This is the accepted manuscript of the article that appeared in final form in **Energy** 148 : 676-686 (2018) , which has been published in final form at https://doi.org/10.1016/j.energy.2018.01.160 © 2018 Elsevier under CC BY-NC-ND license (http://creativecommons.org/licenses/by-nc-nd/4.0/)

Improvement of auxiliary BI-DRUM boiler operation by dynamic simulation

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Abstract

Due to Horizon 2030 objectives, industry is committed to improving traditional auxiliary bi-drum boilers performance in existing plants.

In this study, a new operation methodology for auxiliary bi-drum boilers has been developed to reduce the operational cost of the plant.

Two new functioning modes have been defined as “hot standby mode” (HSM) and “fast hot startup” (FHS). Both of them will be studied using thermal dynamic calculations. In addition, Power Plant Simulation & Design (PPSD) commercial software has been used for dynamic performance analysis.

Results show that auxiliary bi-drum boilers will be able to operate under those conditions only if some additional technological modifications are implemented. For example, automated burners that are adapted to current standards, feed water control valve capable of regulating the flow during HSM, automate superheater valve drains and a continuous blow down valve. This new operation methodology will result in energy savings and reduction in operational cost.

Keywords: Dynamic energy behaviour; Hot startup; Hot standby mode; Natural circulation; Bi-drum boiler

Nomenclature

C

empirical value that depends on fuel type (−)

$\dot{Q}_N$

Boiler heat output at 100% MCR (kW)

$\dot{Q}_{EN}$

Boiler heat losses (kW)

$\dot{Q}$

Heat transfer flow (kW)

ΔT

Temperature difference (ºC)

R

Total Thermal Resistance (K/W)

U

General heat transfer coefficient (W/m^2K)

A

area (m2)

$R_{conv\&radiation}$

Convection Thermal Resistance (K/W)

R_{wall}

Conduction thermal Resistance (K/W)

h

convection heat transfer coefficient (W/m^2K)

k

conduction heat transfer coefficient (W/mK)

x

insulation thickness (m)

A_s

surface area (m^2)

n

number of cycles (−)

$2f_{va}$

variable stress (N/mm^2)

f_v

mean stress (N/mm^2)

t*

reference temperature (ºC)

N

load cycles (−)

Ct*

Fatigue curve stress range temperature correction (−)

2fa*t*

Temperature corrected stress range (N/mm^2)

Ss

Fatigue curve statistical scatter corrector factor (−)

2fas

Final notch, mean stress/plasticity, temperature and statistical corrected stress range (N/mm^2)

Nas

minimum cycles to cracking at 2fas (−)

SL

statistical factor to adjust from mean to minimum cycles to cracking (−)

mass flow (t/h)

steam production at boiler battery limit

Pressure (barg)

steam pressure at boiler battery limit

$T_{H20,SH3}$ (ºC)

superheater third tube bundle steam temperature

$T_{metal,SH3}$ (ºC)

superheater third bundle tube metal temperature

$T_{H20,SH2}$ (ºC)

superheater second tube bundle steam temperature

$T_{metal,SH2}$ (ºC)

superheater second tube bundle metal temperature

$T_{H20,SH1}$ (ºC)

superheater first tube bundle steam temperature

$T_{metal,SH1}$ (ºC)

superheater first bundle tube metal temperature

Abbreviations

PPSD

Power Plant Simulation & Design

CCGT

Combined Cycle Gas Turbines

HRSG

Heat Recovery Steam Generator

MCR

Maximum Continuous Rating

WHB

Waste Heat Boiler

BL

Battery Limit

HSM

hot standby mode

FHS

fast hot startup

FG

Flue gas; (full stop is missing) SH: Superhetaer; (full stop is missing) ECO: Economizer

1 Introduction

Industry is interested in keeping auxiliary steam boilers operating at HSM and proceeded to a FHS (in around 10 min), as it enables a considerable energy and cost reduction as well as increase production efficiency.

The present work defines two new operational modes, namely: Hot standby mode (HSM), an operation mode where the boiler is operating at nominal pressure condition but with a steam production and fuel consumption almost non-existent. The other operational mode, fast hot startup (FHS), has the aim to increase from minimum load to maximum designed load in the minimum of time.

Industrial auxiliary steam boilers are essential equipment in conventional industrial plants, such as, thermal power plants, cogenerations, Combined Cycle Gas Turbines (CCGT), refineries, paper industries, as well as, renewable power plants. Considering their presence in different industries and the current economical situation it is crucial to carry out research and improve their design [1] to increase energy savings [2,3] with minimum capital investment.

Industrial boilers have to face new challenges because they have to provide a fast reaction to the steam demand in the plant and also reduce environmental impact by minimizing emissions. As an example, Spain has regulated the emissions in standard R.D. 815/2013 [4]. Thanks to dynamic calculation, designers are able to study non-stationary situations. Therefore, boiler startup load changes and shut down operation cases can be studied and improved. During the past few years, boiler dynamics have been studied. The basis of these studies considered the conservation of mass, energy and momentum. The first studies were focused on natural circulation fluctuations during load change. Adam [5] developed two nonlinear analytical models to study evaporation dynamics in vertical tubes and phase separation in steam drum, which is so-called natural circulation. The model was applied to a bi-drum water-tube boiler of 30 MW with natural circulation. Åström [6] developed a nonlinear analytical model describing drum, evaporator and downcomer loops dynamics. The proposed model was based on empirical formulations derived from measurement data which reduced the number of unknown parameters in dynamic study. This article was the starting point for several researches such as, Franke [7], Kim [8] and Emara-Shabaik [9]. Franke [7] implemented in

Modelica Åström [6] non-linear boiler drum model. Kim [8] developed a model for water level prediction when steam demand is changing in a natural circulation drum-type boiler. Obtained results were compared with Åström's [6] results. Emara-Shabaik [9] developed a dynamic model based on Åströms [6] to predict the temperature of the riser tubes in a water tube boiler. In previous literature, the temperature of the riser tubes was supposed to be equal to steam temperature.

Dynamic calculations have also been used for startup simulations. In CCGTs, for example, it is important for a fast startup to follow the startup curve for gas turbines. The most critical fact during a startup is the fast increase of temperature, pressure and steam flow. During this process, especially the steam drum and the superheater collector suffer thermal stresses as temperature is not properly distributed along the thickness of the material. Kim [10] developed a model to predict transient characteristics of HRSG. Startup behaviour was also analysed paying special attention to thermal stress on the steam drum. In this model, water-steam convection coefficient was calculated using Dittus-Boelter formulation and the temperature of the metal was considered to be equal to steam temperature. Krüger [11] studied drum-type boiler startup optimization based on a non-linear model. Thermal stress calculations were considered in the steam drum assuming that there was an unsteady flux. The optimization model was done using the commercial Dymola environment based on the open Modelica language [12].

During dynamic calculations, special attention is required to two other factors namely, thermal stress and the metal superheater tube temperature.

Regarding thermal stress, its determination in thick wall components is very complicated as boundary conditions must be correctly defined. Boiler designers always define some values to avoid thermal stress problems during startup. These values are defined from a conservative point of view for boiler lifetime. For example, the rate of fluid temperature variation in a steam drum must be constant and no more than 1 °C/min. This steam temperature variation will imply a uniform temperature distribution in thick walled components during startup. But when the startup time needs to be reduced and the constant temperature rate of 1 °C/min is not fulfilled thermal stress should be studied. The most common method to study thermal stress in thick walled components is EN-12952-3 European standard [13]. Taler developed a methodology to reduce startup time avoiding thermal stress in thick walled components [14]. He proposed to fill up the steam drum with hot water before startup. The results showed that startup time could be reduced from 6.5 h to 2 h using his method. These calculations were made based on the mathematical dynamic model developed in Ref. [15]. To overcome thermal stress in thick wall components, new techniques for measuring the transient temperature of fluid flowing in the pipeline have also been studied [16].

Regarding the temperature of the metal tube, when the temperature is higher than expected, tubes could break. Therefore, it is very important to select the correct material at design stage. During load changes or boiler startup the temperature in the tubes increase rapidly as there is no cold water or steam inside the tubes to cool it down. Considering these situations several studies were carried out [27,28] [29]. In all of them tube failure locations occurred in the superheater tubes.

Apart from the analytical nonlinear models developed for boiler dynamic study, commercial software is also used. In the literature, four commercial software were found; Modelica, Simulink&Matlab, ASPEN and APROS. Modelica was used by Franke [7] implementing Astrom [6] non-linear model. The goal was to improve boiler startup in order to minimize cost and environmental impact during plant operation on a 700 kW coal fired drum boiler. Casella [17] simulated a combined cycle power plant startup and reduced its time from 25,300s to 19,200s by removing intermediate stops during startup. In this new situation thermal stress in thick walled components was studied in detail. Sedić [18] used Matlab and Simulink to simulate a dynamic behaviour of a fuel oil single drum boiler and in contrast with [6] empirical relationships were avoided. The model was made to be applicable to any type of boiler facility. Alobaid developed several models based on Advanced Process Simulation Software (APROS) studying cold, warm and hot start ups in different types of boilers, for instance a vertical HRSG boiler [19], a triple pressure supercritical one-through HRSG [20] and Benson one through sub-critical boiler [21] were simulated. ASPEN was also used to simulate three pressures HRSG boilers during startup procedure [22]. A comparison between ASPEN and APROS tools was made considering a three pressure forced circulation HRSG [23,24]. In conclusion, similar results were obtained with both models. Nicolas used APROS [25] to compare warm and hot startup time in HRSG and one through HRSG boiler and to study dynamics on a three vertical HRSG pressure boilers [26].

Research was made regarding fast startup and dynamic calculations using commercial software or developed numerical models. In all of them, special attention was paid to the temperature of the superheater tubes and the steam drum thermal stress. However, neither studied the fast startup of 10 min from HSM, which is below the technical minimum load. The proposed methodology in this article differs from other authors' methodology on HSM. It is an operating mode where boilers are operating below their minimum technical load at nominal pressure conditions. Therefore, the boilers are kept in hot conditions with a steam production almost non-existent. As the boilers are already prepared they are able to do a fast startup, in the case study the plant needs the boilers at 100% Maximum Continuous Rate (MCR) within 10 min.

Other methodologies study boiler startups by considering them as shut down, a so-called “cold startup”. The advantage of this methodology is that boilers are not consuming any fuel as they are shut down. The disadvantage of shutting down the boilers is that it is not possible to do a fast startup in 10 min. The cold startup time takes around 6 h for these specific boilers as thermal stress must be checked. As an example, Taler [14] developed an operating methodology to reduce cold startup time from 6.5 h to 2 h, but with this methodology startup in 10 min is still not possible.

Others authors studied startup when boilers are operating at their minimum load [21]. In this case, the advantage is that as boilers are already hot, it is possible to do a startup in shorter time. Moreover, this methodology does a fast startup from minimum technical load to maximum load 100%MCR. According to boiler designers companies, boilers' minimum load is 10% MCR. The disadvantage of this methodology is that if there is no steam demand on the plant, there is a waste of energy, as boilers are not able to supply a steam production below their minimum technical load.

The proposed methodology in this article takes advantage and combines the benefits of the other methodologies proposed by previously named authors: (1) At HSM the boilers have an almost non-existent steam production. (2) At HSM boilers are already hot so a FHS in 10 min is possible. The disadvantage of this methodology is the required initial capital investment. Burner design is special, not standardised, and FHS control philosophy description is much more complex.

In conclusion, HSM implies an increase in boiler flexibility and energy and cost savings and so is considered a very attractive operational strategy.

2 Problem set up

For this study a petrochemical plant located in the north of Spain is selected. This petrochemical plant reuses wasted heat to produce steam for the plant functioning. Mainly gas turbines and waste heat boilers (WHB) are used in this plant. At the same time, three oversized auxiliary bi-drum boilers are also operating at their minimum load so that WHB and other oversized auxiliary boilers operating at their minimum load can cover the base steam of the plant.

Auxiliary bi-drum boilers are designed oversized to produce the necessary steam demanded from the plant when WHB stop working for any circumstance, such as maintenance or breakdown and also during peak and fluctuations of the steam demand. Oversized auxiliary bi-drum boilers are normally operating at their minimum load 10% MCR and when it is required, they start working at full load 100%MCR.

Recently, the plant made some changes in their process so that waste heat was better reused. Due to these improvements, overall plant efficiency increased and the steam demanded from the plant was reduced. Therefore, WHB cover the base steam demanded from the plant and it is not necessary to produce steam using the auxiliary boilers. The traditional technology of auxiliary boilers do not allow them to work below their minimum load (10%MCR) so there is a steam production which is wasted in the plant. A new operational methodology is necessary on auxiliary bi-drum boilers to reduce the cost of the plant and energy waste.

The solution to this problem could be to shut down auxiliary boilers. In any case this is not a possibility because when there is a peak or a variation in steam demand or when WHB stop working, auxiliary boilers cover those steam demand fluctuations. Shutting down the boilers means losing the possibility of doing a fast startup within 10 min. Boilers would be cold and it would take six hours to prepare them for full operational load. This is not acceptable as the plant requires a FHS in 10 min and for this reason the plant requires that boilers are kept operating in hot conditions and constant pressure conditions.

HSM and FHS operation modes are the new operation modes necessary for the three auxiliary steam boilers. It is not feasible to shut down one boiler and maintain the other two in HSM. When WHB stop working, it is necessary to produce steam in the three auxiliary boilers at 100% MCR to provide steam for plant operations. Therefore, the three auxiliary steam boilers must be ready for FHS. Due to plant characteristics, fast startup must be feasible within 10 min, so HSM and FHS are needed requiring the development of the necessary technology to modify and include those operations modes in traditional steam generators.

3 Description of the auxiliary bi-drum boiler

The described boilers started operation in February 1983. The water tube bi-drum boiler is a two drum, bottom supported, natural circulation boiler with multiple gas pass designed to burn fuel gas. The furnace is fully water cooled using the membrane wall construction. A unique plenum encloses the upper front wall, roof and rear of the unit. The design enables the flue gases to flow horizontally and parallel to the drums through the evaporating bank.

An inverted loop drainable superheater is located behind a screen at the furnace outlet to protect it from direct furnace radiation. This location gives a semi-radiant heat transfer characteristic, which guarantees superheater temperatures at low loads (see Fig. 1). An economizer is used to reduce exhaust gas temperature and improve boiler efficiency.

Fig. 1 Steam generator.

alt-text: Fig. 1

Cyclone steam separators with secondary scrubbers are included in the steam drum to produce high quality steam.

The steam generators are designed for a steam production of 42.5 bar(g) at 371 °C and 90,000 kg/h firing fuel gas. The boiler and auxiliary equipment are designed to operate from 10% MCR to 100% MCR.

Additionally, the described boiler is a natural circulation boiler, where water-steam mixture flows naturally inside the tubes. The main components forming a natural circulation loop are a steam drum, downcomer tubes and riser tubes. Feedwater enters the boiler steam drum from the economizer. Water flows to the water drum through a number of downcomer tubes after which the water-steam mixture flows back to the steam drum through a much larger number of riser tubes. Furnace heat is transferred to riser wall tubes where steam is starting to form. As steam-water mixture is less dense than water in downcomer tubes, gravity will cause water to move downwards while water-steam mixture moves upwards. The rate of water flow depends on the average density difference between the unheated water and the heated steam-water mixture. In natural circulation, the density difference between water and the steam-water mixture produces enough force to overcome frictional and gravitational resistance to flow and produce a velocity of circulation sufficient to always maintain a flow of steam through the tubes, which avoids overheating of the tubes.

Boilers have four fuel gas burners arranged in two levels, with a distribution of two burners per level. Each burner is designed for a power release of 33%MCR with a turn down ratio of 1:7. They are designed for manual operation where the furnace is purged and the burner started, ignited and stopped manually, each burner's air register is also manually operated.

A schematic flow diagram of one auxiliary bi-drum boiler is shown in Fig. 2.

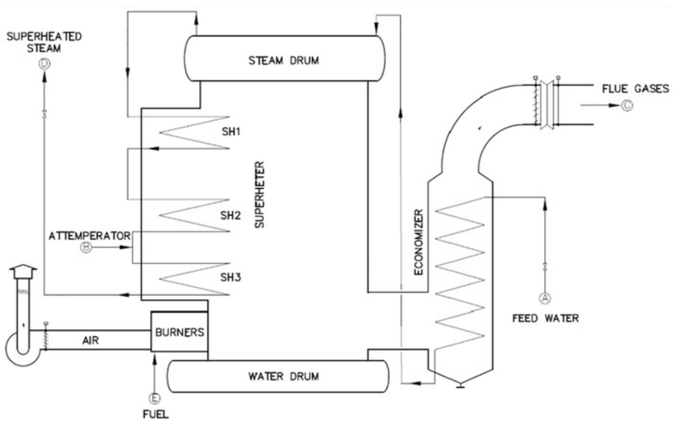

Fig. 2 Flow diagram of auxiliary bi-drum boiler.

alt-text: Fig. 2

The main characteristics of the boiler are summarized in Table 1.

Table 1 Boiler characteristics.

alt-text: Table 1

Variable	Value	Units
Boiler type	Bi-drum water tube boiler	
Circulation	Natural circulation	
Furnace dimensions	Height 4100 Width 4420 Length 5750	mm
Refractory in furnace floor	YES	
Number of burners	4	
Burners arrangement	Two levels	
Steam Flow (100% MCR)	90,000	Kg/h
Steam Pressure	42.5	bar
Steam Temperature	371	ºC
Thermal power input	69	MW

Current auxiliary boilers are working at their minimum load, 10%MCR. Produced steam is wasted in the plant, this corresponds to a steam mass flow of 9000 kg/h equivalent to a thermal power of 6.9 MW and a loss of 178€/h per boiler. Auxiliary boilers are operating at standby mode 30% of the year, that is, 2628 h/year. The rest are supplying steam to overcome plant steam fluctuations demand.

4 Results

The main result of this article is the methodology proposed as differs from other authors' methodology on HSM. The confirmation that this operation methodology is applicable is the main conclusion. Thanks to HSM, boilers are already prepared to do a FHS in 10 min while operating below 10% MCR.

Other methodologies study boiler cold startups, in these cases the boilers are not hot so it is not possible to do a fast startup within 10 min. For the boilers described in this paper 6 h are required to heat the boilers to operating at 100%MCR when initially the boilers were shut down. Despite the fact that some research has shown the possibility to reduce this time substantially [14], a startup within 10 min was still considered impossible.

Previous authors studied startups when boilers were operating at their minimum load [21]. In this case it is possible to do a startup in shorter time, but these methodologies do a fast startup from 10% MCR to 100% MCR. However, if there is no steam demand on the plant there is a waste of energy as boilers are not able to supply a steam production below their minimum technical load.

With the proposed methodology boilers operating at HSM could realize a FHS in 10 min starting with almost zero steam production. Although the presented methodology will result in cost and energy savings, the main drawback is that the initial investment is high. Special equipment will need to be purchased in order to be able to work under these conditions.

In order to validate proposed new operation methodology, HSM and FHS must be studied.

4.1 Hot standby mode (HSM)

HSM is defined as the minimum steam load at which a boiler is operating with constant nominal pressure condition. The heat loss through the boiler enclosure will suppose a pressure reduction. In order to maintain a constant steam pressure it is necessary to introduce a power input equal to the heat loss through the enclosures. That power input is introduced by boiler burners; therefore, HSM is defined with burner power input equal to heat loss. To achieve that, heat transfer loss through boiler enclosures must be calculated. There are two different methods:

Method 1: Heat transfer loss is calculated applying radiation, convection and conduction heat loss formulation.

Considering boiler wall tubes like a plane wall and applying thermal electrical analogy, heat transfer and thermal resistances are defined as:

$$\dot{Q} = \frac{\Delta T}{R} = UA\Delta T \quad (1)$$

$$R_{conv\&radiation} = \frac{1}{hA_S} \quad (2)$$

$$R_{wall} = \frac{x}{kA} \quad (3)$$

For the calculation it is assumed that:

- Tube wall temperature is equal to steam temperature (255 °C)
- Ambient temperature is equal to 25 °C
- Ambient combined radiation and convection heat transfer coefficient is 9.6 W/m^{2}K.
- Insulation material is mineral wood with a thermal conductivity of 0.051 W/mK

Applying the above formulation and assumption, calculated heat loss is showed in Table 2.

Table 2 Heat loss calculation.

alt-text: Table 2

Heat loss calculated with insulation		
Boiler surface	273.1	m^2
Others items surface	20	m^2
Economizer surface	108	m^2
Boiler tube wall temperature	255	ºC
Ambient temperature	25	ºC
Insulation material	mineral wool	
Insulation thickness	75	mm
Insulation conductivity	0.051	W/mK
Radiation and convection coefficient	9.6	W/m^{2}K
Heat Flux	146.05	W/m^2
Surface temperature	40.21	ºC
Insulated surface	401.1	m^2
Heat loss through insulated surface	58.58	kW
Uninsulated surface	2	m^2

Heat lose uninsulated surface	4.42	kW
Total heat loss 63 kW		

Method 2: To apply European standard EN 12952-15 30] (Corresponds to reference [30] of bibliographic references).

According to EN12952-15 European standard heat losses are described by:

$$\dot{Q}_{EN} = C * \dot{Q}_N^{0.7} \quad \textbf{(4)}$$

The obtained results can be seen in Table 3.

Table 3 EN 12952-15 heat loss calculation.

alt-text: Table 3

Heat loos according to EN 12952-15		
Heat output water/steam	67,408	kW
Type of fuel	Light Oil or natural gas	
C	0.0113	
$\dot{Q}_{EN}$	**215.4**	**kW**

Comparing the heat loss calculated with both methods, the result obtained from EN-12952-15 standard is the least favourable. The least favourable value is selected, so that the boiler will have a heat transfer loss through boiler enclosures of 215 kW. To overcome heat transfer losses and guarantee HSM the minimum power necessary to introduce to boiler furnace is 215 kW.

As mentioned before, each existing manual burner is designed for a power release of 33%MCR with a turn down ratio of 1:7. Considering that the power input necessary at full load is 69 MW, each burner minimum load is 3.3 MW. Therefore, existing burners are not capable of supplying 215 kW. Considering those values, a new design for the combustion system is necessary for HSM.

The results obtained from thermal calculations at the defined minimum load called HSM are shown in Table 4.

Table 4 HSM process data.

alt-text: Table 4

Variable	Value	Units
Steam flow	< 100	Kg/h
Steam temperature	254	ºC
Steam pressure	42.5	bar
Combustion air flow	500	Nm^3/h
Feed water temperature	142	ºC
Fuel gas flow	16	Nm^3/h
Burner power	215	kW
Boiler pressure drop	0	mm.w.c.

According to Table 4, defined boiler HSM is suitable to a steam production of 100 kg/h and a fuel consumption of 16 Nm^3/h. Comparing with the existing available minimum load, 10%MCR, this new operation mode will suppose energy savings and cost reduction.

4.2 Fast hot startup study (FHS)

When there is a problem in the plant and WHB break down auxiliary boilers should be able to supply a steam production of 90,000 kg/h at nominal conditions in no more than 10 min from HSM. To guarantee FHS it is necessary to do thermal dynamic calculations. However, the burner ramp up shall first be defined as it is the set point for the dynamic calculations. To validate defined burner ramp and dynamic calculations the following items must be checked:

1. Drum swell, to guaranty an accurate steam drum level control.
2. Superheater tube metal temperature, to avoid tube overheating and rupture.
3 Thermal stress in steam drum.

Power Plant Simulator & Designer (PPSD) commercial software was used to study thermal dynamic calculation. PPSD is a software of the company KED GmbH for the modeling of thermodynamic processes and steam generators. PPSD is mainly used in calculations and design of steam generators and power plants. The software was developed in 1993 by Grigory Doverman and Christian Daublebsky of Eichhain as a training simulator for PC-DOS-based power plants with a graphical interface. Later, the program was adapted for the design of steam generators and heat exchangers, subsequently extended to entire power stations_]31] ~~[30~~ (This reference must be replace by reference 31)].

PPSD includes flue gas, water/steam, natural circulation and fuel schemes. Flue gas path is modeled on flue gas scheme from atmospheric air supply to economizer exit, see Fig. 3. It consists of atmospheric air supply, a furnace, a superheater heat exchanger, a radiation cavity, an evaporator and an economizer. Division walls and water-tube walls are also represented. An air controller connects the air supply to the furnace. In the diagram flue gas velocity on each element and flue gas temperature between each element can be checked.

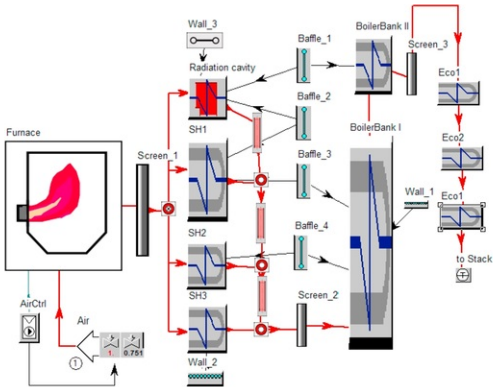

Fig. 3 Fuel gas scheme.

alt-text: Fig. 3

The water/steam path, see Fig. 4, is modeled on another scheme from feed water supply to steam battery limit. Battery limit (LS) is modeled as inlet boundary condition together with temperature, mass flow and the defined pressure parameters. The water level of the steam drum is controlled by supply water and the steam produced in the evaporator. Water is heated by flue gas in the evaporator and converted into saturated steam in the steam drum after passing through steam separators. The saturated water is transported to the evaporator, thanks to the natural circulation, and then reheated. The dry steam exits the steam drum and flows through the superheater, which is divided into three tube bundles. Between the first and second superheater tube bundle an attemperator is located. The attemperator has another control which regulates water supply to guarantee 371 °C at battery limit. At battery limit, the last control can be found, which regulates the fuel gas supply to guarantee 90,000 kg/h.

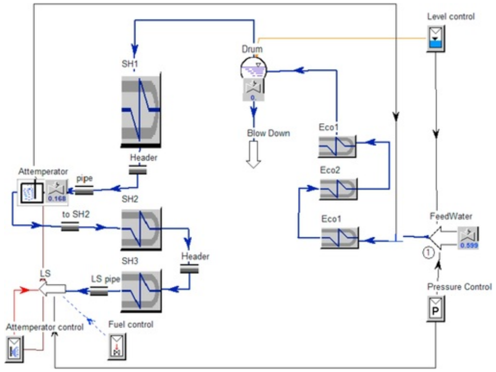

Fig. 4 Water/steam scheme.

alt-text: Fig. 4

Heat exchange bundles commonly called superheaters, evaporators and economizers have been implemented using real dimensional data. Dimensional data is preferably defined on the flue gas scheme as evaporators do not appear in the water/steam scheme. After the element connections, circuit control elements are defined. It is basic procedure to use and define controllers for dynamic behaviour calculation, such as, the drum level, the attemperator and the fuel controller. In the natural circulation scheme all the equipment engaged in the water/steam path is displayed. Finally, in the fuel scheme the different fuel compositions are defined.

4.2.1 Steady state model validation

Auxiliary bi-drum boiler model verification and evaluation with the design data for steady state 100% full load is performed. Another examination is made testing auxiliary bi-drum boiler at 110%, 70%, and 30% part loads. Therefore steam mass flow, excess air, pressure outlet and feed water inlet temperature boundary conditions are changed for part load simulation. Fig. 5 presents the superheater steam temperature, economizer flue gas outlet temperature and boiler flue gas outlet temperature; Fig. 6 shows the fuel mass flow at different loads.

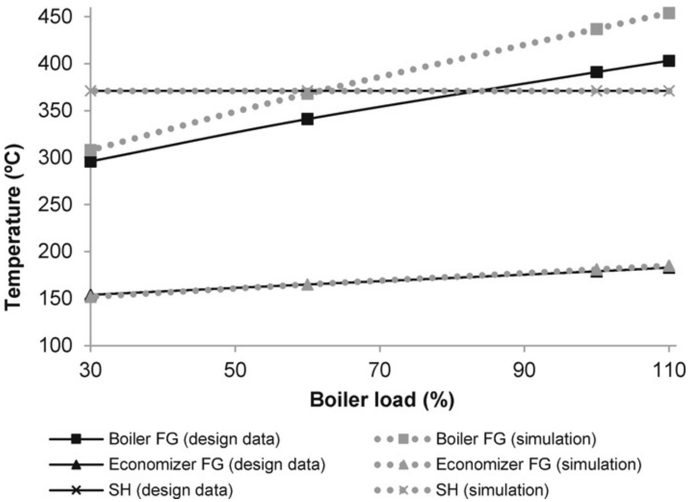

Fig. 5 Auxiliary bi-drum boiler temperatures outlet at different loads.

alt-text: Fig. 5

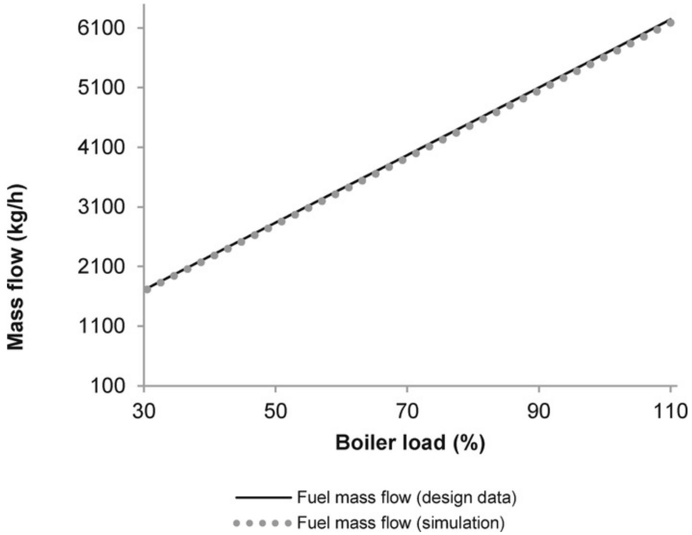

Fig. 6 Auxiliary bi-drum boiler fuel mass flow outlet at different loads.

alt-text: Fig. 6

The comparison between the simulations and the design data correlate well with each other. Thanks to the attemperator, superheater steam outlet temperature is kept constant equal to 371 °C. It is independent of the load change in both the simulation and the design data. Economizer flue gas outlet temperature results match well with design data. At 110% and 100% load the model incorrectly predicts the flue gas temperature at the boiler outlet. However, the model does follow the same tendency as the design data. Although the simulated flue gas boiler outlet temperature is 11% higher than the given one, the economizer flue gas outlet temperature has less than 1.5% error between design and simulation results. Furthermore, the computed fuel mass flow consumption corresponds accurately to the design flow consumption, where the relative error between the simulation and the design is less than 1.2%.

Once the auxiliary bi-drum boiler model has been compared to the design data for steady state and different load changes have been performed with high accuracy, the fast startup simulations with validated model can be produced.

4.2.2 Burner ramp

Burner ramp is the burners' power release. The higher the power liberation the lower the time needed to achieve nominal conditions. Burner power liberation is defined as a function of time to allow auxiliary boilers operating at full load in 10 min.

The basic considerations for FHS and burner ramp definitions are:

- The steam generator is in HSM at 42.5 bar(g).
- The steam generator is kept in HSM with a burner load of 215 kW
- Superheater drain valves must be opened during the first minutes of the startup.
- With 2000 kW burner power, there must be a minimum flow through the terminal steam valve of at least 1 t/h (in this situation it is recommended to close superheater drain valves).
- After 10 min from startup signal the steam generator must be delivering a steam production of 90,000 kg/h, 42.5 bar(g) and 371 °C.
- The live steam terminal valve is controlling the pressure during the startup
- Burner's ramp power release during FHS follows Fig. 7.
- Air excess during FHS shall be according to Table 5.

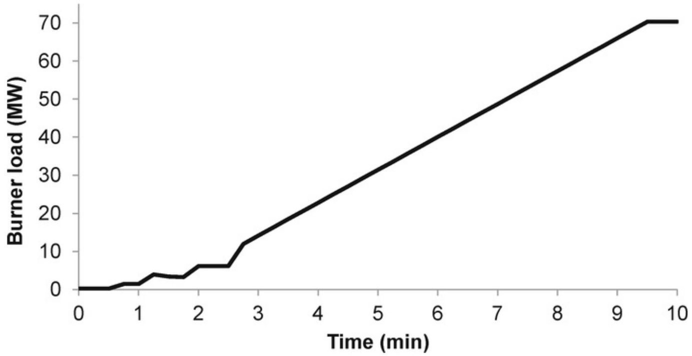

Fig. 7 Burner ramp.

alt-text: Fig. 7

Table 5 Excess air definition.

alt-text: Table 5

%MCR	25	50	75	100
Excess Air	1.35	1.22	1.15	1.12

Defined excess air during FHS shall follow the data of Table 5.

As illustrated in Table 5, the excess air is higher at lower performance load. This is to get a complete combustion and to maintain the flame direction in the furnace, thus avoiding flame flash-back.

The burner's defined minimum load is 215 kW. Existing burners are not able to supply such a minimum power, so existing combustion systems shall be replaced. Besides, they are manual burners; so one operator per burner is needed during startup to manually modify the air register position which takes more than 10 min to startup.

The proposed design is to replace the four manual burners by four new automatic burners of 17.5 MW in order to guarantee a FHS.

To guarantee a power release of 215 kW and in order to have a homogeneous power release to the furnace, the two burners on the first level should have special igniters capable of releasing a power from 100 kW to 700 kW. With such power output, the boiler should operate in HSM by the special igniters located on the burners on the first level.

Following the defined burner ramp, FHS could be made within 10 min. In order to validate defined burner ramp and FHS, three items must be checked: drum swell, superheated tube metal temperature and thermal stress in steam drum.

In addition the volume of water and weight to be heated during FHS is considered in thermal dynamic calculation, the values of which are shown in Table 6.

Table 6 Steam generator volume and weight data.

alt-text: Table 6

	Volume (m3)	Weight (kg)
Steam generator	20	50,000
Steam drum	11.6	24,000
Mud/Water drum	1.4	6000
Economizer	10	32,000
Superheater & headers	3.7	12,500

4.2.3 Drum swell

Due to the rapid increase in burner ramp, a fictitious increase in boiler water level is produced by the increase of bubbles in the water-steam circuit due to heating. This water level increase must be studied to confirm that there will not be steam drum water level problems with the defined burner ramp for FHS.

For the drum swell calculation the water mass storage in the pressure boiler before and after the startup should be compared. The water mass difference corresponds to the water stored in the steam drum. This water mass accumulation will increase the water level. It is important to check that this situation does not happen. In Table 7 the data obtained for drum swell calculation can be seen.

Table 7 Drum swell calculation.

alt-text: Table 7

	Tubes volume	H_2O mass before startup	H_2O mass after startup
	m3	kg	kg
Screen	0.67	555	454
Baffle	0.9	745	642
Boiler bank	8.85	7313	6621
Furnace floor	0.79	652	560
Furnace front&rear wall	1.51	1243	720
Side walls	3.65	3012	2630
Total	**16.37**	**13,520**	**11,627**

As it is shown, the volume of the tubes of the steam generation is 16.36 m^3, equivalent to a mass of water at HSM of 13,520 kg. After startup, once vaporization has started, there is a water mass of 11,627 kg distributed along boiler pressure parts. The remaining water mass of 1893 kg is stored in the steam drum, increasing the water volume. The water mass (1893 kg) stored in the steam drum corresponds to a water level increase of 250 mm.

According to the dynamic calculations the increase of the level happens in the first 75 s of the startup. As the steam drum's diameter is 1524 mm, swell problems will not happen. However, it is important to keep the balance between the supply of water and the steam production. During startup there will be 10 min to control the drum level and with steam releasing from the drum, the drum water level will decrease.

To avoid steam drum water level fluctuation problems during FHS it is recommended to fully open the continuous blow down valve.

4.2.4 Superheater tube metal temperature

During load changes, the superheater metal tubes temperature is the most sensitive part of the FHS and is thus another parameter that needs to be studied to validate the defined burner ramp.

The superheater is simulated in commercial software and divided into three tube bundles. The first one is located after the steam drum, the second and third one after the attemperator (de-superheater). The highest metal temperature is in the third tube bundle as this tube is cooled less due to the higher superheater steam temperature. Following from this, the third superheater metal tube temperature evolution must be checked during FHS. Fig. 8 shows the superheater tube metal temperature evolution as a function of time during FHS for the three superheater tube bundles. The rectangle's line represents the third tube bundle metal temperature.

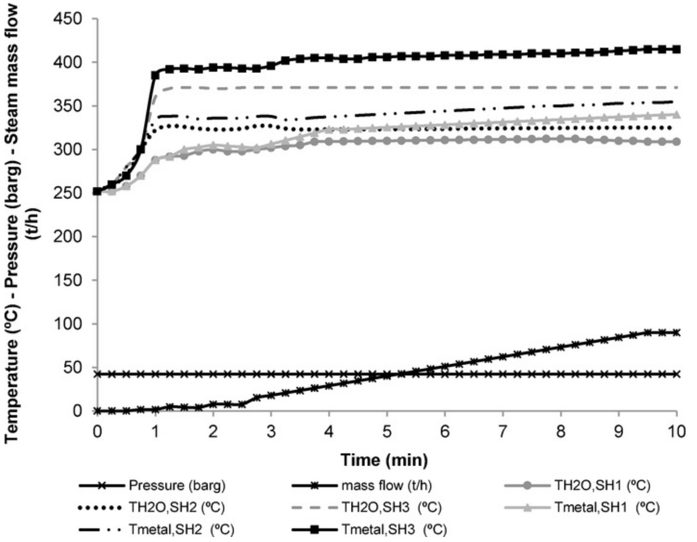

Fig. 8 Superheater tube metal temperature evolution.

alt-text: Fig. 8

According to Fig. 8, the maximum temperature gradient happens during the first minute of the FHS. That is when the burner load changes. At this stage, the burners will stop working with the special igniters and start working with burner lances. However, superheater temperature change is on the safe side as the superheater tube material is SA213T22 [32] and the metal temperature will always remain below 450 °C.

4.2.5 Thermal stress in steam drum

Thermal stress is caused when there is a temperature change. Materials expand when temperature increases and contract when temperature decreases. The most common issues that occur in thick walled components are so called creep and fatigue effects. The former depends on temporal peak stress suffered whereas the latter is caused by repeated application of stress or thermal expansions and contractions on the boiler metal.

Temperature gradients appear in the auxiliary steam boilers during the startup and load change. The thickest component on a steam generator is the steam drum, so thermal stress in the steam drum during startup must be checked to validate defined burner ramp.

The drum thermal stress is related to boiler life cycles. Due to this, auxiliary boiler admissible numbers of cycles are verified considering temperature gradients in the steam drum during FHS.

The calculations have been made following EN 12953-3 [13] standard, section 13, Appendix B and C. Fatigue life usage factor is defined considering linearly accumulated damage:

$$\sum_{1}^{k} \frac{n_k}{N_k} = \left(\frac{n_1}{N_1} + \frac{n_2}{N_2} + \ldots + \frac{n_k}{N_k} \right) \qquad \textbf{(5)}$$

The sum is contrasted with fatigue limiting usage factor. Fatigue limiting usage factor value is 0.4 considering the number of cycles (n) equal to 2000. In Table 8 there is a resume of the calculations.

Table 8 Resume fatigue calculations.

alt-text: Table 8

Fatigue life calculations and Corrosion Fatigue Cracking Checks				
			SH header	Steam drum
			hot start up	cold start up
Fatigue curve stress range temperature correction, Ct*	Ct*		0.80	0.84

Fatigue curve stress range temperature correction, Ct*	Ct*		0.80	0.84
Temperature corrected stress range, 2fa*(t*) = 2fa*/Ct*	2fa*t*	N/mm^2	16.66	341.58
Fatigue curve statistical scatter stress corrector factor; Ss	Ss		1.50	1.50
Final notch, mean stress/plasticity, temperature and statistical corrected stress range, 2fas = 2fa*(t*) x Ss	2fas	N/mm^2	24.99	512.37
Endurance limit			330.90	330.90
Minimum cycles to cracking at 2fas; Nas	Nas		10^7	$2.8*10^5$
2faL = 2fa*(t*)	2faL	N/mm^2	16.66	341.58
Mean cycle to cracking at 2fal, NAL	NAL		10^7	$5*10^7$
Statistical factor to adjust from mean to minimum cycles to cracking, SL	SL		10	10
Minimum cycles to cracking with factor of SL on cycles to cracking: NAL/SL	NAL/SL		10^7	$5*10^7$
Number of cycles supposed	n		$2*10^3$	$2*10^3$
Fatigue usage factor = n/MIN (Nas, NAL/SL)	n/N		$2*10^{-4}$	$7*10^{-2}$
Limiting usage factor	$(n/N)_{limiting}$		0.4	0.4

As it is shown in Table 8, the fatigue usage factor is below the limiting usage factor so there will be no issues regarding thermal stress during the FHS in fact, there is no limit with the number of FHS for the whole life cycle of the boiler.

5 Conclussions

The boiler should be able to operate in HSM with a burner power of 215 kW, resulting in energy savings and operation cost reductions. Special ignition lances will be used in first level burners to supply that power.

Moreover, FHS could be achieved from HSM in 10 min without any problem on the boiler if the described startup procedure and burner ramp are applied.

There is no danger in having too high a metal tube temperature on the superheaters during FHS when implementing defined burner ramp. However, after long period of HSM, the saturated steam in the superheater header will condensate and will block the superheater tubes. This can cause incorrect steam distribution and cause overheating of the superheater tubes. To avoid superheater tubes from overheating, its header drain size and location must be checked. In the worst case 3.75 m^3 water will need to be drained within 1 min.

There is no limit to the number of FHS when considering the life time cycle consumption with planned startup time and conditions.

In order to assure HSM and FHS, the following modifications must be made to the existing industrial boiler and auxiliary equipment:

- New burners shall be supplied that are able to operate at defined HSM as current burners are not able to operate at such low power.
- Combustion system gas skid shall be changed to adapt to the current standard [33].
- The combustion system shall be automated as manual operation does not enable FHS.
- A new feed water control valve in parallel position for the HSM will need to be implemented as existing feed water control valve is not able to operate in HSM.
- Superheater manual drain valve will need to be automated in order to be opened before FHS.
- Continuous blow down valve will need to be automated to avoid drum level fluctuations due to drum swell. It shall be fully opened during startup procedure.
- It is important to maintain the supply of feed water equilibrated to the steam produced during FHS to avoid steam drum level problems.
- A new control philosophy will need to be defined and programmed in the burner management system [33] for FHS.

- Defined burner ramp shall be followed.

The proposed new operation methodology reduces the cost in auxiliary bi-drum boilers operation. Following the recommendations and applying detailed technological changes, HSM and FHS in 10 min are possible, with no issues occurring related to drum level, overheating tubes or thermal stress.

The necessary capital investment per boiler is around 600,000€. Each boiler will be operating at HSM 30% time of a year, 2628 h. If boiler operation methodology is not improved boilers will be operating at their minimum load, 10%MCR, wasting steam in the plant with a fuel consumption of 0.5 t/h. The energy waste has a cost of 178 €/h. Operating hours in this conditions are 2628 h, so the total cost per boiler is 467,784€. In HSM this cost would not exist, so the initial investment is paid back in 600,000/467,784 = 1.28 years.

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Highlights

- A new operation methodology for auxiliary bi-drum boilers has been developed.
- Operational cost of the plant has been reduced thanks to HSM.
- HSM is an operating mode below minimum technical load.
- HSM allows for a 10 min startup, so-called FHS.
- Modifications to current technology have been defined.

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