

**EFFECT OF CHANGES IN THE THICKNESS OF  
A PERFORATED PLATE OF THE HEAT EXCHANGER ON  
ITS STRUCTURAL STABILITY<sup>1</sup>**

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One of the most important elements of heat exchangers are perforated bottoms. These are perforated plates of significant thickness, in which heating cartridge pipes are mounted. For the designers of heat exchangers, the strength analysis of the connection between the elements, which allows for the determination of the state of stress and calculation of displacements of the perforated bottom after fixing the pipes in it, is of special importance. It results from carried out computations that the maximum reduced stresses in the perforated bottom are significantly lower than the values of admissible stresses. In this paper, computations of stresses and strains for various thicknesses of the perforated bottom have been presented. As a result of a decrease in the thickness of the perforated bottom, its deflection increases, and thus the heating system pipes are more exposed to strains and losses of stability.

*Key words:* finite element method, heat exchangers, heating pipes

## 1. Introduction

Generally, heat exchangers are used in industry, mainly in heat and power generation. The number of pipes, and thus of holes in a perforated plate,

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can reach a few thousands. These holes are separated from one another by thin bridges. The structure of a plate, which is a round axially symmetrical plate mounted on an elastic base, prompts one to apply relevant equations of strength of materials and of the theory of elasticity. In order to do this, computations of the so-called equivalent rigidities and equivalent mechanical properties of the plate material have been made for many years. For the plane state of stress in the plate (e.g. disk tension), the results of this approximation agree with the results of experimental investigations. For actual working conditions (plate bending), an accurate enough approximation has not been obtained up till now.

The mounting of pipes by pipe expanding is an important problem in perforated bottoms. As a result of this process, a permanent plastic strain around holes takes place, which affects the shape of adjacent holes, and – consequently – the operation of the whole plate and all pipes. This paper presents calculations of stresses and strains for various thicknesses of perforated bottoms. Due to a decrease in the thickness of the perforated bottom, its deflection increases, making thus the heating cartridge pipes more liable to a loss of stability.

## 2. Object under analysis

Heat exchangers operate in very high temperatures. The fact that differences in temperatures of particular elements of heat exchangers has to be taken into account often results in serious problems in their designing process.

It is known that the volume of materials used to build heat exchangers increases with an increase in temperature (the dimensions of a jacket, perforated plates, etc., change). They are the so-called heat dilatations. The non-uniformity of temperatures that characterises each heat exchanger causes that pipes elongate in a different way than the jacket, and the perforated plate alters its diameters differently in comparison with the outer wall, etc. The differences in individual elongations that arise are sometimes followed by very serious strains of the material, which can lead to permanent deformations or even a rupture (Hobler, 1986).

The jacket, pipes of the heating cartridge with the perforated plate are a place that requires most care in this respect. The jacket reaches a temperature close to the temperature of the working medium with which it comes into contact, whereas the pipes have an intermediate temperature between the temperatures of both the media. It is followed by a considerable difference

in heat elongations and, in turn, these elongations can be the reason of high strains in the material the pipes, jacket and perforated plate are built of.

### 3. FEM calculations

#### 3.1. Object under analysis

The numerical computations have been carried out for a single-cycle heat exchanger, Fig. 1, whose model was developed on the basis of the technical specification of the Py-100-020 decarbonised water heater [2].

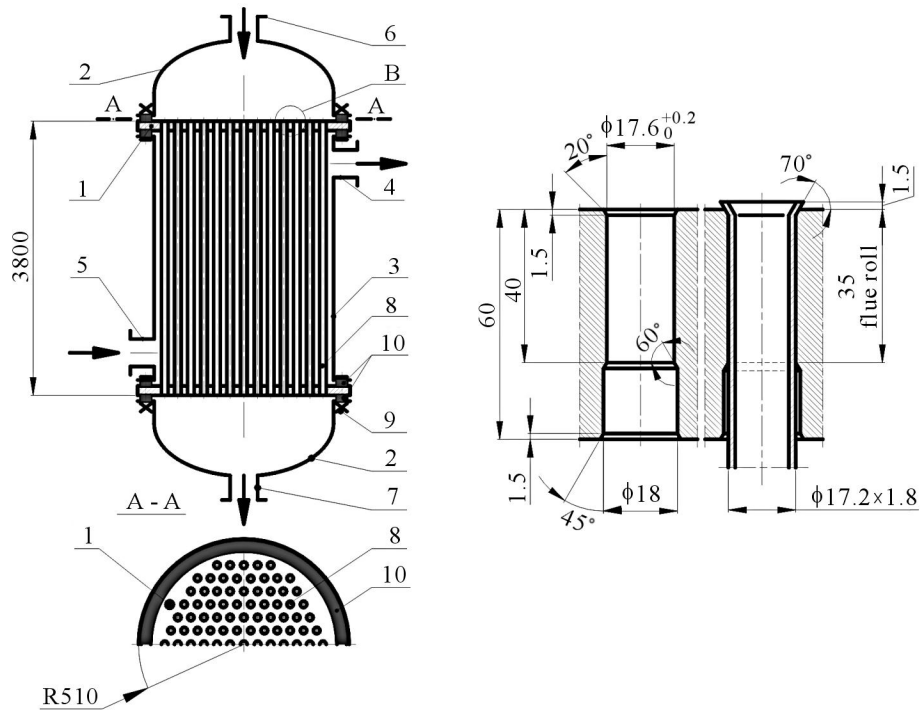


Fig. 1. Schematic view of the heat exchanger

The basic elements of the heat exchanger are as follows:

- perforated walls (1)
- bottoms (2)
- jacket (3)

- heating cartridge pipes (8)
- heating cartridge gaskets (10).

The perforated walls are mounted to the jacket and bottoms with screw fasteners (9). Connector pipes (6) and (7) supply water and take it off, whereas connecting pipes (4) and (5) – deliver steam. (B) shows the dimensions of a hole in the perforated bottom before expansion of pipes and the view of the pipe-perforated bottom connection after the pipe expansion.

#### 4. FEM models assumed

The finite element method with conventional assumptions have been applied in the numerical computations. Employing the conditions of symmetry in the computations, a model depicted in Fig. 2 that represents 1/8 of the heat exchanger has been developed. A non-linear model of the material with a linear reinforcement has been assumed. The model has been divided into 39243 8-node elements of the SOLID45 type (Chudzik, 2002).

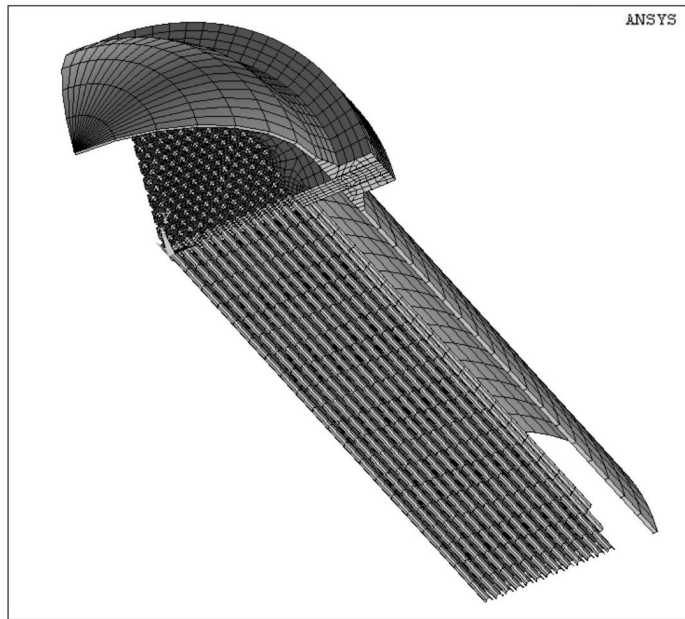


Fig. 2. Numerical model

The calculations of the strength of materials have been made for two thicknesses of the plate, namely

- according to the specification – 60 mm
- the thickness of the plate has been decreased by 1:2.

The calculations have been carried out for the case of a failure, and they have been aimed at the determination of strains occurring in the pipes, perforated bottoms and jacket. A failure occurs when the inflow and outflow of water is closed, and the heating steam is still delivered to the heat exchanger.

**Table 1.** Properties of the materials assumed in the FEM computations

	$R_e$ [MPa]	$E$ [MPa]	$\nu$ [-]	Coefficient of reinforcement
St36K PN-75/H-92123 bottom	196	$2 \cdot 10^5$	0.3	0.99
St36K PN-75/H-92123 jacket	196	$2 \cdot 10^5$	0.3	0.99
St41K PN-75/H-92123 perforated bottom	255	$2 \cdot 10^5$	0.3	0.99
I-K10 PN-74/H-74252 heat exchanger pipes	235	$2 \cdot 10^5$	0.3	0.99
Polonit PN-79/M-11022.02 gasket	–	$0.03 \cdot 10^5$	0.4	–

The load acting on the gasket has been calculated on the basis of the initial tension of screws in the bottom-gasket-perforated bottom-gasket-jacket connection, quoted in the technical specification of the heat exchanger.

**Table 2.** Values of loads that result from the assembly

Load acting on the gasket	7.1 MPa
Pipe-perforated bottom expanding pressure	176 MPa

**Table 3.** Emergency operation conditions of the heat exchanger

Parameters	Water chamber	Steam chamber
Pressure	–	$p_0 = 1.17$ MPa
Temperature	–	$T_0 = 523$ K

**Table 4.** Values of heat transfer coefficients

Steam-liquid convective heat transfer coefficient	$\alpha_w = 17 \text{ W/m}^2\text{K}$
Liquid convective heat transfer coefficient	$\alpha_w = 29 \text{ W/m}^2\text{K}$
Thermal conductivity coefficient through pipes	$\lambda = 49 \text{ W/mK}$
Thermal conductivity coefficient through the perforated bottom	$\lambda = 49 \text{ W/mK}$
Air convective heat transfer coefficient	$\alpha_w = 30 \text{ W/m}^2\text{K}$

## 5. Results of the numerical computations

In both analysis cases, the strain distributions in the heat exchanger and the stability modes corresponding to the critical load have been obtained. The results of the numerical computations for the first case are presented in Fig. 3 and Fig. 4, and for the second case in Fig. 5 and Fig. 6, correspondingly.

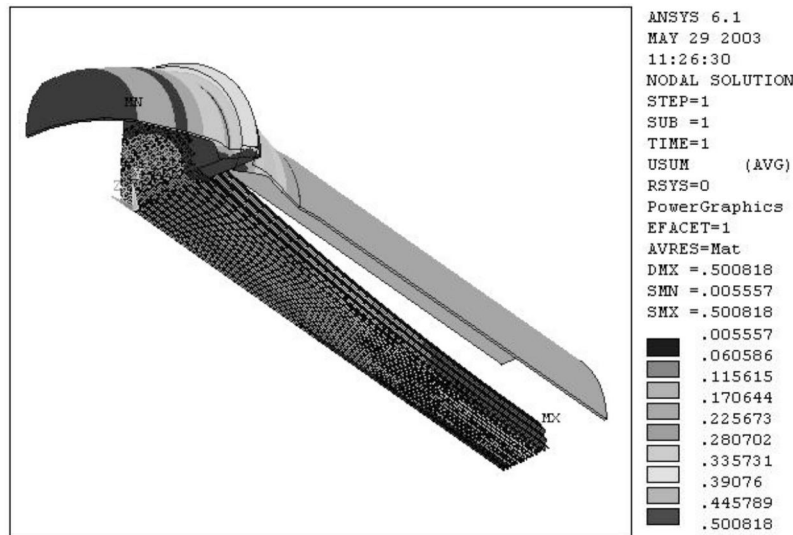


Fig. 3. Total strains of the heat exchanger [mm] (case no. 1)

Figure 3 shows the strains of the heat exchanger during the failure for the thickness of the perforated bottom equal to 60 mm. The pipes of the heat exchangers "collapse" towards the centre of the vessel.

Figure 4 presents the first mode of buckling that corresponds to the load which is four times higher than at the moment of the failure for the thickness

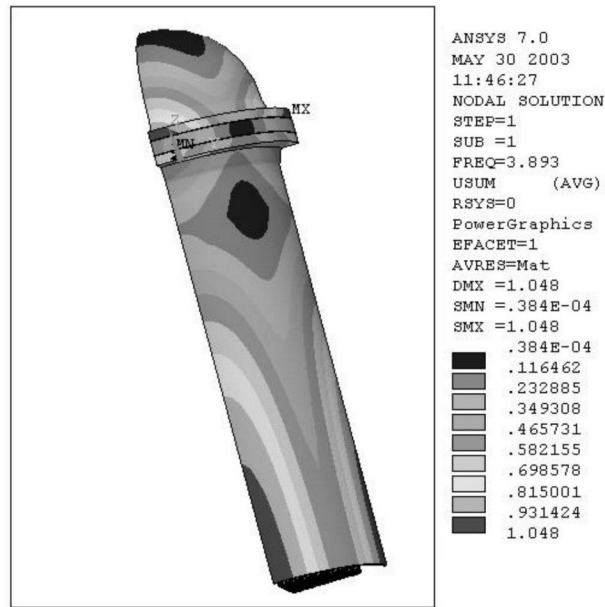


Fig. 4. Buckling mode corresponding to the lowest critical load



Fig. 5. Total strain of the heat exchanger [mm] (case no. 2)

of the perforated bottom equal to 60 mm. For these dimensions of the plate, the jacket of the heat exchanger is subjected to buckling.

Figure 5 depicts the strains of the heat exchanger during the failure for the thickness of the perforated bottom equal to 30 mm. The pipes of the heat exchanger undergo a deformation towards the vessel.



Fig. 6. Buckling mode corresponding to the lowest critical load

In Fig. 6, the first mode of buckling that corresponds to the load higher by approximately 30% during the failure, for the perforated bottom thickness of 30 mm, is presented. For these dimensions of the plate, the pipes situated in the central part of the plate are subjected to buckling.

## 6. Conclusions

On the basis of the numerical analysis, it can be stated that the vessel under analysis will not be subjected to buckling during a failure. In the case of the perforated bottom of the thickness equal to 60 mm, the critical load is almost four times higher than the actual load of the heat exchanger. When the critical load is achieved, the thick plate of the heat exchanger causes that the vessel jacket is subjected to a loss of stability as the first one.



In the case when the thickness of the perforated bottom plate is decreased by half, a margin of the critical load is about 30%, and the pipes that are closest to the axis of symmetry of the vessel is subjected to buckling under the critical load. The remaining elements of the heat exchanger have a four-time margin with respect to the critical load.

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### Wpływ zmian grubości płyty perforowanej wymiennika ciepła na stateczność konstrukcji

#### Streszczenie

Jednym z istotnych elementów wymiennika ciepła są dna sitowe – płyty perforowane o znacznej grubości, w których zamocowane są rurki wkładu grzewczego. Dla konstruktorów wymienników ciepła istotna jest analiza wytrzymałościowa połączenia pozwalająca na określenie stanu naprężeń oraz obliczenie przemieszczeń dna sitowego

po zamocowaniu rurek. Z przeprowadzonych obliczeń wynika, że maksymalne naprężenia zredukowane w dnie sitowym są dużo niższe od wartości naprężeń dopuszczalnych. W pracy przedstawiono obliczenia naprężeń i odkształceń dla różnych grubości dna sitowego. Na skutek zmniejszenia grubości dna sitowego zwiększa się jego napięcie, przez co rurki wkładu grzewczego bardziej są narażone na utratę stateczności.

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