ESTIMATION OF CONDENSATE MASS FLOW RATE DURING PURGING TIME IN HEAT RECOVERY STEAM GENERATOR OF COMBINED CYCLE POWER PLANT

by

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In this paper the transient modeling of heat recovery steam generator in purging time was considered. In purging time, compressed air from the gas turbine was used to purge a combustible gas from heat recovery steam generator. During this time; steam condensate was formed in the superheater stage which should be drained completely to avoid some problems such as deformation of superheaters. Because of this reason, estimation of drain formation is essential to avoid this problem. In this paper an energy model was provided and this model was solved by MATLAB software. Average model error is about 5%. Results show that, during purge time, steam temperature was decreased from 502 °C (superheater 2), 392 °C (superheater1), and 266 °C (Evaporators 1 and 2) to 130 °C, 130 °C and 220 °C, respectively, and also steam pressure was decreased from 52 bar to 23 bar during purge time. At end of purge time, condensate formation was about 220 (l) when inlet gas temperature was equal to 100 °C and purge gas mass flow rate was equal to 386.86 kg/s.

Key words: *purging, condensate, heat, recovery*

Introduction

Heat recovery steam generator (HRSG) is an energy recovery heat exchanger that recovers heat from a hot gas stream. It produces steam that can be used in a process or used to drive a steam turbine. A common application for an HRSG is in a combined-cycle power station, where hot exhaust from a gas turbine is fed to a HRSG to generate steam which in turn drives a steam turbine. This combination produces electricity more efficiently than either the gas turbine or steam turbine alone [1]. Heat recovery steam generation has three different operation modes, normal, startup and shutdown that in a combined-cycle, startup procedure is separated into three primary phases, purging of HRSG; gas turbine speed-up, synchronization, and loading; and steam turbine speed-up, synchronization, and loading [2].

In purging time, compressed air from the gas turbine is used to purge combustible gases from HRSG, during this time; condensate is formed in the superheater. Purging required as a precondition to start the gas turbine through the boiler, is a common requirement of all boiler codes to ensure safety operation of the plant. This rule is historically evolved, since in the

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beginning of boiler operation severe accidents occurred [3, 4]. During purging time, condensate is formed in the superheater . If the condensate is not completely drained from all superheater tubes before steam flow is established during the start up, two adverse consequences are result [5].

- First, condensate that often is still sub-cooled is ejected in large quantity into the outlet header and pipe manifold where it quench-cools hotter material. On hot starts after trips, the outlet header and manifold can be more than 200 °C above saturation temperature. During warm starts, the outlet header will be close to saturation temperature but the outlet manifold temperature will still be substantially higher.
- Second, tubes have different average temperatures at different times because they do not clear of condensate simultaneously. Some tubes will be clear of condensate and therefore will be cooled by steam flow, while others will have part of the tube filled with condensate at saturation or lower temperature. Still other tubes will be stagnant and therefore at the purge air or exhaust gas temperature. Designs that have stiff tube arrangements connected to headers at each end develop. Thus, the unit will develop large stresses at tube-attachment welds when other tubes in the same row are at different temperature.

To eliminate both of these effects, the condensate needs to be removed from the lower headers at the peak rate at which it forms. But that is not easily accomplished. Many researchers have been conducted about modeling of HRSG [6-11]. Dechamps [6] described a method used to compute the transient performances of assisted circulation heat recovery steam generators. In his mathematical modeling bundle of tubes in each part of HRSG replaced with *equivalent* linear heat exchanger. Valdes *et al*. [7] proposed a methodology to identify the most relevant design parameters that impact on the thermal efficiency and the economic results of combined cycle gas turbine power plants in full and part loads. Mohagheghi and Shayegan [8] developed a new method for modeling a steam cycle in advanced combined cycles by organizing non-linear equations and their simultaneous solutions by use of the hybrid Newton methods. Godoy *et al*. [9] proposed optimal designs of a CCGT power plant characterized by maximum second law efficiency values. These thermodynamic optimal solutions were found within a feasible operation region by means of a non-linear mathematical programming (NLP) model, where decision variables (*i. e*. transfer areas, power production, mass flow rates, temperatures and pressures) could vary freely. Woudstra *et al.* [10] investigated internal exergy efficiency for combined cycles used the same gas turbine but have different steam bottoming cycles. Differences did originate from the number of pressure levels at which steam is generated in the HRSG . The evaluation includes respectively a single pressure, double pressures and triple pressures HRSG. The steam pressures were optimized with regard to overall plant efficiency using a multi-parameter optimization procedure. They showed that internal exergy evaluation was useful method to cycle performance promotion. Bahadori and Vuthaluru [11] provided a simple model to estimation of the percent of blowdown that is flashed to steam as a function of flash drum pressure and operating boiler drum pressure followed by the calculation of the amount of heat recoverable from the condensate.

Due to literature survey, no research has been conducted the HRSG during purging time. In this paper, transient modeling of HRSG to estimate of condensate mass flow rate during purging time was conducted. For this purpose, a transient analytical modeling of HRSG was prepared and the effect of cooling gas temperature and mass flow rate on condensate was investigated. In addition effect of purge time on condensate was also considered. In summary, the followings are the specific contribution of this study in the subject matter area:

– transient modeling of HRSG, especially during purge time,
– comparing the numerical output of the model with the actual measurements at a real power plant, and
– a computer program code was developed in the MATLAB which could be applied for simulation of any types of HRSG.

System description

The typical power and steam generation plant, which we considered in this research consists of the followings items:
– gas turbine and generator: Four 4 V94.2 gas turbine generator and auxiliaries,
– HRSG : two 2 single pressure level waste heat recovery steam generators, and
– common system: Two 2 spray tray type deaerators, seven 7 boiler feed water pumps (six 6 electric motor driven and one 1 steam turbine driven), two 2 re-circulation pumps, four 4 booster pumps, two 2 fuel oil pump, blow down system, sampling system and dosing system.

Waste heat from gas turbine exhaust heat passes through the individual HRSG will generate steam. Each HRSG generates HP steam to HP steam common header. Also LP steam is produced through let down station which its source is HP steam extracted from HP common header. Demine water is supplied to external deaerators through preheater which its temperature shall be increased before enters to preheater by re-circulation pumps. Feed water to the HRSG and auxiliary boiler economizer shall be fed by a common feed pumps, which shall take their suction from the feed water storage tank.

This HRSG has the following condition:
– each HRSG maximum capacity: 340000 kg/h (gross),
– installation: outdoor,
– type: fired, natural circulation,
– operating mode: constant pressure,
– duct and casing will be insulated for surface temperature below 60 °C at ambient temperature 37 °C,
– gas pressure drop: 300.00 mmH_2O based on static head, and
– HRSG configuration: length (from diverter center to outlet stack center): 34.32 m and inlet duct angle: 51 deg.

Figure 1 shows the schematic diagram of system. For mathematical modeling, three control volumes are considered. These control volumes are two superheaters 1, 2 and evaporator, which are shown in fig. 1. During purging time, inlet HP steam drum and at temperator are closed. Purging time is about 300 s, which condensate is formed in superheaters 1 and 2, and evaporator are drained through a drain which is located below superheaters.

Specification of HRSG heating surface is shown in tab. 1. Steam conditions in evaporator and superheaters 1 and 2 are shown in tab. 2. The purge gas specification is shown in tab. 3.

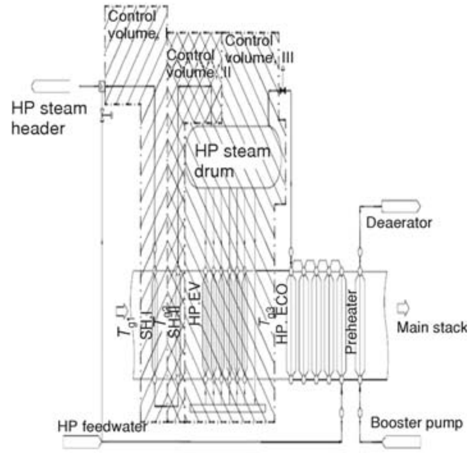

Figure 1. Schematic diagram of the system

Table 1. Specification of HRSG heating surface

Component	Geometry data						Fin data							Weight		
	S_L [mm]	S_T [mm]	S_{TN} [–]	S_{LN} [–]	HS [m^{-2}]	No tubes [–]	OD [mm]	Thickness [mm]	Finned length [m]	No fins [/m]	Thickness [mm]	Height [mm]	Type [–]	Tube [ton]	Fin [ton]	Total [ton]
SH-HP2	101.6	114.3	64	3	1647	192	50.8	2.7	17.0	100	1.2	9.0	Solid	10.5	5.2	15.7
SH-HP1	101.6	114.3	64	2	2206	128	50.8	2.7	17.0	142	1.0	11.0	Solid	7.0	5.2	12.1
EV-HP2	90.0	96.0	78	6	13797	468	38.1	2.4	17.0	260	1.0	19.0	Ser-rated	16.8	43.2	60.0
EV-HP1	90.0	96.0	78	12	27594	936	38.1	2.4	17.0	260	1.0	19.0	Ser-rated	33.6	86.3	119.9

Table 2. Steam conditions in evaporator and superheaters 1 and 2

Component	Pressure [bar]	Temperature [°C]	Steam condition
SH-HP2	52.00	502.00	Superheat
SH-HP1	52.00	392.00	Superheat
EV-HP2	52.00	266.00	Saturate
EV-HP1	52.00	266.00	Saturate

Table 3. Purge gas specification

Gas component	Temperature [°C]	Pressure [bar]	m_P [kgs^{-1}]	Time [s]
N_2 (76%)	100	3	386.86	300
O_2 (21%)				
CO_2 (1%)				
Others (1%)				

Mathematical modeling

The heat balance for each control volume can be written as:

$$m_p C_P \frac{dT_P}{dt} + m_f C_f \frac{dT_f}{dt} + m_s \frac{dU_s}{dt} + \dot{m}_{wp} h_{fgp} = \dot{m}_{gp} C_g (T_{gin} - T_{fout}) \tag{1}$$

Here m_p is the total pipe mass in each control volume, C_p – the specific heat of pipe, T_p – the pipe temperature, t – the time, m_f – the total fin mass in each control volume, C_f – the specific heat of fin, T_f – the fin temperature, m_s – the total steam mass inside pipes in each control volume, U_s – the internal energy of steam, $\dot{m}_{gp}$ – the purge gas mass flow rate, C_g – the constant pressure specific heat of purge gas, T_{gin} – the inlet purge gas temperature, T_{gout} – the outlet purge gas temperature, $\dot{m}_{wp}$ – the condensate mass flow rate, C_w – the specific heat of condensate, and h_{fgp} – the condensate enthalpy. Equation (1) can be written for three control volumes which are shown in fig. 1. In general, this equation shows energy conservation in control volumes. First to fourth terms in left hand of the equation, show pipes, fins, steam, condensation energy variation with cool gas flowing. Right side of the equation shows energy variation of purge air which it flows control volumes showed in fig. 1.

Also, energy gas reduction rate is equal to heat transfer between gas and HRSG control volume:

$$U_t A_t \Delta T_{ln} = \dot{m}_g C_g (T_{gin} - T_{gout}) \tag{2}$$

Here U_t is the overall heat transfer conductivity, A_t – the overall heat transfer surface, and ΔT_{ln} – the log-mean temperature.

Fin temperature can be calculated by the following equation [12]:

$$T_{\mathrm{f}} = \frac{U_{\mathrm{i}} A_{\mathrm{i}} T_{\mathrm{s}} + U_{\mathrm{t}} A_{\mathrm{t}} T_{\mathrm{gout}}}{U_{\mathrm{i}} A_{\mathrm{i}} + U_{\mathrm{t}} A_{\mathrm{t}}} \tag{3}$$

Here, U_{i} is the inside tubes overall heat transfer coefficient and A_{i} – the internal surface.

Overall heat transfer coefficient can be calculated by the following equation [12]:

$$\frac{1}{U_{\mathrm{t}}} = ff_{\mathrm{o}} + \frac{1}{\eta h_{\mathrm{o}}} + ff_{\mathrm{i}} \frac{A_{\mathrm{t}}}{A_{\mathrm{i}}} + \frac{A_{\mathrm{t}}}{2\pi k_{\mathrm{m}}} \ln\left(\frac{d_{\mathrm{o}}}{d_{\mathrm{i}}}\right) + \frac{A_{\mathrm{t}}}{A_{\mathrm{i}} h_{\mathrm{i}}} \tag{4}$$

Here, ff_{i}, ff_{o} are fouling factors inside and outside of tubes, k_{m} is the thermal conductivity of tube wall, d_{o}, d_{i} are outlet and inlet pipe diameters, h_{o}, h_{i} are outlet and inlet heat convection coefficient, and η is the fin effectiveness. Inlet and outlet overall heat transfer coefficients can be calculated by the following equations [12]:

$$\frac{1}{U_{\mathrm{i}}} = ff_{\mathrm{i}} + \frac{A_{\mathrm{i}}}{2\pi k_{\mathrm{m}}} \ln\left(\frac{d_{\mathrm{o}}}{d_{\mathrm{i}}}\right) \tag{5}$$

$$\frac{1}{U_{\mathrm{o}}} = ff_{\mathrm{o}} + \frac{1}{\eta h_{\mathrm{o}}} \tag{6}$$

Outlet overall heat transfer coefficient can be calculated by the following equation [12]:

$$h_{\mathrm{o}} = C_1 C_3 C_5 \left(\frac{d_{\mathrm{o}} + 2h}{d_{\mathrm{o}}}\right) \left(\frac{T_{\mathrm{gm}} + 273.15}{T_{\mathrm{f}} + 273.15}\right) G Cp_{\mathrm{m}} \,\mathrm{Pr}^{-0.67} \tag{7}$$

Here, C_1, C_3, C_5 are constants, d_{o} is the outlet pipe diameter, h – the fin height, T_{gm} – the average purge gas temperature, G – the gas mass velocity, Cp_{m} – the mixture gas heat capacity, and Pr – the Prandtl number. G can be calculated by the equation:

$$G = \frac{\dot{m}_{\mathrm{g}}}{N_{\mathrm{w}} L (S_{\mathrm{T}} - d_{\mathrm{o}} - 2nbh)} \tag{8}$$

Here, N_{w} is the number of pipes wide, L – the effectiveness length of pipe, S_{T} – the transverse pitch, n – the number of fins per meter, b – the fin thickness, and h – the fin height.

C_1, C_2, and C_3 are obtained from tab. 4. In this table S_{T} and S_{L} are transverse and longitudinal pitches and s is the fin clearance, which can be calculated by the equation:

$$s = \frac{1}{n} - b \tag{9}$$

and the Reynolds number:

$$\mathrm{Re} = \frac{Gd}{\mu} \tag{10}$$

For average inside pipe heat transfer coefficient, we use the equation [13]:

$$\bar{h}_{\mathrm{i}} = 0.065 \frac{\sqrt{k_{\mathrm{f}} \rho_{\mathrm{f}}}}{\mu_{\mathrm{f}}} \sqrt{\mathrm{Pr}} \sqrt{\tau_{\mathrm{i}}} \tag{11}$$

Table 4. Factors C_1, C_3, and C_5 for solid and serrated fins in in-line and staggered arrangements

Solid fins: $C_1 = 0.25\,\mathrm{Re}^{-0.35}$
In-line: $C_3 = 0.2 + 0.65\exp(-0.25\,h/s)$ $C_5 = 1.1 - [0.75 - 1.5\exp(-0.7Nd)]\exp(-2S_{\mathrm{L}}/S_{\mathrm{T}})$
Staggered: $C_3 = 0.35 + 0.65\exp(-0.25h/s)$ $C_5 = 0.7 + [0.7 - 0.8\exp(-0.7Nd^2)]\exp(-S_{\mathrm{L}}/S_{\mathrm{T}})$
Serrated fins: $C_1 = 0.25\,\mathrm{Re}^{-0.35}$
In-line: $C_3 = 0.35 + 0.5\exp(-0.35\,h/s)$ $C_5 = 1.1 - (0.75 - 1.5\exp[-0.7\,Nd)]\exp(-2S_{\mathrm{L}}/S_{\mathrm{T}})$
Staggered: $C_3 = 0.55 + 0.45\exp(-0.35\,\mathrm{h/s})$ $C_5 = 0.7 + [0.7 - 0.8\exp(-0.15Nd^2)]\exp(-S_{\mathrm{L}}/S_{\mathrm{T}})$

Here k_f is the thermal conductivity of liquid phase inside pipe, ρ_f – the liquid phase density inside pipe, μ_f – the liquid phase viscosity, Pr_f – the Prandtl number of liquid phase, and τ_i – the interfacial shear stress. τ_i is calculated by the equation [14]:

$$\tau_i = f_i \frac{\overline{G}_g^2}{2\rho g} \tag{12}$$

Here f_i is an interfacial friction factor evaluated as for single-phase pipe flow at a mean vapor mass velocity G_g. For both solid and serrated fins, effectiveness η is [13]:

$$\eta = 1 - (1 - E)\frac{A_f}{A_t} \tag{13}$$

For solid fins:

$$A_f = \pi n[2h^2 + 2d_o h + d_o b + 2bh] \tag{14}$$

$$A_t = A_f + \pi d_o(1 - nb) \tag{15}$$

E is fin efficiency can be calculated by the equation [13]:

$$E = \frac{1}{1 + 0.330034 m^2 h^2 \left[\dfrac{d_o + 2h}{d_o}\right]^{0.5}} \tag{16}$$

m is a factor which can be calculated by the equation [14]:

$$m = \left(\frac{2h_o}{kb}\right)^{0.5} \tag{17}$$

Here k is the fin thermal conductivity. For serrated fins [13]:

$$A_f = \pi d_o n\left[0.5b + h + \frac{2bh}{w_s}\right] \tag{18}$$

Here w_s is the fin width.

Factor m can be calculated by the equation [14]:

$$m = \left(\frac{2h_o(b + w_s)}{kbw_s}\right)^{0.5} \tag{19}$$

The calculation process is as follows:

(1) Equation (1) which shows energy balance between cool gas (purge air) and control volumes is coupled with eq. (2) which shows heat transfer between purge air and HRSG.
(2) Parameters included in eqs. (1) and (2) are calculated by eqs. (3) to (19).
(3) One second is considered as step calculation.

Results and discussion

For validation the results, the comparison between model and experimental data is applied. Figure 2 shows comparison of system pressure variation during purging time between model and measured data. A good agreement between model and measured data is observed.

Figure 3 shows variation of outlet purge gas temperature from superheaters 1 and 2 and evaporator during purge time. At first, purge gas entered the first control volume (supeheater 2),

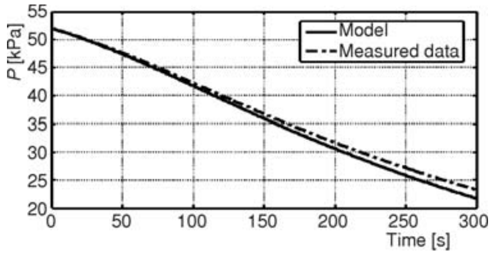

Figure 2. Comparison of system pressure variation during purging time between model and measured data

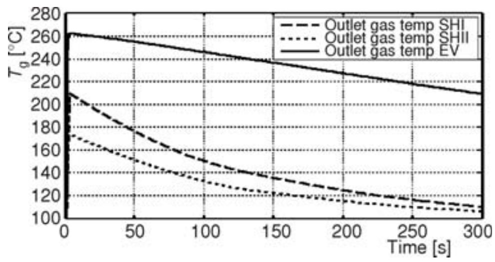

Figure 3. Variation of outlet purge gas temperature from superheaters 1 and 2 and evaporator during purge time

and it is warmed in this control volume, after it, purge gas entered the second and third control volumes (superheater 1 and evaporator) and also it is warmed in these control volumes. It is observed that, outlet purge gas temperature from evaporator is higher than superheater 1 and also outlet purge gas temperature from superheater 1 is higher than superheater 2 during purge time. Figure 4 shows the steam temperature variation in super heaters 1 and 2 and evaporator during purge time. Since, purge cool gas has a minimum temperature in this control volume, and has not been warmed yet, steam temperature in superheater 2 has the most temperature decrease rate during purge time, so steam temperature is decreased from 502 °C to 130 °C. Minimum decrease steam temperature rate is occurred in evaporator. It can be seen from fig. 4, this reduction is about 47 °C during purge time. This variation has two reasons; first reason is related to purge gas which it has been warmed in superheaters 1 and 2 before it entered to evaporator, so the effect of purge gas cooling is reduced; second reason is related to mass of steam, pipes and fins which are more than superheaters 1 and 2.

Figure 5 shows the pipe temperature variation in superheaters 1 and 2 and evaporator during purge time. This variation is similar to fig. 4.

Figure 6 shows condensate mass formation during purge time. It can be seen, condensate formation starts after 80 s of purge start. At the end of purge time, condensate formation is about 238 kg or 238 l which it should be drained from superheaters. Figure 7 shows variation of condensate mass formation through purging

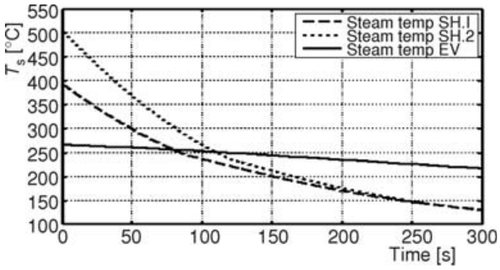

Figure 4. Steam temperature variation in superheaters 1 and 2 and evaporator during

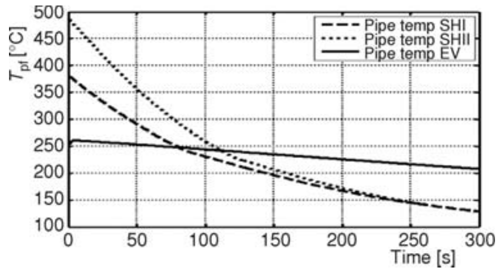

Figure 5. Pipe temperature variation in superheaters 1 and 2 and evaporator during

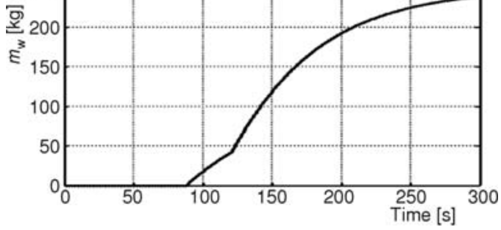

Figure 6. Condensate mass formation during purge time

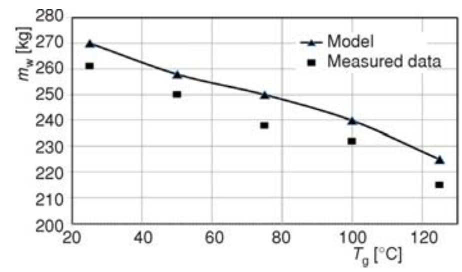

Figure 7. Variation of condensate mass formation through purging time with purge gas temperature

time with purge gas temperature (model and measured output data). Average model error is about 5%.

From this figure, it can be concluded that, if purge gas temperature decreases from 125 °C to 25 °C with same purge gas mass flow rate (386.86 kg/s), condensate mass formation increases from 238 kg to 258 kg.

Conclusions

Boiler and combustion safety codes require that HRSG's are purged prior to start-up to ensure that any remaining combustible gases from the gas turbine are removed and do not present a risk of explosion. This purging process requires the movement of large volumes of cold ambient air through the HRSG which cools the steam within the hot tubes and results in condensation. Large volumes of cool condensate can be formed which then fall onto bottom headers which, because of their thick walls and mass, remain hot. This causes local quenching and possible cracking at stress concentrations such as tube penetrations. It is important that the high volumes of condensate which may be generated during the purge are drained prior to re-start. Otherwise, this condensate would remain at the lower saturation temperature (337 °C at 140 bar) until evaporated, cooling areas of the superheater and reheater even though adjacent sections may have reached the superheater temperature (560 °C). Again this can give rise to large thermal stresses and possible permanent deformation of tubes known as tube bowing. Tube bowing, once it has occurred, can lead to by-passing of hot gases which can then impinge directly on cooler downstream components and result in vibration issues, causing further fatigue failures.In this paper investigated the mathematical modeling of transient HRSG during purge time in order to drain condensate formation in superheaters. For this purpose, a computer program code was written in MATLAB to simulate HRSG during purge time. For validation the model, numerical output of program was compared with measured data in real power plant which average error is about 5%. During purge time, steam temperature decreases from 502 °C, 392 °C, and 266 °C to 130 °C, 130 °C, and 220 °C, respectively, and also steam pressure decreases from 52 bar to 23 bar during purge time. At end of purge time, condensate formation is about 220 l when inlet gas temperature is equal to 100 °C and purge gas mass flow rate is equal to 386.86 kg/s.

Nomenclature

A_i	– internal surface, [m^2]
A_t	– overall heat transfer surface, [m^2]
b	– fin thickness, [m]
C_f	– specific heat of fin, [kJkg^{-1}°C^{-1}]
C_g	– specific heat of purge gas, [kJkg^{-1}°C^{-1}]
Cp_m	– the mixture gas heat capacity, [kJkg^{-1}°C^{-1}]
C_p	– specific heat of pipe, [kJkg^{-1}°C^{-1}]
C_s	– specific heat of steam, [kJkg^{-1}°C^{-1}]
C_w	– specific heat of condensate, [kJkg^{-1}°C^{-1}]
C_1, C_2, C_3, C_5	– constants
d_i	– inlet pipe diameters, [m]
d_o	– outlet pipe diameters, [m]
E	– fins efficiency
ff_i	– fouling factors inside of tubes, [m^2°CW^{-1}]
ff_o	– fouling factors outside of tubes, [m^2°CW^{-1}]
f_i	– interfacial friction factor
G	– gas mass velocity, [kgm^{-2}s^{-1}]
G_g	– mean vapor mass velocity
h	– fin height, [m]
$h_{fg\,p}$	– condensate enthalpy, [(kJkg^{-1}]
h_i	– inlet heat convection coefficient, [Wm^{-2}°C^{-1}]
h_o	– outlet heat convection coefficient, [Wm^{-2}°C^{-1}]

k – fin thermal conductivity, [$Wm^{-1}°C^{-1}$]
k_f – thermal conductivity of liquid phase inside pipe, [$Wm^{-1}°C^{-1}$]
k_m – thermal conductivity of tube wall, [$Wm^{-1}°C$]
L – length of pipe, [m]
m_f – total fin mass in each control volume, [kg]
$m_{g\,p}$ – purge gas mass flow rate, [kgs^{-1}]
m_p – total pipe mass in each control volume, [kg]
m_s – total steam mass inside pipe in each control volume, [kg]
m_{wp} – condensate mass flow rate, [kgs^{-1}]
N – number
N_w – number of pipes wide
n – number of fins per meter, [Nom^{-1}]
Pr – Prandtl number
Pr_f – Prandtl number of liquid phase
Re – Reynolds number
S_L – longitudinal pitch, [m]
S_T – transverse pitch, [m]
s – fin clearance, [m]
T_f – fin temperature, [°C]
$T_{g\,in}$ – inlet purge gas temperature, [°C]
$T_{g\,out}$ – outlet purge gas temperature, [°C]
T_{gm} – average purge gas temperature, [°C]
ΔT_{ln} – log-mean temperature, [°C]
T_p – pipe temperature, [°C]
T_s – steam temperature, [°C]
t – time, [s]
U_i – inside tubes overall heat transfer
U_s – steam internal energy, [$kJkg^{-1}$] coefficient, [$Wm^{-2}°C^{-1}$]
U_t – overall heat transfer conductivity, [$Wm^{-2}°C^{-1}$]
w_s – fin width, [m]

Greek symbols

η – fin effectiveness
μ_f – liquid phase viscosity, [NSm^{-2}]
ρ_f – liquid phase density inside pipe, [kgm^{-3}]
τ_i – interfacial shear stress, [Nm^{-2}]

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