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PERFORMANCE MODELING OF UNFIRED STEAM CYCLE USING SINGLE & DUAL PRESSURE ONCE-THROUGH STEAM GENERATOR

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ABSTRACT

The high thermal efficiency and the use of low carbon content fuel (e.g., natural gas) have made the Combined Cycle Power Plant (CCPP) one of the best choices for power generation due to its benefits associate with low cost and low environmental impact. The performance of Unfired Steam Cycle (USC) as a part of the CCPP has significant impact on the performance of the whole power plant as it provides the CCPP with around one third of the total useful power. An accurate performance simulation of the USC is therefore necessary to analyze the effects of various operating parameters on the performance of combined cycle power plant. In this paper, a performance simulation approach for an unfired steam cycle using single and dual pressure-level of an OTSG is presented. The developed modeling method has been applied to the performance simulation of an existing unfired steam cycle power generation unit installed at Manx Electricity Authority and the results are promising. A comparison between simulated and actual performance at design and off design operating conditions of the same USC has shown a remarkable agreement with errors values below 1%.

Keywords: CCPP, UCS, HRSG, Steam cycle, performance, OTSG

INTRODUCTION

In recent years, the rising cost of energy and global warming have highlighted the need to develop an advanced energy system with increased efficiency and reduced CO_2 emissions. Therefore, the world wide acceptance of steam and gas turbine combined cycle for electrical power generation is a result of the outstanding thermal efficiency and environmental compliance, as well as the low installed cost, high reliability, and good operating flexibility that has been demonstrated by operating experience of CCPP. The use of steam cycle driven by a gas cycle evolved from the demand of high overall thermal efficiency of a gas turbine power plant by utilizing the heat available in the exhaust gas. In a CCPP, two thirds of the gas turbine exhaust heat is captured by the heat recovery steam generator (HRSG) with the remaining third being lost in the exit stack. The steam cycle contributes approximately 30% of the CCPP total useful power output by steam turbine generators.

A schematic diagram of a typical gas turbine combined cycle system is shown in Figure 1 which consisting of a simple cycle of gas turbine driving a generator, a HRSG, a steam turbine driving a generator, a condenser and the associated auxiliary system.

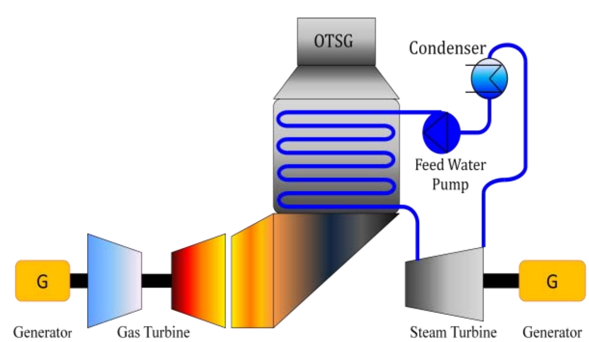

Figure 1 Schematics Of Single-pressure CCPP

In the past two decades remarkable improvements have been seen in combined cycle power plants to claim a net efficiency of up to 60%, due to the following changes:

1. Improvements in gas turbine technology
2. Improvements in heat recovery steam generator technology (dual pressure, reheat).
3. Trimming the operating parameters of the unfired steam cycle power plant.

Current advanced gas turbine combined cycle plants utilize two or three pressure levels, reheat unfired steam cycle and have exhaust gas temperature more than 500℃ [1],[2],[3].This paper presents a methodology that can be used to evaluate the performance of unfired steam cycle power plants using OTSG. The USC performance simulation program was created from the integration of new performance simulation codes of the main components of steam cycle. These include OTSG, steam turbine, pump and condenser. Steamomatch, is another performance simulation code developed at Cranfield University for steam cycle utilizing conventional HRSG [4].There are also some performance simulation codes available in wide ranges of commercial and academic for conventional steam cycle and combined cycle power plant that utilizes conventional HRSG [5],[6],[7]. The attention of this work focused on the operating parameters of the USC and its behavior at design and off-design performance. To simplify the calculations of the USC performance, the gas turbine operating conditions assumed to be constant. This allows the performance engineer to analysis the impact of the steam cycle operating parameters on the USC and thus CCPP performances. The simulated performance is then compared with the measured performance for an existing USC power plant installed and operated at Manx Electricity Authority in Isle of Man.

NOMENCLATURE

d_O	Tube Outer Diameter[m]
DSP	down Stream Pressure [Bar]
DP	Orifice Pressure Drop [Bar]
DP_O	Orifice Reference Pressure Drop [Bar]
F	Tube Arrangement Factor [-]
h	Enthalpy [kJ/kg]
k	Thermal Conductivity [W/m ˚C]
LSP	Live-Steam Pressure [Bar]
LST	Live-Steam Temperature [˚C]
m	Mass Flow Rate [kg/s]
m_O	Reference Mass Flow Rate[kg/s]
Nu	Nusselt Number [-]
Re	Rynoulds Number [-]
Pr	Prandtle Number
P	Pressure [Bar]
P_O	Reference Pressure [Bar]
P_T	Transverse Pitch Ratio [-]
P_L	Longitudinal Pitch Ratio [-]
Q_{pre}	Preheated Heat Transfer Rate [W, kJ/s]
Q_{ev}	Evaporation Heat Transfer Rate [W, kJ/s]
Q_{su}	Superheated Heat Transfer Rate [W, kJ/s]
SSC	Specific Steam Consumption [kg/kW.h]
T_{sat}	Saturation Temperature [˚C]
T_{stk}	Stack Temperature [˚C]
T	Temperature [˚C]
T_O	Reference Temperature [˚C]
U	Overall Heat Transfer Coefficient [W/m^2. ˚C]
Ū	Average Heat Transfer coefficient [W/m^2. ˚C]
W	Power Output [MW]
X	Steam Quality [-]

Greek Symbols

φ	Constant Factor for Steam Turbine
ΔT_{lm}	Logarithm Mean Temperature difference [˚C]
Δh	Steam Enthalpy Drop [kJ/kg]
η	Efficiency [%]

Subscripts

av	Available
cond	Condenser
d_O	Tube Outer Diameter[m]
ev	Evaporated
g	Exhaust Gas
max	Maximum
o	Reference
pre	Preheated
req	Required
sat	Saturated
sup	super heated
w	water

Abbreviations

CCPP	Combined Cycle Power Plant
HP	High Pressure
HRSG	Heat Recovery Steam Generator
LP	Low Pressure
OTSG	Once Through Steam Generator
SC	Steam Cycle
ST	Steam Turbine
USC	Unfired Steam Cycle

UNFIRED STEAM CYCLE MODELING

The unfired steam cycle or USC makes use of the available heat from the exhaust of gas turbine to raise superheated steam for driving the steam turbine and thus generate additional power from the same amount of fuel compared with simple gas turbine cycle and therefore the CO_2 emissions remain unchanged. Single and multiple pressures USC are available in the industrial market because of the remarkable improvements in industrial gas turbine and heat recovery steam generator [3].

SINGLE PRESSURE MODE

The simplest form of an unfired steam cycle consists of a single pressure level unfired OTSG, steam turbine, condenser, water pumps, etc. Schematic of such a unit is shown in Figure 1. Unlike a conventional drum-type HRSG that has an economizer, evaporator and a super heater [8]. The OTSG has continuous pipes where the point at which the water/steam interface exists is free to move through the horizontal tube bank depending on the heat input, mass flow rate and pressure of the water. The basic idea for modeling the USC with OTSG is presented in the T-Q diagram depicted in Figure 2. In this diagram, the red line with notations (1,2,3,4) represent the exhaust gas from gas turbine, while the blue area represents the four stages of the unfired steam cycle starting from water compression (a-b), water evaporation (b-c), steam superheating (c-d), steam expansion (d-e). The T-Q diagram illustrates the relationship between the exhaust gas temperature and the heat flow rate. This shows that when the heat energy transferred from the hot side to the cold water, the temperature of the exhaust gas drops simultaneously. Moreover, the approach point temperature is no longer exists in the heat transfer diagram since there is no drum required for OTSG.

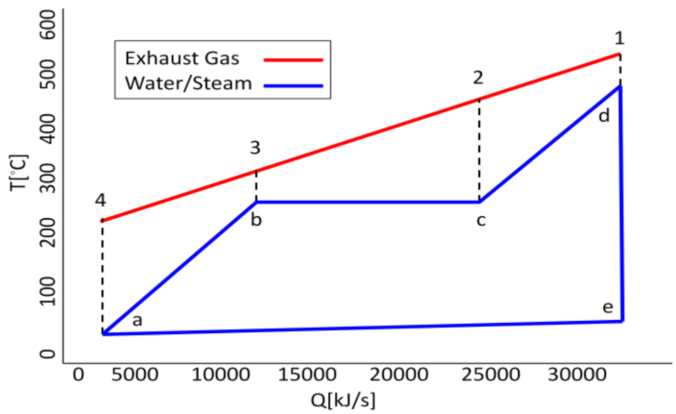

Figure 2 Heat Transfer Diagram and Notations for Single-Pressure CCPP

HEAT TRANSFER COEFFICIENT

The heat-transfer between exhaust gas and water/steam is mainly due to convection. The heat transfer coefficient is determined by many factors, such as surface geometry, fluid properties and flow arrangement. For gas to liquid OTSG, two main properties in the fluid that have significant impact on the heat transfer coefficient are density and conductivity. The average heat-transfer coefficient for entire tube bundle in OTSG can be evaluated from the following empirical correlation developed by Grimison [9]:

$$\bar{U}_g = \frac{Nu_{d_o} k}{d_o} \qquad (1)$$

Where k is a thermal conductivity of the exhausted gas, Nu_{do} is the Nusselt number given by Equation (2) [9], and d_o is the tube outer diameter.

$$Nu_{d_o} = 0.32 F\, \mathrm{Re}_{d_O,\max}{}^{0.61} \mathrm{Pr}^{0.31} \qquad (2)$$

Hausen [10] has modified the empirical correlation proposed by Grimison [9] and developed an empirical relationship for the tube arrangement factor (F) that presented in Equation (3). This factor refers to the tube arrangement in the HRSG and it is evaluated for two different tube layouts (aligned and staggered). For aligned tube arrangement,

$$Nu_{d_o} = 0.34 F\, \mathrm{Re}_{d_O,\max}{}^{0.61} \mathrm{Pr}^{0.31} \qquad (3)$$

Where the tube arrangement factor (F) calculated by the following equation:

$$F = 1 + \left(P_L + \frac{7.17}{P_L} - 6.52\right)\left(\frac{0.266}{(P_T - 0.8)^2} - 0.12\right)\left(\frac{1000}{\mathrm{Re}_{d_O,\max}}\right)^{0.5} \qquad (4)$$

For staggered tube arrangement

$$Nu_{d_o} = 0.35 F\, \mathrm{Re}_{d_O,\max}{}^{0.57} \mathrm{Pr}^{0.31} \qquad (5)$$

Where the tube arrangement factor calculated by the following equation:

$$F = 1 + 0.1 P_L + \frac{0.34}{P_T} \qquad (6)$$

Where P_T and P_L are tube pitch ratio, refer to the ratio of the transverse pitch (S_T) and longitudinal pitch (S_L) to tube outer diameter (d_O):

$$P_T = S_T / d_O \qquad (7)$$

$$P_L = S_L / d_O \qquad (8)$$

CYCLE ENERGY BALANCE

The amount of the heat energy available in the exhaust gas which is required to generate super heated steam is presented by the area denoted surrounded by (1, 2, c, d) in Figure 2 and can be calculated from:-

$$Q_{cd,req} = m_{\sup}(h_d - h_c) \qquad (9)$$

$$Q_{12,av} = m_g(h_1 - h_2) \qquad (10)$$

Similar equations can be applied to evaluate the heat balance between region (2-3) of the exhaust gas and (b-c) of the saturated water/steam in the evaporating stage .These are given below:

$$Q_{bc,req} = m_{ev}(h_c - h_b) \tag{11}$$

$$Q_{23,av} = m_g(h_2 - h_3) \tag{12}$$

Finally, the remaining heat in the exhaust gas will be recovered by the preheating stage before leaving through the stack of OTSG and the heat transfer is represented by Equations (13) and (14).

$$Q_{ab,req} = m_{pre}(h_b - h_a) \tag{13}$$

$$Q_{34,av} = m_g(h_3 - h_4) \tag{14}$$

The steam turbine inlet conditions are given by the orifice system of OTSG. The live-steam temperature is an important parameter for power output. This temperature can be determined by the Stodola-like choking relation [11] at the inlet of the steam turbine for a given inlet area:

$$\frac{m_{\sup}\sqrt{T_{\sup}}}{p_{\sup}} = \varphi * A_{ST} \tag{15}$$

Where m_{sup} is steam mass flow rate, T_{sup} and P_{sup} are the steam turbine inlet temperature and pressure respectively, A_{ST} is the turbine entry area, and φ is a constant factor for the steam turbine. Figure 3 represents the flow chart of the performance simulation process of unfired steam cycle operates with single pressure mode of OTSG. For most HRSG applications, a value of 45 W/m^{2}C is typical overall heat transfer coefficient for each section of the conventional HRSG [5]. Furthermore, coefficient values of 56.41, 62.45 and 48.15 were used by [8] for preheating, evaporating, and superheating in OTSG. It was assumed that the inlet conditions of the OTSG hot side such as exhaust gas mass flow and temperature are known and remain unchanged during the design and off-design of the USC performance simulation.

The first step of system equations execution for the developed model is starts from equation (15) where the guessed values of mass flow rate used as input, and the down steam pressure (Steam turbine inlet pressure P_{sup}) is a function of the mass flow rate for a given orifice sizes and upstream pressure as illustrated in Figure 4. Both parameters are used to solve for the superheated steam temperature (Steam turbine inlet temperature T_{sup}) at design and off-design performance. The next step is calculating the heat flow rate that captured by the superheated steam ($Q_{1,2,c,d}$) with the aid of equations (9) and (10). However, same calculations should carried out to determine the evaporating and preheating heat transfer load as well as the unknown variables based on the equations (11), (12), (13) and (14).

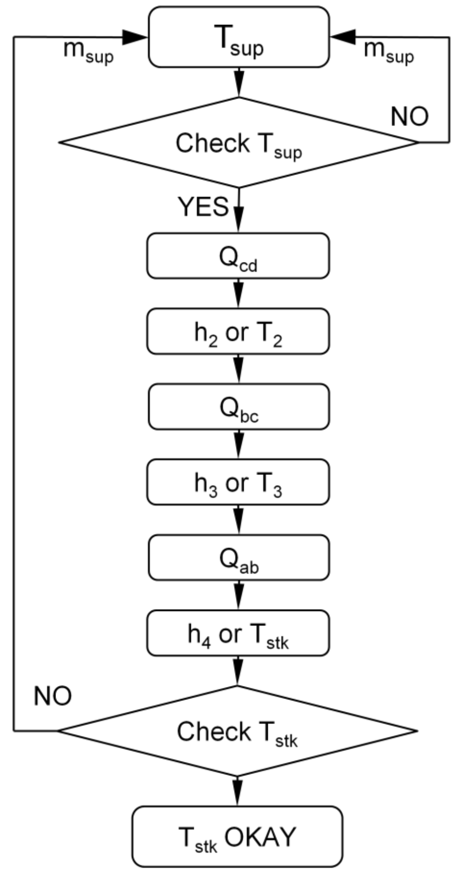

Figure 3 USC Single-Pressure Model

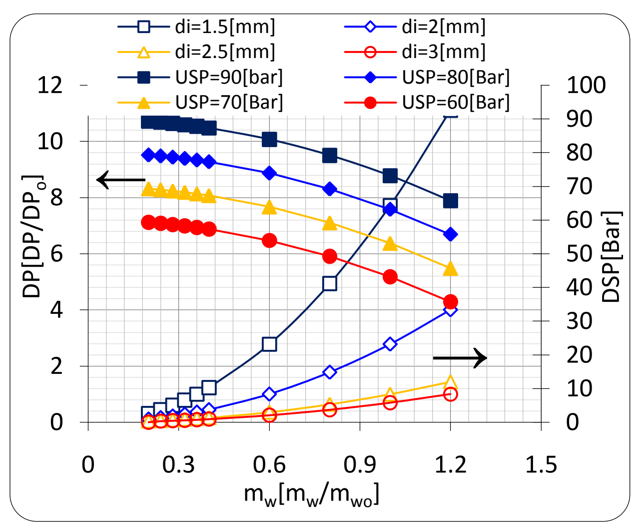

Figure 4 Pressure Drop and Down Stream Pressure Variation with Mass Flow Rate through an Orifice

DOUBLE PRESSURE MODE

The following T-Q diagram represent the unfired steam cycle which operates with dual pressure mode combined with the gas cycle in CCPP configuration. These two pressure circuits will determine separately since there is no connection between them due to disappearance of water/steam drum in OTSG technology. Low pressure level will recover the left amount of heat carried out by the exhaust gas after leaving the preheating section (a-b) of high pressure level.

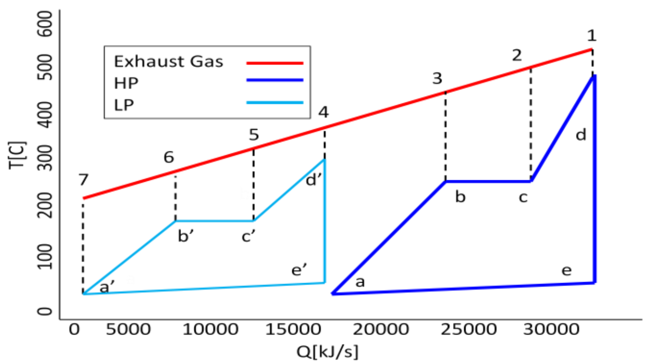

Figure 5 Heat Transfer Diagram and Notations for Dual-Pressure CCPP

Dual pressure OTSG offers higher steam turbine power output, increased boiler efficiency and thus better overall unfired steam cycle efficiency. In the case of natural gas combined cycle power plant, the feed water temperature can be further reduced. This enables more exhaust gas energy to be recovered by lowering the stack temperatures [12].

In dual pressure mode, the stack temperature or enthalpy of the exhaust stack was used as a control variable instead of the mass flow rate, while in HP level, the mass flow rate as well as the exhaust stack temperature used as control variables to achieve the required live-steam temperature that limited by the steam turbine material. However the same set of non-linear equations (9) to (15) should be executed to determine the heat balance between the LP and the uncovered energy of the exhaust gas. Figure 6 represents a flow chart of the performance simulation process of unfired steam cycle operates with dual pressure mode of the OTSG.

CASE STUDY

The following case study is based on the data of an existing unfired steam cycle part of combined cycle power plant installed at Manx Electricity Authority. The configuration of this USC and their operating conditions is given in Table 1. These data were also used to validate the developed model at design performance.

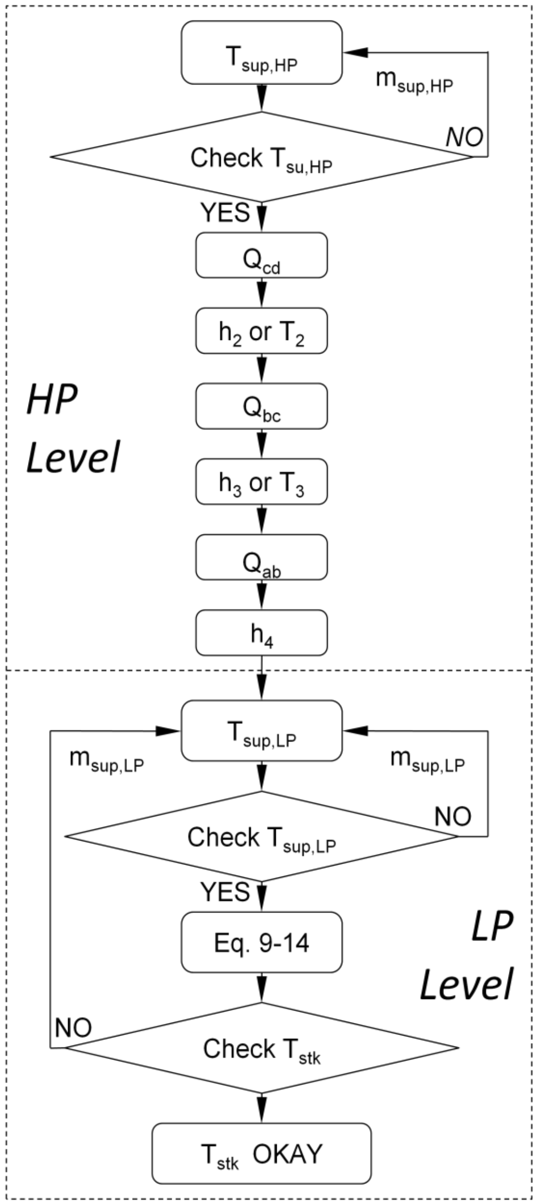

Figure 6 USC Dual-Pressure Mode

Table 1Unfired Steam Cycle Characteristics

Boiler	*OTSG Double Pressure*
Steam Turbine	*Induction Double Flow*
Exhaust Gas Inlet Temperature	*500°C*
Exhaust Gas Outlet Temperature	*120°C*
Exhaust Mass Flow Rate	*80kg/s*
(HP) Water Inlet Temperature	*35°C*
(HP) Live-steam Temperature	*480°C*
(HP) Water Mass Flow Rate	*9.6kg/s*
(HP)Live-steam Pressure	*50bar*
(LP)Water Inlet Temperature	*35°C*
(LP) Live-steam Temperature	*256°C*
(LP)Water Mass Flow Rate	*2.25kg/s*
(LP)Live-steam Pressure	*5bar*
Condenser Pressure	*0.06bar*

RESULTS AND DISCUSSIONS

For this particular steam cycle power plant, slight variation of the major operating parameters of the cycle may have significant impact on the USC and thus CCPP performance. These parameters are the live steam pressure (LSP), live steam temperature (LST) and the condenser pressure (P_{cond}). In the following sections, the impact of each of the parameters will be simulated and analyzed.

LIVE-STEAM PRESSURE (LSP)

The effects of the live-steam pressure on the USC performance are illustrated in Figure 8 to Figure 15. Two factors must be considered when selecting the HP and LP live-steam pressure for a dual-pressure unfired steam cycle. Firstly, The HP live-steam pressure must be relatively high for good exergetic utilization of the exhaust gas heat. Secondly, the LP live-steam pressure must be low to attain good energetic and exergetic utilization of the uncovered exhaust gas heat and therefore achieve a higher steam turbine output [12]. The USC developed model has successfully shown that the steam turbine is mainly driven by the HP live-steam which provides approximately 85% of the total power output and the remaining 15% contributed by the LP live-steam as illustrated in Figure 7.

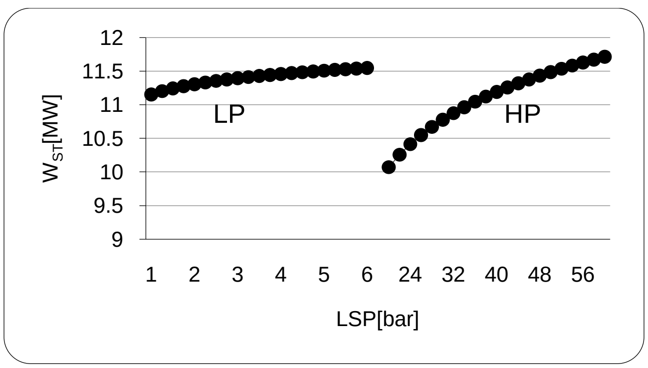

Figure 7 Steam Turbine Power Output Variation with LSP

When analyzing the performance of the USC and CCPP, the system efficiency should be investigated as a first criterion. It is clearly demonstrated in Figure 8 and Figure 9 that expanding the super heated steam at a higher live-steam pressure will make the steam cycle to achieve higher thermal efficiency and power output at different condenser pressures. This is due to a greater enthalpy drop in the steam turbine as depicted in Figure 10. The efficiency of the CCPP also increases due to this affection (see Figure 11). Furthermore, at higher evaporating pressure, less heat will be recovered in the OTSG resulting in higher stack temperature, lower OTSG efficiency and therefore less steam will be generated as illustrated in Figure 12 and Figure 13 respectively.

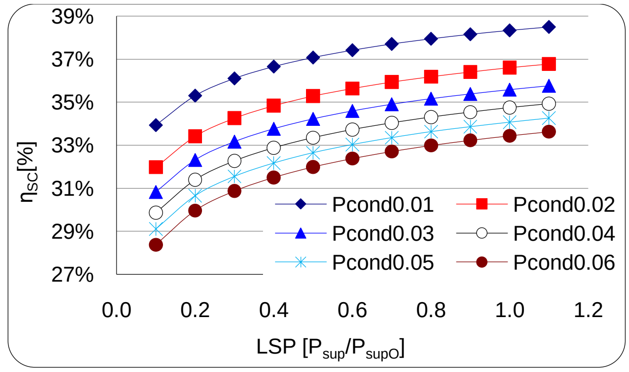

Figure 8 USC Efficiency Variation with LSP at Different P_{cond}

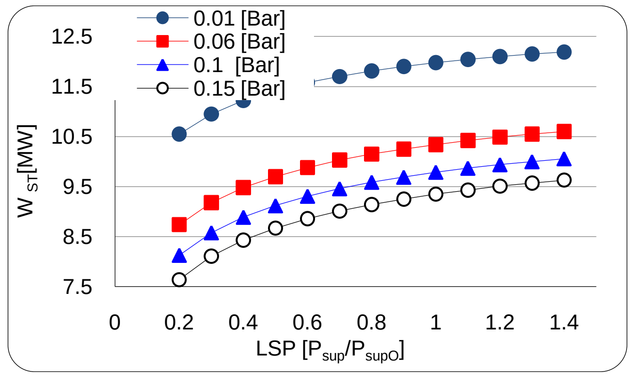

Figure 9 Steam Turbine Power Output Variation with LSP at Different P_{cond}

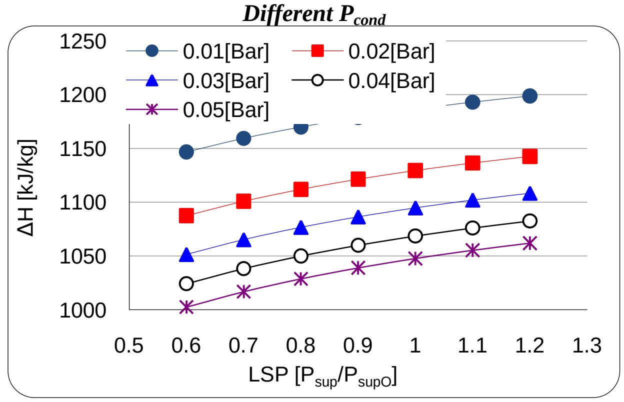

Figure 10 Enthalpy Drop as Variation of LSP at Different P_{cond}

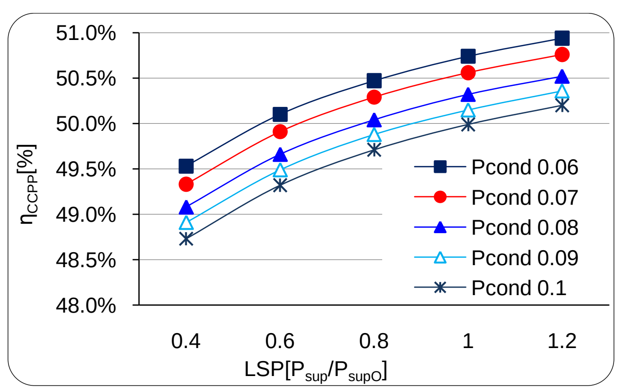

Figure 11 CCPP Efficiency Variation with LSP at Different P_{cond}

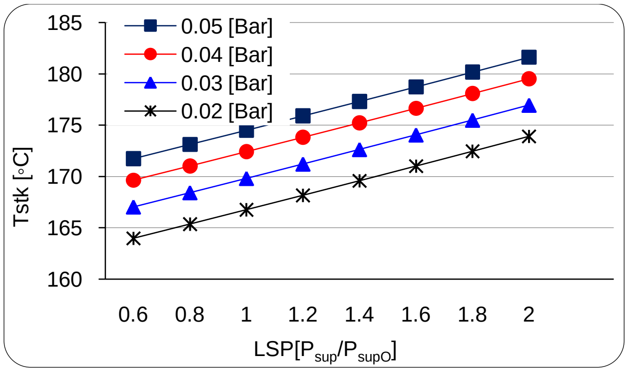

Figure 12 Variation of Stack Temperature with LSP at Different P_{cond}

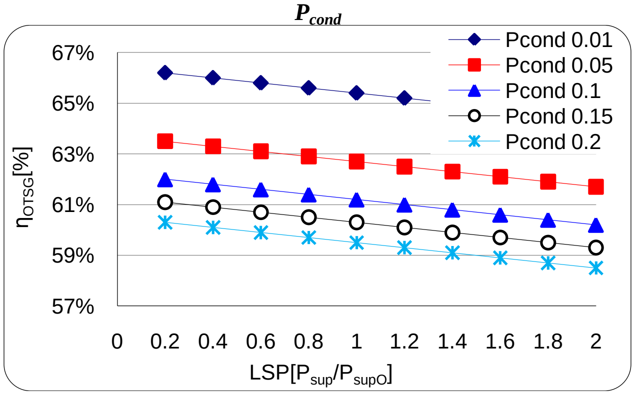

Figure 13 OTSG Efficiency Variation with LSP at Different P_{cond}

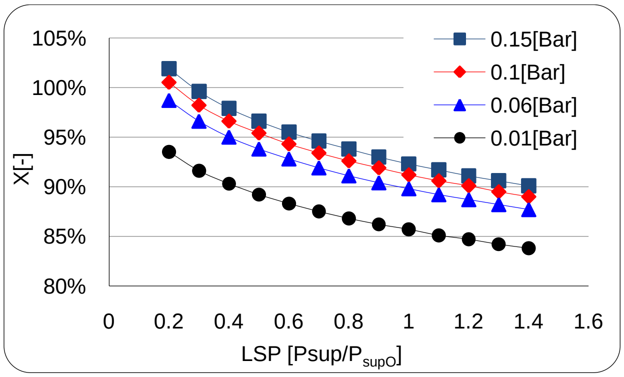

Figure 14 Steam Quality Variation with LSP and at Different P_{cond}

A negative aspect of a higher live-steam pressure in a single cycle is increasing the moisture content at the end of the steam turbine. Too much moisture increases the risk of erosion in the last stages of the steam turbine. A limit is set at approximately 12% for conventional steam cycle and 15% for nuclear power plants which corresponds to steam quality ranging from 85% (nuclear) and 88% (conventional) [13]. The moisture content is heavily dependent on the condenser pressure: the higher the pressure, the lower the moisture content. The opposite parameter to the moisture content is the steam quality which is defined as the friction of the total mass flow composed of a vapor or liquid. However, as demonstrated in Figure 14, increasing the live-steam pressure will decrease the steam quality due to a higher specific volume of saturated water and lower specific volume of the saturated vapor. The steam wetness in such turbines is widely known to be extremely undesirable as it leads to both a loss of efficiency and mechanical damage (erosion) of machine components (usually the turbine blades). Therefore, several compromised actions can be taken to improve the steam quality at the end of the expansion of the steam turbine:

- Decrease the cycle live-steam pressure, which however lowers cycle efficiency as illustrated in Figure 8.
- Increase the live-steam temperature that is normally limited by the material (the highest temperature is around 540°C).
- Increase the condenser pressure, which however lowers the cycle efficiency, and therefore the power output.

The most appropriate solution for this problem is to accommodate the single housing steam turbine with the LP circuit at low pressure to keep the steam quality at or above the risk limit. Finally, the live-steam pressure has a significant effect on the specific steam consumption and hence, the power output. As the LSP increases at different condenser pressure, the SSC decreases due to the greater enthalpy drop as depicted in Figure 15. This means that the steam turbine power output increases as LSP increases and consequently more power output can be obtained with less consumption of superheated steam. SSC it is an important parameter to look at when assessing the performance of the steam turbine and thus whole steam cycle.

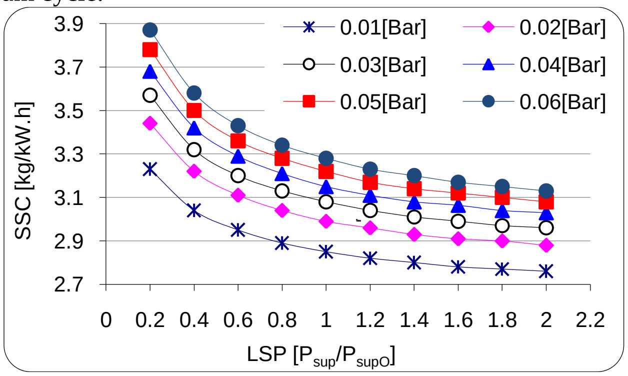

Figure 15 Variation of Specific Steam Consumption with LSP at Different P_{cond}

LIVE-STEAM TEMPERATURE (LST)

Live-steam temperature has a significant impact on the unfired steam cycle performance and therefore on the combined cycle power plant performance. Such impact is demonstrated with the simulated results of the USC model. Increasing LST provides a substantial improvement in the steam cycle efficiency, power output, and specific steam consumption. LST has an almost same effect as the LSP does on the USC performance. Figure 16 has shown similar behavior as Figure 7 but in terms of live-steam temperature.

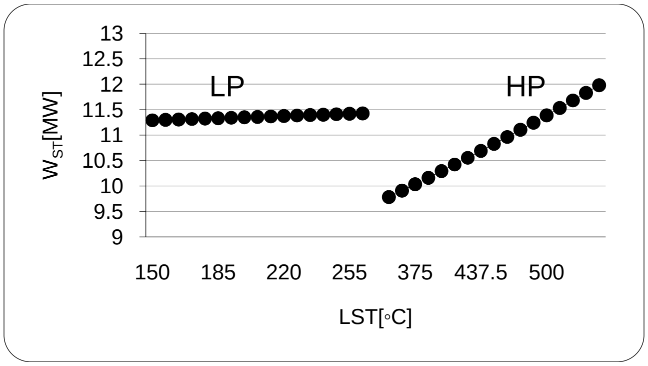

Figure 16 Turbine Power Output Variation with LST

The USC efficiency and steam turbine power output have an almost linear relationship with the live-steam temperature as demonstrated in Figure 17 and Figure 18 respectively. Such behavior is due to the greater enthalpy drop in the steam turbine.

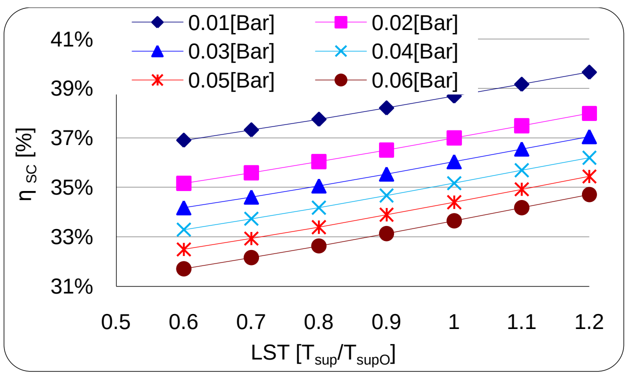

Figure 17 USC Efficiency Variation with LST at Different P_{cond}

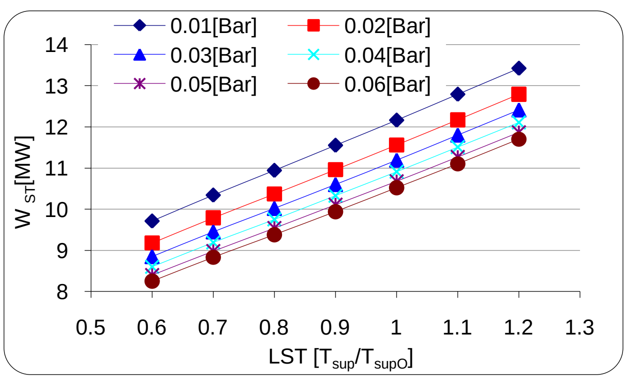

Figure 18 Turbine Power Output Variation with LST at Different P_{cond}

Figure 19 has demonstrated that the specific steam consumption varies inversely with the LST. Therefore increasing the LST will lower the SSC due to the greater enthalpy drop in the steam turbine which mean less amount of super heated steam needed to generate one mega watt at high LST and vice versa. Live-steam temperature has also an effect on the performance of the OTSG. Based on the assumption that has been made in the introduction (gas turbine operate at constant operating conditions), the variation of LST depends on the feeding water mass flow rate. As the mass flow rate decreases, the live-steam temperature increases and this will result in higher exhaust gas stack temperature and thus lower boiler efficiency due to less energy being utilized in the OTSG as illustrated in Figure 20.

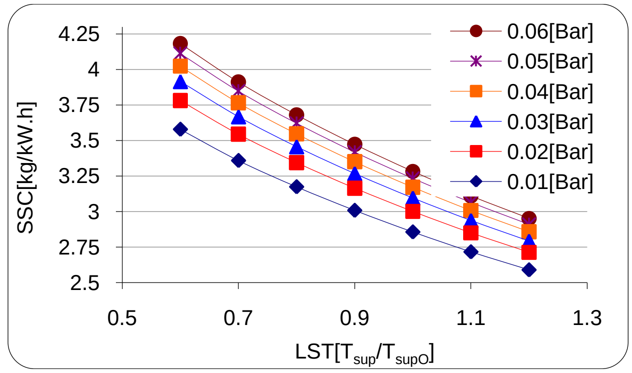

Figure 19 Specific Steam Consumption Variation with LST at Different P_{cond}

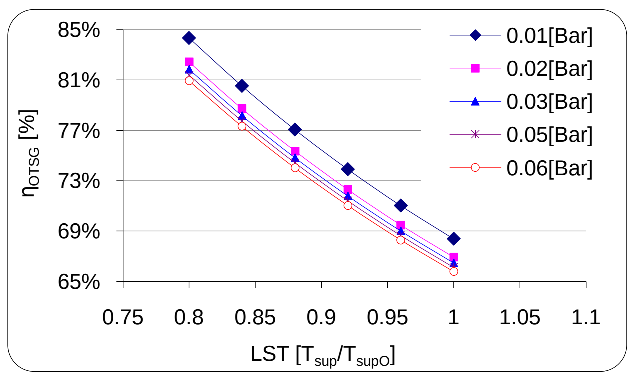

Figure 20 OTSG Efficiency Variation with of LST at Different P_{cond}

Finally, the combined cycle power plant has a high sensitivity performance to any variation in the LST. The CCPP efficiency varies directly with the super heated steam temperature. In Figure 21 it was clearly demonstrated that increasing the LST results in linear increasing of the thermal efficiency of CCPP due to the linear increase in USC efficiency as depicted in Figure 17.

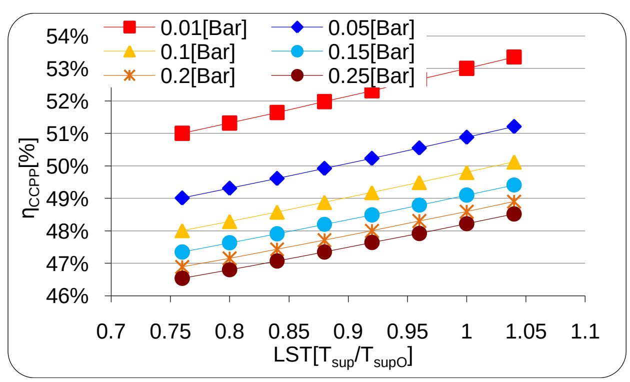

Figure 21 CCPP Efficiency Variation with LST at Different P_{cond}

CONDENSER PRESSURE (P_{COND})

The low condenser pressure allows the cycle to operate with high temperature and pressure drop between the source (OTSG) and the sink (condenser). This has significant effect on the performance of the USC and thus CCPP as well as the exhaust steam quality. It was illustrated in Figure 22 and Figure 23 that the increases in the condenser pressure (or decreases in vacuum), results in decreases enthalpy drop across the steam turbine. Therefore, this lowers the steam cycle efficiency and power output due to the resistance of the condenser pressure to the stream of the superheated steam mass flow through the steam turbine. Furthermore, increasing the condenser pressure increases the exhaust gas stack temperature, and lowers the steam turbine power output due to building up pressure against the steam flow through the steam turbine. Thus increasing the LSP, increases saturated pressure and hence the saturated temperature, resulting in less steam generated.

Figure 23 and Figure 24; illustrates the effect of the condenser pressure on the turbine power output, stack temperature and indirectly on the saturation temperature. The condenser pressure has also an effect on the specific steam consumption (SSC) as shown in Figure 25. The SSC increases as the condenser pressure increases due to lower power output per unit of superheated steam mass flow rate. However, at high condenser pressure, more superheated steam will be consumed to overcome the resistance that results from raising the condenser pressure and thus the whole cycle will operate at poor efficiency.

The cross point in Figure 23 and Figure 25 represents the optimum value of the condenser pressure that can achieve the best efficiency and power output with optimum stack temperature. Figure 26 shows the efficiency of the OTSG drops as the condenser pressure moves toward the ambient pressure (reduces vacuum). The stack temperature increases as a result of the drop in boiler efficiency. The CCPP efficiency presented in Figure 27 has adversely affected by the condenser pressure, therefore it decreases as the condenser pressure increases.

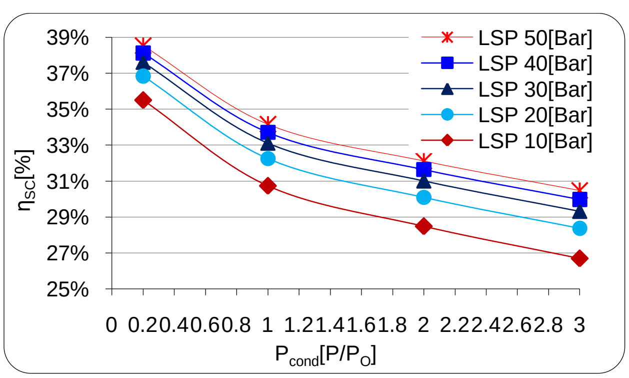

Figure 22 USC Efficiency Variation with P_{cond} at Different LSP

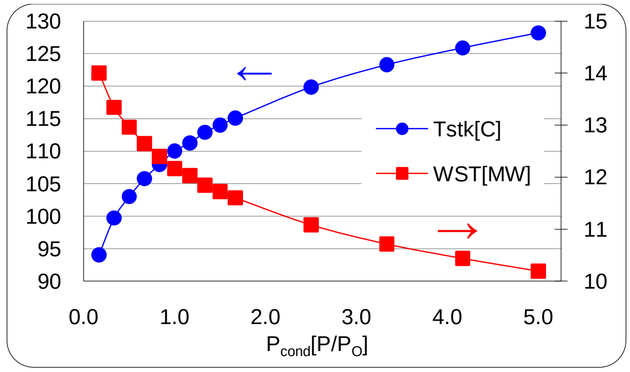

Figure 23 Stack Temperature & Turbine Output Variation of P_{cond}

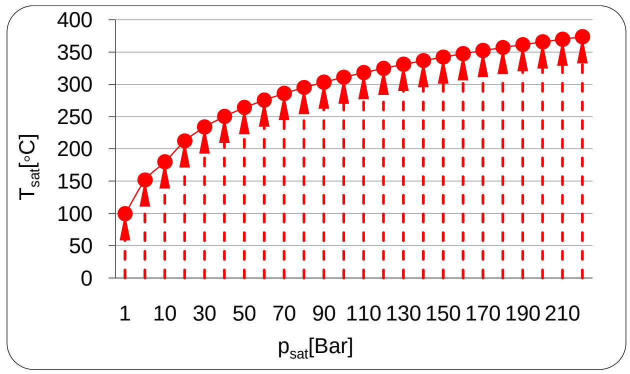

Figure 24 Saturated Temperature as a Function of P_{sat} (LSP)

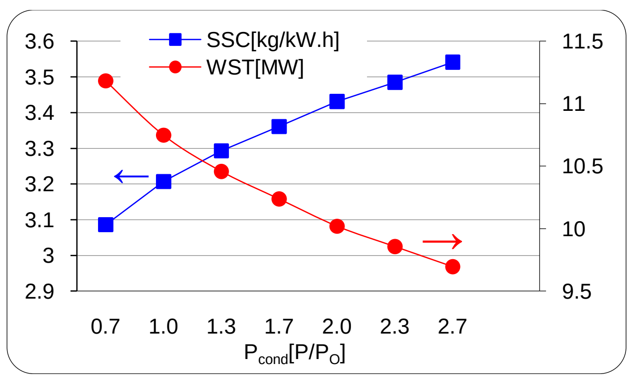

Figure 25 Specific Steam Consumption and turbine power output Variation with the P_{cond}

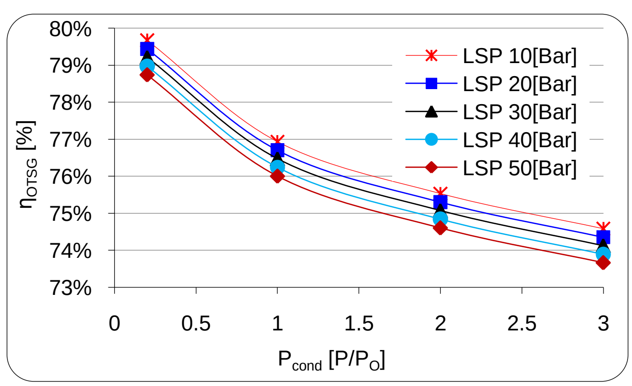

Figure 26 OTSG Efficiency Function of P_{cond} at different LSP

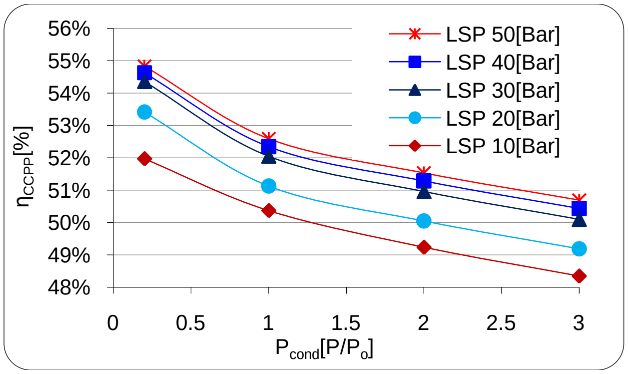

Figure 27 CCPP Efficiency Variation with P_{cond} at different LSP

Finally, the cycle condenser pressure has shown significant impact on steam quality of the steam turbine exhaust. Steam quality must be kept at limited level to avoid the erosion that might occur in the last stages of the steam turbine. Steam quality and condenser pressure are logarithmically related as illustrated in Figure 28. It is showing that when the USC is operated in a single pressure mode (HP only), will always risks erosion in the last stages of the Induction steam turbine due to the high moisture content (see the blue curve of Figure 28)To compensate for the poor steam quality, LP circuit (light blue curve of Figure 5 denoted by a', b', c', d', e') should start up after the HP circuit in order to keep the steam quality at/or above an accepted level (see the red curve in Figure 28). Furthermore, LP circuit must always be kept on as long as HP circuit is on.

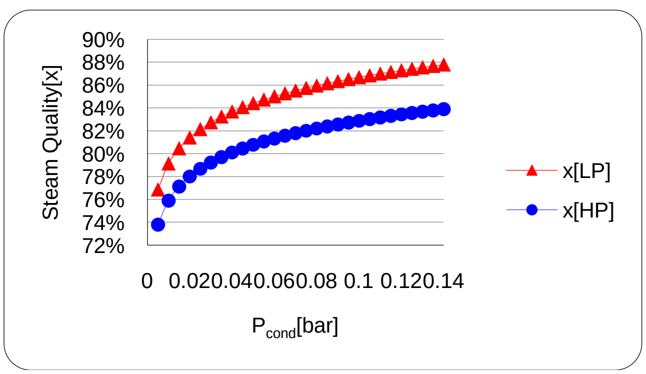

Figure 28 Variation of Exhaust Steam with P_{cond}

MODEL VALIDATION

A comparison between the results of the developed model and the observed performance of an existing combined cycle power plant has been carried out to validate the USC performance simulation model based on the following assumption: the operating conditions of the gas turbine remain unchanged over predicting the design and off design performance of the unfired steam cycle power plant. This will help the performance and design engineer to have a clear picture about the impact of the steam cycle operating parameters on the USC and thus CCPP as well as the OTSG.

The developed model has shown a general agreement compared with the realistic unfired steam cycle performance. Each calculated and measured parameters provided by MEA have been used to evaluate the accuracy of USC model. The obtained results from the developed model have made full satisfactory for the MEA power plant with neglected errors. Below is a comparison table between the results of MEA and USC model.

Table 2 Model Validation and Comparison Results

Variables	*USC Model*	*MEA*	*Error %*
η_{OTSG}	*68.8%*	*69%*	*-0.29%*
η_{SC}	*28.75%*	*29%*	*-0.9%*
η_{CCPP}	*50%*	*49.8%*	*0.4%*
W_{ST}	*21.12*	*21.1*	*0.9%*
T_{stk}	*120*	*120*	*0.0%*
m_w	*35229.4*	*35200*	*0.08%*
m_g	*77.4*	*81.75*	*-5.33%*

The only variable that shows disagreement is the gas turbine exhaust mass flow given maximum error -5.33% as shown in Table 2. This difference is due to the uncertain calculation that has been done by MEA. However, the exhaust mass flow rate (m_g) calculated by the USC model is based on the heat balance between the steam cycle and the heat load of the exhaust gas from the gas turbine. Finally, the use of an existing academic and commercial performance simulation tools is not adequate due to the lack of performance simulation capability of USC utilizing OTSG as a heat recovery boiler. These tools were appearing to be capable for performance simulation of the conventional steam cycle with drum-HRSG only. For this reason it is necessary to develop an advance and reliable model that is capable to achieve this requirement.

CONCLUSION

A novel performance simulation approach for steam cycle was presented. The model was specifically developed for an Unfired Steam Cycle using single and dual pressure-level of Once-Through Steam Generator (OTSG). The USC performance model was created in a user friendly and flexible environment into Microsoft Excel with excellent feedback from MEA. The developed model was applied to predict the performance of an existing steam cycle power plant installed at Manx Electricity Authority and the obtained results were promising. It has showed that the trimming of operating parameters is essential for efficient operation of the USC. The plotted figures that are presented in this paper have shown various effects of the steam cycle operating conditions on the performance of OTSG, USC and thus CCPP. However, huge benefits can be obtained by optimizing the USC operating parameters.

Special attention was paid to the result presented in Figure 28 which illustrates the logarithm behavior of the exhaust steam quality with the condenser pressure. It is showing that as the condenser pressure increases, the steam quality increases simultaneously. The key result concluded from Figure 28 is that the LP exhaust steam has higher quality than the HP due to the lower operating pressure as well as less steam turbine stages number. Figure 7 and Figure 16 are illustrated that the steam turbine is mainly driven by HP circuit. Therefore, the main purpose of implementing LP circuit in parallel with HP circuit in induction steam turbine is to keep the exhaust steam quality at/ or above the accepted level which in turn works as protection for the last stages from the risk of erosion. The conclusion made from the intersection points in Figure 23 and Figure 25 give the optimum value of the condenser pressure that can achieve the best cycle efficiency and power output with optimum stack temperature and specific steam consumption.

Finally, Figure 22, Figure 26 and Figure 27 represent remarkable result with minimal error of the USC, OTSG and CCPP thermal efficiency respectively. The thermal efficiency in these three figures is inversely proportional to the condenser pressure. Therefore, high thermal efficiency can be achieved at a lower condenser pressure due to the greater pressure ratio and vice verse

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