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Comparative investigation of drum-type and once-through heat recovery steam generator during start-up

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HIGHLIGHTS

- First start-up comparative study of once-through and natural-circulation heat recovery steam generator.
- HP heat exchanger surface is 8% lower for the natural-circulation heat recovery steam generator.
- Faster pressure build-up in the once-through evaporator.
- High temperature gradients in the HP drum wall.
- Uniform wall temperature re-establishes more than 90 min faster in the HP separator.

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ABSTRACT

This study investigates the impact of the design of a heat recovery steam generator (HRSG) on its dynamic behaviour under the boundary condition of a gas turbine start-up. For that purpose, a validated HRSG model with three pressure stages and reheater section is modified by replacing the once-through evaporator in the high pressure circuit with a natural circulation evaporator, including the associated control circuits. Both models are designed to supply equal steam mass flows with equal steam parameters (temperature, pressure) at full load, which enables a balanced assessment of the two technologies. After an extensive description of the modelling approach and its practical realisation, detailed simulation results for start-up procedures from warm and hot initial conditions are presented. Differences in the transient behaviour of the HRSGs are highlighted and discussed. In industrial practice, frequent start-ups cause increased material fatigue, which in turn has an adverse effect on the operating lifetime of a power plant. Hence, the present work is complemented by an analysis of the temperature gradients in the most critical components with respect to thermal stress. Results generally show similar responses of the high pressure systems to gas turbine start-up with the exception of accelerated pressure build-up in the once-through evaporator. Greater temperature deviations are observed in the natural-circulation HRSG across the wall of the high-pressure drum.

1. Introduction

Combined cycle power plants (CCPP) differ from other thermal power plant technologies due to high process efficiency and operational flexibility, low emission level and moderate investment costs. In CCPPs the waste heat of a gas turbine (GT) is used to power a water/steam cycle by means of a heat recovery steam generator, which is installed downstream in the flue gas path. Whereas early plants still used single-pressure HRSGs, more pressure stages were subsequently added to the water/steam cycle. Hence, more exergy can be recovered from the hot flue gas. Subcritical heat recovery steam generators with three pressure circuits and adjacent reheater section are considered state of the art [1]. Modern plants reach net efficiency factors of more than 60% at full load, with gas turbine inlet temperatures ranging from 1773 K to 1873 K [2]. Current research is aimed at further increasing the inlet temperature by developing innovative cooling concepts, new materials and thermal barrier coatings for the combustion chamber and the first gas turbine stages [3]. This enhancement will also benefit the bottoming cycle due to higher steam parameters. Thermodynamic calculations show that the CCPP net efficiency may reach up to 65%, given a gas turbine inlet temperature of around 2023 K [4]. Furthermore, the integrated gasification combined

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Nomenclature

A	cross-section area (m^2)
c_k	two-phase friction multiplier (–)
D_H	hydraulic diameter (m)
E	rate of entrainment (–) or modulus of elasticity (MPa)
F	force/volume (N/m^3)
f	friction coefficient (–)
g_z	gravitational component in z-direction (m/s^2)
H	height (m)
h	static enthalpy (kJ/kg)
h_0	total enthalpy (kJ/kg)
k	heat transfer coefficient (kg/s), [W/(m^2 K)]
$\dot{m}$	mass flow rate (kg/s)
n_w	number of control volumes (–)
L	liquid level (m)
l	length (m)
P	power (MW)
p	static pressure (bar)
$\dot{Q}$	heat flow/volume (kW/m^3)
$\dot{q}$	heat flow/area (kW/m^2)
R	rate of stratification (–)
T	temperature (K)
t	time (s)
u	fluid velocity (m/s)
z	spatial coordinate (m)
α	void fraction (–), stress concentration factor (–)
β	thermal expansion coefficient (1/K)
Γ	mass exchange rate [kg/(m^3 s)]
δ	wall thickness (mm)
λ	heat conductivity [W/(m K)]
η	dynamic viscosity [kg/(m s)]
ν	Poisson's ratio (approximately 0.3 for steel)
ρ	density (kg/m^3)
σ	stress, surface tension (MPa)

Subscripts

a	annular flow
av	average
b	bubbly flow
d	droplet flow
el	electric
fc	forced convection
fl	form loss
g	gas phase
i	component index or interface between phases
ig	interaction between phase interface and gas phase
ik	interaction between phase interface and liquid/gas phase
il	interaction between phase interface and liquid phase
in	inner wall
k	liquid or gas phase
l	liquid phase
lin	linear
max	maximum
nb	nucleate boiling
nc	natural convection
ns	non-stratified flow
pb	pool boiling
pu	pump
s	stratified flow
sat	saturation
sp	single phase
t	tangential
th	thermal
T	with respect to temperature
va	valve
w	wall
wg	interaction between wall and gas phase
wk	interaction between wall and liquid/gas phase
wl	interaction between wall and liquid phase

Abbreviations

BFP	boiler feed pump
CCPP	combined cycle power plant
CPH	condensate preheater
ECO	economiser
EV	evaporator
FEM	finite-element method
FG	flue gas
GT	gas turbine
HRSG	heat recovery steam generator
HP	high pressure
IP	intermediate pressure
LCF	low cycle fatigue
LP	low pressure
PI	proportional–integral controller
RH	reheater
SH	superheater
ST	steam turbine

cycle is a promising concept for the future in order to make the high-efficiency combined cycle available to solid fuels such as coal and biomass [5].

In the recent decade, plant operators have increasingly shifted their focus from high efficiency at design load to increased operational flexibility, namely fast start-up procedures. The rising share of renewables in the overall electricity feed-in introduces a significant source of intermittent power supply into the grid. In order to mitigate the negative dynamics that wind energy and photovoltaic may trigger, a flexible reserve of conventional power generation is required. Combined cycle power plants are particularly suited to compensate for fluctuations in the electricity grid, since modern gas turbines only need 20 min to reach 100% of their nominal load for cold, warm and hot start-ups. Seven minutes after gas turbine start, 70% of the nominal exhaust temperature and 60% of the nominal flue gas temperature are available already [6]. However, start-up ramps are limited by thermal stresses in the heat recovery steam generator due to the large wall thickness of the high pressure circuit [7]. Whereas conventional HRSG design is largely based on simple steady-state models, detailed modelling and dynamic simulation of the relevant components are necessary in order to evaluate and optimize HRSG design with respect to fast start-up capability. Heat recovery steam generators are divided into vertical and horizontal designs, which in turn can be realized with a combination of drum-type or once-through heat exchangers.

Drum-type circuits typically use natural circulation for horizontal designs and forced circulation for vertical designs. This is due to the fact that horizontal evaporator pipes are more susceptible to backflow so that pumps are required for preventing system instability. Operational experience shows that combined cycle plants with vertical HRSGs are cycling tolerant systems, as the design permits the tubes to expand/contract freely and independently of one another [8]. In contrast, evaporator tubes for horizontal designs are hanging vertically in a more rigid harp structure. In order to support their own weight, a larger wall thickness must be selected