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THE PERFORMANCE MODELLING OF A SINGLE AND DUAL PRESSURE UNFIRED ONCE THROUGH STEAM GENERATOR

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ABSTRACT

During the worldwide combined cycle power plant (CCPP) power plant building boom high pressure steam cycle has been the design of choice for the most power plant developers. In combined cycle power plant; there are many key areas that can be focused on for improving the thermal efficiency of the cycle and their performances. One of these areas is the unfired heat recovery steam generator so-called once through steam generator (OTSG). The unfired utility scale OTSG is a critical component in the combined cycle power plant, since it is the connection between the gas turbine (GT) and the steam turbine (ST) power units. In this paper, an adequate model has been developed for performance simulation of the once through steam generator OTSG in CCPP application. This model is well applicable for single and two-level of the steam cycle operation pressure. The developed model was tested against an existing OTSG installed at Manx Electricity Authority, and the obtained results and conclusions from this effort are satisfactory with a neglected error and will be a magnificent starting point for future improvements.

Keywords CCPP, Gas Turbine, Steam Turbine, HRSG, OTSG:

INTRODUCTION

Today's prevailing market conditions challenge owners and operators of combined cycle power plants (CCPP) to add new capacity at lower cost, operate more efficiently and cycle more frequently to meet stringent environmental requirements. However, one of the key areas that can be focused on for improving the thermal efficiency of the CCPP is the heat recovery so-called once-through steam generator (OTSG). This type of boiler is the first new technology to be introduced in the heat recovery field since the wide scale introduction of combined cycle power plant. The performance of once through steam generator (OTSG) strongly affects the overall performance of a combined-cycle power plant. The OTSG is a critical component in the combined cycle power plant, since it is the connection between the gas turbine (GT) and the steam turbine (ST) power units. OTSG in its simplest form, is a continuous tube heat exchanger, in which preheating, evaporating, and superheating of feed water takes place consecutively as illustrated in Figure 1

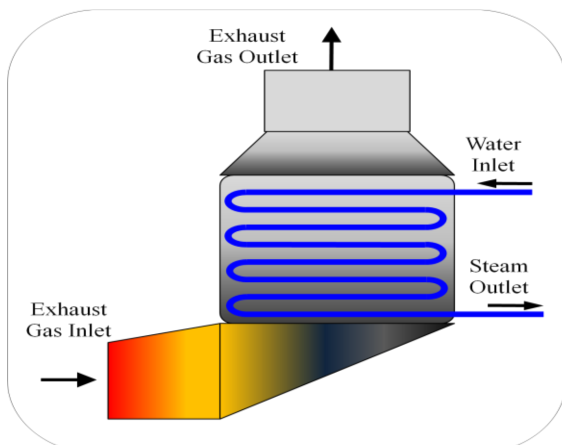

Figure 1 Schematic of OTSG

The key design of the Once-Through boiler is the elimination of a thick-walled, thermally sluggish high-pressure steam drum, and hence, the faster start-up times of OTSG when compared with the conventional HRSG in steam cycle is due to disappearance of the drum and the circulating accessories. This type of steam generator provides higher thermal efficiency because of its high reliability for high cycle operation pressure in the range of 165 Bar. The higher efficiency means lower cost per MW unit output and this is in turn increase the same unit output per unit of CO_2 emission, which often facilitates the approval of legislative permits.

Two-level of steam operation pressure in the OTSG is possible due to the improvement in gas turbine technology.The Once-Through heat recovery steam generator provides:

- Rapid start-up (25 minutes cold start)
- No delay for the warm-up of drums (the boiler follows the gas turbine on start-up)
- Unlimited daily cycling
- Fully sequenced automatic start-up

Currently, the Once- through heat recovery steam generators OTSGs are employed in a number of applications. The largest unites are used in combined cycle power plants recovering heat from exhaust gas of gas turbines (GTs). These are referred to as utility scale OTSG. This type of boiler is also used in various industrial processes and this is referred to as industrial OTSGs. The use of once through steam generator for combined cycle gas turbine power plant applications relatively new, with some designs including supplementary firing systems that allow the production of steam when gas turbine is not available [1]. The nuclear power generation industry has also adopted this type of heat exchanger with particularities such as helical tubing design [2] [3].

The structure robustness of this type of boiler permits the generation of steam at extreme pressure conditions. This is the case with the fossil-fuel fired once-through boilers for supercritical water pressure conditions used in traditional steam power plant [4][5][6]. The operation behaviour and performance characterisation of an OTSG as part of a combined cycle power plant have been presented and discussed by [7]. Research on the OTSG performance simulation is not abundant, although publications devoted to the analysis of the thermo-hydraulic performance of OTSGs are more frequent. Mathematical model and design of an advanced once-through heat recovery steam generator has been developed by [8].

Unlike conventional heat recovery steam generator (HRSG), Once-through steam generator (OTSG) does not have defined economizer, evaporator or super heater sections. The point at which the water-steam interface exists is free to move forward and back ward through the horizontal tube bundle depending on the available heat from the gas turbine, water mass flow rate and the pressure of the water. The simplicity of OTSG justifies its employment in combined cycle configurations compared with the conventional boiler (drum HRSG) where, many items of equipments installed as illustrated in Figure 2.

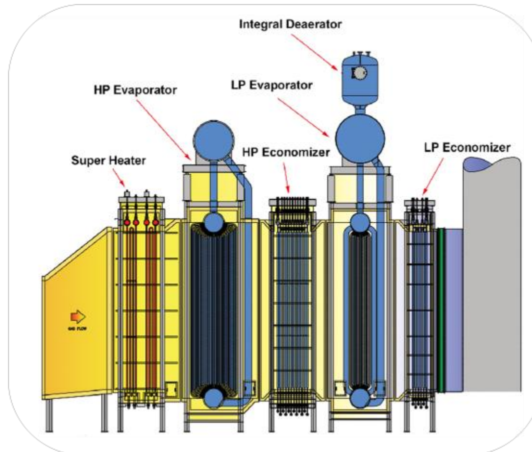

Figure 2 Schematic of Conventional HRSG

The performance modelling of OTSG is totaly different from the conventional boiler where each tube plays well-defined role: economizer, evaporator, super heater and the water/steam separator (drum). Each section in the conventional HRSG has specific amount of surface area, geometry, and tube layout etc to achieve particular heat transfer process dependent on the phase of the fluid. (i.e., the evaporator required more surface area due to the phase changes in the fluid compared with the economizer and super heater where less area needed for single phase).

In conventional HRSG, the heat transfer regime in each section is fixed by the design. The disappearance of the drum and the HRSG's sections introduces different understanding to the OTSG behaviour. However, due to the unfixed boiling point in the tubes rows of OTSG, the flow regions considered as a moveable.

According to this phenomenon, the surface area of tubes bundles should be designed and sized in a way that ensure equality between all tubes to make them capable to transfer sufficient heat flow rate when the phase changes process occurs at the inlet of the first or last tubes row. In this case, the OTSG tubes bank should be consider as evaporator, where more surface area required due to the fluid phase change (huge heat flow rate will be transferred through the surface area during the boiling and evaporating process) and thus to ensure that the tubes in each pass are capable to absorb and transfer enough heat when the boiling point travelling among the tubes bundles. In this paper a realistic model of OTSG performance simulation presented.

This model is well applicable for single and two-level of the steam cycle operation pressure. Furthermore, the proposed method appeared to be not limited for OTSG in combined cycle gas turbine (CCPP), but well fit for combined heat and power (CHP) (industrial scale) where no phase change in the fluid steams flowing through the exchanger takes place. The developed model was tested against an existing OTSG installed at Manx Electricity Authority, and the obtained results and conclusions from this effort are satisfactory with a neglected error and will be a magnificent starting point for future improvements in OTSG field.

NOMENCLATURE

A_{exh}	Exhaust Gas Surface Area
Bo	Boiling Number
Cp	Fluid Specific Heat
C*	Heat Capacity ratio
di	Tube Inner Diameter
do	Tube Outer Diameter
E	Convention Enhancement Factor
F	Tube Arrangement Factor
f	Tube Friction Factor
G	Water Mass Velocity
L	Tube Length
LSP	Live-steam pressure
LST	Live-steam temperature
LST_O	Reference Live-steam temperature
m_w	Water Mass Flow Rate
m_{wO}	Reference Water Mass Flow Rate
M	Molecular Weight
Nu	Nusselt Number
Δi	Enthalpy Drop
h	Fluid Heat Transfer Coefficient
N_{TU}	Number Transfer Unit
Nu	Nusselt Number
k	Fluids Thermal Conductivity
Pr	Prandtle Number
PP	Pinch Point
P_r	Reduced Pressure
Re	Reynolds Number
S	Boiling Suppression Factor
S_D	Diagonal Tube Pitch
S_L	Longitudinal Tube Pitch
S_T	Transverse Tube Pitch
ΔT	Temperature Difference
U	Overall Heat Transfer Coefficient
V	Exhaust Gas Inlet Velocity
Q	Heat Flow rate

Greek Symbols

ζ	Exhaust Gas Friction Factor
ε	OTSG effectiveness
μ	Dynamic Viscosity
Φ	Heat Flux
η	Efficiency

Subscripts

av	Available
act	Actual
cb	Convection Boiling
c	Cold
exh	Exhaust Gas
ev	Evaporating process
f	Water
g	Vapor
h	Hot
HP	High Pressure Circuit
in	Inlet
lm	Long Mean
L	Liquid
LP	Low Pressure Circuit
i	Inner
max	Maximum
nb	Nucleate boiling
out	Outlet
o	Outer
ρ	Fluids Density
pre	Preheating process
sup	Super heating process
sp	Single phase
t	Tube
tp	Two Phase
x	Vapor Quality
Xtt	Martinelli parameter

Abbreviations

CCPP	Combined Cycle Power Plant
HRSG	Heat Recovery Steam Generator
OTSG	Once-Through Steam Generator

ASSUMPTIONS FOR HEAT TRANSFER ANALYSIS

The analysis of the heat transfer mechanisms inside any type of heat exchanger involves the appropriate definition of the thermodynamic systems and volumes of control. The control volume is a region of space bounded by a control surface through which energy pass. The boundaries of the volume are the OTSG wall through which two fluids streams cross the boundaries without mixing: water mass flow rate m_c and gas turbine exhaust gas mass flow m_h as illustrated in Figure 3.

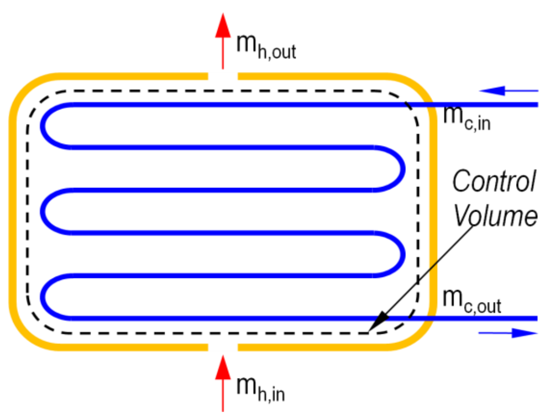

Figure 3Thermodynamics System of OTSG

To analyze the heat exchanger transfer problem, a set of assumptions are introduced so that the resulting theoretical model is simple enough for the analysis. The following assumptions are made for the OTSG rating (performance) problem formulations: the energy balances, and rate equation.

1. The OTSG operate under steady-state conditions [i.e., constant flow rates].
2. The fluid flow rate is uniformly distributed through the OTSG on each fluid side in each pass.
3. Overall extended surface efficiency $\eta_{\circ}$ is considered uniform, constant and unity (neglecting fin ineffectiveness).
4. In multi-pass OTSG, heat transfer area is distributed uniformly in each pass due to the unfixed boiling point (moveable flow regions).
5. Overall multi-pass counters flow arrangement.
6. Each fluid is considered unmixed between passed.
7. The overall heat transfer coefficient (U) is constant (independent of position) throughout the tubes bank of OTSG.
8. The velocity and temperature at the entrance of the OTSG are uniform over the flow cross section. There is no gross misdistribution at the inlet of the OTSG.
9. No fluid leakage and flow bypassing of either of two fluid streams occur in the OTSG.
10. Heat loss to or from the surroundings are negligible [i.e., OTSG outside walls are adiabatic, $\theta=0$].
11. No work interaction for each fluid streams ($W=0$).
12. Kinetic and potential energy changes are assumed negligible ($ke=0, pe=0$).

ENTHALPY & HEAT TRANSFER RATES EQUATIONS

To design or to predict the performance of the OTSG, it is essential to relate the total heat transfer rate to quantities such as the inlet and outlet fluids temperature, the overall heat transfer coefficient, and the total heat transfer area. Two such relations may readily be obtained by applying overall energy balances to the hot and cold fluids as shown in Figure 4

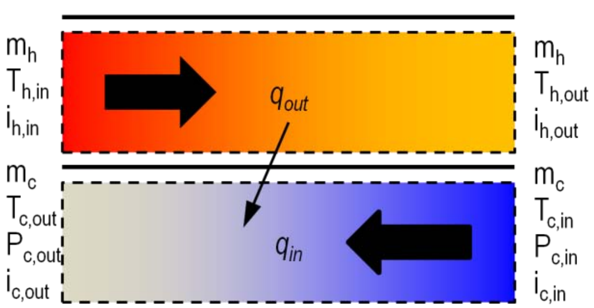

Figure 4 Overall Heat Balance for Hot and Cold Fluids of OTSG

The overall heat-transfer between the hot and cold fluids is given in the following equations:

$$Q_{out} = m_{exh}\Delta i_{exh} \qquad (1)$$

$$Q_{in} = m_{f,g}\Delta i_{f,g} \qquad (2)$$

For single phase fluids, the overall heat flow rates can be evaluated in terms of specific heat and temperature difference and is equal to:

$$Q_{out} = m_{exh}\Delta i_{exh} = m_{exh}Cp_{exh}\Delta T_{exh} \qquad (3)$$

$$Q_{in} = m_{f,g}\Delta i_{f,g} = m_{f,g}Cp_{f,g}\Delta T_{f,g} \qquad (4)$$

The overall heat transfer rate can also be evaluated in terms of overall heat transfer coefficient, surface area, and long mean temperature difference as:

$$Q = UA\Delta T_{lm} \qquad (5)$$

Equations (1) or (2) and (5) are well-known relationship from thermodynamics. They relate the heat transfer rate Q with the enthalpy rate change for an open no adiabatic system with a single bulk stream (*water*) entering and leaving the OTSG under isobaric conditions. Equation (5) reflects a convection-conduction heat transfer phenomenon in the OTSG. The heat transfer rate (Q) is proportional to the heat transfer area (*A*) and log-mean temperature difference (ΔT_{lm}) (for counter flow) or related to it in a way that involves terminal temperature differences between the water/steam and exhaust gas of the gas turbine passing through the OTSG such as ($T_{h,i}$-$T_{c,o}$) and ($T_{h,o}$-$T_{c,i}$). Accordingly, log mean temperature different may take the following form:

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} \qquad (6)$$

OTSG EFFECTIVENESS

Effectiveness (ε) is a measure of thermal performance of a boiler. It is defined for a given boiler of any flow arrangement as a ratio of actual heat transfer rate from the hot fluid to the cold fluid to maximum possible heat transfer rate thermodynamically permitted:

$$\varepsilon = \frac{Q_{act}}{Q_{max}}$$

The boiler effectiveness is function of heat capacity ratio (C*), number transfer unit (N_{TU}), and flow arrangement:

$$\varepsilon = f(C^*, N_{TU}, FlowArrangement)$$

The fluid heat capacity ratio (C*) is simply the ratio of mass flow rate times specific heat capacity for two streams can be calculated from:

$$C^* = (mc_p)_{\min} / (mc_p)_{\max} \tag{7}$$

In boiling process, heat capacity ratio will equal to zero (C*=0, constant Cp), therefore, OTSG effectiveness will be function of (N_{TU}) and flow arrangement only. However, due to unfixed flow boiling regions in the horizontal tube bundle, the effectiveness of the OTSG is given by the following expression:

$$\varepsilon = 1 - \exp(-N_{TU}) \tag{8}$$

Equation (8) is well-known relation for heat exchanger effectiveness with capacity ratio(C*=0) (evaporating or condensation) of one or two streams.

T-Q DIAGRAM

The heat balance between the two fluids (exhaust gas and water/steam) is illustrated in Figure 5. The T-Q diagram represents the relationship between the exhaust gas temperature and the heat flow rate. This shows that more heat will be recovered by the cold side (blue lines) as the temperature of the exhaust gas (red lines) drops. As a result of this, exergy losses decrease due to better matching of the gas-cooling curve with the water/steam curve in the heat exchanger [9]

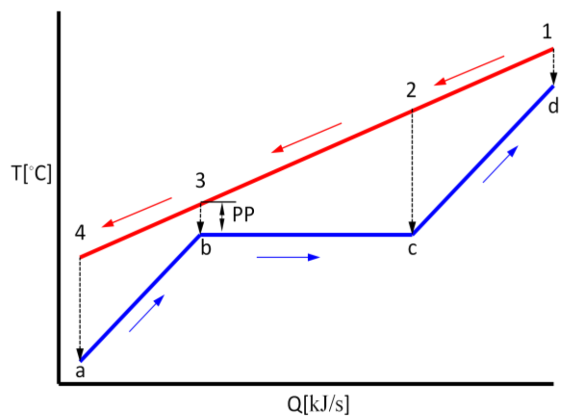

Figure 5 T-Q Diagram for Single Pressure of OTSG

PINCH POINT

The critical temperature difference that influences the amount of heat transfer surface area in the boiler is the pinch point. It is the smallest temperature difference between water/steam and exhaust gas. It is designated by (PP) in Figure 5 as the difference between the exhaust gas at point (3) and the water/steam at point (b). The pinch point limits the amount of heat that can be recovered in most HRSG designs. For many general purpose of heat recovery steam generator, such as those found in CCPP, refineries and chemical plants, a pinch in range of 10-15°C provides an economical design with a realistic payout.

FLOW REGIONS

The OTSG for CCPP applications utilized to produce steady state superheated steam condition to drive the steam turbine in the bottom cycle (steam cycle). Therefore, the design engineer must ensure that the boiling and evaporating process complete before the last few rows leading to the OTSG outlet header. 5-10 tubes row is extremely important as a safety margin to make sure that the steam quality reach to the greater than unity (superheated). The water boiling process inside horizontal tubes rows of OTSG is illustrated in Figure 6[10]

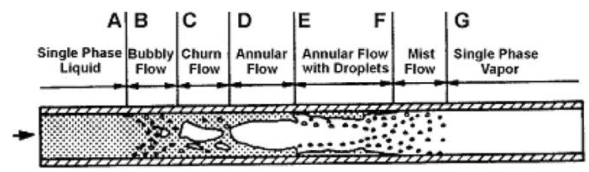

Figure 6 Forced Convection Boiling Process in Horizontal Tube [10]

Figure 6; clearly illustrate the unfixed flow regions of single and multi-phase existing together inside horizontal plain tube. These regions are always traveling in a horizontal tubes backward and forward dependent on the water mass flow rate, operation pressure and available heat.

Reducing the water mass flow rate for a fix heat flow rate (kJ/s) will shift the boiling and evaporating regions (B-F) backward towards the single phase liquid (A) and thus the phase change process complete in a short time and therefore, steady state superheated steam condition reached before/or within the last tubes rows (safety margin tubes rows). In the other case scenario, increasing the water mass flow rate will reposition the region (B-F) forward towards the single phase vapor region (G) and thus the phase change process will takes long time and requires many tubes rows number till complete and may reach beyond the safety margin (last few tubes rows) causing the disappearance of region (G) and therefore, steady state superheated steam condition will not achieved.

HEAT TRANSFER COEFFICIENT

1. COLD SIDE COEFFICIENT (h_c)

The heat transfer phenomena associated with liquid-vapor phase change play an important role in boiling process present, among others, in the heat exchangers of power plans. Example of special interest is nowadays the heat recovery steam generator (conventional HRSG or OTSG), component of the combined cycle power plants [11].

1.1 SINGLE PHASE (h_{sp})

As indicated in Figure 3, water enters the horizontal tube below its boiling point as a compressed liquid. The heat-transfer coefficient of water or vapor as pure single phase is determined by the following equation:

$$h_{sp} = \frac{Nu_{di}\, k}{d_i} \qquad (9)$$

Where the subscript (*sp*) refer to single phase (liquid or vapor) and (*i*) is the inner tube diameter.

1.1.1 Laminar Flow

The Nusselt number in equation (9) is average, non-dimensional and given by the following empirical relation proposed by Sieder and Tate [12] for laminar flow condition $(\mathrm{Re}_{di} \cong 2300)$:

$$Nu_{di} = 1.86(\mathrm{Re}_{di}\,\mathrm{Pr})^{1/3}\left(\frac{d_i}{L_t}\right)^{1/3} \qquad (10)$$

1.1.2 Turbulent Flow

For fully developed turbulent flow condition in smooth tube, Petukhov[13] has developed a more accurate, although more complicated, expression to evaluate the Nusselt number taken the following form:

$$Nu_{di} = \frac{(f/8)(\mathrm{Re}_{di} - 1000)\mathrm{Pr}}{1.07 + 12.7(\mathrm{Pr}^{2/3} - 1)(f/8)^{1/2}} \qquad (11)$$

The application range of this empirical correlation is $0.5 \le \mathrm{Pr} \le 2000$, $3000 \le \mathrm{Re}_{di} \le 5*10^6$.The friction factor may be obtained from the following equation:

$$f = (1.82 \log \mathrm{Re}_{di} - 1.64)^{-2} \qquad (12)$$

1.2 MULTI-PHASE (h_{tp})

Flow boiling in tube is subject to two important heat transfer mechanisms:

- Nucleate boiling heat transfer h_{nb}.
- Convective boiling heat transfer h_{cb}.

These two mechanisms are combined to provide local predictions for flow boiling heat transfer coefficients (h_{tp}) in horizontal and smooth tube under isobaric condition. The basic equation for two-phase flow boiling proposed by Gungor and Winterton [14] is:

$$h_{tp} = Eh_f + Sh_{nb} \qquad (13)$$

The liquid phase convective heat transfer coefficient (h_L) is given by the Dittus-Boetler [15] correlation for the fraction of liquid flowing alone in a tube of internal diameter (d_i) using mass velocity of *G* (*1*-*x*), as:

$$h_f = 0.023 R_f^{0.8}\,\mathrm{Pr}_f^{0.4}\left(\frac{k_f}{d_i}\right) \qquad (14)$$

The nucleate pool boiling coefficient is obtained with the aid of Cooper [16] equation:

$$h_{nb} = 55 p_r^{0.12}(-0.4343 \ln p_r)^{-0.55} M^{-0.5} \phi^{0.67} \qquad (15)$$

Equation (15) is dimensional and gives the heat transfer coefficient in (W/m^2.C). The heat flux ϕ must be introduced in (W/m^2). M is the molecular weight and (p_r) is the reduced pressure, which is the ratio of the saturation pressure (p_{sat}) to the critical pressure (p_{crit}). The new enhancement factor (*E*) is a function of the Martinelli parameter and also the heat flux via the Boiling number:

$$E = 1 + 24000\, Bo^{1.16} + 1.37\left(\frac{1}{X_{tt}}\right)^{0.86} \qquad (16)$$

Where X_{tt} is Martinelli parameter given by the following expression:

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_f}\right)^{0.5}\left(\frac{\mu_f}{\mu_g}\right)^{0.1} \qquad (17)$$

The effect of the heat flux on nucleate boiling is characterized by the boiling number Bo, which is defined as:

$$Bo = \frac{\phi}{G i_{fg}} \qquad (18)$$

The new completely empirical boiling suppression factor (S) is:

$$S = (1 + 0.00000115\, E^2\, \mathrm{Re}_L^{1.117})^{-1} \qquad (19)$$

With (Re_L) based on *G* (*1*-*x*).

2. SMOKE SIDE COEFFICIENT (h_h)

Heat-transfer between a tube bundles (or bank) and a fluid flowing normal to the tubes' axes depends on the geometrical arrangement of the tubes. Experimental results are mainly available for two arrangements; namely the aligned and staggered tube arrangement (see.Figure 7).

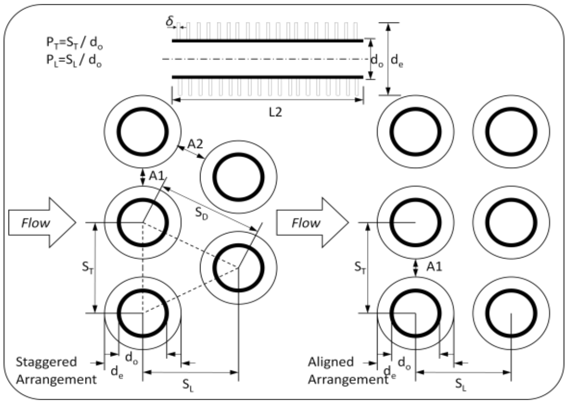

Figure 7 Tube Arrangement in a Bank

A configuration is characterized by the tube outer diameter (d_o) and by the transverse pitch (S_T) and longitudinal pitch (S_L), measured between centre of adjacent tubes. The average heat-transfer coefficient for an entire tube bundle can be evaluated by the following equation:

$$h_{exh} = \frac{Nu_{do}\,k}{d_o} \tag{20}$$

The average Nusselt number is given by the following empirical correlation developed by Hausen[17]: For aligned arrangement

$$Nu_{do} = 0.34F\,\mathrm{Re}_{do,\max}{}^{0.61}\,\mathrm{Pr}^{0.31} \tag{21}$$

And

$$F = 1 + \left(P_L + \frac{7.17}{P_L} - 6.52\right)\left(\frac{0.266}{(P_T - 0.8)^2} - 0.12\right)\left(\frac{1000}{\mathrm{Re}_{do,\max}}\right)^{0.5} \tag{22}$$

For staggered arrangement

$$Nu_{do} = 0.35F\,\mathrm{Re}_{do,\max}{}^{0.57}\,\mathrm{Pr}^{0.31} \tag{23}$$

And

$$F = 1 + 0.1P_L + \frac{0.34}{P_T} \tag{24}$$

And P_T and P_L are transverse and longitudinal ratio respectively. ($P_T=S_T/d_o$), ($P_L=S_L/d_o$).

OVERALL HEAT TRANSFER COEFFICIENT (U)

The coefficient (U) presented in equation (5) is the overall heat-transfer coefficient that is important to predict the boiler performance and may be expressed in terms of water/steam (internal) and exhaust gas (external) heat transfer coefficients taken the following form (assuming wall and fouling resistance is negligible):

$$U = \frac{1}{(1/h_i) + (1/h_o)} \tag{25}$$

Equation (25) is showing that the overall coefficient. It is a component of an internal and external heat transfer coefficient. However, In the case of gas-liquid, or gas-phase change heat-transfer, the thermal resistance associated with the gas side is significantly higher than the one associated with liquid or phase change, due to the poorness of the exhaust gas thermodynamic properties such as thermal conductivity, density, and viscosity etc and thus low heat –transfer coefficient h_o.

Hence a low (h_o) is often compensate by a high surface area (A_o) to make $(\eta hA)_o \approx (\eta hA)_i$ assuming surface efficiency (η) is unity[18] .This is the reason the surface area on the gas side is about 5 to 10 times higher than that on the liquid-side, or phase change-side when the liquid or phase-change heat transfer coefficient (h_i) is 5 to 10 times higher than the (h_o) of the gas side. This would explain why fins are used on the gas sides in a gas-liquid or gas-to-phase change exchanger. The overall heat transfer coefficient (U) may be defined optionally in terms of the surface area of either the gas-side or liquid-side. Thus

$$UA = U_o A_o = U_i A_i \tag{26}$$

Note, that calculation of UA product does not requires designation of the inside or outside since ($U_iA_i=U_oA_o$). However, calculation of an overall coefficient depends on controlling resistance of the fluids and in turn this could be based on the inside or outside side surface area, since $U_i \neq U_o$. Nevertheless, the coefficient proportionality in equation (5), (25) and (26) is the overall heat transfer coefficient (U).

According to [19], for most HRSG applications a value of 45 W/m^{2}C is typical for each sections of the conventional HRSG. The coefficient values of 56.41, 62.45 and 48.15 were used by [20] for preheating, evaporating, and superheating in OTSG respectively. Moreover, a value between 40-50 W/m^{2}C for the overall heat transfer coefficient was found in [21], [22], and [23]. However, the value of (U) given by [20] will be used with the aid of equation (5) to specify the water flow regions in the OTSG's tubes bank.

PRESSURE DROP

The pressure loss of the unfired OTSG is extremely important due to its significant impact on the gas turbine performance. Furthermore, the water side or two-phase pressure drop will not be discussed in this paper due to its small effect on the boiler or steam cycle performance and considered as neglected.

SMOKE SIDE

For the flow gases normal to banks of tube may be calculated with the aid of the following equation recommended by Holman [23] and Kreith[24]

$$\Delta p = 1.995 N_r \zeta V_{\max}^2 \rho_{exh} \tag{27}$$

The empirical friction factor ζ is given by Jakob [25] as:

$$\zeta = \left\{ 0.25 + \frac{0.118}{\left[(S_T - d_o)/d_o \right]^{1.08}} \right\} \left(\mathrm{Re}_{do,\max} \right)^{-0.16} \tag{28}$$

For staggered tube arrangements, and

$$\zeta = \left\{ 0.04 + \frac{0.08 S_L / d_o}{\left[(S_T - d_o)/d_o \right]^{0.43+1.13 d_o/S_L}} \right\} \left(\mathrm{Re}_{do,\max} \right)^{-0.15} \tag{29}$$

Exhaust gas maximum velocity is usually occurs at transverse or diagonal planes (A1) or (A2) respectively as illustrated in Figure 7.For flows normal to in-line tube banks the maximum exhaust gas velocity will occur at the transverse plane (*A1*) of Figure 7. Based on mass continuity for an incompressible fluid V_{max} can be calculated by [26]:

$$V_{\max} = \frac{S_T}{S_T - d_o} V \tag{30}$$

Where (V) is OTSG exhaust inlet velocity. For staggered configuration, the maximum exhaust gas velocity may occur at either the transverse plane (A1) or the diagonal plane (A2) of Figure 7. It will occur at (A2) if the tube rows are spaced such that

$$2(S_D\text{-}d) < (S_T\text{-}d)$$

Hence V_{max} occur at (*A2*) if

$$S_D = \sqrt{S_L^2 + \left(\frac{S_T}{2} \right)^2} \langle \frac{S_T + D}{2}$$

In which case it is given by

$$V_{\max} = \frac{S_T}{2(S_D - d_o)} V \tag{31}$$

OTSG PERFORMANCE SYSTEM EQUATIONS

The flow chart presented in Figure 5 represents the order for solving the system equations of OTSG performance simulations model based on equations (1) to (29). The inlet operation conditions of the hot and cold side are known (point (1) and (a)). For given water inlet saturated pressure (P_{sat}), the saturated temperature (T_{sat}) will be determined from saturated table of water. In the other hand, the only variables considered as unknown are that denoted by (2), (4) and (d).

Table 1 Known and Unknown variables based on Figure 5

Exhaust gas side(red curve)		*Cold side(blue curve)*	
1	√	*a*	√
2	*X*	*b*	√
3	√	*c*	√
4	*X*	*d*	*X*

√ Known , X Unknown

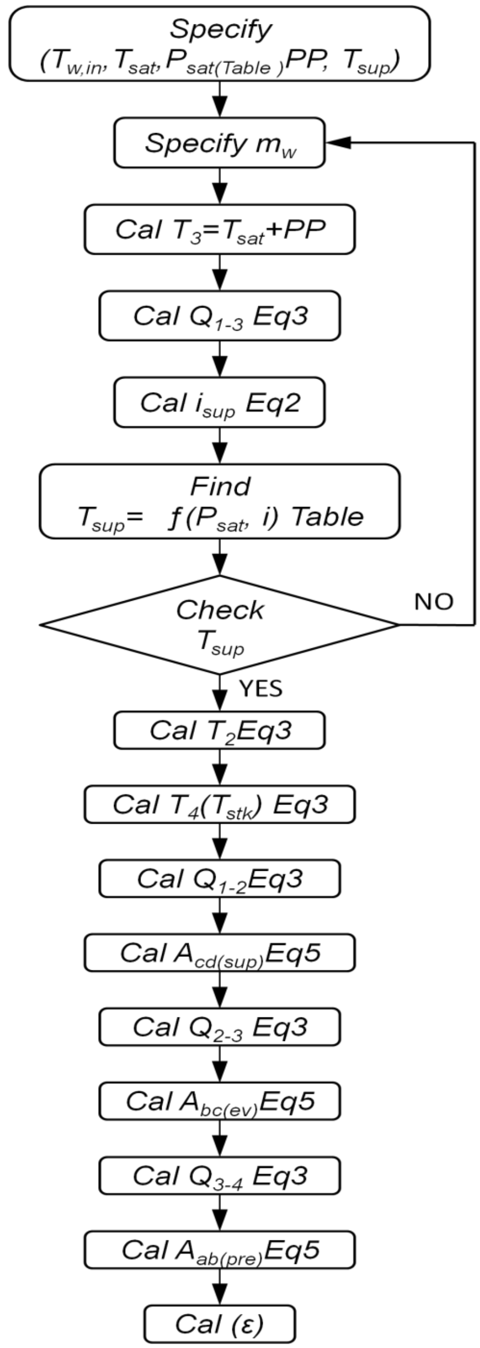

Figure 8 OTSG Performance Flow Chart For single and double pressure mode

MODEL VALIDATION

In this section, the developed model has been applied to simulate an existing Unfired Once-Through Steam Generator installed in Manx Electricity Authority power plant as a critical connection point between gas and steam cycle. The model validation has been carried out based on the following assumption: the operating conditions of the exhaust gas remain unchanged over predicting the design and off design performance of the unfired once-through steam generator. This will help the performance and design engineer to have a clear picture about the impact of the UOTSG operating parameters on its performance.

A comparison between simulated and observed results is presented. Table 2 shows exceptional results with error values below 1%. The developed model has shown a very good agreement compared with the realistic unfired OTSG performance. Each calculated and measured parameters provided by MEA have been used to evaluate the accuracy of UOTSG model.

Table 2 Model Validation and Comparison Results

Variables	*OTSG Model*	*MEA*	*deviation*
$\eta_{OTSG,single}$	67.99	68.5	-0.74%
$\eta_{OTSG,double}$	80.36	79.8	0.71%
LSP_{HP}	49.54	50	-0.92%
LST_{HP}	484.35	485	-0.13%
$m_{w,HP}$	34450	34560	-0.32%
$T_{stk,single}$	179.4	180	-0.33%
LSP_{LP}	4.97	5	-0.60%
LST_{LP}	255	256	-0.39%
$m_{w,LP}$	7910	7920	-0.13%
$T_{stk,double}$	120.4	120	0.33%

RESULTS AND DISCUSSIONS

When the OTSG operate in single pressure mode (HP only), the water mass flow rate was used as a control variable. However, it is actuation depends on predefined operating condition of the steam cycle. The super heated steam temperature has been used as a control parameter due to the limited steam turbine material temperature.

Figure 9 illustrates the variation of the OTSG efficiency with water mass flow. Therefore, as mass flow increases, the OTSG efficiency increases due to the drop in stack temperature. Mass flow rate has also a significant effect on the live-steam temperature as illustrated in Figure 10. Therefore, when the water mass flow increases, the superheated steam temperature decreases at fix heat flow rate. Furthermore, the stack temperature was used as a control parameter for low pressure circuit (LP) instead of the superheated steam temperature due to the importance for keeping stack temperature above the dew point (100°C). Figure 11 illustrate that increases water mass flow rate will decrease the stack temperature, in another word; more heat will be recovered at high mass flow rate.

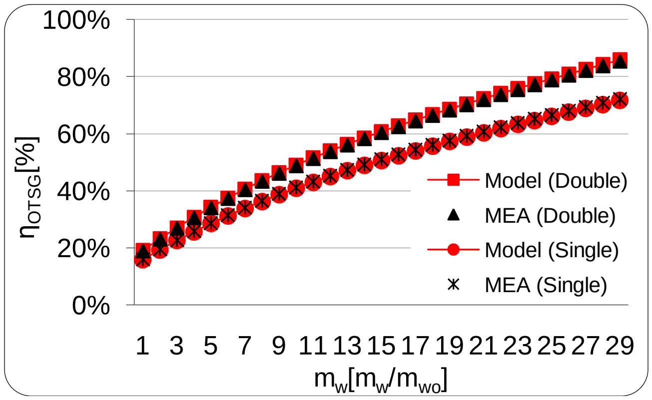

Figure 9 Variation of OTSG Efficiency with feeding water mass flow rate

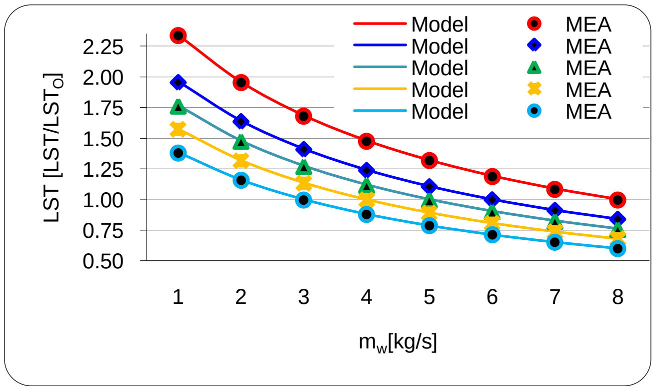

Figure 10 Variation of live steam temperature with water mass flow rate

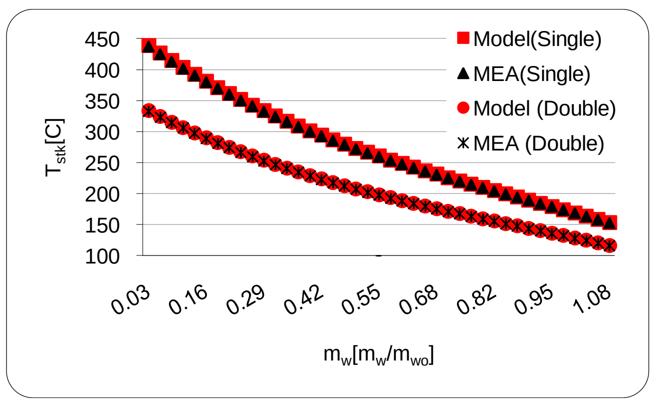

Figure 11 Variation of stack temperature with water mass flow rate

Figure 12 Illustrate the OTSG effectiveness as function of number transfer unit (heat transfer size). Therefore, when the N_{TU} small the OTSG effectiveness is low and when N_{TU} large, effectiveness approach asymptotically the limit imposed by flow arrangement and thermodynamic considerations.

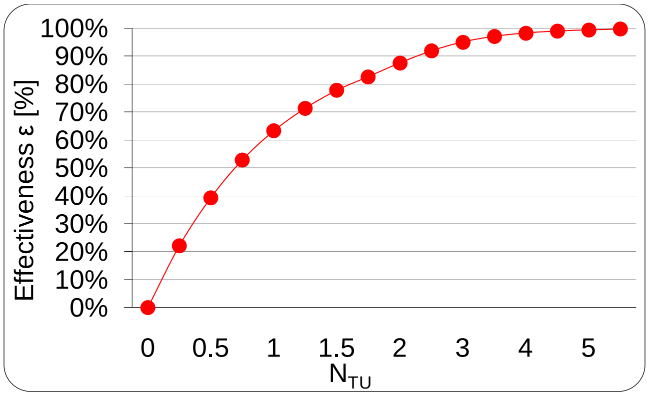

Figure 12 OTSG Effectiveness

Figure 13 represents multi-pass overall-counter flow arrangements with unmixed between the hot and cold fluids. It is clearly shown that OTSG with this tube configuration is the best ever design for CCPP application where demands of sufficient superheated steam generation at high heat transfer effectiveness such as 90%.

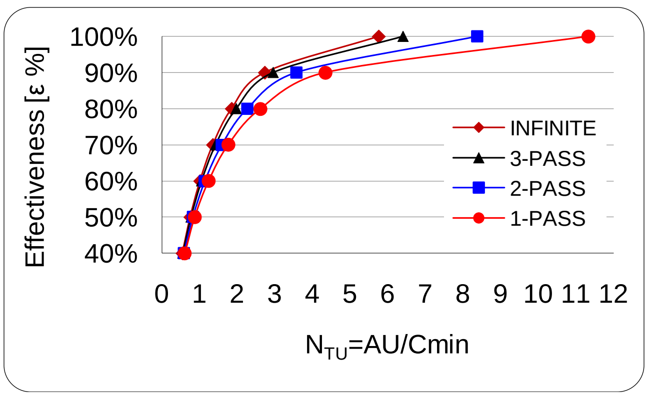

Figure 13 Multi-pass effectiveness

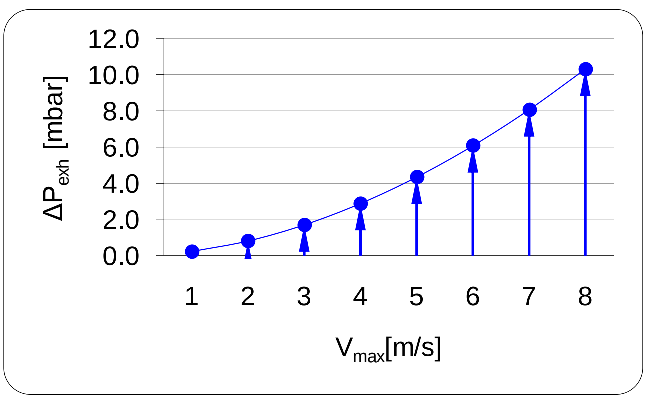

Figure 14 Variation of exhaust gas pressure drop with maximum velocity

Exhaust gas pressure drop is an important parameter in evaluating the OTSG performance analysis due to it is significant impact on the gas turbine performance. Increasing the OTSG pressure drop will increase the negative effective on the Bryton cycle efficiency, and thus CC efficiency. OTSG pressure drop is function of exhaust maximum velocity as illustrated in Figure 14.

CONCLUSION

New approach for Performance simulation of Unfired Once-Through Steam Generator for CCPP application has been presented. The UOTSG model was created in a user friendly and flexible environment into Microsoft Excel with excellent feedback from Manx Electricity Authority. The adequate developed model has been used to predict the UOTSG performance at design and off design point. This model is well applicable for single and two-level of steam cycle operation pressure and also capable to cover wide range of applications including combined heat and power (CHP). The importance of this generic model is to simulate the performance of OTSG for any application at different operation conditions. However, the code allows the user to adapt the model to any physical specifications and operating conditions required. The obtained results and conclusions from this effort are satisfactory with errors value below 1% and will be a magnificent starting point for future improvements.

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REFERENCES

[1]Duffy, T. E. and Schneider, P. H. (1993), "Advanced combined-cycles from 8 MW to 23MW". Proceedings of the 55th Annual Meeting of the American Power Conference. Part 1 (of 2); Chicago, IL, USA. Publ by Illinois Inst of Technology; 1733-1741.

[2]Abdalla, M. A. (1994). "Four-region, moving-boundary model of a once-through, helicalcoil steam generator," Annals of Nuclear Energy. 21(9):541-562.

[3]Bruens, N. W. S. (1981), "PWR plant and steam generators dynamics". Afdeling der Werktuigbouwkunde (Report) WTHD ed. Delft, Holland: Technische Hogeschool Delft.; 128.

[4]Bethune III, J. L. (1974), "Use of the computer for dynamic analysis of a power plants performance". 17th Int ISA Power In strums Symp; Boston, Mass. ISA; 17-26.

[5]Leithner, R. (1983), "Comparison of Once-through Forced Flow Boilers, Once-through Forced Flow Boilers with Full Capacity Circulation, and Natural Circulation Boilers". VGB-Kraftwerkstechnik.; 63(7):553-568.

[6]Jarkovsky, J.; Fessl, J., and Medulova, V. (1989). “Steam generator dynamic mathematical modeling and its using for adaptive control systems testing. Power Systems Modeling and Control Applications” - Selected Papers from the IFAC Symposium; Brussels, Belg. Publ by Pergamon Press Inc; 167-174.

[7]Dechamps, P. J. and Galopin, J.-F. (1997), “Once-through heat recovery steam generators working with sub- and supercritical steam conditions for combined cycles. Proceedings of the 1997 International Gas Turbine & Aeroengine Congress & Exposition; Orlando, FL. USA: ASME; c199710p.

[8]Marie-Noëlle, D., Georges, H. (2004),”Mathematical modelling and design of an advanced once-through heat recovery steam generator”. Computer and chemical engineering 28:651-660.

[9]Rolf, Kehlhofer. Frank, Hannemann. Franz, Stirnimann. (2009), “Combined-Cycle Gas & Steam Turbine Power Plants”. 3rd Ed. PennWell.

[10]John, G. Collier. John, R. Thome.(1994),”Convective Boiling and Condensation”, 3rd Edition. Oxford Science Publications.

[11]Horlock, J. H., (2003), “Advanced Gas Turbine Cycles” Elsevier Science, Oxford.

[12]Sieder, E. N and Tate, G. E. (1936).”Heat transfer and pressure drop of liquids in tubes. Industrial and Engineering Chemistry, 28, 1429-1435.

[13]Petukhov, B. S. (1970). “ Heat transfer and friction in turbulent pipe flow with variable physical properties”.ln advanced in heat transfer, Vol.6, edited by Hartnett, J P. and Irvine, T. F. Academic Press, New York, 504-564.

[14]Gungor, K. E. and Winterton, R. H. S. (1986).”A general correlation for flow boiling in tubes and annuli”. Int. J. Heat Mass Transfer, 29, 351-358.

[15]Dittus, F. W. and Boetler, L. M. K. (1930). “Heat transfer in automobile radiators of the tubular type. Ln: University of California (Berkeley) Publications on Engineering, October 2, 443-461.

[16]Cooper, M. G. (1984). “Heat flow rates in saturated nucleate pool boiling a wide-ranging examination using reduced properties”. Adv. Heat Transfer, 16, 157-239.

[17]Hausen, H. (1983). “Heat transfer in counter flow, parallel flow and cross flow. McGraw-Hill, New York.

[18]Shah, R, K., (2003) “Fundamental of heat exchanger design”, WILEY.

[19]General Electric. Gate Cycle Software [Web Page]. Accessed 2010 Nov12. Available at: http://www.gepower.com/prod_serv/products/oc/en/opt_diagsw/gatecycle.htm.

[20]CMI ENERGY, Specialized in boilers design and manufacturing. Accessed 2010 Nov12. Available at http://www.cmigroupe.com/

[21]Kay, W. M. & London, A. L. (1998) “London Compact heat exchanger”, 3rd Edition.

[22]Frank, P. Incropera & David, P. De witt, (1996), “Fundamental of Heat and Mass Transfer”, 3rd Edition.

[23]Holman, J.P. (1997), “Heat Transfer”, 8th Ed. McGraw-Hill, New York.

[24]Kreith, F. (1973).”Principle of heat transfer, 3rd Ed. Harber & Row Publishers, New York.

[25]Jakob, M. (1938). “Heat transfer and flow resistance in cross flow of gases over tube banks. Transactions ASME, 60,384.

[26] Frank, P. Incropera & David, P. De witt, (1996), “Fundamental of Heat and Mass Transfer”, 3rd Edition.

[27]Grimison, E. D. (1937). “Correlation and utilization of new data on flow resistance and heat transfer for cross flow of gases over tube banks. Transactions ASME, 59, 583-594.

