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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:

- 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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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

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MARIAN TRELA[a*], EUGENIUSZ IHNATOWICZ[a], ANDRZEJ KRUPA[a], MAŁGORZATA NAJWER[b] and JÓZEF BANASIEWICZ[b]

Monitoring of air content in air-vapour mixture removed from power plant condensers

[a] *Institute of Fluid-Flow Machinery, Polish Academy of Sciences, Fiszera 14, 80-231 Gdańsk, Poland,*

[b] *Power Station Turów, Młodych Energetyków 12, 59-916 Bogatynia 3, Poland*

Abstract

The paper considers the problem of measurement of air amount in air–vapour mixture removed from condensers by steam ejectors. An efficient method for the air inleakage rate measurement is proposed. It allows for the accurate measurement and monitoring of air inleakage rate in the wide range from 0.5 to 150 kg/h of dry air. The results of such measurements may be easily implemented in diagnostic codes to evaluate the power losses resulting from the excess air influx into condensers. The monitoring systems based on the above method have been constructed and implemented at the Turów power station for turbine units of 200 MW.

Keywords: Air-vapour mixture; Condenser; Monitoring of air content in vapour

Nomenclature

A	–	area
B	–	coefficient
c_D	–	orifice discharge coefficient
c_p	–	specific heat, J/(kg K)
d	–	throat diameter
D	–	pipe diameter
i	–	specific enthalpy, J/kg
m	–	dimensionless air mass flow rate, $\dot{m}_a/\dot{m}_{ao}$
$\dot{m}$	–	mass flow rate, kg/s or kg/h
N	–	identification number of the orifice
p	–	pressure, Pa

*Corresponding author. E-mail address: mtr@imp.gda.pl

Q – heat transfer rate, J/s
R – thermal resistance, $(\mathrm{m}^2\mathrm{K})/\mathrm{W}$
T – temperature
T_o – reference temperature
t – time
V – velocity
U – overall heat transfer coefficient, $\mathrm{W}/(\mathrm{m}^2\mathrm{K})$
δT_o – terminal temperature difference

Subscripts

1 – initial
2 – outlet
a – air
b – barometric
c – condensate
f – fouling
m – mixture
o – reference value
t – tube wall
v – vapour
vac – vacuum
w – water
log – log mean difference

1 Introduction

The paper considers the measurement of mass flow rate of air removed from condensers in steam power plants. The excessive amount of air in the condenser diminishes the heat transfer and leads to the increase of condensation pressure, thus reducing the efficiency of the cycle. In the paper, two modern systems of continuous measurement and monitoring of the mass flow rate of air removed from condensers are presented. They enable to measure, precisely and in a wide range, the mass flow rate of discharged air, with simultaneous data acquisition. Due to the relatively wide measurement range, the systems can be used in the power plants including turbine units of very low up to very high outputs. The paper also presents the calculations of the losses resulting from the excessive air-inleakage into the condenser.

2 Basic formulas for evaluation of condenser performance

Air enters the condenser along with the steam and subsequently leaks through the turbine and other gaps in the vacuum part of the turbine unit. The excessive amount of air affects unfavourably the heat transfer during vapour condensation process, thus decreasing the overall heat transfer coefficient U. This leads to the increase of condensation temperature T_c, i.e. to the increase of condensation pressure p_c and the decrease of the turbine performance. The excessive amount of gases in the condenser promotes also the corrosion, resulting from the dissolution of O_2 and CO_2 in the condensate. Therefore, the concentration of air in the condenser has to be controlled and kept below a certain limiting values, which depends on the mass flow rate of vapour entering the condenser and on the condenser geometry.

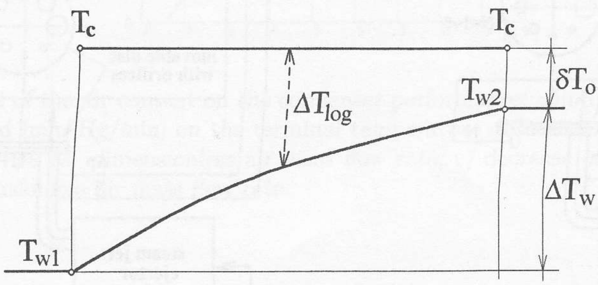

Figure 1. Relationship between the heat rejection parameters.

The temperature distribution in the condenser is shown schematically in Fig. 1. The condensation temperature T_c depends on the inlet cooling water temperature T_{w1}, the temperature rise of circulating water ΔT_w, and the terminal temperature difference, δT_o,

$$T_c = T_{w1} + \Delta T_w + \delta T_o. \tag{1}$$

Taking into consideration the following relations:

- logarithmic mean temperature difference

$$\Delta T_{\log} = \frac{\Delta T_w}{\ln\left(1 + \frac{\Delta T_w}{\delta T_o}\right)}, \tag{2}$$

- energy balance formula

$$Q = \dot{m}_v (i_v - i_c), \quad Q = \dot{m}_w c_{pw} \Delta T_w, \tag{3}$$

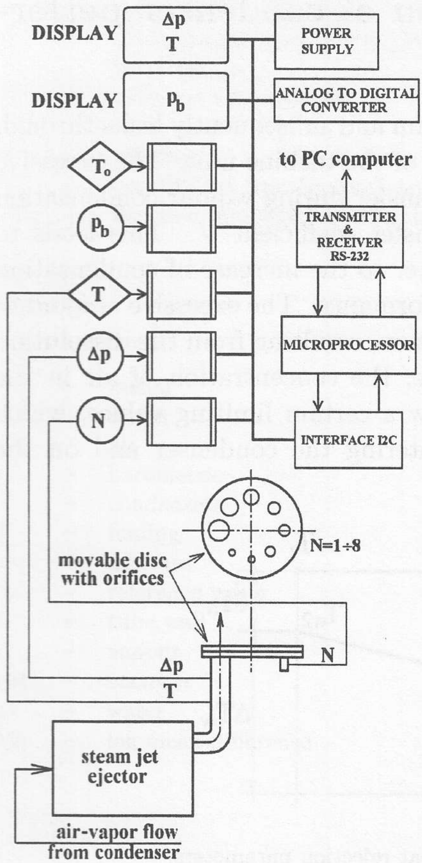

Figure 2. Schematic diagram of the apparatus in the first type of monitoring systems.

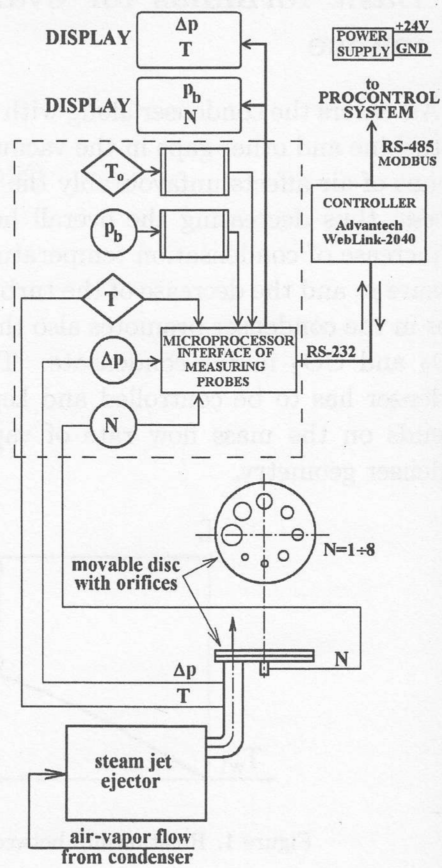

Figure 3. Schematic diagram of the second monitoring systems.

- Peclet equation

$$Q = UA\Delta T_{log} \tag{4}$$

for $\dot{m}_v = \text{const}$ one obtains

$$\Delta T_w = \frac{\dot{m}_v}{\dot{m}_w}\left(\frac{i_v - i_c}{c_{pw}}\right) \sim \frac{\dot{m}_v}{\dot{m}_w}. \tag{5}$$

Taking into account Eqs. (3) and (4) one arrives at the terminal temperature difference

$$\delta T_o = \frac{\Delta T_w}{\exp(UB) - 1} \tag{6}$$

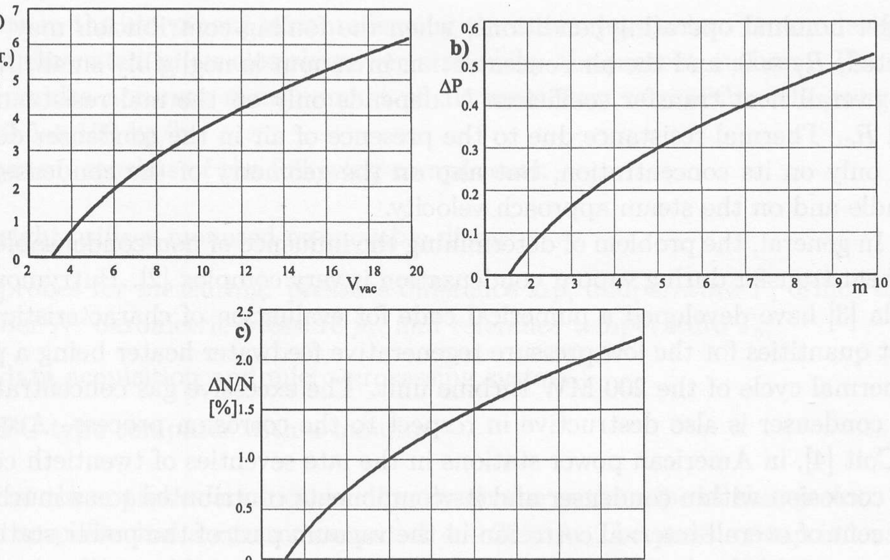

Figure 4. Influence of the air content on the condenser performance: a)influence of the vacuum loss speed [mm Hg/min] on the terminal temperature difference [7], b) condensation pressure rise vs. dimensionless air mass flow rate, c) decrease of turbine set output vs. dimensionless air mass flow rate.

where $B = A/(\dot{m}_w c_{pw})$. It is seen that for $A, \dot{m}_w, c_{pw} =$ const, the temperature difference δT_o is only a function of the overall heat transfer coefficient U:

$$\delta T_o = f(U). \tag{7}$$

3 The effect air concentration on overall heat transfer coefficient

The above analysis and the resulting relations (4) and (6) reveal that the overall heat transfer coefficient is a dominant factor as far as the heat ejection parameters are considered. Decreasing the heat transfer coefficient U results in the increase of values of $\Delta T_{log}, \delta T_o$ and p_c.

The total heat transfer resistance R is the sum of all individual resistances and may be expressed as:

$$R = \frac{1}{U} = R_w + R_t + R_c + R_f + R_a \tag{8}$$

Thermal resistances involved in Eq. (8) can be calculated from various semi-empirical heat transfer correlations, which may be found elsewhere, e.g. in [1].

Under nominal operating conditions, when the fouling contribution may be neglected, $R_f \approx 0$, and the air concentration in vapour is negligibly small, $R_a \approx 0$, the overall heat transfer coefficient U depends only on thermal resistances R_w and R_c. Thermal resistance due to the presence of air in the condenser depends not only on its concentration, but also on the geometry of the condenser tube bundle and on the steam approach velocity.

In general, the problem of determining the influence of non-condensable gases on heat transfer during vapour condensation is very complex [2]. Butrymowicz & Trela [3] have developed a numerical code for evaluation of characteristic pertinent quantities for the low-pressure regenerative feedwater heater being a part of a thermal cycle of the 200 MW turbine unit. The excessive gas concentration in the condenser is also destructive in respect to the corrosion process. According to Coit [4], in American power stations in the late seventies of twentieth century the corrosion within condenser and its components contributed to as much as 70 per cent of overall internal corrosion in the vacuum part of the power station. In order to avoid these two unfavourable effects i.e. on the turbine performance and corrosion, the concentration of gases in the condenser should be controlled and kept below permissible limiting values. These limiting values depend mainly on the mass flow rate of vapour entering the condenser and on the condenser design. The other important factor is the venting system which has to operate properly.

4 Air-inleakage measurement system

The measurement system of gases removed from the condenser by using a steam jet ejector consists generally of two parts: one device to measure the mass flow rate of vapour-air mixture leaving the ejector and the other one to determine its physical parameters [5]. The condition of air-vapour mixture exiting the condenser can be determined by accurately measured pressure and temperature at the ejector vent connection. The amount of vapour to saturate non-condensable gases can be calculated then from the pertinent formula in thermodynamics of air-vapour mixture [6]. The mass flow rate of the mixture can be measured with the aid of one of standard method, for instance by using the orifice. Two different kinds of monitoring systems have been developed at the Institute of Fluid-Flow Machinery of PAS in Gdańsk:

1. the autonomic system,
2. the system connected to the main power station recording system e.g. PROCONTROL.

Figure 2 presents a schematic diagram of the novel autonomic system for continuous measurement, monitoring and recording of the mass flow rate and air

contents in the air-vapour mixture leaving the steam jet ejector. It was designed and implemented in an electric power station by the present authors. The system provides accurate measurement of air mass flow rate within the range of $\dot{m}_a = 0.5 \div 150$ kg/h.
The system consists of the following components:

- eight orifices mounted on movable discs,
- probes for measuring: pressure difference Δp, temperature T, orifice number N, barometric pressure p_b, and reference temperature T_o,
- data acquisition and micro-processing system,
- PC-type computer with a monitor.

The first three probes are located directly on the outlet pipe of the ejector. The sensor amplifier has a current output of $4 \div 20$ mA. The pressure tap is located at one pipe diameter upstream from the orifice plate. The barometric pressure gauge is installed in a box containing the microprocessor-based measuring system. The probe embraces a self-contained thermometer. Its role is to compensate the temperature change of the probe. Thermal compensation is done by the software. The temperature coefficients of a sensor-amplifier system were evaluated experimentally for each barometric pressure probe and were stored in EPROM memory. In the course of measurement the relevant correction coefficients obtained during calibration were used. Temperature was measured with the aid of thermocouples. Reference temperatures were measured using resistor thermometers.

The device identifying the orifice number N was mounted beneath the movable disc. It was mechanically coupled with the disc. The device plays a role of a potentiometer, which is a part of the bridge connected to the amplifier system of $4 \div 20$ mA current signal. Therefore the disc position gave an input signal for the device identifying the number N. All system components are integrated by the I2C bus. The only exception is the RS-232 interface with optical insulation, which is controlled directly by the microprocessor. Via this interface the data are transmitted to the PC-type master computer located in the turbine set control room. In the box containing the microprocessor system, the temperature and pressure displays are located as well.

In order to determine characteristic curves for eight measuring orifices a series of laboratory tests were carried out in aerodynamic tunnel at the Institute of Fluid-Flow Machinery, PAS. These tests led to evaluation of the discharge coefficient C_D for those orifices. During the tests the similarity rules were preserved. The test results were presented in the form of the following formula:

$$C_D = k\mathrm{Re}^n, \tag{9}$$

where the coefficients k and n are the functions of the Reynolds number and the ratio d/D. The Reynolds number was derived basing on the internal exit pipe diameter D. The data was obtained for the Reynolds number range: $\mathrm{Re} = (0.2 \div 17) \cdot 10^3$.

Figure 3 shows a schematic diagram of the second type of monitoring system. Here, the PC computer is replaced by the controller connected to the power station recording system in MODBUS standard. This version of the monitoring system was also implemented at the Turów power station. With the help of the above presented formulas a numerical algorithm was developed for determination of the mass flow rate of air-vapour mixture leaving the steam jet ejector and its fraction i.e. the amount of dry air in the mixture. The current values of the mass flow rates of dry air, $\dot{m}_a$, and mixture, $\dot{m}_m$, averaged over the period of 1 minute, are displayed on the computer monitor and then recorded and stored on the computer hard disc for the autonomic system. In a case of the second system the data are transmitted to the station controlling and recording system e.g. PROCONTROL. The procedure mentioned above is used to calculate the averaged values of $\dot{m}_a$ for one hour, day, etc. Due to very high accuracy of the probes used, the measurement ranges were $0.03 < \Delta p < 0.3$ kPa, for the orifices marked by numbers $N = 1 \div 8$. When these limits are exceeded, a message appears on the monitor screen which urges the operator to make the necessary change of the orifice into a new one marked by a given number N. There is also a possibility for introducing a limiting value of the air mass flow rate $\dot{m}_a$, beyond which the system alarms the operator.

In the version described above, the monitoring system can be used for measurements and monitoring of the mass flow rate of air within the range of $\dot{m}_a = 0.5 \div 150$ kg/h for one jet ejector.

5 Condenser performance analysis with respect to air content

The losses of the condenser performance may be caused by several reasons. For the loss resulting from the excessive amount of air in the condenser it can be evaluated very easily if one has an empirical relation between the terminal temperature difference and the air-inlekage. Folwarczny & Magiera [7] obtained an empirical relation combining the change of terminal temperature difference $\Delta(\delta T_o)$ and the vacuum loss rate (Fig. 4a) for the TK 200 turbine unit. This kind of examination of the condenser tightness is used in some countries together with the measurement of the mass flow rate of removed mixture. During this examination the air removal system has to be shut down then the air inflow affects the increase of condensation pressure. On the other hand the results of experimental investigations performed by the authors of this paper show that in the case of the TK 200 turbine units the vacuum loss rate of 1 mm Hg/min corresponds to the mass flow rate of ejected air by approximately 6 kg/h. It is obvious that the increase of terminal temperature difference by $\Delta(\delta T_o)$ corresponds to the condensation pressure increase by Δp_c. It may be shown, that within the pressure range $p_c = 5 \div 12$ kPa, an average pressure change p_c, due to the saturation temperature change, equals 0.423 kPa/K. This allows to derive the relation $\Delta P = f(m)$, basing on the graph shown in Fig. 4a, which is shown in Fig. 4b (where $\Delta P = \delta p_c/p_{co}$, $m = \dot{m}_a/\dot{m}_{ao}$, $\dot{m}_{ao} = 12$ kg/h, $p_{co} = 5$ kPa). The increase of the condensation pressure by Δp_c corresponds to the turbine power output loss by ΔN. This loss can be evaluated from the following formula [6]:

$$\Delta N = k_c \Delta p_c \tag{10}$$

where the coefficient k_c depends on the condenser geometry. For the turbogenerator unit TK 200 it equals $k_c = 1.7$ MW/kPa [7]. Fig. 4c shows the final estimation of the power output losses resulting from the excessive air-inleakage i.e. the function of power loss versus dimensionless air mass flow rate removed from the condenser [8].

The numerical results presented in Fig. 4c reveal a significant influence of the excessive air amount in the condenser on the power losses. Decreasing the air inleakage (by eliminating some leaks), for instance from the value of $m = 6$ (often recorded in some power stations) to $m = 2.5$ improves the output power by, *approximately, 1%*.

Exact power losses ΔN of the turbine unit due to the excessive air inflow to the condenser can be evaluated based on the method of diagnosing thermal cycles and their components, used for instance in the code DIAGAR [9].

6 Conclusions

The presented monitoring systems of the air removed from the condenser have been successfully implemented in the six power units of 200 MW output at Turów power station. For the three turbine units the autonomic monitoring systems have been used. They have been working successfully for 5 years. The next three monitoring systems of the second kind have been applied at the Turów power station this year.

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