffness which must be at least **S***

$$S = \frac{C \cdot E \cdot I}{L^3} \geq S^* \tag{12}$$

where **C** is only a constant which depends on the distribution of the loads. The second moment of area is equal to

$$I = \frac{b^4}{12} = \frac{A^2}{12} \tag{13}$$

Again we substitute the free variable. Here it is square section **A**. After substitution we obtain the following equation

$$m = \left(\frac{12 \cdot S^* \cdot L^3}{C}\right)^{\frac{1}{2}} \cdot (L) \cdot \left(\frac{\rho}{E^{\frac{1}{2}}}\right) \tag{14}$$

Material index is taken from the last part of the equation again

$$M = \frac{\rho}{E^{\frac{1}{2}}} \quad (15)$$

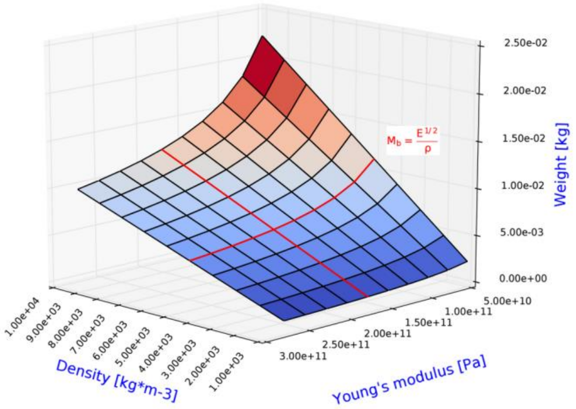

Figure 5. Material index for a beam

4. ENGINEERING DESIGN-DRIVEN MATERIALS SELECTION

A circular cylinder is defined by radius r, length L and thickness t. If is ratio radius to thickness r/t higher than 10, we talk about thin-walled cylindrical tube or cylindrical shell. From a mathematical point of view cylinder is a very stiff shape, but that can be applied only on a perfect model. In reality every product has a material, geometrical or mechanical defect of a little or greater importance.

Buckling is a very common load case of circular cylinders shells. Cylinders under external pressure are often used as submarine hulls, pressure vessels, off-shore drilling rigs, submarine pipelines, tubular tunnels, etc.

We distinguish two main types of buckling – a global (longitudinal) buckling and a lateral (transvers) buckling.

The global buckling is a buckling, when a cylindrical shell is loaded on both sides of its longitudinal axis. This type of loading occurs for example in the off-shore industry during drilling. When oil platform starts to spud a new well, there is a drilling pipeline which is several kilometers long. A part of this drilling pipeline is between a vessel and a seabed in the water without any support. This has a consequence that when a drilling rig encounters hard subsoil, two forces starts to acts against each other there. All parts of the drilling pipeline which is submerged in water begins to bend and if the force is too high, then the weak/most stressed part of the pipeline collapses.

A similar phenomenon occurs in submarine pipelines, where high pressure and temperature cause deformations. [OMMUNDSEN 2009]

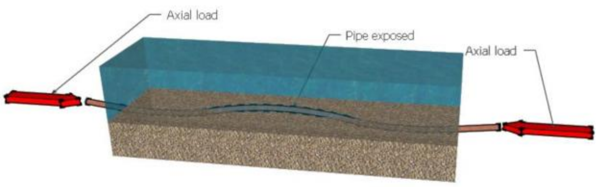

Figure 6. Global buckling of submarine pipeline [OMMUNDSEN 2009]

The second type of the buckling is the lateral buckling, when cylindrical vessel is subjected to external overpressure. One of the best examples are submarines hulls.

If a cylindrical vessel is exposed to external pressure and pressure is too high, then vessel collapses. We call this failure non-symmetric bifurcation buckling or shell instability which is show in Fig. 7 [ROSS 2010].

Figure 7. Shell instability of a circular cylinder

To avoid this problem the cylinder vessels can be reinforced by rings or rigs which are equally spaced along the longitudinal axis of the cylinder. Reinforced vessels can be buckle in different ways.

The first type of failure is shell instability, it is characteristic by one longitudinal deflection as shown in Fig. 8 [SREELATHA 2012]. It appears when the vessels are weakly reinforced.

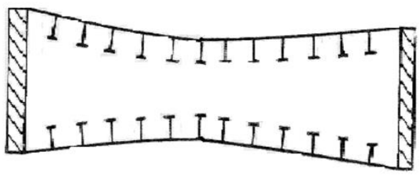

Figure 8. Shell instability

The second type of failure is general instability, it is characteristic by many longitudinal and circumferential deflections as shown in Fig. 9 [SREELATHA 2012]. It is specific for very strongly reinforced vessels.

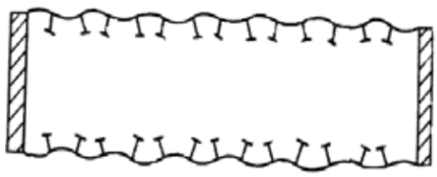

Figure 9. General instability

The third type is also specific for strongly reinforced vessels, but in this case the spaces between stiffening rings are small. This type is called axisymmetric deformation and it is characteristic by the fact that the vessel keeps its circular form while imploding inwards, as shown in Fig. 10 [ROSS 2010]. This mode of failure is more predictable. [ROSS 2010], [SREELATHA 2012]

Figure 10. Axisymmetric collapse

5. DETERMINATION OF A NEW MATERIAL INDEX FOR BUCKLING OF A CYLINDER

The second part of this paper is devoted to derivation of a new material index using finite element method. Let's assume we would like to investigate buckling behavior of a thin-walled cylinder. There are two different load cases. Firstly the cylinder is exposed to an external pressure and secondly to an axial force. The height as well as diameter of the cylinder is set to 1m. The thickness of the wall and material properties are subjected to a design of experiment simulation. The FEM model of the cylinder including two loading scenarios is shown in Fig. 11.

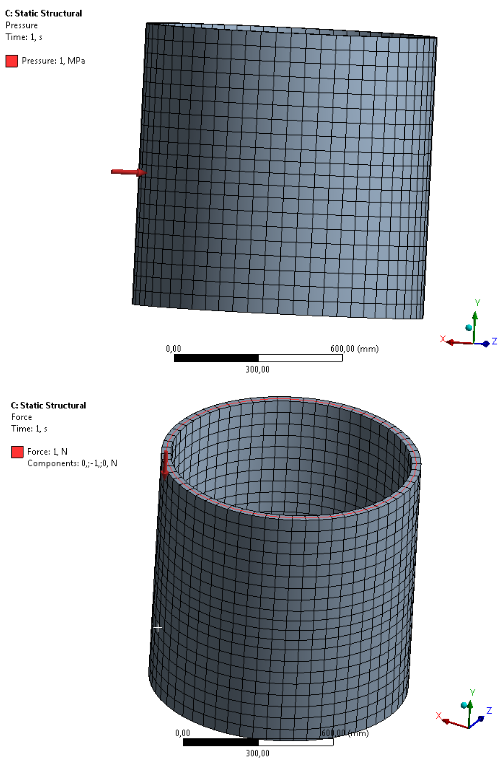

Figure 11. FEM model of the cylinder

As mentioned above in this case there are three variables - wall thickness, Young's modulus and density. Unlike previous problems we have no coupling equation reducing the number of variables so the first step is to derive the coupling equation. In order to do so, an additional design of experiment simulation was conducted. In this case density was not considered as one of the variables and the task was to obtain values of Load Multiplier as a function of wall thickness and Young's modulus. Using Matlab Curve Fitting Toolbox the coupling equation was derived:

$$t = \left(\frac{LM}{k_1 E + k_2}\right)^{k_3} \quad (16)$$

Thanks to this equation the relationship between wall thickness and Young's modulus is established. The general form of this equation is the same for both load cases; however, the constants vary. Having obtained the coupling equation it is possible to proceed to the second step – the second design of experiment simulation including density as one of the variables. Using Matlab Curve Fitting Toolbox the material index for buckling of a light thin-walled cylinder loaded by external pressure is:

$$M = \frac{k_4 \rho}{E^{0,4214}} \quad (17)$$

Fig. 12 shows the buckling failure mode I case of a radial pressure loading.

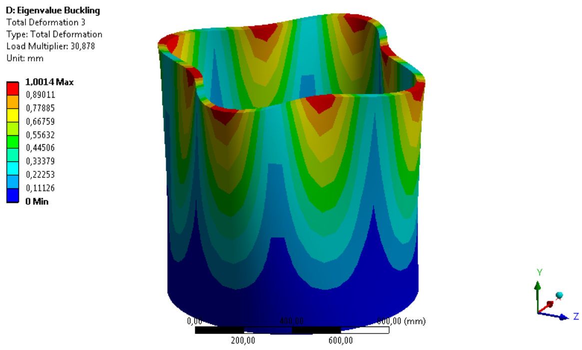

Figure 12. Buckling failure mode - radial pressure loading

The material index for buckling of a light thin-walled cylinder loaded by axial force is:

$$M = \frac{k_5 \rho}{E^{0,5255}} \quad (18)$$

Fig. 13 shows the buckling failure mode in case of axial force loading.

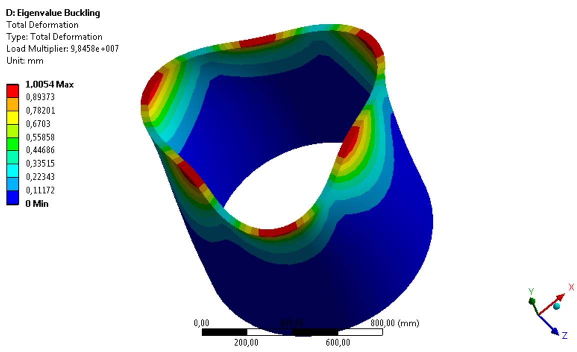

Figure 13. Buckling failure mode – axial force loading

The indices for buckling of a cylinder mentioned above were determined using linear buckling analysis. This type of analysis is based on modal analysis. A different approach is also possible using nonlinear buckling analysis which is based on nonlinear large deformation static analysis. This approach tends to be more conservative. Nonlinear buckling analysis was applied on a cylinder loaded by axial force and the appropriate material index was determined as follows:

$$M = \frac{k_6 \rho}{E^{0,5238}} \quad (19)$$

The slight difference in exponent in Eq. 19 confirms the assumption that the nonlinear analysis is more conservative than linear buckling analysis.

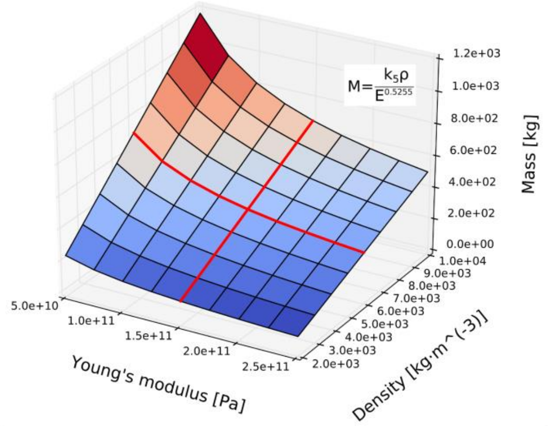

Figure 14. Material index for buckling of a cylinder

The choice of geometric parameters in the example above might not be valid in a larger range of dimensions. Therefore the next step was to generalize the problem and introduce two more variables – height and diameter. Density was omitted for a total of 4 variables. This led to a significant increase in computational time since the number of design points was much higher. In Fig. 15 there are Design Points Results for the mentioned problem. The increase in the number of variables resulted in more complicated process of function data fitting and this part is still under development.

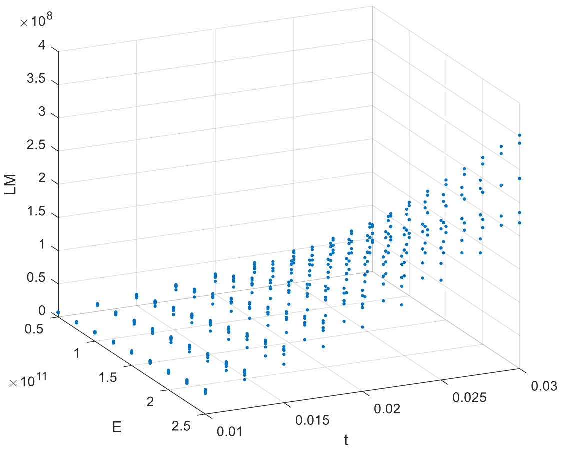

Figure 15. Design Points Results

6. CONCLUSION

In this paper it was shown that it is possible to create connection between material selection methodology based on material indices and numerical finite element method analyses conducted in ANSYS. Firstly, some of the existing material indices were verified by performing design of experiment simulations in ANSYS and the actual process of numerical verification was established. Secondly, new material indices were determined for buckling of a cylinder. Both linear buckling analysis and nonlinear buckling analysis were utilized and the expected variation was confirmed. Matlab Curve Fitting Toolbox was used to fit the numerical results with a custom equation to derive the indices. The developed methodology is also applicable for more complicated problems which is the next step in this research.

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