Analysis of the Plate Heat Exchanger Failure

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Abstract— *The present article describes the effect of the pressure on the operability of the plate exchanger of the SWEEP IC 5 x 20 type as well as the results of the numerical analysis of the stresses at three different loads. The exchanger was tested at the testing pressure defined by the manufacturer (2 MPa). Maximum stresses were observed at the places of soldered joints on individual exchanger plates and the stress was highly above the carrying capacity of the solder joint. With an increasing distance from the place where the boards were connected, stress exhibited a sharp increase. The testing was also focused on the stresses on the surface area (shroud) of the exchanger at the permissible operating pressure of 1.6 MPa defined by the manufacturer. An important observation was the stress identified at the measured real operating pressure of 0.7 MPa ‒destruction of the exchanger used by the operator.*

Keywords— plate heat exchangers, numerical analysis of stress.

I. INTRODUCTION

The supply of heat via heat distribution systems may be accompanied with failures of plate heat exchangers which may occur in the following two cases:

1) At the formation of the total hydraulic impact in the system when the pressure in the conveyed fluid exceeds the permissible value of the stress in the material of the soldered joint;

2) If there is a hidden defect in the exchanger which is detected during the exchanger operation.

The article presents the case of analysing the plate heat exchanger destruction which occurred during the unsteady flow of water in the distribution system, characterised with time changes in the flow rate and pressure of water. Such unstable condition may be induced by various causes, most frequently by a failure of the pump or a sudden opening or closing of the regulating element in the pipeline during the steady flow of the fluid from the heat exchanger to a consumer. The shorter the time of closing the valve, the higher the proportion of kinetic energy of water transforms into the deformation work, and the more intensive the exerted hydraulic impact force in the system [1], [2], [3]. The intensity of the hydraulic impact may be one of the causes of the failures of the inner and outer walls of the exchanger, as shown in Fig. 1 and Fig. 2.

FIGURE 1: Deformation of the outer wall of the exchanger

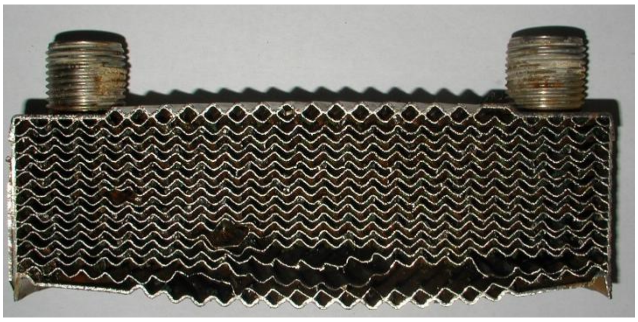

FIGURE 2: Deformation of the soldered joints of boards

II. ANALYSIS OF HYDRAULIC IMPACT

The outer smoother plate (shroud) of the analysed plate heat exchanger was 1.3 mm thick. This parameter of the exchanger plate was measured after the destruction thereof during the operation (Fig. 2). The inner profiled heat-transfer surface of the plate was 0.43 mm thick. The values required for the identification of an increase in the pressure in the distribution system were obtained by measuring the volumetric flow rate of water in the heat distribution system Q_V and subsequent analytical calculations. Table 1 contains the parameters corresponding to the water with the temperature of 55 °C.

The calculations were made using the following formulas:

$$\Delta p = \rho \cdot \kappa \cdot a_{\text{theor}} \cdot \upsilon \text{ (Pa)} \tag{1}$$

Where ρ is the density of water at the given temperature of water ($kg \cdot m^{-3}$); υ is the water flow velocity ($m \cdot s^{-1}$); a_{theor} is the theoretical speed of sound in the fluid ($m \cdot s^{-1}$); and κ is the pipeline elasticity coefficient (1).

The theoretical speed of sound in the fluid is calculated using the following formula:

$$a_{\text{theor}} = \sqrt{\frac{K}{\rho}} \, (\text{m} \cdot \text{s}^{-1}) \tag{2}$$

Where K is the volume modulus of elasticity of the fluid (for water it was $2.3 \cdot 10^9$ Pa).

The pipeline elasticity coefficient κ was identified using the following formula:

$$\kappa = \frac{1}{\sqrt{1 + \frac{K \cdot d_1}{E \cdot s}}} \tag{3}$$

Where E is the modulus of elasticity of the pipeline material (for steel pipes it was $2.1 \cdot 10^{11}$ Pa); d_1 is the external diameter of the pipeline; and s is the thickness of the pipe wall.

The real speed of sound in water was calculated as follows:

$$a_{\text{real}} = \kappa \cdot a_{\text{theor}} \, (\text{m} \cdot \text{s}^{-1}) \tag{4}$$

TABLE 1
PARAMETERS REQUIRED TO EXPRESS INCREASED PRESSURE IN THE DISTRIBUTION SYSTEM

ρ ($kg \cdot m^{-3}$)	a_{theor} ($m \cdot s^{-1}$)	κ (1)	a_{real} ($m \cdot s^{-1}$)	Q_V ($m^3 \cdot s^{-1}$)	υ ($m \cdot s^{-1}$)	Δp (MPa)
988	1,525.8	0.9756	1,488.6	0.00015	1.3263	**1.9505**

The length l of the circulation pipeline (from the heat exchanger to the customer) was 22.4 m. The blast (shock) wave would pass this route from the valve to the exchange, where it expands, in the time $t = l/a_{real} = 0.015$ s. The time in which the shock wave returns back to the place where it began, i.e., to the regulating element, was $T = 2 \cdot l/a_{real} = 0.030$ s.

The pressure in the supply pipeline of the distribution system was scanned using the ALMEMO MA 2390-8 measuring device by AHLBORN equipped with the FD 8214 17U pressure sensor and with the measuring range from 0 to 10 MPa. This device applied the pressure scanning frequency with the sampling frequency of 10 data per second. When investigating the presence of the total hydraulic impact in the given circuit, the used device for measuring the pressure is able to detect only a partially damped shock wave of such impact.

The values of the pressure p measured during the morning peak hours at the outlet from the exchanger ranged from 0.44 MPa to 0.59 MPa. During the evening peak hours, the pressure ranged from 0.44 MPa to 0.61 MPa.

The total increase in the pressure in the system Δp_{tot}, at a measurable pressure increase in the distribution system at the outlet from the heat exchanger, may be expressed as the pressure in the circulation pipeline p, increased in the value of the pressure Δp which may occur during the hydraulic impact, as follows:

$$\Delta p_{\text{tot}} = p + \Delta p \text{ (Pa)} \tag{4}$$

The manufacturer of the analysed exchangers tests their performance at the maximum pressure of 2.0 MPa and the permissible operating pressure of 1.6 MPa. The distribution system into which the exchanger was incorporated exhibited the deformation of the soldered joints of the inner plates of the exchanger at the operating pressure of only 0.7 MPa. The numerical calculation of the stress in the analysed plate exchanger was therefore made applying this particular pressure as the boundary condition.

During the measurable partial hydraulic impact, the pressure in the system increased in $\Delta p = 1.9505$ MPa. At the pressure in the system, for example, during the evening peak hours, with the value of 0.61 MPa, the total pressure of water in the system would be approximately 2.56 MPa; this exceeds the value of the testing pressure defined by the manufacturer in approximately 28 %.

The SWEEP IC 5 x 20 exchanger was analysed applying numerical mathematical methods in the COSMOS/M and ANSYS_10.0 environments [4], [5]. The boundary condition for the analysis of the stresses in the body of the exchanger was the pressure acting on the inner wall of the exchanger shroud, as shown in Table 2.

TABLE 2
BOUNDARY CONDITIONS OF THE ANALYSIS

Maximum testing pressure defined by the manufacturer (MPa)	Permissible operating pressure defined by the manufacturer (MPa)	Pressure during the real operation of the exchanger (MPa)	Measured deformation of the shroud δ (mm)
2.0	1.6	0.7	10.5

The calculation was made using the maximum testing pressure defined by the manufacturer, i.e., 2.0 MPa, and the permissible operating pressure, i.e., 1.6 MPa. Subsequently, the exchanger was tested for the condition corresponding to the real plastic deformation of the front external wall of the exchanger (Fig. 1). This deformation was identified on the basis of the measurement of the deformed exchanger and represented the value of 10.5 mm. It was detected by the operator during the real operation of the exchanger at the pressure of 0.7 MPa.

III. VERIFICATION OF THE STRESSES IN THE SWEEP IC 5 X 20 EXCHANGER

The basic parameters of the tested exchanger were: height – 190 mm; width – 73 mm; thickness of the inner heat-transfer surface 0.43 mm; thickness of the external plate – 1.3 mm; material – unknown nickel alloy; testing pressure – 2.0 MPa; permissible operating pressure – 1.6 MPa; and pressure during the real operation of the exchanger – 0.7 MPa.

The calculation of the stress was made for the area of linear and non-linear statics. The stress analysis in the area of the linear statics was performed applying the following boundary conditions: pressure on the inner side of the exchanger plate $p = 2$ MPa and 1.6 MPa; modulus of elasticity of the material $E = 2.1 \cdot 10^5$ MPa; and Poisson's ratio $\mu = 0.3$. For the non-linear stress analysis, the boundary conditions were as follows: pressure on the inner side $p = 0.7$ MPa; modulus of elasticity of the material $E = 2.1 \cdot 10^5$ MPa; and Poisson's ratio $\mu = 0.3$. For each calculation, the exchanger plate was fixed along its edges, in the openings of the water inlet and outlet on the exchanger, and at the points of soldered joints where the deformation was hindered in all directions ($x = y = z = 0$). With regard to the shape of the exchanger, the analysis was performed as a symmetrical task. The result of the stress analysis of the heat exchanger was the information on the *stress distribution* across the exchanger shroud and the *deformation extent* at individual loads.

At the *testing* pressure of 2 MPa, maximum stresses were formed at the points of soldered joints on individual exchanger plates. As the distance from the plate joints increased, the stress sharply increased as well. At the given pressure, the peak stress σ reached the value of as much as 425 MPa (Fig. 3).

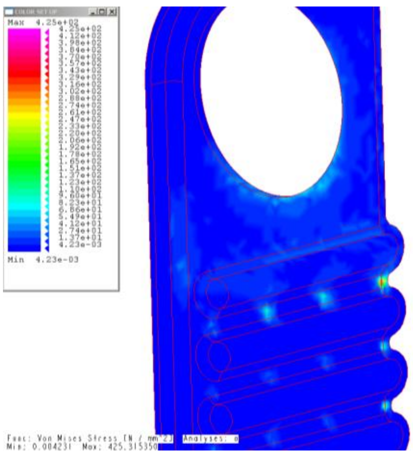

FIGURE 3: Detail of the stress distribution in soldered joints (p = 2 MPa)

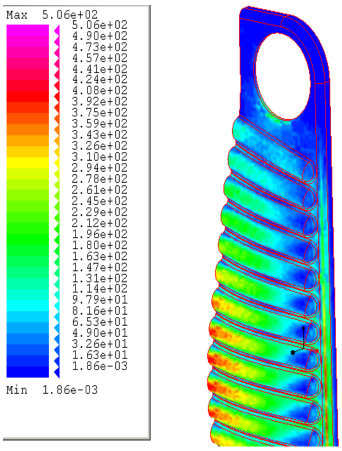

FIGURE 4: Stress distribution on the external wall (p = 1.6 MPa)

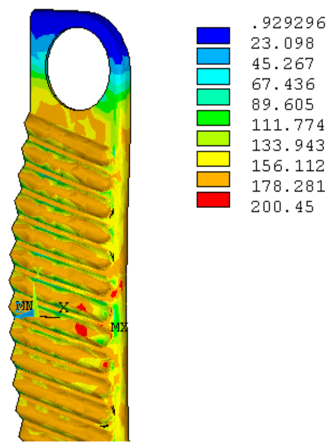

FIGURE 5: Stress distribution on the external wall (p = 0.7 MPa)

At the *permissible* operating pressure declared by the manufacturer (1.6 MPa), the stress was observed on the surface area (shroud) of the exchanger. The maximum stress σ on the wall reached the value of as much as 506 MPa (Fig. 4).

At the *measured* operating pressure of 0.7 MPa, the stress was observed in the area of the non-linear statics. For the available solders, the distribution of the yield stress values ranged from Re = 170 MPa to Re = 1,030 MPa. Due to unknown physical properties of the used solder, the calculation of the yield stress was made on the basis of the known deformation of the exchanger wall representing 10.5 mm. The approximation method of calculation, with the known value of the deformation of the exchanger front plate, the corresponding value of the yield strength was identified as 178 MPa. This value was affected by the number of kinematic reinforcements. The given value of the yield strength corresponds to the NICOLOY 800HT solder material. With this material and at the pressure of 0.7 MPa, the stresses in the top plate of the exchanger are shown in Fig. 5. The maximum stress σ on the exchanger plate achieved the value of 200 MPa.

On the basis of the performed calculations and analysis it is possible to state that the design stress in the walls of the front and rear plates of the exchanger must not exceed the following value:

$$\sigma_{\mathrm{D}} = \frac{\mathrm{Re}}{1.5} = \frac{178}{1.5} = 115\ \mathrm{MPa} \tag{5}$$

Corresponding to the design pressure p_{d} of 0.47 MPa. The constant 1.5 represents the safety criterion. Once this value is exceeded, plastic deformation gradually develops in the exchanger plates and the exchanger loses its function.

IV. CONCLUSION

Provided that the soldered joints exhibit full and functional carrying capacity, the exchanger may be pressurised up to the value of $p = 2$ MPa. At such pressure, the walls of the front plate are exposed to the stress σ of only 10 MPa which is sufficiently below the permissible value $\sigma_{\mathrm{P}} = 115$ MPa. All the primary forces are then transferred through the soldered joints. Therefore, the exchanger components that are loaded the most are the soldered joints. The exchanger works reliably until these joints are destructed. The soldered joints become destroyed, and hence the exchanger losses its function, in the following two cases:

1) A sudden increase in the pressure in the exchanger above the carrying capacity of the soldered joint;
2) Due to the effects of the cyclic fatigue life of the soldered joint which gradually loses its carrying capacity due to the cyclic loading, and after the material reaches its kinetic carrying capacity, the joint becomes destructed and the exchanger walls gradually experience plastic deformation. This is a well-known correlation between the “stress (force) and the number of cycles” equal to the fatigue life.

Reliable operation of SWEEP exchangers, IC 5 x 20 types, depends on the carrying capacity of the soldered joints. As operators cannot affect this qualitative property of the exchanger, the problem of exchanger deformation, with the given solder material and during the operations at pressures above 0.47 MPa, must be addressed by the manufacturer.

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