OPTIMUM DESIGN OF TANKS FOR SEASONAL THERMAL ENERGY STORAGE

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ABSTRACT

Energy demands in buildings vary on a daily, weekly and seasonal basis. These demands can be matched with the help of thermal energy storage systems (TES). TES systems have the potential of making the use of thermal equipment more effective, and they are important means of offsetting the mismatch between thermal energy availability and demand. The peaking power problem arising in the case of a discrepancy between energy supply and expenditure can be resolved by using energy accumulation. The construction costs for energy accumulation can be lower than those for special peak energy equipment. The performance and design problems of an advanced type of reinforced concrete thermal energy storage tank with a "hot" inner steel liner have been studied. In the case of the system with a thin steel liner the thermal buckling optimisation problem of a steel lining shell has to be solved. By using the linear theory of cylindrical shells in the case of thermal action only, a critical temperature interval is determined depending on the shell geometry and the stiffness of the basic reinforced concrete structure. On the basis of multi-objective optimisation, the design methods for optimum weight and fastening of the lining shell to the basic structure are derived.

Key words: Critical temperature, elastic foundation, multivariable optimization, steel liner, thermal buckling

INTRODUCTION

Demands for thermal energy vary on a seasonal basis. These demands can be matched with the help of Thermal Energy Storage (TES) systems that operate synergistically and are carefully matched to each specific application. TES systems have the potential of making the use of thermal equipment more effective, and are important means of offsetting the mismatch between thermal energy availability and demand. Well-designed systems can reduce initial and maintenance costs and improve energy efficiency (Dincer et al., 1997).

A variety of TES techniques for heating and cooling applications have been developed over the past decades. Increasing energy demands, shortages of fossil fuels and environmental concerns are increasing the interest in the development of economically competitive and reliable means of seasonal storage of thermal energy.

Different examples about the efficient utilization of natural and renewable energy sources, cost savings and increased efficiency achievable through the use of seasonal TES can be considered (Dincer and Rosen, 2011).

Any system providing energy consists of the source of primary energy, a subsystem of transformation and consumers of the transformed energy. In such systems, discrepancies can arise in time and space between energy supply and expenditure. The peaking power problem can be resolved by using energy accumulation. The construction costs for energy accumulation can be lower than those for special peak energy equipment (Beckmann and Gilli, 1984).

This study examines an advanced type of thermobattery with a "hot" inner steel liner (Fritz and Nemet, 1983). In order to prevent the penetration of vapour, gas and liquid into the base structure formed as a reinforced concrete structure, a thin steel liner is used. Between the liner and reinforced concrete vessel there is a thermal insulation layer (Fig. 1). The vessel acts in a plane stress state and must be resistant to normal inner pressure and its design has to be performed according to building codes (EN 1992-1-1, 2004).

During transient regimes with low inner pressure and high temperature, the steel liner is in a state of thermal deformation as a result of thermal expansion (Beckmann and Gilli, 1984). Thermal stresses can cause buckling of the lining shell. During repeated buckling processes of the facing shell, permanent deformation can occur, which causes deterioration of shell stiffness and durability of the structure. In Fig. 1 the cross section of layered tank is shown and the inner steel layer is depicted in the post-buckling form. By using fasteners in definite places along the circumference, it is possible to force the steel liner to buckle with the predefined form. In general, the buckling form of the lining shell resting on the elastic support depends on the shell geometry, stiffness of basic structure or the interlayer of the thermal insulation, as well as on the type of fastening to the basic structure.

To regulate the distribution of buckling waves along the length of the shell and around its circumference, and to improve the load carrying capacity of the steel shell during thermal action, a definite system of fasteners is needed. For a given length and radius of a lining shell, the critical temperature interval depends on the thickness of the shell and the stiffness of the basic structure and parameters of the buckling form. The purpose of this study is to develop an analytical method for optimization of a steel liner and the determination of an optimum system for fastening the liner to the basic structure.

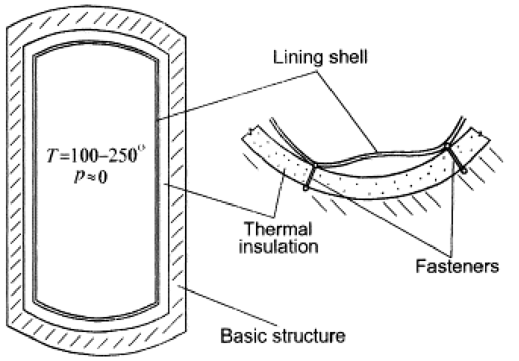

Figure 1. Scheme of thermal energy storage tank and cross section with buckled liner

STEEL LINER BEHAVIOUR UNDER THERMAL ACTION

Analytical solution to the thermal buckling problem

The objective of this research is to study the thermal buckling problem of a steel shell resting on an elastic foundation. It is assumed that the temperature of the outer "cool" basic construction is fixed and that there is no temperature gradient throughout the thickness of the steel shell. For the given temperature interval, the modulus of elasticity of the shell does not change. Both ends of the shell are pinned. In-plane and out-of-plane (lateral) displacements are not possible (Fig. 2).

The steel shell is considered as a cylinder of length L and radius R resting on an elastic foundation. It is assumed that the reaction offered by the basic construction to the thermal lateral deflection w of the steel shell is proportional to the deflection. Thus, the reaction per unit area of the shell is Kw, where K is constant, called the modulus of foundation.

According to the linear theory of cylindrical shells in the case of thermal action only, the equilibrium equation is given by

$$\frac{D}{h}\nabla^4 w = \frac{1}{R}\frac{\partial^2\varphi}{\partial x^2} - \sigma_x\frac{\partial^2 w}{\partial x^2} + \sigma_y\frac{\partial^2 w}{\partial y^2} - \frac{Eh\alpha}{12(1-\nu)}\nabla^2\theta \tag{1}$$

and the deformation compatibility equation is

$$\frac{1}{E}\nabla^4\varphi = \frac{1}{R}\frac{\partial^2 w}{\partial x^2} - \alpha\nabla^2 T\,, \tag{2}$$

where θ – effective value of thermal moment (Volmir, 1963; Brauns, 1988);
T – average temperature along thickness of the shell;
α – coefficient of thermal expansion;
ν – Poisson's ratio of steel;
h – thickness of shell.

Cylindrical stiffness D is determined in the following way:

$$D = \frac{Eh^3}{12(1-\nu^2)}\,. \tag{3}$$

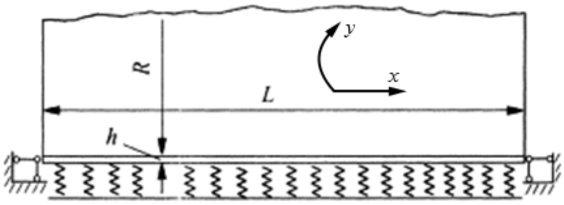

Figure 2. Analysis scheme of cylindrical part of lining shell resting on elastic foundation

The coordinates x and y are oriented in a lengthways and circumferential direction, respectively.

Elimination Airy stress function φ in Eqs (1) and (2), and taking into account the reaction of the basic structure to thermal expansion of the shell yields the result

$$\frac{D}{h}\nabla^8 w + \frac{E}{R^2}\frac{\partial^4 w}{\partial x^4} + \sigma_x\nabla^4\frac{\partial^2 w}{\partial x^2} + \sigma_y\nabla^4\frac{\partial^2 w}{\partial y^2} + \frac{K}{h}\nabla^4 w = 0, \tag{4}$$

where σ_x and σ_y - normal stresses in direction; x and y, respectively;
K – modulus of foundation in post-buckling stage.

Differential operators in expressions (1), (2), and (4) are expressed in the following forms:

$$\nabla^4 = \frac{\partial^4}{\partial x^4} + 2\frac{\partial^4}{\partial x^2\partial y^2} + \frac{\partial^4}{\partial y^4};\ \nabla^8 = \nabla^4\nabla^4\,. \tag{5}$$

The following function is assumed for the radial deflection of lining shell:

$$w(x,y) = \sin\frac{m\pi x}{L}\sin\frac{ny}{R}, \tag{6}$$

where m and n – numbers of half waves in the lengthways direction of the shell and waves in the circumferential direction of the shell, respectively.

Equation (6) satisfies boundary conditions on the $x = 0, L$.
By substituting representation of the lateral deflexion (6) into the governing differential equation (4) the following result is obtained:

$$\begin{aligned}&\frac{D}{h}\left(\frac{m^2\pi^2}{L^2}+\frac{n^2}{R^2}\right)^4+\frac{E}{R^2}\frac{m^4\pi^4}{L^4}-\\&-\sigma_x\left(\frac{m^2\pi^2}{L^2}+\frac{n^2}{R^2}\right)^2\frac{m^2\pi^2}{R^2}-\\&-\sigma_y\left(\frac{m^2\pi^2}{L^2}+\frac{n^2}{R^2}\right)^2\frac{n^2}{R^2}+\\&+\frac{K}{h}\left(\frac{m^2\pi^2}{L^2}+\frac{n^2}{R^2}\right)^2=0.\end{aligned} \tag{7}$$

The thermal stress component σ_y is defined by

$$\begin{aligned}\sigma_y&=\frac{R}{h}K_0w_0=\\&=\frac{R}{h}K_0\left[\alpha\Delta TR-\frac{\sigma_y\Delta T^2R}{E}(1-\nu^2)\right],\end{aligned} \tag{8}$$

where w_0 is deflection of the lining shell and K_0 is modulus of foundation in the pre-buckling stage, respectively.

The temperature increase is determined as $\Delta T = T - T_0$, where T_0 is initial temperature. By solving Eqs (5) and (8) for ΔT_0, the critical temperature interval ΔT^{cr} is determined.

Structural optimization of lining shell

If the system to be optimized does not yet exist, or if experimentation on an existing system is not feasible, due to high costs or for other practical reasons, the only approach is through an analytical model. The structural optimization problem considered consists of the weight $W(\xi)$ minimization of a steel shell lining including fastening to the basic structure. The design variables ξ_i are the length $(\xi_1 = L)$ and radius $(\xi_2 = R)$ of the shell at the given tank volume and design temperature interval ΔT, shell thickness $(\xi_3 = h)$, stiffness $(\xi_4 = K_0)$ of support (interlayer) in pre-buckling stage and wave numbers $(\xi_5 = m, \xi_6 = n)$.The entire problem can be expressed in terms of the design variables as follows: find a vector ξ such that

$$\begin{aligned}&W(\xi)=\gamma(2\pi RLh+V_{bot}+V_{top}+\\&+V_{fast})\rightarrow \min\end{aligned} \tag{9}$$

Subject it to behavioural constraints

$$G_j(\xi)=g_j^U-g(\xi)\geq 0,\ j\in Q_R \tag{10}$$

and side constraints

$$\xi_i^L \leq \xi_i \leq \xi_i^U . \tag{11}$$

Here Q_R denotes the set of retained constraints:
g_j^U – the upper bond to a response quantity $g_j(\xi)$;
ξ_i^L and ξ_i^U – the lower and upper limit of the independent design variables ξ_i, respectively;
γ – the specific gravity of steel;
V_{bot}, V_{top} and V_{fast} – volumes of the steel liner in the bottom and top part of the battery and the volume of fasteners, respectively.
One important feature of constrained optimization is the difficulty in showing that a local optimum is in fact a global value. In general, however, starting from different base points, and if all searches lead to the same solution, it is likely that this is the global optimum. These procedures can be used freely in unconstrained problems. With restrictions, however, it is not so easy to obtain valid alternative starting points that satisfy all the constraints and which are significantly different.

Numerical results and discussion

The problem is solved with a linear elastic statement. The multi-objective optimization by applying Optimization Toolbox used together with MATLAB is performed. By applying the conjugate gradient-type minimizer, the optimum fastening system, depending on temperature interval and lining shell geometry as well as stiffness of the basic structure, according to Eq. (11) is performed. In practical cases, it is important not only to locate the optimum, but also to examine the nature of function in this neighbourhood, since it is unlikely that we can exactly maintain the optimal conditions. Certainly, each adjacent point should have a worse value of the objective function than that at the optimum, but this is not enough. It is also necessary to know the sensitivity of the designed system.
The purpose of the investigations is not only to find the local or global optimum but also to carry out analysis and perform the design of the structure. An algorithm and PC programme have been developed

for drawing isolines in sections $\xi_i \xi_j$ $(i \neq j)$with all the factors fixed except two. After determination of buckling form, i.e. m and n, providing the given critical temperature interval at fixed length L, radius R and thickness h (Fig. 3), graphs for the practical design of the lining shells according to the given conditions are derived (Fig. 4).

Note that, in order to prevent the thermal buckling below critical temperature for the given stiffness of the basic structure or insulation layer, a fixed thickness of lining shell is needed. To increase the critical temperature at the given stiffness of support structure and shell thickness $n-1$ or $n-2$, rows of fasteners in a circumferential direction can be used. The fasteners fix the shell at determined points along the circumference against the radial displacement resulting in buckling form with decreased number of waves.

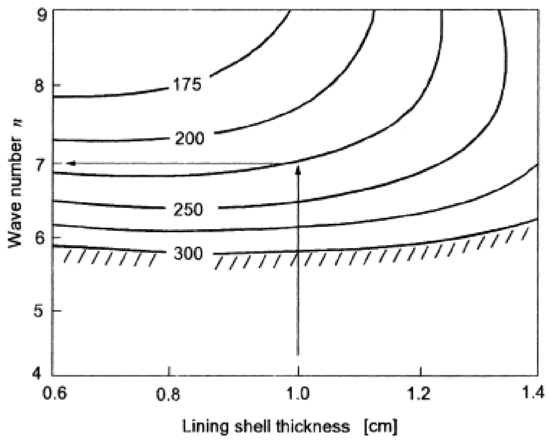

Figure 3. Isolines of critical temperature interval (◦C) of lining shell: $m = 1$, K_0 0.75 MPa, $L = 20$ m, $R = 2.5$ m

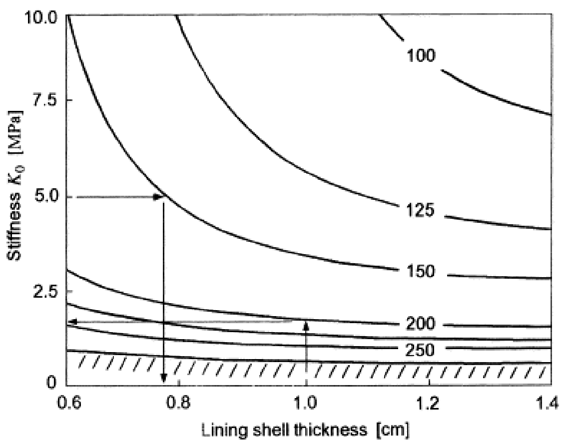

Figure 4. Critical temperature interval (◦C) isolines of lining shell: $m = 1$, $n = 5$, $L = 20$ m, $R = 2.5$ m

Fig. 5 shows the isolines of the lining shell weight with shell length and radius. Line 1 shows the weight variation in the cylindrical part of the liner at a fixed volume of thermal energy storage tank $V = 400$ m^3 and liner thickness 1.0 cm.

The weight difference between tank with length $L = 5$ m and $L = 20$ m is a factor of two. Taking into account the weight of the liner in semi-spherical bottoms, the difference is approximately 30%, i.e. 33 and 24 tons, respectively. Because the construction of the cover (top) is complicated in the case of large diameter, a tank with a length to diameter ratio of 3-5 is more preferable.

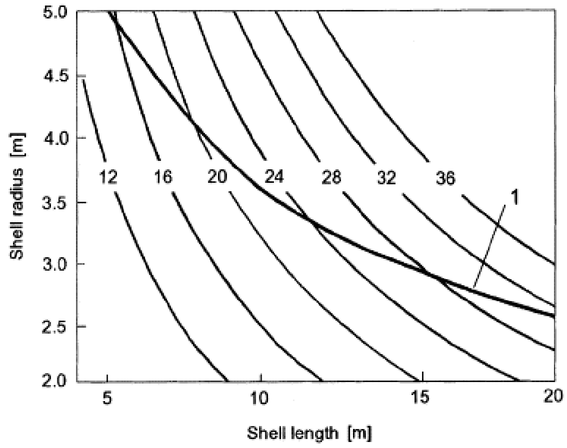

Figure 5. Isolines of lining shell weight (tons); 1 – cylindrical part of liner at tank volume $V = 400$ m^3 and liner thickness $h = 1.0$ cm

On the basis of multi-variable optimization (Beveridge and Schechter, 1970) for the designed thermal energy storage tank of volume V, the minimum weight problem of the lining shell, including fastening system, can be solved by taking into account the behavioural and side constraints as well as the given degree of safety (Malmeister et al., 1980) for thermal buckling.

CONCLUSIONS

An analytical method for the thermal buckling of a lining steel shell in a holder has been developed. On the basis of multi-objective optimization, the design methods for optimum weight and fastening system of the lining shell, depending on temperature interval, have been derived. By using fasteners, the lining shell is fixed to base structure at determined points along the circumference against radial displacement and buckling form with $n-1$ or $n-2$ waves realized, and the degree of safety for thermal buckling of the liner for a given geometry has been increased.

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