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# OPTIMAL DESIGN OF SANDWICH PANELS WITH A SOFT CORE

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The main issue taken up in the paper is to find optimal designs of multi-span sandwich panels with slightly profiled steel facings and polyurethane foam core (PUR), which would satisfy conflicting demands of the market, i.e. minimal variance in types of panels, maximum range of application and minimum cost. The aim is to find dimensional and material parameters of panels which generate minimum cost and maximum length of span under prescribed loads in ultimate and serviceability limit states. The multi-criterion optimization problem is formulated in such a way, where the length of the span plays two roles, namely a design variable and a component of a vector objective function. An evolutionary algorithm is used. Numerous inequality constraints are introduced in two ways: directly and by external penalty functions.

*Key words:* sandwich panels, soft core, Pareto optimization, soft computing, genetic algorithm

## 1. Introduction

In recent years, we have observed an increase in practical applications of sandwich panels in civil engineering and mechanical constructions (Hassinen *et al.*, 1997; Magnucka-Blandzi and Magnucki, 2007). High bending stiffness coupled with small weight and very good thermal and damping properties make sandwich panels an attractive structure for designers (Craig and Norman, 2004). Easiness of transport and assembly in all weather conditions are additional advantages. These aspects have also generated a growth in investment outlays for computational and experimental research.

Sandwich panels most often used in civil engineering consist of two steel facings and a soft core. The facings carry normal stresses, while the three principal roles of the core are to carry shear stresses, to protect the compressed facing against buckling and to provide expected thermal insulation (Chuda-Kowalska *et al.*, 2007). Note that a greater elasticity modulus of the core, which usually improves the mechanical response of a slab, is accompanied by greater weight and higher cost of the material, as well as worse insulation coefficient. Another conflict is observed between mechanical features and price of a panel when variable thickness or depth of the profiling of facings is considered. The above conflicting relations make the traditional design difficult and give reasons for using methods of the optimal design.

Optimal design of sandwich panels has been widely discussed in literature. Bozhevolnaya and Lyckegaard (2006) examined sandwich panels with junctions between different core materials. The application of different core materials in the same panel allows designers to improve the capability of the core to withstand varying external shear loads. The optimal choice of the core material was also discussed in Czarnecki *et al.* (2008), where the basic aim was to find plates of minimal compliance. The comparison of sandwich panels behaviour with various combinations of materials in three-point bending was described in Craig and Norman (2004). This analysis served as a basis for creating failure maps which make it possible to optimize the geometry and mass of sandwich panels. Tan and Soh (2007) discussed the two-criterion optimization (minimization of the weight and heat transfer) for sandwich panels with a prismatic core using genetic algorithms.

The present paper extends the class of problems discussed in literature by considering conflicting market demands for panels of minimum cost and the widest range of application. To fulfill the latter demand, we will look for solutions applicable as one- or multi-span structural elements satisfying design requirements for the maximum length of the span in typical load conditions. A two-span plate can be considered as a representative case, because it exhibits simultaneously the field and support phenomena, which are dominant in design. It is assumed that the panels are subjected to mechanical and thermal loading. Thermal action induces high strains and stresses in multi-span panels. Its destructive effects can be reduced by using methods of optimal structural design. The interrelation of optimal values of design variables can be used in manufacturing of panels and their engineering applications.

## 2. Sandwich panel theory

This paper discuses only panels with parallel facings and soft core. The panel under consideration with the assumed support and loading conditions is shown in Fig. 1.

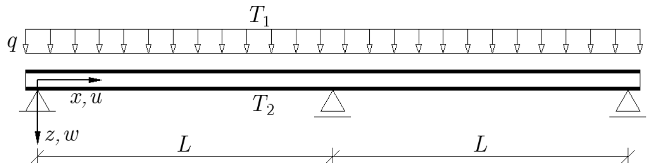


Fig. 1. Two-span sandwich panel loaded mechanically and thermally. $q$ represents uniformly distributed external load. $T_1$ and $T_2$ denote changes of temperature of the upper and lower facings, respectively

In this case of support, uniformly distributed loading and thermal actions, a sufficiently adequate theory is the Timoshenko beam theory generalized to sandwich sections. This is valid however, only when the ratio of the span length $L$ to the plate width $b$ is greater than 2. According to the modified Timoshenko and Reissner theories (Stamm and Witte, 1970) we assume that: the strains are small; the materials (steel in the facings and foam in the core) are isotropic, homogeneous and linearly elastic; normal stress in the foam core is negligible $(\sigma_{xC} = \sigma_{yC} = 0)$. From the last assumption it follows that the shear stresses in the core are constant along the transverse axis $z$ $(\tau_{xzC} = \tau_{yzC} = \text{const})$. The mechanical model for the structural analysis, i.e. for the evaluation of stresses, strains and displacements, depends on the type of facings (flat or profiled). Deeply profiled facings made of thin sheets fall into the same category as thick facings. Therefore, panels which have at least one deeply profiled facing must be considered separately from those that have two flat facings. Our attention will be focused on panels with flat or slightly profiled facings. However, for the sake of completeness, we start the discussion with the general case, in which at least one facing is deeply profiled or thick. The cross sectional equilibrium condition can be written in the form of two uncoupled differential equations $(2.1)_1$ for vertical deflection w and $(2.1)_2$ for the shear strain $\gamma$ (Stamm and Witte, 1970)

$$
\begin{aligned}
&-\frac{B_{F1}+B_{F2}}{G_C A_C} w^{VI} + \frac{B}{B_S} w^{IV} = \frac{q}{B_S} - \frac{q''}{G_C A_C} - \theta'' \\
&-\frac{B_{F1}+B_{F2}}{G_C A_C} \gamma^{IV} + \frac{B}{B_S} \gamma'' = -\frac{q'}{G_C A_C} - \frac{B_{F1}+B_{F2}}{G_C A_C} \theta'''
\end{aligned}
\tag{2.1}
$$

where $w$ and $\gamma$ are functions of the position coordinate x, which is measured along the length of the panel (see Fig. 1). The denominator $G_C A_C$ represents the shear stiffness of the core, $q$ is the uniformly distributed transverse load and $\theta$ is the initial curvature induced by a temperature difference $\Delta T$ between the lower and upper face sheets. $\theta$ is positive when associated with stretching of the lower face. The total bending stiffness $B$ is given by

$$B = B_{F1} + B_{F2} + B_S + B_C \tag{2.2}$$

where $B_{F1}$, $B_{F2}$ are the bending stiffness of the upper and lower facings with respect to their own centre lines $(2.3)_1$, BS is the bending stiffness of the facings with respect to the global centre line of the sandwich panel $(2.3)_2$ and $B_C$ is the bending stiffness of the core with respect to its own centre line $(2.3)_3$

$$\begin{aligned} B_{Fi} &= \frac{bt_i^3}{12} E_{Fi} \qquad\qquad B_S = bt_i e_i^2 E_{Fi} \qquad\qquad (i = 1, 2) \\ B_C &= \frac{bd^3}{12} E_C \end{aligned} \tag{2.3}$$

Table 1 demonstrates typical ratios of stiffness parameters encountered in sandwich panels with soft core.

**Table 1.** Stiffness parameters

| Stiffness ratios | Range [‰] |
|---|---|
| $B_C/B_S$ | 0.470-0.630 |
| $\sum B_{Fi}/B_S$ (flat facings) | 0.008-0.015 |
| $\sum B_{Fi}/B_S$ (slightly profiled facings) | 0.020-0.150 |
| $\sum B_{Fi}/B_S$ (profiled facings) | 1.00-15.00 |

This table demonstrates that $B_{F1}$ and $B_{F2}$ in sandwich panels with flat and slightly profiled facings as well as $B_C$ are negligible. Neglecting $B_{F1}$, $B_{F2}$ and $B_C$ in (2.1) and (2.2), one arrives at (2.4) for sandwich beams with flat or slightly profiled facings

$$w^{IV} = \frac{q}{B_S} - \frac{q''}{G_C A_C} - \theta'' \qquad\qquad \gamma'' = -\frac{q'}{G_C A_C} \tag{2.4}$$

Integrating twice (2.4) and using equilibrium equations $M' = Q$, $Q' = -q$, we obtain constitutive equations (2.5) of the Timoshenko beam

$$M = B_S(\gamma' - w'' - \theta) \qquad\qquad Q = G_C A_C \gamma \tag{2.5}$$

where $M = M_y$ and $Q = Q_z$ denote the bending moment and shear force, respectively.

The normal stresses $\sigma_{F1}$, $\sigma_{F2}$ in the metal face sheets and shear stress $\tau$ in the core are expressed by $(2.6)_1$ and $(2.6)_2$, respectively (see Fig. 2)

$$\sigma_{Fi} = (-1)^i \frac{M}{eA_{Fi}} \qquad \tau = \frac{Q}{A_C} \qquad (i = 1, 2) \tag{2.6}$$

where $A_{Fi}$ $(i = 1, 2)$ represents cross-sectional areas of the face sheets and $A_C$ represents the cross-sectional area of the core. The shear stresses $\tau$ across the facings $t_1$ and $t_2$ have the 2-nd order parabolic distribution (Fig. 2). Since $t_1$ and $t_2$ are very small (usually 0.5 mm), the shear stresses in the facings can be neglected. This is represented by $(2.6)_2$.

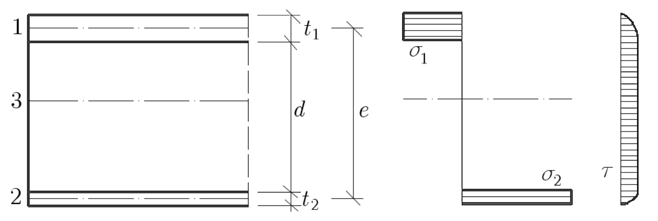


Fig. 2. Distribution of normal stresses $(\sigma_1, \sigma_2)$ and shear stress $(\tau)$ in a sandwich panel with thin flat facings

## 3. Formulation and solution of the optimization problem

Consider a two-span sandwich panel with flat or slightly profiled steel facings and with equal span lengths $L$. The panel is subjected to temperature action and to a uniformly distributed load, resulting from snow or wind actions. The temperature of the face exposed to sunshine depends on the colour of the face.

The design vector $\boldsymbol{x}$ comprises five variables which specify geometric and material parameters of the panel and the length of spans

$$\boldsymbol{x} = [L, t_1, t_2, d, G_C] \tag{3.1}$$

where $L$, $t_1$, $t_2$, $d$ and $G_C$ denote the length of the span, thickness of the external and internal face sheets, the thickness and the shear modulus of the soft core, respectively.

The aim of the optimization is to find a structure of the maximum length $L$ and minimum cost. Hence, the optimal design problem is formulated as a two-criterion optimization with a vector objective function $\varGamma$. Our task is to find the optimal design vector $\boldsymbol{x}$, satisfying the constraints and providing minimum of $\varGamma$

$$\boldsymbol{\varGamma}(\boldsymbol{x}) = [\varGamma_1, \varGamma_2] = [F_C, \alpha/L] \rightarrow \min_{\boldsymbol{x}\in X_0} \tag{3.2}$$

Here $F_C$ is the total cost, $L$ is the span length and $\alpha$ is the scaling coefficient. $X_0$ is the allowable domain of design vectors $\boldsymbol{x}$, specified by constraints

$$g_i(\boldsymbol{x}) \leqslant 0 \qquad\qquad i = 1, 2, \ldots, n \tag{3.3}$$

Constraints (3.3) fall into one of two categories. The first one represents behavioral constraints, which are implicit functions of $\boldsymbol{x}$. They follow from requirements of the ultimate and serviceability limit states and are listed in Tables 2 and 3.

**Table 2.** Constraints following from the ultimate limit state

| Constraint | Comments |
|---|---|
| $g_i(\boldsymbol{x}) = \dfrac{\lvert\tau_i\rvert}{f_{cvd}} - 1 \leqslant 0$ | $\tau_i$ – shear stress<br>$f_{cvd}$ – design shear strength of core |
| $g_i(\boldsymbol{x}) = \dfrac{\lvert\sigma_i^M\rvert}{f_{yd}} - 1 \leqslant 0$ | $\sigma_i^M$ – normal stress in span or at support<br>$f_{yd}$ – design yield strength of face sheet |
| $g_i(\boldsymbol{x}) = \dfrac{\lvert\sigma_i^w\rvert}{f_{swi}} - 1 \leqslant 0$ | $\sigma_i^w$ – wrinkling stress in span or at support<br>$f_{swi}$ – design wrinkling stress in span or at support with consequential failure |
| $g_i(\boldsymbol{x}) = \dfrac{\lvert\sigma_i^d\rvert}{f_{ccd}} - 1 \leqslant 0$ | $\sigma_i^d$ – crushing stress at support<br>$f_{ccd}$ – compressive strength of core |
| $g_i(\boldsymbol{x}) = \dfrac{F_i^L}{f_{dl}} - 1 \leqslant 0$ | $F_i^L$ – reaction at support<br>$f_{dl}$ – design fastener capacity |

The second group takes the form of box conditions on the components of $\boldsymbol{x}$, they are explicit functions. The total cost $F_C(\boldsymbol{x})$ in (3.2) is expressed by the cost $c_0$ and specific costs of materials $c_1$, $c_2$. Here $c_0$ contains constant manufacturing costs independent of the design variables $\boldsymbol{x}$. The parameters $c_1$ and $c_2$ specify the cost of steel in face sheets and of PUR in the core. The parameters $c_0$, $c_1$ and $c_2$ were evaluated based on market data

$$F_C(\boldsymbol{x}) = c_0 + c_1(t_1 + t_2) + c_2(G_C)d \tag{3.4}$$

**Table 3.** Constraints following from the serviceability limit state

| Constraint | Comments |
|---|---|
| $g_i(\boldsymbol{x}) = \dfrac{\lvert\sigma_i^M\rvert}{f_{yk}} - 1 \leqslant 0$ | $\sigma_i^M$ – normal stress in span or at support<br>$f_{yk}$ – characteristic yield strength of face sheet |
| $g_i(\boldsymbol{x}) = \dfrac{\lvert\sigma_i^w\rvert}{f_{k\,swi}} - 1 \leqslant 0$ | $\sigma_i^w$ – wrinkling stress in span or at support<br>$f_{k\,swi}$ – characteristic wrinkling stress in span or at support without consequential failure |
| $g_i(\boldsymbol{x}) = \dfrac{u_i^{max}}{u_{lim}} - 1 \leqslant 0$ | $u_i^{max}$ – maximum deflection in span<br>$u_{lim}$ – limit deflection in span |

It is assumed that $c_0 = 35\,\mathrm{m}^{-1}$, $c_1 = 19200\,\mathrm{m}^{-2}$. The cost of material $c_2$ is considered to be a function of the shear modulus $G_C$:

$$c_2(G_C) = [a(G_C - G_C^0) + 100]\ \mathrm{m}^{-2} \qquad\qquad G_C^0 = 2500\ \mathrm{kPa} \tag{3.5}$$

The function $c_2(G_C)$ with different values of the parameter $a$ is illustrated in Fig. 3.

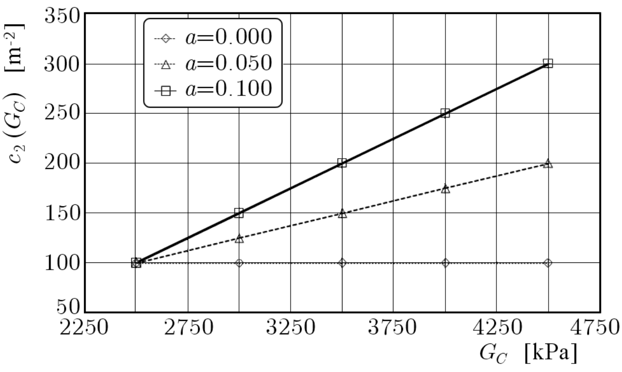


Fig. 3. Unit cost of core material $c_2$ as a function of the shear elasticity modulus of the core $G_C$

The implicit behavioral constraints are introduced by means of external penalty functions. Penalized objective function (3.2) takes the following form

$$\varGamma_1^P(\boldsymbol{x}) = \varGamma_1(\boldsymbol{x}) + \sum_{i=1}^{24} P_i(\boldsymbol{x}) \to \min_{\boldsymbol{x}} \tag{3.6}$$

where $P_i(\boldsymbol{x})$ are penalty functions $(i = 1, 2, \ldots, 24)$

$$P_i(\boldsymbol{x}) = \begin{cases} 0 & \text{for} \quad g_i(\boldsymbol{x}) \leqslant 0 \\ \beta_i g_i(\boldsymbol{x}) & \text{for} \quad g_i(\boldsymbol{x}) > 0 \end{cases} \tag{3.7}$$

The implicit constraints are evaluated from the stress and displacement, computed using FEM based on the modified Timoshenko theory (Pozorski *et al.*, 2007).

In practical applications, the specific costs and box conditions can be validated by manufacturing companies. The explicit constraints for $t_1$, $t_2$, $d$ and $G_C$ were assumed: $t_1, t_2 \in \langle 0.0004\,\mathrm{m}, 0.0010\,\mathrm{m}\rangle$, $d \in \langle 0.06\,\mathrm{m}, 0.20\,\mathrm{m}\rangle$ and $G_C \in \langle 2500\,\mathrm{kPa}, 4500\,\mathrm{kPa}\rangle$. The above box conditions refer to current the technology of manufacturing and demands of the market.

These constraints were explicitly introduced into the optimization algorithm.

The optimization problem was solved using the DPEA (Distributed Parallel Evolutionary Algorithms) program (Burczyński and Kuś, 2004). The DPEA can operate on many subpopulations. The following evolutionary parameters and operators were defined: number of subpopulation 1, number of chromosomes 20, number of genes 4, probability of mutation 0.1, probability of crossover 1. A tournament selection was performed.

## 4. Optimization results: discussion

Within a general framework of the above presented formulation, a number of specific problems have been solved in order to study the influence of rigidity and variable cost of the core material, the temperature level and intensity of the external load. Table 4 describes these three classes of problems. Assumed temperatures on the lower and upper face sheets follow demands of design codes. Both face sheets have the same thermal expansion coefficient $\alpha = 0.000012\,1/{}^\circ\mathrm{C}$.

### 4.1. Optimal shear rigidity of the core

Pareto optimal solutions for different parameters $a$ (Table 4) specifying the cost of the core material as a function of $G_C$ are shown in Fig. 4.

The plots show how the total cost of a panel depends on its allowable length of the span $L$. A drastic increase of the cost is observed for $L > 3.5\,\mathrm{m}$. The

**Table 4.** The optimization problems

| $q$ [kN/m] | $\Delta T = T_2 - T_1$ [°C] | $a$ [1/(kPa m$^2$)] |
|---|---|---|
| Variable $a$ in the cost function of the core material | | |
| 0.385 | −55°C (summer), +50°C (winter) | 0.000 |
| 0.385 | −55°C (summer), +50°C (winter) | 0.050 |
| 0.385 | −55°C (summer), +50°C (winter) | 0.100 |
| Variable temperature $T_1$ and $T_2$ depending on the colour of the facing | | |
| 0.385 | −40°C (summer) | 0.050 |
| 0.385 | −55°C (summer) | 0.050 |
| – | −40°C/−65°C/−90°C | 0.000 |
| Variable load level $q$ | | |
| 0.385 | −40°C (summer) | 0.050 |
| 1.540 | −40°C (summer) | 0.050 |
| 0.75/1.00/2.00 | – | 0.000 |

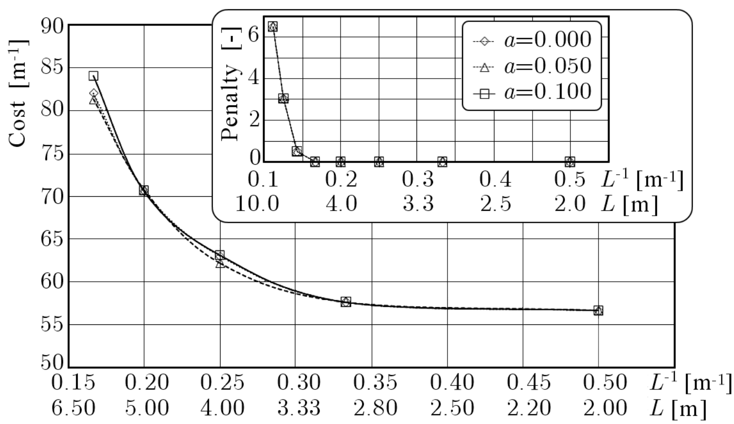


Fig. 4. Pareto optimal solutions for different costs of the core material

plots can be a good basis for a compromise decision as to what type of panel is worth producing. This compromise decision must account for a percentage demand for panels of greater $L$, however $L = 3.5$ m or 4.0 m can be suggested. Note that panels with $L > 5.9$ m are unacceptable because they violate the behavioral constraints in the presence of the assumed box conditions. This is clearly shown in the window of Fig. 4. The external penalty function is equal to zero only in the admissible region, where all constraints are satisfied. It is unexpected and interesting that all three plots shown in Fig. 4 are nearly identical. Hence, they demonstrate that the parameter $a$ in (3.5) and in Fig. 3

has no influence on the total cost of the panel. The reason can be found in the window of Fig. 5.

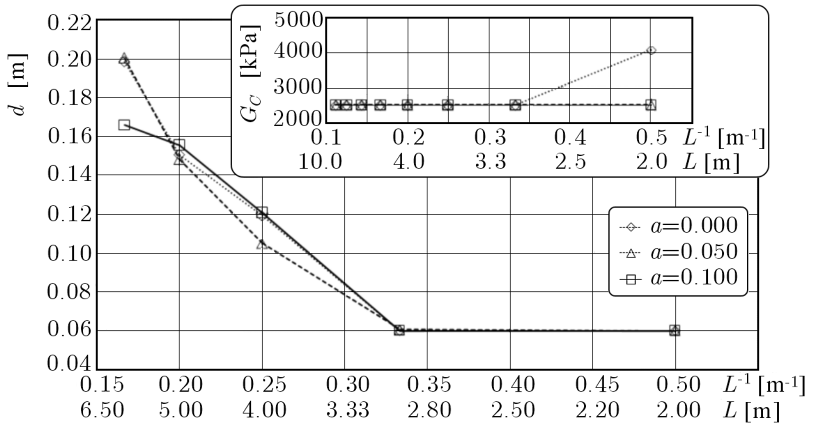


Fig. 5. Optimal slab thickness d for variable span length $L$

It can be seen that in the whole range of variable L the optimal value of the shear coefficient is equal to its lower bound $G = G_0 = 2500\,\mathrm{kPa}$. One exceptional point for $L = 2.0\,\mathrm{m}$ in Fig. 5 does not disturb the cost function $C$ in Fig. 4, because it is connected with the case when $a = 0$.

The plot of the optimal depth of the slab $d$ is shown in Fig. 5. It is interesting that it attains its lower bound $d = 0.06\,\mathrm{m}$ for $L \leqslant 3.0\,\mathrm{m}$. In the region $L > 3.0\,\mathrm{m}$ small differences in $d$ for variable $a$ result from the interchange of active constraints. Figure 6 shows the optimal thickness of facings.

The thickness of the upper facing attains its lower bound $t_1 = 0.4\,\mathrm{mm}$ for $L \leqslant 4.0\,\mathrm{m}$. The thickness of the lower facing $t_2$ slightly increases for $L = 3.0\,\mathrm{m}$. This is an effect of wrinkling at the middle support. For $L = 4.0\,\mathrm{m}$, the wrinkling effect is not so decisive because the plate depth $d$ is twice as great. In the region of $L > 4.0\,\mathrm{m}$, a rapid increase in thickness of both facings is observed.

### 4.2. The influence of temperature and load intensity on the optimal design

Pareto optimal solutions referring to a uniformly distributed loading combined with two temperature levels are shown in Fig. 7.

The Pareto solutions for the two load levels combined with temperature action (see Table 4) are presented in Fig. 8.

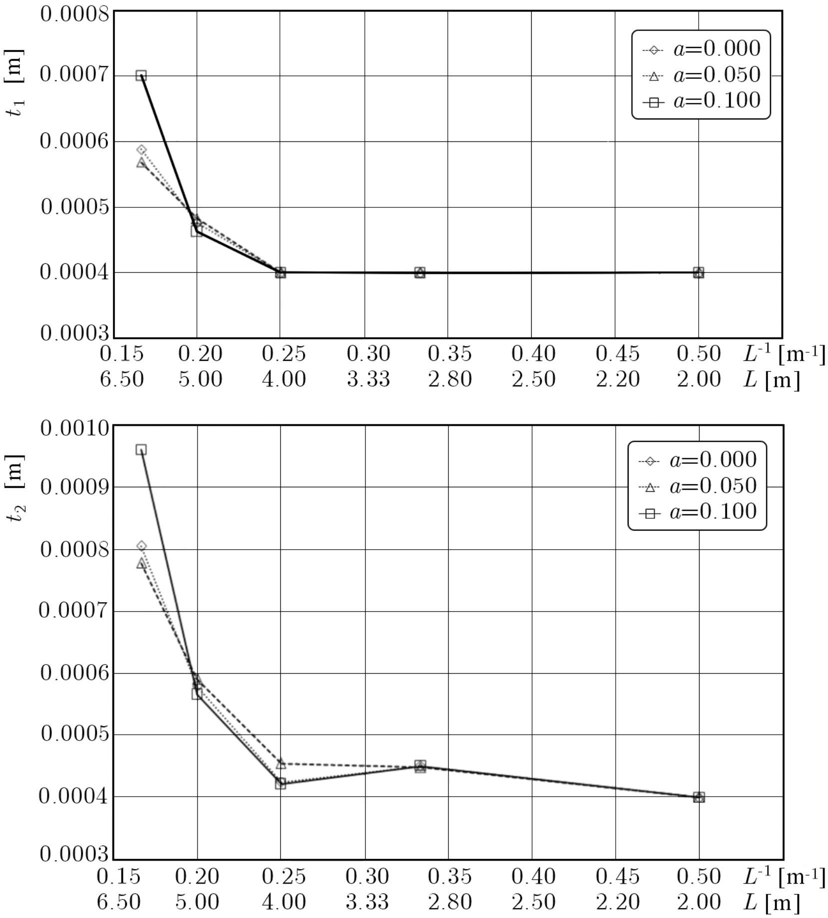


Fig. 6. Optimal thickness of facings for variable span length $L$: (a) upper facing $t_1$, (b) lower facing $t_2$

The respective penalty functions are shown in the windows. As it was expected, the cost increases for more severe loading and temperature conditions, yet the quantitative relations between $L$ and cost $C$ presented in Fig. 7 and Fig. 8 are valuable, because they enable making compromise decisions with regard to the final optimal design. It is surprising that variation of the temperature conditions does not influence the Pareto solution (Fig. 7) in the range $L < 3\,\mathrm{m}$. This can be explained by the fact that for small $L$ the optimal cross sectional variables attain their lower bounds. To better illustrate the influence of loading and temperature action on the optimal design of a sandwich slab next examples were solved. Figure 9 presents the Pareto optimal solutions

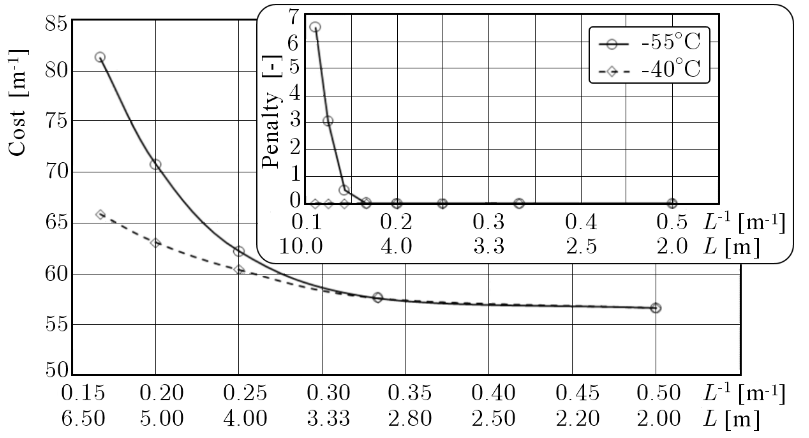


Fig. 7. Pareto optimal solutions for two temperature conditions. Window: respective penalty functions

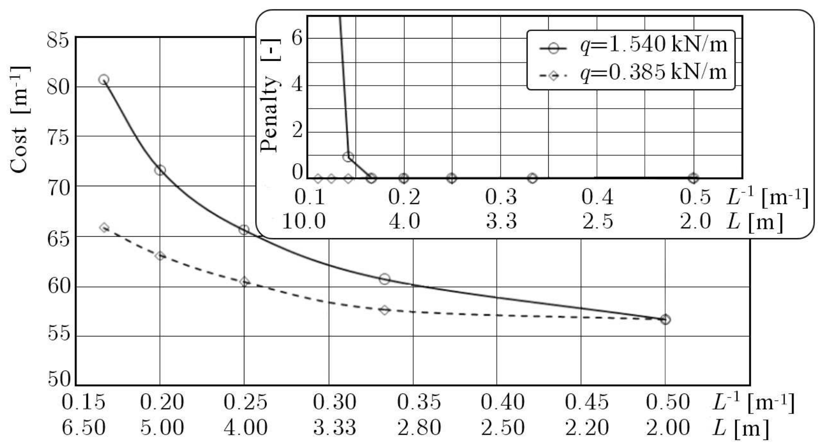


Fig. 8. Pareto optimal solutions for two load levels. Window: respective penalty functions

when the sandwich is subjected solely to the load or temperature actions of variable intensity specified in the Figure.

In the case of mechanical load, the plots (continuous lines) demonstrate an exponential increase of cost with respect to the span $L$. In the case of thermal loads, the wrinkling failure at the internal support is an active constraint. The plots (dashed lines) demonstrate a nearly linear relation between the span $L$ and the cost. Note that for small $L$, the temperature action may more significantly contribute to the total cost than the loading.

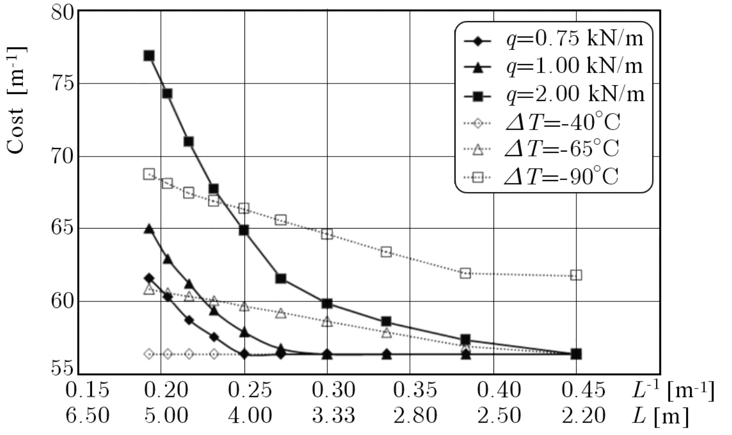


Fig. 9. Pareto optimal solutions for slabs subjected solely to loading (continuous line) or to temperature action (dashed line)

## 5. Concluding remarks

The paper presents an optimization problem, where a compromise between the cost of the product and its widest range of applications can be reached. The problem was formulated as a two-objective optimization of sandwich panels under prescribed loading and temperature induced distortions, aiming at the minimum cost and maximum length of the span. The length of the panel span plays two roles, namely the design variable and a component of the vector objective function. Pareto optimal solutions were computed. The associated optimal values of design variables were presented and discussed.

The quantitative representation of the Pareto solutions can be used in making compromise decisions by producers of sandwich panels with soft cores. In making final decisions, the actual cost factors must be used in the objective function, because Pareto solutions and associated optimal values of design variables depend on the cost function. Nevertheless, the analysis carried out in the paper allow us to draw general conclusions.

Since both the facings and the core contribute to the total cost and to the bearing capacity of the panels, they are mutually interrelated depending on external actions and length of the span. In principle, an increase of the thickness of facings generates higher cost than an increase of thickness of the core. Therefore, in the analysed cases, the optimal thickness of the facings was often equal to its lower bound. This is in agreement with general policy of producers to make the facings as thin as possible.

Several optimization problems (different forms of the cost function, various load and temperature actions) for a two-span panel were carried out. The analysis showed that for design parameters referred to the current technological limitations, a compromise optimal panel should be designed for the range of span lengths $L$ up to 3.5 m or 4.5 m.

A further increase of the span length leads to a drastic increase of cost generated by greater thickness of the facings and core. To respond to the market demand for panels applicable to cases of larger spans – reaching 6.0 m – a separate type of panels should be designed. Manufacturing one type of panels to cover the spans $L \in \langle 2.0\,\mathrm{m}, 6.0\,\mathrm{m}\rangle$ is not economically viable.

The paper also shows how the interaction of the external load and temperature influences the optimal solutions. Former experience of authors allows one to state that in the case when the temperature dominates, a smaller shear stiffness $G_C$ of the core is preferable. This is a general principle valid for multispan panels. For one-span simply supported panels, cores with higher $G_C$ are optimal. This is observed particularly in the case, when deflection under a mechanical load is an active behavioral constraint.

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## Optymalne projektowanie płyt warstwowych z miękkim rdzeniem

Streszczenie

W pracy podejmuje się problem optymalizacji wieloprzęsłowych płyt warstwowych z rdzeniem z poliuretanu (PUR) i okładzinami stalowymi lekko profilowanymi. Poszukuje się rozwiązań, które spełnią sprzeczne wymagania rynku, mianowicie: minimalizację typoszeregu płyt, maksymalizację zakresu ich zastosowania oraz minimalizację kosztu produkcji. Celem optymalizacji jest znalezienie parametrów geometrycznych i materiałowych płyt warstwowych, które minimalizują koszt oraz maksymalizują dopuszczalną rozpiętość dla ustalonych obciążeń i przy spełnieniu stanów granicznych nośności i użytkowalności. W wielokryterialnym sformułowaniu problemu optymalizacyjnego rozpiętość pełni dwie funkcje. Jest ona równocześnie zmienną projektową i składową wektora funkcji celu. Jako narzędzie optymalizacji wykorzystano algorytmy genetyczne. Ograniczenia nierównościowe wprowadzono do procedury optymalizacyjnej za pomocą zewnętrznej funkcji kary oraz jawnie.

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