Density Wave Oscillation in the Horizontal Parallel Tube Paths of the Evaporator of a Natural Circulation Heat Recovery Steam Generator – A Theoretical Investigation

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Abstract: - The paper presents the results of a theoretical stability analysis on a vertical type natural circulation heat recovery steam generator. The evaporator of the boiler consists of four parallel horizontal tube paths, which are connected at both ends with headers. For different operation pressures and geometries the dynamic instability and in particular the density wave oscillation (DWO) were analysed. The investigations show that the stability of the boiler can be improved by increasing the system pressure. A faster decay of the DWO can be achieved by the implementation of an orifice at the tube inlet of the evaporator (single phase flow), whereas the installation of an orifice at the tube outlet (two phase flow) results in a more unstable behaviour.

Key-Words: - Density wave oscillation, Dynamic instability, Natural circulation, Horizontal parallel tubes, Heat recovery steam generator

1 Introduction

Working media in process facilities or power plants e. g. heat exchangers, heat recovery steam generators (HRSG) which are arranged behind a gas turbine or a blast furnace, are operated in either single phase or multiphase condition. Water and steam are the most widely used working media. Multiphase flow has a stronger tendency towards unstable behavior than single phase flow. These so-called fluid-dynamic instabilities can manifest themselves in the form of pressure drop or mass flow fluctuations, a periodical change of the flow pattern of the two phase mixture, or fluctuations of the wall temperatures. The instabilities can cause vibrations of plant components, a non-uniform mass flow distribution in the tube bank. They can accelerate the initiation of a boiling crisis, which is characterized by a sudden rise of surface temperature due to the drop of heat transfer coefficient, a non-uniform heat flux or a difficulty measurement and control of system parameters. Thermo-hydraulic oscillations and flow instabilities can influence the operation and the control of the plant in a negative way and therefore they are undesirable.

The instability of a system can be either static or dynamic. Flow instabilities and thermo-hydraulic oscillations have been analysed for a long period of time and many works have been performed by many researchers since flow excursion was at first investigated by Ledinegg [1]. A main classification of the thermo-hydrodynamic instabilities can be seen e. g. in Bouré et al. [2], Bergles [3] or Yadigaroglu [4]. These classifications are based on the distinction between the static and dynamic character of the conservation laws, which are used to explain the dynamics of the unstable equilibrium state. A flow is subject to a static instability if the steady state flow becomes unstable under certain conditions. It translates to another quite different operational condition or to a periodic behavior. These types of instability can be predicted with the help of a steady-state analysis. Studies on the static instability are presented in [5] – [13].

In contrast to the static instability, the dynamic instability is caused by transient inertia or dynamic feedback effects if they have essential impact in the process. For the determination of the stability boundaries of such a dynamic instability the steady state principles are not sufficient enough, because several elementary physical instability mechanisms often contribute to the overall behaviour of the system. A large part of the studies on dynamic instabilities is focused on forced circulation ([14] - [18]) and relatively less work has been done on natural circulation systems [19] - [23].

Generally, the density wave oscillation is the most common type of dynamic instability encountered in two-phase flow systems [4]. The instability is a result of the multiple feedback effects in relationship between the flow rate, steam generation and pressure drop in a boiling channel. The DWO is characterised by large amplitudes and a nearly sinusoidal period. The DWO is a low frequency oscillation (~1 Hz) related to a period of one or two times the duration which is necessary for a fluid particle to travel along the channel [24]. The transportation delays in the channels are of high importance for the stability of the system. The oscillation of the pressure and the mass flux are in phase. The physical mechanism of the density wave oscillation is described in detail in [4].

Natural circulation is an important operation mode in industrial heating processes, nuclear power plants and chemical plants e. g. heat recovery steam generators or boiling water nuclear reactors (BWR). Beside the BWR the natural circulation operation mode is also used in light water nuclear reactors for removing the shutdown decay heat during a reactor accident. Therefore many of the work on thermo-hydraulic instabilities, especially on density wave oscillation (DWO), are done from the viewpoint of nuclear reactor safety. In these cases the investigated boiling channels are arranged vertically.

Chang and Lahey Jr. [23] made an analytic model for their investigation of the non-linear dynamics of the boiling system of a BWR. The one-dimensional model for the boiling channel comprised a nodal formulation that uses a homogeneous equilibrium assumption for the adiabatic two-phase flow, a lumped parameter approach for the heated wall dynamics and a point neutrons kinetic for the consideration of the nuclear feedback in the BWR loop. The analysis of the nuclear coupled DWO in a simplified BWR (SBWR) has shown a limit cycle may occur for abnormal operating conditions.

Fukuda and Kobori [25] have shown analytically and experimentally that two different types of DWO (Type 1 and 2) exist. For a heated channel with a subcooled mass flow at the inlet a dimensionless phase change number

$$N_{pch} = \frac{\dot{Q}}{A w_{in} \rho_f r} \frac{\rho_f - \rho_g}{\rho_g} \tag{1}$$

and a dimensionless subcooling number

$$N_{sub} = \frac{h_f - h_{in}}{r} \frac{\rho_f - \rho_g}{\rho_g} \tag{2}$$

can be calculated. With the help of these numbers a stability diagram can be developed for the channel as shown schematically in Fig. 1.

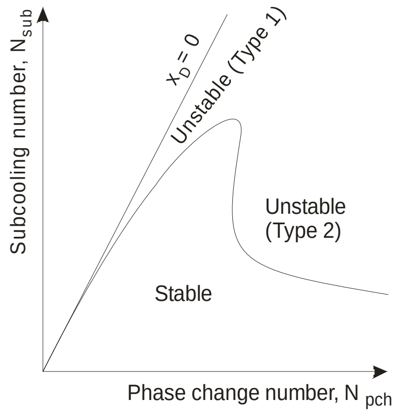

Fig. 1: Schematically stability diagram for DWO [26]

The type 1 instability is dominant when the steam quality is low at channel outlet, while the type 2 instability is relevant at high steam quality. The type 2 is well known as the typical density wave oscillation. But the mechanisms of both DWO - types are approximately equal.

Nayak et al. [19] investigated theoretically the nuclear coupled DWO in the Indian advanced heavy water reactor (AHWR). They used for their analytical model a point kinetics model for the neutron dynamics and a lumped parameter model for the fuel thermal dynamics. The linear stability theory was used to reveal the instability of in-phase and out-of-phase modes in the boiling channels of the AHWR. They found, that both in-phase and out-of-phase thermo-hydraulic instability may occur in the AHWR channels depending on the channel power and core inlet subcooling. The frequency of the Type 2 instability is larger than that for the Type 1 instability.

Aritomi and coworkers have conducted a series of experiments using two parallel channels to simulate the reactor core. They reported that flow instabilities may occur depending on the start-up procedure of an SBWR ([33]-[35]).

The studies done with a horizontal tube arrangement for the understanding of the dynamic instability are rarely reported.

Ding et al. [36] analysed the dynamic instabilities pressure drop oscillation, DWO and thermal oscillation in a horizontal, forced convection, in-tube flow boiling system. They found, that the oscillation can start earlier at a higher mass flow rate compared to an upward flow system.

Yüncü et al. [37] performed experimental and theoretical study to investigate the influence of the heat flux, exit orifice diameter and mass flow rate on two-phase flow instabilities (pressure drop oscillation and DWO) in a horizontal, forced convection, single channel system. The mathematical model is based on homogenous flow assumption and thermodynamic equilibrium between the two phases. The transient characteristics of the boiling two-phase flow horizontal system are obtained by perturbing the governing equations around a steady state. The experimental and theoretical results have been compared.

Karsli et al. [18] carried out an experimental work to study the effects of heat transfer enhancement on two-phase flow instabilities in a horizontal, forced convection, in-tube flow boiling system. They investigated five different heat transfer surface configurations and five inlet temperatures at constant heat input, system pressure and exit restriction. Pressure drop oscillation, DWO and thermal oscillation are found for all configurations. They found that the amplitudes and periods of the pressure drop oscillation and DWO for tubes with enhanced surfaces are higher than those of the bare tube.

Walter et al. [22] studied theoretical the density wave oscillations in the horizontal, parallel channels of an evaporator of a heat recovery steam generator (HRSG) under natural circulation conditions. They investigated various effects e. g. tube roughness, drum height, downcomer diameter and an evenly distribution of the heat flow to the tubes of the bundle heating surface of the evaporator on the DWO. They found that changes in the drum height and in the tube roughness has no significant influence on the oscillation amplitudes of the DWO. A decrease of the downcomer diameter and a homogenization of the heat absorption in the individual layers of the bundle heating surface have shown that the flow stability improves under operating conditions where density wave oscillations occur.

As seen from the literature reviewed, research into dynamic instabilities in horizontal flows under natural circulation conditions has been rarely reported. However, natural circulation systems with a horizontal tube arrangement are used widely in the industry.

In the present paper the author has performed a theoretical stability analysis for a vertical type heat recovery steam generator for the most important type of dynamic instability, namely the density wave oscillation.

2 Simulated Steam Generator

Figure 2 shows a sketch of a vertical type HRSG. The bundle heating surfaces are arranged horizontally and the flue gas enters the boiler at the bottom and leaves it on the top. The typical tube length per layer in such a boiler configuration is up to 20 m.

In natural circulation systems with horizontal arranged evaporators the water steam mixture has no preferred flow direction during start-up. Therefore the vertical type natural circulation HRSG is more unstable than a system with vertical evaporator tubes. Especially vaporizers at low operation pressure show tendency to flow instabilities.

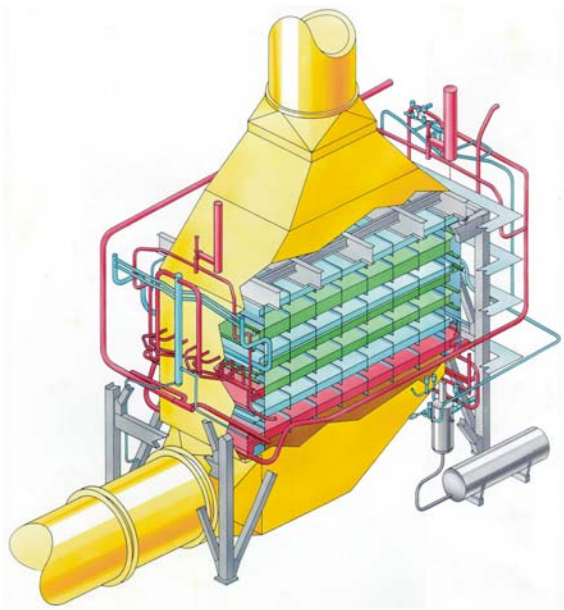

Fig. 2: Sketch of a vertical type HRSG [27]

A model of the analysed evaporator of the vertical type HRSG is shown in Fig. 3. The model includes a drum, a downcomer with siphon, a lower and an upper header, a heated tube bank with four horizontal parallel tube paths (the length of the tube per layer is 20 m) and a riser pipe.

The working medium enters the drum subcooled through the feed water tube and leaves the drum saturated through the downcomer. The water steam mixture, which is produced in the evaporator, enters the drum through the riser tubes. The two-phase flow will be separated in the drum. The steam leaves the drum at the top in direction to the superheater. The difference in height of the evaporator can be seen in Fig. 3. The drum level is controlled and hold at the drum centerline. For the bundle heating surface helically coiled finned tubes with segments are used. The tubes in the tube bank are arranged in a staggered way. The dimension of the evaporator tubes for the analyses was defined with Ø48.3 x 3.2 mm.

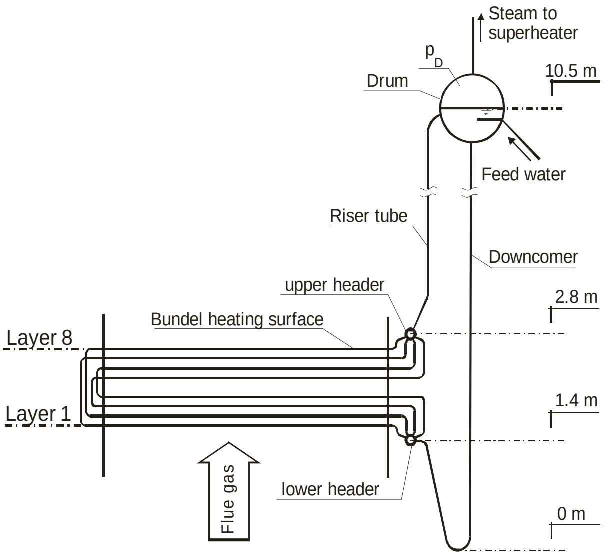

Fig. 3: Sketch of the simulated evaporator of the natural circulation HRSG

2.1 Initial and boundary condition

The start-up behaviour of the boiler depends on the time difference between the shutdown and restart of the boiler. For the start-up from the cold condition (cold start, the system pressure in the drum is equal to the atmospheric pressure) or part load condition (warm start) the admissible rate of temperature (pressure) change in the natural circulation systems is mostly conditioned by the thermal stresses in the large diameter components in particular headers and drums. This limitation applies both to load increase and reduction. A load change linked to a pressure change is accompanied by a change of the saturation temperature. Therefore variable pressure operation of a natural circulation system is only acceptable under certain conditions. Especially in the low pressure region the rate of change of the saturation temperature as a function of pressure is high. Tables with an acceptable start-up time for drums, cyclones and headers for different design pressures and wall thickness can be seen in [28].

The present study was done for a hot start of the boiler. In this case the time difference between the shutdown and restart of the HRSG boiler is up to approximately one day (overnight standstill) and the operation pressure is supposed unchanged during this time. Therefore the limitation of the pressure (temperature) change is not taken into account and the boiler can start in a faster way. Gradients in superheater headers are not within the scope of this analysis. The following initial conditions are used for the dynamic simulations:

- the steam generator is filled with water near boiling conditions
- the pressure distribution of the fluid is due to gravity
- the velocity of the working fluid in the tube network of the evaporator is zero and
- the fluid temperature in the evaporator of the boiler is equivalent to the boiling temperature at drum pressure.

The drum pressure, which is constant during the whole simulation, and the gas turbine exhaust flue gas temperature and mass flow (see Fig. 4), which are given as a function of time, are boundary conditions for the simulation.

Fig. 4 shows the time evolution of the gas turbine exhaust flue gas temperature (full line) and mass flow (broken line) for the hot start-up of the steam generator. The gas turbine reaches the steady state approximately 350 s after the start of simulation. After 350 s the gas turbine works at full load. The total time for the simulation was 2000 s. The flue gas ramps are equal for all simulations.

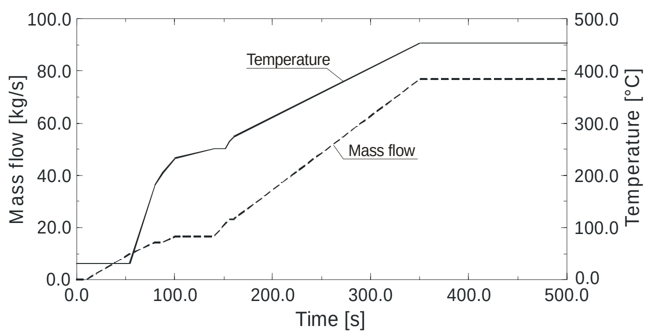

Fig. 4: Flue gas mass flow and temperature

3 Analysed Parameters

In the present study the following parameters are analysed:

- Operation pressure (variation of the drum pressure between 9 and 20 bars) (test case 1).
- Evaporator tube diameter in the connection zone between the last fin of the evaporator tubes and the upper header (Note: this variation of the tube diameter at evaporator outlet creates an effect similar to an orifice). In test case 2 the tube diameter for this connection zone was decreased from 48.3 mm to 44.5 mm.
- Implementation of an additional flow resistance at the inlet of the first four tube layers (test case 3).

At each simulation only one of the investigated parameters was changed.

4 Mathematical model

The computer program DBS (**D**ynamic **B**oiler **S**imulation) which was developed at the Vienna University of Technology, Institute for Thermodynamics and Energy Conversion, was used in these evaluations. The program was designed to analyze the dynamic behavior of steam generators, especially natural circulation HRSGs.

4.1 Model of the fluid flow in the tube

The mass flow in the tubes of a steam generator can be assumed to be one-dimensional, as the length of the tubes is much greater compared to their diameter. For the model under consideration, the topology of the tube network, the number of parallel tubes, the geometry in terms of outer diameter and wall thickness of the tubes, the fin geometry and the dimensions of the gas ducts are necessary. Furthermore, the thermodynamic data – such as mass flow, pressure and temperature – of the heat exchanging streams are used as input data.

The mathematical model [29] for the working medium is one-dimensional in flow direction and uses a homogeneous equilibrium model for two-phase flow and applies a correction factor for the two phase pressure loss according to Friedel [30]. For a straight tube with constant cross section the governing equations in flow direction can be written for the conservation of the mass:

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho w}{\partial x} = 0 \tag{3}$$

and for the conservation of the momentum:

$$\frac{\partial \rho w}{\partial t} + \frac{\partial \rho w w}{\partial x} = -\frac{\partial p}{\partial x} - \rho g_x + \left(\frac{\partial p}{\partial x}\right)_{Friction} . \tag{4}$$

The density ρ and the velocity w are averaged values over the cross section of the tube.

Considering the fluid flow in steam boilers, the thermal energy is much higher than the kinetic and the potential energy as well as the expansion work. Therefore, the balance equation for the thermal energy can be simplified to:

$$\frac{\partial \rho h}{\partial t} + \frac{\partial \rho h w}{\partial x} = \dot{q}\frac{U}{A} \tag{5}$$

The heat exchange between fluid and wall is governed by Newton's law and the heat transfer through the wall is assumed to be in radial direction only. The heat transfer models used in DBS for the single and two-phase flow of the working medium includes correlations for horizontal as well vertical tubes and is described in detail in [29].

4.2 Model of the header

For the calculations the following assumptions may be made for the header:

- Assuming that the distribution of the thermodynamic state of the header is homogeneous, the collector can be seen as one control volume for the calculation.
- The gravity distribution of density and pressure of the fluid inside the header can be neglected because the vertical dimension of the header is small compared to that of the remaining tube system.
- The huge difference in the cross section area of the header and the connected tubes causes strong turbulence, avoiding thus a segregation of the fluid in the header.

Because the headers are assumed to be a single control volume, the equations for the mass and energy balance are ordinary differential equations with time t as independent variable:

$$\frac{d}{dt}\left(\rho_C V_C\right) = \sum_j \rho_j w_j A_j - \sum_k \rho_k w_k A_k , \tag{6}$$

$$\frac{d}{dt}\left(\rho_C h_C V_C\right) = \sum_j \rho_j w_j h_j A_j - \sum_k \rho_k w_k h_k A_k \tag{7}$$

The variables of the header are denoted with the index C; j represents values at the header entrance and k values at the outlet. V_C is the volume of the fluid inside the header and A the cross sectional area of the connected tubes. Similar to the treatment of the fluid flow in the tube, kinetic energy as well as expansion work is neglected in the energy balance.

Because momentum is a vector quantity having the direction of the tube axis, the momentum fluxes must not be added arithmetically but rather as vectors. This is the case at the inlet and outlet of the headers, where the individual tubes are connected under different angles. The velocity of the fluid in the header is rather small compared to that inside the tubes. So it can be assumed that the momentum of the fluid will be lost at the inlet of the header and has to be rebuilt at the outlet. Based on this assumption, the momentum balance of the header reduces to a pressure balance. The changes of the momentum at the inlet and the outlet can be taken into account by a pressure loss coefficient ζ:

$$p_C = p_j - \frac{\zeta_j}{2}\rho_j w_j \left|w_j\right| \tag{8}$$

$$p_C = p_k + \frac{\zeta_k}{2}\rho_k w_k \left|w_k\right| \tag{9}$$

The discretization of the partial differential equations for the conservation laws was done with the aid of the finite-volume-method. The pressure-velocity coupling and overall solution procedure are based on the SIMPLER [31] and the PISO [32] algorithm. Both algorithms are included in DBS for a faster convergence of the code. Dependent on the convergence rate the program switches between both algorithms. To prevent checkerboard pressure fields a staggered grid is employed and for the convective term the UPWIND scheme is used.

4.3 Model of the flue gas

For the description of the flue gas flow through the boiler the one-dimensional partial differential equation of the conservation law for the energy is used. The flue gas mass flow is calculated quasi-stationary, while the energy balance is calculated unsteady. The momentum balance for the flue gas is neglected, because the pressure drop of the flue gas is out of interest. The discretization of the energy balance is done with the finite-volume-method.

The convective heat transfer coefficient between the flue gas and the tubes can be calculated with different correlations for plain or finned inline or staggered tube banks and is described in detail in [29].

5 Discussion of the results

Figure 5 shows as a result of the numerical simulation the development of the mass flow in selected tubes of the natural circulation HRSG over the time at a drum pressure of 12 bars. In this case the overall steady state circulation ratio was approximately 14.4.

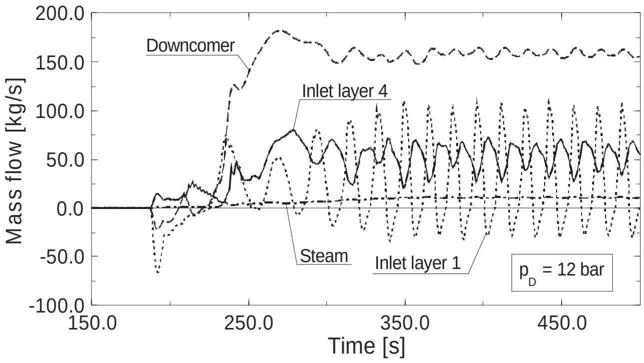

Fig. 5: Mass flow in selected tubes of the HRSG

It can be seen that the mass flow in the four tube paths starts approximately 200 s after start of simulation to oscillate. In the following the mass flows in the downcomer and the riser as well as in the bundle heating surface oscillate. This detected oscillation does not decay and also exists at steady state. The oscillation is identified as a Type 2 density wave oscillation according to [25].

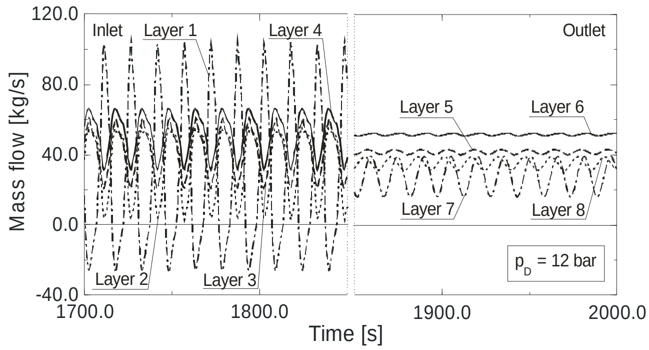

Fig. 6: Mass flow at the inlet of the layers 1 to 4 and at the outlet of the layers 6 to 8

The mass flow at the inlet of the layers 1 to 4 and at the outlet of the layers 6 to 8 is shown in Fig. 6. It can be seen, that the phase displacement of the oscillation in the four tubes paths of the evaporator is shifted in time. The oscillation amplitude at the inlet of the first four layers is higher than at to the outlet of the four lower heated layers. Therefore the oscillation is damped during its way through the tube paths from e. g. the inlet of layer 1 (approximately 130 kg/s at steady state) to the outlet of the associated layer 7 (approximately 23 kg/s at steady state). This behaviour of the DWO results in a low influence on the circulation mass flow in the downcomer and riser as well as on the steam rate (see Fig. 5).

The amplitudes of the DWO at the inlet of the four tube layers are very high, therefore the reverse flow can occur for a short time period during oscillation (see Fig. 6 the negative mass flow).

5.1 Influence of the operation pressure

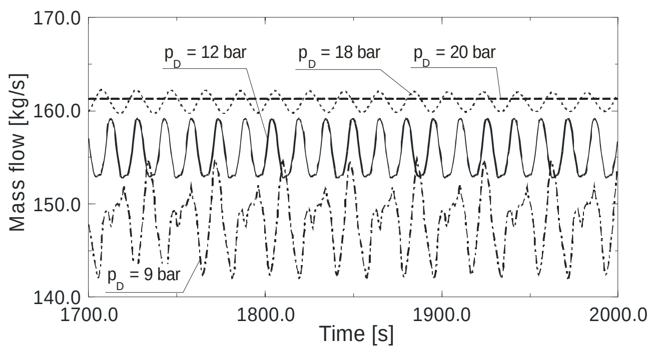

Fig. 7: Mass flow in the downcomer at different drum pressures

The influence of the operation pressure on the DWO can be seen in Fig. 7. The simulations are done without any changes in the geometry of the boiler.

The different curves represent the mass flow in the downcomer of the evaporator at full load and different drum pressure. An increase of the drum pressure at constant heat flux results in an increase of the circulation mass flow and consequently in an enhanced overall circulation ratio U_D (U_D = 13.1 at 9 bar and U_D = 15.7 at 20 bar).

As mentioned above the oscillation of the DWO is characterized by a nearly sinusoidal period, which can be seen in Fig. 7 for a drum pressure of 12 and 18 bars. With decreasing system pressure at constant heat input and geometry the DWO loses the sinusoidal form. This is a result of the interaction of the fluid flow in the upper header between the four tube paths. The mass flow in the single tubes of the four tube paths oscillate nearly sinusoidal but with a phase displacement (see the steam quality in Fig. 8, which is a result of the different heat fluxes to the individual tube paths).

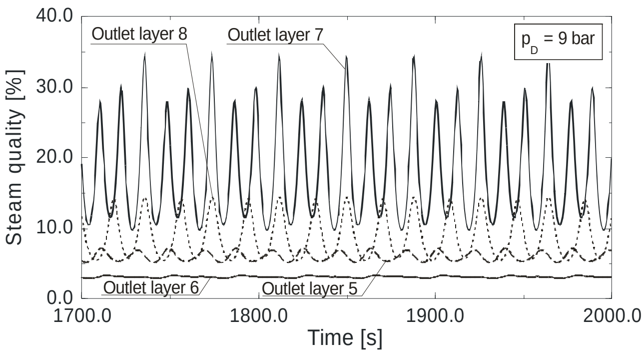

Fig. 8: Steam quality at the outlet of the four tube paths at a drum pressure of 9 bars

With decreasing system pressure at given heat input the steam quality at the tube outlet increases, which results in a higher two-phase pressure drop and mass flow amplitude.

On the basis of the unbalanced heat absorption of the four tube paths the steam quality at the tube outlet is different. This results also in a different two-phase pressure drop in the four tube paths of the evaporator. But the total pressure difference between the upper and lower header must be the same for the single tube paths and therefore a different mass flow inside the tube paths will be the consequence. The superposition of the four mass flows in the upper header can result in a non-sinusoidal period of the DWO.

The variation of the operation pressure has shown that a system pressure of 18 bars works very close to the stability boundary of the analysed system. The oscillation amplitude of the DWO decreases very slowly (see Fig. 9). Approximately 5000 s after the simulation start the oscillation disappears.

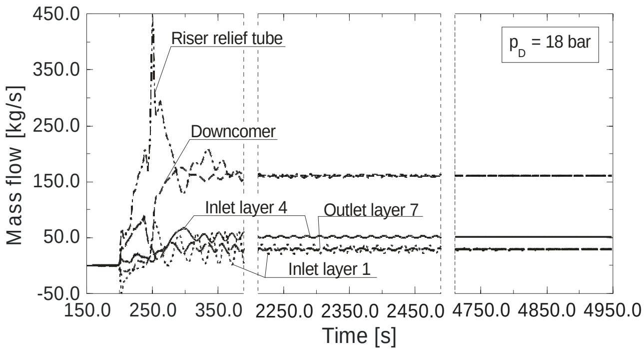

Fig. 9: Mass flow in selected tubes of the HRSG at a drum pressure of 18 bars

With an increase in system pressure at constant heat input, the steam void fraction at evaporator outlet decreases. The lower void fraction generation leads to a decrease of the two-phase friction pressure drop. A damping of the DWO is the consequence of the raised system pressure, which is seen in the decrease of the oscillation amplitude.

Tab. 1: Phase change number and mass flow for the different tube paths at full load and 18 bars

Tube layer	Phase change number [-]	Mass flow [kg/s]
1	4.88	30.006
2	3.84	36.852
3	3.85	42.888
4	3.05	51.297

The dimensionless subcooling number N_{sub} is only a function of the thermodynamic variables of state and can be formed independently from the number of tubes. For a drum pressure of 18 bars the N_{sub} for the HRSG evaporator is 0.252. In comparison to the subcooling number the phase change number is also depending on the fluid velocity at tube inlet and the heat absorption. Therefore N_{pch} must be calculated for every single evaporator tube bank layer.

With the aid of

$$\overline{N}_{pch} = \frac{\sum_{i=1}^{4} N_{pch,i}\dot{m}_i}{\sum_{i=1}^{4} \dot{m}_i} \qquad (10)$$

an averaged phase change number for the tube bundle of the HRSG evaporator was calculated and given with 3.78. Based on these two dimensionless numbers the oscillation was identified as a Type 2 DWO according to [25].

5.2 Influence of the flow resistance at tube inlet and tube outlet

In this paragraph the results of test case 2 and 3 will be summarized.

The thermo-hydraulic behaviour is depending on the flow resistance in the individual sections of the boiler. It is well known that additional flow resistance can improve the stability of heated circulation systems (see e. g. [2], [11], [22]). In this part we should investigate the effect of higher flow resistance in the evaporator tubes by changing their diameter at connection to upper header (test case 2) and the arrangement of orifices at the inlet of the single tubes (lower header; test case 3). The additional flow resistance by the orifice is defined to

$$\Delta p = \xi_{ad} \frac{\rho w^2}{2} . \qquad (11)$$

In test case 2 the outer tube diameter of the connecting tubes between the upper header and the finned layer tubes of the layers 5 to 8 was changed from 48.3 mm to 44.5 mm. The increase of the flow resistance by smaller tube diameter leads to an increase of the oscillation amplitude. The absolute values of the amplitudes at the different layer inlets (1 to 4) are higher as compared to test case 1. The analysis of this test case has shown that the mass flow of the working fluid at the tube inlet of layer 1 to 4 changes between forward and reverse flow due to the oscillation.

The additional flow resistance at the outlet of the tube increases the two-phase pressure drop, which is not in phase with the inlet flow. This leads to a more unstable behaviour of the boiler.

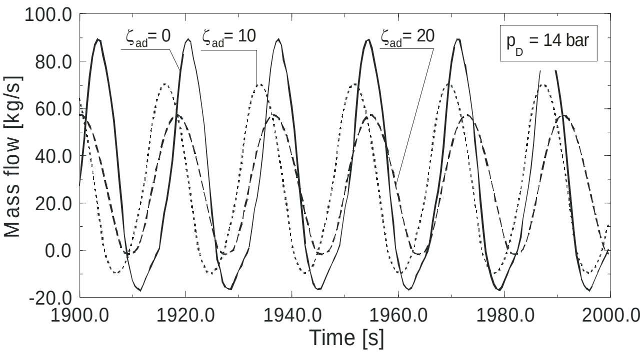

Fig. 10: Mass flow at the inlet of layer 1 at different additional flow resistance

The results for the start-up simulations of the HRSG with additional flow resistance implemented at the inlet of the four tube paths (layer 1 to 4) are summarized in Fig. 10. The simulations are done for a drum pressure of 14 bars and an additional flow resistance ξ_{ad} of 0, 10 and 20. In Fig. 10 the mass flow of the working medium at the inlet of the most heated layer 1 is shown.

It can be seen, that the implementation of the orifice at the tube inlet reduces the amplitude of the DWO. The additional flow resistance at the inlet of the tube increases the single phase pressure drop, which is in phase with the mass flow at the tube inlet. Therefore the stability of the boiler will be improved.

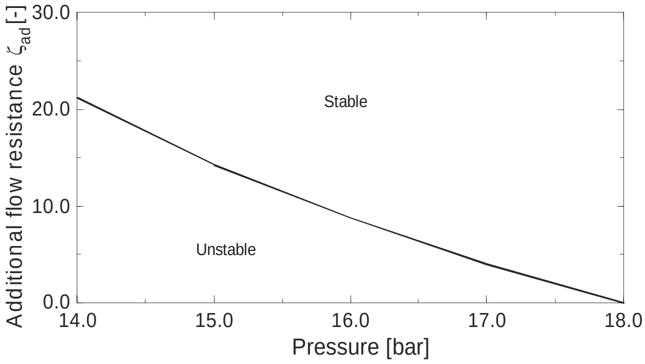

Fig. 11: Stability map for the additional flow resistance at the tube path inlet

Figure 11 represents a stability map for the investigated HRSG boiler. At the horizontal axis the drum pressure and at the vertical axis the additional flow resistance is outlined. The curve in Fig. 11 shows the boundary between a possible stable or unstable circulation of the working fluid. With decreasing drum pressure a higher flow resistance is necessary to get a stable circulation of the mass flow in the evaporator of the HRSG boiler.

The very high values for the additional flow resistance ξ_{ad} at low operation pressure are only theoretical values, because such orifices cannot be realized in practice. Therefore this method to damp the DWO is useable only within certain limits.

In Table 2 the corresponding subcooling numbers and the averaged phase change numbers for the stability boundary presented in Fig.11, are listed.

Tab. 2: Averaged phase change and subcooling number at the stability boundary of the HRSG

Drum pressure [bar]	Averaged phase change number [-]	Subcooling number [-]
14	3.034	0.4117
15	3.228	0.3602
16	3.415	0.3175
17	3.597	0.2823
18	3.784	0.2520

A comparison of the test cases 2 and 3 shows, that an increase of the flow resistance at the inlet of the tubes improves the flow stability, while a flow restriction at the exit of the tubes results in higher instability. The higher flow resistance at the tube inlet increases the single phase flow resistance which is in phase with the flow at the tube inlet. In the opposite to this fact the higher flow resistance at the tube outlet increases the two-phase flow resistance at the tube outlet, which is not in phase with the change of the mass flow at the tube inlet. The consequence is an amplification of the oscillation by additional frictional pressure loss at the tube outlet.

6 Conclusion

In this article the results of a theoretical stability analysis are presented for a vertical type natural circulation HRSG. The evaporator of the HRSG consists of four parallel horizontal tube paths which are connected at both ends with headers.

The variation of the system pressure has clearly demonstrated its important influence to the boiler stability. With decreasing system pressure the boiler tends to a more unstable behaviour.

To improve the stability the flow resistance at the inlet of all tubes (flow restriction, e. g. orifice) of the bundle heating surface can be increased within certain limits. An increase of the flow resistance at the tube outlet results in a more unstable behaviour of the boiler.

7 Nomenclature

A	Cross section area [m^2]
g_x	Component of the gravity in direction of the tube axis [m/s^2]
h	Spec. enthalpy [J/kg]
h_C	Spec. enthalpy of the fluid inside the header [J/kg]
h_f	Spec. enthalpy at saturation line [J/kg]
h_{in}	Spec. enthalpy of the fluid at channel inlet [J/kg]
$\dot{m}$	Mass flow [kg/s]
N_{pch}	Phase change number [-]
$\overline{N}_{pch}$	Averaged phase change number [-]
N_{sub}	Subcooling number [-]
p	Pressure [Pa]
p_C	Pressure of the fluid inside the header [Pa]
p_D	Drum pressure [bar]
Δp	Pressure difference [Pa]
$\dot{q}$	Heat flux [W/m^2]
$\dot{Q}$	Heat flow to the channel [W]
r	Heat of evaporation [J/kg]
t	Time [s]
U	Perimeter [m]
U_D	Overall circulation ratio [-]
V_C	Volume of the fluid inside the header [m^3]
w	Fluid velocity [m/s]
w_{in}	Fluid velocity at channel inlet [m/s]
x	Length [m]
x_D	Steam quality [-]
ρ	Density [kg/m^3]
ρ_C	Density of the fluid inside the header [kg/m^3]
ρ_f	Liquid density [kg/m^3]
ρ_g	Gas density [kg/m^3]
ζ	Pressure loss coefficient [-]
ξ_{ad}	Additional flow resistance [-]

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