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POLISH ACADEMY OF SCIENCES



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## 113

Selected papers from the International Conference
on *Turbines of Large Output*
devoted to 100th Anniversary of
Prof. Robert Szewalski Birthday,
Gdańsk, September 22-24, 2003

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*Transactions of the Institute of Fluid-Flow Machinery* have primarily been established to publish papers from four disciplines represented at the Institute of Fluid-Flow Machinery of Polish Academy of Sciences, such as:

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TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

No. 113, 2003, 3-4

## Editorial

These Special Issues of the *Transactions of Fluid-Flow Machinery*, Nos. 113 and 114 contain selected papers from the International Conference on *Turbines of Large Output* devoted to commemorate 100$^{\text{th}}$ Birthday Anniversary of Prof. Robert Szewalski. The conference was held in Gdańsk, Poland on September 22-24, 2003.

The Conference is a continuation of previous conferences held at the Institute of Fluid-Flow Machinery PAS in former years dedicated to technology of steam turbines. Series of conferences bearing the same name took place in the years 1962, 1965, 1968, 1993. In 1997 organised has been a conference on steam turbines and related topics, but with a slightly amended title – *Problems of Fluid-Flow Machinery.* At present at the Institute of Fluid Flow Machinery there are conduced research of fundamental character encompassing both the issues related to steam turbines and fundamentals of power engineering.

Organisers of the present conference have returned to the traditional name, i.e. Conference on *Turbines of Large Output* to mark the respect to the memory of Professor Robert Szewalski (1903-1993), the founder and a director of the Institute of Fluid-Flow Machinery for several years, and initiator of the conference series devoted to the problems of steam turbines.

The Editors are very grateful to the referees of the papers presented in this issue of the *Transactions of Fluid-Flow Machinery*: J. Badur, W. Batko, E. S. Burka, J.T. Cieśliński, P. Doerffer, A. Gardzilewicz, B. Grochal, J. Kiciński, G. Kosman, T. Król, J. Krzyżanowski, J. Mikielewicz, A. Neyman, W. Ostachowicz, R. Puzyrewski, R Rządkowski, J. Świrydczuk, M. Trela, T. Uhl and Z. Walczyk.

We appreciate cordially the manifested authors interest in our conference. Special thanks are conveyed to Ms 'Maya' Bagińska, for her fruitful editorial help related to directing the papers to reviewers and correspondence with the authors of contributions.

*Prof. Jarosław Mikielewicz*
Editor-in-Chief

*Dr Edward Śliwicki*
Managing Editor

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

No. 113, 2003, 107-118

DARIUSZ BUTRYMOWICZ* and MARIAN TRELA

# Problems of condensation heat transfer in power plant heat exchangers

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## Abstract

The paper deals with the problems of the influence of inert gases, vapour superheating, and fouling on condensation heat transfer in power plant heat exchangers. The performance estimation of a recuperative heat exchanger under conditions of fouling and presence of inert gases and vapour superheating is extremely important for the diagnostic codes of the turbine units. Basing on the one dimensional model the influence of fouling and air-concentration on the distribution of both the pertinent heater temperatures and heat flux was investigated. It appeared that inert gases play a dominant role in the case when its concentration is sufficiently high. It may be concluded that the proposed 1D model provides a very efficient method for heat transfer evaluation of heat exchangers. The effect of a strong vapour superheating on performance of heat exchanger has been discussed. It may be concluded that non-condensation zone in these heaters depends not only on vapour superheating but also on the heat transfer surface area.

**Keywords:** Power plant heat exchangers; Condensation heat transfer; Inert gases; Fouling

## Nomenclature

$A_o$ – outside surface area of tubes, $m^2$
$c_{pv}$ – specific heat of vapour, J/(kgK)
$c_{pw}$ – specific heat of water, J/(kgK)
$D_i$ – tube inner diameter, m
$D_m$ – log mean tube diameter, $D_m = (D_o - D_i) / \ln (D_o/D_i)$, m
$D_o$ – tube outer diameter, m
$h_{fg}$ – specific enthalpy of vaporisation, J/kg
$k$ – overall heat transfer coefficient, $W/(m^2K)$

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| | | |
|---|---|---|
| $L$ | – | current path length of water inside a tube, m |
| $L_w$ | – | total path length of water inside a tube, m |
| $\dot{m}$ | – | mass flow rate, kg/s |
| $n_b$ | – | number of passages of water in a heat exchanger, |
| $n_t$ | – | total number of horizontal tube rows, |
| $n_{tb}$ | – | number of horizontal tube rows in one passage, |
| $p$ | – | pressure, Pa |
| $q$ | – | heat flux, $q = \dot{Q}/A_o$, W/m$^2$ |
| $\dot{Q}$ | – | heat transfer rate, W |
| $R_a$ | – | thermal resistance of inert gas, m$^2$K/W |
| $R_{fi}$ | – | thermal resistance of fouling inside a tube, m$^2$K/W |
| $R_{fo}$ | – | thermal resistance of fouling outside a tube, m$^2$K/W |
| $R_t$ | – | thermal resistance of tube wall, $R_t = (D_o - D_i)/2\lambda_t$, m$^2$K/W |
| $T$ | – | temperature, K |
| $\delta T_o$ | – | terminal temperature difference, Eq. (1), K |
| $\alpha_v$ | – | vapour-side heat transfer coefficient, W/(m$^2$K) |
| $\alpha_w$ | – | water-side heat transfer coefficient, W/(m$^2$K) |
| $\beta$ | – | row effect coefficient, dimensionless |
| $\xi$ | – | coefficient defined by Eq. (11), dimensionless |
| $\lambda_t$ | – | tube wall thermal conductivity, W/(mK) |

## Subscripts

| | | |
|---|---|---|
| $a$ | – | air; inert gases, |
| $b$ | – | interface, |
| $c$ | – | condensate, |
| $s$ | – | saturation for pure vapour, |
| $to$ | – | outside tube surface, |
| $v$ | – | vapour, |
| $vg$ | – | vapour at the boundary between condensation and non-condensation zones, |
| $vs$ | – | local saturation conditions for vapour containing inert gases, |
| $w$ | – | water inside tubes, |
| $wi$ | – | water at the inlet to the heat exchanger, |
| $wg$ | – | water at the boundary between condensation and non-condensation zones, |
| $wo$ | – | water at the outlet from the heat exchanger. |

# 1 Introduction

The influence of fouling as well as the inert gas content in vapour on the heat transfer rate in power plant heat exchangers is considered in the paper. This type of heat exchanger is characterized by the following features:

- there exists a strong relationship between the heat transfer coefficient on the surface temperature, where the condensation process takes place,
- the condensation heat transfer coefficient depends strongly on both the

steam velocity and the tube flooding (the so called row effect),

- the mixture (vapour – inert gas) temperature is equal to the local vapour saturation temperature which depends on the local gas content in the mixture,
- the interface temperature (condensate layer-vapour and air mixture) depends on the local gas content in the mixture,
- the strong vapour superheating causes the non-condensation zone in the heat exchanger.

The above features make difficult the proper calculation of the performance of the regenerative heaters. It is widely assumed [1, 2], in the classical theory of heat exchangers, that heat transfer coefficients are either constant throughout the heat exchanger or explicitly described in function of fluids temperatures on both sides of the wall dividing both media. Additionally, classical theories do not *consider the flow of both media with the change of phases.* Each of such cases involves an individual approach and it is difficult to work out a general mathematical model to tackle that problem [1, 2].

A general schematic of the considered regenerative heater is presented in Fig. 1. A heated water, with the flow rate $\dot{m}_w$, flows through a system of a horizontal tube bundle, whereas the steam condenses on the outer wall of that tube bundle.

The diagnostic heat exchanger analysis requires determination of the outlet parameters ($\dot{m}$, $T_{wo}$, $T_c$) at a given heat transfer area and the inlet parameters ($\dot{m}$, $p_v$, $T_v$, $T_{wi}$). In such analysis the heat transfer rate needs to be calculated and one of the most important parameters describing performance of the heat exchanger is the so called terminal temperature difference:

$$\delta T_o = T_s - T_{wo}. \tag{1}$$

The authors have concentrated their efforts on the simplest models of heat exchangers, which allowed to incorporate adequate calculations into the diagnostic analysis.

# 2 Mathematical models of power plant heat exchangers

## 2.1 Global and lumped-parameter models

The present authors showed, [3], that at least three types of the models of power plant heat exchangers suitable for diagnostic analysis can be derived.

The simplest one is the co called global model. It is assumed in this model that the value of the overall heat transfer coefficient $k$ for the entire heat exchanger is known. The authors showed, [3], that the global model is usually too simple and at the same time least accurate for the considered heat exchangers.

The lumped-parameter model considers the system of algebraic balance equations. The present authors showed, [3], that such model for power plant heat exchanger has to consist of at least three balance equations. This system of equations ought to be solved numerically with respect to the outlet water temperature, mean outside temperature of the tube surface and the steam mass flow rate.

A significant drawback of the lumped-parameter model comes from its lack of possibility to include both the local effects stemming from the presence of inert gases in the steam and the row effect. This model assumes the logarithmic temperature distribution of heated water. It will be shown later, in the case of a presence of inert gases in the vapour, that this is a significant simplification.

## 2.2 One-dimensional model of power plant heat exchanger

The one-dimensional model of the power plant heat exchanger takes into account variations of the conditions of heat transfer along the current path of water inside tubes. This has a particular importance in the case of analysis of the influence of inert gases, when the conditions of heat transfer are highly non-homogeneous. Due to the presence of inert gases there is also a necessity to consider the change of the local saturation temperature $T_{vs}$ as well as temperature of the interface $T_b$. Due to the possibility of inclusion of the above parameters in the particular flow passages as well as the local row effect coefficient $\beta$ in the model (in the direction perpendicular to the water flow passage), it may be regarded as a quasi-two-dimensional model. The model utilises the following differential balance equation:

$$\dot{m}_w c_{pw} \frac{dT_w}{dA_o} - k\left(T_w,\, T_b,\, T_{to},\, \dot{m}_v\right) \cdot \left(T_{vs} - T_w\right) = 0, \tag{2}$$

where $T_w$ is the local water temperature, and the overall heat transfer coefficient is provided by the equation:

$$\frac{1}{k} = \frac{1}{\alpha_v\left(T_b,\, T_{to},\, \dot{m}_v\right)\beta\left(\dot{m}_v\right)} + R_a\left(T_b,\, \dot{m}_v\right) + R_{fo} + R_f \frac{D_o}{D_m} + \left(\frac{1}{\alpha\left(T_w\right)} + R_{fi}\right)\frac{D_o}{D_i}. \tag{3}$$

The resistance $R_a$ can be determined from a well validated, and often used in numerical calculations, Berman relation [4].

The considered model is supplemented with the following relations:

- equation linking the heat transfer coefficient on the steam side $\alpha_v$ with temperatures $T_{to}$, $T_b$ and the steam mass flow rate $\dot{m}$

$$d\dot{m}_v \left[h_{fg} + c_{pv} \left(T_v - T_{vs}\right)\right] = \alpha_v \, dA_o \left(T_b - T_{to}\right), \tag{4}$$

- energy balance equation for the elementary part of the heater tube:

$$d\dot{m}_v \left[h_{fg} + c_{pv} \left(T_v - T_{vs}\right)\right] = \dot{m}_w c_{pw} dT_w, \tag{5}$$

- equation linking the local steam partial pressure with the air content:

$$p_{vs} = \frac{p_v}{1 + 0.622 \dfrac{\dot{m}_a}{\dot{m}_v}}. \tag{6}$$

Based on equation (6) the local value of steam saturation temperature $T_{vs}$ can be determined under condition that the distribution of air mass flow rate $\dot{m}_a$ is known. Dependent local variables, which must be determined from the above system of equations are: water temperature $T_w$, steam mass flow rate $\dot{m}_v$, temperature of interface condensate film-air and vapour mixture surface $T_b$, temperature of tube outside surface $T_{to}$ and the steam saturation temperature $T_{vs}$.

In the case of multi-pass heat exchangers with horizontal tubes the conditions of heat transfer on the steam side change remarkably at the end of each pass due to a sudden change of the tube location in vertical direction (the elbow effect). This causes discontinuities in heat transfer conditions and makes difficult the above differential equations to solve. In order to overcome this problem, the above system of equations has been solved by the authors by transformation of differential equations into the difference equations. The details of such transformation can be found in [5]. Averaged value of the row coefficient $\beta$ for the given tube-pass $n_b$ has been determined from the formula:

$$\beta\left(n_b\right) = \frac{1}{n_t - \left(n_{tb} - 1\right) - 1} \int\limits_{n=(n_b-1)n_{tb}+1}^{n=n_t} \beta\left(n_t - n + 1\right) dn, \tag{7}$$

where the local value of the row coefficient $\beta$ has been determined from Kern relation [6]:

$$\beta\left(n\right) = n^{\frac{5}{6}} - \left(n-1\right)^{\frac{5}{6}}. \tag{8}$$

The application of the above simplified form of the row coefficient is justified by the fact that for the case of one-dimensional model, which considers the local influence of thermal-hydraulic parameters on heat transfer, such coefficient

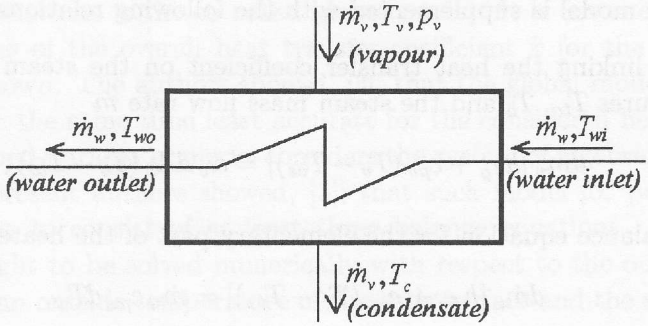


Figure 1. Schematic diagram of the heat exchanger (feed-water heater).

describes solely the effect of the condensate flow from the tubes located above the considered one. The effects linked with the velocity change of steam flow and the change of concentration of inert gases as well as the changes of saturation and interface temperatures have already been incorporated into the model. For that reason, the Kern relation, based on the experimental data, which estimated higher values of $\beta$ – gives better results than the Nusselt relation derived theoretically in the form:

$$\beta(n) = n^{\frac{3}{4}} - (n-1)^{\frac{3}{4}}. \tag{9}$$

The one-dimensional model of the power plant heat exchanger, discussed above, ought to be regarded as the most convenient for the case of diagnostic analysis. Then at a relatively small effort of numerical calculations it enables to take into account all important effects (in respect to heat transfer).

## 3 Influence of surface fouling and inert gases

The influence of surface fouling and the presence of inert gases on the performance of the power plant heat exchanger is based on the example of the one of the low pressure regenerative feed-water heaters of the turbine unit of 200 MW (Fig. 2). Such heater is mounted in the casing of condensers and its heat transfer surface is in form of the bundle of horizontal tubes. The basic geometrical dimensions of the sample heater, for which the heat transfer calculations have been performed, are as follows: number of tubes – 670; outside tube diameter – 19 mm; internal tube diameter – 17 mm; tube material - brass; length – 3.642 m; width – 0.358 m; number of vertical rows in the bundles – 52; number of water-passes – 4; inlet dimensions of the steam socket: 2.548 m×0.200 m; tube spacing – 44 mm

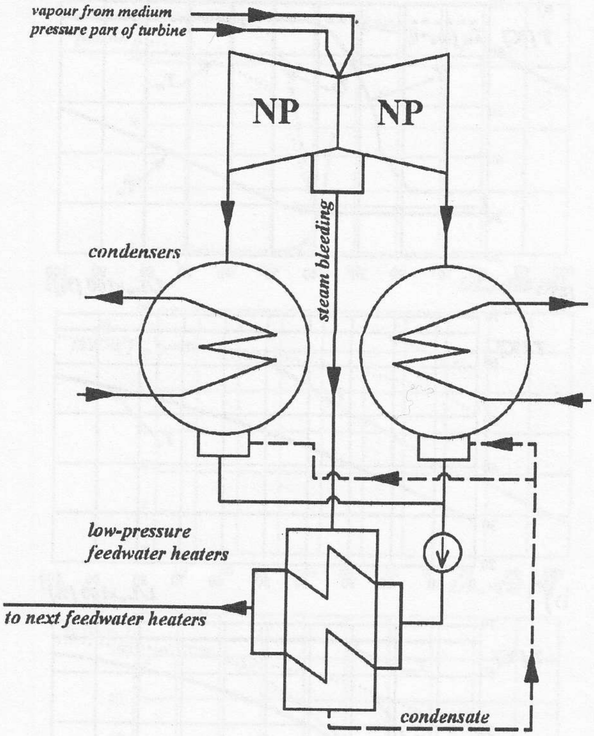


Figure 2. A part of schematic diagram of the power station installation typical for the turbo-units of the output of 200 MW.

in the triangle configuration. The following quantities have been assumed in the numerical calculations: mass flow rate of water $\dot{m}_w = 63.74$ kg/s, inlet water temperature $T_{wi} = 28.51°C$, inlet steam temperature $T_v = 93.06°C$, the inlet steam pressure $p_v = 0.0271$ MPa. The details of calculation procedure have been summarised in [5].

Temperature distribution in the sample regenerative heater are presented in Fig. 3 for three cases: fouled heat exchanger in the presence of air in the steam (Fig. 3a), fouled heat exchanger without air in the steam (Fig. 3b) and the clean heater also without the influence of inert gases (Fig. 3c). It results from the presented graphs that the distribution of water temperature $T_w$ in the heater operating under presence of air (Fig. 3a) strongly departs from the logarithmic behaviour, pertinent to the operation of the clean heater without air (Fig. 3c). This justifies the necessity of using a more complex numerical one-dimensional model.

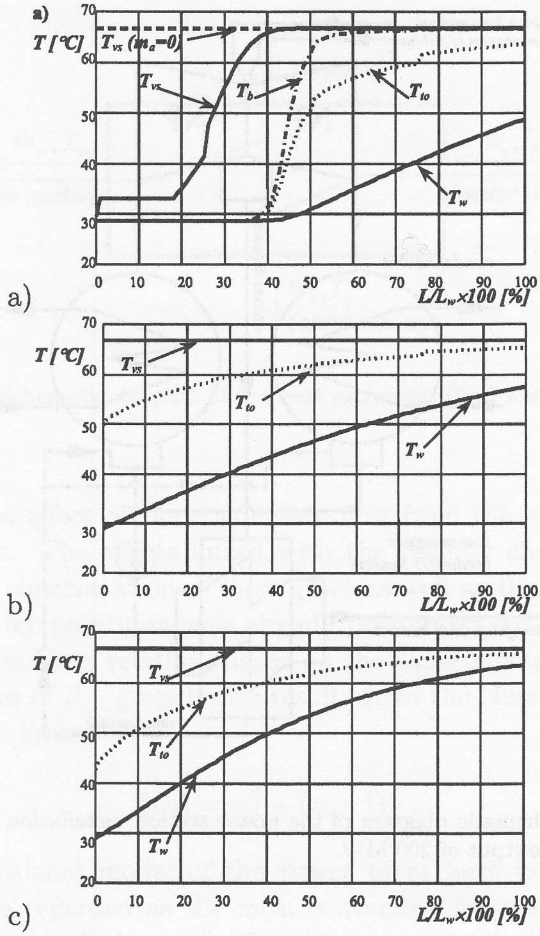


Figure 3. Variation of temperatures in feed-water heater along the water flow path inside tubes for: a) $R_{fi} = 1.5 \cdot 10^{-4}$ m$^2$K/W, $\dot{m}_a = 1 \cdot 10^{-3}$ kg/s; b) $R_{fi} = 1.5 \cdot 10^{-4}$ m$^2$K/W, $\dot{m}_a = 0$ c) $R_{fi} = 0$, $\dot{m}_a = 0$.

The corresponding variation of the heat flux $q$, along the water path in the tubes is presented in Fig.4. A characteristic feature of these distributions comes from the presence of the maximum of $q$ on these curves. This is justified by the fact that in the lower part of the heater (dimensionless water path $L/L_w$ is approaching zero) there are no favourable conditions for vapour condensation due to: low steam velocity, tube flooding, highest content of air in vapour (for the case "a"). A very low heat flux $q$ appears in the case "a" even though the temperature difference between the water and the vapour steam is the highest one at that zone. On the other hand, in the upper part of the heater (dimensionless

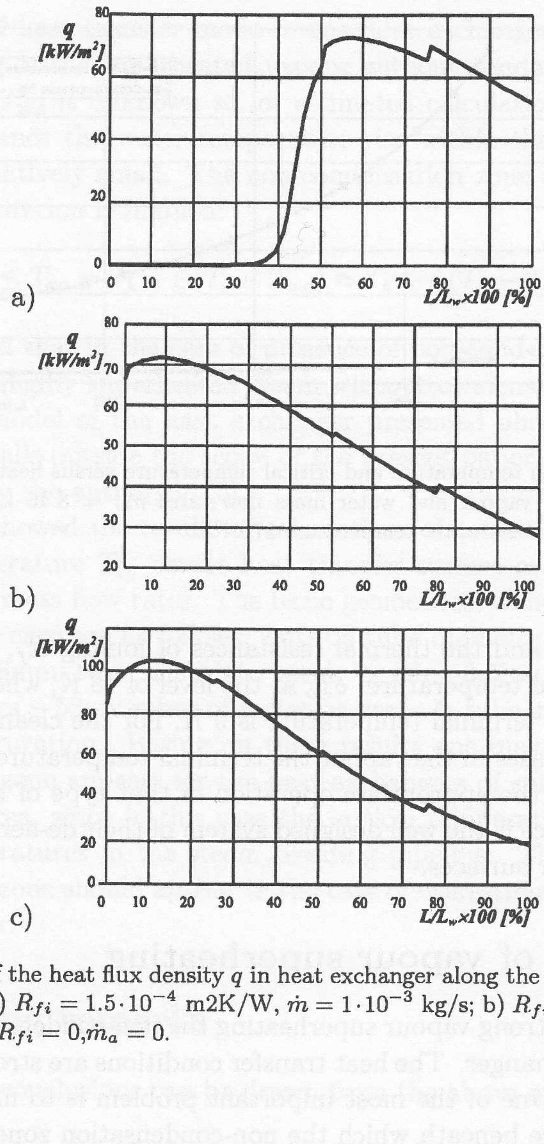


Figure 4. Variation of the heat flux density $q$ in heat exchanger along the water flow path inside tubes for: a) $R_{fi} = 1.5 \cdot 10^{-4}$ m2K/W, $\dot{m} = 1 \cdot 10^{-3}$ kg/s; b) $R_{fi} = 1.5 \cdot 10^{-4}$ m2K/W, $\dot{m}_a = 0$; c) $R_{fi} = 0, \dot{m}_a = 0$.

water path $L/L_w$ is approaching unity) the conditions for heat transfer are more favourable, however the temperature difference between steam and the heated water, which is the driving force for heat transfer is the lowest.

In the case of the heater operating under the presence of air in vapour (Fig. 4a), the maximum of heat flux $q$ takes place in the upper part of the heat exchanger, whereas in the case of lack of inert gases, independently from the existence of surface fouling, the maximum falls in the lower part of the heat exchanger (Fig. 4b, c).

Air content in vapour, assumed in the calculations, corresponding to the ratio

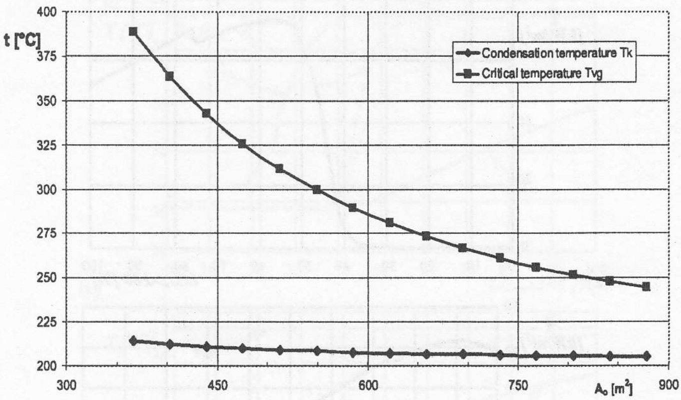


Figure 5. Condensation temperature and critical temperature versus heat transfer surface area for constant vapour and water mass flow rates $\dot{m}_v$ = 8.46 kg/s, $\dot{m}_w$ = 122 kg/s respectively, inlet water temperature 170.4°C.

$(\dot{m}_a/\dot{m}_v) = 0.31\%$ and the thermal resistances of fouling $R_f$, cause the appearance of the terminal temperature: $\delta T_o$ at the level of 18 K, whereas in the lack of the inert gases the terminal temperature is 9 K. For the clean heater, operating without the inert gases in the vapour the terminal temperature is about of 3.5 K. In order to ensure the appropriate operation of that type of heat exchangers of principal importance is the well designed system of their de-aeration and cleaning of the heat transfer surfaces.

# 4 Influence of vapour superheating

In the case of strong vapour superheating the non-condensation zone may exists in the heat exchanger. The heat transfer conditions are strongly unfavourable within this zone. One of the most important problem is to find out the critical vapour temperature beneath which the non-condensation zone exists.

The following relation can be derived for critical vapour temperature in the boundary between condensation zone and non-condensation zone in the heat exchanger:

$$\frac{T_{vg} - T_k}{T_k - T_{wg}} = \frac{\dfrac{1}{\alpha_v}}{\left(\dfrac{1}{\alpha_i} + R_{fi}\right)\dfrac{D_o}{D_i} + R_t\dfrac{D_o}{D_m} + R_{fo}} = \xi. \tag{10}$$

It is important to note that Eq. (10) describes the local values of heat transfer for the boundary between the two zones. Hence this equation can be treated as the criterion of heat transfer mode in the heat exchanger, i.e. whether the convective cooling of the superheated vapour without condensation is possible. The temperature $T_{wg}$ is unknown so for estimated calculations one can assume that $T_{wg} \approx T_{wo}$ since the water temperature rise within the non-condensation zone is usually relatively small. The non-condensation zone does not exist only if the simplified criterion is fulfilled:

$$T_v \leq T_{vg} = T_k + \xi \left( T_k - T_{wg} \right) \approx T_k + \xi \left( T_k - T_{wo} \right). \tag{11}$$

It must be stressed that in the case of presence of non-condensation zone a convective cooling of highly superheated steam without condensation occurs, so the one-dimensional model of the heat exchanger presented above requires further extension, which falls outside the scope of the present paper. That problem has been considered by the authors in [5].

In Fig. 5 are showed the results of calculations of condensation temperature and critical temperature $T_{vg}$ versus heat transfer surface area $A_o$ for constant vapour and water mass flow rates. The basic geometrical dimensions of the sample heater for this case are as follows: outside tube diameter – 32 mm; internal tube diameter – 28mm; tube material – steel; length – 3.5 m; number of vertical rows in the bundles – 52; number of water-passes – 4; tube spacing – 44 mm in the triangle configuration. Basing on these results one may conclude that the non-condensation zone appears for the heat exchangers of sufficiently large heat transfer surface area, since in this case the critical temperature $T_{vg}$ falls below the vapour temperatures in the steam bleeding pipeline. This means that the non-condensation zone should appear in the case of overestimated surface area of the heat exchanger.

# 5 Concluding remarks

The following conclusions can be drawn from the above analysis:

1. Inclusion of the presence of inert gases in the condensing steam requires implementation of at least one-dimensional model of the heat exchanger.
2. The results of calculation show that there is a maximum of heat flux $q$, which location in the flow domain is dependent both on: the inert gases content as well as fouling of heat transfer surface.
3. The presence of inert gases is the reason for the significant increase of the terminal temperature difference $\delta T_o$. Fouling of the heat transfer surface is also responsible for the significant increase of the temperatures difference $\delta T_o$.

4. The strong vapour superheating may cause non-condensation zone in the heat exchanger. The presence of this zone depends on the criterion formulated in the paper. It was shown that the critical vapour temperature $T_{vg}$ depends on the surface area of the heat exchanger.

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