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DESIGN PRINCIPLES AND SIZING APPROACH OF UNFIRED ONCE-THROUGH STEAM GENERATORS

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ABSTRACT

One of the key elements of increasing the thermal efficiency of a combined cycle power plant (CCPP) is to improve the design and operation of the heat recovery steam generators (HRSG) utilized in the cycle. Once-through steam generator (OTSG) is a new technology introduced for heat recovery in power systems. It eliminates boiler drums and other components of conventional HRSGs. The simplicity and compactness of an OTSG justifies its application in combined cycle power plants. This paper describes a design principle and an analytical sizing approach that will assist OTSG's designers to achieve a good design by determining the core dimension, volume of an OTSG for given flow rates and their entering and leaving temperatures as well as the heat transfer area on the smoke side. The developed model has been tested with reference to a scaled model of an existing OTSG that is installed at Manx Electricity Authority and the results were promising. The overall characteristics of heat transfer and pressure drop distributions of the OTSG "scaled model" shows general agreement with the real characteristics of the existing OTSG with error values less than 1%.

Keywords: HRSG, OTSG, design, sizing, CCPP

INTRODUCTION

In the last two decade, combined cycle power plants (CCPP) become a good choice to produce energy among various power plants because of their high efficiency and low environment impact. Most combined-cycle power plants burn natural gas, therefore, they produce low exhaust emissions. Their high efficiency results in low air emissions per MWh of electrical power produced and a low amount of waste heat. CCPP couples a topping gas-turbine (Brayton) cycle with a bottoming steam-turbine (Rankine) cycle via a heat recovery steam generator (HRSG) therefore, it offers a higher thermal efficiency than either of the cycles executed individually. The basic idea is to utilize the hot exhaust gas energy available from the Brayton cycle to produce super heated steam at high pressure and temperature to drive a steam-turbine for additional electric power generation. A heat recovery steam generator (HRSG) plays a unique role in coupling the topping and bottoming cycle.

The new once-through heat recovery steam generators (OTSGs) have been developed to improve efficiency and costs of conventional HRSGs. OTSGs are well applicable for combined cycle power plant with steam turbine throttle pressure from 120 bars up to supercritical pressure. This helps realize high efficiency cycles. In an OTSG, water flows in a continuous tube heat exchanger where preheating, evaporation, and superheating of the feed-water take place consecutively. Two levels of steam cycle operation pressure in an OTSG are possible.

This type of steam generator provides the following advantages compared to a conventional HRSG:

- No low efficient thick-wall, thermally sluggish high-pressure water/steam drum exists
- Simple in configuration
- Rapid start-up
- Unlimited daily cycling is acceptable

The use of once through steam generator (OTSG) for combined cycle gas turbine (CCGT) power plant applications is relatively new compared to conventional HRSG and published research on OTSG is limited. Some typical conventional HRSG designs including supplementary and auxiliary firing systems were described by Duffy and Schneider [2]. Practical design of a conventional HRSG is usually based on the concept of pinch and approach points that govern the gas and steam temperature profile [3],[4],[5]. A general guideline for the optimum design of conventional heat recovery steam generators was presented by Alessandro and Nicola [6]. However, the detailed methodology for an optimum heat exchanger design is much more complicated, not only because of the arithmetic involved, but also of many qualitative judgments to be made [7]. An advanced modelling and design of OTSG was given by Marie-Noëlle and Georges [1]. In this paper, a design principle and sizing method for OTSGs are introduced and tested against a mock design of an existing OTSG. Some assessment of the principle and method is carried out.

NOMENCLATURE

A1 Transverse plane [-]
A2 Diagonal plane [-]
A_f Fin area [m^2]
A_g Tube smoke side area [m^2]
A_w Tube wetted area [m^2]
A_o Free flow area [m^2]
A_{fr} Frontal area [m^2]
A_{wall} Tube wall area [m^2]
C_{min} Minimum fluid heat capacity
C_{max} Maximum fluid heat capacity
C* Heat capacity ratio [C_{min}/C_{max}]
Cp Exhaust specific heat [kJ/kg.˚C]
d_i Tube inner diameter [m]
$d_{orif,in}$ Orifice inner diameter[m]
d_o Tube outer diameter [m]
d_e Fin outer diameter[m]
Δp Fluid pressure drop [Pascal]
f Friction factor [-]
g_c Proportionality factor in Newton's second law [-]
G Specific mass flow rate [kg/m.s]
h Heat transfer coefficient [W/m^2.˚C]
k Thermal conductivity [W/m.˚C]
l Fin length form root[m]
L1 Hot flow length[m]
L2 Cold flow length [m]
L3 No flow length[m]
m Fin effectiveness parameter [-]
m_g Exhaust gas mass flow [kg/s]
m_w Header water mass flow [kg/s]
$m_{w,st}$ Tube water mass flow [kg/s]
N_r Tube row number [-]
N_{tu} Number transfer unit [-]
N_t Tube number in one row [-]
r_h Hydraulic radius [m]
ac Cold frontal area [m^2]
bc Hot frontal area [m^2]
P_L Longitudinal ration ($P_L=S_L/d_o$)
Pr Prandtl number [-]
P_T Transverse Ratio ($P_T=S_T/d_o$)
P_w Water Pressure [Bar]
P_s Steam Pressure [Bar]
Re_{do} Reynolds number [-]
St Stanton number ($h/G{*}c_p$)
S_D diagonal tube pith[m]
S_L Longitudinal tube pitch[m]
S_T Transverse tube pitch[m]
T_w Water temperature [˚C]
T_s Steam temperature [˚C]
V_g Exhaust gas velocity [m/s]
V_w Water velocity [m/s]

Greek Symbols

ρ Density [kg/m^3]
δ Fin thickness[m]
η_f Fin temperature effectiveness[%]
η_o Surface effectiveness[%]
ε Effectiveness[%]
ε_p Effectiveness of one pass of a multi-pass OTSG[%]
σ Ratio of free-flow area to frontal area [%]
Β Ratio of total transfer area to total volume of OTSG [m^2/m^3]
υ_m Specific volume($1/\rho$)
θ° Tube layout angle[-]

Subscripts

c Cold-fluid side of OTSG
g Exhaust gas
h Hot-fluid side of OTSG
min Minimum
max Maximum
n Number of identical passes in overall counter flow arrangement
in Inlet
out Outlet
s Steam
st Single tube
w Water

OTSG AND STEAM CYCLE

A typical gas turbine combined cycle power plant (CCPP) is schematically shown in Figure 1.

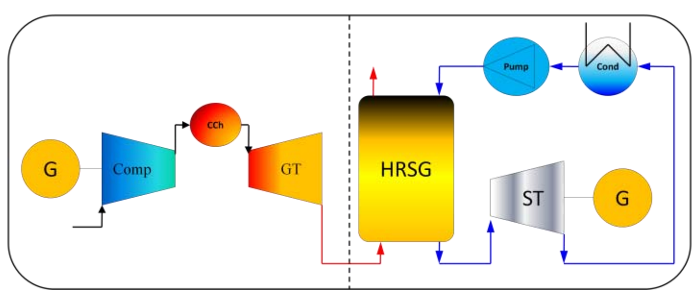

Figure 1 Schematics of CCGT Power Plant

Once-through Steam Generator (OTSG) is an alternative to the conventional Heat-Recovery Steam Generator (HRSG). Unlike a HRSG, an OTSG illustrated in Figure 2, does not have defined economizer, evaporator or super heater sections. Thus all feed water entering the OTSG convert to steam in a bundle of once through tubes.

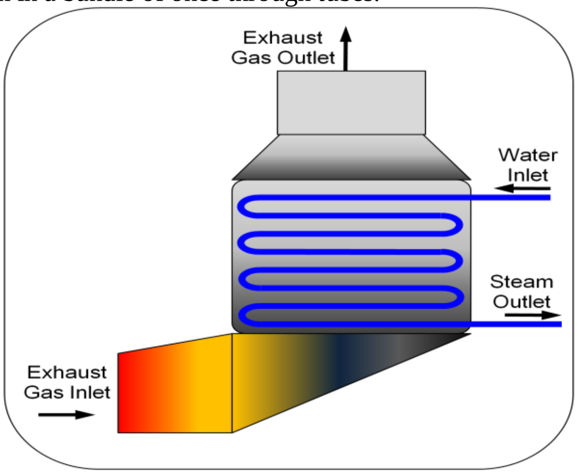

Figure 2 Schematic of OTSG

The point at which the steam-water interface exists inside the tubes is free to move upstream and downstream through the tube bundle depending on available heat from the exhaust gas at hot side of the tube. Therefore, design engineers should pay great attention to this phenomenon by adding an extra tube circuit (increase the multi-pass) as a safety margin to ensure that the saturated steam has reached to the superheated condition that the steam quality above unity or in another word the steam temperature above saturation . However, the steam production starts shortly after admission of feed water and hot exhaust gas into the OTSG. The generated super heated steam is then admitted into the steam-turbine to generate an extra useful power.

OTSG DESIGN CONSIDERATIONS

To schematically illustrate the design procedure, Figure 3 shows a heat exchanger design process based on the methodologies given by Marson [8], Taborek [9], and Shah [10]. The major design considerations for a new heat exchanger are including:

- Process specification
- Thermal and hydraulic design
- Mechanical design

The first and most important consideration is to select the design basis (i.e., design conditions). The design basis would require the specification of operating conditions that may include fluid type and rates, inlet temperatures and pressure of both fluid streams, required heat duty and maximum allowed pressure drops on both fluid sides. Based on the problem specifications and the design engineer's experience, the exchanger construction type and flow arrangement are first selected. Selection of the construction type depends on the fluids (gas, liquid, or condensing/evaporating) used on each side of a two fluid exchanger, operating pressures and temperatures, while the choice of a particular flow arrangement is dependent on the required exchanger effectiveness, exchanger construction type, upstream and downstream ducting, and footprint. The next step is the surface configuration and material selection. The qualitative criteria for surface and material selection are the operating temperature, pressure, design engineer's experience and judgment, the consideration of corrosion, erosion, and many other criteria, which will not be discussed in this study. The first dashed block discussed in more details in the next section as it is the focus of this study.

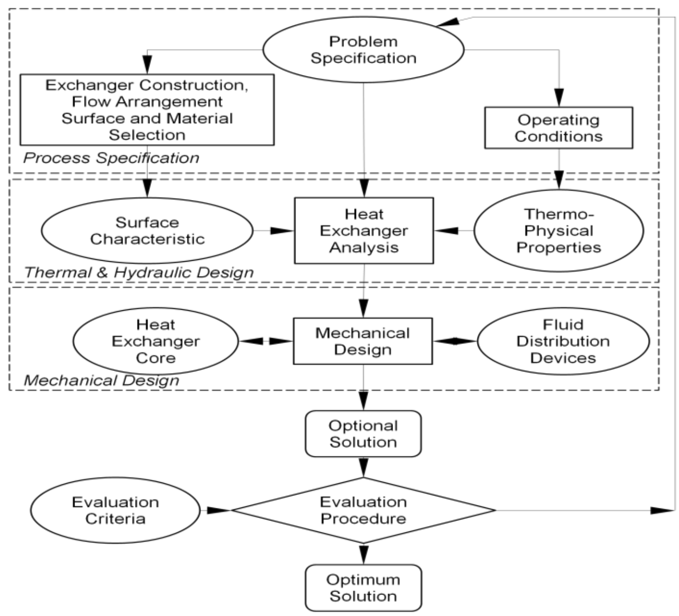

Figure 3 Methodology of Heat Exchanger Design [7] [11]

OTSG PROCESS SPECIFICATIONS

The process specification of a heat exchanger includes operating conditions, heat exchanger type, flow arrangement, materials, surface configuration and material selection. It is the first step of OTSG design process and the OTSG sizing problem is part of the process. The sizing of an OTSG mainly deals with the sizing of the core of the OTSG, given the flow rates of exhaust gas at hot side and water/steam at cold side and their entering and leaving temperatures.

OPERATING CONDITION AND TUBE TYPE

The OTSG in operation must withstand the stresses produced by operating pressure and temperature difference between fluids at hot and cold sides. Tubes boiler may be designed for water/steam operating pressure inside the tubes up to about 300Bar with exhaust gas outside the tubes being at low pressure about 1 bar. They also should be designed to cover the operating temperature from ambient temperature up to about 1143.15°F. In gas-to-liquid application the heat transfer coefficient at gas side may be much lower than the liquid side. For example, the heat transfer coefficient on the gas side may be around 0.1 to 0.001 of that on the liquid side. Therefore, for a thermally balanced design (i.e. having the product of heat transfer coefficient and area of the same order of magnitude on each fluid side of the exchanger) or ultrahigh heat exchange effectiveness is desired, fins may be employed to increase the smoke-side surface area to compensate for low heat transfer coefficients [11]. For combined cycle applications, the tube-fin heat exchanger may be selected for OTSG with the tubes being round shape as shown in Figure 4.

Figure 4 Typical Finned Tubes in OTSG

OTSG CONSTRUCTION & FLOW ARRANGEMENT

To achieve high OTSG heat transfer effectiveness, counter flow arrangement with unmixed fluid stream is preferred. A fluid stream is considered unmixed when it passes through individual flow tubes with no fluid mixing between adjacent flow tube as illustrated in Figure 5.

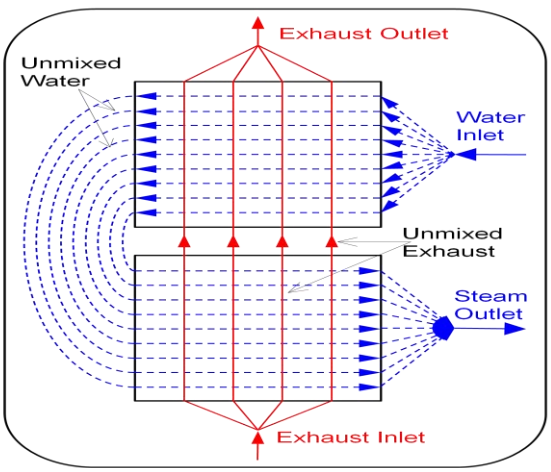

Figure 5 Multi-pass& unmixed counter flow

The OTSG effectiveness generally decreases with increasing mixing on any fluid side even if counter flow arrangement has selected [11]. One of the major advantages of proper multi-passing arrangement is to increase the exchanger overall effectiveness over the individual pass effectiveness, but with increased pressure drop on the multi-pass side [11]. If the overall direction of the two fluids is chosen as counter flow, the exchanger overall effectiveness approaches that for pure counter flow as the number of passes increases as illustrated in Figure 6 (dashed blue arrow is counter flow with the sold red arrow).

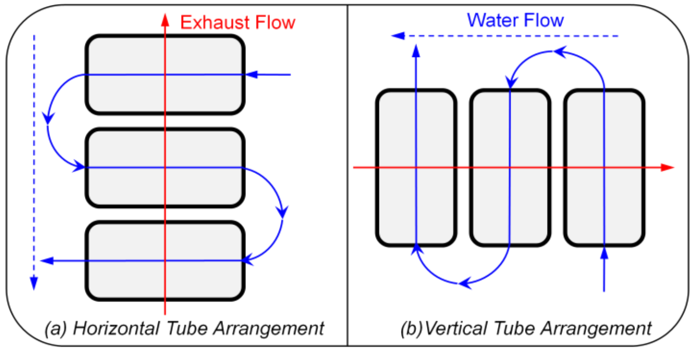

Figure 6 Multi-Pass Tube Arrangement

Two types of multi pass tube arrangements are normally used in an OTSG and they are illustrated in Figure 6.The majority of the OTSG units installed accommodate exhaust gas that flows vertically upward and the water flow enters at the top and flows downward through the serpentine tube bundle to exit at the bottom as superheated steam as shown in Figure 2 and Figure 6 (a).

OTSG TUBE-FIN PITCH AND LAYOUT

The selection of tube pitch is a compromise between a close pitch (small values of $P_T=S_T/d_o$) for increasing shell-side (finned-side) heat transfer and surface compactness, and an open pitch (large values of $P_L=S_L/d_o$) for decreasing shell-side plugging and easing shell-side cleaning (see Figure 7). For most OTSG heat exchangers, it is recommended that the ratio of the tube pitch to tube outer diameter may vary between 2 and 3.

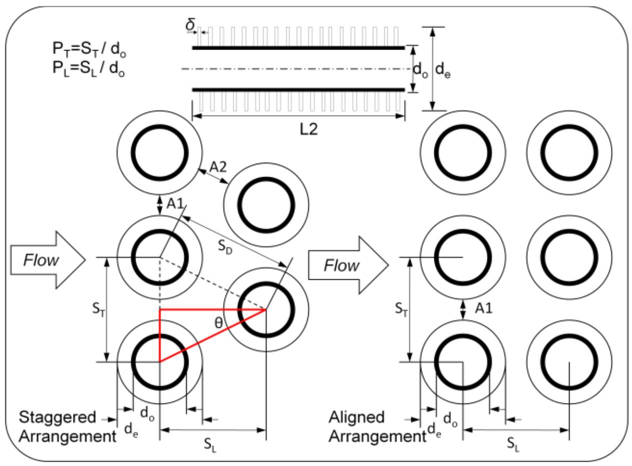

Figure 7 Tube Arrangement in OTSG Tube Banks

Two standard types of tube layouts are shown in Figure 8[11]; they called "square" and the "equilateral triangle". The square pitch is used when mechanical cleaning is necessary on the shell side, while the triangular pitch provides a more compact arrangement, usually resulting in the strongest tube bank. Hence, it is preferred when the operating pressure difference between the two fluids large.

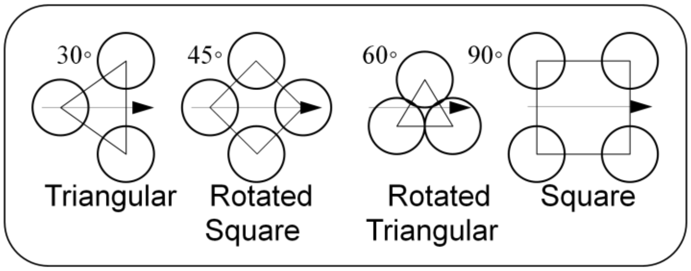

Figure 8 Tube layout Angle [11]

Note that the 30˚, 45˚, 60˚, arrangements are staggered and 90◦ is inline. The tube layouts in increase order of shell-side heat transfer and pressure drop are: 30˚, 45˚, 60˚, and 90˚. Thus the 90˚ layout will have the lowest heat transfer coefficient and the lowest pressure drop while 30˚ triangular tube layout arrangement is preferable in the OTSG for better heat transfer effectiveness and compactness [11].

OTSG TUBE-FIN MATERIAL

Material selection and compatibility between construction material and working fluids are important issues in OTSG design, in particular with regard to corrosion. Dry operation on the evaporation side of OTSG tubes with most current gas turbine allows the use of carbon steel fins, which are currently installed on many OTSGs for most effective heat transfer surfaces. Stainless steel fins are employed when the ambient temperature severe. The nickel stainless steel tubes permits the use of passive water treatment (wetlands) with the OTSGs. OTSG systems normally employ alloy 825 and stainless steel [12] fins in the inlet rows at all pressure level. These materials minimize the effects of corrosion and therefore allow the OTSG systems to accommodate low feed water temperature (as low as 15˚C) and thus low stack temperatures [12].

OTSG TUBE-FIN DIMENSION

From heat transfer point of view, small-diameter tubes (i.e. 12.75mm, 25.4mm) yield high heat transfer coefficients and result in a more compact HRSG [11]. However, it is strongly recommend not using such type of boiler with a compact heat transfer area to avoid the effect of the back pressure when the boiler installed behind gas-turbine generator. Therefore, back pressure various between15-30mbar is permitted. Usually, a utility scale OTSG for power generation has 60-400m^2/m^3 [11] to achieve reasonable heat transfer and is able to avoid high back pressure.

OTSG SIZING METHODOLOGY

➢ CORE MASS VELOCITY APPROACH:

Some assumptions are made as follows before the sizing method is introduced. It is assumed that the operating condition parameters (i.e. $T_{w,in}$, $T_{s,out}$, $P_{w,in}$, $P_{s,out}$, m_w, $T_{g,in}$, $T_{g,out}$, and m_g) are available and the OTSG type is gas to liquid among others such as liquid to liquid. The flow arrangement is counter flow and the finned-tube surface configuration is selected for smoke side. Moreover the tube pitch ratio is assumed to vary between 2 and 3 and tube layout angle to be 30^o. For high OTSG effectiveness (ε), it is assumed that each pass of fluid is unmixed as illustrated in Figure 5 and the mass flow rate in each pass is the same. Therefore, placing a calibrated orifice inside the tube is important to ensure equality of mass flow rate in each pass. Fluid passing through an orifice experiences a drop in pressure and acceleration. However as long as the fluid speed is sufficiently low (V_w<0.3 Mach number) the incompressible Bernoulli's equation describes the flow reasonably well.

➢ WATER VELOCITY INSIDE THE TUBE:

It is assumed that the water velocity inside the tube is in the range (0.9-2.4m/s) [11] to ensure good heat transfer process. The design water velocity is set at 1.9m/s inside each tube. Water velocity inside each tube should remain floating in that limited range when the OTSG running at off-design performance. To avoid exceeding this water velocity range, the number of tubes N_t connected to the feeding header is the only major variable available to the design engineer. However, water velocity can be calculated based on tube inner area, water mass flow rate (or orifice pressure drop) and water density from equation (1):

$$V_w = \frac{m_w}{\rho_w A_w} \tag{1}$$

Where A_w is tube inner area (wetted). The water velocity and pressure drop both are function of water mass flow rate across the calibrated orifice as it will be illustrated later on in this paper. The transverse tube number N_t connected to the feeding header is determined from equation (2):

$$N_T = \frac{m_w}{m_{w,st}} \tag{2}$$

Where $m_{w,st}$ is single tube water mass flow rate. The procedure for sizing the OTSG is an iterative process. To illustrate this procedure, a simple multi pass cross flow yielding counter flow arrangement will be considered. The problem then is to determine the dimensions of the overall OTSG (core dimensions L1, L2, and L3) and thus the volume. Figure 9 below shows the geometry for a multi-pass cross-counter flow OTSG.

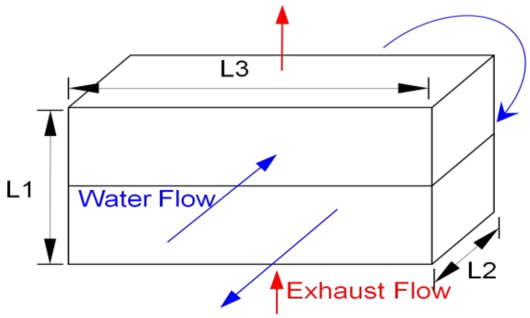

Figure 9 Core Dimensions of Multi-pass OTSG

Note that L1 is the flow length for the hot fluid, L2 is the flow length for the cold fluid, and L3 is a "no-flow" dimension. The frontal area for the hot fluid (i.e., the cross section facing fluid m_g) is (L2*L3). A very convenient variable is the exhaust gas mass velocity (G) for the hot side flowing normal to the tube bank into the OTSG. Note that (G) is the exhaust gas mass flow rate (m_g) divided by the cross-sectional free flow area (A_o) as illustrated in equation (3)

$$G = \frac{m_g}{A_o} \tag{3}$$

A first estimate of (*G*) can be made using the following approximate relation proposed by [7]:

$$G \approx \sqrt{\left(\frac{St\,\mathrm{Pr}^{2/3}}{f}\right)\left(\frac{\Delta P}{N_{tu}}\right)\left(\frac{g_c}{\upsilon_m\,\mathrm{Pr}^{2/3}}\right)} \tag{4}$$

Where the Stanton number ratio ($St\mathrm{Pr}^{2/3}/f$) tends to be reasonably constant for most surfaces and does not vary greatly from surface to surface [7]. A figure of 0.3 suggested by [7] provides a good initial estimate for an iterative procedure (see Figure 10).

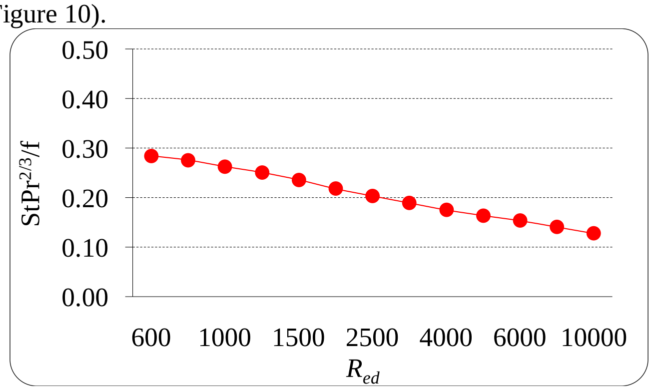

Figure 10 Stanton Number Ratio as a Function of Reynolds Number

The pressure drop ΔP in the second approximation of equation (4) is the desired pressure drop for one side (usually, exhaust gas side due to the impact of exhaust back pressure on Gas Turbine Performance). Dechamps[16]recommends adjusting for a pressure drop in the range 1.5 kPa to 3.0 kPa". N_{tu} is number transfer unit, a non-dimensional expression of the heat transfer size of the OTSG calculated with Equation (5) [11]. N_{tu} is determined for heat exchanger desired effectiveness ε and heat capacity ratio ($C^*=C_{min}/C_{max}$) derived from the given flow rates. The exhaust gas thermal-capacity rate is considered as maximum, while the water thermal-capacity rate is a minimum. Figure 11 represents the relationship between the OTSG effectiveness ε and number transfer unit N_{tu} through the heat capacity ratio C^*.

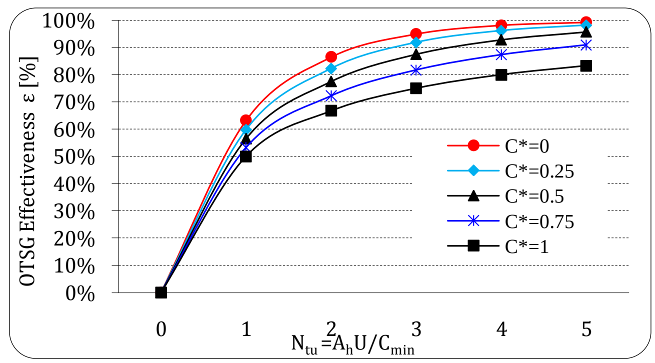

Figure 11 Map of Multi-pass Counter Flow OTSG

$$N_{tu} = \frac{1}{1-C^*}\ln\frac{1-C^*\varepsilon}{1-\varepsilon} \tag{5}$$

In an OTSG boiler, the multi-pass tube is one of the important features for gas-liquid, or gas-phase change application to achieve ultrahigh boiler effectiveness in the range of 80-90% [7]. As the number of passes increase, the effectiveness increases logarithmically as shown in Figure 12.

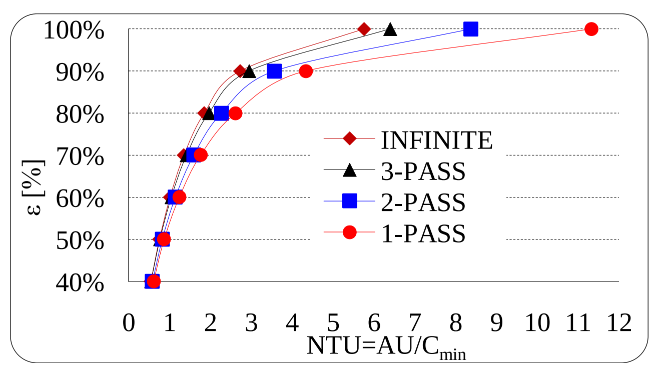

Figure 12 Variation of Tube Effectiveness with Number Transfer Unit at Different Pass Number

The results of multi-pass overall-counter flow effectiveness have obtained from the developed model with the aid of equation (6) [7]:

$$\varepsilon = \frac{\left[\left(1-\varepsilon_p C^*\right)/\left(1-\varepsilon_p\right)\right]^{n-1}}{\left[\left(1-\varepsilon_p C^*\right)/\left(1-\varepsilon_p\right)\right]^{n-C^*}} \quad (6)$$

Where n is the number of identical passes in overall-counter flow arrangement and ε_p is the effectiveness of each pass, a function of N_{tu}/n. for the special cases of C*=1, the effectiveness can be obtained from equation (7)

$$\varepsilon = \frac{n\varepsilon_P}{1+(n-1)\varepsilon_P} \quad (7)$$

The dimensionless proportionality factor g_c in the third approximation of equation (4) can be approximated as unity. The specific volume υ_m and Prandtl number Pr are the exhaust gas properties.

➢ EXHAUST GAS FRONTAL AREA (A_{fr}) OR (L2*L3):

The free flow area A_o is defined as the space between tubes where the exhaust gas passes through between the tubes banks. It is related to the frontal area (A_{fr} = L2*L3) through the ratio of free flow area to frontal area σ that typically has a value between 0.44 and 0.59 [7]. Thus when G is specified, the hot side frontal area is fixed and is given with the aid of equation (3) by:

$$A_{fr} = A_o / \sigma \quad (8)$$

➢ EXHAUST GAS FLOW LENGTH (L1):

To estimate the exhaust gas flow length, a convenient variable is the tube row number for a desired pressure drop between the tube banks of the OTSG. Such pressure drop can be estimated with aid of Equation (9) recommended by [17] and [18] which is suitable for flow of gases over a bank of tubes taking the following form:

$$\Delta p_g = \frac{2 f N_r G^2}{\rho_g} \quad (9)$$

For staggered configurations, the empirical friction factor *f* is given by [19] as:

$$f = \left(0.25 + \frac{0.118}{\left[\left(S_T - d_o\right)/d_o\right]^{1.08}}\right)\left(\mathrm{Re}_{do,\max}\right)^{-0.16} \quad (10)$$

For aligned arrangements:

$$f = \left(0.04 + \frac{0.08 S_L / d_o}{\left[\left(S_T - d_o\right)/d_o\right]^{0.43+1.13d/S_L}}\right)\left(\mathrm{Re}_{do,\max}\right)^{-0.15} \quad (11)$$

The Rynolds number $\mathrm{Re}_{d,\max}$ is based on the maximum exhaust gas velocity V_{max} occurring within the tube bank and is given by[20]:

$$\mathrm{Re}_{do,\max} = \frac{\rho d_o V_{\max}}{\mu} \quad (12)$$

The tubes row number N_r should be iterated to obtain acceptable pressure drops. Finally, the exhaust gas flow length L1 can be evaluated by the following equation:

$$L1 = N_r * S_L \quad (13)$$

The longitudinal tube pitches S_L that typically has values between 53mm and 73mm [7].

➢ EXHAUST GAS MAXIMUM VELOCITY (V_{max}):

For flows normal to in-line tube banks the maximum exhaust gas velocity presented in equation (12) will occur at the transverse plane A1 of Figure 7. Based on mass continuity for an incompressible fluid, $V_{g,max}$ can be calculated by [20]

$$V_{g,\max} = \frac{S_T}{S_T - d_o} V_g \quad (14)$$

For a 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

$$2\left(S_D - d_o\right) < \left(S_T - d_o\right)$$

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

$$V_{g,\max} = \frac{S_T}{2(S_D - d_o)} V_g \quad (15)$$

Where V_g is exhaust gas entry velocity. The pressure drop results taken from equation (9) are then compared with those specified for the design in Equation (4). The new estimates of G are made according to the following approximation (see equation (4) and (9)):

$$G = \sqrt{\Delta P}$$

➢ OTSG CORE VOLUME (V_{OTSG}):

The last step in sizing is to determine the OTSG core volume for a desired pressure drop of the exhaust gas into the tube bank. The size of the OTSG can be evaluated by:

$$V_{OTSG} = A_{fr} * L1 \quad (16)$$

Where L1 is the exhaust flow length determined from Equation (13) and (A_{fr} = L2*L3) is exhaust gas frontal area obtained from Equation (8). Furthermore, with the aid of the following equation (17), the designer can calculate the no-flow length L3 for a given transverse pitch S_T and calculated tube number for one row N_t can be calculated from equation (17):

$$L3 = S_T * N_t \quad (17)$$

Where N_t it is the tube number in single row. For a calculated mass velocity G, free flow area A_o and frontal area A_{fr}, the cold flow length L2 can be calculated from:

$$L2 = \frac{A_{fr}}{L3} \quad (18)$$

➢ SIZING PROCEDURE FLOW CHART:

The sizing procedure for a typical OTSG is summarized in Figure 13 and explained as follows:

1. Choose the ratio of Stanton number such as 0.3 [7] as a starting point.
2. Choose Number Transfer Unit for a given heat capacity ratio and desired effectiveness from Figure 11. It is recommended in [7] that values of 0.9 be used for the capacity ratio and the effectiveness.
3. Specify a desired pressure drop over the tube bank of the OTSG. It is typically in the range 1.5 kPa to 3.0 kPa to minimize the effect of back pressure on the gas turbine performance.
4. Calculate first estimated value of specific mass flow (G).
5. Calculate the free flow area (A_o) and frontal area (A_{fr}).
6. Calculate ($V_{g,max}$) and ($Re_{d,max}$).
7. Calculate pressure drop (ΔP).
8. Compare calculated and specified pressure drop
9. Make decision
10. Guess tube row numbers
11. Make decision
12. Calculated exhaust gas flow length (L1).
13. Calculate OTSG volume (V_{OTSG}).

All these steps involve OTSG sizing procedure are presented in the following flow chart.

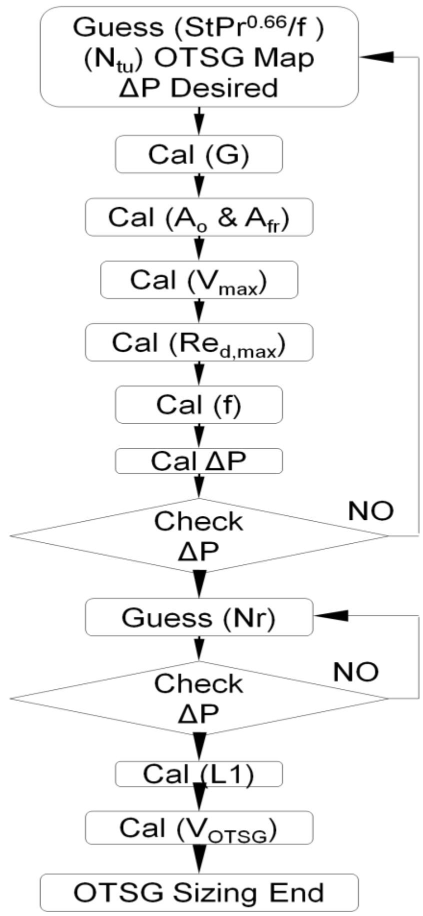

Figure 13 Sizing Procedure Flow Chart

CASE STUDY, RESULTS AND DISCUSSIONS

The following case study is based on the operating conditions of the OTSG operating at the Manx Electricity Authority (MEA). Figure 14 represent the boiler configurations and operating conditions of MEA.

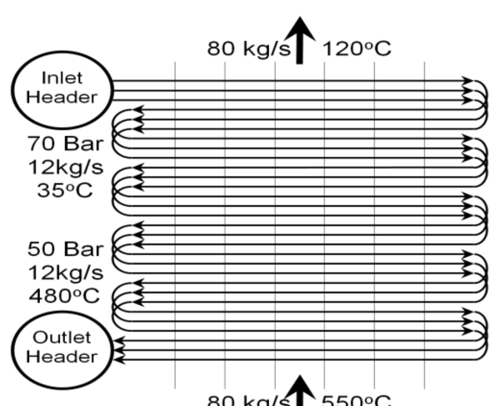

Figure 14 OTSG Configuration and Operating Conditions

The following results are very satisfactory between the real characteristics of OTSG installed at MEA and the standard characteristics for particular finned tube steam generator found in [7] and [16].

Table 1 OTSG sizing Model Deviations

Items	*deviation*
V_{OTSG}	*0.01%*
Hot Flow [L1]	*0.012%*
Cold Flow [L2]	*0.011%*
No-Flow [L3]	*0.01%*
Tube Layout Angle [θ]	*0.015%*
Transverse Pitch [S_T]	*0.0%*
Longitudinal Pitch [S_L]	*0.0%*
Diagonal Pitch[S_D]	*0.0%*
Transverse Ratio[P_T]	*0.0%*
Longitudinal Ratio [P_L]	*0.0%*

The proposed OTSG sizing approach has been applied to an existing utility scale once through steam generator installed at Manx Electricity Authority. The information of the OTSG and the test data provided by MEA were used to validate the developed model with the purpose of evaluating the accuracy of OTSG sizing model. In the existing reference OTSG, the gas turbine exhaust gas is ducted horizontally through the casing of the OTSG and then passes cross the tube bank vertically to a stack, while the water flow inside the multi-pass tubes row horizontally (see Figure 2and Figure 14). However, based on the results obtained from the model and MEA test data the following figure has illustrated the model validation in terms of pressure drop and heat transfer coefficient.

The MEA simulated data have shown general agreement with the standard boiler design characteristics presented in [7] and [16]. The total core dimensions of OTSG is a function of the hot side frontal area (A_{fr} = L2*L3), free flow area A_o and exhaust flow length L1 or row number N_r and tube longitudinal pitch S_L. They are illustrated in the following Figure 16, Figure 17 and Figure 18.

Figure 18 has also illustrating the variation of the core pressure drop with the row number. This figure is based on the results obtained from equation (9). The reverse effect of the free flow area on the core pressure drop is illustrated in Figure 19. Increases in free flow area (A_o) will decrease the mass velocity (G) and thus the pressure drops (see Figure 22).

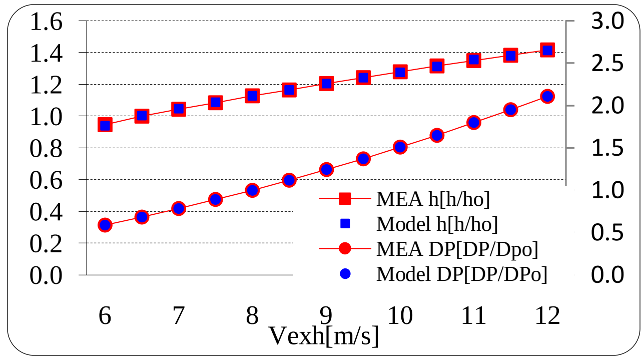

Figure 15 Variation of Heat Transfer and Pressure Drop with Exhaust Velocity

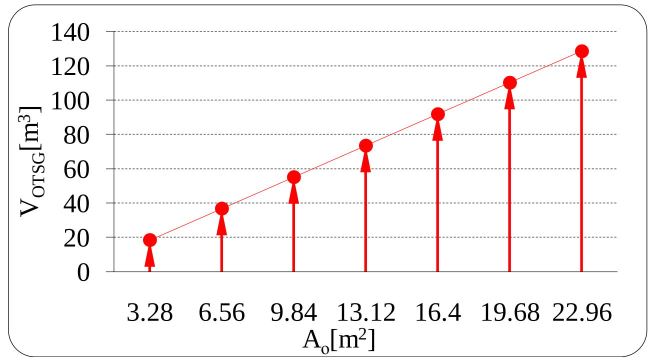

Figure 16 OTSG Volume as function of Free Flow Area

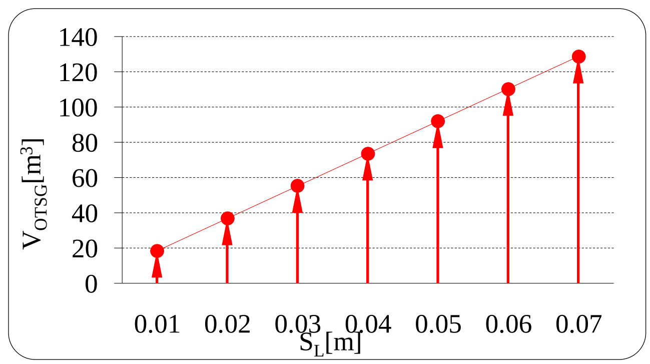

Figure 17 Variation of OTSG Volume with Longitudinal Pitch

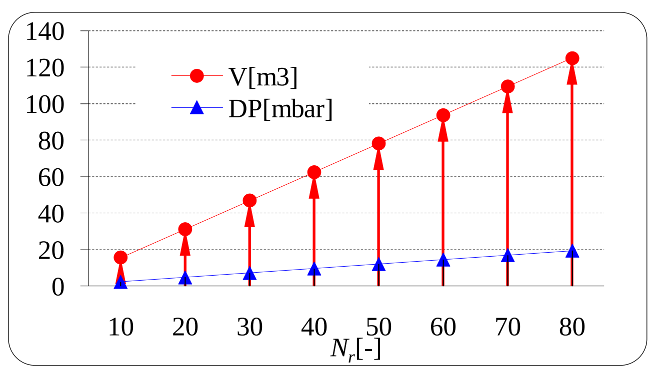

Figure 18 OTSG Volume as Function of Tube Row Number

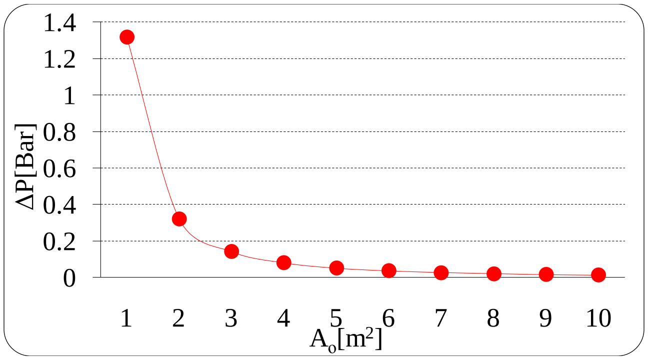

Figure 19 Variation of Pressure Drop with Minimum Flow Area

Furthermore, as the free flow area increases, exhaust gas maximum velocity decrease as illustrated in Figure 20.

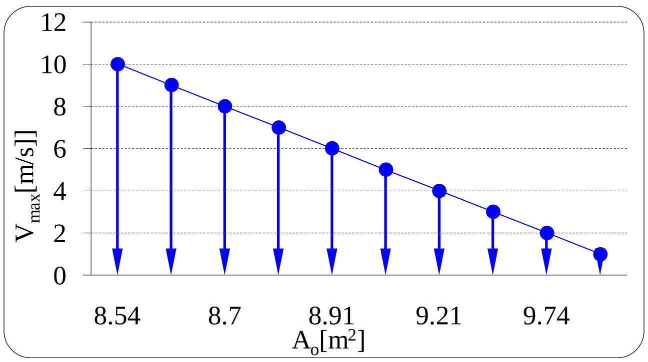

Figure 20 Variation of Maximum Exhaust Velocity with Free Flow Area

For different free flow area A_o, increasing exhaust mass flow rate will increase pressure drop as illustrated in Figure 21.

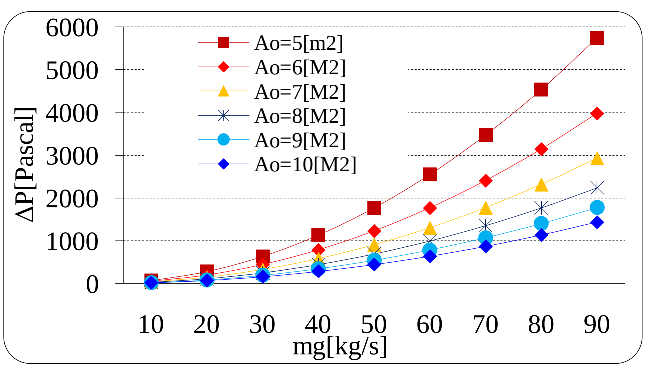

Figure 21 Variation of Pressure Drop with Exhaust Mass Flow Rate

For a given free flow are A_o, increasing the specific mass flow rate mass G will increase the core pressure drop as illustrated in Figure 22.

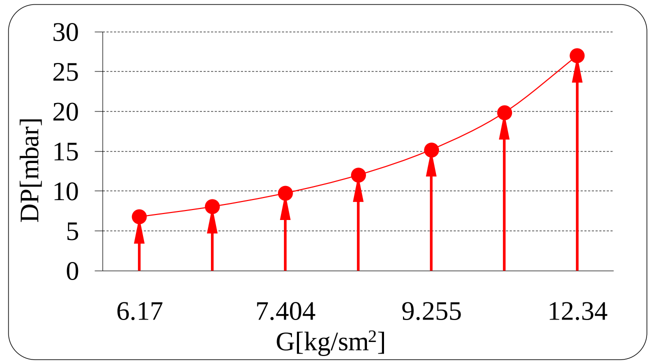

Figure 22 Pressure Drop Cross Tube Bank as a Function of Mass Velocity

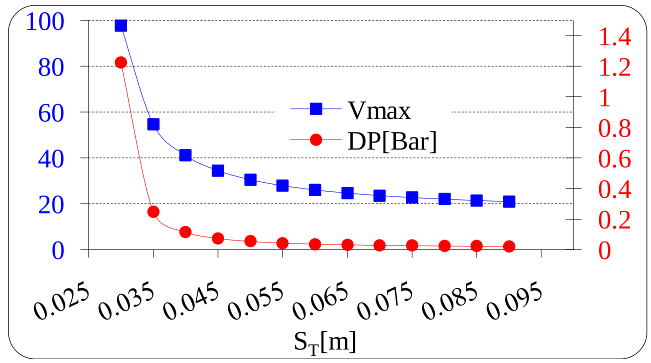

Figure 23 Variation of Pressure Drop and Maximum Exhaust Velocity with Tube Transverse Pitch

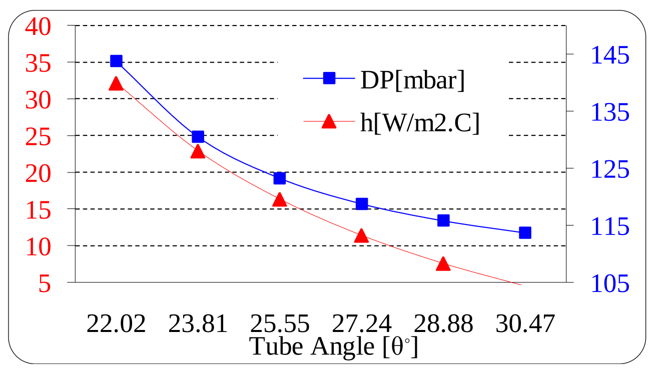

Figure 24 Variation of Pressure Drop and Heat Transfer Coefficient with Tube Layout Angle

Transverse tube pitch has also significant effect on the OTSG performance. As S_T increase, the core pressure drop, exhaust maximum velocity and thus heat transverse coefficient decrease as illustrated in Figure 23 and Figure 24. The tube layout angle θ° Figure 7 has significant effect on the heat transfer load and the pressure drop over the tube bank. As the tube angle increases, the heat load and pressure drop will decrease as result of wide transverse S_T and narrow longitudinal S_L pitches between the tubes. This behavior is clearly illustrated in Figure 24. Furthermore, this tube layout angle θ° has almost a linear relationship with the transverse pitch S_T as illustrated in Figure 25.

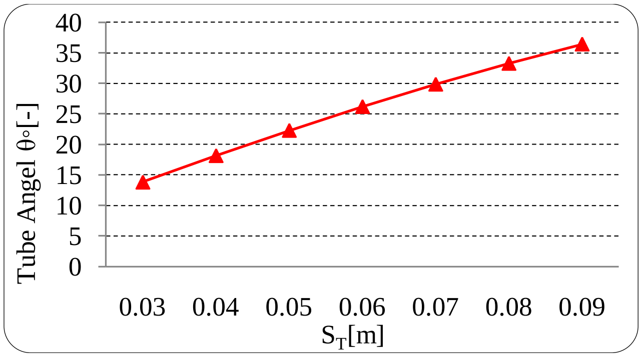

Figure 25 Variation of Tube Layout Angle with Transverse Pitch

CONCLUSION

A methodology for sizing of the utility scale once through steam generator for combined cycle gas turbine application has been outlined. The methods proposed in the present paper appear to be particularly suitable for an unfired OTSG for power and industrial use with multi-pass counter flow tube arrangement. The importance of the unfired OTSG sizing model lies in the need to achieve an efficient heat recovery system. Optimum OTSG core dimensions are obtained iteratively to maximize the heat recovery process for a steam cycle and minimize the impact of back pressure on gas-turbine cycle performances.

ACKNOWLEDGMENTS

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