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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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GDAŃSK 2003

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Aims and Scope

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:

- Liquid flows in hydraulic machinery including exploitation problems,
- Gas and liquid flows with heat transport, particularly two-phase flows,
- Various aspects of development of plasma and laser engineering,
- Solid mechanics, machine mechanics including exploitation problems.

The periodical, where originally were published papers describing the research conducted at the Institute, has now appeared to be the place for publication of works by authors both from Poland and abroad. A traditional scope of topics has been preserved.

Only original and written in English works are published, which represent both theoretical and applied sciences. All papers are reviewed by two independent referees.

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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 100th 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
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TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

No. 113, 2003, 225-234

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Analysis of transient heat transfer operation in tube wall

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Abstract

Paper presents the problem connected with non-steady state operation of a regenerative heat exchanger. A simple mathematical model is proposed for estimation response time for non-steady temperature distribution in tube wall. The solution presented in paper was obtained from the analytical calculations.

Keywords: Heat transfer; Regenerative heat exchanger; Non-steady state operation

Nomenclature

A	–	area, m^2
a	–	thermal diffusivity, m^2/s
c_p	–	specific heat capacity at constant pressure, J/(kgK)
$\dot{E}$	–	energy flow rate, W
H	–	enthalpy, J
h	–	specific enthalpy, J/kg
m	–	mass, kg
$\dot{m}$	–	mass flow rate, kg/s
p	–	*pressure, Pa (bar)*
Q	–	*heat, J*
$\dot{Q}$	–	heat flow rate, W
$\dot{q}$	–	heat flux density, W/m^2
T	–	*temperature, K*
t	–	*time*, s
U	–	internal energy, J
α	–	convective heat transfer coefficient, $W/(m^2K)$
λ	–	conductive heat transfer coefficient, W/(mK)
ρ	–	density, kg/m^3

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1 Introduction

Heat exchangers are the important elements in the cycles with steam turbines. The role of these elements is to heat the feed water. Water on its way to the boiler is going through the heaters. In effect in the steam generator smaller heat flux is needed for the production of fresh steam with pressure p and temperature T compared to the cycle without regeneration. In effect the cycle efficiency is increasing.

The all elements in the cycle are designed for nominal load conditions. This calculation has been done making assumption of the steady state operation of the system. The dynamic aspects of elements characteristics are not taken into account. This type of approach concerns not only calculations in nominal load but also in part load.

The assumption of steady state condition allows to omit all the phenomena connected with accumulation of mass flow, heat and energy in the system. Heat exchangers found in steam cycles are in the tubular form, predominantly. The feed water goes inside the tube, while the steam passes outside the tube. In the case of largest plants, the dimensions of heaters are for example 2 m in diameter and 10 m length. According to E. Mitterecker, H. Kallenberg (1985) the total mass of such element is about 110 tons [1].

The heating steam is taken from the extraction points in turbine. Deviations of turbine load have an influence on the pressure at the extraction point. Consequently, the saturation pressure and temperature in the heater are not constant. If there is an increase of steam temperature, then the heater needs a time to adapt to a new stationary state. In this time, the mass flow rate of steam to heater is greater compared to stationary work. Part of the heat is accumulated in the material of the heat exchanger.

We have the inverse situation, when the steam pressure in extraction points is going down. In the time between the two stable work states, the consumption of steam is smaller compared to steady state of work. The temperature of heater material must go down.

The parameter important for this unsteady state performance of the heater is the response time. This parameter describes the time between two steady states of work after a jump of temperature.

For example, we have a heat exchanger working in the steam cycle. Under stable work the pressure of steam is 1.2 bar. The temperature of feed water is 75°C, mass flow rate of water is 100 kg/s. The response time for the heater was assumed to be about 20 s. Consider, we have a jump of steam pressure to 1.3 bar. The heat accumulated in the material of the heat exchanger as a consequence of pressure jump and is equal to 17.8 MJ. Fig. 1 presents the mass flow rate of steam as a function of pressure. Line (a) in this figure represents the mass flow rate for the increase of pressure from 1.2 bar to 1.3 bar. We have the opposite situation for the jump of pressure from 1.3 bar to 1.2 bar Fig. 1. – line (b). The calculation of mass flow rate for steady state work is presented as line (c).

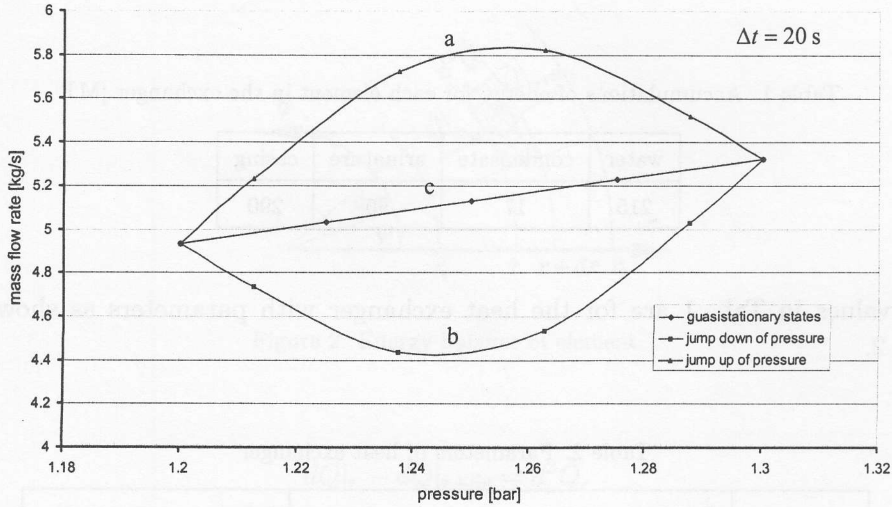

Figure 1. Hysteresis of change in mass flow rate for steam passing through the exchanger, brought about by sudden changes in pressure [3].

2 Assumptions

The assumptions presented below were formulated to obtain a simple mathematical model of heat transfer in non-stable state in tube wall, which would take into account transient effects in the process. The aim was to obtain a solution in analytic form.

2.1 Simplifying assumptions

In the thermodynamic system of a heat exchanger, a number of elements are responsible for heat energy accumulation. These include structural parts of the exchanger such as: the armature, casing, tube sheet; and the fluids contained within the body of the exchanger: water, steam, condensate. The total amount of accumulated energy is obviously related to the mass of the accumulating body; thus the largest amount of energy will be accumulated in the casing, followed by the feed water, armature, condensate, and steam in that order. The casing, however, has less importance as it does not directly take part in heat transfer from steam to water. The key accumulating elements influencing transient process behavior are the feed water and the armature. The description that follows takes into account heat accumulation in the pipes only.

Table 1 shows thermal energy accumulation in the respective exchanger elements at external temperature of 20 °C.

Table 1. Accumulations of energy for each element in the exchanger [MJ]

water	condensate	armature	casing
215	17	90	290

The values in Tab. 1 are for the heat exchanger with parameters as shown in Tab. 2.

Table 2. Parameters of heat exchanger

feed water	Pressure	6 bar
	inlet temperature	75°C
	mass flow rate	49 kg/s
steam	saturation pressure	1.33 bar
geometry and material data	casing internal diameter	2.6 m
	casing external diameter	2.66 m
	tube internal diameter	0.020 m
	tube external diameter	0.024 m
	mean U-tube length	15 m
	number of U-tubes	160
	thermal conductivity (copper)	397 W/m/K

2.2 Fundamental equation

The fundamental equation has been derived from the energy conservation principle on the assumption of constant physical properties of the armature. For solids, the change of internal energy related to volume and pressure changes is negligible. Therefore

$$\hat{d}Q \cong dH \cong dU, \tag{1}$$

so that the energy conservation equation reduces to the special case of balancing heat influx and outflow. A mass element within the tube wall is shown in polar coordinates in Fig. 1 below. For the element in Fig. 2 one can write

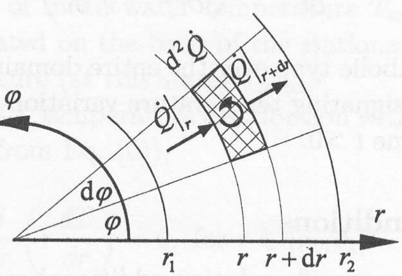

Figure 2. Energy balance of element.

$$d\dot{Q}|_r - d\dot{Q}|_{r+dr} = d^2\dot{Q}. \tag{2}$$

The next equation represents the influx and outflow of heat from the element according to Fourier law; with the assumption that heat is conducted in the radial direction only:

$$d\dot{E}_{in} = d\dot{Q}|_r = \dot{q}dA = -\lambda\frac{\partial T}{\partial r}zrd\varphi, \tag{3a}$$

$$d\dot{E}_{out} = d\dot{Q}|_{r+dr} = (\dot{q}+d\dot{q})d(A+dA) = -\lambda\frac{\partial}{\partial r}\left(T+\frac{\partial T}{\partial r}\right)z(r+dr)d\varphi \tag{3b}$$

The part of heat flux being accumulated in the element is

$$d(d\dot{E}_{acc}) = d^2\dot{Q} = d^2m\frac{dh}{dt} = d^2m\frac{\partial h}{\partial t} = d^2m\left(\frac{\partial h}{\partial T}\right)_p\frac{\partial T}{\partial t}. \tag{4}$$

Introducing the definition for specific heat at constant pressure

$$c_p = \left(\frac{\partial h}{\partial T}\right)_p,$$

and including density ρ, one transforms formula (4) into the form

$$d^2\dot{Q} = \rho zr\;dr\;d\varphi\;c_p\;\frac{\partial T}{\partial t}. \tag{5}$$

After including relations (3) and (5) in Eq. (2), simplifying and discarding small second-order terms, one obtains the following partial differential equation

$$\frac{\partial T}{\partial t} = a\left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial t}\right). \tag{6}$$

Equation (6) is of parabolic type over the entire domain of r and t. Its solution is the function $T(r,t)$ designating temperature variation within the region between radii r_1 i r_2, at any time $t>0$.

2.3 Boundary conditions

In order to obtain a specific solution additional constraints must be made in the form of boundary and initial conditions. Heat exchange at the edges of the region occurs in accordance with Newton's law. Boundary conditions, therefore, are of the third kind (Fourier's conditions):

$$\left(l_w\frac{\partial T}{\partial r} - T\right)_{r=r_1} = T_{w_o\infty}, \quad \left(l_p\frac{\partial T}{\partial r} - T\right)_{r=r_2} = T_{p_n\infty}, \tag{7}$$

where:

$$l_w = \frac{\lambda}{\alpha_w}, \; l_p = \frac{\lambda}{\alpha_p}.$$

The meaning of individual symbols in this formulation is shown in Fig. 3. The

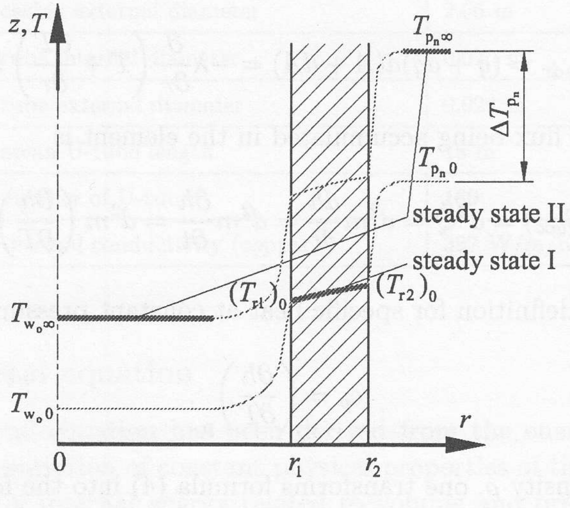

Figure 3. Illustration of boundary conditions.

conditions (7) describe a jump in steam and water temperature between two steady-state conditions. Taking into account low thermal inertia of steam, the assumption of a jump transition is justifiable for steam. The corresponding assumption for feed water (no thermal inertia) on the other hand, is a major simplification. The value of mean water temperature $T_{w_o\infty}$ at the new steady (bounding) state is calculated on the basis of the stationary model for the given steam saturation temperature (at this state) $T_{p_n\infty}$.

At the beginning ($t = 0$), temperature distribution within the cylindrical wall is given by the equation (from Eq. (6)),

$$\frac{d}{dr}\left(r\frac{dT}{dr}\right) = 0, \text{ for } r \in \langle r_1, r_2\rangle \tag{8}$$

for steady state conditions, and fulfilling the conditions:

$$\left(l_w\frac{\partial T}{\partial r} - T\right)_{r=r_1} = T_{w_o0}, \quad \left(l_p\frac{\partial T}{\partial r} - T\right)_{r=r_2} = T_{p_n0}. \tag{9}$$

The values of mean temperatures T_{w_o0} i T_{p_n0} for the initial steady–state are determined using the same method as in the boundary states. A solution of (8) which fulfils the conditions (9) is obtained as the function

$$T(r,0) = T_0(r) = \left(T_{r1} - \frac{T_{r1} - T_{r2}}{\ln\frac{r_2}{r_1}}\ln\frac{r}{r_1}\right)_0, \tag{10}$$

where wall temperatures are given by the formulas:

$$T_{r1} = \frac{\frac{l_w}{r_1}(T_{p_n} - T_{w_o})}{\frac{l_w}{r_1} + \frac{l_p}{r_2} + \ln\frac{r_2}{r_1}}, \quad T_{r2} = \frac{\frac{l_p}{r_2}(T_{p_n} - T_{w_o})}{\frac{l_w}{r_1} + \frac{l_p}{r_2} + \ln\frac{r_2}{r_1}}. \tag{11}$$

3 Calculation example

The solution presented below was obtained from the analytic calculation. The parameters for numerical calculations are presented in Tab. 3.

The values in Tab. 3 are the results of prior calculations utilizing the steady–state model of the heat exchanger, based on data from Tab. 1. For details of the steady–state model, see [2]. The illustration below (Fig. 4) shows the temperature distribution across the tube wall in the two steady states I and II corresponding to values in Tab. 3.

Table 3. Input parameters

		state I	state II			
α_w	W/m^2/K	6568	6604	ρ	kg/m^3	8960
α_p		5056	4967	r_1	m	0.01
T_{w_o}	K	362.8	363.8	r_2		0.012
T_{p_n}		380.8	383.1	λ	W/m/K	397
T_{r_1}		368.9	370.3	c_p	J/kg/K	386
T_{r_1}		374.2	375.9			

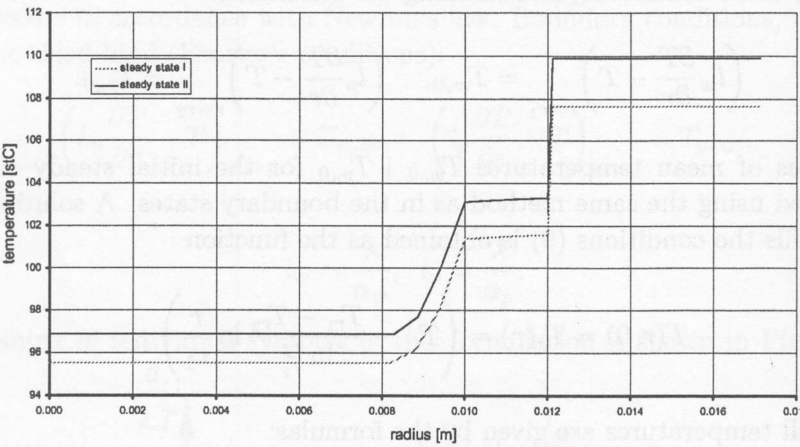

Figure 4. Temperature distributions for two selected steady state conditions.

The above distributions were obtained with the following assumptions: in the case of state I, saturated steam pressure 1.33 bar; for state II, 1.43 bar. This variation in pressure corresponds to a change in saturated steam temperature of about 2.25 K.

For a jump up of steam pressure, temperature distributions in pipe walls have been calculated; they are shown in Fig. 5 as function of radial distance and time. Note, in Fig. 5b, that the response time of the system in question is about 4 s. This information is essential for determining the path of hysteresis, and therefore for quantifying the actual changes in operating parameters of the exchanger during transient-state operation.

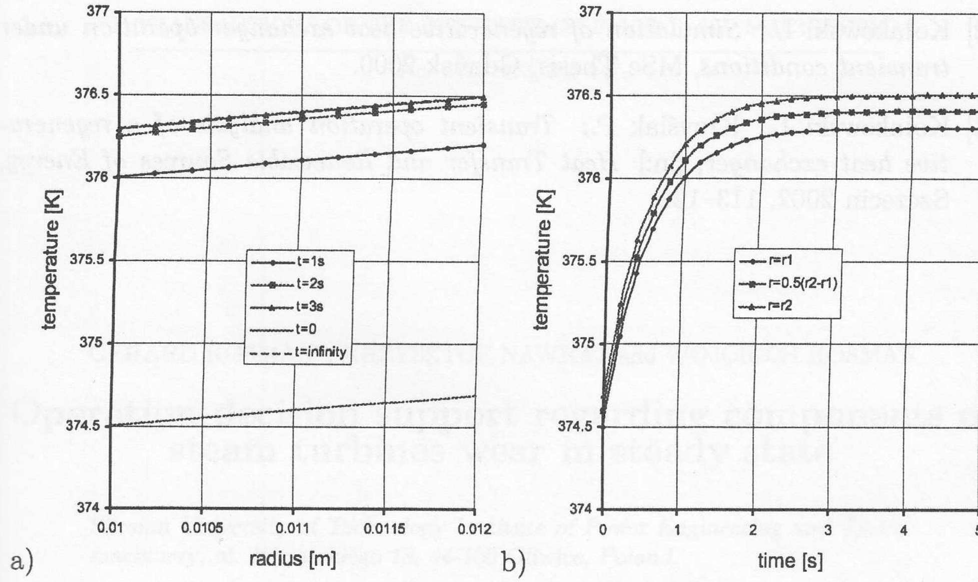

Figure 5. Dependence of temperature distribution in tube wall on: a) radius, b) time.

4 Conclusions

The paper presents a simplified model of a heat transfer in transient operating conditions in tube wall, based on Fourier's conduction equation. It takes into account only one of the constructional elements, one tube in the heat exchanger.

The computational scheme proposed in this paper makes it possible, for changes in the medium temperatures expressed as sudden jumps, to determine the response time of the tube wall. This information cannot be obtained within the framework of the steady-state heat exchanger model. Obtained results estimate the response time to be about 4s.

Advantage of the methodology outlined in this paper is the feasibility of obtaining an actual analytic solution. Further work in this area should lead to an improved model, in which the assumption of a jump change refers only to steam–side parameters, while changes in water temperature with time will be the resulting function.

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