Stability analysis of natural circulation systems

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Abstract: In natural circulation systems with unequally heated risers which are connected by a header the static instability, namely reverse flow, can occur. This paper presents different possibilities to determine the mass flow distribution in the tube network of such systems. To calculate the mass flow distribution a graphical and the numerical analysis (stationary and dynamical) was used. The investigation was done for a two-pass boiler under variable operating pressures and heat loads. The dependence of the critical heat absorption ratio $V_{crit}$ from the pressure is discussed. A lower boundary for the part load operation of the analysed natural circulation steam generator was determined. Modifications of the boiler geometry were analysed and the calculation results were presented and discussed.

Key-Words: Natural circulation, Reverse flow, Stability analysis, Steam generator, Two-pass boiler

1 Introduction

Natural circulation steam generators have many advantages compared to forced- and once-through circulation boilers. The main advantage of the natural circulation system is the simple design which results in an unproblematic operation. The part load operation of such a boiler is possible without special auxiliary equipment. Compared with the forced circulation boilers the power for the feed water pump is lower.

The steam quality of the water/steam mixture in the evaporator tubes of natural circulation steam generators is in the normal operation case not higher than approximately 30%. Therefore film boiling respectively critical heat flux can only occur under unfavorable conditions.

Steam generators with natural circulation consist of a more or less complicated tube network. To avoid instable circulation certain design criteria must be taken into consideration. Unstable conditions can be the result of static [1]-[4] (e. g. reverses flow) and/or dynamic instabilities [5]-[7] (e. g. density wave oscillation).

In this paper the static instability, namely the reverse flow, will be illustrated. Flow reversal can occur in systems with unequal heated tubes which are connected by a header (see Fig. 1). The investigation to identify reverse flow in a natural circulation system can be done with the aid of a graphical and a numerical analysis.

2 Investigated boiler

Figure 1 shows a sketch of the investigated two-pass natural circulation steam generator. The higher heated riser system 2 corresponds to the furnace and the lower heated riser system 1 represents the tubes of the wall in the second pass. The large diameter downcomer is unheated.

In the present study two different boiler designs are analysed:

1. The tubes of both riser systems are connected with the drum (design case 1). No intermediate headers and riser relief tubes (displayed with broken lines in Fig. 1) are implemented in this case.
2. The unheated part of both riser systems is replaced by an intermediate header and riser relief tubes (design case 2). The ratio of the cross section area of the riser tubes to the relief tubes is 3.05.


3 Graphical Analysis

The graphical analysis is used to study the mass flow distribution under steady state conditions. For the determination of the mass flow rate in the individual tubes of the boiler it is necessary to combine the different tube characteristics. The characteristic curve for a tube can be obtained by plotting the total pressure difference between the drum and the lower header $\Delta p_{DH}$ against the mass flow rate (see Fig. 2). The total pressure difference $\Delta p_{DH}$ can be calculated by

$$\Delta p_{DH} = \Delta p_{Fric} + \Delta p_{H} + \Delta p_{Accel}. \tag{1}$$

$\Delta p_{Fric}$ includes the pressure loss to friction as well as the pressure loss to e. g. in- and outflow and bends. The pressure difference due to acceleration $\Delta p_{Accel}$ is identical to zero in case of an unheated tube. $\Delta p_{H}$ describes the pressure difference of the static head.

For the graphical analysis the total pressure difference $\Delta p_{DH}$ for every individual tube must be equal (equation (2)) and also the mass balance for the lower header (equation (3)) must be fulfilled.

$$\Delta p_{DH1} = \Delta p_{DH2} = \Delta p_{DH3} \tag{2}$$

$$\dot{m}_h = |\dot{m}_1| + \dot{m}_2 \tag{3}$$

The negative sign in equation (3) is used if the mass flow in the lower heated tube is directed downward. The counterclockwise flow direction is counted positive.

Figure 2 should help to explain the procedure for the graphical determination of the mass flow distribution in the tubes of the boiler design 1. At the horizontal axis of Fig. 2 the mass flow and at the vertical axis $\Delta p_{DH}$ is outlined.

The characteristic curve for the downcomer is illustrated in Fig. 2 by the full line and that for the riser system 1 and 2 by the dotted lines. Dependent on the flow direction of riser system 1 the mass flow for the unequally heated riser systems must be added or subtracted at constant pressure difference $\Delta p_{DH}$. In case of reverse flow in the lower heated riser system the characteristic curve for $\dot{m}_1 < 0$ must be subtracted (see arrows in Fig. 2) from the characteristic $\dot{m}_2 > 0$. This results in a circulation characteristic (broken lines) dependent on the flow direction of riser system 1.

It can be seen that the circulation characteristic for upward flow direction in both riser systems (signed with $\dot{m}_1 > 0$ and $\dot{m}_2 > 0$) has only one intersection point with the characteristic of the downcomer (point A). In this intersection point equation (2) and (3) are fulfilled. The associated circulation mass flow $\dot{m}_1$, $\dot{m}_1$ and $\dot{m}_2$ as well as $\Delta p_{DH}$ can be read from the axis. In case of reverse flow in riser system 1 the circulation characteristic (signed with $\dot{m}_1 < 0$ and $\dot{m}_2 > 0$) has two intersection points with the graph for the downcomer (point B and C). Only solution B is stable.

Higher heat absorption at the lower heated system 1 moves the circulation characteristic for reverse flow in direction of lower mass flow rate and $\Delta p_{DH}$. This leads to the consequence that the solutions B and C move together. The point where B and C are merged represents the limit of univocal flow direction in the riser system with the lower heat absorption. For further increase of the heat flow $\dot{q}_1$ the downcomer characteristic does not cross the graph for the circulation characteristic with the mass flow $\dot{m}_1 < 0$ and $\dot{m}_2 > 0$. In this case only upward flow is possible in both riser tubes.

The graphical solution is very illustrative but also very time-consuming. Therefore a numerical simulation is preferable to find design criteria to prevent reverse flow. In the following paragraphs the results of such numerical analysis (stationary and dynamical) will be presented.

4 Stationary Analysis

The evaluation of the stationary behavior of steam generators is a standard procedure for the design of heating surfaces. Many computer programs e. g. NOWA [8] were developed in the past to calculate the stationary operation conditions of steam generators. The data for the geometry (e.g. tube diameter and lengths, wall thickness, etc.), the system pressure as well as the heat absorption profile for the heated tubes are necessary for the input of such a code. The temperature-, density-, pressure- or mass flow distributions in the tube network of the boiler are the result of such a calculation.
Figure 3 shows as a result of a static stability analysis the mass flow distribution in the tubes of the steam generator. The calculations are done for the boiler design shown in Fig. 1 without the riser relief tubes and the intermediate headers (design case 1). The heat flux for riser system 2 with \( q_2 = 320 \text{ kW/m}^2 \) was kept constant while the heat flux for \( \dot{q}_1 \) decreases. A negative sign in Fig. 3 describes a flow direction from the drum to the lower header.

Up to a heat absorption ratio of \( V = \dot{q}_2 / \dot{q}_1 = 7 \) only one solution (A) with upward flow direction in both riser systems can be found. With a heat absorption ratio higher than 7 two stable (A and B) and one unstable (C) solutions are possible for the circulation mass flow. The solution C in Fig. 3 was found with the help of the graphical method presented in the paragraph above. The heat absorption ratio of 7 represents the critical value \( V_{\text{crit}} \) between a stable upward flow in both riser systems and a possible operation condition with reverse flow in the lower heated riser system 1.

A complete solution for the mass flow in the lower heated riser system 1 \( \dot{m}_1 \) for all possible heat absorption ratios is presented in Fig. 4. The Fig. shows that reverse flow is possible for a heat absorption ratio \( V > 7 \) (point C) at the constant heat flux for riser system 2 with \( \dot{q}_2 = 320 \text{ kW/m}^2 \). The critical heat absorption ratio \( V_{\text{crit}} \) decreases with lower total heat flux. This can be seen for the curve for \( \dot{q}_2 = 75 \text{ kW/m}^2 \) (point D).

The static analysis has shown that both stable solutions (A and B) for the mass flow distribution can be found. But a static analysis can only find the solution under stationary conditions. As shown in [2] the correct static results alone are not a guaranty for a safe operation of a natural circulation system with unequally heated risers under all conditions. Therefore the prediction of the transient behavior of the boiler e. g. under hot start-up conditions or heavy load changes must be done by a dynamic analysis.

5 Dynamic Analysis

The one-dimensional mathematical model for the flow in the tubes, which was used for all calculations in this paragraph, includes a homogeneous equilibrium model for the two phase flow and applies a correction factor for the two phase pressure loss according to Friedel [10]. The collectors and distributors respectively are assumed to be points. The steam, which enters the header, is evenly distributed to the outlet tubes. A detailed description of the computer code is given in [11].

5.1 Boundary and initial conditions

The following initial conditions are used for the dynamic simulations:

- the steam generators are filled with water near boiling condition
- the pressure distribution of the fluid is due to gravity and 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 of the steam generator is constant during the whole simulation. The heat fluxes to the two riser systems of the two-pass boiler are boundary conditions for the simulation. The water level was controlled at the drum centerline.

The system pressure will be varied in steps between 80 bars and 12 bars (80 bar, 65 bar, 50 bar, 40 bar, 30 bar 20 bar and 12 bar). The heating ramp for the simulated hot start-up was selected according to [9] and is shown in Fig. 5. It can be seen that in the first period of the simulation the heat flow increases up to 30 % of the full load. Between 30 s and 150 s after the ignition of the burner the load is constant. During the time period between 150 s and 400 s the ramp-like heat input reaches the full load. The total time for the simulation is 10000 s.

The total heat absorption (i.e. the heat absorption for both riser systems) will be varied in steps between 8 MW and 0.8 MW (8 MW, 6.4 MW, 4.8 MW, 3.2 MW and 1.6 MW). The value of 8 MW was chosen as upper extreme. For industrial furnaces with oil firing the corresponding heat absorption can be assumed between 5 MW - 6 MW. The value of 0.8 MW can represent the condition for low load operation or special equipment in process engineering (e. g. refuse incineration plant).

5.2 Results of the dynamic analysis

Figure 6 shows as a result of the dynamic simulation the mass flow in the different tubes of the boiler. The simulation refers to the boiler design without the intermediate headers and riser relief tubes at 80 bars, a total heat flow of 4.8 MW and a heat absorption ratio of $V = 9$ (full line) and $V = 9.5$ (broken line).

In the following the results for the heat absorption ratio of $V = 9$ will be described in detail:

It can be seen that the lower heated riser system 1 starts in the downward direction with a relatively high mass flow rate. The higher heated riser system 2 starts in the upward direction with approximately the double mass flow rate as the riser system 1. With the beginning of the steam production in the lower heated riser system 1, the mass flow starts to slow down. Approximately 500 seconds after the start of the simulation the flow direction of the riser system 1 changes rapidly to upward flow. Following the change of the flow direction, the mass flow tends after a short peak to a much lower mass flow. After about 600 seconds the mass flows in the downcomer and both riser systems have achieved the steady state values.

The situation for the circulation mass flow after a moderate change of the heat absorption ratio from $V = 9$ to $V = 9.5$ is different. The mass flow in the tubes of the boiler at $V = 9.5$ starts similar to that shown for a heat absorption ratio of $V = 9$. The mass flow of the working fluid in riser system 1 starts in the downward direction too. With the beginning of the steam production in the lower heated riser system 1, the mass flow starts to slow down. But, compared to the case with $V = 9$ the heat flux to the riser system 1 is not high enough to change the flow direction. Therefore the flow direction remains different in both riser systems at steady state.

Figure 7 shows the pressure difference of the static head $\Delta p_H$ for both riser systems as well as the total pressure difference between the drum and the lower header $\Delta p_{DH}$. 
At first the results for the heat absorption ratio of $V = 9$ will be described:

During the first phase of the hot start-up $\Delta p_{DH}$ is smaller than the pressure difference of the static head and therefore the mass flow in the lower heated riser system 1 is directed downward. With the beginning of the steam production $\Delta p_H$ starts to decrease. Between 480 and 520 seconds after the start of the simulation the increasing steam production in the riser system 1 reduces the static head $\Delta p_H$ and crosses the graph of the pressure difference $\Delta p_{DH}$. In this period, the mass flow changes from downward to upward direction.

The pressure difference due to height $\Delta p_H$ to the case of a heat absorption ratio of $V = 9.5$ is always higher than $\Delta p_{DH}$ (see Fig. 7). Therefore the mass flow is always directed downward in the lower heated riser system 1 also at steady state.

Figure 8 shows the critical heat absorption ratio $V_{crit}$ for the boiler design 1 at different system pressures and heat loads.

A heat absorption ratio of $V_{crit} \geq 5$ is assumed to be a design criterion for the two-pass boiler for stable operation at all heat loads. A curve with the constant heat absorption ratio of $V_{crit} = 5$ (broken line) is included in Fig. 8. It can be seen that the system tends to flow reversal for low heat loads even at 80 bars.

Beside the overall circulation ratio the fluid velocities are also of important interest for the boiler design. In a certain temperature (pressure) range high velocities can be the reason for erosion-corrosion ([13] [14]).

Figure 9 shows the velocity in the higher heated tubes. It can be seen that the velocity corresponds over a wide range of pressures and heat loads with the usual design rules (The velocity of the steam-water mixture should be approximately $\leq 12 \text{ m/s}$ and the velocity in the downcomer $\leq 4\text{ m/s}$ [12]). A curve with the constant velocity of 12 m/s is included in Fig. 9.

The analysis of the basic design has shown that the boiler should not operate in the pressure range < 30 bars and in the very low load region, even at 80 bars. Therefore modifications on the system geometry are necessary. One of the most successful methods to get a higher stability is to modify the thermo-hydraulic behaviour of the circulation system. This can be done by reducing the pressure loss in the downcomer or increasing the pressure loss in the riser systems [4], [12].

The investigation on the design case 1 has also shown that the mass flow in the higher heated tubes is higher than necessary for cooling the tubes. Therefore a redesigning of the unheated section of the riser systems should be done with the aim to reduce the mass flow in the higher heated riser system. This can be done by replacing the unheated part of both riser systems by an intermediate header and riser relief tubes (see Fig. 1) [4], [12].

In the following the results for design case 2 will be presented.

The investigation for this test case is subdivided into two parts:

1. The number of unheated riser relief tubes for the lower heated system 1 is two and for the higher heated system 2 is four.
2. The number of unheated riser relief tubes for the lower heated system 1 is four and for the higher heated system 2 is two.

At first the design case with the four riser relief tubes implemented in the higher heated riser system 2 will be described.

Figure 10 shows the critical heat absorption ratios $V_{crit}$ for the first analysed configuration of design case 2. The modifications at the unheated part of the riser systems has moved the lower limit for stable upward flow of $V_{crit} = 5$ in direction to the lower heat loads. Therefore no operation restriction at lower heat loads is given for the pressure range down to approximately 60 bars.
The simulation result for the velocity in the riser relief tubes of the higher heated riser system 2 is shown in Fig. 11. It can be seen that the curve with the constant velocity of 12 m/s is shifted in direction of higher pressures and lower heat loads. A detailed analysis of the data in the pressure region between 30 bars and 50 bars and higher heat loads has shown that the velocities are close to 12 m/s. Therefore a safe operation is also given for this boiler design down to 30 bars.

The installation of four riser relief tubes in the lower heated system 1 increases the critical heat absorption ratio especially in the high pressure/high heat load region (see Fig. 12). The limit for stable upward flow of $V_{crit} = 5$ is also moved to lower heat loads.

It can be seen that this boiler design has achieved the highest values for $V_{crit}$ from all analysed configurations. A comparison of the data has shown that the pressure in the lower header for the design case 2 with four riser relief tubes at the lower heated riser system 1 was higher than in the other cases while the pressure in the lower header of the basic design was the lowest. The higher pressure in the lower header moves the critical heat absorption ratio to higher values. This was demonstrated also in [12].

On the basis of this result the current design case can be used only in a very restricted operation range. Therefore a further redesign (e.g. larger dimension for the riser relief tubes and intermediate headers) of the boiler is necessary.

### 6 Conclusion

A stability analysis for a natural circulation system with two unequally heated riser systems which are connected by a header with an unheated downcomer was performed. For the determination of the static instability, namely reverse flow, different methods have been presented. The investigation was done for different operation pressures and heat loads.
The study has shown that for every system pressure, heat load and boiler configuration a special critical heat absorption ratio $V_{crit}$ exists. This critical heat absorption ratio must be determined for every configuration of such a boiler. The critical heat absorption ratio depends also on the thermo-hydraulic behaviour of the circulation system. Heating conditions which exceed this critical value can lead to unstable conditions including the possibility of flow stagnation and or reverse flow.

The analysis of the data has shown that all modifications which lead to a higher pressure in the lower header results in a higher stability.

In boiler configurations with unequally heated tube systems the number of riser relief tubes which are connected with the lower heated tubes should be smaller or equal to the number of riser relief tubes which are connected with the higher heated tubes. Otherwise the velocity in the relief tubes can exceed the upper limit.

## 7 Nomenclature

- $m_0$: Mass flow in the downcomer [kg/s]
- $m_1$: Mass flow in riser system 1 [kg/s]
- $m_2$: Mass flow in riser system 2 [kg/s]
- $\dot{q}_1$: Heat flow to riser system 1 [kW/m$^2$]
- $\dot{q}_2$: Heat flow to riser system 2 [kW/m$^2$]
- $V$: Heat absorption ratio [-]
- $V_{crit}$: Critical heat absorption ratio [-]
- $\Delta p_{Accel}$: Pressure difference due to acceleration [bar]
- $\Delta p_{DH}$: Total pressure difference between the drum and the lower header [bar]
- $\Delta p_{Fric}$: Pressure difference due to friction [bar]
- $\Delta p_{H}$: Pressure difference of the static head [bar]

## References:


