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Diploma thesis

Advanced Tube Rupture Pressure Evolution Calculation and Examination of Preventive Measures

carried out for the purpose of obtaining the degree of Master of Science (MSc or Dipl.-Ing. or DI), submitted at TU Wien, Faculty of Mechanical and Industrial Engineering, by

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Vienna, March 2025

I confirm, that going to press of this thesis needs the confirmation of the examination committee.

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I declare in lieu of oath, that I wrote this thesis and performed the associated research myself, using only literature cited in this volume. If text passages from sources are used literally, they are marked as such.

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English Abstract

Steam generators provide steam for power- and heat generation as well as for the process technology. For bigger capacities, water tube boilers are necessary. For ensuring safe operation, all possible damages and their causes must be considered. The rupture of a water tube – with water-steam mixture flowing into the combustion chamber, followed by an explosion-like pressure increase resulting in a burst of the boiler – has faced a problem at existing plants. Several occurs not just caused whole plants to be shut down and bigger repairments to be done but also demanded human loss. Explosion flaps or burst disks must be installed as counter measure for avoiding the pressure inside the boiler to reach critical values.

In a previous released bachelor's thesis, a simplified model was programmed, showing the functionality of a tube rupture simulation. The present work presents a more detailed analysis of the topic, also considering the water side of the problem: A numerical model of an existing water tube boiler has been created, programmed with MATLAB. A simulation of a tube rupture shows the evolution of pressure in the different parts of the flue gas channel and the effects on the water tube side. An explosion flap has been considered. Different dimensions of the flap and positions within the flue gas channel have been tested. Simulations show that the explosion flap with a cross section of 1-2 m², positioned near the membrane wall result in a faster decrease of pressure. This numeric model can be used for counterchecking when designing a steam generator.

Deutsche Kurzfassung

Dampferzeuger stellen Dampf für die Energie- und Wärmeerzeugung, sowie die Prozesstechnik zur Verfügung. Bei größeren Kapazitäten sind Wasserrohrkessel im Einsatz. Um einen sicheren Betrieb zu gewährleisten, muss jeder mögliche Schaden und dessen Ursache bedacht werden. Das Reißen eines Wasserrohres – wobei ein Wasser-Dampf-Gemisch in die Brennkammer strömt, einen explosionsartigen Druckanstieg hervorruft und ein Zerbersten des Kessels verursacht – stellt ein Problem für bestehende Anlagen dar. Mehrere Vorkommnisse haben nicht nur ein Abschalten der Anlage und große Reparaturen verursacht, sondern auch Menschenleben gekostet. Gegenmaßnahmen, wie Explosionsklappen oder Berstscheiben müssen installiert werden, um einen Anstieg des Druckes im Kessel überhalb kritischer Werte zu verhindern.

In einer zuvor veröffentlichten Bachelorarbeit wurde ein vereinfachtes Model, welches die Funktionalität einer Simulation eines Rohrreißers aufzeigen soll, programmiert. Die vorliegende Arbeit stellt eine detailliertere Untersuchung des Themas dar, welches auch die Wasserseite des Problems berücksichtigt: Ein numerisches Modell eines existierenden Wasserrohrkessels, erstellt mit der Programmierumgebung MATLAB. Eine Simulation eines Rohrreißers zeigt die Druckentwicklung in den verschiedenen Segmenten des Rauchgaskanals. Eine Explosionsklappe wurde berücksichtigt. Verschiedene Dimensionen der Klappe, sowie Positionen im Rauchgaskanal wurden getestet. Simulationen zeigen, dass eine Explosionsklappe mit einem Querschnitt von 1-2m², angebracht an der Membranwand, den schnellsten Druckabfall verursacht. Dieses numerische Modell kann als Gegenprüfung beim Entwurf eines Dampferzeugers dienen.

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List of Abbreviations

Abbreviation:	Meaning:
DNB	departure of nucleate boiling
ECO	economizer
e.g.	exempli gratia, for example
Eq.	equation
EU	European Union
Fig.	figure
PDE	partial differential equation
sec.	seconds
spec.	specific
TRPEC	tube rupture pressure evolution calculation

List of Symbols

Abbreviation:	Unit:	Meaning:
A	m²	cross section of control volume
A_{CC}	m²	surface inside combustion chamber
$A_{CC,proj}$	m²	projected surface inside combustion chamber
A_{EF}	m²	cross section of explosion flap
A_{th}	m²	narrowest cross section of ruptured tube
A_{Ti}	m²	cross section of tube connection to turbine inlet
a, a_P, a_E, a_W, $a_P{}^0$, a_{11}, b_1, ...	-	coefficients of calculation points for numerical description of heat transfer
a_{TR}	-	exponent of disequilibrium factor
$\boldsymbol{a}$, $\boldsymbol{b}$	-	matrix of coefficients of calculation points for numerical description of heat transfer
b_{FO}	m	width of fouling layer
b_{IN}	m	width of insulation
b_T	m	width of tube wall
C	-	dimensionless mass flow of ruptured tube
c	J/kgK	specific heat capacity
c_p	J/kgK	isobaric specific heat capacity
$c_{P,FG}$	J/kgK	isobaric specific heat capacity of flue gas
c_{pgs}	J/kgK	isobaric specific heat capacity of gas phase at boiling
c_{pi}	J/kgK	isobaric specific heat capacity of control volume
c_{pl0}	J/kgK	isobaric specific heat capacity of liquid phase at nozzle entrance
$c_{P,N2}$, $c_{P,O2}$, $c_{P,CO2}$, $c_{P,H2O}$	J/kgK	isobaric specific heat capacity of flue gas components N2, O2, CO2, H2O
$c_{V,FG}$	J/kgK	isochoric specific heat capacity of flue gas
c_{vi}	J/kgK	isochoric specific heat capacity of control volume
$c_{v,FO}$	J/kgK	isochoric specific heat capacity of fouling layer
$c_{v,Tube}$	J/kgK	isochoric specific heat capacity of tube wall

Abbreviation:	Unit:	Meaning:
d_{Ti}	m	diameter of tube connection to turbine inlet
E	J	energy inside control volume
f	-	factor to change between explicit or implicit or between (semi-parallel)
$\dot{H}_i$	W	enthalpy flow of control volume
h_i	J/K	specific enthalpy of control volume
h_{FW}	J/K	specific enthalpy of feed water
h_S	J/K	specific enthalpy of produced steam
h_{TR}	J/K	specific enthalpy of mass going out of ruptured tube
i	-	count value of calculation node
k	-	factor for ω_{Frozen}
L	m	characteristic length
l_{EF}	m	length of exhaust channel after explosion flap
l_{Ti}	m	length of tube connection to turbine inlet
$\dot{M}$	kg/s	mass flow out of ruptured tube
$\dot{M}_{max}$	kg/s	maximum (critical) mass flow out of ruptured tube
m	kg	mass
$\dot{m}_{EF}$	kg/s	mass flow out of explosion flap
m_{FO}	kg	mass of fouling layer
m_i	kg	mass inside control volume
$\dot{m}_{FW}$	kg/s	feed water mass flow
$\dot{m}_S$	kg/s	produced steam amount
$\dot{m}_{TR}$	kg/s	mass flow out of ruptured tube
m_{Tube}	kg	mass of tube wall (membrane wall)
m_{WS}, $(\dot{m}_{WS})$	kg/s	mass (flow) ratio of water / steam
N	-	disequilibrium factor
n	-	time step
Nu	-	Nusselt number
p_{Am}	Pa	pressure of ambience
$p_{EF,act}$	Pa	activating pressure of explosion flap
p_{evap}	bar (abs)	pressure inside evaporator

Abbreviation:	Unit:	Meaning:
p_{FG}	Pa	pressure of flue gas
p_i	Pa	pressure of control volume
Pr	-	Prandtl number
p_{WS}	Pa	pressure of water / steam
Δp	Pa	pressure loss in flue gas channel
p_0	Pa	pressure at nozzle entrance
$\dot{Q}$	W	heat flow towards control volume
$\dot{Q}_{FG \rightarrow FO}$	W	heat flow from flue gas to fouling layer
$\dot{Q}_{FG \rightarrow FO,conv}$	W	heat flow from flue gas to fouling layer, convective part
$\dot{Q}_{FG \rightarrow FO,rad}$	W	heat flow from flue gas to fouling layer, radiant part
$\dot{Q}_{FO \rightarrow Ti}$	W	heat flow from fouling layer to tube surface, inside
$\dot{Q}_{IN \rightarrow AM}$	W	heat flow from insulation to ambience
$\dot{Q}_{Ti \rightarrow WS}$	W	heat flow from tube surface, inside to water / steam mixture
$\dot{Q}_{To \rightarrow IN}$	W	heat flow from tube surface, outside to insulation
$\dot{Q}_{WS \rightarrow To}$	W	heat flow from water / steam mixture to tube surface, outside
R	J/kgK	specific gas constant of whet flue gas
Re	-	Reynolds number
R_i	J/kgK	specific gas constant inside control volume
R_I	W/m²K	heat conductance for sum of all heat transfer coefficients of fouling and tube, outside
R_o	W/m²K	heat conductance for sum of all heat transfer coefficients of tube, inside and insulation
R_{FG}	J/kgK	specific gas constant of dry flue gas
R_{WS}	J/kgK	specific gas constant of water / steam
$S_p \Delta x$	-	well term
$\boldsymbol{T}$	K	temperature matrix
T_i	K	temperature of control volume
T_{AM}	K	temperature of ambience
T_{FG}	K	temperature of flue gas
T_{FO}	K	temperature of fouling layer
T_{IN}	K	temperature of insulation

Abbreviation:	Unit:	Meaning:
T_S	K	boiling temperature
T_{Ti}	K	temperature of tube surface, inside
T_{To}	K	temperature of tube surface, outside
T_{WS}	K	temperature of water / steam
T_W, T_P, T_E	K	temperatures of calculation points for numerical description of heat transfer
T_0	K	temperature at nozzle entrance
t	sec.	time
Δt	sec./sec.	time step width
t_{EF}	sec.	time in which explosion flap is fully opened
U	-	circulation number
U_{crit}	-	critical circulation number
U_i	J	inner energy of control volume
u_i	J/kg	specific inner energy of control volume
u_{WS}	J/kg	specific inner energy of water / steam
V_i	m³	volume of control volume
v	m/s	speed of flue gas flow
v_{FG}	m/s	speed of flue gas flow through explosion flap
v	m³/kg	specific volume
v_{l0}	m³/kg	specific volume of liquid at nozzle entrance
v_{gs}	m³/kg	specific volume of gas at boiling curve
v_{g0}	m³/kg	specific volume of gas at nozzle entrance
v_0	m³/kg	specific volume at nozzle entrance
v_i', v_i''	m³/kg	specific volume of water and steam inside control volume
v_S	m³/kg	specific volume of steam
v_W	m³/kg	specific volume of water
V_{WS}	m³	volume of water / steam
v_{WS}	m³/kg	specific volume of water / steam
x	kg/kg	amount of steam
Δx	-	size of calculation point for numerical description of heat transfer

Abbreviation:	Unit:	Meaning:
$\dot{x}_{eq,th}$	-	vapour mass flow fraction at narrowest cross section at thermodynamic equilibrium
$\dot{x}_0$	-	vapour mass flow fraction at entrance
$\alpha_{FG \rightarrow FO}$	W/m²K	heat transfer coefficient for forced convection, from flue gas to fouling layer
$\alpha_{FG,rad}$	W/m²K	heat transfer coefficient for radiation, from flue gas to fouling layer
$\alpha_{FG,rad,current}$	W/m²K	value of new derived heat transfer coefficient for radiation, from flue gas to fouling layer
$\alpha_{FG,rad,old}$	W/m²K	value of preceding heat transfer coefficient for radiation, from flue gas to fouling layer
$\alpha_{IN \rightarrow AM}$	W/m²K	heat transfer coefficient for natural circulation, from insulation to ambience
$\alpha_{Ti \rightarrow WS}$	W/m²K	heat transfer coefficient for forced convection, from tube, inside to water / steam mixture
$\alpha_{WS \rightarrow To}$	W/m²K	heat transfer coefficient for forced convection, from water / steam mixture to tube, outside
Γ	-	factor for exponent of disequilibrium factor
$\Gamma_1, \Gamma_2\ \Gamma_3$	-	factors for η_{crit}
ε	-	emission number
ζ	-	friction coefficient
ζ_{EF}	-	friction coefficient of explosion flap (when fully opened)
η_b	-	general pressure ratio
η_{crit}	-	critical pressure ratio
η_g	-	pressure ratio of gas phase
$\eta_{g,crit}$	-	critical pressure ratio of gas phase
η_s	-	subcooling factor
η_{th}	-	pressure ratio of narrowest cross section of nozzle
κ	-	isentropic exponent
λ	W/mK	thermal conductivity
λ_{FO}	W/mK	thermal conductivity of fouling layer

Abbreviation:	Unit:	Meaning:
λ_{IN}	W/mK	thermal conductivity of insulation
λ_T	W/mK	thermal conductivity of tube wall
ν	m²/s	kinematic viscosity of the flue gas
ξ	kg/kg	amount of water inside control volume
ξ_{CO2}	kg/kg	amount of CO2 in flue gas
ξ_{H2O}	kg/kg	amount of H2O in flue gas
ξ_{N2}	kg/kg	amount of N2 in flue gas
ρ	kg/m³	density of flue gas
ρ_{FG}	kg/m³	density of flue gas at explosion flap
ρ_i	kg/m³	density inside control volume
Σ	-	sum formula
σ	W/m²K^4	Stefan-Boltzmann-number
Ω	-	factor for factors Γ_1, Γ_2 Γ_3
ω	-	compressibility factor
ω_{Flash}	-	compressibility due to evaporation of liquid or condensation of gas
ω_{Frozen}	-	compressibility of steam resulting from pressure changes in nozzle

1 Introduction and Motivation

In today's landscape of energy supply and process technology, steam generation is still essential for many applications. Thus, research and development are necessary to assure a perpetual improvement. Safety related topics need to be addressed as well. The case of a ruptured tube of a membrane wall at water tube boilers is an important scenario that needs to be analyzed when designing a boiler. Corresponding counter measures must be dimensioned, and its impact must be cross checked.

The principle of a steam generator is the production of superheated steam by combustion of a fuel which can be oil, gas, coal or any biomass as well as various alternative fuels (bio ethanol or waste liquids). Combustion takes place inside a combustion chamber which is surrounded by a water containing body. Heat transfer is used to warm the water until steam is generated. For most industrial purposes, superheated steam is demanded. There are two major boiler concepts: shell tube boilers and water tube boilers. The first one is designed for producing steam with up to 30 bar, mostly for supplying process steam. The last one – upon which this work is referring to – can work with up to 300 bar, mainly used for generation of electric energy [1] (chapter 1.4). A Clausius Rankine cycle describes a typical thermodynamic cycle of a steam-to-electric energy production: 1) Water is compressed with a pump, 2) evaporated and superheated, then 3) expanded through a steam turbine and 4) cooled down by a condenser.

1.1 Water tube boiler

Heat, coming from a combustion chamber, is transferred to the water by using a water tube heat transformer, by convection and radiation. The location where heat transfer takes place is divided into several zones, corresponding to the different steps of steam formation: preheating at the so-called Economizer, evaporating at the evaporator and superheating at the superheater.

The evaporator is usually placed at the hottest zone of the flue gas side: the furnace. A membrane wall is built out of vertical steel tubes, filled with water, which are welded together with metal strips in between to form a combustion chamber.

Following the furnace, the flue gas is forced by an induced fan, flowing through a channel, in which several heat transfer tube bundles (for superheating and preheating) are passed. The flue gas is then treated as appropriate and finally led into the environment through a stack.

To avoid unnecessary erection heights, the channel can be divided into two or three channels (as shown in Fig. 1).

There are three designs of water tube boilers which differ in their steam output pressure. “Natural circulation” delivers steam at 160 bar, “forced circulation” and a “once-through boiler” at 180 bar and 300 bar respectively, see [1] (chapter 1.4).

At “natural circulation”, feeding water, coming from a water treatment system (mainly for desalination), is preheated by the Economizer and fed into the boiler drum. At the boiler drum liquid water and steam (which has already been generated) are separated through gravitation, so that water gathers at the bottom and steam at the top. The water is flowing downwards through downcomer tubes and into the membrane wall through lower wall-headers. Water is heated and evaporated into unsaturated wet steam. The steam which is separated at the drum is heated up more at the superheaters and led to the turbine. The difference between the density of the saturated water inside the downcomer tubes and the water / steam mixture inside the membrane wall leads to a circulation without an external impulse (pump), giving the name “natural circulation”. The circulation number U which is defined by the ratio of ingoing water mass and outgoing vapour mass, is in a range of 6-14, see [2] (chapter O1 2.5).

When steam with higher pressure is demanded, “forced circulation” is necessary. At higher pressure, the difference of density in the membrane wall and the downcomer tubes becomes too small to assure automotive circulation, therefor a pump is needed to enable a flow. Here circulation numbers up to 40 are possible, see [2] (chapter O1 2.5).

For providing higher pressurized steam, also beyond the critical point (around 221 bar), an overcritical steam generator or “once through boiler” is applied. Inside the heat transfer tube bundles, water is fully evaporated and superheated into overcritical state without circulation. When water is present in overcritical state, there is no difference between liquid or gaseous state.

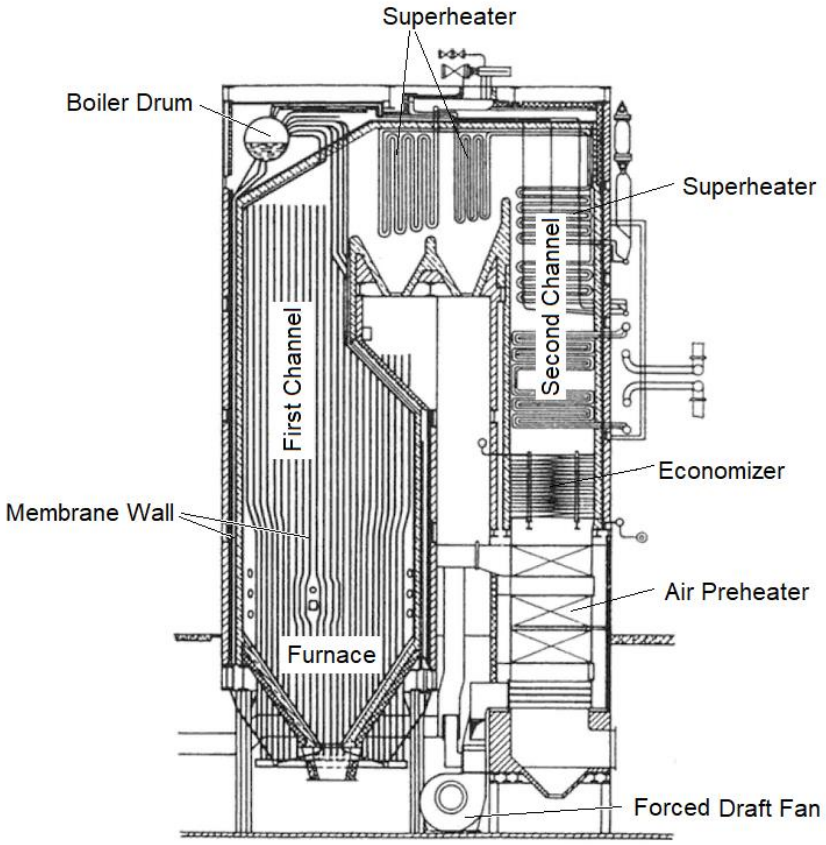

Fig. 1 water tube boiler, two channel-natural circulation-steam generator, pulverized coal firing [3] (chapter 1).

Designing a boiler must be according to the ruling standards and directives of the country the device is distributed in. Pressure equipment with a maximum design pressure of 0.5 bar which is distributed in the EU, obtain to the pressure equipment directive (PED). Also, the EN12952 must be followed. When applying the PED, mechanical loads are considered, corresponding materials are demanded.

At water tube boilers, possible occurrents like overheating, pollution, abrasion or corrosion of heat exchanger tubes must be considered.

Though, all these standards and directives must be followed, damages and accidents can happen. One occurrent is the rupture of a heat exchanger tube. Causes can be an overheating or abrasion of the metal material. If a water / steam tube brakes, water is going out into the hot environment of flue gas. This leads to vaporization of water which causes an increase of pressure, like an explosion. This pressure level can be too high for the membrane wall of the boiler which can lead to a burst of the combustion chamber. At a steam generator in Austria a tube rupture occurred in 2014 which led to a burst of a combustion chamber.

The aim of this work is to have a better view on the topic of a tube rupture at water tube steam generators and deliver a simulation tool for its pressure evolution. Thus, the name Tube Rupture Pressure Evolution Calculation (TRPEC) was given.

2 Basics

2.1 Steam formation and possible damages

Combustion inside the boiler results in a heat flux (as radiation and forced convection) towards the colder heat exchanger tubes. Since, the biggest heat flux in value is transferred at the evaporator, the mechanical stress of the material at this location must be especially considered. Boilers always have a two-phase mixture of liquid and gaseous water inside the membrane wall tubes at the evaporator section.

As water is fed into the rising tubes, steam is generated along the height and a ratio of steam ξ is present. The formation of steam in a vertical tube of an evaporator can have various evolutions. Studies carried out by VDI heat atlas [4] (chapter H3.5) show the following:

There are several regimes of steam formation, as can be seen at Fig. 2. Water in two-phase state has always a higher heat transfer than water or saturated steam alone. The tube temperature must not exceed material limits. Since liquid water has a better heat transfer than steam, the formation of steam at the tube wall is to be avoided. If the heat transfer does not exceed a "critical" value, the tube wall stays wetted by a liquid layer. When the heat flow is too high, two scenarios can occur: "film boiling" (or "departure of nucleate boiling", short DNB) describes the formation of steam directly at the tube surface and therefor a worsening of heat transfer which leads to an overheating of the tube material. When steam formation is that far evolved that the whole cross section of the tube is filled with steam (ξ=1), "dryout" is reached. See Fig. 3.

The size of this critical heat flow is depending on the amount of steam, which can be seen at Fig. 4. This scheme also depends on the material of the tube, the shape of the surface and the steam formation of the water.

Natural- and forced circulation are operated in a way that the water mass flow is kept in a certain range, to prevent reaching critical heat flow. Circulation number U must be bigger than a critical value ($U > U_{crit}$).

Once-through boilers operate with super critical steam – dryout is demanded. Hence, heat transfer must be kept inside the "post dry out zone" (right bottom corner of figure) to avoid material damages.

When above described measures are not taken into account, overheating and weakening of the material results. Since high pressure loading is at hand, a burst of a tube is possible.

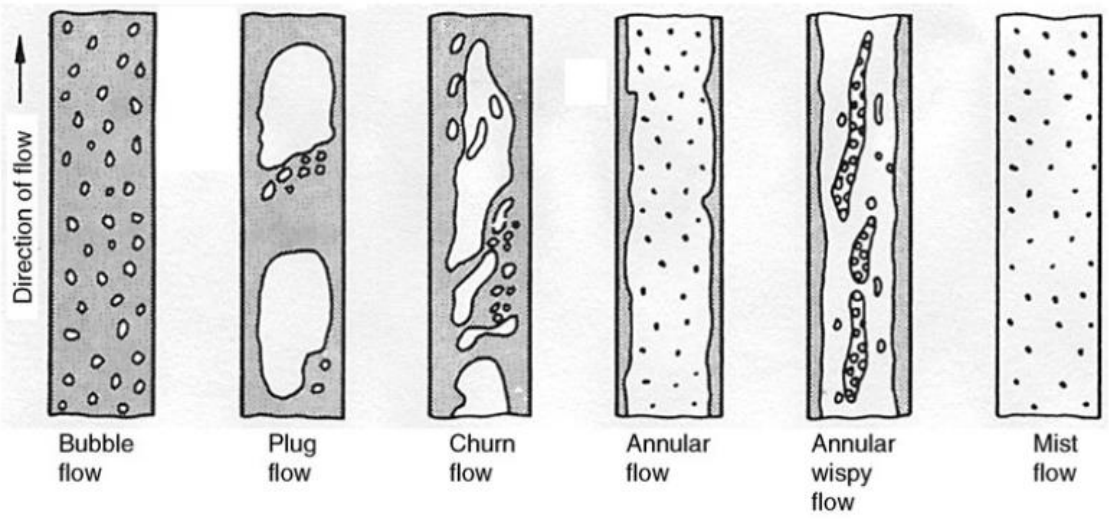

Fig. 2 flow patterns for upward flow in vertical tubes [4] (chapter H3.1)

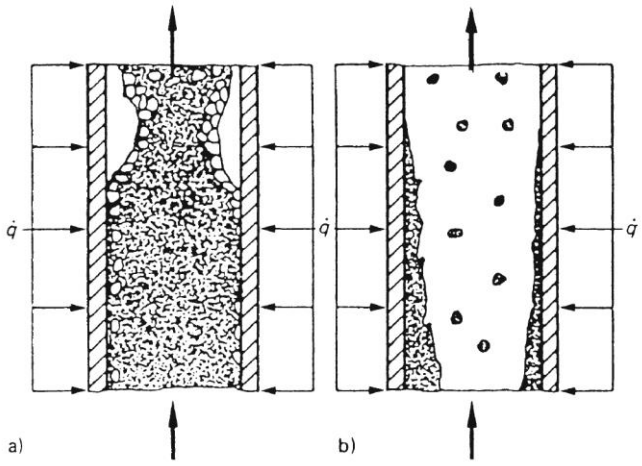

Fig. 3 film boiling (a) and dry out (b) [4] (chapter H3.5)

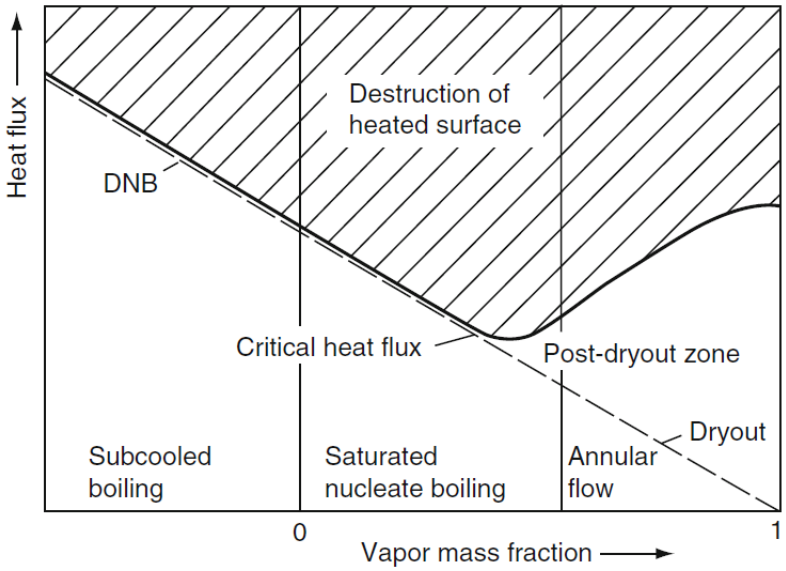

Fig. 4 critical heat flux for the system water / steam [4] (chapter H3.5)

2.2 Pollution and corrosion in water / steam tubes

Babcock et al. [5] (chapter 42), lays the foundation of pollution assembly in water/steam tubes: There can be impurities in the water and / or the steam. These can be caused by interactions between water and the material it's in contact with. E.g. dissolved H^+ or OH^- ions can form oxides, hydroxides and hydrogen. This interaction typically happens at locations where process values (temperature, pressure, amount of steam, velocity) change. But impurities can also come from the feeding water. Thus, a control of the water quality is inevitable. Cleaning strategies for the water/steam must be implemented. Also, the feeding water needs to be treated by adding chemicals to control its pH and oxygen concentration. All these measures can lead to a more sufficient steam generation and longevity of all mechanical components (pumps, tubes, …).

At boilers driven by natural circulation, the concentration of the circulated water needs to be monitored. At once through boilers, no circulation of water occurs, therefore this step is not applicable.

When looking at the fin-tubes of a boiler at the place of highest heat flux, the concentration of impurities (hardness deposits like calcium and magnesium and metallic oxides) increases due to evaporation, as follows: When steam is generated at film boiling, dissolved solids reside. The steam is pushed away by the impulse of the flowing water, but the solids stay. Also, the water which is near the tube wall is superheated and therefor the solubility of solids decreases. At already deposited minerals, accumulation of additional solids is very likely. Other effects, like hideout (and hideout return) where minerals only are dissolved at full load and are solved again at half load and therefor difficult to monitor, must be considered.

If deposits block the heat flow through the metal tube, the material can be overheated and weakened, eventually resulting in a rupture.

Corrosion of the metal material can occur underneath deposit layers, where concentration of minerals is possible.

However, today's water treatment strategies are satisfying and the likelihood of a tube rupture due to deposits or corrosion is very low.

2.3 Additional damage sources

Abrasion, due to a leakage of a surrounded pressure containing tube or valve that results in an uncontrolled jet of water or steam, can have a weakening effect. Dimensioning according to given standards are meant to avoid damages at welding seams or the material itself. As practice proofs, leakages still can occur.

2.4 Outlet at ruptured tube

Feeding water (as liquid) is fed into the tubes of the membrane wall, where evaporation takes place. At natural and forced circulation, the evaporation reaches a certain steam ratio before entering the super heaters, where at once-through boilers, the evaporator can already contain satisfied steam. The location of the tube rupture is assumed to be at half the height of the membrane wall, see Fig. 5. Thus, an overall state of the water inside the evaporator of a membrane wall can be supposed as two-phase state (liquid / saturated steam).

The pressure inside the membrane wall is much bigger than the pressure inside the combustion chamber (which is in most cases at under pressure - around -5mbarg – to avoid flue gas flowing out at leakages). If a rupture occurs, the outgoing water/steam mass flow of the opening can be compared to a mass flow out of a nozzle: at constant inlet pressure, a decrease of outlet pressure leads to an increase of the velocity of the outgoing media (potential energy is transformed into kinetic energy). This increase of velocity is limited by the speed of sound. In other words, when reaching a certain pressure difference (=critical pressure ratio), the mass flow will not exceed a certain value (= critical mass flow). For liquids or gases, this critical mass flow can be calculated easily, but for two phased media, different conditions must be defined: Depending on whether an evaporating or non-evaporating flow is present, the geometry of the outflowing cross section and the operating conditions, different calculation methods are available. [4] (chapter L2.4)

The flashing flow-2 phase model of VDI-Wärmeatlas [2] (chapter L2.4) is describing a flow through a nozzle of a two-phased media with higher amount of steam.

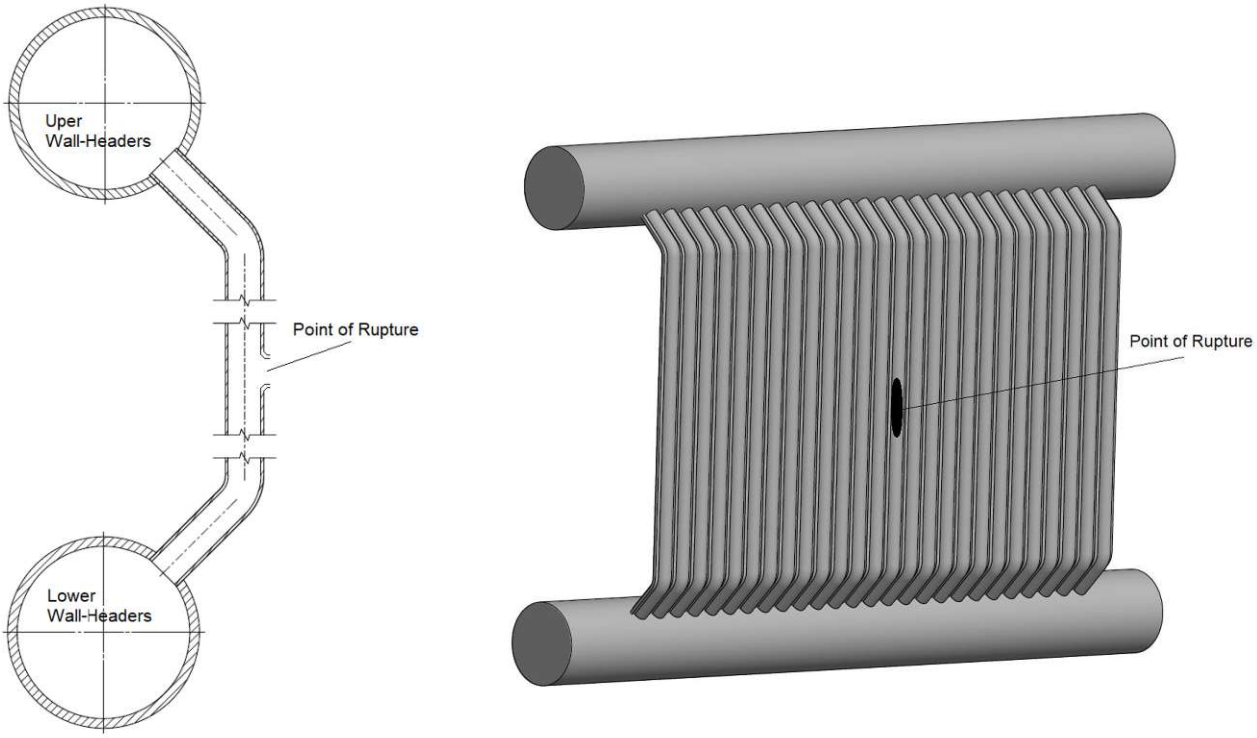

Fig. 5 point of rupture at membrane wall

2.5 Analysis of the problem

Simulating a steam generator can be done by using commercial software like IPSEpro or EBSOLIN. But in order to simulate a tube rupture and take assumptions of possible effects in detail, a new model must be developed. Therefor a simulation is written in MATLAB. An Excel file is used as user-friendly surface for data input and output.

The description of a tube rupture can be achieved in various ways. Erler [6] describes the steam generator as a model of the flue gas part, calculated with one calculation node. The combustion chamber and the sum of the following components of the flue gas duct are summed up to one control volume which interacts with the incoming water from the tube. This strategy is a simple model for gaining results in a realistic range. Since the different sections of the flue gas channel are not described, process data (pressure, temperature, …) are not known but can be useful for dimensioning. Also, the heat transfer is very much simplified.

Within this work, the flue gas side of the boiler is described with a chain of calculation nodes. First, the combustion chamber, as well as the superheater(s) and the economizer. At last, the other parts of a typical duct (like flue gas filter up to the induction fan) are represented by one node.

In the following chapter, the modelling of the problem is displayed. For each part of the boiler that is involved in the tube rupture, a proper way of mathematical description is needed. Thus, the two major parts of this simulation are the flue gas side and the water / steam side.

3 Model

There are two segments which are needed for describing a tube rupture and therefor need to be described numerically: the flue gas side (including the combustion room) and the water / steam side (including the membrane wall). Both problems will be described with separate equation systems which are linked by using results from each other as boundary condition. Together with the given boundary conditions it is possible to solve the equations of each system by setting up a boundary value problem.

3.1 Flue gas side

Combustion takes place in the combustion chamber, which is surrounded by tubes filled with water. A heat flow to the colder water tubes results. A fan is forcing a mass flow of the flue gas along a channel, passing several heat exchanger bundles, an ECO, flue gas treatment and finally going to the stack.

Segments that follow the combustion chamber are included, to have more information about the mass flow, pressure and temperature of flue gas at the ambience of the combustion chamber. This information can help at dimensioning the volumes of each segment of the channel.

The processes inside the area of the flue gas channel are mathematically formulated with help of partial differential equations (PDE), also called “transport equations”. These equations describe the transport of characteristics of the media inside the channel. The PDEs are discretized to formulate algebraic equations. Discretization needs the flue gas channel to be separated into a net of calculation nodes by applying a finite volume method. This method divides the concerned area into control volumes, see Fig. 6. Each control volume is fixed-in-space and has a calculation node at its center, which corresponds to an average value of each characteristic within the control volume.

Balances of fixed-in-space control volumes are used as transport equations. A balance describes an evolution of a characteristic by interaction with its ambience.

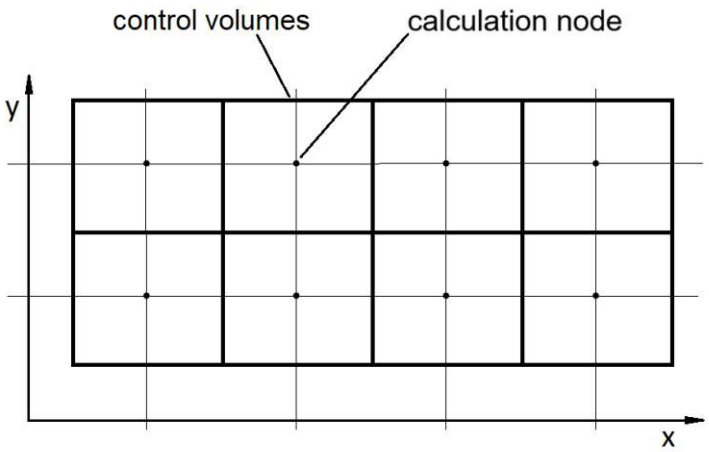

Fig. 6 control volumes with calculation nodes

Thus, the different segments of the flue gas channel are presented as calculation nodes. The net of nodes is one dimensional. The calculation nodes are numbered, as can be seen in Fig. 7. The count value of the nodes is defined with i, the preceding node as $i-1$ and the following with $i+1$.

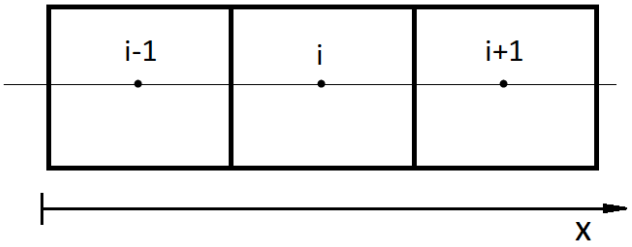

Fig. 7 linear calculation area

Each node is described as a finite volume and has its state values (absolute pressure p_i, flue gas temperature T_i, the volume V_i and the mass m_i inside the control volume). Other information that is necessary for calculation is considered as well (the density ρ_i, enthalpy h_i, the inner energy u_i, specific volume of water and steam v_i', v_i'', the specific gas constant R_i and many others) and will be mentioned in the corresponding formulas. As described before, the finite volumes interact with each other. For this, a mass flow $\dot{m}_i$ (in one direction) which is exiting the control volume is defined. Interacting with the ambience (other than the neighboured control volumes) is realized by a heat flow $\dot{Q}_i$ as an energy input into the thermodynamic system. See Fig. 8 for a definition of an example control volume i,.

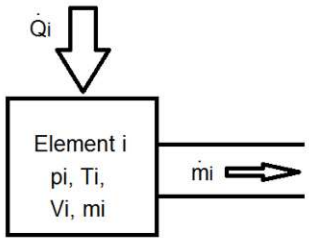

Fig. 8 control volume i

A one-dimensional model has two borders to the ambience which are defined by boundary conditions. The flue gas channel can be separated into as many control volumes as needed. The following structure has been chosen: The initial node represents a fictional node, just having a mass flow serving as boundary condition for the next node. The second node displays the combustion chamber, the third the flue gas channel surrounding the first super heater, the fourth surrounding the evaporator. The fifth node is the one interacting with the ECO and the last one represents also a fictional one, defining a pressure level for keeping a stable flow of flue gas. Then, a heat flux $\dot{Q}_i$, describing heat input from the heat exchanger tube bundles is defined. See Fig. 9 for arrangement of nodes.

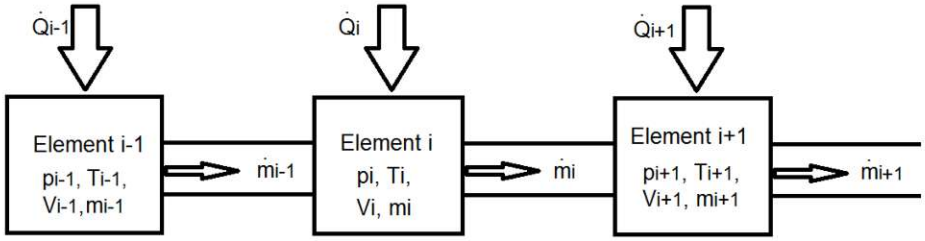

Fig. 9 model of flue gas channel

A mass balance (Eq. (1)) describes the change of mass inside the control volume m_i as the difference of incoming and outgoing mass flow. The relation of the outgoing mass flow $\dot{m}(t)$ to the pressure loss in the flue gas channel $\Delta p(t)$ is calculated with an impulse balance (Eq. (2)), where a friction coefficient ζ, cross section A and the specific volume v is needed. Else, an energy balance (Eq. (3)) which describes the change of energy inside the control volume E by calculating the difference of in- and outgoing enthalpy flow $\dot{H}_{i-1} - \dot{H}_i$ and all acting heat flows $\sum \dot{Q}_i$.

$$\frac{\mathrm{d}m_i}{\mathrm{d}t} = (\dot{m}_{i-1} - \dot{m}_i) \tag{1}$$

$$\Delta p(t) = \zeta \ \frac{\dot{m}(t)^2}{2A^2} \ v \tag{2}$$

$$\frac{\mathrm{d}E}{\mathrm{d}t} = (\dot{H}_{i-1} - \dot{H}_i + \sum \dot{Q}_i) \tag{3}$$

Linearizing of these differential equations is necessary. A forward and backward difference method with the time step n is introduced. The preceding time step is defined as $n-1$ and the following with $n+1$. Thus, the mass-, impulse- and energy balance are adapted (Eq. (4), (5) and (6)). The mass balance using a backward differential method, the impulse balance a forward and the energy balance a backward method. For the impulse balance, an average of the specific volume of two control volumes is needed. The energy balance takes the difference of inner energy of two time steps U_i^{n+1} and U_i^n (by using the mass m_i, the isochoric specific heat capacity c_{vi} and temperature T_i). The enthalpy flow $\dot{H}_i$ for the incoming mass flow ($\dot{m}_{i-1}h_{i-1}$) and the outgoing one ($\dot{m}_i h_i$) is calculated. First for the dry flue gas ratio ($\dot{m}(1-\xi)$), the isobaric spec. heat capacity c_{pi} and the temperature T_i are needed. Second for the ratio of water ($\dot{m}\xi$), the spec. enthalpy of water / steam h_{WS_i} is necessary.

$$m_i^{n+1} - m_i^n = (\dot{m}_{i-1}^n - \dot{m}_i^n)\,\Delta t \tag{4}$$

$$p_i^n - p_{i+1}^n = \zeta \frac{\dot{m}_i^{n\,2}}{2A_i^2}\left(\frac{v_i^n + v_{i+1}^n}{2}\right) \tag{5}$$

$$\underbrace{m_i^{n+1}\, c_{v_i}^{\ n+1}\, T_i^{n+1}}_{U_i^{n+1}} - \underbrace{m_i^n c_{v_i}^{\ n}\, T_i^n}_{U_i^n} =$$

$$= \Bigg[\underbrace{\left(\dot{m}_{i-1}^n (1-\xi_{i-1}^n) c_{p_{i-1}}^{\ n}\, T_{i-1}^n + \dot{m}_{i-1}^n \xi_{i-1}^n h_{WS_{i-1}}\right)}_{\dot{m}_{i-1}^n h_{i-1}^n} \tag{6}$$

$$- \underbrace{\left(\dot{m}_i^n (1-\xi_i^n) c_{p_i}^{\ n}\, T_i^n + \dot{m}_i^n \xi_i^n h_{WS_i}\right)}_{\dot{m}_i^n h_i^n} + \sum \dot{Q}_i^n \Bigg] \Delta t$$

3.1.1 Linearizing

The set up equations can either be solved explicit or implicit. Explicit calculation takes results from the previous time step and derives values of the following time step. A linear formulation of the equations is possible and therefore not time consuming (compared to implicit calculation). Initial- and boundary conditions are needed, and solving the transport equations is done time step by time step. The stability of an explicit method is depending on the initial values and the time step width Δt between each time step. Since initial values are mostly predefined, the time step width must be adapted as needed to assure convergence.

An implicit formulation is executed with nonlinear equations, which must be solved simultaneously and need iteration. The stability is given unconditionally. But when initial values diverge too far from realistic values or the time step width is too big, the solutions can diverge too.

The formulation of the differential coefficients can be explicit or implicit, depending on weather using the values of the present time step n or the following $n+1$. e.g. $(\dot{m}_{i-1}^{n} - \dot{m}_{i}^{n})$ or $(\dot{m}_{i-1}^{n+1} - \dot{m}_{i}^{n+1})$. Since the model of the flue gas will be combined with the water side model, the calculation should be easily changeable. To change between explicit and implicit, a formulation which enables the calculation to be semi-parallel is chosen. By defining a factor f, the calculation can vary between fully explicit ($f = 0$), fully implicit ($f = 1$) or any value in between. The mass- and energy balance are rewritten correspondingly (Eq. (7), (8)).

$$m_i^{n+1} - m_i^n = \left[(\dot{m}_{i-1}^{n+1} - \dot{m}_i^{n+1})\, f + (\dot{m}_{i-1}^{n} - \dot{m}_i^{n})\,(1-f)\right] \Delta t \tag{7}$$

$$\begin{aligned} m_i^{n+1}\, {c_{v_i}}^{n+1}\, T_i^{n+1} - m_i^n\, {c_{v_i}}^n\, T_i^n = \\ = \Big[\Big[\Big(\dot{m}_{i-1}^{n+1}\,(1-\xi_{i-1}^{n+1})\, {c_{p_{i-1}}}^{n+1}\, T_{i-1}^{n+1} + \dot{m}_{i-1}^{n+1}\,\xi_{i-1}^{n+1} h_{i-1}\Big) \\ - \Big(\dot{m}_i^{n+1}\,(1-\xi_i^{n+1})\, {c_{p_i}}^{n+1}\, T_i^{n+1} + \dot{m}_i^{n+1}\,\xi_i^{n+1}\, h_i\Big) + \dot{Q}_i^{n+1}\Big] f \\ + \Big[\Big(\dot{m}_{i-1}^{n}\,(1-\xi_{i-1}^{n})\, {c_{p_{i-1}}}^{n}\, T_{i-1}^{n} + \dot{m}_{i-1}^{n}\,\xi_{i-1}^{n}\, h_{i-1}\Big) \\ - \Big(\dot{m}_i^{n}\,(1-\xi_i^{n})\, {c_{p_i}}^{n}\, T_i^{n} + \dot{m}_i^{n}\,\xi_i^{n}\, h_i\Big) + \dot{Q}_i^{n}\Big](1-f)\Big]\Delta t \end{aligned} \tag{8}$$

Another needed size is the amount of water inside the flue gas ξ, which is defined by the ratio of the mass flow of water $\dot{m}_{WS}$ and the mass flow of the mixture $\dot{m}$, see Eq. (9). ξ can be derived from an adaption of the initial mass balance for the water ratio of the flue gas (ξm), see Eq. (10).

$$\xi = \dot{m}_{WS}/\dot{m} \tag{9}$$

$$\xi_i^{n+1}\, m_i^{n+1} - \xi_i^n\, m_i^n = [(\xi_{i-1}^n\, \dot{m}_{i-1}^{n+1} - \xi_i^n\, \dot{m}_i^{n+1})\, f + (\xi_{i-1}^n\, \dot{m}_{i-1}^n - \xi_i^n\, \dot{m}_i^n)\,(1-f)]\,\Delta t \tag{10}$$

Also, the specific gas constant of the flue gas R_i must be described with Eq. (11) by using the amount of water ξ_i and the corresponding gas constants (R_{WS} and R_{RG}). The specific volume v is calculated with Eq. (12). Finally, the ideal gas law is used for calculating the pressure of the flue gas in Eq. (13).

$$R_i^{n+1} = R_{WS}\ \xi_i^{n+1} + R_{RG}\,(1-\xi_i^{n+1}) \tag{11}$$

$$v_i^{n+1} = \frac{V_i}{m_i^{n+1}} \tag{12}$$

$$p_i^{n+1}\, V_i^{n+1} = m_i^{n+1}\, R_i^{n+1}\, T_i^{n+1} \tag{13}$$

The equation system can be used to elaborate the values of the new time step $n+1$, like m_i^{n+1}, T_i^{n+1}. The used program MATLAB supplies a solving system for equations, named *fsolve*. The formulas need to have the structure $f(x) = 0$.

Calculation of searched values is done according to the following sequence:

$m_i^{n+1}$with Eq. (7) → ξ_i^{n+1} with Eq. (10) → R_i^{n+1} with Eq. (11) → v_i^{n+1} with Eq. (12) → T_i^{n+1} with Eq. (8) → ${p_i}^n$ with Eq. (13) → $\dot{m}_i^n$ with Eq. (5)

3.1.2 Boundary conditions

Known values are the generated amount of flue gas which already describes a fully burned fuel, all corresponding data of the flue gas, the heat flow towards the water tubes or tube bundles, a friction coefficient ζ_i for each calculation node (which is used to describe the pressure drop through every segment), the volume and cross section of each segment. Also, the pressure and temperature of the last control volume are always known. When a tube rupture occurs, the incoming water / steam mixture is also given.

3.1.3 Initial values

As initial values, the operation temperature, pressure and mass flow of each control volume are used. Initial values describe a steady operation, no start-up process of the boiler is described.

3.2 Water side

Before describing the side of the boiler which contains water/steam, the system borders need to be defined, as seen in Fig. 10. The water system of a typical boiler contains the tube walls around the combustion chamber, the tube bundles of the superheaters, the water drum, collecting and connecting tubes, as well as the line to the steam turbine. Simplification has been done by summing up all water containing volumes into one "pot" which contains all water mass within the system. This pot only interacts with the combustion chamber. All other calculation nodes of the flue gas tract have a heat transfer to their surrounding described as a constant heat flow. This simplification is acceptable, as it considers the heat capacity of the steel tubes. The time of the pressure increase inside the combustion chamber (during a tube rupture), is shorter than the time of the temperature decrease of the tube material. Thus, in the most critical first 60 seconds of the tube rupture, the heat flow of the heat exchangers towards the flue gas duct remains.

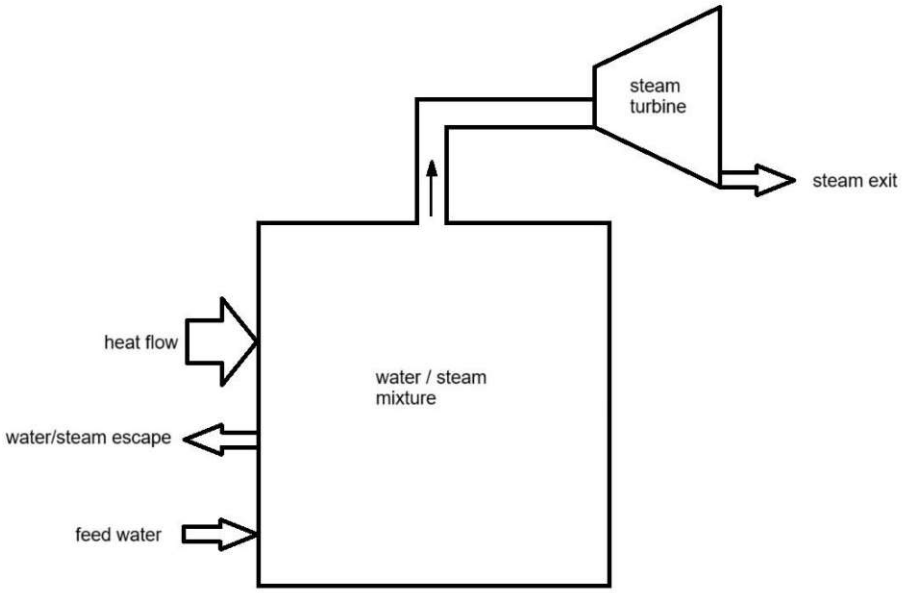

Fig. 10 model of water / steam pot

A mass- and energy balance can be set up to describe the processes going on inside the pot. Then, an impulse balance can be used to calculate the outgoing steam mass flow towards the turbine. The mass balance of the water / steam pot (Eq. (14)) calculates the change of the mass of the water / steam mixture m_{WS} by using the in- and outgoing mass flows, which are the incoming feed water $\dot{m}_{FW}$, the outflowing steam $\dot{m}_S$ towards the turbine and the mass flowing out the ruptured tube $\dot{m}_{TR}$ (if rupture happening). The energy balance Eq. (15) takes the difference of inner energy ($m_{WS}u_{WS}$) and matches it with the enthalpy flow of the incoming

feed water ($\dot{m}_{FW} h_{FW}$), of the exiting steam ($\dot{m}_S h_S$) and the one for the tube rupture ($\dot{m}_{TR} h_{TR}$) together with all acting heat flows $\sum \dot{Q}_i$. Also, an impulse balance who describes the flow of steam to the turbine Eq. (16) is used. Needed values are: The pressure inside the pot p_{WS}, of the ambience p_{Am}, a friction coefficient of the tube towards the turbine ζ, the steam mass flow towards the turbine $\dot{m}_S$, the cross section of the tube connection to the turbine inlet A_{Ti}, also its length l_{Ti} and diameter d_{Ti} and finally the specific volume of the steam v_S.

$$m_{WS}^{n+1} - m_{WS}^{n} = (\dot{m}_{FW}^{n} - \dot{m}_S^{n} - \dot{m}_{TR}^{n})\,\Delta t \tag{14}$$

$$m_{WS}{}^{n+1} u_{WS}{}^{n+1} - m_{WS}{}^{n} u_{WS}{}^{n} = \left(\dot{m}_{FW}{}^{n}\, h_{FW}{}^{n} - \dot{m}_S{}^{n}\, h_S{}^{n} - \dot{m}_{TR}{}^{n}\, h_{TR}{}^{n} + \sum \dot{Q}_i^{n}\right)\Delta t \tag{15}$$

$$p_{WS}^{n+1} - p_{Am} = \zeta \frac{\dot{m}_S^{n\,2}}{2A_{Ti}^2} \frac{l_{Ti}}{d_{Ti}} v_S^{n} \tag{16}$$

Sought values for a new time step $n+1$ are elaborated after the following order:

$$m_{WS}^{n+1} \rightarrow u_{WS}^{n+1} \rightarrow p_{WS}^{n+1}, h_S^{n+1}, v_S^{n+1}, v_W^{n+1}, T_{WS}^{n+1} \rightarrow v_{WS}^{n+1} \rightarrow x^{n+1} \rightarrow \dot{m}_S^{n+1}$$

Namely, the mass of water / steam mixture m_{WS}^{n+1}will be calculated with the mass balance (Eq. (14)). Next, the inner energy of the mixture u_{WS}^{n+1} by using the energy balance (Eq. (15)). With the mass and the inner energy, other values (pressure of mixture p_{WS}^{n+1}, enthalpy of the steam amount h_S^{n+1}, specific volume of the steam amount v_S^{n+1}, spec. volume of the water amount v_W^{n+1}, temperature of the mixture T_{WS}^{n+1}) can be derived with help of the water-steam-table. The specific volume of water / steam mixture v_{WS} is calculated with Eq. (17) by using the absolute volume of water-steam V_{WS}. The amount of steam x^{n+1} is calculated with the definition of it, see Eq. (18).

$$v_{WS}{}^{n+1} = \frac{V_{WS}}{m_{WS}{}^{n+1}} \tag{17}$$

$$x^{n+1} = \frac{v_{WS}^{n+1} - v_W^{n+1}}{v_S^{n+1} - v_W^{n+1}} \tag{18}$$

3.2.1 Boundary conditions

The water input through the feeding pump is defined as a constant mass flow. The steam outlet, representing a turbine (implying a heat-to-energy system), is calculated by using an impulse balance. Dimensions, like the volume of the pot can be calculated by summing up all water containing parts. Also, the sum of the heat input from the flue gas side is given and calculated by the heat transfer model, see next chapter.

3.2.2 Initial values

As initial values, the volume, mass and mass flows of a steady operation need to be supplied.

3.3 Heat transfer of water / steam pot

The interaction between the combustion chamber and the water / steam pot needs to be thought of, since the outflow of a ruptured tube can be derived in more detail when its surrounding is defined well and considered. The temperatures and inner energy of the system must be known.

For describing the interaction between the flue gas and the water / steam pot, the heat transfer from the hot gas towards the colder water / steam mixture must be derived. The boiler wall is built up as follows (see Fig. 11): The membrane wall (carbon steel), insulated by a material like glass wool on the outside and a layer of fouling inside the combustion chamber must be considered. For fouling, a thin layer with a composition of ash is assumed.

A conjugate heat transfer is present: a combination of a radiant, a convective and a conduction part. Calculation points, where the temperature must be known, have been defined, as displayed in Fig. 12. The heat transfer through the fouling contains a radiant part coming from the hot ambience of the combustion chamber facing the membrane wall, the convection from the hot flue gas flowing along the membrane wall and conduction through the fouling and metallic material of the membrane wall. The heat transfer towards the ambience consists of conduction through the outer tube wall and the insulation.

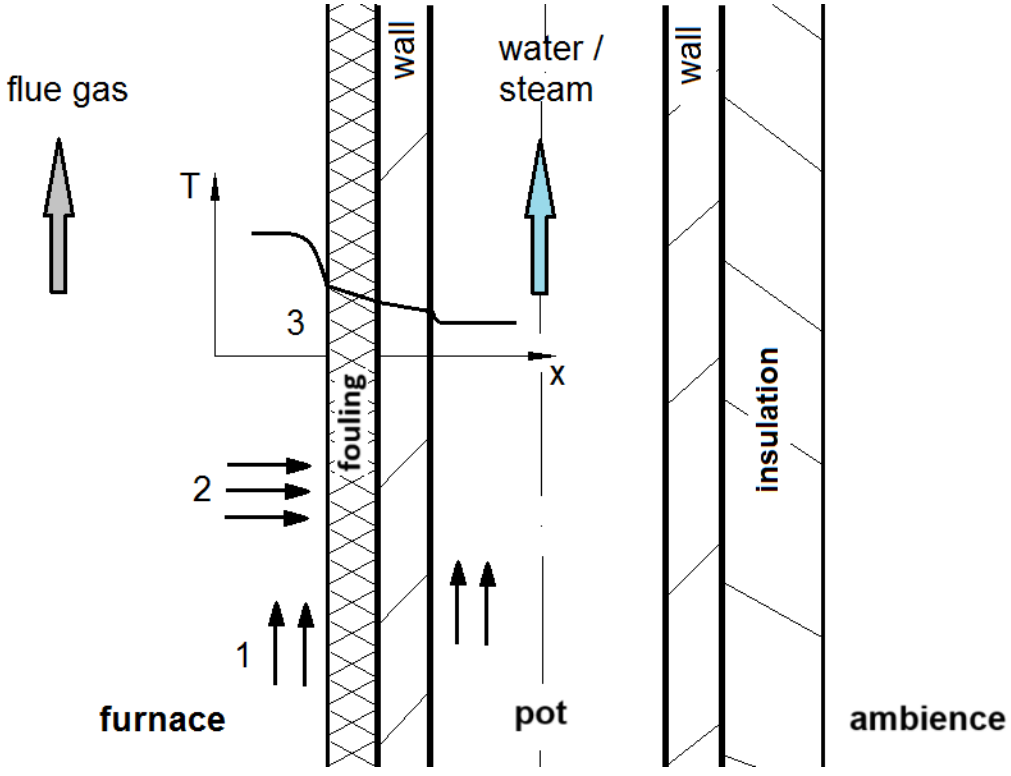

Fig. 11 heat transfer from combustion chamber to water / steam and to ambience

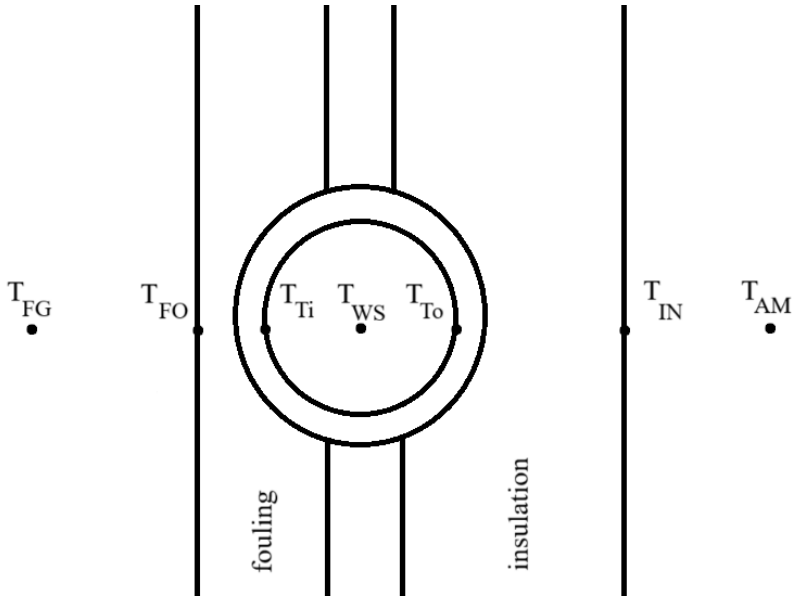

Fig. 12 heat transfer calculation points (FG…flue gas, FO…fouling, Ti…tube inside, WS…water steam mixture, To…tube outside, IN…insulation, AM…ambience)

The temperature of the flue gas T_{FG} and of the ambience T_{AM} are taken as boundary values. For deriving the remaining temperatures, an energy balance for each calculation point needs to be set up.

3.3.1 Heat transfer at fouling

In Eq. (19), an energy balance of the fouling layer is reviewed. The corresponding calculation point is located at the outer surface of the fouling. The change of inner energy is described by a difference of the temperatures of the fouling layer T_{FO} with the corresponding mass m_{FO} and spec. heat capacity $c_{v,FO}$. The in- and outgoing heat flows ($\dot{Q}_{FG\rightarrow FO}$ from the fuel gas to fouling and $\dot{Q}_{FO\rightarrow Ti}$ from fouling to the tube wall inside) are considered. The heat transfer between the combustion chamber and the fouling layer contains a radiant $\dot{Q}_{FG\rightarrow FO,rad}$ and a convective part (forced convection) $\dot{Q}_{FG\rightarrow FO,conv}$. The radiant flow Eq. (20) is calculated with σ as the Stefan-Boltzmann-number, ε as the emission number and the projected surface $A_{CC,proj}$ facing the combustion chamber. For the convective flow Eq.) the surface of the combustion chamber A_{CC} is used and a heat transfer coefficient for forced convection $\alpha_{FG\rightarrow FO}$ must be estimated (see Eq. (22)). The Nusselt number Nu as function of Reynolds number Re and Prandtl number Pr (see Eq.(23)) and a thermal conductivity λ divided by a characteristic length L are needed. For the Nusselt number, an attempt for a laminar flow over a flat plate is used. At Eq. (24), Reynolds Re needs the speed of the flow v ,the characteristic length L and the kinematic viscosity of the

flue gas ν. For the Prandtl number Pr, the viscosity ν, the density ρ, the spec. heat capacity c_p and the thermal conductivity λ are used.

$$m_{FO} c_{v,FO} \left({T_{FO}}^{i+1} - {T_{FO}}^{i} \right) = (\dot{Q}_{FG \to FO} - \dot{Q}_{FO \to Ti}) \Delta t \tag{19}$$

$$\dot{Q}_{FG \to FO,rad} = \sigma \varepsilon \left({T_{FG}}^{i^4} - {T_{FO}}^{i+1^4} \right) A_{CC,proj} \tag{20}$$

$$\dot{Q}_{FG \to FO,conv} = \alpha_{FG \to FO} \left({T_{FG}}^{i} - {T_{FO}}^{i+1} \right) A_{CC} \tag{21}$$

$$\alpha_{FG \to FO} = \frac{\lambda}{L} Nu(Re, Pr) \tag{22}$$

$$Nu = 0.664 \, Re^{\frac{1}{2}} Pr^{\frac{1}{3}} \tag{23}$$

$$Re = \frac{\mathrm{v} L}{\nu}, \quad Pr = \frac{\nu \rho c_p}{\lambda} \tag{24}$$

To establish a linear equation system, the radiant heat flow, which is described by a temperature with the power of 4, must be avoided. Thus, a heat transfer coefficient $\alpha_{FG,rad}$ is defined in Eq. (25) to receive an adapted heat transfer definition, Eq. (26).

$$\alpha_{FG,rad} = \frac{\left({T_{FG}}^{4} - {T_{FO}}^{4} \right)}{(T_{FG} - T_{FO})} \sigma \varepsilon \tag{25}$$

$$\dot{Q}_{FG \to FO,rad} = \alpha_{FG,rad} A_{CC,proj} (T_{FG} - T_{FO}) \tag{26}$$

The heat transfer from the fouling to the inner pipe surface $\dot{Q}_{FO \to Ti}$ is described as a conduction process. As described in Eq. (27), it contains a heat conductance R_I (which is displaying the sum of all heat transfer coefficients of the fouling and the tube, see Eq. (28) with the width b_{FO} and heat conductance λ_{FO} of the fouling and of the tube wall, b_T and λ_T, respectively).

$$\dot{Q}_{FO \to Ti} = R_I A_{CC} (T_{FO} - T_{Ti}) \tag{27}$$

$$R_i = \frac{1}{\frac{b_{FO}}{\lambda_{FO}} + \frac{b_T}{\lambda_T}} \tag{28}$$

The resulting energy balance, Eq. (29), can be used for building an equation system:

$$m_{FO}c_{v,FO}\left({T_{FO}}^{i+1}-{T_{FO}}^{i}\right)= \left[\alpha_{FG,rad}\left({T_{FG}}^{i}-{T_{FO}}^{i+1}\right)A_{CC,proj}+\alpha_{FG\to FO}\left({T_{FG}}^{i}-{T_{FO}}^{i+1}\right)A_{CC}-R_i\left({T_{FO}}^{i+1}-{T_{Ti}}^{i+1}\right)A_{CC}\right]\Delta t. \tag{29}$$

Next, an energy balance of the inside of the membrane wall is set up, Eq. (30). The corresponding calculation point is located at the inner surface of the tube. Since the membrane wall is divided into an inner and outer part, the mass must be separated as well. Therefor the half of the mass m_{Tube} is considered. A spec. heat capacity of the tube $c_{v,Tube}$ is needed. The in- and outgoing heat flows ($\dot{Q}_{FO\to Ti}$ from the fouling to the inside tube wall and $\dot{Q}_{Ti\to WS}$ from the tube wall to the water / steam mixture) are included. The heat transfer from the tube towards the water / steam is described with forced convection in Eq. (31) with a corresponding heat transfer coefficient $\alpha_{Ti\to WS}$ (which can be derived as $\alpha_{FG\to FO}$). The resulting energy balance Eq. (32) can be used for building an equation system.

$$\frac{m_{Tube}}{2}c_{v,Tube}\left({T_{Ti}}^{i+1}-{T_{Ti}}^{i}\right)=\left(\dot{Q}_{FO\to Ti}-\dot{Q}_{Ti\to WS}\right)\Delta t \tag{30}$$

$$\dot{Q}_{Ti\to WS}=\alpha_{Ti\to WS}A_{CC}\left(T_{Ti}-T_{WS}\right) \tag{31}$$

$$\frac{m_{Tube}}{2}c_{v,Tube}\left({T_{Ti}}^{i+1}-{T_{Ti}}^{i}\right)= =\left[R_i\left({T_{FO}}^{i+1}-{T_{Ti}}^{i+1}\right)A_{CC}+\alpha_{Ti\to WS}\left({T_{Ti}}^{i+1}-{T_{WS}}^{i}\right)A_{CC}\right]\Delta t \tag{32}$$

To derive the temperatures ${T_{FO}}^{i+1}$ and ${T_{Ti}}^{i+1}$, a calculation system is set up as shown in Eq. (33).

$$\begin{pmatrix} a\ \dots \\ \quad\ \dots \end{pmatrix}\begin{pmatrix} T_{FO} \\ T_{Ti} \end{pmatrix}=\begin{pmatrix} b\ \dots \\ \quad\ \dots \end{pmatrix} \tag{33}$$

Since, the calculation of the flue gas side is half-implicit-explicit, the numerical description of this heat transfer problem should be implicit to avoid any incompatibility. A numerical approach for solving heat transfer problems with simple models is supposed by Patankar [7] (chapter 4.3-3). Since the heat transfer is conjugate, the first simplification is, all sorts of transfer to be described or handled as conduction. Patankar is giving a model for calculating an unsteady one-dimensional conduction with help of a fully implicit discretization equation. Here, three calculation nodes, defined as “W”, “P” and “E”, with their temperatures T_W, T_P and T_E

are used. The temperature at the time points Δt and $\Delta t + 1$ for each node is considered. The points are connected with each other with coefficients a, defined in Eq. (34):

$$a_P = a_E + a_W + {a_P}^0 - S_p \Delta x \tag{34}$$

The term $S_p \Delta x$ describes a well, which is not present and therefor irrelevant. ${a_P}^0$ describes the coefficient for point P at a previous time step and is defined with Eq. (35):

$${a_P}^0 = \frac{\rho c \Delta x}{\Delta t} \tag{35}$$

While using the mass m and the coefficient of heat capacity c of the calculation node, this term can be used as shown in Eq. (36):

$${a_P}^0 = \frac{mc}{\Delta t} \tag{36}$$

Corresponding re-sorting the equation leads to Eq. (37), and therefor to Eq. (38):

$$\begin{aligned} &\underbrace{T_{FO}{}^{i+1}}_{T_1} \underbrace{\left(\frac{m_{FO} c_{v,FO}}{\Delta t} + \alpha_{FG,rad} A_{CC,proj} + \alpha_{FG \to FO} A_{CC} - R_i A_{CC}\right)}_{a_{11}} + \\ &+ \underbrace{T_{Ti}{}^{i+1}}_{T_2} \underbrace{(-R_I A_{CC})}_{-a_{12}} = \\ &= \underbrace{T_{FG}{}^{i}}_{T_0} \underbrace{\left(\alpha_{FG,rad} A_{CC,proj} + \alpha_{FG \to FO} A_{CC}\right)}_{a_0} + \\ &+ \underbrace{T_{FO}{}^{i} \left(\frac{m_{FO} c_{v,FO}}{\Delta t}\right)}_{b_1} \end{aligned} \tag{37}$$

$$\begin{aligned} a_{11} T_1 &= a_{12} T_2 + a_0 T_0 + b_1 \\ \rightarrow a_{11} T_1 - a_{12} T_2 &= a_0 T_0 + b_1 \end{aligned} \tag{38}$$

The same is done with the energy balance of the inside tube wall respectively, see Eq. (39). A requested structure is established in Eq. (40).

$$\begin{aligned}
&\underbrace{T_{FO}{}^{i+1}}_{T_1}\underbrace{(-R_I A_{CC})}_{-a_{21}} + \\
&+\underbrace{\mathrm{T_{Ti}}^{i+1}}_{T_2}\underbrace{\left(\frac{m_{Tube}c_{v,Tube}}{\Delta t} + R_I A_{CC} + \alpha_{Ti\to WS}A_{CC}\right)}_{a_{22}} = \\
&= \underbrace{T_{WS}{}^{i}}_{T_N}\underbrace{(\alpha_{Ti\to WS}A_{CC})}_{a_N} \\
&+\underbrace{T_{Ti}{}^{i}\left(\frac{m_{Tube}c_{v,Tube}}{2\Delta t}\right)}_{b_2}
\end{aligned} \tag{39}$$

$$\begin{aligned}
a_{22}T_2 &= a_N T_N + a_{21}T_1 + b_2 \\
a_{22}T_2 - a_{21}T_1 &= a_N T_N + b_2
\end{aligned} \tag{40}$$

Fitting it all into the equation system, leads to Eq. (41). After converting it by using the inversion of matrix $\boldsymbol{a}$, $\boldsymbol{T}$ can be calculated, see Eq. (42).

$$\begin{pmatrix} a_{11} & -a_{12} \\ -a_{21} & a_{22} \end{pmatrix}\begin{pmatrix} T_1 \\ T_2 \end{pmatrix} = \begin{pmatrix} a_0 T_0 + b_1 \\ a_N T_N + b_2 \end{pmatrix} \tag{41}$$

$$\boldsymbol{T} = inv(\boldsymbol{a})\,\boldsymbol{b} \tag{42}$$

3.3.2 Heat transfer at insulation

For deriving the temperatures $T_{To}{}^{i+1}$ and $T_{IN}{}^{i+1}$, the same approach is executed. The heat transfer between the pot and the ambience is calculated. First the heat transfer from the water / steam mixture towards the tube is described at Eq. (43).

$$\dot{Q}_{WS\to To} = \alpha_{WS\to To}A_{CC}(T_{WS} - T_{To}) \tag{43}$$

Then from the tube towards the insulation in Eq. (44).

$$\dot{Q}_{To\to IN} = R_o A_{CC}(T_{To} - T_{IN}) \tag{44}$$

with R_o as the sum of all heat conductance of the tube and the insulation in Eq. (45).

$$R_o = \frac{1}{\frac{b_T}{\lambda_T} + \frac{b_{IN}}{\lambda_{IN}}} \tag{45}$$

At last, the heat transfer from the insulation towards the ambience is described in Eq. (46).

$$\dot{Q}_{IN \to AM} = \alpha_{IN \to AM} A_{CC} (T_{IN} - T_{AM}) \tag{46}$$

To get a similar equation system for solving it, the same approach is taken, which is not necessary to be described in detail.

3.3.3 Iteration

Solving the equation system of the fouling layer delivers the temperature of the fouling layer ${T_{FO}}^{i+1}$ and the inside tube wall ${T_{Ti}}^{i+1}$. Corresponding for the insulation layer, the temperature of the outside tube wall ${T_{To}}^{i+1}$ and the insulation ${T_{IN}}^{i+1}$ are derived.

Since the heat transfer coefficient for radiation of the flue gas towards the fouling layer is a function of the calculated temperatures $\alpha_{FG,rad} = f(T_{FG}, T_{FO})$, a fixed-point iteration is necessary. It is iterated in every time step: after the mentioned equation system is solved and temperatures are derived, a new heat transfer coefficient is calculated. With this new coefficient the whole calculation is repeated. Iteration is executed until a defined stop criterium is reached. Iteration stops when the difference of the new derived heat transfer coefficient $\alpha_{FG,rad,current}$ and of the preceding one $\alpha_{FG,rad,old}$ is smaller than three decimals: $abs(\alpha_{FG,rad,current} - \alpha_{FG,rad,old}) \leq 10^{-3}$.

3.3.4 Boundary conditions

As mentioned before, the temperature of the flue gas T_{FG} and of the ambience T_{AM} are taken as boundary values. Also, the temperature of the water / steam mixture is known, since it was just derived in the corresponding calculation.

3.3.5 Initial values

First, the surface of all water containing tubes is necessary. For calculating the radiant heat transfer the projected surface facing the combustion chamber is also needed.

Then, the heat transfer coefficients $\alpha_{FG\rightarrow FO}$, $\alpha_{\mathrm{FG,rad}}$ must be elaborated. Also, the heat conductance of all material is needed.

3.4 Outlet of water / steam – Flashing flow model

The model of flashing flow which is introduced by VDI-Wärmeatlas [2] (chapter L2.4), describes a flow through a nozzle of a two-phased medium where the amount of steam changes while flowing through the nozzle. The mass flow $\dot{M}$ and critical pressure ratio η_{crit} are calculated with a homogeneous equilibrium model, which assumes a homogeneous mixed flow in thermodynamic equilibrium. Then, correction factors are introduced.

The maximum possible mass flow out of a nozzle (=critical mass flow) $\dot{M}_{max}$ is given in Eq. (47) with the narrowest cross section A_{th}, the pressure and specific volume at the nozzle entrance p_0, v_0. A frictionless flow, incompressible medium and a pressure at full vacuum is assumed. For comparing mass flows, a dimensionless mass flow C is defined in Eq. (48) as the ratio of the actual mass flow $\dot{M}$ to $\dot{M}_{max}$.

$$\dot{M}_{max} = A_{th}\sqrt{2\frac{p_0}{v_0}} \tag{47}$$

$$C = \frac{\dot{M}}{\dot{M}_{max}} \tag{48}$$

The critical pressure ratio of a two-phase medium is larger than of a compressible gas. This issue is considered with the compressibility factor ω, first introduced by Leung [8]. With this factor and the pressure ratio in the narrowest cross section of a nozzle η_{th} (which will be set equal to η_{crit}), the dimensionless mass flow is calculated, see Eq.(49). The subcooling factor η_s is used, which equals 1 for mixtures with steam at saturation. The pressure ratio η_{th} is not known from the beginning and an initial value must be defined, which will be explained later.

$$C = \frac{\sqrt{\omega(N)\eta_s ln\left(\frac{\eta_s}{\eta_{th}}\right) - (\omega(N)-1)(\eta_s-\eta_{th})}}{\omega(N)\left(\frac{\eta_s}{\eta_{th}}-1\right)+1} \tag{49}$$

First, the compressibility factor $\omega(N)$ needs to be calculated as a function of the disequilibrium

factor N, see Eq. (50) with help of Eq. (51), (52) and (53). The disequilibrium factor N describes the deviation of actual steam amount to the thermodynamic equilibrium. The compressibility of steam resulting from pressure changes in the nozzle are defined with ω_{Frozen}. With ω_{Flash}, the compressibility due to evaporation of liquid or condensation of gas is described. N is calculated with an exponent a_{TR} and $\dot{x}_{eq,th}$ (vapour mass flow fraction at narrowest cross section at thermodynamic equilibrium). a_{TR} is calculated with Eq. (54) by using a factor Γ which is depending on $\dot{x}_0$ (vapour mass flow fraction at entrance) as follows: if $\dot{x}_0 < 0.003$ than $\Gamma = \dot{x}_0/0.003$, else $\Gamma = 1$. The fraction $\dot{x}_{eq,th}$ is calculated with Eq. (55). The specific volume of liquid at the entrance v_{l0} needs to be known.

$$\omega(N) = \omega_{Frozen} + \omega_{Flash} N \tag{50}$$

$$\omega_{Frozen} = \dot{x}_0 \frac{v_{gs}}{v_0}\left[(1-k) + \frac{k}{\left(\frac{1}{\eta_g}\right) - 1}\left(\left(\frac{1}{\eta_g}\right)^{\frac{1}{\kappa}} - 1\right)\right] \tag{51}$$

$$\omega_{Flash} = \frac{c_{pls} T_0 p_0 \eta_s}{v_0} \frac{\left(v_{gs} - v_{l0}\right)^2}{{\Delta h_{vs}}^2} \tag{52}$$

$$N = \left(\dot{x}_{eq,th}\right)^{a_{TR}} \tag{53}$$

$$a_{TR} = {\eta_s}^{-0.6} + \frac{3}{5}\Gamma \tag{54}$$

$$\dot{x}_{eq,th} = \dot{x}_0 + \omega_{Flash} \frac{v_0}{v_{gs} - v_{l0}} \ln\left(\frac{\eta_s}{\eta_{th}}\right) \tag{55}$$

Values that are needed for ω_{Frozen} and ω_{Flash} must be elaborated as follows: v_{gs} is the specific volume of the gas at boiling curve, Eq. (56) with T_S as the boiling temperature, T_0 as the temperature at the entrance and v_{g0} as the spec. volume of gas at the entrance. Then, the pressure ratio for the gas phase η_g is elaborated with Eq. (57) with the general pressure ratio η_b, the critical pressure ratio of the gas phase $\eta_{g,crit}$ (with Eq. (58)) which is calculated by help of the isentropic exponent κ. Also a factor k is defined by Eq. (59) with $\dot{x}_0$ (vapour mass flow fraction at entrance), the heat capacity of the liquid phase at the entrance c_{pl0} and the heat capacity of the gas phase at boiling c_{pgs}.

$$v_{gs} = \frac{1}{\eta_s}\frac{T_S}{T_0}v_{g0} \tag{56}$$

$$\eta_b \leq \eta_{g,crit} \dots \eta_g = \eta_{g,crit}$$
$$\eta_b > \eta_{g,crit} \dots \eta_g = \eta_b \tag{57}$$

$$\eta_{g,crit} = \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}} \tag{58}$$

$$k = \frac{\dot{x}_0}{\dot{x}_0 + (1-\dot{x}_0)\frac{c_{pl0}}{c_{pgs}}}; k \in (0 \dots 1) \tag{59}$$

Now three factors Γ_1, Γ_2 and Γ_3 are calculated by using Eq. (60), with help of factor Ω (Eq. (61)) and $\dot{x}_{eq,th}$.

$$\Gamma_1 = \frac{(1-\omega)}{2}[\Omega - (1-\omega)]$$
$$\Gamma_2 = \frac{\Omega}{2}\left[\frac{2}{\eta_s} - 1 - \frac{\dot{x}_0}{\omega_{Flash}}\frac{v_{gs}-v_{l0}}{v_0}(1+\omega)\right] + \frac{1}{2}a_{TR}(\omega-\omega_{Frozen})(1+\omega) - \omega^2 + 1 \tag{60}$$
$$\Gamma_3 = \omega\left(\frac{1}{\eta_s}-1\right) - \frac{1}{2} + \left[\frac{\dot{x}_0}{\omega_{Flash}}\frac{v_{gs}-v_{l0}}{v_0} - \frac{a_{TR}(\omega-\omega_{Frozen})}{\Omega}\right]\left(\frac{\Omega}{2}-\omega^2\right)$$

$$\Omega = a_{TR}\frac{v_0}{v_{gs}-v_{l0}}\,{\omega_{Flash}}^2\left(\dot{x}_{eq,th}\right)^{a_{TR}-1} \tag{61}$$

Then, the critical pressure ratio η_{crit} is derived from Eq. (62). Calculation does not allow complex values, therefor the value under the root must not be negative. Results must be in the range of $\eta_b \leq \eta_{crit} \leq 1$.

$$\eta_{crit} = \eta_s\left(1 + \frac{1}{2\Gamma_1}\left(\Gamma_2 \pm \sqrt{{\Gamma_2}^2 - 4\Gamma_1\Gamma_3}\right)\right);\ \eta_{crit} \in (\eta_b \dots 1) \tag{62}$$

The calculation of η_{crit} (equals η_{th}) needs an iteration to find a destiny value, since the pressure ratio η_{th} is not known at the beginning. VDI-Wärmeatlas [2] (chapter L2.4) suggests for the starting value to be in the range of $\eta_b \leq \eta_{th} \leq 1$. Iteration has been achieved by using a secant method which needs results from three preceding iteration steps $\eta_{th,i}$, $\eta_{th,i-1}$ and $\eta_{th,i-2}$, see Eq. (63). A precision of 5 decimal places is achieved, $abs(\eta_{th,new} - \eta_{th,old}) \leq 10^{-5}$.

$$\eta_{th,i+1} = \eta_{th,i-1} - 0.5(\eta_{th,i-1} - \eta_{th,i}) \frac{(\eta_{th,i-1} - \eta_{th,i-2})}{(\eta_{th,i-1} - \eta_{th,i}) - (\eta_{th,i-2} - \eta_{th,i-1})} \tag{63}$$

With it, the dimension-less mass flow C is calculated with Eq. (49). By transforming Eq. (47) and Eq. (48) into Eq. (64), the mass flow $\dot{M}$ is elaborated.

$$\dot{M} = C A_{th} \sqrt{2 \frac{p_0}{v_0}} \tag{64}$$

3.4.1 Boundary conditions

The diameter of the opening should be known, also the amount of steam of the water / steam mixture. Wellensiek [9] suggests using twice the diameter of a rising tube of the membrane wall to consider the maximum possible mass flow.

Then the pressure of the mixture and of the flue gas side are needed. Finally, the temperature of the mixture should be known.

3.4.2 Initial values

The thermodynamic values, such as the heat capacity of the liquid and gaseous phase inside the pot or the pressure of saturated state or the critical pressure, just to name a few, must be known. As described before, an initial value of the pressure ratio η_{th} in the range of $\eta_b \leq \eta_{th} \leq 1$ is necessary.

3.5 Explosion flap

A mechanical flap which is attached to the flue gas channel serves as overpressure safety valve to avoid an overload of the combustion chamber. The location is depending on where the highest pressure at a tube rupture is present.

Erler [6] gives an example how to model such a flap: An impulse balance is set up to describe the pressure difference between the flue gas side p_{FG} and the ambience p_{Am} over the explosion flap opening with Eq. (65). Herein the flow through the flap is defined with the velocity of flue gas through the flap v_{FG}, the friction coefficient of the flap ζ_{EF} (when fully opened) and the density of the flue gas ρ_{FG}. The second term implies the acceleration of the air inside a possible exhaust channel with length l_{EF}. Here, this exhaust channel is not considered. The mass flow

of the flue gas exiting through the explosion flap $\dot{m}_{EF}$ is defined at Eq. (66) with the velocity v_{FG}, the cross section of the flap opening A_{EF} and the density of flue gas ρ_{FG}. Rearrangement of Eq. (65) leads to Eq. (67). To consider the inertia of mass of the explosion flap, two measures are taken: 1. The mechanism of the flap is activated after a certain activating pressure $p_{EF,act}$ is reached. 2. The cross section A_{EF} is gradually opened and defined by a time t_{EF} in which the flap is fully opened.

$$p_{FG} - p_{Am} = \zeta_{EF}\rho_{FG}\frac{v_{FG}^{\,2}}{2} + \rho_{FG}l_{EF}\frac{d\,v_{FG}}{dt} \tag{65}$$

$$\dot{m}_{EF} = v_{FG}A_{EF}\rho_{FG} \tag{66}$$

$$\dot{m}_{EF} = \sqrt{\frac{2A_{EF}^{\,2}\rho_{FG}(p_{FG} - p_{Am})}{\zeta_{EF}}} \tag{67}$$

3.5.1 Boundary conditions

The pressure of the flue gas channel and the ambience are known. The cross section, the density of flue gas, the friction coefficient and the activating pressure, the opening time are defined as well.

3.5.2 Initial values

Before starting the opening process, the mass flow is set to $\dot{m}_{EF} = 0$.

4 Results

Since modelling the problem has been achieved, the simulation needs to be defined as follows: Simulation can be executed for steady operation, operation with a tube rupture and with an additional counter measure (explosion flap). To analyze the position and cross section that has the best impact as a reduction of the pressure increase inside the combustion chamber, different locations at the flue gas channel and different cross sections of the flap have been tested.

Data of an existing steam generator was provided for testing reasons. The information is taken to countercheck the results of the steady solution of the simulation. The referred plant called Kogeban, is a biomass boiler in France which is designed for natural circulation. All necessary dimensions of the boiler room, all water piping and process data was made accessible by the Institute of Energy Systems and Thermodynamics (of Technical University of Vienna) for research purposes and used as initial values.
To simplify the user's experience, an excel file was created. There, the input of all necessary data can be done and all important results are displayed in diagrams, see Fig. 13.

The diagrams shown in the following chapters have been created with MATLAB.

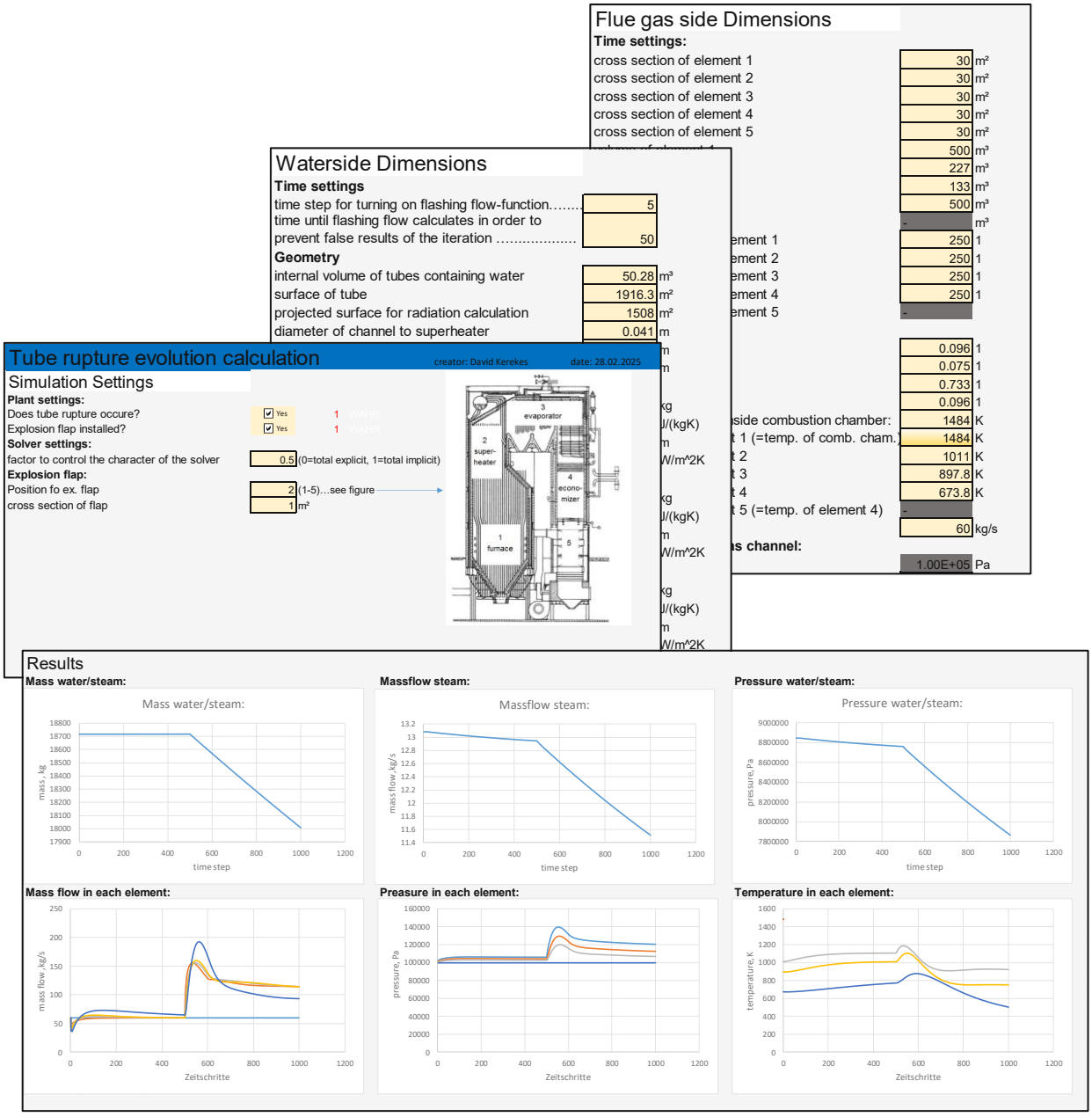

Fig. 13 excel table for in- and output of TRPEC data

4.1 Initial values

The following data is taken from the documentation of the reference plant:

For the flue gas side, the inside dimensions of the combustion chamber and the following flue gas draft are used.

Table 1: Boiler parameters used as initial values

Parameter:		Amount:	Unit:
Produced steam amount $\dot{m}_S$		13.0805	kg/s
Pressure inside evaporator p_{evap}		88.45	bar (abs)
Circulation number U		16.23	-
Cross section of all control volumes A (average value)		30	m²
Size of control volume			
	1	-	-
	2	500	m³
	3	227	m³
	4	133	m³
	5	500	m³
	6	-	-
Flue gas temperature			
	1	1484	K
	2	1484	K
	3	1011	K
	4	897.8	K
	5	673.8	K
	6	673.8	K
Flue gas mass flow		60	kg/s
Friction coefficient of all control volumes ζ (average value)		250	-
Total surface of all water containing segments A_{CC}		1916.3	m²
Projected total surface of all water containing segments $A_{CC,proj}$		1508	m²

Continue of Table 1:

Parameter:		Amount:	Unit:
Heat flow between flue gas side and water side			
	1	-	-
	2	32.279	MW
	3	29.750	W/m²
	4	7.150	W/m²
	5	14.100	W/m²
	6	-	-
Volume of all water / steam containing volumes		50.28	m³

Note: The initial node (Nr. 1) is a fictional one, with a mass flow value serving as boundary condition for the second one. The last node (Nr. 6) is also a fictional one, defining a pressure level for keeping a stable flow of flue gas.

The used fuel is biomass (wood and other organic natural waste). For no further information of the properties of the fuel is given, the resulting fuel gas properties must be assumed as follows: Combustion of 1 kg air-dried wood with an air excess of $\lambda = 2$ leads to the following flue gas composition (mass ratio values):

$\xi_{CO2} = 9.6$ m% $\quad CO_2$
$\xi_{H2O} = 7.5$ m% $\quad H_2O$
$\xi_{N2} = 73.3$ m% $\quad N_2$
$\xi_{O2} = 9.6$ m% $\quad O_2$

By using the composition, the specific gas constant R_{FG} can be derived according to FDBR - Handbuch [10] with Eq. (68).

$$R_{FG} = (0.2869 - 0.0746\,\xi_{CO2} + 0.1745\,\xi_{H2O})\,10^3 \tag{68}$$

The specific heat capacity (isobaric $c_{P,FG}$ and isochor $c_{V,FG}$) can be calculated by using the known capacities of each flue gas element, implementing Eq. (69) and (70).

$$c_{P,FG} = \left(c_{P,N2}\,\xi_{N2} + c_{P,O2}\,\xi_{O2} + c_{P,CO2}\,\xi_{CO2} + c_{P,H2O}\,\xi_{H2O}\right) 10^3 \tag{69}$$

$$c_{V,FG} = c_{P,FG} - R_{FG} \tag{70}$$

With the gas constant of the flue gas R_{FG} and of water R_{WS}, the gas constant of the mixture R can be derived with Eq. (71).

$$R = R_{FG}\,(1 - \xi) + R_{WS}\,\xi \tag{71}$$

For calculating the water side of the model, data from a water-steam table is necessary. The open-source library “CoolProp” is used.

4.2 Conditions and assumptions

The incident of a tube rupture is analysed within the first couple of seconds after the rupture occurs. Within this short time the safety monitoring of the plant has no effect on the boiler yet. Therefor the following assumptions are defined as follows:

- The amount of steam at the place of the tube rupture equals the steam amount of the boiler.
- The combustion chamber pressure is close to the ambience pressure. The pressure level of all remaining control volumes is calculated by a defined friction coefficient ζ to get a linear pressure decrease towards the ambience.
- When a tube rupture occurs, the flue gas blower stays in operation. Therefor the flue gas mass input remains.
- When a tube rupture occurs, the feed water pump stays in operation. Therefor the feed water input remains.
- When a tube rupture occurs, the steam turbine remains. Therefor the steam mass output remains (but not as constant value).

4.3 Steady operation

The simulation of a steady operation of the boiler is shown. No tube rupture occurs.

time period: 10 sec.

time step width: $\Delta t = 0.01$ sec.

factor for solver (explicit/implicit): $f = 0.5$

Calculation time (orientation): 12 sec.

The courses of mass flow, pressure and temperature of each control volume are showing, that a stable solution is found after 150 time steps. As can be seen in Fig. 14, even though the initial values of mass flow were the same for all control volumes (corresponding to the mass balance), iteration resulted in alterations for the first 150 time steps but converged to a steady state. This can be explained by the inertia of the system. The bigger the control volume, the longer it takes to converge. The last element of the flue gas channel still has some overswinging which lasts around 1000 time steps. Values for each control volume are displayed with different colors. Since the sixth element corresponds to the ambience, no mass flow or pressure is displayed. Similar for the pressure and the temperature: steady state is reached for each control volume at its own pressure and temperature level after 150 time steps, see Fig. 15 and Fig. 16.

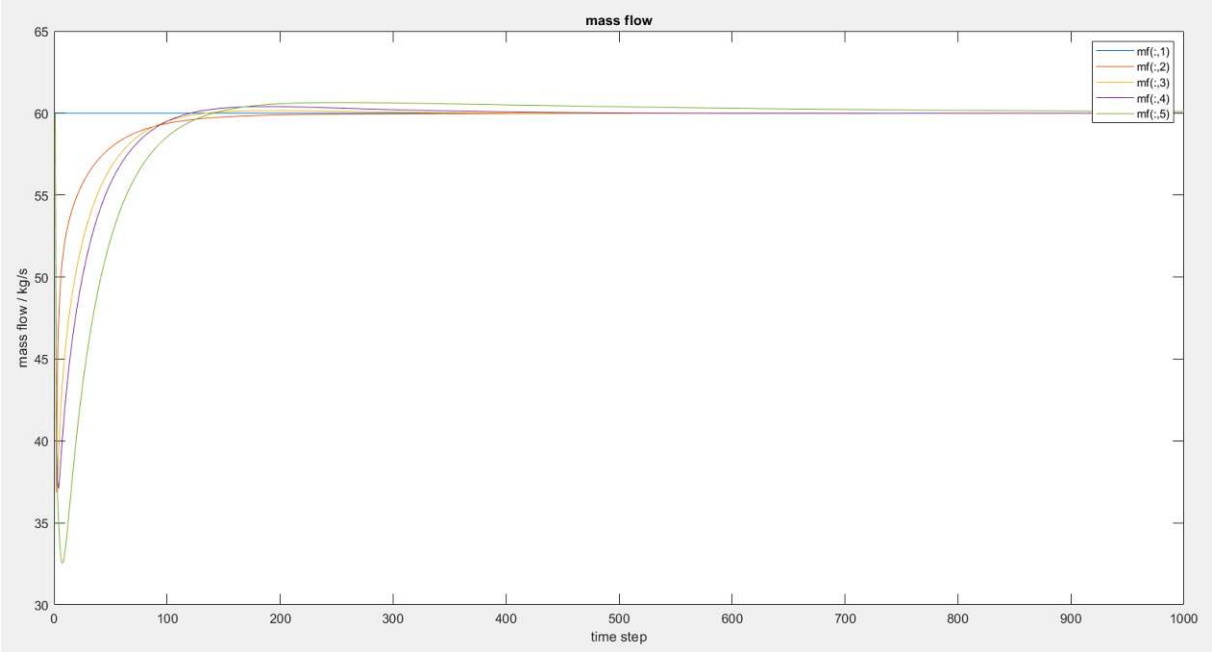

Fig. 14 mass flow of flue gas channel

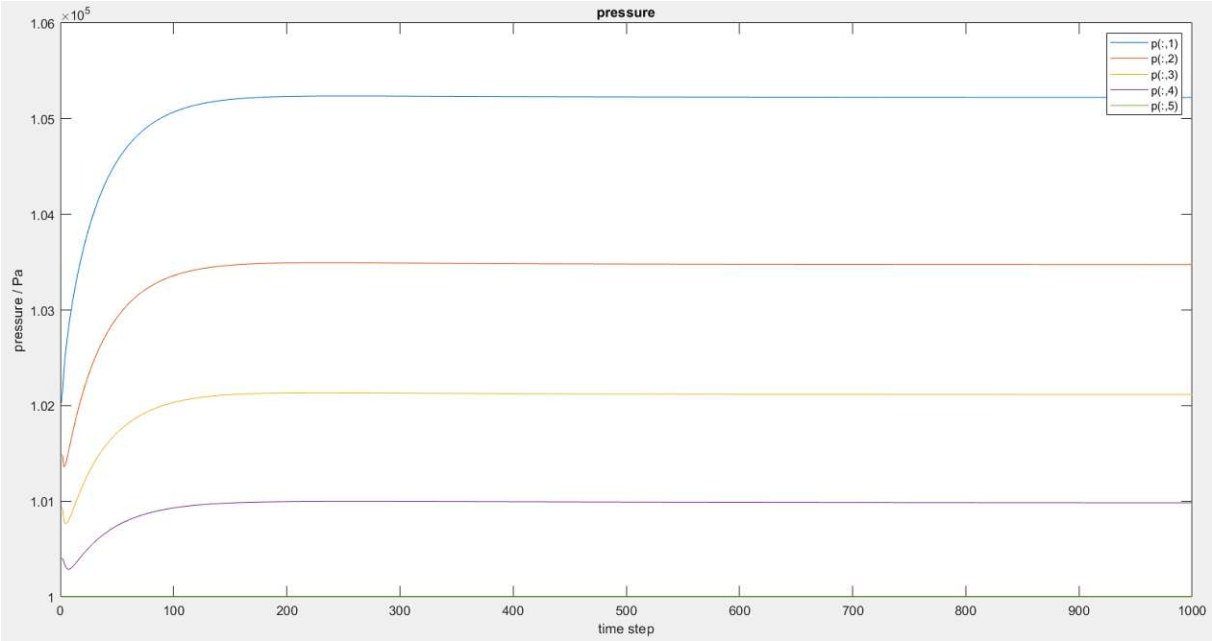

Fig. 15 pressure of flue gas channel

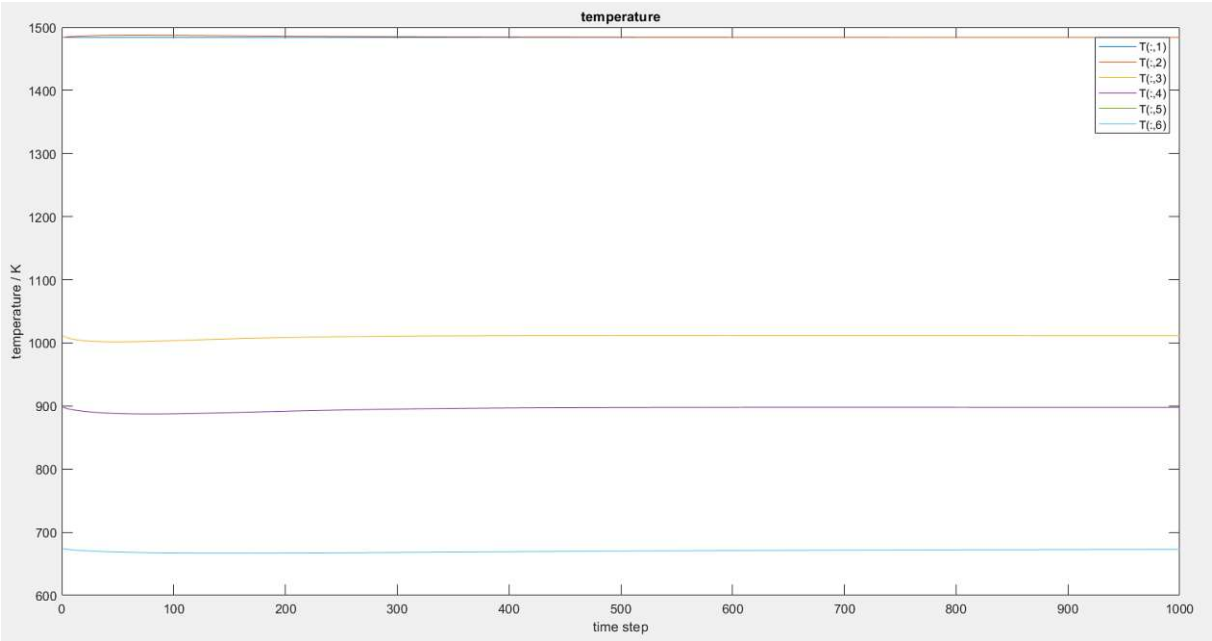

Fig. 16 temperature of flue gas channel

4.4 Operation with tube rupture

time period: 10 sec.

time step width: $\Delta t = 0.01$ sec.

factor for solver (explicit/implicit): $f = 0.5$

Calculation time (orientation): 13 sec.

The tube rupture process starts 5 seconds after simulation start in order to prevent a disruption of the tuning phase into steady state condition. As can be seen mass flows, pressure and temperature increase, see Fig. 17, Fig. 18 and Fig. 19.

The pressure inside the combustion chamber (control element 1) rises from 1 barg to 1.77 barg (1* 10^5 Pa to 1.77* 10^5 Pa), resulting in a pressure increase of 770 mbarg. The maximum pressure is reached 1.5 seconds after the rupture occurs. This pressure would be too high for the membrane wall and would lead to bulging of the boiler wall.

The pressure in the following control volumes is increasing respectively.

The mass flows of each control volume rise as well. The last control volume has a significant high increase which can be explained by the impulse effecting the big volume of this calculation point to answer with an increased mass flow.

The pressure of water /steam inside the pot is decreasing respectively, see Fig. 20. The mass flow out of the ruptured tube is calculated with a maximum value of 148 kg/s, see Fig. 21

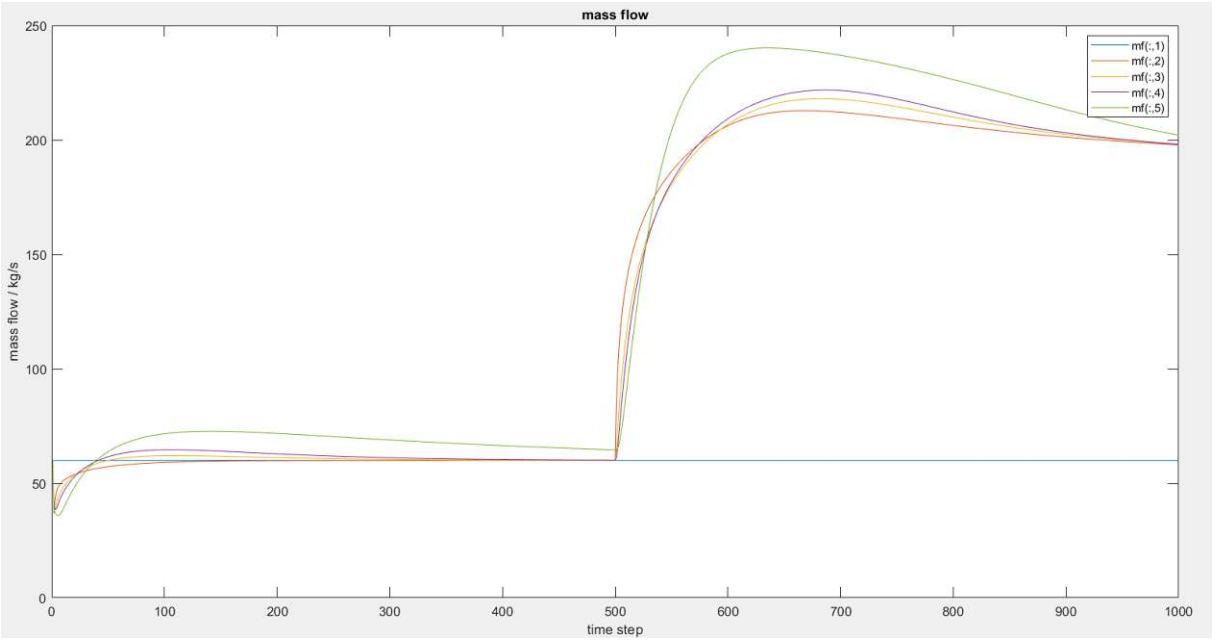

Fig. 17 mass flow of flue gas channel

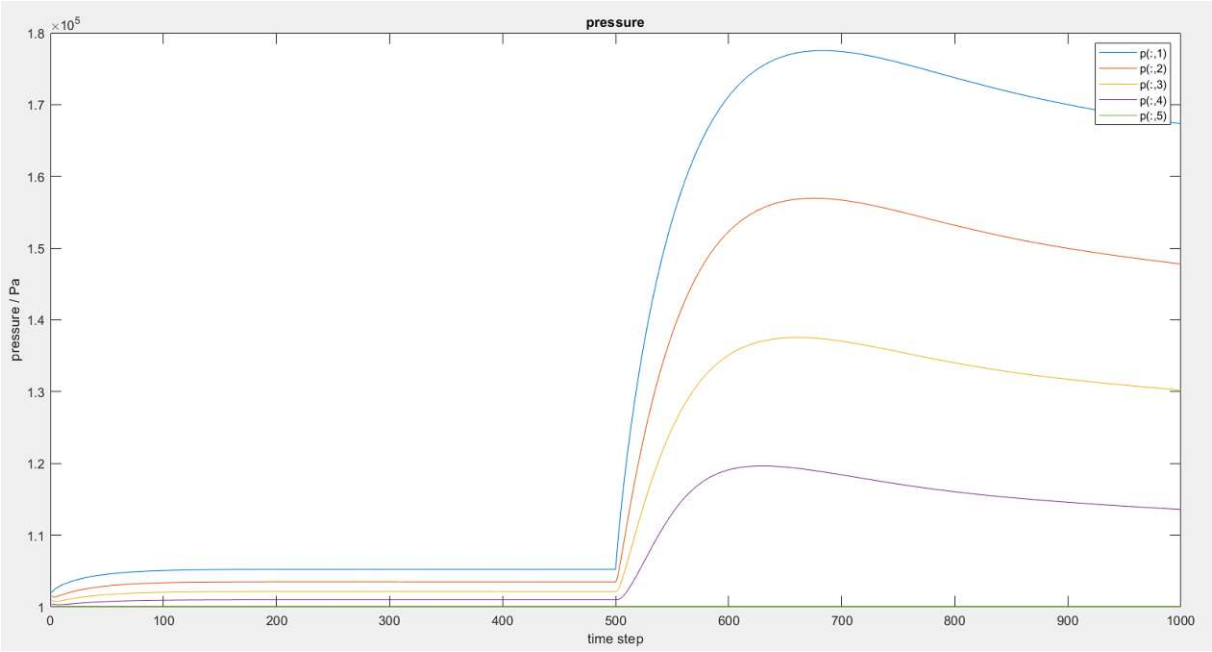

Fig. 18 pressure of flue gas channel

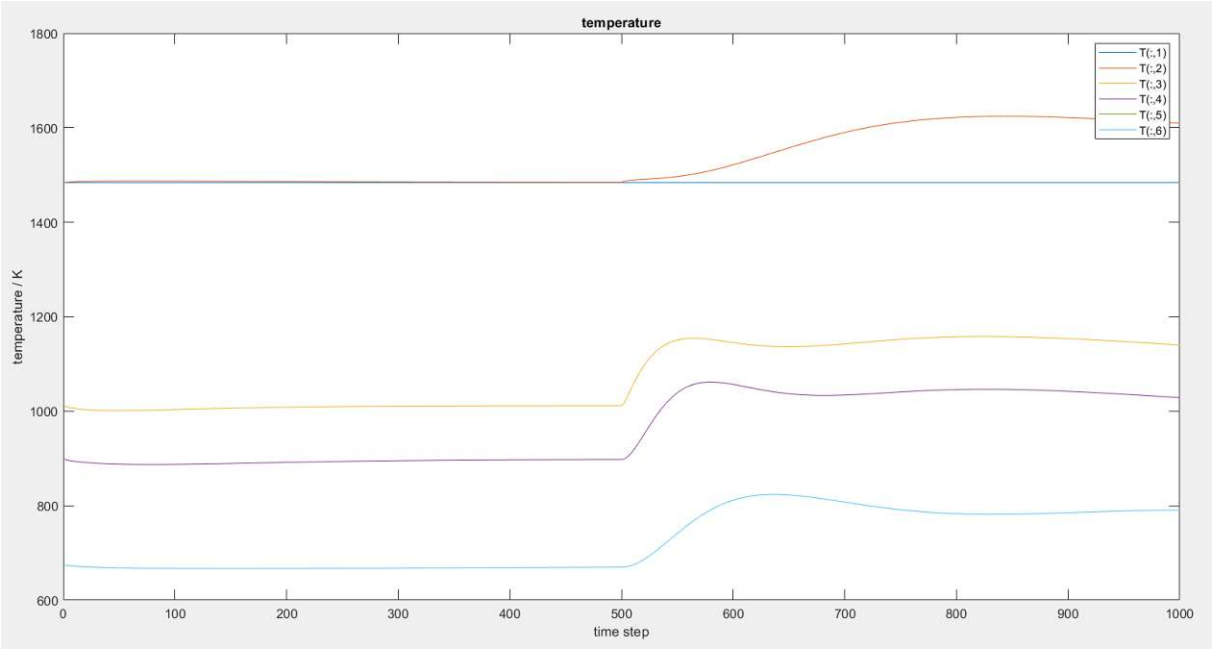

Fig. 19 temperature of flue gas channel

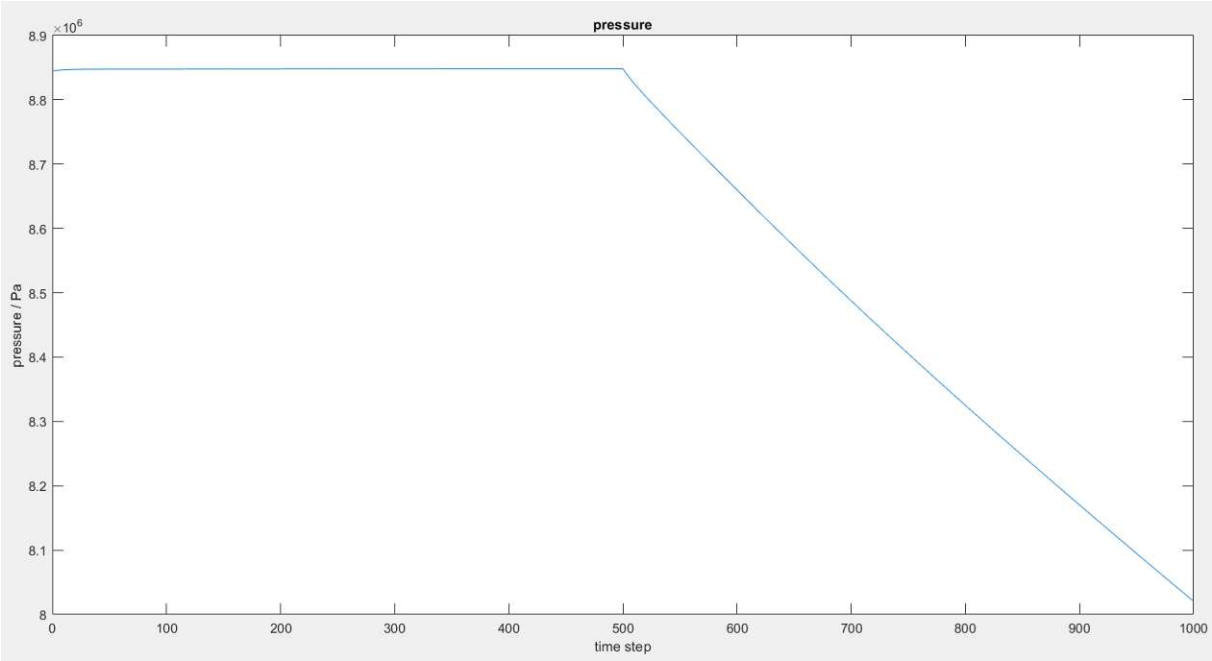

Fig. 20 pressure of water / steam side

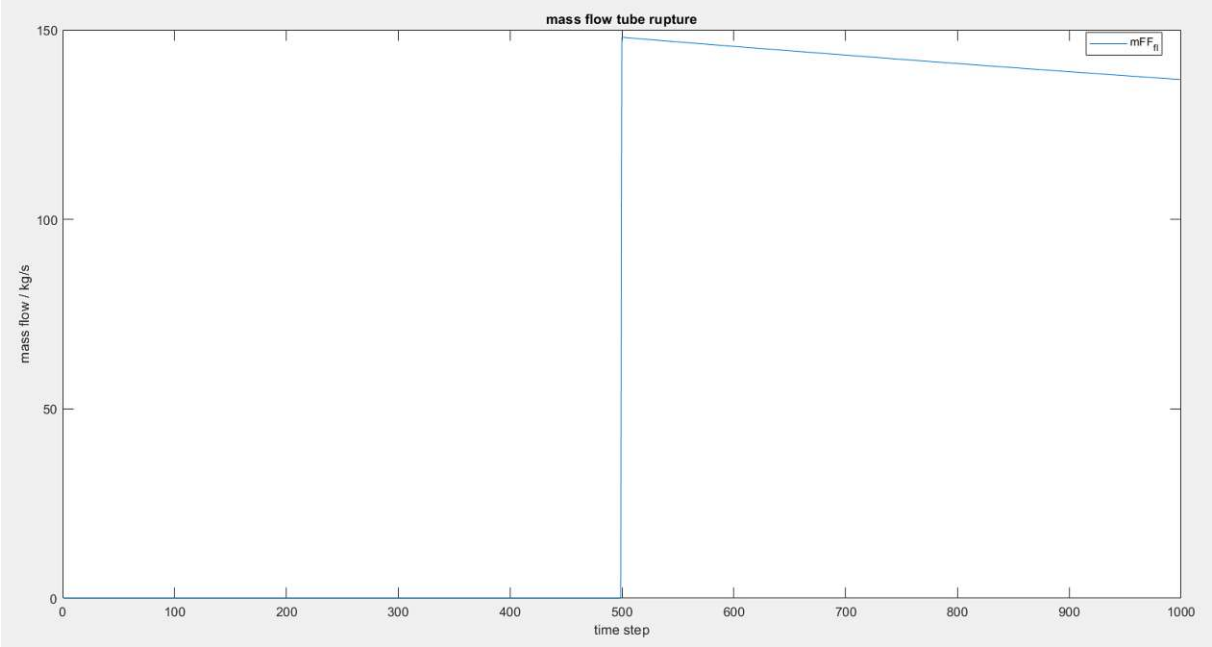

Fig. 21 mass flow out of ruptured tube

4.5 Tube rupture with explosion flap

time period: 10 sec

time step width: Δt = 0.01 sec

factor for solver (explicit/implicit): f = 0.5

Calculation time (orientation): 17 sec

In the following chapters, different locations and flap cross sections are tested.

4.5.1 Explosion flap at evaporator, 1 m²

Place of explosion flap: Evaporator

Cross section of explosion flap: 1 m²

Different locations of the explosion flaps are tried out. First location is the evaporator. When the explosion flap is installed at the membrane wall itself, the geometry of the tubing must be led around the flap. As described before, the flap begins to open when the flue gas pressure reaches an activating pressure $p_{EF,act}$. Its opening cross section A_{EF} is enlarging until a time t_{EF} is reached and the flap is fully opened.

As can be seen in Fig. 22, the mass flow of each control volume rises to a lower value than without the explosion flap. A maximum flow rate of 195 kg/s is reached at the last control volume and is reduced to 100 kg/s within 4 seconds. For the others, the mass flow is limited at around 150 kg/s and decreases to around 120 kg/s.

Here as well, the mass flow of the last control volume is higher than the others due to the effects, as described earlier. The maximum pressure in the combustion chamber (first control volume) is reduced from 1.77 barg (without explosion flap) to around 1.4 barg, see Fig. 23. All other pressure values are reduced respectively. The mass flow out of the explosion flap is displayed at Fig. 24 and has a maximum value of 117 kg/s.

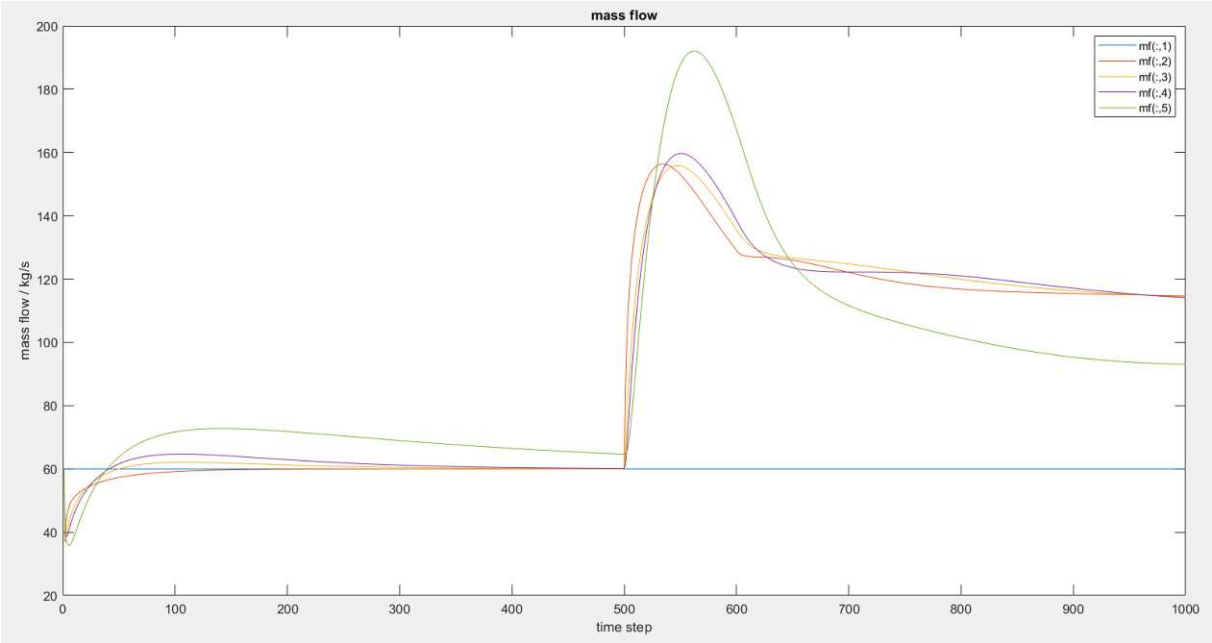

Fig. 22 mass flow of flue gas channel

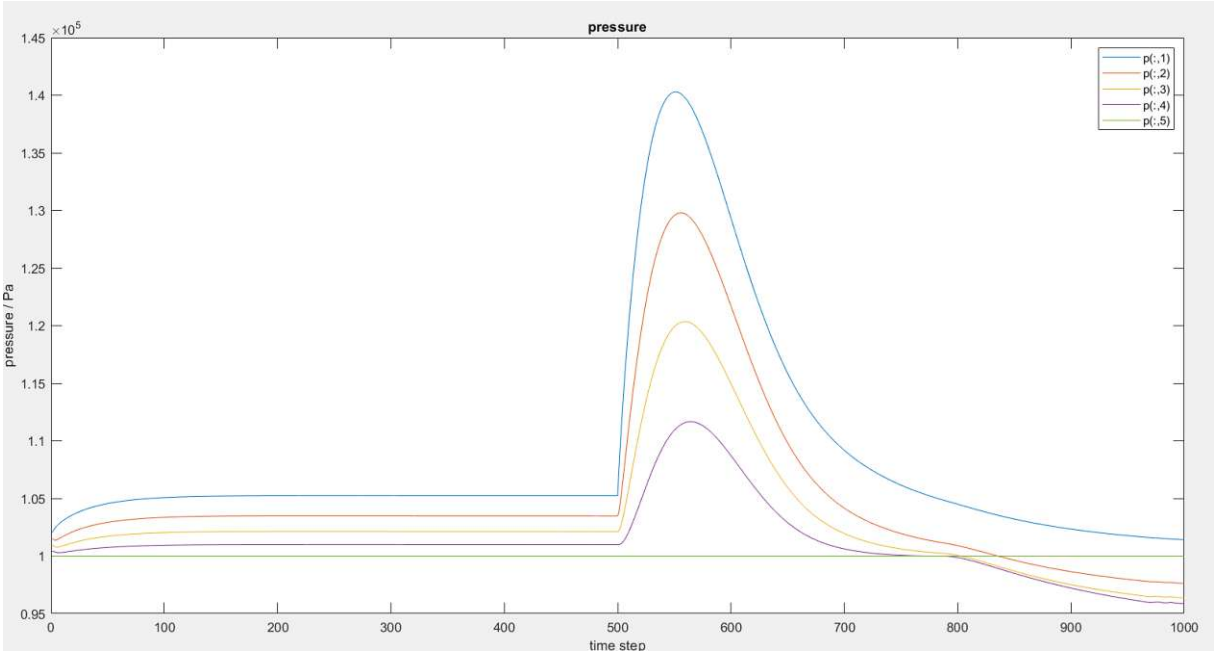

Fig. 23 pressure of flue gas channel, explosion flap at combustion chamber, 1 m² cross section

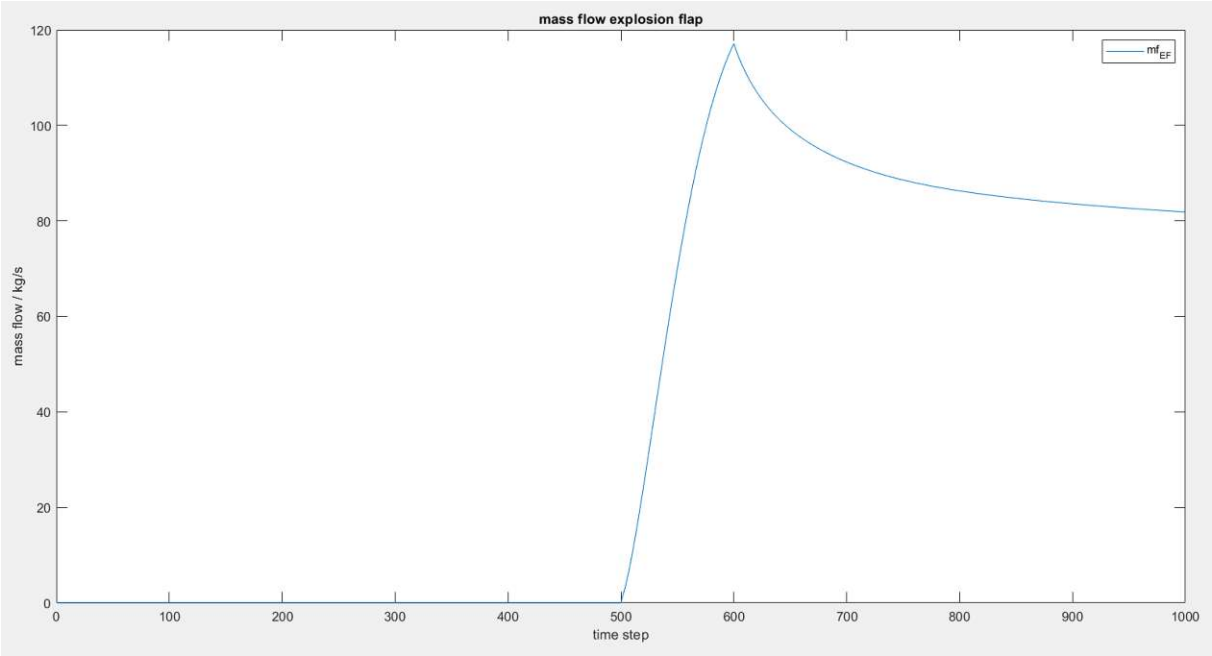

Fig. 24 mass flow out of explosion flap

4.5.2 Explosion flap at evaporator, 2 m²

Place of explosion flap: Evaporator

Cross section of explosion flap: 2 m²

Next, the cross section of the flap is increased. A pressure decrease to 1.32 barg (instead of 1.4 barg at 1m² flap cross section) is accomplished, as seen in Fig. 25.

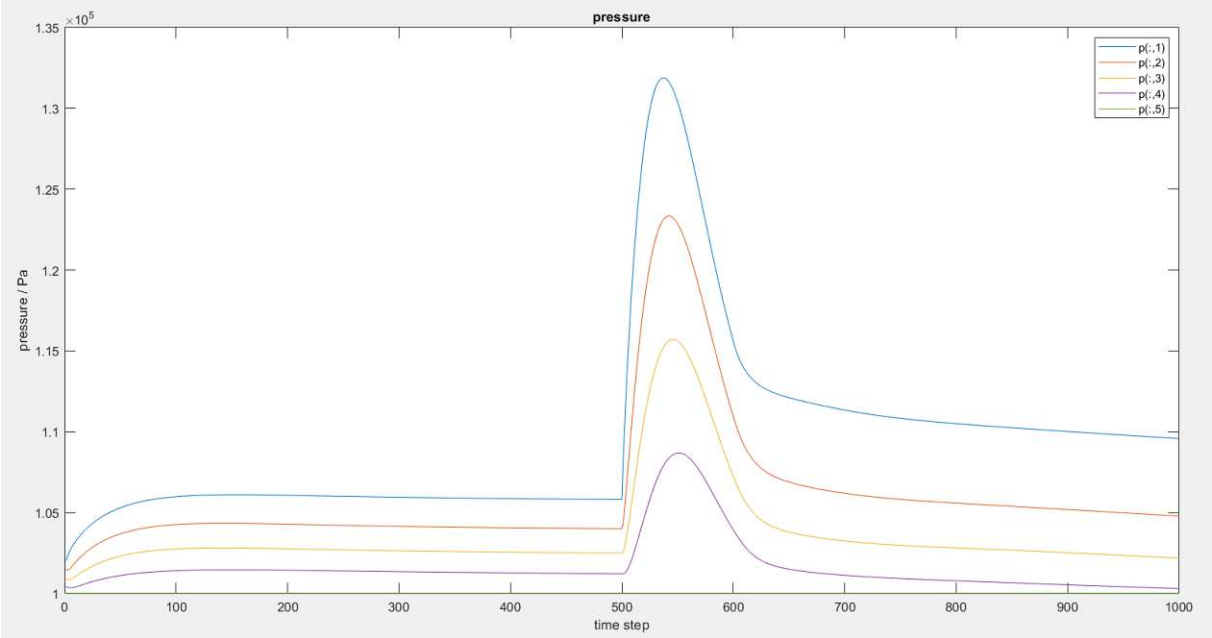

Fig. 25 pressure of flue gas channel, explosion flap at combustion chamber, 2 m² cross section

4.5.3 Explosion flap at super heater, 2 m²

Place of explosion flap: Superheater

Cross section of explosion flap: 2 m²

When using a flap with the same cross section but positioned near the super heaters, the pressure inside the evaporator decreases to 1.44 barg, as seen in Fig. 26. This implies that the explosion flap should be placed somewhere between the combustion chamber and the super heater part, somewhere at the membrane wall but not further down the flue gas channel to assure highest possible pressure reduction.

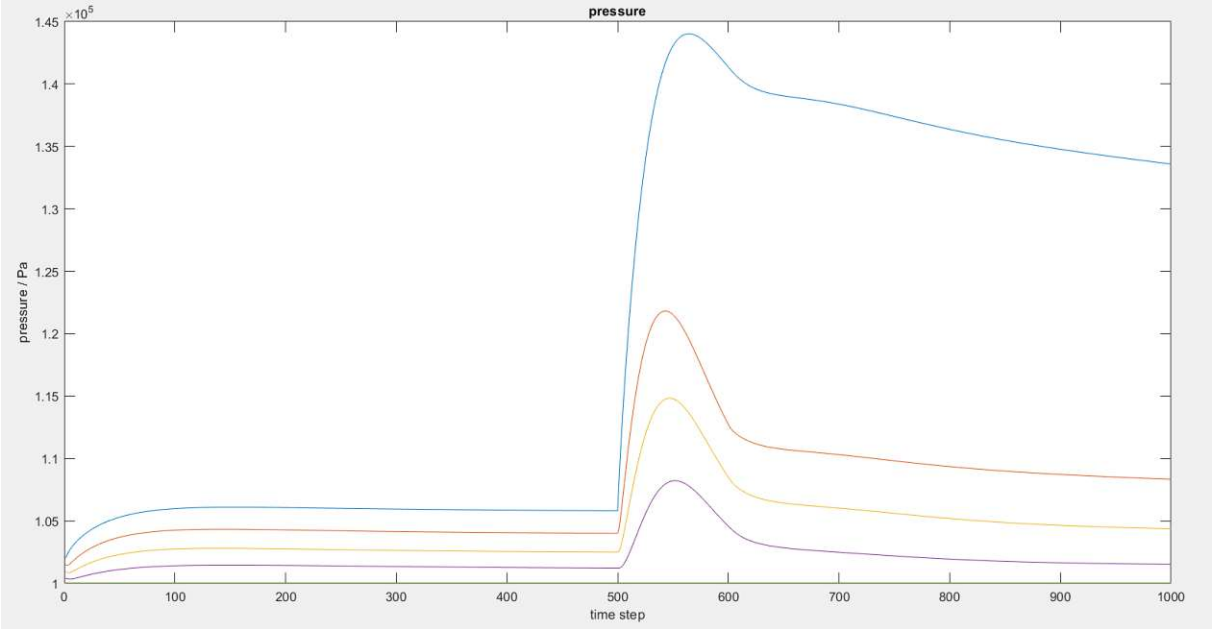

Fig. 26 pressure of flue gas channel, explosion flap at super heaters, 2 m² cross section

Conclusion

A program is written in MATLAB to be used as simple tool for simulating water tube boilers and the effect of a tube rupture and counter measures. An Excel file is available to serve as in- and output interface.

With described models of the flue gas side and the water / steam side, a stable operation of an existing plant is achieved.

With modelling a tube rupture according to a two-phased model, a possible pressure increase inside the boiler room can be predicted.

Trying out different positions of the explosion flap showed that the maximum effect of pressure relief is achieved when placing it at the membrane wall near the evaporator and the super heaters. The cross section of 1 to 2 m² have been tested. Of course, the bigger the flap, the more mass flow can exit the high-pressured zone. But to meet technical realistic expectations, 1-2 m² are sufficient for assuring a quick and safe decrease of pressure.

The TRPEC simulation can serve for designing new water tube boilers or as an analyzing tool for existing ones. Many dimension factors and process values are needed as input and must be available or assumed (for new design).

Process values such as pressure, mass flow and temperature of every part of the flue gas channel can be analyzed. Also, the process values of the water / steam side are calculated. By modelling the mass flow going out the ruptured tube according to the “Flashing flow” model, introduced by VDI-Wärmeatlas [2] (chapter L2.4), the whole problem is considered mathematically.

This simulation can also be used for further applications for future research. Since the structure is simple, additional features can be programmed as needed.

Note from author:

If there is interest in the MATLAB code, the author can be contacted.

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