LAPPEENRANNAN TEKNILLINEN KORKEAKOULU
LAPPEENRANTA UNIVERSITY OF TECHNOLOGY

TIETEELLISIÄ JULKAISUJA 27
RESEARCH PAPERS

ESA K. VAKKILAINEN

Offdesign Operation of Kraft Recovery Boiler

ISBN 978-952-214-753-0 (PDF)

LAPPEENRANNAN TEKNILLINEN KORKEAKOULU
LAPPEENRANTA UNIVERSITY OF TECHNOLOGY

UDK 621.181
672.082
519.876
536.2

TIETEELLISIÄ JULKAISUJA
RESEARCH PAPERS

27

ESA K. VAKKILAINEN

Offdesign Operation of Kraft Recovery Boiler

Thesis for the degree of Doctor of Technology to be presented with due permission for public examination and critisism in the Auditorium 2 at Lappeenranta University of Technology (Lappeenranta, Finland) on the 1st of October, 1993, at noon

LAPPEENRANTA
1993

ISBN 951-763-763-2

ISSN 0356-8210

Vakkilainen, Esa K., Offdesign operation of kraft recovery boiler.
UDK 621.181 : 519.876 : 536.2 : 676.08
Key words: steam generators, recovery boiler, simulation, heat transfer

ABSTRACT

A model to solve heat and mass balances during the offdesign load calculations was created. These equations are complex and nonlinear. The main new ideas used in the created offdesign model of a kraft recovery boiler are the use of heat flows as torn iteration variables instead of the current practice of using the mass flows, vectorizing equation solving, thus speeding up the process, using non-dimensional variables for solving the multiple heat transfer surface problem and using a new procedure for calculating pressure losses.

Recovery boiler heat and mass balances are reduced to vector form. It is shown that these vectorized equations can be solved virtually without iteration. The iteration speed is enhanced by the use of the derived method of calculating multiple heat transfer surfaces simultaneously. To achieve this quick convergence the heat flows were used as the torn iteration parameters.

A new method to handle pressure loss calculations with linearization was presented. This method enabled less time to be spent calculating pressure losses.

The derived vector representation of the steam generator was used to calculate offdesign operation parameters for a 3000 tds/d example recovery boiler. The model was used to study recovery boiler part load operation and the effect of the black liquor dry solids increase on recovery boiler dimensioning.

Heat flows to surface elements for part load calculations can be closely approximated with a previously defined exponent function. The exponential method can be used for the prediction of fouling in kraft recovery boilers.

For similar furnaces the firing of 80 % dry solids liquor produces lower hearth heat release rate than the 65 % dry solids liquor if we fire at constant steam flow. The furnace outlet temperatures show that capacity increase with firing rate increase produces higher loadings than capacity increase with dry solids increase.

The economizers, boiler banks and furnaces can be dimensioned smaller if we increase the black liquor dry solids content.

The main problem with increased black liquor dry solids content is the decrease in the heat available to superheat. Whenever possible the furnace exit temperature should be increased by decreasing the furnace height.

The increase in the furnace exit temperature is usually opposed because of fear of increased corrosion.

ACKNOWLEDGEMENTS

This work started in 1984 at the Lappeenranta University of Technology. The idea of using vector representation to decrease independent variables was invented then. It was not until 1987 that I had the chance to create the code to test my ideas. The code was finished during 1989 at my current employment the A. Ahlstrom Corporation, Ahlstrom Machinery, Recovery Boilers in Varkaus Finland.

I wish to express my sincere thanks to my supervisor professor Reino T. Huovilainen for his encouragement and guidance during this work. I am also grateful to Director of Recovery Boilers Harry Rickman for the special arrangements that enabled me to finish my dissertation.

The 'gang of 1984' in the Lappeenranta University of Technology, you encouraged in the growing of academic spirits. I thank Risto Leukkunen, Timo Talonpoika and Juha Pesari for many informal discussions and co-operation and Timo Hyppänen for enlightening pondering. The professors Seppo Korpela, Jaakko Larjola and Pertti Sarkomaa showed me what scientific work really is. Special thanks to Ms. Maisa Raatikainen who has always encouraged me to finish my graduate work and displayed enthusiasm that I would do so.

Varkaus, November 1992

Esa Vakkilainen

CONTENTS

TABLES

FIGURES

NOMENCLATURE

A	surface area, m^2
b	correction coefficient, -
C	heat capacity, J/K
c_p	specific heat capacity, J/kgK
d	diameter, m
e	damping parameter, -
G	thermal conductance, W/K
H	enthalpy, J
h	specific enthalpy, J/kg
k	tube roughness, m
L	length, m
l	dimensionless length, -
n	number of, -
p	pressure, Pa
q	flow, -/s
R	ratio of minimum heat capacity flow to maximum heat capacity flow, -
r	radius, m
s	radiation beam length, pitch, m
T	temperature, K
U	overall heat transfer coefficient, W/m^2K
w	flow velocity, m/s
x	torn iteration variable, mole ratio, -
y	design variable
z	number of transfer units, -

Greek symbols

α	heat transfer coefficient, W/m^2K
ϵ	ratio, -
ζ	loss coefficient, -
η	dynamic viscosity, Pa s
θ	total temperature difference, K
λ	heat conductivity, W/mK
ξ	friction factor, -
ρ	density, kg/m^3
Φ	heat flow, W

Superscripts

n	iteration
*	corrected

Subscripts

A	attemperating
a	air, arrangement
b	bend
c	convection
C	heat capacity
d	dynamic
e	external
eff	effective
ev	evaporative
ex	external
f	fouling, friction
fw	feedwater
g	gas
i	in, inside, inlet
i	index referring to elements
io	inlet and outlet
j	index referring to process units
l	laminar, longitudinal, losses
ln	logarithmic
m	mass
max	maximum
min	minimum
o	out, outside, outlet
pa	primary air
q	fuel, generation
r	radiation, row
s	steam, surface
sa	secondary air
t	tube, total, turbulent, transverse
ta	tertiary air
ts	tube side
w	wall

1. INTRODUCTION

A time consuming processes in the recovery boiler manufacture is the design phase. Typically tens of thousands of hours are used. Designing similar boilers is seldom possible. Some dozens of boilers are sold each year by any large boiler manufacturer. Similar-looking boilers are often designed for different fuel analysis and for different steam conditions.

Computers can be used to help in the design of modern recovery boilers. A kraft recovery boiler can be modelled by defining equations that describe it. Using appropriate program these equations can be solved and the performance of the recovery boiler evaluated. During the proposal phase of the kraft recovery boiler design these heat and mass balances must be repetitively solved.

Typically steam generator models have been used to solve the performance as a function of the main design parameters. The main parameters of recovery boilers depend on the allowable investment costs, desired heat and power generation efficiency. The main design variables vary frequently from project to project. Changes such as required flue gas outlet temperature, design pressure, main steam temperature and fuel analysis are fairly typical even during the proposal phase.

Today the offdesign performance of the kraft recovery boiler must be evaluated, for each proposal. This is computationally intensive. Typical offdesign calculation is performance during increase and decrease of boiler load from design load. Often other additional calculations are needed such as change of the main steam pressure or change of the air ratio. For retrofits the typical information is the effect of increase or decrease in area of a single heat transfer surface.

1.1 Steady state simulation

According to Kaijaluoto (84) the two common methods of modelling processes are equation oriented and procedure-oriented. Gundersen and Hertzberg (84) claim that most of the practical engineering programs use a procedure-oriented approach where we string separately modelled processes to form larger process modules. The solving of those models is done by iterative loops in a sequence determined by the process topology. The solution process is usually close to the way an engineer would solve the process by hand.

The procedure-oriented or modular approach is widely used in commercial steady state simulators (ASPEN Plus and PRO/II). Use of such software for steam generator design is limited because they lack suitable, ready made process units. Virtually all of

The steam generator models reported in the literature, PROSIM Fogelholm (89), He and Foster (86), Stecco (86), Sciubba and Su (86), RANKINE Somerton et al. (86) and Raiko (82), use iterative techniques to solve mass and energy balances.

The procedure-oriented approach is used also in the steam generator manufacturers' design programs. Examples are Deutche Babcock Sonnenchein (82), CE Robinson and Shiue (87), IVO Asikainen (83) and Pyropower Reese and McHugh (87). The origin of those programs was the automation of manual design procedures.

The main disadvantages of the procedure-oriented method are according to Kaijaluoto (84); Multiple nested iteration sequence leads often to conversion problems. For each iterative calculation, a separate iteration algorithm has to be included in the code. Suitable convergence tolerances for the nested iteration loops must be found. Rigid information structure restricts the use of the programs to only a few applications. Direction of information can be different from direction of iteration.

The newer method is the equation-oriented approach, where processes are modelled by a large set of linear and nonlinear equations describing the process units and the process topology. Suitable numerical methods are used for solving the equations.

There are several models that use matrix operations for the solving of the mass, energy and exergy balances; CASCAN El-Masri (86), Rosen and Scott (80), GAUDEAMO Valero (86) and Vakkilainen (86). A good example of the equation-oriented steam generator model is a model by Nishio (77), that uses linear programming.

The main drawbacks in the models using matrix operations are the nonlinearity of physical properties of gases, water and heat transfer equations. These programs must use linearization and several steps of iterative matrix solving. Perez (90) states that large matrices require large amounts of memory.

1.2 Recovery boiler modelling

There are only a few recovery boiler models presented in the literature. The most notable one is the model of the University of Idaho by Shiang (86). Most of the work has been concentrated to finding proper operating practices. See for example Haynes et al. (88) and Lundborg et al. (92). Adams (87) presents an offdesign study of the effect of black liquor solids content on the flue gas temperatures.

In the University of Idaho model the heat transfer calculations are very limited. Sootblowing, various flow paths of the flue gases and radiative transfer through surfaces is not accounted for. In the cases presented process units with multiple heat transfer surfaces were often calculated as a single heat transfer surface process unit.

One of the main research areas in the recovery boiler modelling has been the behavior of the recovery boiler furnace. Detailed flow modelling has improved our understanding of the phenomena in the furnace. Such studies have been done by Horton and Vakkilainen (93) and Karvinen et al. (91) Also performance of boiler components such as superheaters has been investigated. See for example an article by Vakkilainen et al. (91). The use of detailed flow programs for design of recovery boilers is limited because design and performance analysis requires moderate accuracy without a detailed flow field.

1.3 Offdesign model for recovery boilers

The aim of this work was to create a program capable of handling efficiently the offdesign calculations of recovery boilers. Computer models of steam generators and recovery boilers have been presented before. Typically they were designed to solve the heat and mass balances for one single case. The stimulus of this program was to produce results more quickly and assure the quality of engineering. According to Winter (90) these are the major challenges in computer aided process engineering.

To model kraft recovery boilers, the following new features will be presented

1. Usage of the heat flows as torn iteration variables instead of the current practice of using the mass flows.
2. New equation solving scheme using vectors to decrease the number of independent variables in heat transfer, mass and energy balances. Gundersen (90) presents decomposition algorithms and describes their use. Instead of using a algorithm for decomposition it was decided to try to represent the governing equations in a new way.
3. Usage of near analytical solution of the multiple heat transfer surface case.
4. Pressure losses calculation is speeded up by a new procedure and use of linearization.

Examples of the vectorized model of the steam generator are presented. In the vectorized model the heat transfer surfaces are connected one after another in a string. The same fluid flows through all of the connected heat transfer surfaces. Vectorizing is one form of decomposition of governing equations.

The dimensionless equations for the single surface heat exchangers are derived for the heat exchange case with multiple heat transfer surfaces. In the kraft recovery boiler superheaters the heat transfer to membrane tube walls, roof tubes and screens requires solution of multiple heat transfer at the same time.

For the offdesign study the effect of load on the kraft recovery boiler gas side temperatures is shown. The derived vector representation of the steam generator will be used to study the problem of estimating heat flows for a change of load case. The same example recovery boiler will be used to study the effect of the black liquor dry solids increase on recovery boiler dimensioning.

2. OFFDESIGN PROGRAM CRITERIA

The proposal phase is the first step in the recovery boiler design. During the proposal phase the main dimensions of the recovery boiler are chosen. The operation of the boiler with required fuels and loads must be calculated. The proposal phase design process of a recovery boiler can be divided into four separate stages as shown in *Figure 1.*

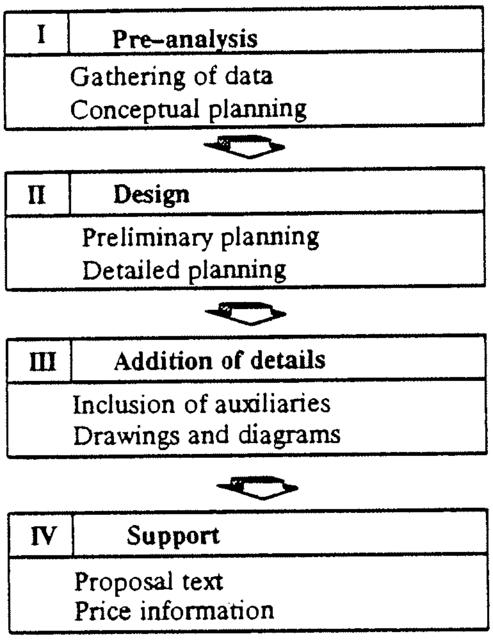

Figure 1. Stages in recovery boiler design at proposal phase

In the beginning of the pre-analysis stage, the data given by the customer is analyzed and checked. Fuel properties are checked for each of the fuels of the recovery boiler. Properties of unspecified variables input mass flows and air temperatures are fixed.

After sufficient input data has been gathered, the main variables for the boiler are chosen; main steam flow, main steam pressure, main steam temperature and required flue gas exit temperature.

With this data the overall mass and energy balances are calculated. Then the boiler type with the water / steam circulation type and the heat transfer surface order is chosen. Finally the preliminary lay-out is created.

The pre-analysis phase is followed by the design phase. With data from preliminary analysis the designer sizes the main equipment. All the time that the design proceeds, further input from the designer is demanded. The desired input might be as simple as accepting the data calculated by the computer but might lead to the abandonment of the chosen boiler look. Giving input means that the execution of the design programs stops until the desired data is given.

During the design phase the optimum main dimensions of each heat transfer surface must be determined based on the chosen type of operation. This means calculating the boiler operation with different input variable values. The results are verified and the best possible configuration is iteratively searched.

In the third stage we size the auxiliary equipment. The operation of the auxiliary equipment, valves, blowers and electrostatic precipitators is checked against previous assumptions in the design phase. During this phase the final lay-out and instrumentation drawings are made.

The remaining task is to record all this information and determine the price for the desired equipment. The proposal text contains the descriptions of each piece of equipment, used materials and the main operating values.

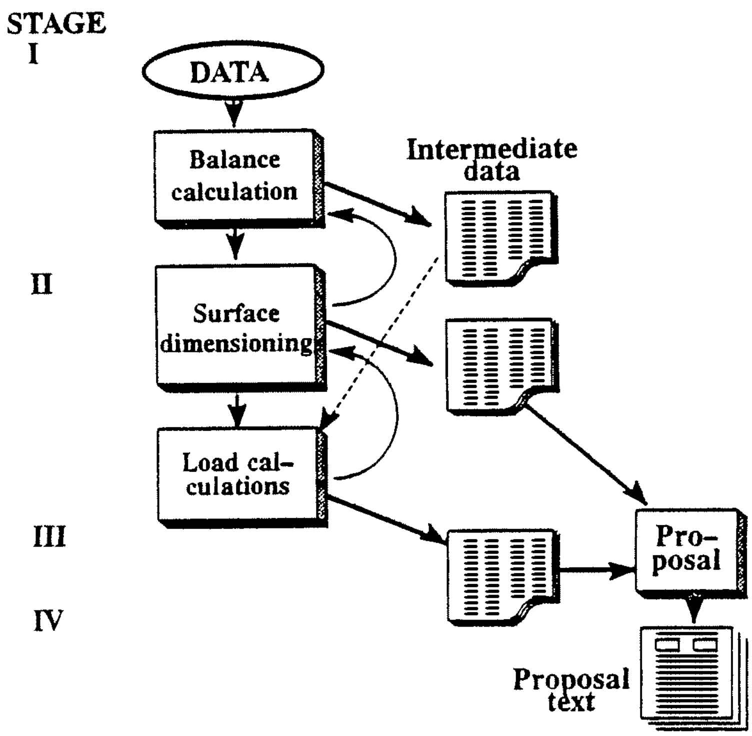

Figure 2. Programs for recovery boiler proposal phase

To improve and speed up the proposal creation different types of programs for the different stages of design work are needed. During this thesis, computer programs were created to support the whole proposal phase of the recovery boiler design. The phases of design process are shown in *Figure 2*. One of these programs was material and energy balance for recovery boilers, **SOMAT**. PC programs for preliminary design of individual heat transfer surfaces included **RADI** for superheaters, **VEBA** for vertical boiler banks and **SEKO** for economizers. The thesis concentrated on a program for the offdesign calculations of boilers, **BOFD**.

The main function of an offdesign program is to perform the calculations needed for the analysis of the boiler partial load behavior. Usually some 20 to 30 different loads are calculated. The guarantees are usually given for the design, minimum and maximum loads. The offdesign program is also used when more than a single design fuel is used. The most typical fuel change for recovery boilers is the change in the black liquor dry solids.

The preliminary objective was to make a program capable of designing and calculating both circulating fluidized bed boilers and recovery boilers. It was considered helpful if the program could also be used to design common pulverized coal, oil fired and grate boilers.

Ahmavaara et. al. (87) have created a detailed specification of the required features and design criteria for the offdesign program. A outline of it is included in *Appendix A*.

3. RECOVERY BOILER MODEL

Recovery boiler is a dynamic process where parameter values fluctuate and change. To model it a very complex set of nonlinear, time dependent, partial differential equations must be used. A water / steam side calculation is presented by Juslin et al. (92). Such models are far too complicated to be used for design and performance analysis.

According to Perregaard et al. (92) when processes are modelled a simplified description of the process is given by sets of equations. The full static mathematical model of a recovery boiler contains; mass balances, energy balances, heat transfer equations, constraints and thermophysical equations.

These equations are solved after all equations have been defined. Equation solving is not easy, because many of the equations describing heat transfer and thermophysical properties are nonlinear. Attention must be paid to the solution algorithm.

3.1 Process units

To create a model for the whole recovery boiler we divide it into several process units, 1 ... j as shown in *Figure 3.* Each process unit represents a smaller subprocess, can be modelled accurately. The process units are connected by mass and energy flows.

The recovery boiler is a special case of general heat exchanger network. In it one main fluid flow, flue gas, gives heat to multiple flows; steam, water and air.

The starting point for dividing the process into process units is the process flowsheet. Each process unit represents the largest entity for which we can define both the heat and the mass transfer equations with required accuracy. The process units model all the important subprocesses in the whole recovery boiler. Each main heat transfer surface; furnace, superheater, economizer, etc. forms a process unit.

Process unit boundaries are typically recovery boiler outer walls and other process units. They are connected with mass and energy flows. This way the calculation of heat transfer and chemical processes can be separated from the calculation of flow values.

In each process unit heat can be generated by combustion. Several heat exchanger surfaces and surrounding outer boundaries can transfer heat between the flows through the process units. Heat can be transferred from one process unit to the next process unit through boundaries by radiation and convection.

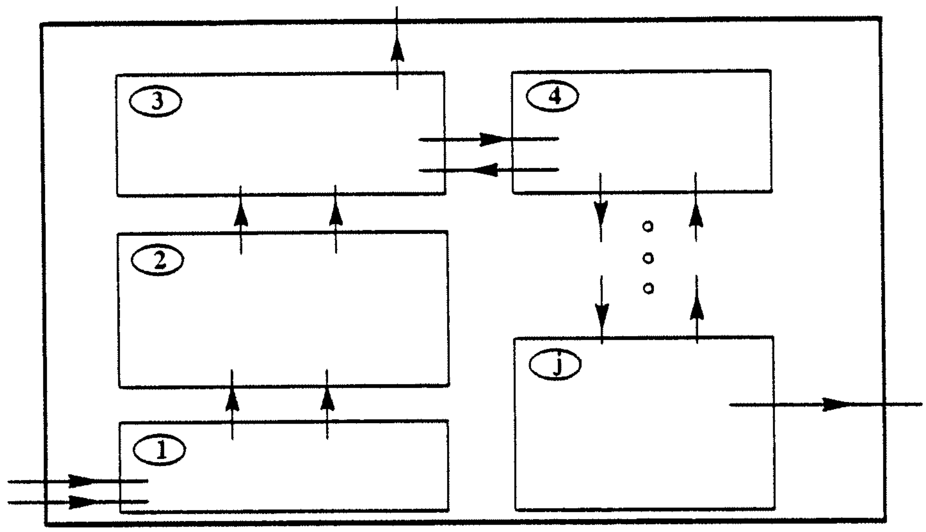

Figure 3. Model with process units 1, ..., j and connecting streams shown

Heat transfer in the process unit is modelled as a heat exchange between at least one heat transfer surface and the heat source. The heat source can be the flue gas or another fluid.

In each process unit the heat transfer processes are described by suitable equations. As these equations are independent from the heat and mass balances, schemes involving different and changing equations can easily be applied. The accuracy of our overall model depends on the accuracy of the equations used to model individual process units.

Actual heat exchanger surfaces can extend into several process units. A typical example is a water-cooled wall. Large physical surfaces have to be divided into smaller calculation surfaces.

3.2 Main variables

The most important task is to identify the primary unknown variables. They are used to define the physical model of the process. The rest of the values are either functions of primary unknowns or design parameters that have a constant, given value.

The main variables are
1. mass flows
2. mass flow inlet and outlet states

3. heat flows; heat losses, heat generation by combustion, heat absorbed by the heat transfer surface elements.

3.2.1 Torn iteration variables

Torn iteration variables are those values of the main variables that best describe the process and result in a quickly solvable physical model.

The torn iteration variables are used to solve the rest of the unknown variables. The solution of all of the main variables is then reduced to the solution of the torn iteration variables.

3.2.2 Design variables

There are typically more unknown variables than equations. The number of unknowns is reduced by choosing some of them as design variables. Design variables are those of the main variables and boundary conditions that either can be assumed to remain constant or the value of which is already specified.

Typical design variables are
1. mass flow inlet states; air inlet temperature, feedwater inlet pressure and temperature
2. desired main steam flow outlet state; main steam outlet pressure and main steam outlet temperature
3. desired combustion rate; liquor feed rate
4. boundary values; ambient temperature and pressure, rate of blowdown, sootblowing steam consumption
5. operating variables; division of attemperating, division of airflows, use of recirculation, combustion of liquor.

For any practical design case these values are specified by the designer based on industry practice. The value of a design variable can change for different loads.

3.2.3 Configuration variables

Configuration variables are parameters or boundary conditions. They have a fixed value for all calculated cases.

Typical configuration variables are dimension values; tube diameters and thicknesses, heights, widths, depths and materials, surface element connection order; location of sweet water condensers and desuperheating and process unit type and order; number and type of superheaters and economizers.

A typical design rule is; The boiler bank inlet temperature must for all load cases be lower than the first melting temperature of deposits. If the calculated boiler bank inlet temperature is too high, the superheater dimensioning should be changed.

Configuration variables do change from one recovery boiler to another based on the calculated values of variables and design rules.

3.3 Example

To clarify the equations in the recovery boiler model the recovery boiler shown in *Figure 4* is modelled. It is a very simple recovery boiler consisting only of furnace made of panel welded tubes, cage where radiative heat transfer takes place, secondary superheater (SH II), primary superheater (SH I), economizer (Eco), walls of the cage and the superheaters, which are cooled by evaporating water, air heater and sweet water condenser.

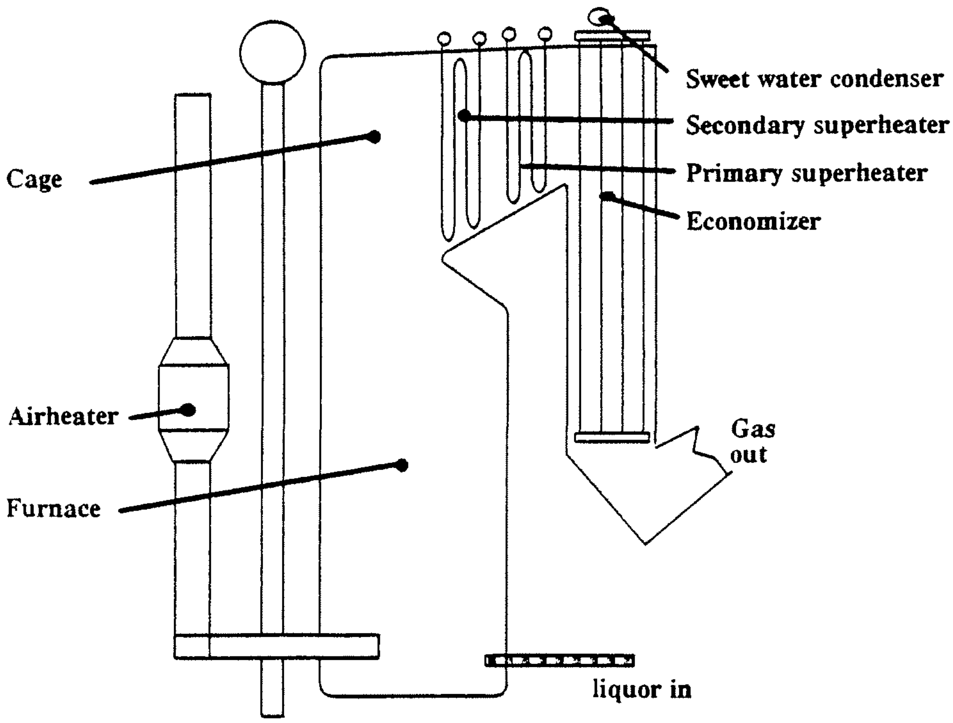

Figure 4. Example recovery boiler

The example recovery boiler is of very simplified Alhstrom type. The air enters through air heater to the furnace. The fuel is fed to the furnace by the liquor guns and the combustion takes place mainly at the lower section of the furnace.

The feedwater enters the economizer, where it is heated near boiling point. It continues through attemperating heat exchanger to the superheater and convection cage walls. In the furnace walls water is evaporated and steam is led to the primary superheater and lastly to the final superheater. To control steam temperatures feedwater attemperating is used between primary and final superheater.

To model the recovery boiler we must describe all the flows in the recovery boiler to define mass exchange processes and divide the whole gas side into smaller process units for defining the heat transfer processes.

3.3.1 *Process units of the example recovery boiler*

The model of the example recovery boiler is shown in *Figure 5. It* consists of seven process units. The effect of sootblowing to the flue gas molar composition is ignored in this simplified model and the flue gas is a constant composition flow.

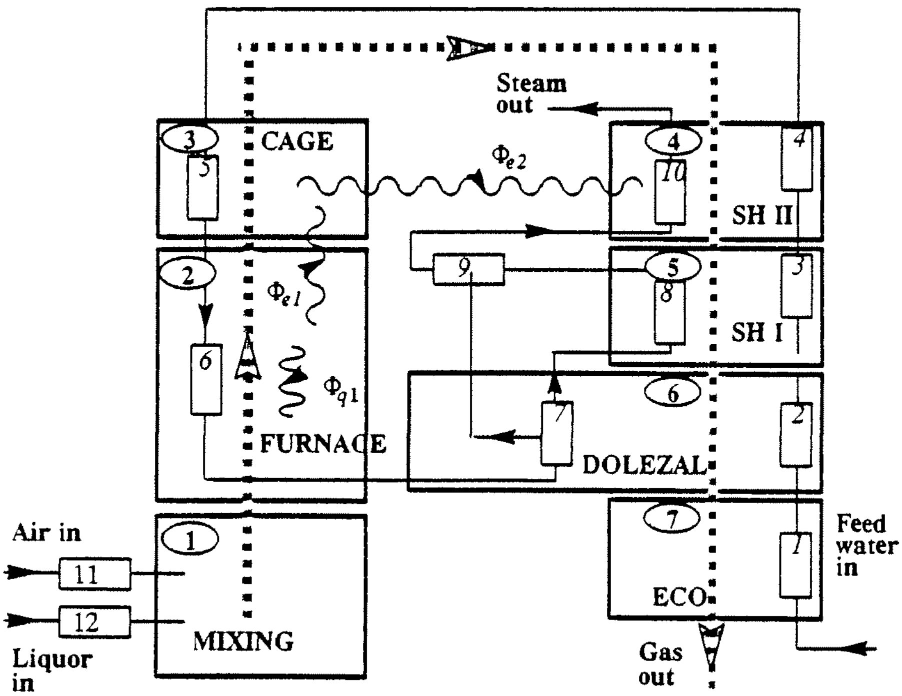

Figure 5. **Flowsheet of example recovery boiler**

The recovery boiler furnace consists of two process units: mass addition process unit and combustion and heat transfer process unit. Liquor and most of the air is typically added at the lowest part of the furnace.

The mass addition process unit is used to calculate the flue gas flow by adding the net liquor *12* and air *11* flows. The flue gas then flows through all of the process units. Net liquor flow replaces the true liquor flow and the smelt flow.

The second process unit is the furnace unit. In the furnace process unit the net combustion heat is added. In furnace there is a single heat exchange element *6* the furnace walls. There is radiation into the cage process unit. The radiation into the superheaters is assumed to be negligible.

The third process unit is the cage unit in which heat is removed from the flue gas by the side walls *5* and through radiation to the secondary superheater process unit.

The fourth process unit is the secondary superheater unit consisting of the secondary superheater surface *10* and the side walls *4*. Secondary superheater receives radiation heat from the cage process unit.

Typical mass addition is the attemperating flow and typical mass extraction is the blowout flow. Modelling mass flow changes outside the process unit boundary does not affect the consistency of the overall process model. If mass flow changes do occur within the process unit boundary, those changes must be accounted for also in the process unit heat transfer model.

The mixing of fluid mass flows takes place outside the heat transfer elements. The mass flow into the heat transfer element equals the mass flow out of the heat transfer element.

Mass division and summation elements are needed for attemperating, blowdown, gas recirculation and air flow separations. In mass exchange modules the mass flow out of element equals the sum of the mass flows into the element.

The fifth process unit is the primary superheater unit consisting of the primary superheater surface *8* and the side walls *3*. There is no radiative heat exchange from previous process units.

The sixth process unit is the sweet water condenser unit where no gas side heat transfer takes place. The attemperating flow is condensed from the saturated steam *7* and the condensing heat is used to heat the economizer *2* outlet water.

The seventh process unit is the economizer unit consisting of the economizer surface *1* in which entering feedwater is heated near the boiling point.

The attemperating in element *9* is modelled outside the process unit boundaries.

3.4 Process unit models

In a typical recovery boiler process unit the flue gas, j, cools. The heat is absorbed by some of the flows 1.. i which cross the process unit boundaries.

The heat exchanging process units for recovery boiler flue gas side can roughly be divided into three types. Single heat transfer surface, in which heat is exchanged between flue gas and a single receiving media. Multiple heat transfer surface, in which heat is exchanged between flue gas and several receiving flows, is the major process unit type. Special process units for example the recovery boiler furnace process unit model the more complicated process units.

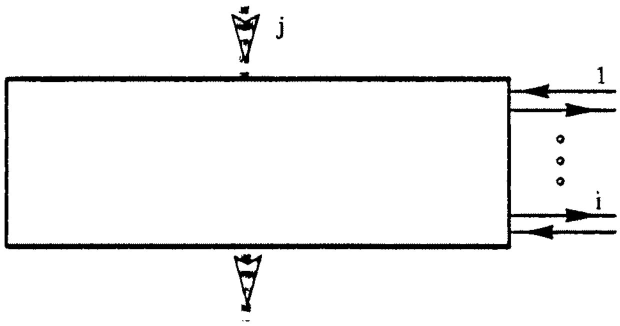

Figure 6. Flows to and from process unit

The heat transfer functions for each process unit model are different. The heat exchange in a process unit is modelled independently from the heat transfer.

The problem
i) from given heat absorbing fluid inlet and outlet conditions
ii) from given heat supplying media inlet conditions
iii) an expression for the heat flow from the heat supplying media and into the heat absorbing media should be found.

The convention, that when the temperature of the fluid raises the heat flow is positive and when the temperature of the fluid decreases the heat flow is negative, was used. In addition it has been assumed that heat flow through boundary into the process unit is positive and out of the process unit negative.

3.4.1 Model of process unit containing a single heat transfer surface

Heat transfer process units of the recovery boiler have been modelled as single heat transfer surface type. See Lundborg et al. (92) and Haynes et al. (88). In fact current heat transfer as taught in universities deals almost exclusively with this kind of situation.

A process unit j containing a single heat transfer surface *l* is shown in *Figure 7*. Fluid j exchanges heat with fluid 1. No mixing of fluids occurs. Heat losses, internal and external heat sources are assumed to equal zero. The heat supplying media can be either fluid 1 or fluid j.

The heat transfer surface *l* is an element through which the mass flow 1 passes.

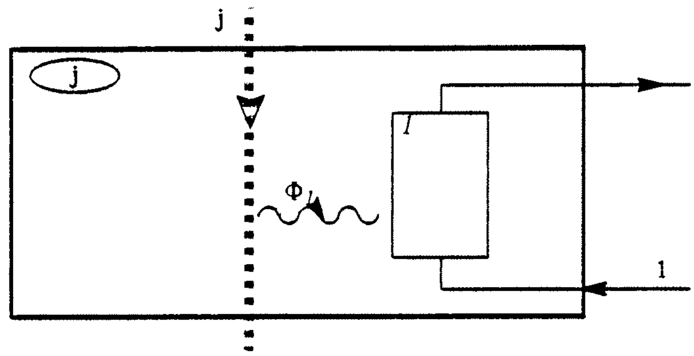

Figure 7. Model of the process unit j containing single heat transfer surface

We apply the convention of identifying the heat transfer element with the mass flow through it with the same element number and the process unit with the flue gas mass flow through it with the same process unit number.

We model the heat flow Φ_l in the process unit j between fluid j and the element *l* by applying the laws of conservation of energy, conservation of mass and specifying the heat transfer equation.

3.4.1.1 Conservation of energy

There is no heat exchange through boundaries

$$\Phi_1 = \Phi_j = \Phi_l \qquad (1)$$

where

Φ_1	is the heat flow recieved by fluid 1 in process unit j
Φ_j	is the heat recieved by fluid j in process unit j
Φ_l	is the heat flow to element *l*

We can define two of the heat flows through changes in the fluid enthalpies

$$\Phi_1 = h_{1i} q_{m1i} + h_{1o} q_{m1o} \qquad (2)$$

and

$$\Phi_j = h_{ji} q_{mji} + h_{jo} q_{mjo} \qquad (3)$$

where

h_{1i}	is the specific enthalpy of fluid 1 at the inlet of process unit j
h_{1o}	is the specific enthalpy of fluid 1 at the outlet of process unit j
h_{ji}	is the specific enthalpy of fluid j at the inlet of process unit j
h_{jo}	is the specific enthalpy of fluid j at the outlet of process unit j
q_{m1i}	is the inlet mass flow of fluid 1 in process unit j
q_{m1o}	is the outlet mass flow of fluid 1 in process unit j
q_{mji}	is the inlet mass flow of fluid j in process unit j
q_{mjo}	is the outlet mass flow of fluid j in process unit j

3.4.1.2 Conservation of mass

In the recovery boiler heat exchangers the fluids do not mix

$$q_{m1i} - q_{m1o} = 0 \qquad (4)$$

and

$$q_{mji} - q_{mjo} = 0 \qquad (5)$$

3.4.1.3 Heat transfer

The third set of equations are the heat transfer equations. Heat flow to a surface element is usually expressed as

$$\Phi_l = U_l A_l \theta_l \qquad (6)$$

where

$$\theta_l = \theta_l (T_{1i}, T_{1o}, T_{ji}, T_{jo}) \qquad (7)$$

and

U_l	is the overall heat transfer coefficient to element l

A_l	is the heat transfer area of element *l*
θ_l	is the effective temperature difference to element *l*
T_{1i}	is the inlet temperature of fluid 1
T_{1o}	is the outlet temperature of fluid 1
T_{ji}	is the inlet temperature of fluid j
T_{j0}	is the outlet temperature of fluid j

The effective temperature difference in equation *7* is usually expressed with the logarithmic temperature difference function.

The overall heat transfer coefficient U for process unit j is a function of the inside 1 and the outside j flow properties and the form variables of the heat exchanger surface *l*.

$$U_l = U_l\ (Re_1, Pr_1, Re_j, Pr_j, d_l\ , A_l\ , ...) \qquad (8)$$

Reynolds and Prandtl numbers are

$$Re_i = Re_i\ (q_{mi}, d_i, A_i, \eta_i\ ) \qquad (9)$$

where

$$Pr_i = Pr_i\ (\lambda_i, c_{pi}, \eta_i\ ) \qquad (10)$$

and

A_i	is the flow area of side *i*
d_i	is the hydraulic diameter of side *i*
q_{mi}	is the mass flow of fluid i at side *i*

so we express *11* as

$$U_l = U_l\ (q_{m1}\ , q_{mj}\ , d_l\ \ , A_l\ \ , ...) \qquad (11)$$

We could express the heat flow Φ_l with equations *6*, ..., *11* if we knew the outlet temperature T_{jo}.

3.4.1.4 Outlet temperature

We can express the outlet temperature T_{jo} with the dimensionless variables ϵ, R, z, using the non dimensional method of Ryti (69). When the inlet temperature of fluid j exceeds the inlet temperature of fluid 1,

$$\epsilon = \epsilon (T_{1i}, T_{1o}, T_{ji}, T_{jo}) \qquad (12)$$

$$= \Delta T_{max} / \theta \qquad (13)$$

$$\theta = (T_{ji} - T_{1i}) \qquad (14)$$

$$\Delta T_{max} = \mathrm{Max}\,(\mathrm{Abs}(T_{1i} - T_{1o}),\ \mathrm{Abs}(T_{ji} - T_{jo})) \qquad (15)$$

$$R = R(q_{m1}, c_{p1},\ q_{mj},\ c_{pj}) \qquad (16)$$

$$= q_{C\,min} / q_{C\,max} \qquad (17)$$

$$q_{C\,min} = \mathrm{Min}\ (q_{m1} c_{p1},\ q_{mj} c_{pj}) \qquad (18)$$

$$q_{C\,max} = \mathrm{Max}\ (q_{m1} c_{p1},\ q_{mj} c_{pj}) \qquad (19)$$

$$z = z (U_1,\ q_{m1} c_{p1},\ q_{mj} c_{pj}, A_1) \qquad (20)$$

$$= G / q_{C\,min} \qquad (21)$$

$$G = U_1\ A_1 \qquad (22)$$

For the countercurrent flow when the fluid j temperature drop is greater than the fluid 1 temperature gain

$$T_{jo} = T_{ji} - \epsilon\,\theta \qquad (23)$$

$$\epsilon = \frac{1 - e^{-z(1-R)}}{1 - e^{-z(1-R)} R} \qquad (24)$$

For parallel flow when the fluid j temperature drop is greater than the fluid 1 temperature gain

$$T_{jo} = T_{ji} - \epsilon\,\theta \qquad (25)$$

$$\epsilon = \frac{1 - e^{-z(1+R)}}{1+R} \qquad (26)$$

Similar types of expressions can be derived for other cases occurring in practice.

Expressions *24* and *26* are not completely analytic. The flue gas properties used in the calculation of heat capacity ratio R and dimensionless conductance z depend weakly on outlet temperature. In practice from two to five iterations are required.

3.4.2 Model of process unit containing multiple heat transfer surfaces

Recovery boiler process units exchange heat through boundaries. They often contain multiple heat transfer surfaces. There can also be heat generation. More complex representation is needed to model recovery boiler process units with high accuracy.

Model of the process unit j in *Figure 8*, consists of m heat transfer surfaces *1, 2, ..., m* through which heat absorbing fluid flows 1, 2, ..., m pass. Through the process unit j flows a heat supplying fluid j.

We can define the heat flows $\Phi_1, \ldots, \Phi_m$ in the process unit j between fluid j and the elements *1, ..., m* by applying the laws of conservation of mass, conservation of energy and specifying the heat transfer equations.

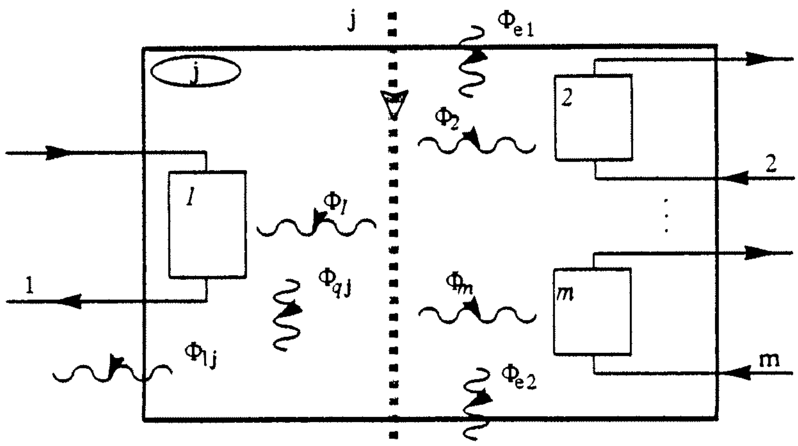

Figure 8. Model of process unit j with m heat transfer surfaces 1, 2, ..., m

Inside the process unit j a heat flow Φ_{qj} is generated. The heat exchange between two other process units through boundaries is represented by external heat flows e1 and e2. Through the process unit j boundary a heat loss lj occurs.

The law of conservation of energy to both volume j and elements *1, ..., m* with heat transfer equations to elements *1, ..., m* define the heat exchange in process unit j between gas flow j and surfaces *1, 2, ..., m*.

3.4.2.1 Conservation of energy

$$\Phi_j + \Phi_1 + \Phi_2 + \dots + \Phi_m + \Phi_{e1} - \Phi_{e2} + \Phi_{lj} + \Phi_{qj} = 0 \quad (27)$$

where

Φ_j	is the heat received by gas flow j in process unit j
Φ_1	is the heat received by surface *1* in process unit j
Φ_2	is the heat received by surface *2* in process unit j
Φ_m	is the heat received by surface *m* in process unit j
Φ_{e1}	is the 1 external heat flow to process unit j
Φ_{e2}	is the 2 external heat flow to process unit j
Φ_{lj}	is the heat losses from process unit j
Φ_{qj}	is the heat generated in process unit j

The heat flow from gas and the heat flows into surfaces can be expressed through changes in the fluid enthalpies

$$\Phi_j = h_{ji} q_{m\,ji} - h_{jo} q_{m\,jo} \quad (28)$$

and

$$\Phi_1 = h_{1i} q_{m\,1i} - h_{1o} q_{m\,1o} \quad (29)$$

$$\Phi_2 = h_{2i} q_{m\,2i} - h_{2o} q_{m\,2o} \quad (30)$$

$$\dots$$

$$\Phi_m = h_{mi} q_{m\,mi} - h_{mo} q_{m\,mo} \quad (31)$$

3.4.2.2 Heat transfer

The heat flows to the surfaces *1*, ..., *m* are expressed by

$$\Phi_1 = U_1\, A_1\, \theta_1 \quad (32)$$

$$\Phi_2 = U_1\, A_2\, \theta_2 \quad (33)$$

$$\dots$$

$$\Phi_m = U_m\, A_m\, \theta_m \quad (34)$$

where

$$\theta_1 = \theta_1\,(T_{1i}, T_{1o}, T_{ji}, T_{jo}) \qquad (35)$$

$$\theta_2 = \theta_2\,(T_{2i}, T_{2o}, T_{ji}, T_{jo}) \qquad (36)$$

$$\ldots$$

$$\theta_m = \theta_m\,(T_{mi}, T_{mo}, T_{ji}, T_{jo}) \qquad (37)$$

Heat transfer coefficients for each element *i* depend on the inside and outside flow properties and the design variables of the heat transfer surface element *i*.

$$U_i = U_i\,(Re_i, Pr_i, Re_j, Pr_j, d_i, A_i, \ldots) \qquad (38)$$

and as before

$$U_i = U_i\,(q_{mi}, q_{mj}, d_i, A_i, \ldots) \qquad (39)$$

For more detailed analysis of actual heat transfer calculation methods, applicable to the design of recover boiler surfaces see Rahtu (90).

The heat flows $\Phi_1, \ldots, \Phi_m$ can be expressed with equations *28, ..., 39* if the outlet temperature T_{go} is known.

3.4.2.3 Outlet temperature

Dimensionless variables ϵ, R^*, z are used to define an expression for the outlet temperature T_{go} when the inlet temperature of the fluid j exceeds the inlet temperature of fluids 1, 2, ..., m. The heat flow to fluid 1 is assumed to be greater or equal to heat flows to fluids 2, ..., m respectively.

$$R^* = R^*(q_{m1}, c_{p1}, q_{mj}, c_{pj}, b) \qquad (40)$$

$$= q^*_{Cmin} / q^*_{Cmax} \qquad (41)$$

$$q^*_{Cmin} = \text{Min}\,(q_{m1}c_{p1},\ b\,q_{mj}c_{pj}) \qquad (42)$$

$$q^*_{Cmax} = \text{Max}\,(q_{m1}c_{p1},\ b\,q_{mj}c_{pj}) \qquad (43)$$

The correction factor b defines R^* so that it equals the ratio of temperature changes.

$$R^* = \Delta T_{min} / \Delta T_{max} \qquad (44)$$

$$\Delta T_{max} = \text{Max}\,[\,\text{Abs}(T_{1i} - T_{1o}),\ \text{Abs}(T_{ji} - T_{jo})\,] \qquad (45)$$

$$\Delta T_{min} = \mathrm{Min}\ [\ \mathrm{Abs}(T_{1i} - T_{1o}\ ),\ \mathrm{Abs}(T_{ji} - T_{jo})\ ] \qquad (46)$$

We further define ϵ, G and θ as previously with temperatures of fluids j and 1.

$$\epsilon = \epsilon\ (\ T_{1i}\ , T_{1o}\ , T_{ji}\ , T_{jo}\ ) \qquad (47)$$

$$= \Delta T_{max} / \theta \qquad (48)$$

$$\theta = (\ T_{qi} - T_{1i}\ ) \qquad (49)$$

$$z = z\ (\ U_l\ , q_{m1} c_{p1}\ , q_{mj}\ c_{pj}\ , A_l\ ) \qquad (50)$$

$$= G / q_{C_{min}}^{\cdot} \qquad (51)$$

$$G = U_l\ A_l \qquad (52)$$

Outlet temperature of fluid 1 can then be expressed using equation *24* or *26*.

The correction coefficient b can be defined using the overall heat balance *27*. The effect of the other heat capacity flows is subtracted from the gas side heat capacity. For the multiple heat transfer surface case the correction factor b is

$$b = \frac{q_{mj} c_{pj} - \dfrac{\Phi_{qj} + \Phi_2 + \ldots + \Phi_m - \Phi_{e2} + \Phi_{e1} + \Phi_{1j}}{T_{ji} - T_{jo}}}{q_{mj} c_{pj}} \qquad (53)$$

Expressions *28*, ..., *53* are suitable for iterative solving of T_{jo}. The iteration converges quickly because usually the heat flows to surfaces *2*, ..., *m* are an order of magnitude smaller than the heat flow to main surface. No additional iterations are needed as the flow properties require a few iterations.

3.4.3 Model of the recovery boiler furnace

The recovery boiler furnace outlet temperature can not be calculated with the models presented previously. This is because there is heat generation in recovery boiler furnace, which varies with height. Furnace temperature profiles have been published by Jones (89), Purola (82), Tolvanen (80) and Jutila et al. (78).

Model of the recovery boiler furnace as shown in *Figure 9*, consists of lower furnace model, process unit j and upper furnace model, process unit j+1. The furnace processes consist then of a separate heat transfer and mass addition process unit.

In the upper furnace model there are two surface elements *1* and *2* through which two heat absorbing fluid flows 1 and 2 pass. Heat is generated by combustion and exchanged through radiation with external surfaces *e1* through *em*. Through the process unit j+1 boundary a heat loss occurs. Heat generated by combustion is first absorbed by the fluid flow j+1. Elements *1* and *2* then absorb some of the heat.

Inside the lower furnace process unit j a heat flow is generated and heat is exchanged between upper and lower furnace process units through common boundary. The flue gas flow j+1 is also generated in the lower furnace process unit by primary air flow pa, secondary air flow sa, tertiary air flow ta and net liquor flow lq.

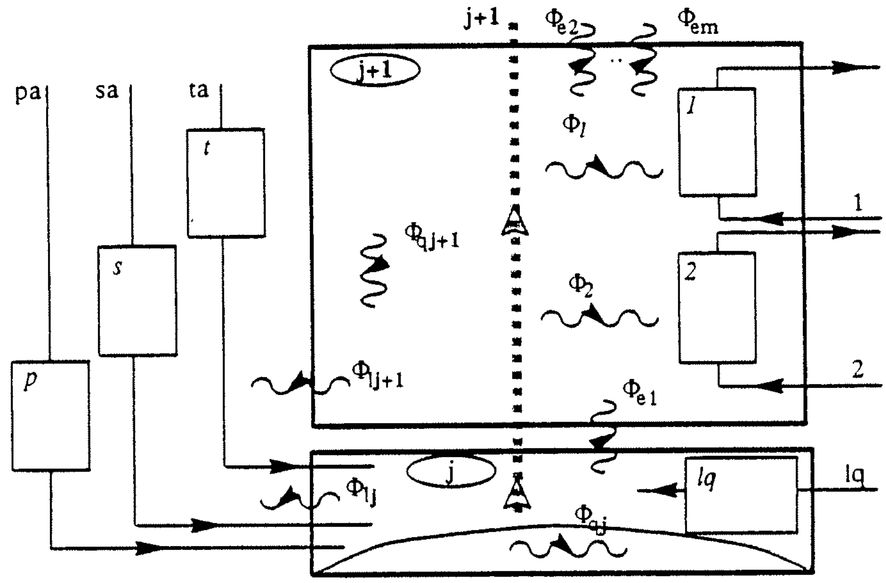

Figure 9. Model of recovery boiler furnace

The heat exchange in process unit j+1 between gas flow j+1 and surfaces *1* and *2* is defined by applying laws of

3.4.3.1 Conservation of energy

For process unit j+1

$$\Phi_{j+1} + \Phi_{1} + \Phi_{2} - \Phi_{e1} + .. + \Phi_{em} + \Phi_{l\,j+1} + \Phi_{q\,j+1} = 0 \qquad (54)$$

heat given by flue gas flow j+1 + heat to surface *1* + heat to surface *2* - external heat 1 + .. + external heat m + heat losses from j+1 + heat generated in j+1 = 0

For the mass addition process unit j

$$\Phi_{j} + \Phi_{e1} + \Phi_{pa} + \Phi_{sa} + \Phi_{ta} + \Phi_{lq} + \Phi_{l\,j} + \Phi_{qj} = 0 \qquad (55)$$

heat given by flue gas flow j + external heat flow 1 + heat given by primary air + heat given by secondary air + heat given by tertiary air + heat given by net liquor flow + heat losses from j + heat generated in j = 0

Changes in the fluid enthalpies define heat given by gas and all heat flows to surfaces.

$$\Phi_{j+1} = h_{j+1i}\,q_{m\,j+1i} - h_{j+1o}\,q_{m\,j+1o} \qquad (56)$$

$$\Phi_{j} = h_{ji}\,q_{m\,ji} - h_{jo}\,q_{m\,jo} \qquad (57)$$

and

$$\Phi_{1} = h_{1i}\,q_{m1i} - h_{1o}\,q_{m1o} \qquad (58)$$

$$\Phi_{2} = h_{2i}\,q_{m2i} - h_{2o}\,q_{m2o} \qquad (59)$$

3.4.3.2 Heat transfer

The heat flow to the surfaces is

$$\Phi_{1} = U_{1}\,A_{1}\,\theta_{1} \qquad (60)$$

$$\Phi_{2} = U_{2}\,A_{2}\,\theta_{2} \qquad (61)$$

and

$$\theta_1 = \theta_1 (T_{1i}, T_{1o}, T_{ji}, T_{jo}) \qquad (62)$$

$$\theta_2 = \theta_2 (T_{2i}, T_{2o}, T_{ji}, T_{jo}) \qquad (63)$$

Heat transfer coefficients for each process unit j are defined as functions of the inside and outside flow properties and the design variables of heat exchanger surface element *i*.

$$U_i = U_i (Re_i, Pr_i, Re_j, Pr_j, d_i, A_i, \ldots) \qquad (64)$$

And as before

$$U_i = U_i (q_{mi}, q_{mj}, d_i, A_i, \ldots) \qquad (65)$$

3.4.3.3 Outlet temperature

Heat transfer in recovery boiler furnace is mainly radiative. Newton's method for outlet temperature estimation sometimes diverges and usually converges poorly, when stirred reactor furnace model was used.

First the combination method of Demidovich and Maron (76) was tried to decrease divergence. In the combination method the method of proportional parts is used to ensure convergence. Even with the combination method the convergence was slow.

Then an empirical expression for outlet temperature was used which achieved convergence to 0.01 degrees within ten - fifteen iterations

$$T^{n+1}_{j+1o} = [(T^{n}_{j+1o})^{a} (T^{n-1}_{j+1o})^{b}]^{1/(a+b)} \qquad (66)$$

For calculated recovery boiler cases the best convergence was achieved when $a = 1.5$ and $b = 2.5$.

We can use expression *66* for iterative solving of the outlet temperature. From equations *56* and *54* the heat flows Φ_1 and Φ_2 can be solved.

In addition to a stirred reactor furnace model, a one-dimensional furnace model can be used. In the one-dimensional model a multiple-point furnace temperature profile is calculated instead of constant temperature profile.

The heat release profile can be found out through fitting to achieve a known temperature profile. The fine tuning can be made by comparing industrial measurements to achieve a correct furnace outlet temperature. The furnace outlet temperature predictions with this model are more accurate than with other models.

With a new type of calculation point ordering the heat transfer could be made non-iterative. Some 20 calculation points are needed to represent the furnace temperature profile. The profile model is a little slower than the other furnace models.

Prediction of recovery boiler furnace outlet temperatures is dependent on the liquor droplet size distribution, liquor temperature, liquor gun pressure, number and location of liquor guns, liquor gun angle, air distribution, number and location of air ports and air temperatures.

All these operation factors are dependent on boiler design and operational practices. There is no consensus at the moment of optimum values nor optimum practices. The temperature distributions vary from one recovery boiler to another recovery boiler. The outlet temperatures can be predicted with some ± 20 ...30 °C accuracy without accurate knowledge of operation factors.

3.5 Governing equations

The governing equations for the model of the example recovery boiler consist of large number of expressions.

3.5.1 Conservation of mass

Sum of mass flows to a element equals the mass flow from the element. Connecting the steam / water side flows we get eleven equations.

$$
\begin{aligned}
&q_{m1i} - q_{mfw} = 0\\
&q_{m2i} - q_{m1o} = 0\\
&q_{m3i} - q_{m2o} = 0\\
&q_{m4i} - q_{m3o} = 0\\
&q_{m5i} - q_{m4o} = 0\\
&q_{m6i} - q_{m5o} = 0 \qquad (67)\\
&q_{m7i} + q_{mA} - q_{m6o} = 0\\
&q_{m8i} - q_{m7o} = 0
\end{aligned}
$$

$$q_{m9i} - q_{m8o} - q_{mA} = 0$$
$$q_{m10i} - q_{m9o} = 0$$
$$q_{ms} - q_{m10o} = 0$$

Connecting the flue gas side flows we get eight equations.

$$q_{m1i} - q_{m11} - q_{m12} = 0$$
$$q_{m2i} - q_{m1o} = 0$$
$$q_{m3i} - q_{m2o} = 0$$
$$q_{m4i} - q_{m3o} = 0 \qquad (68)$$
$$q_{m5i} - q_{m4o} = 0$$
$$q_{m6i} - q_{m5o} = 0$$
$$q_{m7i} - q_{m6o} = 0$$
$$q_{mfg} - q_{m7o} = 0$$

3.5.2 *Conservation of energy*

Applying the conservation of energy for each mass exchanging process unit we get three equations.

$$\Phi_1 = h_{11o}q_{m11o} + h_{12o}q_{m12o}$$
$$\Phi_{11} = 0 \qquad (69)$$
$$\Phi_{12} = 0$$

Applying the conservation of energy for each heat transfer process unit we get five equations.

$$\Phi_2 - \Phi_6 - \Phi_{e1} + \Phi_{q1} = 0$$
$$\Phi_3 - \Phi_5 + \Phi_{e1} - \Phi_{e2} = 0$$
$$\Phi_4 - \Phi_{10} - \Phi_4 + \Phi_{e2} = 0 \qquad (70)$$
$$\Phi_5 - \Phi_8 - \Phi_3 = 0$$
$$\Phi_7 - \Phi_1 = 0$$

Applying the conservation of energy for desuperheating process unit we get three equations.

$$\Phi_6 = 0$$
$$\Phi_7 - \Phi_2 = 0 \qquad (71)$$
$$\Phi_9 = 0$$

Process unit flue gas side heat flow can be expressed with enthalpy difference.

$$\Phi_1 = h_{1i} q_{m1i} - h_{1o} q_{m1o}$$
$$\Phi_2 = h_{2i} q_{m2i} - h_{2o} q_{m2o}$$
$$\Phi_3 = h_{3i} q_{m3i} - h_{3o} q_{m3o}$$
$$\Phi_4 = h_{4i} q_{m4i} - h_{4o} q_{m4o} \qquad (72)$$
$$\Phi_5 = h_{5i} q_{m5i} - h_{5o} q_{m5o}$$
$$\Phi_6 = h_{6i} q_{m6i} - h_{6o} q_{m6o}$$
$$\Phi_7 = h_{7i} q_{m7i} - h_{7o} q_{m7o}$$

Surface element heat flows can be expressed with enthalpy difference.

$$\Phi_1 = h_{1o} q_{m1o} - h_{fw} q_{mfw}$$
$$\Phi_2 = h_{2o} q_{m2o} - h_{2i} q_{m2i}$$
$$\Phi_3 = h_{3o} q_{m3o} - h_{3i} q_{m3i}$$
$$\Phi_4 = h_{4o} q_{m4o} - h_{4i} q_{m4i}$$
$$\Phi_5 = h_{5o} q_{m5o} - h_{5i} q_{m5i}$$
$$\Phi_6 = h_{6o} q_{m6o} - h_{6i} q_{m6i} \qquad (73)$$
$$\Phi_7 = h_{7o} q_{m7o} - h_{7i} q_{m7i} + h_A q_{mA}$$
$$\Phi_8 = h_{8o} q_{m8o} - h_{8i} q_{m8i}$$
$$\Phi_9 = h_{9o} q_{m9o} - h_{9i} q_{m9i} - h_A q_{mA}$$
$$\Phi_{10} = h_{10o} q_{m10o} - h_{10i} q_{m10i}$$
$$\Phi_{11} = h_{11o} q_{m11o} - h_{11i} q_{m11i}$$
$$\Phi_{12} = h_s q_{ms} - h_{12i} q_{m12i}$$

3.5.3 Heat transfer

For each heat transfer surface element the heat flow can be expressed like equations *6* and *7*. There are seven heat transfer surface elements so we get seven equations.

$$
\begin{aligned}
\Phi_1 &= U_1\ A_1\ \theta_1\ (T_{1i}\ ,T_{1o}\ ,T_{7i}\ ,T_{7o}) \\
\Phi_3 &= U_3\ A_3\ \theta_3\ (T_{3i}\ ,T_{3o}\ ,T_{5i}\ ,T_{5o}) \\
\Phi_4 &= U_4\ A_4\ \theta_4\ (T_{4i}\ ,T_{4o}\ ,T_{4i}\ ,T_{4o}) \\
\Phi_5 &= U_5\ A_5\ \theta_5\ (T_{5i}\ ,T_{5o}\ ,T_{3i}\ ,T_{3o}) \\
\Phi_6 &= U_6\ A_6\ \theta_6\ (T_{6i}\ ,T_{6i}\ ,T_{2i}\ ,T_{2o}) \\
\Phi_8 &= U_8\ A_8\ \theta_8\ (T_{8i}\ ,T_{8o}\ ,T_{5i}\ ,T_{5o}) \\
\Phi_{10} &= U_{10} A_{10} \theta_{10} (T_{10i}\ ,T_{10o}\ ,T_{4i}\ ,T_{4o})
\end{aligned}
\qquad (74)
$$

4. SOLVING THE RECOVERY BOILER MODEL

During recovery boiler model solution mass balances, energy balances and heat transfer equations are solved to the specified accuracy.

Solving the model is complicated because the heat transfer functions are nonlinear. Also the thermodynamic properties of water and steam are nonlinear. According to Westerlund (84) they require complicated functions to calculate.

The solving of the recovery boiler model is reduced to finding the solution to a set of nonlinear equations by suitable algorithm.

$$F(x,y) = 0 \tag{75}$$

subject to

$$g(x,y) = 0 \tag{76}$$

$$h(x,y) \geq 0 \tag{77}$$

where

F	is the vector of error functions
x	is the torn iteration variables
y	is the design variables
g	is the vector of equality constraints
h	is the vector of inequality constraints.

The model should be defined so that a quick solution is possible. Gundersen (90) presents decomposition algorithms and describes their use. Instead of using a algorithm for decomposition it was decided to try to represent the governing equations in a new way. In chapters 4.1 to 4.5 recovery boiler model is reduced to a subsets of equations. In chapter 5 it is shown that these equations are readily solvable.

4.1 Variables

The first task is to choose suitable torn iteration variables. The choice is important as it will affect the whole solution procedure. There seems to be a practice to choose mass flows as torn iteration variables, when calculating complex heat transfer situations in steam generating equipment. At least Nishio (77), Sandner (83), Adibhatla (81) and Asikainen (83) have done so. Heat flows and temperatures are then solved after the mass flows are determined.

When considering the recovery boiler partial load situation, other more appropriate choices can be made. In the offdesign program developed the heat flows to the heat transfer surface elements have been chosen as torn iteration variables. In the following the solution procedure for the unknown variables through heat flows will be more thoroughly described.

First we will describe what is meant by vectorized mass flow representation. Then the example recovery boiler generator model is described as consisting several mass flow vectors. Lastly through the mass and energy balances of the example the solution of such elements is described.

4.2 Vector representation of mass flows

The complex processes that a fluid undergoes in a recovery boiler can be represented with smaller, simpler subprocesses. These subprocesses will be called elements. The change in the fluid state occurs within the element. Elements are connected by mass flows.

The flow of mass through a string of mass exchanging and heat transferring surface elements, *1 ... i*, is a single mass flow vector as shown in *Figure 10*.

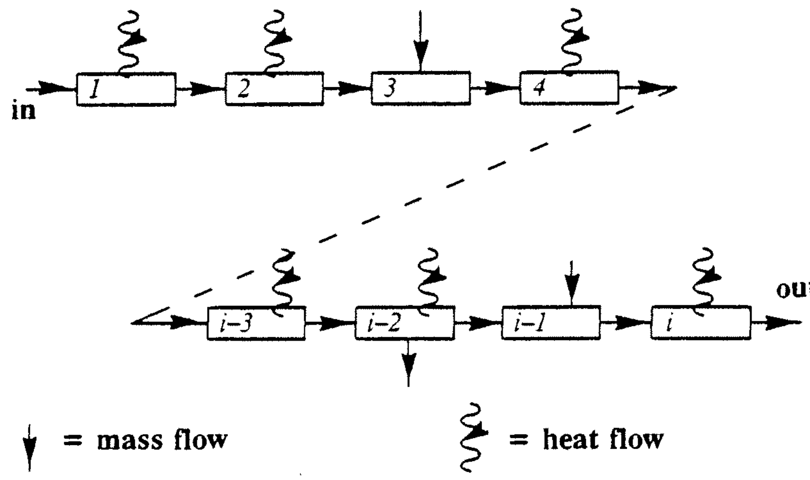

Figure 10. Typical vector representation of a mass flow through elements *1 ... i*

In the *Figure 10* elements *1, 2, ... ,i* can be separated into;

i) elements where heat is exchanged with the surroundings; element is a part of a heat transfer surface

ii) elements where a mass flow is added to or removed from the incoming flow; element is used to describe attemperating, blowdown, sootblowing, etc.

Surfaces *3* and *i–1* receive a mass flow and surface *i–2* extracts a mass flow. Surfaces *1*, *2*, *4*, *i–3* and *i* are heat transfer elements. Each of those elements is a part of some heat transfer surface.

4.2.1 *Mass flow vectors of the example recovery boiler*

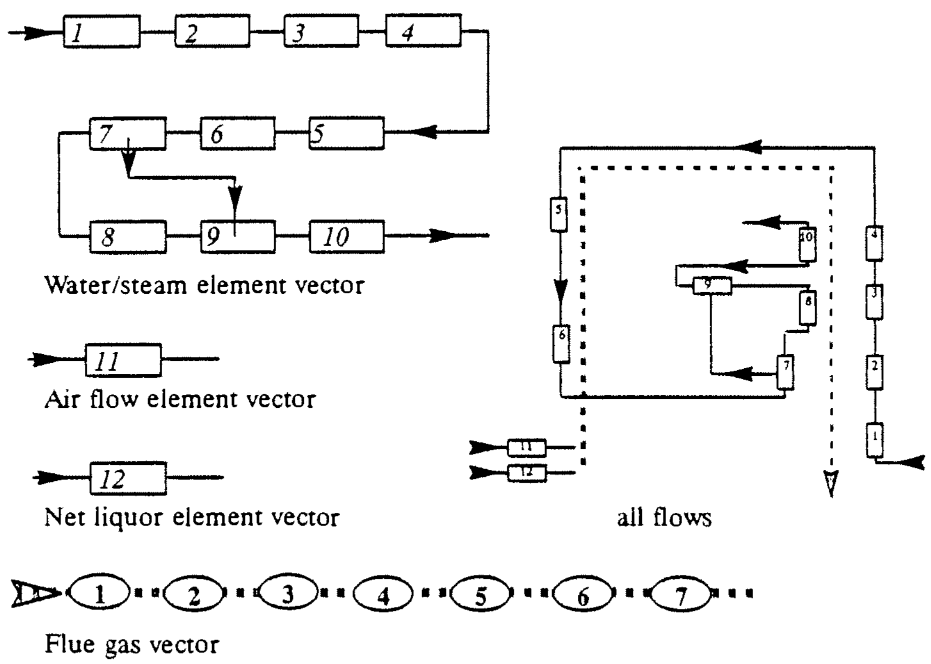

Figure 11. Mass flow vector data for example recovery boiler

Figure 11 shows the four different mass flow vectors in the example recovery boiler model.

The first mass flow vector is the water / steam element vector in which the whole water / steam circulation is described by a single vector. The attemperating flow *7 –> 9* is determined in the sweet water condenser process unit. Because we are modelling heat transfer, we do not try to model the real water circulation. For dimensioning purposes a simplified model in which temperatures and pressures in the waterside elements are correct is enough.

Flue gas vector describes the flow of flue gas through the recovery boiler process units. The flue gas flow in the furnace of the recovery boiler is a sum of net liquor *12* and combustion air *11*.

4.2.2 *Sample values for the mass flow vectors*

We can from the calculation of heat and mass balances estimate relatively close input values for the volume and element side vectors.

Combustion calculations define the net heat to furnace. Radiation and convection losses are given by overall balances. Preliminary furnace outlet temperature for the design load, preliminary economizer outlet temperature for the design load and flue gas mass flow are given by overall balances.

Table 1. Flue gas vector values

Process unit number	1	2	3	4	5	6	7
Process unit name	MIX	FURN	CAGE	SH II	SH I	SWC	ECO
Net heat to subvolume, kW	0	60000.	-20.	-20.	-60.	-30.	-30.
Inlet temperature, °C	xx	940.	xx	xx	xx	xx	190.
Outlet temperature, °C	xx	xx	xx	xx	xx	xx	xx
Mass flow, kg/s	22.3	22.3	22.3	22.3	22.3	22.3	22.3

For the water/steam flow vector we know preliminary heat flows to individual heat transfer element surfaces, inlet state; temperature, pressure and steam content, required outlet state; temperature, pressure and steam content, evaporative state; temperatures and pressures for evaporative surfaces, preliminary inlet mass flow from the overall balances and individual pressures from preliminary design pressure losses.

Table 2. Water / steam side vector values

Element number	1	2	3	4	5	6	7	8	9	10
Net heat to element, kW	10000	500.	100.	100.	2000.	36000	0	5000.	0	6000.
Outlet steam content, %	xx	xx	xx	xx	xx	100.	100.	100.	100.	100.
Inlet steam content, %	0	xx	xx	xx	xx	xx	100.	100.	100.	100.
Outlet temperature, °C	xx	xx	269.5	269.5	269.5	269.5	269.5	xx	xx	410.
Inlet temperature, °C	125.	xx	xx	269.5	269.5	269.5	269.5	269.5	xx	xx
Outlet pressure, bar	55.6	54.6	54.6	54.6	54.6	54.6	54.6	52.1	51.8	49.2
Inlet pressure, bar	55.7	55.6	54.6	54.6	54.6	54.6	54.6	54.6	52.1	51.8
Mass flow, kg/s	21.2	21.2	21.2	21.2	21.2	21.2	xx	xx	xx	21.2

For the air flow vector we know heat flow to heat transfer element surface, inlet state; temperature and pressure, outlet pressure and mass flow.

Table 3. Air flow vector values

Element number	11
Net heat to element, kW	3000.
Outlet temperature, °C	xx
Inlet temperature, °C	0
Outlet pressure, bar	1.00
Inlet pressure, bar	1.00
Mass flow, kg/s	18.00

The net liquor is defined as any other incoming mass flow. For the net liquor flow vector we know the heat flow to heat transfer element surface is negligible. The inlet state; temperature and pressure, outlet pressure and mass flow are design values.

Table 4. Net liquor flow vector values

Element number	12
Net heat to element, kW	0.
Outlet temperature, °C	xx
Inlet temperature, °C	110.
Outlet pressure, bar	1.20
Inlet pressure, bar	1.20
Mass flow, kg/s	4.30

The unknown values marked xx in the tables 1, ..., 4 must be solved. We define the needed balances for the solution in chapters 4.3 through 4.5 and describe the chosen iteration method in chapter 5.3 pp. 54 - 57.

4.3 Solving the water / steam vector

The water / steam mass flow matrix for the example recovery boiler consists of nineteen equations.

For all water / steam flow elements the inlet flow equals the outlet flow. There is eleven equations and thirteen mass flows. We choose the attemperating flow and the feedwater inlet flow to be the torn iteration values. The mass flow matrix for steam and water flows is then

$$\mathbf{A}\ \mathbf{q} = \mathbf{B} \qquad (78)$$

where

$$A = \begin{bmatrix} 1 & & & & & & & & & \\ -1 & 1 & & & & & & & & \\ & -1 & 1 & & & & & & & \\ & & -1 & 1 & & & & & & \\ & & & -1 & 1 & & & & & \\ & & & & -1 & 1 & & & & \\ & & & & & -1 & 1 & & & \\ & & & & & & -1 & 1 & & \\ & & & & & & & -1 & 1 & \\ & & & & & & & & -1 & 1 \\ & & & & & & & & & 1 \end{bmatrix} \qquad (79)$$

and

$$q = \begin{bmatrix} q_{m1} \\ q_{m2} \\ q_{m3} \\ q_{m4} \\ q_{m5} \\ q_{m6} \\ q_{m7} \\ q_{m8} \\ q_{m9} \\ q_{m10} \\ q_{ms} \end{bmatrix} \qquad (80)$$

$$B = \begin{bmatrix} q_{mfw} \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ -q_{mA} \\ 0 \\ q_{mA} \\ 0 \\ 0 \end{bmatrix} \qquad (81)$$

Specifying each single mass flow that connects two elements as unknown is the usual practice in current programs. Typical examples are Sandner (83), Gundersen and Herzberg (84). Each connecting stream adds at least one additional unknown. This increases the number of unknowns substantially.

In the recovery boiler case this is unnecessary. The mass flows have been ordered as a vector. All the other mass flows can be readily solved if the feedwater flow q_{mfw} and the attemperating mass flow q_{mA} are known. It should be noted that general equation solution routines can't detect this type of behaviour but have to iterate.

For all elements the outlet from the previous element equals the inlet to the element. The heat balances for the elements give twelve equations for the total enthalpies of outlets

$$E\,H = F \qquad (82)$$

where

$$E = \begin{bmatrix} 1 &&&&&&&&&&& \\ -1 & 1 &&&&&&&&&& \\ & -1 & 1 &&&&&&&&& \\ && -1 & 1 &&&&&&&& \\ &&& -1 & 1 &&&&&&& \\ &&&& -1 & 1 &&&&&& \\ &&&&& -1 & 1 &&&&& \\ &&&&&& -1 & 1 &&&& \\ &&&&&&& -1 & 1 &&& \\ &&&&&&&& -1 & 1 && \\ &&&&&&&&& -1 & 1 & \\ &&&&&&&&&& 1 & 1 \end{bmatrix} \qquad (83)$$

and

$$H = \begin{bmatrix} H_1 \\ H_2 \\ H_3 \\ H_4 \\ H_5 \\ H_6 \\ H_7 \\ H8 \\ H_9 \\ H_{10} \\ H_{11} \\ H_s \end{bmatrix} \quad \text{and} \quad H_i = h_i q_{mi} \qquad (84)$$

$$F = \begin{bmatrix} h_{fw}q_{mfw}+\Phi_{1} \\ \Phi_{2} \\ \Phi_{3} \\ \Phi_{4} \\ \Phi_{5} \\ \Phi_{6} \\ -h_{A}q_{mA}+\Phi_{7} \\ \Phi_{8} \\ h_{A}q_{mA}+\Phi_{9} \\ \Phi_{10} \\ \Phi_{11} \\ \Phi_{12} \end{bmatrix} \qquad (85)$$

If the heat flows, feedwater inlet flow and enthalpy with the attemperating flow and enthalpy are given all the outlet states can be solved.

4.3.1 *General water / steam element vector*

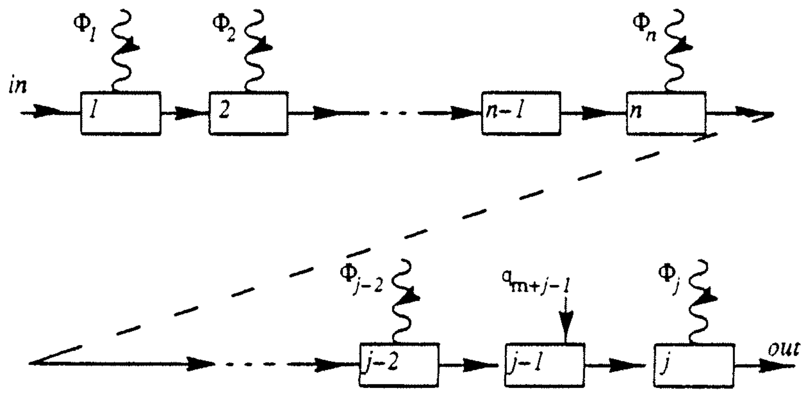

Figure 12. Water / steam element vector

A general water / steam vector is shown in *Figure 12.* It consists of heat transfer elements *1, 2, ... , n, ... j-2* and *j*. It also has mass addition elements , ... , *j-1*. In the mass addition element a mass flow is either extracted or added to the main water / steam flow. The only restriction for the order of mass addition and heat transfer elements is that the first element *1* must be a heat transfer element.

A typical temperature profile of the water / steam vector is shown in *Figure 13.*

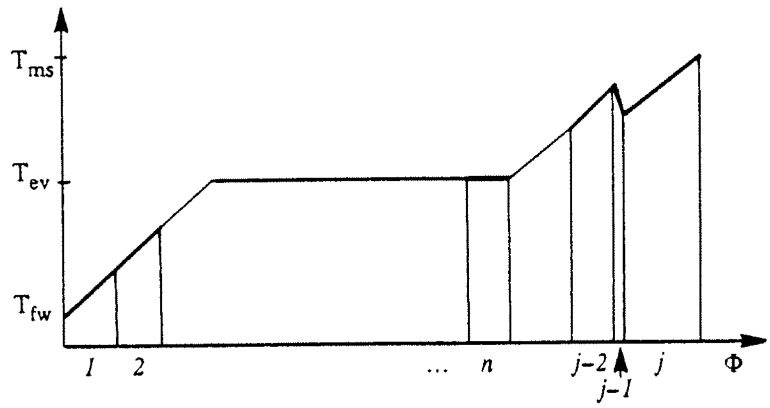

Figure 13. Water / steam vector temperature profile

The water enters at feedwater inlet temperature to element *1*. It is first heated and then evaporated until at the outlet of the element *n* the evaporation is complete. The steam is then superheated and attemperated until it finally reaches the outlet element *j*, from which it exits at main steam temperature.

Chosen boundary conditions for solving the equations are

i) There exists an element *n* after which the mass flow is steam. The number of elements after *n* can be zero if the recovery boiler produces saturated steam. The number of elements before *n* can be zero for a flow through reheater.
ii) The maximum allowable steam outlet temperature is given. Main steam outlet temperature is regulated by attemperating flows.
iii) The mass flow ratios of attemperating flows are given. For example the first attemperating flow is 60 % of the total attemperating flow.
iv) The inlet state to the first element of the vector is given.
v) The states of all mass addition flows are given.
vi) Pressure losses depend linearly on dynamic pressure and the vector outlet pressure is known, see chapter 4.6 pp. 45 - 48.

4.3.2 *Solution procedure*

The target is to determine the main steam mass flow and the attemperating mass flows. The desired maximum main steam temperature and the location of complete evaporation are the given boundary conditions.

We can handle feed forward flows easily within the vector representation but the feed backward flows require several vectors. For boiler applications usually all water / steam flows are of feed forward type.

The above problem can be divided into two parts

Step 1: Solution for evaporative elements

i) Determine feedwater mass flow so the evaporation is complete at the outlet of the last evaporating element *n* from known outlet temperature and pressure with fixed heat flows to elements.

ii) Calculate mass flows through all evaporating elements.

iii) Calculate inlet and outlet enthalpies for all evaporative elements.

Step 2: Solution for superheating elements

i) From known inlet mass flow to first superheating element *n+1* calculate attemperating mass flows to achieve required outlet temperature with fixed heat flows to elements.

ii) Calculate mass flows through all superheating elements.

iii) Calculate inlet and outlet enthalpies for all superheating elements

Feedwater mass flow

We solve the feedwater flow q_{mfw} from the overall heat balance of elements *1 ... n*

$$q_{mfw} = \frac{\sum_i^n \Phi_i - \sum_i^n q_{m+i}(h_n - h_i) - \sum_i^n q_{m-i}(h_i - h_{fw})}{h_n - h_{fw}} \qquad (86)$$

where

$$h_{fw} = h(T_{fw}, p_{fw}) \qquad (87)$$

$$h_n = h(T_{no}, p_{no}) \qquad (88)$$

where

q_{mfw}	is the feedwater mass flow = flow through first element
Φ_i	is the heat flow to element *i*
q_{mi}	is the mass flow through element *i*
q_{m+i}	is the mass flow added into element *i*
q_{m-i}	is the mass flow extracted from element *i*
h_n	is the outlet enthalpy from element *n*
h_{fw}	is the inlet enthalpy to the vector

For each water / steam element we solve directly element by element mass flows of elements *2,...,n*.

$$q_{m\,i} = \begin{cases} q_{m\,i-1} & ;\ \text{mass is not exchanged} \\ q_{m\,i-1} + q_{m+i} & ;\ \text{mass is extracted} \\ q_{m\,i-1} - q_{m-i} & ;\ \text{mass is added} \end{cases} \qquad (89)$$

After which we solve simultaneously for inlet enthalpy of elements $2,\ldots,n$

$$h_{i\,i} = h_{i-1\,o} \tag{90}$$

and outlet enthalpy of elements $1,\ldots,n$

$$h_{i\,o} = \begin{cases} h_{i\,i} + \dfrac{\Phi_i}{q_{m\,i}} & ;\ \text{mass is not exchanged} \\ h_{i\,i} & ;\ \text{mass is extracted} \\ \dfrac{h_{i-1o}\, q_{m\,i-1} + h_{+i}\, q_{m+i}}{q_{m\,i-1} + q_{m+i}} & ;\ \text{mass is added} \end{cases} \tag{91}$$

Attemperating mass flows

Defining the ratio of each attemperating mass flow q_{mai} to the total attemperating flow

$$a_i = \frac{q_{ma\,i}}{\sum_i q_{ma\,i}} \tag{92}$$

We solve each attemperating mass flow q_{mai} using the overall energy balance of elements $n+1,\ \ldots,\ j$

$$q_{ma\,i} = \mathrm{Max}\left(a_i \frac{\sum_{i=n}^{j} \Phi_i - q_{mn}\,(h_j - h_n)}{h_j - \sum_i^j a_i\, h_{ai}},\ 0 \right) \tag{93}$$

where

q_{mai}	is the attemperating flow to element i
h_j	is the outlet enthalpy from the last element j
h_{ai}	is the enthalpy of the attemperating flow to element i

The mass flows of superheating elements are solved with equations *89* as before. In addition to the attemperating flows there are fixed flows. One of the fixed flows is the sootblowing flow.

The inlet and outlet enthalpies of superheating elements are solved with equations *90* and *91* as previously.

If there is not enough superheat then the dividend in equation *93* is negative and the result is 0 mass flow. In practice we should choose a small positive mass flow, because there are leaks in the attemperating control valves.

4.4 Solving the other mass flow vectors

In addition to the steam producing water / steam vectors there are a number of other mass flow vectors in the recovery boiler. The most important ones are airflows. Other flows are flue gas flows and water heating. Usually the inlet mass flows q_{m1} for these flows are design variables.

The unknown values of other heat absorbing vectors are solved similarly as with the water / steam vector. We have a known inlet mass flow, so we can directly proceed to solve for the unknown mass flows.

i) Calculate mass flows through all elements.
ii) Calculate inlet and outlet enthalpies for all elements.

For each vector element the mass flows are for elements *2,...,m* from mass balances

$$q_{m\,i} = \begin{cases} q_{m\,i-1} & ;\text{ no mass exchange} \\ q_{m\,i-1} + q_{m+i} & ;\text{ mass is added} \\ q_{m\,i-1} - q_{m-i} & ;\text{ mass is extracted} \end{cases} \tag{94}$$

After which we solve simultaneously for inlet enthalpy of elements *2,...,m*

$$h_{i\,i} = h_{i-1\,o} \tag{95}$$

and for outlet enthalpies *1,...,m* from the element energy balance

$$h_{i\,o} = \begin{cases} h_{i\,i} + \dfrac{\Phi_i}{q_{m\,i}} & ;\text{ no mass exchange} \\ h_{i\,i} & ;\text{ mass is extraced} \\ \dfrac{h_{i-1\,o}q_{m\,i-1} + h_{+i}\,q_{m+i}}{q_{m\,i-1} + q_{m+i}} & ;\text{ mass is added} \end{cases} \tag{96}$$

4.5 Solving the process unit vector

For all flue gas flow elements the inlet flow equals the outlet flow. There is eight equations and ten mass flows. We choose the flow 11 and the flow 12 to be the torn iteration values. The mass flow matrix for flue gas flows is then

$$C\,q = D \qquad (97)$$

where

$$C = \begin{bmatrix} 1 & & & & & & & \\ -1 & 1 & & & & & & \\ & -1 & 1 & & & & & \\ & & -1 & 1 & & & & \\ & & & -1 & 1 & & & \\ & & & & -1 & 1 & & \\ & & & & & -1 & 1 & \\ & & & & & & 1 & \end{bmatrix} \qquad (98)$$

and

$$q = \begin{bmatrix} q_{m1} \\ q_{m2} \\ q_{m3} \\ q_{m4} \\ q_{m5} \\ q_{m6} \\ q_{m7} \\ q_{m18} \end{bmatrix} \qquad (99)$$

$$D = \begin{bmatrix} q_{m11} + q_{m12} \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} \qquad (100)$$

With given known input flows 11 and 12 all the other mass flows are directly solvable.

The heat balances for the process units give seven equations for the heat flows

$$G\,\Phi = F \qquad (101)$$

where

$$G = \begin{bmatrix} 1 & & & & & \\ & 1 & & & & \\ & & 1 & & & \\ & & & 1 & & \\ & & & & 1 & \\ & & & & & 1 \end{bmatrix} \qquad (102)$$

and

$$\Phi = \begin{bmatrix} \Phi_1 \\ \Phi_2 \\ \Phi_3 \\ \Phi_4 \\ \Phi_5 \\ \Phi_6 \\ \Phi_7 \end{bmatrix} \qquad (103)$$

$$F = \begin{bmatrix} -h_{11o}q_{m11o} - h_{12o}q_{m12o} \\ \Phi_6 + \Phi_{e1} \\ \Phi_5 - \Phi_{e1} + \Phi_{e2} \\ \Phi_{10} + \Phi_4 - \Phi_{e2} \\ \Phi_8 + \Phi_3 \\ 0 \\ \Phi_1 \end{bmatrix} \qquad (104)$$

With given known input flows 11 and 12, external heat flows and heat flows to the elements all the process unit heat flows are directly solvable.

From the process unit heat balances, equation 72, the outlet states can be solved.

$$\mathbf{I}\ H = J \qquad (105)$$

where

$$I = \begin{bmatrix} 1 & & & & & & \\ -1 & 1 & & & & & \\ & -1 & 1 & & & & \\ & & -1 & 1 & & & \\ & & & -1 & 1 & & \\ & & & & -1 & 1 & \\ & & & & & -1 & 1 \end{bmatrix} \qquad (106)$$

and

$$H = \begin{bmatrix} H_1 \\ H_2 \\ H_3 \\ H_4 \\ H_5 \\ H_6 \\ H_7 \end{bmatrix} \quad \text{and} \quad H_i = h_i q_{mi} \qquad (107)$$

$$J = \begin{bmatrix} \Phi_1 \\ \Phi_2 \\ \Phi_3 \\ \Phi_4 \\ \Phi_5 \\ \Phi_6 \\ \Phi_7 \end{bmatrix} \qquad (108)$$

If the process unit heat flows have been solved all the outlet states can be solved.

As with element vectors there exists two types of process units: the mass addition process unit and the heat transfer process unit.

In the mass transfer process unit mass flow is either added or subtracted. There are no heat transfer elements, but external heat flows can be accounted for. If in the process unit new mass is added then new outlet state has to be calculated.

In the heat transfer process unit heat transfer between the flue gas and the elements in the process unit are calculated. The heat exchange with other process units is accounted for. For special process units like the sweet water condenser process unit, no heat exchange with gas and elements occurs.

4.5.1 *Solving the mass exchange process unit*

The element vectors either add or subtract mass flows from process units. For any mass exchange process unit i the mass flow through the process unit is

$$q_{mi} = q_{m\,i-1} + \sum_j q_{mj} \qquad (109)$$

where

q_{mi} is the mass flow through process unit i
q_{mj} is the mass flow j added or subtracted from process unit i

the process unit inlet enthalpy is

$$h_{ii} = h_{i-1o} \qquad (110)$$

where

h_{ii} is the inlet enthalpy of process unit i
h_{i-1o} is the outlet enthalpy of process unit i–1

With known mass flow and inlet enthalpy the outlet enthalpy is calculated from the gas side heat balance

$$h_{io} = \frac{\Phi_{li} + q_{m\,i-1} h_{ii} + \sum_j q_{mj} h_{jo}}{q_{mi}} + h_{ii} \qquad (111)$$

where

Φ_{li} is the heat losses from process unit i

4.5.2 *Solving the heat transfer process unit*

Figure 14 represents a typical process unit side vector for recovery boilers. It consists of one process unit where heat is added 2 and several heat removal process units 3,

... , n, ... j-2 and j. It also has mass addition process units 1, ... , j-1. The order of process units for the general vector can vary.

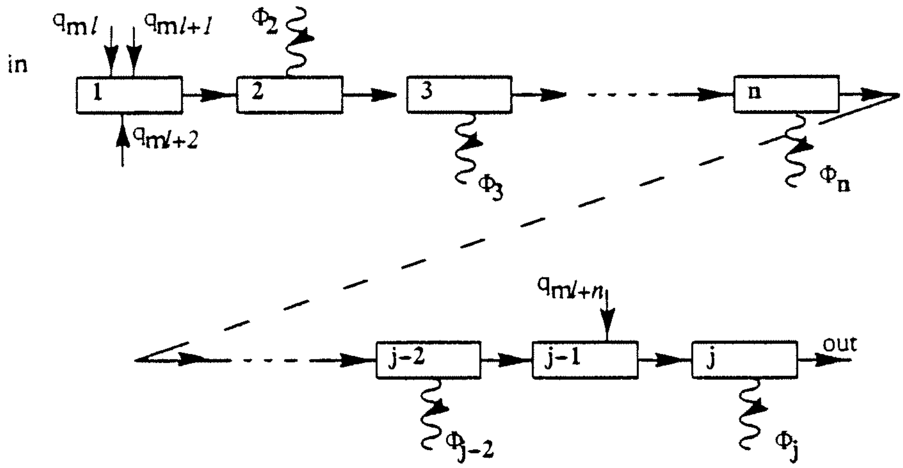

Figure 14. Process unit vector

Figure 15 shows a typical process unit vector temperature profile. The temperature in the process unit vector usually decreases after the initial rise.

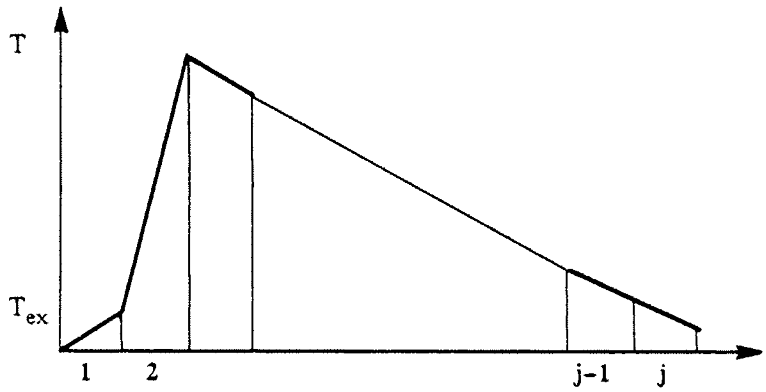

Figure 15. Process unit vector temperature profile

For any heat transfer process unit i the mass flow through it remains constant

$$q_{m\,i} = q_{m\,i-1} \qquad (112)$$

the process unit inlet enthalpy is

$$h_{i\,i} = h_{i-1\,o} \qquad (113)$$

With known inlet mass flow, inlet enthalpy, external heat flows and heat flows to the elements the outlet enthalpy h_{io} can be calculated from the heat balance

$$h_{io} = \frac{q_{m\,i-1} h_{ii} + \sum_j \Phi_j + \sum_j \Phi_{eij} + \Phi_{li} + \Phi_{qi}}{q_{m\,i}} + h_{ii} \qquad (114)$$

where

Φ_{eij} is the external heat flow j to process unit i

4.6 Calculating thermophysical properties

Calculation of enthalpies, heat capacities, thermal conductivities, viscosities and densities is required during the solving of heat and mass balances. Temperature, pressure and when needed steam content were used to record the state of fluid. With physical property algorithms it is easy to express temperatures through known enthalpy and pressure.

$$\Phi_i = q_{m\,i}(h_{li} + h_{io}) \qquad (115)$$

can be expressed with

$$T_{ii} = T_{ii} \, (h_{ii} , p_{ii}) \qquad (116)$$

$$T_{io} = T_{io} \, (h_{io} , p_{io}) \qquad (117)$$

as

$$\Phi_i = \Phi_i(q_{m\,i} , T_{il} , T_{io} , p_{ii} , p_{io}) \qquad (118)$$

For calculation of properties of water and steam, a collection of subprograms, WATLIB created by Talonpoika (85), was used.

Calculating properties for gas mixtures is time consuming. In the recovery boiler the gas composition is almost constant. Speedier calculation procedures for thermophysical properties for gases based on constant mixture are shown in Appendix C.

After revised gas properties calculation still about half of the computing time is used in thermophysical properties calculation. This agrees well with Hooper (92), who found that in his test cases between 50 and 70 % of the time was spent in physical property routines.

4.7 Solving the pressure losses

Accurate calculation of pressure losses is an important part of recovery boiler calculations. The functions are complex and require significant computational effort. For accurate pressure loss calculation the type and design of each heat transfer element must be known.

The main advantage of the new method of calculating pressure losses is that the solution of flow equations and the pressure loss calculation can be separated. The flow solution procedure can then be independent of the type of process units chosen. The calculation speed can be improved because detailed pressure loss equations are not used during flow solution.

The new solution procedure for pressure losses is inspired by an idea expressed by Sas (84) for calculation of the friction factor for the flows in pipe networks.

> *"It is not necessary to iterate pressure losses accurately during each iteration of torn iteration variables, as during each iteration the accuracy of main variables is improved."*

The pressure loss solving proceeds as follows.

i) At first it is assumed that the current pressure loss value corresponds to the current mass flow through each element.
ii) During solution of the surface element vectors new mass flows through elements are calculated. The change in the pressure loss is assumed to be linearly dependent on dynamic pressure, i.e. linearly dependent on the temperature and the square of the mass flow.
iii) New inlet and outlet pressures are calculated based on the pressure loss and flow order information.
iv) Later during the heat transfer coefficient calculation correct water / steam side pressure loss for every element is calculated based on the actual mass flow. Similarly flue gas side pressure losses are calculated. Assuming that the pressure loss is linearly dependent on dynamic pressure a corrected mass flow is substituted for the flow through that element, so that the inlet and outlet pressures remain fixed.
v) The correct flue gas side pressure loss and outlet pressure is calculated during the heat transfer coefficient calculation for every process unit.

When using this technique we sacrifice momentarily the mass balances in the element vectors. However they are always restored during the solution of surface element vectors.

The main assumption used for the solution is that the pressure loss is linearly dependent on the dynamic pressure. Because of changing Reynolds number and nonlinear flow elements, there is small error. The effect of this error to the convergence of tube side and flue gas side pressure loss calculation is examined in the following chapters.

4.7.1 *Linearity of the tube side pressure loss*

The water / steam side solution converges very quickly if the error from linearization assumption is small. To check this we can look at the tube side pressure drop of superheaters, where most of the water / steam side pressure drop occurs. The tube side pressure drop Δp_{ts} is the sum of individual pressure drops

$$\Delta p_{ts} = \Delta p_b + \Delta p_i + \Delta p_o + \Delta p_f \qquad (119)$$

where

Δp_b	is the bend pressure losses
Δp_i	is the inlet pressure losses
Δp_o	is the outlet pressure losses
Δp_f	is the friction losses

All pressure losses are dependent on dynamic pressure p_d

$$p_d = \frac{\rho w^2}{2} = \frac{8 q_m^2}{\pi^2 d_i^4 \rho} \qquad (120)$$

where

w	is the flow velocity
d_i	is the inside diameter of the tube
ρ	is the density
q_m	is the mass flow through one tube

Individual pressure losses are usually expressed as products of loss coefficient and dynamic pressure. If the sum of loss coefficients were constant the pressure loss would be linear.

$$\Delta p_b = n_b \zeta_b \, p_d \qquad (121)$$

$$\Delta p_i + \Delta p_o = \zeta_{io} \, p_d \qquad (122)$$

$$\Delta p_f = \xi \frac{L}{d_i} p_d \qquad (123)$$

where

ξ	is the friction factor
L	is the length of tubes
n_b	is the number of bends
ζ_{io}	is the inlet and outlet loss coefficient
ζ_b	is the bend loss coefficient

For calculation of bend losses the method of Hooper (82) is used.

$$\zeta_b = \frac{b_1}{Re} + b_2\left(1 + \frac{0.0254}{d_i}\right) \qquad (124)$$

The bend tightness coefficients b_1 and b_2 depend on the tube mean bend radius. For each bend type the coefficients must be separately determined. The b_2 coefficient corresponds to the appropriate loss coefficient for the very large Reynolds numbers. The b_1 coefficient represents the increase of resistance with decreasing flow.

According to Kitteredge and Rowley (57) bend losses are strongly influenced by appropriate Reynolds number, so bend losses coefficient ζ_b increases with decreasing flow. When low loads in a kraft recovery boiler are calculated, this effect must be accounted for.

The friction factor ξ is not iterated from the Colebrook equation as is usually the case. Instead a direct functional expression given by Churchill (77) is used. It gives the friction factor as a function of the Reynolds number, Re, the tube diameter, d_i and the tube roughness, k. This expression is valid for both the laminar and the turbulent flow.

$$\begin{aligned} \xi &= 8\left[\left(\frac{8}{Re}\right)^{12} + (A + B)^{-1.5}\right]^{1/12} \\ A &= \left(2.457\ln\left[\left(\frac{7}{Re}\right)^{0.9} + 0.27\frac{k}{d_i}\right]\right)^{16} \\ B &= \left(\frac{37530}{Re}\right)^{16} \end{aligned} \qquad (125)$$

The term $(8/Re)^{12}$ represents the laminar part. If the Reynolds number increases it decreases towards zero. The term $(A+B)^{-1.5}$ represents the turbulent part. If the Reynolds number decreases both A and B increase. The negative power causes the term to decrease towards zero.

For the secondary superheater of the example recovery boiler, see chapter 3.3, the following assumptions can be made. The tube inside diameter is 0.040 m. The tube roughness factor is 0.00005 m. The number of 90° bends is 12. The bend tightness coefficient b_1 for the bends is 1000. The bend tightness coefficient b_2 for the bends is 0.2. For kraft recovery boilers the inlet and outlet loss coefficient ζ_{io} is approximately 1.5. The length of the superheater tubes is 28 m. The mean steam temperature is 380 °C. The mean steam pressure is 50 bar.

The change in the sum of the coefficients from the sum of the coefficients at a mass flow density of 400 kg/m²s for the example secondary superheater is plotted in *Figure 16*. Within the range of about 250 ... 1000 kg/m²s the change is below ±1 %.Within the range of -20 % ... 20 % the change is 0.346 ... -0.272 % respectively.

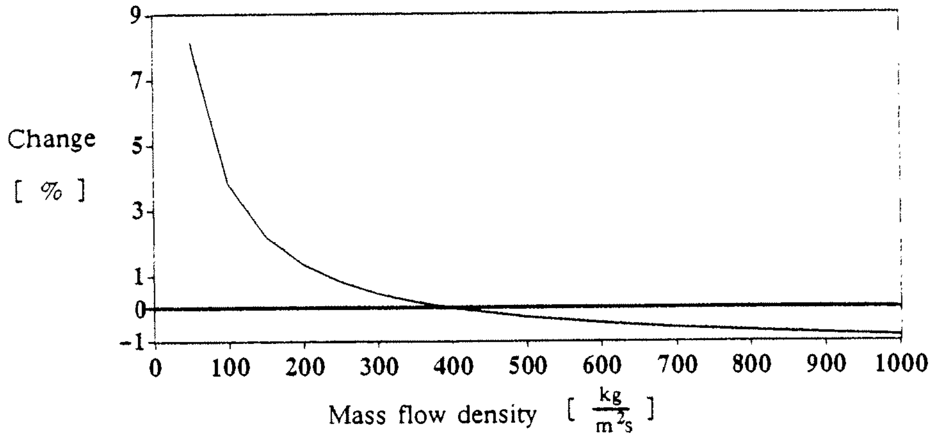

Figure 16. Deviation from linearity for the steam flow

The typical change in the mass flow density during offdesign calculations is about 5 %. For change of this size steam side pressure losses can be iterated almost instantly. In practice the water / steam side pressure losses are always iterated to the accuracy below 0.1 % before the heat flows can be solved.

4.7.3 *Linearity of the flue gas side pressure loss*

Calculation of flue gas pressure drops very accurately is not needed to calculate accurately the flue gas side heat transfer. Convective heat transfer depends very weakly on pressure. A large change in the pressure affects the viscosity and the thermal conductivity only a little. The change in the heat transfer coefficient is even smaller.

Radiative heat transfer depends somewhat on overall pressure. In the furnace and the radiative superheaters where the overall heat transfer is governed by radiative transfer the total pressure remains constant. The recovery boilers are typically operated with a fixed, little below atmospheric pressure in the furnace.

Main pressure drops in the flue gas side occur in the boiler bank and economizer. For tube banks the gas side pressure loss Δp_{gs} is only friction losses

$$\Delta p_{gs} = \Delta p_f \qquad (126)$$

where

Δp_f is the friction losses

For the typical heat transfer surface with horizontal flow and in line arrangement the friction losses are specified according to VDI Wärmeatlas (84).

$$\Delta p_f = n_r \zeta_r \, p_d \qquad (127)$$

where

n_r is the number of rows in heat transfer surface

p_d is the dynamic pressure, calculated at the gas side mean temperature and smallest area

The single row pressure drop ζ_r for in-line tubes is

$$\zeta_r = \zeta_l + \zeta_t \left(1 - e^{-\frac{Re-1000}{2000}}\right)$$

$$\zeta_l = \frac{280\pi \left((s_l^{0.5} - 0.6)^2 + 0.75\right)}{(4 s_l s_t - \pi) s_t^{1.6} \, Re} \qquad (128)$$

$$\zeta_t = 10^{0.47\left(\frac{s_l}{s_t} - 1.5\right)} \left[0.22 + 1.2 \frac{\left(1 - \frac{0.94}{s_l}\right)^{0.6}}{(s_t - 0.85)^{1.3}}\right] + 0.03(s_t - 1)(s_l - 1)$$

where

ζ_l is the laminar part of the pressure drop coefficient

ζ_t is the turbulent part of the pressure drop coefficient

s_t is the dimensionless transverse pitch

s_l is the dimensionless longitudinal pitch

Re is the Reynolds number, calculated at the gas side mean temperature and smallest flow area

For typical boiler banks of recovery boilers, the following assumptions can be made. The tube outside diameter is 0.060 m. The tube longitudinal pitch is 0.150 m. The tube transverse pitch is 0.150 m. The number of tubes is 24. The flue gas temperature is 500 °C. The flue gas pressure is 0.1 MPa. The dynamic viscosity is $33.1 \cdot 10^{-6}$ Pas. The mass flow density is 10 kg/m^{2}s.

The flue gas mass flow does not change within the same load. The change in the coefficient ζ_r with temperature from the same coefficient at 550 °C is shown in *Figure 17*. Within the range of about 460 ... 650 °C the change is below ±1 %. Within the range of -30 ... +30 °C the change is 0.313 ... -0.2852 % respectively.

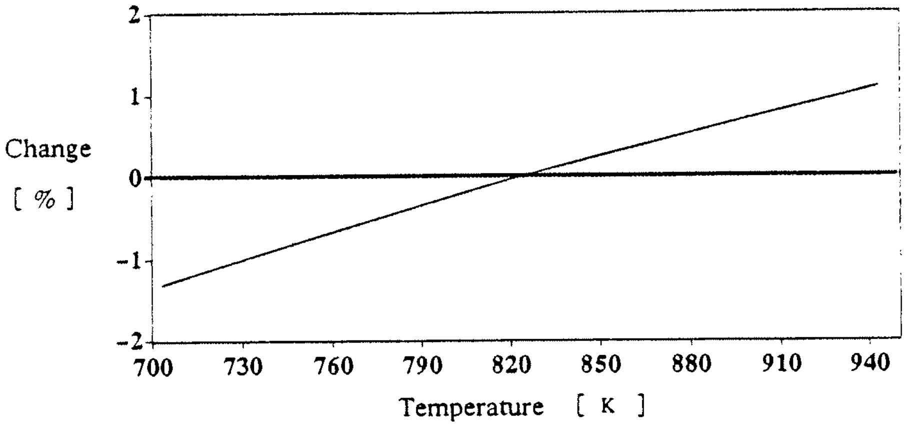

Figure 17. Deviation from linearity for the flue gas flow past boiler bank

The typical change in the average temperature is about 10 °C. Linearization can be used when the flue gas temperature change causes a pressure loss change.

5. STRATEGIES FOR SOLVING TORN ITERATION VARIABLES

There are many methods currently in use for solving new torn iteration parameters. According to Greig (80) the basic ones are Monte-Carlo, direct substitution, gradient methods and Newton's method. A good review of these methods is presented by Gundersen and Hertzberg (83).

The Monte-Carlo method or making random guesses is not the most profitable method in spite of the good results reported by Adibhatla (81). This method was omitted from further study as too computationally intensive.

Gradient methods and the derivatives of Newton's method all require either direct calculation of derivatives or estimation of the Jacobian matrix.

We compare several methods of solving new values of the chosen torn iteration values, heat flows. None of the considered methods requires calculation of the derivatives. One aim of the thesis was to find a method where this could be eliminated.

We study the example steam-generator shown in *Figure 4*. To make our analysis simpler we assume

- furnace is operated so that the exit temperature equals 940 °C
- superheater heat transfer coefficients equal to 40 W/m^2K.
- economizer heat transfer coefficient equals to 20 W/m^2K.
- cage heat transfer coefficient equals to 20 W/m^2K.
- all the rest of the heat transfer coefficients equal to 10 W/m^2K.
- cage heat transfer surface area is 200 m^2
- final superheater heat transfer surface area is 300 m^2
- final superheater side wall heat transfer surface area is 30 m^2
- primary superheater heat transfer surface area is 300 m^2
- primary superheater side wall heat transfer surface area is 30 m^2
- economizer heat transfer surface area is 3000 m^2
- no heat transfer between process units takes place
- flue gas c_p is 1.217 kJ/kgK.

Other values to be used in the example are from chapter 4.2.

5.1 Direct substitution

The most typical engineering approach for solving the problem is to directly substitute the calculated values for the new iteration values.

$$\Phi_i^{n+1} = \Phi_i^n \tag{109}$$

With direct substitution we get the values presented in *Figure 18* for the example boiler. The heat flows for each iteration are scaled with the initial heat flow. Relative variables make it easier to see the magnitude of the variations.

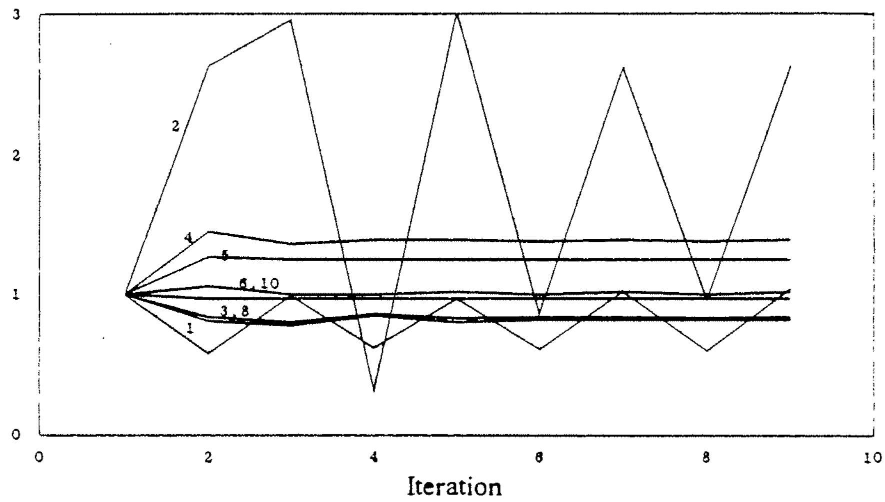

Figure 18. Relative heat flows to elements for direct substitution

It is easy to see that direct substitution is not very successful. Heat to the elements 1 and 2 bounces up and down. The convergence seems to be very slow.

In *Figure 19* the reason for the poor convergence is seen if the magnitudes of the temperature changes are plotted. The process unit outlet temperatures are scaled

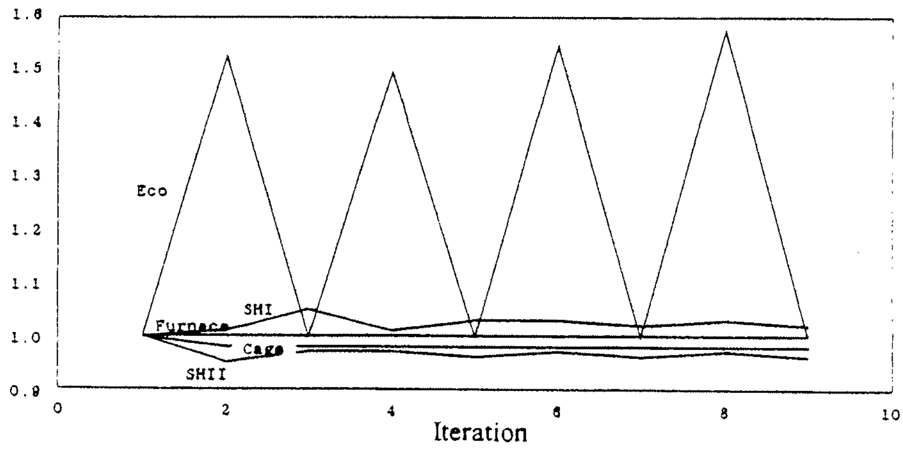

Figure 19. Relative flue gas outlet temperatures for direct substitution

with iteration 1 outlet temperatures. The flue gas exit temperature jumps up and down some 90 °C. The economizer outlet temperature changes are so great that they cause fluctuations in all other variables.

As the economizer outlet temperature does not seem to converge, it seems very unlikely that the whole solution would converge.

5.2 Using a damping parameter

Previously the values were 'bouncing' up and down. A suitable damping parameter e usually speeds up iteration. Heat flows for the new iteration round are expressed as functions of previous values, the damping parameter and calculated values.

$$\Phi_i^{n+1} = e\,\Phi_i^{n} + (1-e)\,\Phi_i^{n_c} \qquad (110)$$

The iteration converges with value 0.5 for the damping parameter as seen in *Figure 20.*

The previously troublesome heat flow to element *2* now converges quickly. Even though there is some fluctuation in all of the heat flows, they all seem to converge in three to five iterations except the sweet water condenser process unit.

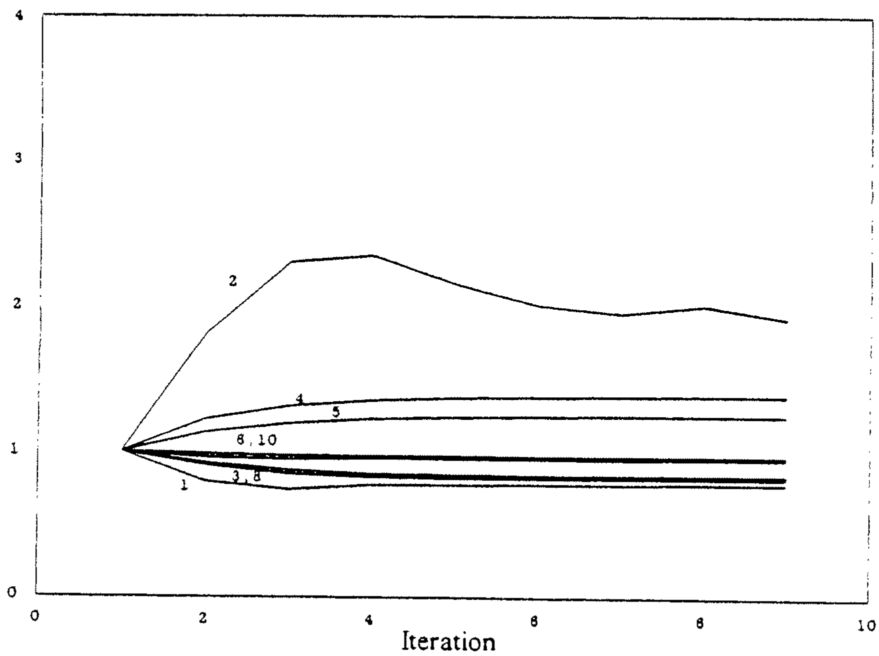

Figure 20. Relative heat flows to elements for damping parameter

The dimensionless temperatures are plotted in *Figure 21*. In it the process unit outlet temperatures are scaled with corresponding initial outlet temperatures.

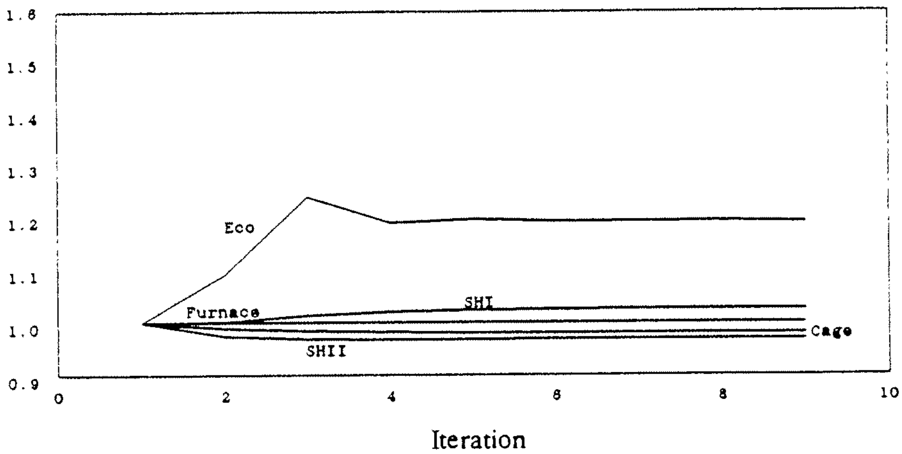

Figure 21. Relative flue gas outlet temperatures for damping parameter

One could argue that with suitable selection of the damping parameter e, we could achieve a much faster convergence, but

I Heat flow 6 (heat to furnace walls) converges towards the correct value. The error is in this case halved each time.

II Heat flow 1 (heat to economizer), heat flow 2 (heat to sweet water condenser) and heat flow 8 (heat to superheater I) change from decreasing values to increasing values or vice versa. The change is over predicted.

III Heat flow 3 (heat to superheater I walls), heat flow 4 (heat to superheater II walls), heat flow 5 (heat to cage walls) and heat flow 10 (heat to superheater II) show constantly decreasing or increasing values. With them the change is under predicted.

There does not exist one good damping parameter value for all of the heat flows. An individual damping parameter should be determined for each individual heat flow. This parameter changes from one case to another. Using the damping parameter approach can be considered a feasible albeit a slow solution.

5.3 Simultaneous calculation of heat flows

In chapters 5.1 and 5.2 we used previously calculated heat flows to determine both the water/steam element vector and the process unit vector state.

In simultaneous calculation the correct heat flows to the elements in a process unit are calculated at the same time as the flue gas outlet state. The heat flows for the new iteration round are determined process unit by process unit as a function of the process unit inlet state.

$$\Phi_i^{n+1} = H\ (\ \Phi_j^{n+1};\ T_j^{n+1};\ p_j^{n+1}) \qquad (\ 111\ )$$

The problem of solving the recovery boiler model is reduced to three steps which are repeated until the desired accuracy is achieved.

Step 1: Torn iteration variables

i) New values of torn iteration variables, i.e. heat flows to heat transfer surfaces, are determined through a suitable algorithm.

Step 2: Solve mass flow vectors

i) Solve steam, water and attemperating mass flows for all water / steam mass flow vectors.

ii) From known heat flows solve temperatures at the inlet and outlet of each surface element for all mass flow vectors.

iii) Scale pressure losses at each element.

iv) At the same time solve the process data, i.e. pressures and thermophysical properties.

Step 3: Solve heat flows vectors

i) Calculate heat transfer coefficients to each heat transfer element in the first process unit

ii) Solve new element heat flows with heat transfer equations.

iii) From heat balance solve new flue gas outlet temperature.

iv) Solve pressure losse for process unit flue gas side and for each element.

v) Repeate steps i) ... iv) for each process unit.

vI) At the same time solve flue gas process data at each process unit.

With the simultaneous method of updating the new torn iteration values, no gradient evaluation is needed. In chapter 7.4 pp. 66 - 68 one way of improving the accuracy of the Step 1 is examined.

Figure 22 shows heat flows to elements for simultaneous calculation. The heat flows to all of the water / steam vector elements converge well. Within four iterations the heat flows have been iterated below an accuracy of $\pm$ 0.1 kW.

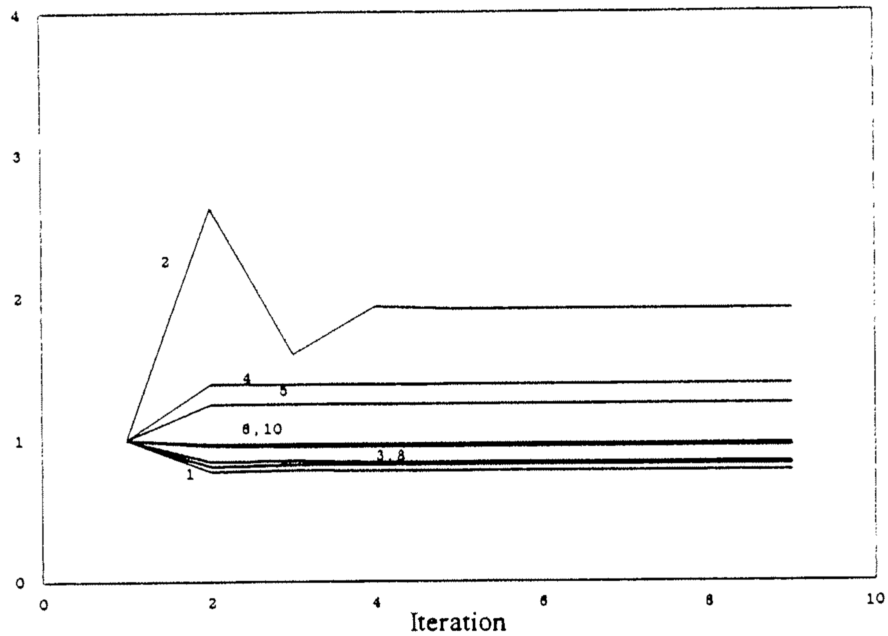

Figure 22. Relative heat flows to elements for simultaneous calculation

The magnitudes of temperature changes are plotted in *Figure 23.* In it the process unit outlet temperatures are scaled with the initial outlet temperatures. The flue gas temperatures convergence also very fast.

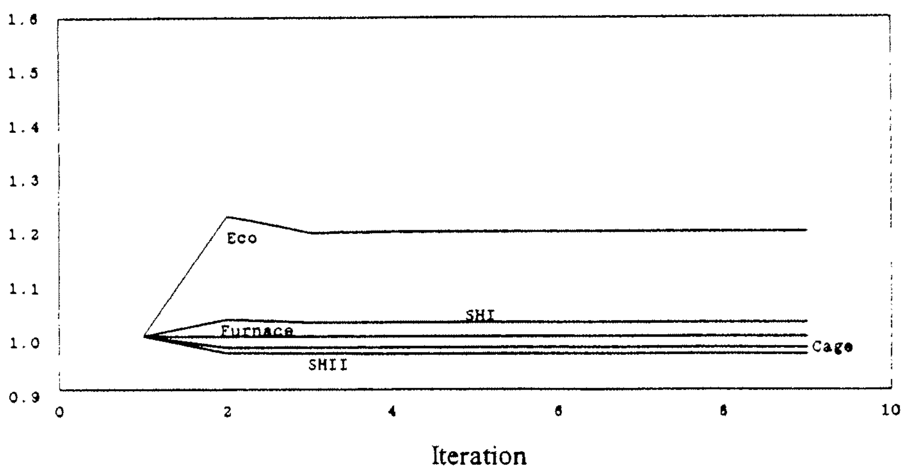

Figure 23. Relative flue gas outlet temperatures for simultaneous calculation

The resulted accuracy is very good. In practice the flue gas temperatures can seldom be measured to the accuracy of ± 1 °C. Measuring heat flows to superheaters with accuracy better than ± 5 % is practically impossible.

To show some of the difficulties involved; when comparing two heat transfer calculation programs, both of which used the same inlet states and the same conductivity for reheater calculations, it was found that the resulted heat flow differed 9 %. The reason was the differences in the water and steam enthalpy calculation routines. Neither routine could be considered more accurate as both routines gave values within the accuracy of the international skeleton table by Schmidt (82) for specific enthalpy.

6. REPRESENTING LOAD-DEPENDENT VARIABLES

Some of the design variables change with the boiler operating conditions. Typical load-dependent variables are fuel flow, air flows and heat generated in the furnace.

The variables which change with load can be linearized through three reference points as shown in *Figure 24*. Usually the reference points can be chosen as the values at minimum load, design load and maximum load.

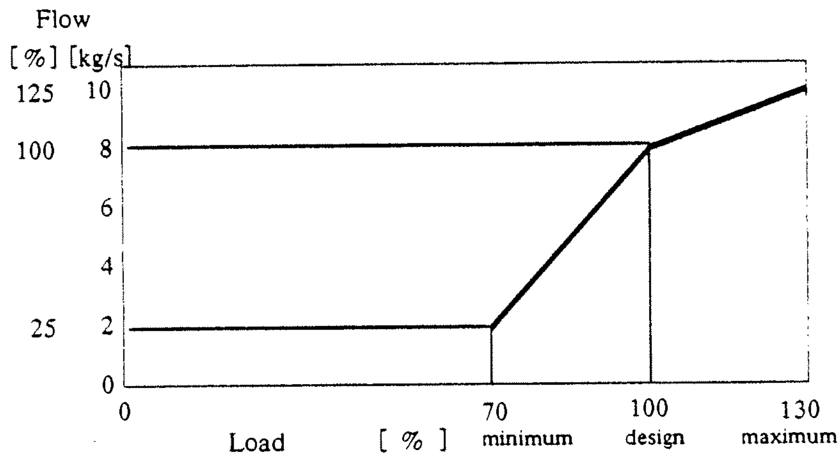

Figure 24. Linearization of an offdesign parameter

The values between defined load points are directly interpolated assuming linear change between the reference points. For example if the tertiary air mass flow at 80 % load case is 4.0 kg/s, then for the 85 % case we add to 4.0 the ratio (85.0–80.0)*6.0/30.0 to get the tertiary air mass flow 5.0 at 85 % load.

Most of the load-dependent variables can be assumed to change linearly. Usually the whole load range can be defined with only three reference points. For clearly nonlinear variables the load range must be divided into smaller subranges. Then for each of these subranges linearization is applied.

6.1 Representing boundary conditions

The water / steam side element vector inlet mass flow is a variable. Other inlet mass flows such as air flows, fuel flows (net liquor), flue gas bypass flows, sootblowing steam flow and continuous blowdown flow need to be defined as functions of load.

To define off design values for the air flows we must consider how we will operate the recovery boiler. Usually the lowest loads have to be considered separately as the air flow ratios tend to change discreetly at the lowest loads.

The net heat input to the recovery boiler furnace is linearly dependent on black liquor flow for design calculations. If needed the heat losses can be separately defined as function of load as well as chemical reactions such as reduction.

6.2 Offdesign gas properties

Recovery boilers are usually operated with fixed excess oxygen content in the exit flue gas. Because the liquor has a fixed composition the flue gas will have a fixed composition except H_2O. The amount of sootblowing and so the amount of H_2O in the flue gas can vary with the load.

To evaluate the thermophysical properties the flue gas can be assumed to be a mixture of seven gaseous components CO_2, H_2O, SO_2, N_2, O_2, CO and H_2. In addition to molar fraction of gaseous species, the amount of dust in the gas has to be defined.

To speed up gas property routines a new calculation method was created as presented in *Appendix C*. Commonly any thermophysical property for the gaseous mixtures is evaluated by first calculating that property for each individual gaseous component. The individual component properties are then used to calculate the mixture property.

Functional representation of important properties can be calculated for the gaseous mixture. This function is similar for the mixture as for each individual gas. By treating the mixture as a separate gas we can reduce the calculation load by a factor of 10.

One set of flue gas mole ratios and one set of air properties is defined for each reference load. Between these loads we assume linear change. Often constant values are used for the whole load range as the properties of flue gas are only weakly dependent on the load.

6.3 Element vector data

In addition to the values already defined for design load there are usually only a few load-dependent variables for typical recovery boiler offdesign cases.

One of the main load parameters is the required exit temperature. We must define this temperature for each surface element vector. For recovery boiler applications the required main steam temperature is constant throughout the whole load range. For large utility and industrial boilers which use sliding pressure load control the main steam temperature is a function of the load.

The water / steam surface element vector exit temperature is the upper limit and we try to attemperate so that this temperature is not exceeded. If there is insufficient heat the final steam temperature will be lower than the maximum and there will be no attemperating.

The inlet temperatures for element vectors do not usually vary with load. This is especially true for recovery boiler applications.

Very seldom do the outlet pressures change. Even the largest recovery boilers operate with fixed steam values.

Offdesign load cases that change element vector data ordering from the design case, such as operation of safety equipment, changes in flow order, etc.must be handled by separate calculations.

6.4 Heat transfer calculation data

The heat transfer calculation data consists of heat transfer process unit dimension data and the heat transfer function to be used. Most of the typical heat transfer functions apply for only a limited range of Prandtl and Reynolds numbers.

A separate process unit heat transfer model could be created for each applicable range. Then the correct process unit should be chosen and the applicability checked during calculation. This approach is clearly unsatisfactory.

The possible process units are listed in *Appendix B*. Heat transfer equations were defined to cover a very wide load range. Both laminar and turbulent areas are included. This requires the usage of several heat transfer models for a single process unit. During low loads the recovery boilers can operate in laminar flow range.

7. OFFDESIGN LOAD CALCULATIONS

The offdesign load calculations are presented using a large modern recovery boiler as an example. The arrangement of heating surfaces is shown in *Figure 25*. The design parameters for the studied 3000 tds/d recovery boiler are shown in *Appendix D*. Even though only one size of recovery boiler is discussed, the results should be applicable to most of the currently operating recovery boilers.

The chosen design features of the boiler are; single drum, furnace with three stage air addition, four stage superheating IA, IB, II and III, vertical boiler bank, evaporator surface, two vertical economizers, I and II and two stage attemperating using sweet water condenser between economizer I and economizer II.

The principal process values are; capacity 3000 tons of black liquor dry solids/day, 34.7 kgds/s, main steam outlet temperature 490 °C and main steam outlet pressure 90 bar.

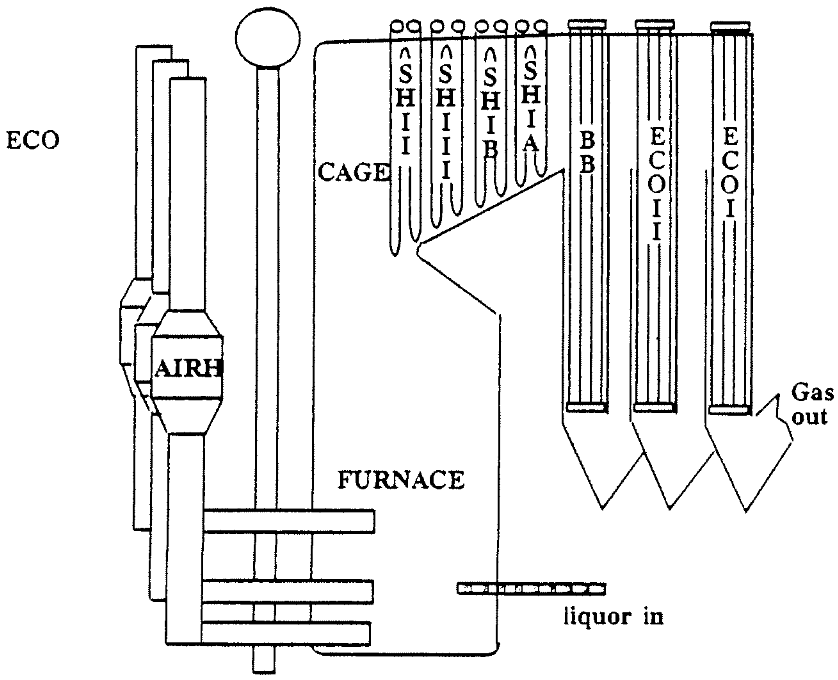

Figure 25. 3000 tds/d recovery boiler

The main steam flow is 135 kg/s. The feedwater inlet temperature is 120 °C. The black liquor dry solids content is 80 % and the black liquor higher heating value is 15 MJ/kgds. The black liquor temperature at guns is 120 °C and the black liquor flow is 43.4 kg/s. The air mass flow is 177.4 kg/s and the flue gas flow is 206.7 kg/s when the excess oxygen is 2.5 %. The smelt flow is 14.0 kg/s.

Using net liquor flow instead of the black liquor flow reduces the number of mass flows. The net liquor flow is defined as the difference of the black liquor flow and the smelt flow.

The furnace exit temperature at nominal load is 940 °C. The cage thermal conductance at nominal load is 17 kW/K. The superheater IA thermal conductance at nominal load is 144 kW/K. The superheater IB thermal conductance at nominal load is 66 kW/K. The superheater II thermal conductance at nominal load is 100 kW/K. The superheater III thermal conductance at nominal load is 68 kW/K. The boiler bank thermal conductance at nominal load is 634 kW/K. The economizer II thermal conductance at nominal load is 343 kW/K. The economizer I thermal conductance at nominal load is 387 kW/K. The radiative heat transfer between process units is accounted for.

The example recovery boiler represents the kind of recovery boilers that will be built during the 1990's. From typical recovery boilers of the 1980's the capacity has about doubled. According to Kiiskilä et al. (92) modern pulp and paper production strives for larger, more integrated plants. Recovery boilers follow this trend.

The required black liquor dry solids content has increased substantially during the last five years. New and better evaporation plants together with advances in black liquor treatment have enabled dry solids to increase by 10 ... 20 percentage points.

The recovery boiler is of modern design. There are no screen tubes as the current industry opinion does not favor them. The one drum design with vertical boiler bank or evaporator has emerged as the current industry standard.

7.1 Vectorized model of 3000 tds/d recovery boiler

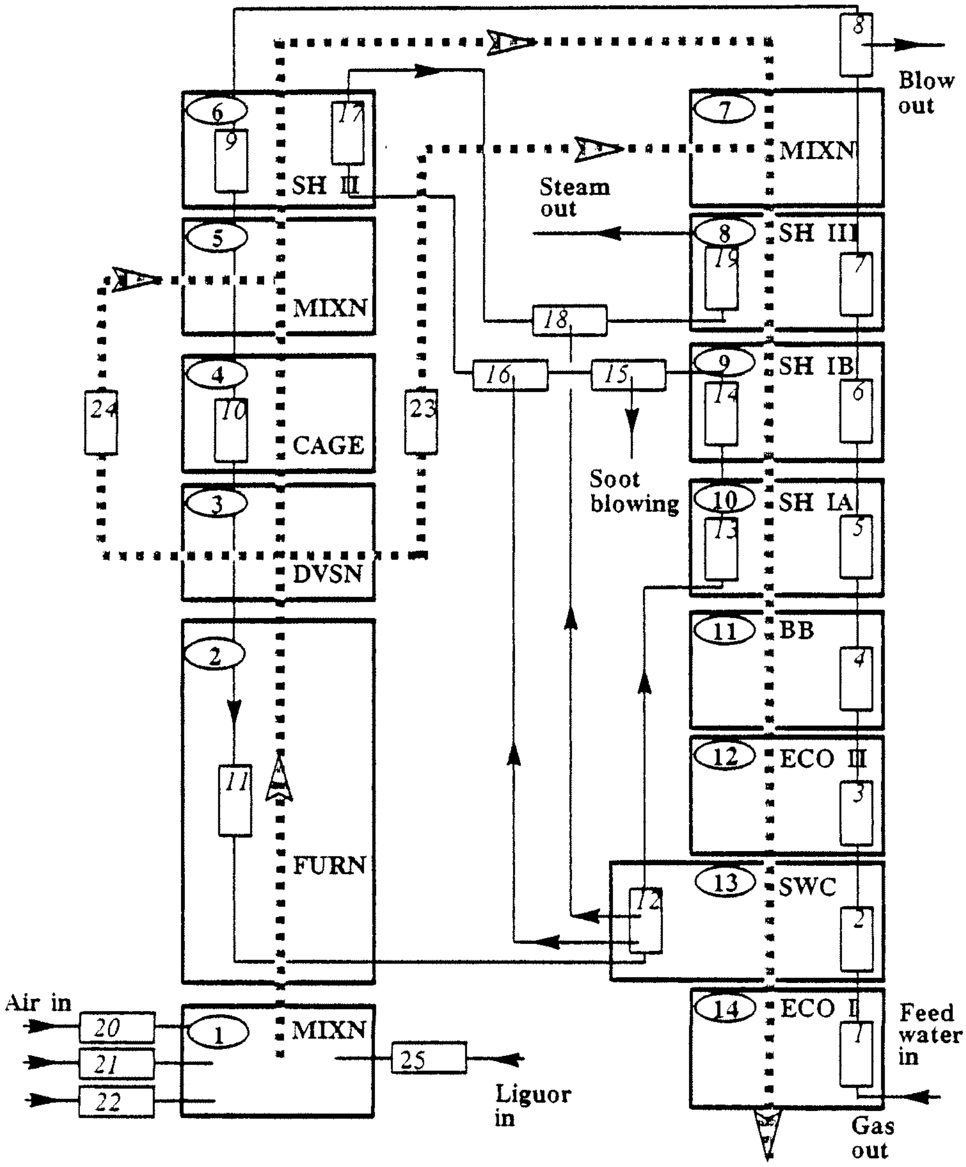

Figure 26. Flowsheet of 3000 tds/d recovery boiler (both process units and surface vectors shown)

There are 14 process units in the example recovery boiler. Compared to the previous example there are two additional superheating surfaces, a boiler bank and a second economizer.

In addition to the extra heat transfer surfaces there are three process units that handle flow splitting at the superheating section. This is done because a simple one-dimensional flow model does not represent the superheater section accurately enough.

Because of more accurate representation there are also elements for the sootblowing steam and the continuous blow down.

7.2 Vectors representing flows

There are seven mass flow vectors in addition to the volume side vector. In this case there is only one water / steam element vector consisting of fourteen heat transferring elements and five mass addition elements. All of the other element vectors consist of a single heat transferring element.

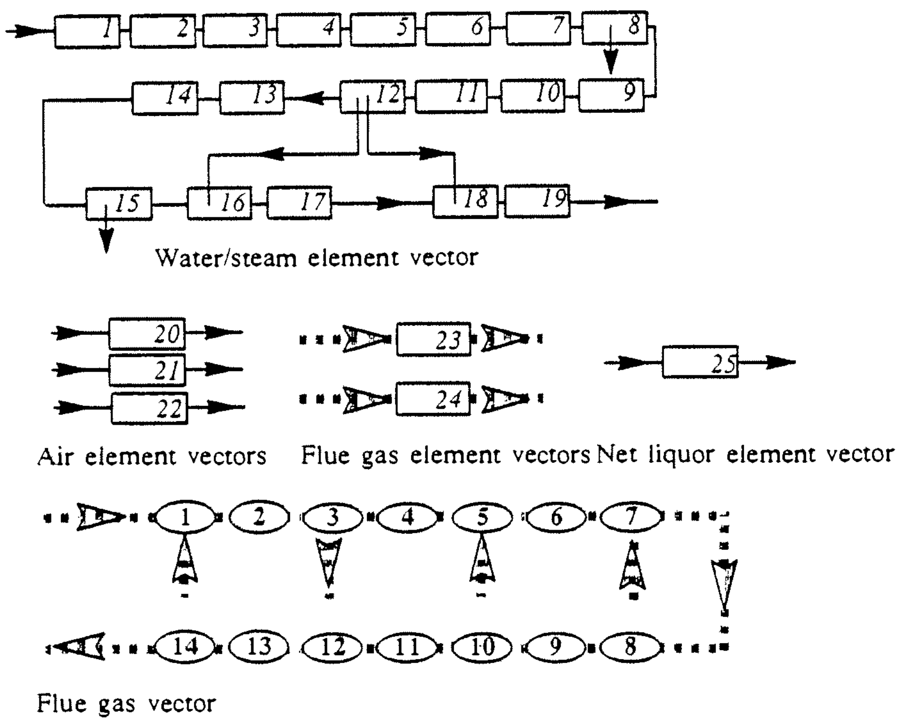

Figure 27. Mass flow vector data of 3000 tds/d recovery boiler

The model has ten heat transferring process units and four mass addition process units. One heat transferring process unit, the sweet water condenser process unit does not transfer heat with the flue gas.

7.3 Recovery boiler temperatures

The example recovery boiler load was varied first from 100 % to 70 % with a 3 % step. Then the load was varied from 100 % to 121 % with 7 % step. In each step only three iteration rounds were needed to achieve convergence below ± 1 kW.

The flue gas side temperatures for the example recovery boiler are shown in *Figure 28*.

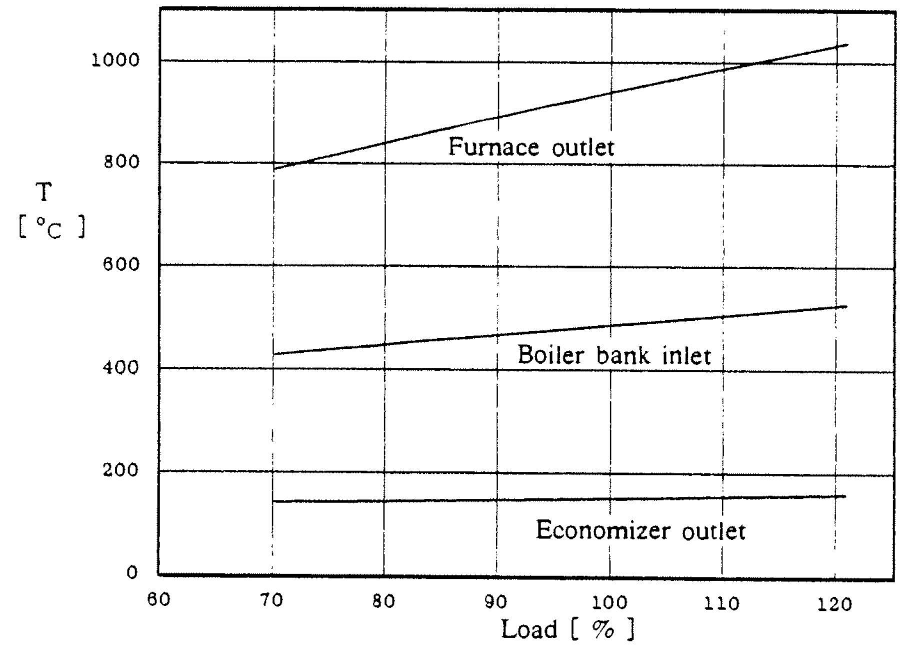

Figure 28. Main flue gas temperatures for 3000 tds/d recovery boiler

In the calculated results the furnace outlet (or bullnose) temperature varies more steeply than with the results presented by Haynes et al. (88). The boiler bank inlet temperature varies less.

The differences are probably caused by the fact that this example boiler is a lot bigger than Haynes'. The example furnace is higher, so the flue gas has more time to cool at low loads. The furnace bottom design is similar.

7.4 Estimation of offdesign heat flows

For the 3000 tds/d example recovery boiler the calculated heat flows are represented in dimensionless form in *Figures 29 .. 32*.

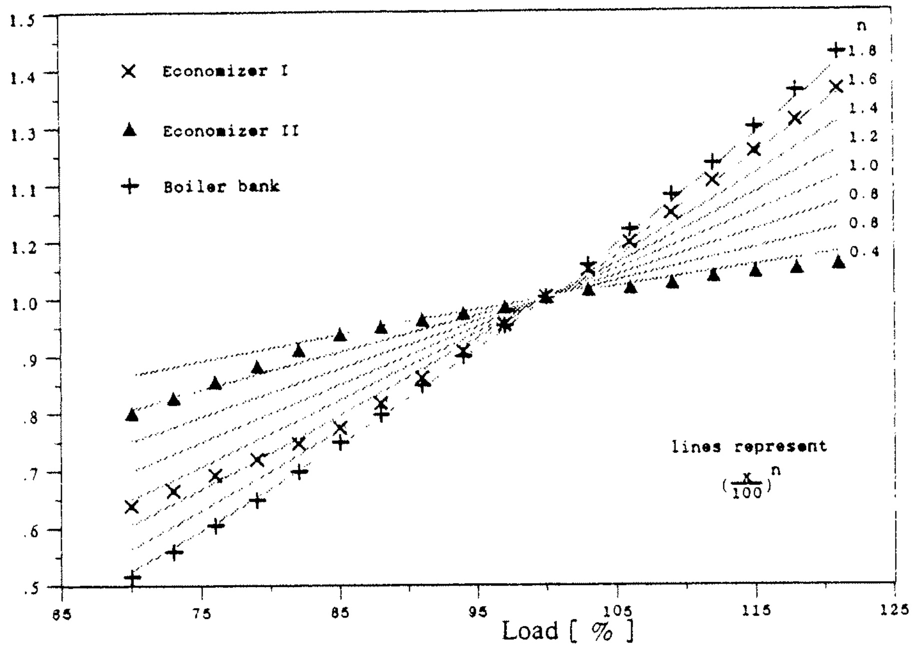

Figure 29. Relative heat flows to economizers and boiler bank

This dimensionless representation has been done to test whether the heat flows for the next offdesign load case can be approximated by the exponent function

$$\Phi_i^{n+1} = \left(\frac{\text{Load}_{n+1}}{\text{Load}_n}\right)^n \Phi_i^n \qquad (112)$$

with a suitable value of n.

Economizer I follows the exponent function with n about 1.6, see *Figure 29*, economizer II follows the exponent function with n about 0.4 and boiler bank follows the exponent function with n about 1.8. The reason for the almost constant heat flow to economizer II is the increase in the inlet temperature because of the attemperating.

In *Figure 30* all of the superheater walls follow the exponent function with n about 1.0.

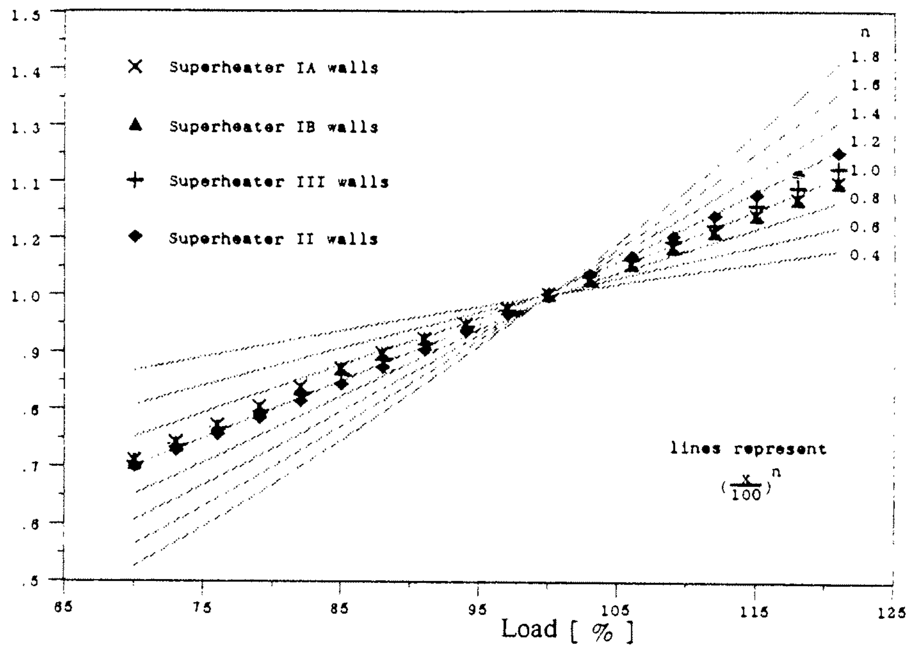

Figure 30. Relative heat flows to side walls

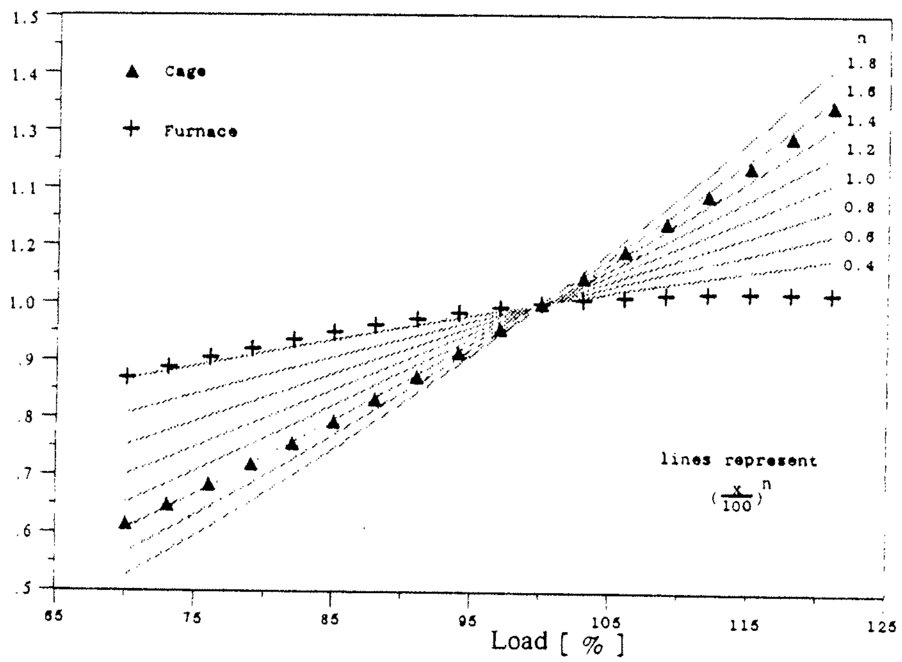

Figure 31. Relative heat flows to furnace and radiative cage

In *Figure 31* furnace follows exponent function with n about 0.2 .. 0.4 and cage follows exponent function with n about 1.5.

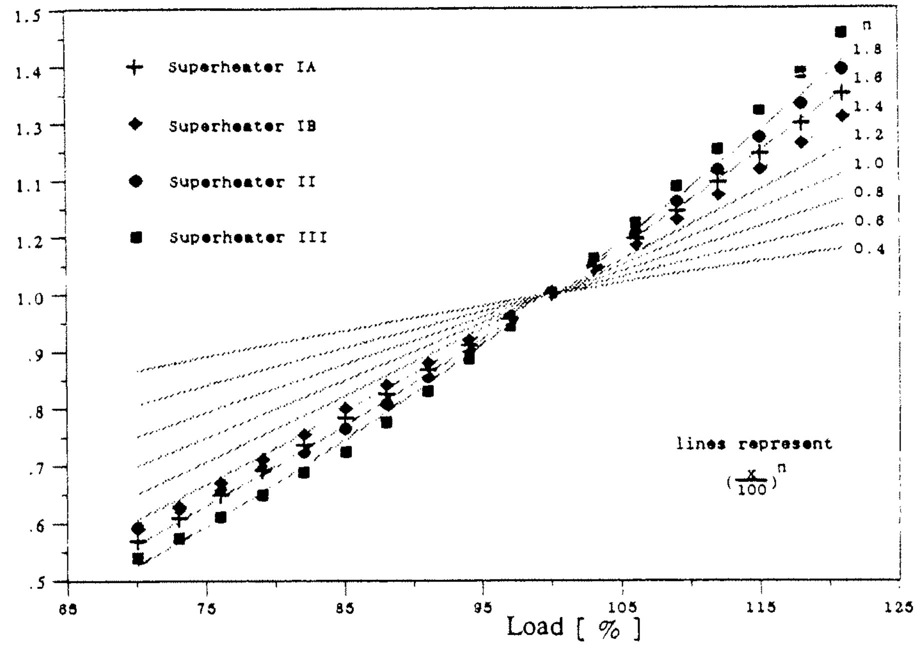

Figure 32. Relative heat flows to superheaters

Superheater III follows the exponent function with n about 2.0, superheater II follows the exponent function with n about 1.7, superheater IA follows the exponent function with n about 1.6 and superheater IA follows the exponent function with n about 1.4. The superheaters have higher exponent than one, because the radiative heat transfer dominates.

For recovery boilers in any given point

1. The change in the heat element heat flows can be approximated with the previously calculated exponent. Vakkilainen and Vihavainen (92) show that this can be used in predicting fouling of the surfaces.
2. If we have not yet calculated the exponent we can choose 1.8 for the superheaters, 1.1 for the superheater walls, 0.3 for the furnace and 1.0 for other types of heat exchanger surfaces. Alternatively values from previous calculation of some other recovery boiler of similar design and capacity can be used.

A value for the exponent is determined after the first offdesign point has been calculated. Afterwards each calculated new load case updates that value. The next calculated point is usually close to the last calculated point. In practice the values from similar types of recovery boilers give a pretty good estimate of the exponent n.

8. EFFECT OF THE BLACK LIQUOR DRY SOLIDS CONTENT ON THE RECOVERY BOILER DESIGN

The dry solids content of the design black liquor of new recovery boilers has been steadily increasing from 65 % to the current level of 70 % - 74 %. There are recovery boilers that burn higher solids than 80 % occasionally, but due to unreliable liquor handling and evaporator operation, long-time operation with higher solids has been infrequent. The main reason is the high viscosities associated with the high solids content. According to Ryham (90) the heat treatment of black liquor, promises high solids with low viscosities enabling high solids operation even in existing boilers.

The effect of black liquor dry solids content increase on furnace, superheater, boiler bank and economizer design is reviewed.

The calculations have been made from a low dry solids value of 60 % to an unrealistically high value of 90 %. The practical case in the existing recovery boilers is the increase of the black liquor dry solids content from 65 % to 80 % dry solids.

This design study was done using typical, current main design values. It is not entirely safe to assume that typical values for today's 70 % dry solids - 2000 tds/d boilers remain valid for future 80 % dry solids - 3000 tds/d boilers.

8.1 Changes in the main design parameters

The material and energy balance calculations were performed using standard methods similar to Adams and Frederick (88). To speed up calculations SOMAT, created recovery boiler material and energy balance program was used.

The material balance values and the assumptions are shown in *Appendix D*. It can be seen that when the dry solids increase the air and black liquor dry solids flows remain constant. Because of the decreased liquor water content the flue gas flow decreases as seen in *Figure 33*. With the same liquor flow, the smelt flow remains constant.

The energy balance shows that when the dry solids increase, the total heat input will slightly decrease as the heat in the black liquor preheat will decrease. Heat available in the furnace decreases because the reduction and smelt losses are constant. The flue gas heat loss decreases with the flue gas mass flow. The H_2O loss will decrease as there is less water to be evaporated. Overall the heat available to generate steam will increase as the losses decrease.

The steam flow increases with increasing black liquor dry solids content, but the sootblowing will remain constant. The blowdown has been increased with increasing steam flow as it depends mainly on the quality of incoming feedwater.

In a recovery boiler with the same main design characteristics but different black liquor dry solids the main operating parameters will change. These changes in the main parameters can be seen in *Figure 33*.

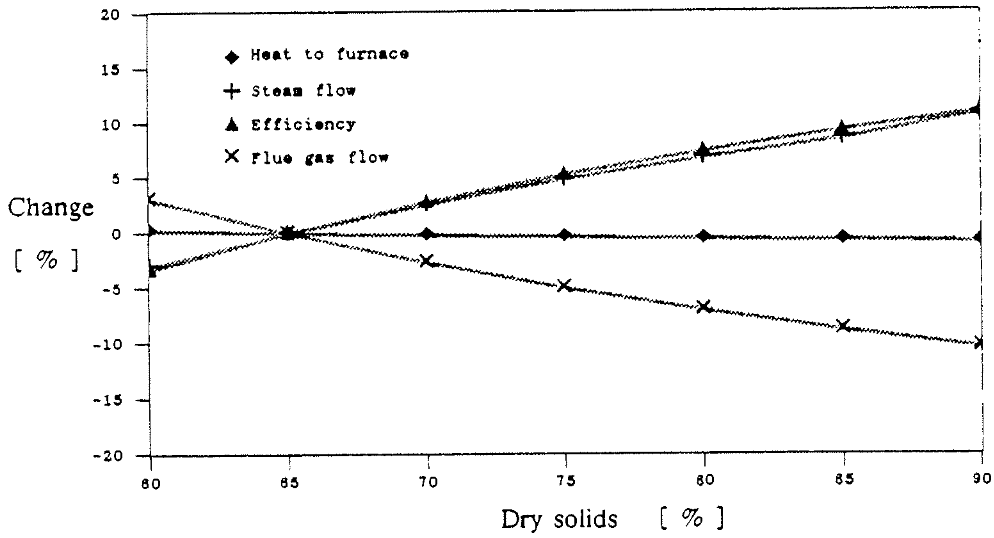

Figure 33. Changes in the main operating parameters

8.1.1 Changes in design basis

The steam generation capability increases with increasing dry solids so for a change of dry solids content from 65 % to 80 % the main steam flow increases 6.9 %. This steam flow increase is about 2.2 % per each 5 % increase in the black liquor dry solids content.

Because the main steam flow increases either larger tubes or more parallel tubes should be used in the water / steam side to achieve the same pressure losses. The required heat transfer area will decrease with increasing black liquor dry solids content because the flue gas flow will decrease.

The steam generation efficiency improves more than the steam generation. This is because less preheating is needed for the black liquor. For the same dry solids change the steam generating efficiency improves from 64.7 % to 69.5 % or 7.4 % (4.8 % percent points).

The increase in the steam generating efficiency is caused mainly by the flue gas flow decrease. The flue gas mass flow decreases 7.0 % for a change of dry solids content from 65 % to 80 %. Because the flue gas flow is smaller the flue gas passages can be designed with smaller cross sectional area and less flue gas fan capacity will be required.

The total heat input to furnace decreases slightly 0.6 % for a change of dry solids content from 65 % to 80 %. The heat in the black liquor is smaller because of reduced black liquor mass flow. The hearth heat release rate in the furnace will remain constant if the furnace bottom area is constant.

The recovery boiler consumes electric power. The main consumption points are the air fans, flue gas fans and the feedwater pump.

The amount of incoming air remains constant with increasing dry solids content. Therefore the power use of the primary, secondary and tertiary air fans remains constant.

The amount of flue gases decreases with increasing dry solids content. The flue gas flow decreases 7.0 %. The flue gas fan power consumption decreases correspondingly.

The amount of steam produced increases with increasing dry solids content. The feedwater flow to produce that steam flow increases 6.9 %. The fan power consumption decrease is an order of magnitude grater than the feedwater pump power increase.

8.1.2 *Unaccounted effects*

The only change in the main operating parameters was the change in the black liquor dry solids content. The flue gas composition remained fixed, except the change in the H_2O content. Further the reduction rate was constant thus the smelt flow didn't change. The amount of sootblowing was kept constant and so the sootblowing mass flow was constant.

The high dry solids black liquor is easier to burn. The current operating experience as presented by Nikkanen (92) suggests, that with increasing black liquor dry solids content the SO_2 emissions are reduced to zero and TRS emissions are almost eliminated. With higher solids less sootblowing is needed. Together with decrease in sulphur containing gases about 1 % higher average reduction (95 .. 96 %) is achievable. The highest reduction rates don't increase, but the variation decreases dramatically.

These changes affect the recovery boiler dimensioning only very little. They were not taken into account to facilitate comparison.

8.2 Heat transfer

The change in black liquor dry solids content will affect the radiative heat transfer, convective heat transfer and heat transfer surface fouling.

8.2.1 Radiative heat transfer

The decrease in water vapor will decrease the overall amount of radiating material in the gas flow. The gas emissivity can be calculated using the band method with known temperature and gas composition according to Rahtu (90).

The radiative heat transfer for large furnaces remains practically constant over the whole dry solids range. The flue gas CO_2 and the H_2O content are high. As the radiative beam length is long, the emissivity changes only slightly with a large change in the H_2O partial pressure. At 1200 °C the change is about 4 % for a change of dry solids content from 65 % to 80 %.

The radiative heat transfer at superheaters decreases only about 10 % with radiative beam length of 1.0 m, when the partial pressure of H_2O decreases from 0.24 to 0.18 bar.

8.2.2 Convective heat transfer

Convective heat transfer is governed by the thermal properties and the flow speed. The change in the flue gas composition affects the specific density, viscosity and thermal conductivity only very slightly. The change in the black liquor dry solids will not affect the convective heat transfer if the design flue gas velocity is constant.

If the heat transfer surfaces remain constant, which is the case in a retrofit, the convective heat transfer will decrease. The increase in dry solids content will decrease gas flow and velocity. The flue gas mass flow decrease of 7.0 % will decrease the convective heat transfer about 5 % for a change of dry solids content from 65 % to 80 %.

8.2.3 Fouling

Nikkanen (92) presents recent experience of high dry solids black liquor. It suggests that fouling would decrease even though the flue gas dust content is increased. In contrast, Jones and Anderson (92) report no change in a CE unit fouling tendencies. Similarly Hagström (91) reports no change in a Tampella unit fouling.

More operating data should be gathered before any firm conclusions can be reached.

8.3 Furnace design

Increasing black liquor dry solids content decreases the furnace loading slightly. The heat input per plan area decreases 0.6 % with increase of dry solids from 65 % to 80 %. The black liquor dry solids flow does not change. The furnace bottom area is typically chosen based on dry solids flow per unit area or the hearth heat release rate. Change in the dry solids does not affect the dry solids flow. The hearth heat release rate in the furnace will remain constant if the furnace bottom area is constant.

The average flue gas flow decreases 7.0 %. If the furnace maximum temperature is constant, this would reduce the maximum flue gas speed by about 0.3 m/s.

8.3.1 Furnace performance

For a similar-sized furnace the operating values with changing black liquor dry solids can be seen in *Table 5*. In The last case 6.9 % more of 65 % dry solids content liquor is fired. The steam generation with the 80 % dry solids and the 6.9 % more of the 65 % dry solids liquor are equal.

Table 5. Furnace outlet temperature for example boiler

Liquor dry solids, %	60	65	70	75	80	85	90	+6.9%
Heat released in furnace, MW	393.0	402.6	410.9	418.1	424.4	430.0	434.9	430.4
Adiabatic flame temperature,°C	1279	1353	1425	1489	1551	1603	1654	1353
Emissivity of gas components, -	0.727	0.720	0.715	0.709	0.703	0.699	0.693	0.702
c_p gas, kJ/kg°C	1.391	1.372	1.355	1.340	1.324	1.314	1.302	1.372
Furnace exit temperature, °C	905	918	928	935	940	944	948	939

The furnace loading is usually determined by either the hearth heat release rate or the firing capacity. Because the black liquor higher heating value and the dry solids flow rate remain constant these parameters show constant loading. This is not the case as we see from the net heat release.

If 80 % dry solids content liquor and 65 % liquor are fired to produce same steam flow, the furnace loading is higher with the 65 % dry solids content black liquor. All furnace loading indicators are higher, including the hearth heat release rate, the firing capacity, total and net heat input.

To generate an equal amount of steam, 1.4 % more net heat must be used when firing 65 % liquor than when firing 80 % liquor. This is because the flue gas losses are greater for the 65 % liquor.

The adiabatic flame temperature increases 200 °C with increase in dry solids from 65 % to 80 %. The increase in maximum furnace temperature is only about half of this.

The actually measured furnace outlet temperature changes range from 20 to 35 °C. The measurement error of furnace outlet temperatures is at least ± 20 °C. The firing practices vary with the changing dry solids levels. This makes reliable comparisons more difficult.

8.3.2 Furnace detail dimensioning

The heat flux to the walls at the lower furnace is increased with increasing dry solids content. More emphasis should be placed to the design of air ports and other openings to avoid increased corrosion rates.

The air flow to recovery boiler furnace remains constant. With present dimensioning criteria the number of air ports, their position and size will not change. As the amount of combustible material will remain constant, there is little incentive to change flow ratios between different air levels.

8.4 Superheater design

As the black liquor dry solids increases the flue gas flow decreases and the steam flow increases. The total heat flow needed for superheating steam increases. Both increase the required flue gas temperature drop.

8.4.1 Heat available to superheat

The black liquor dry solids increase decreases the flue gas mass flow and the flue gas heat capacity. The heat available to superheat is shown in *Table 6*. It decreases 8.8 % with dry solids increase from 65 to 80 %, if the furnace size is constant.

Dimensioning the furnace for constant 918 °C outlet temperature decreases heat to superheat 14.1 %. The decrease is larger than for fixed size furnace as increasing furnace exit temperature compensates the decrease in flue gas flow.

Decrease in the available superheat must be accounted for when recovery boilers are modernized for high solids firing. Jones and Anderson (92) report a significant superheating decrease with CE's Arkansas Kraft boiler.

Table 6. Heat available to superheat

Liquor dry solids, %	60	65	70	75	80	85	90
Flue gas flow, kg/s	217.9	214.1	209.6	206.3	203.4	200.8	198.5
c_p gas, kJ/kg°C	1.391	1.372	1.355	1.340	1.324	1.314	1.302
Steam flow, kg/s	122.8	126.6	129.9	132.8	135.3	137.5	139.4
Furnace exit temperature, °C	905	918	928	935	940	944	948
dry solids affects furnace outlet temperature							
Heat to superheat, kJ/kg_{steam}	894.6	871.3	844.0	819.2	794.3	773.6	753.9
dry solids affects furnace outlet temperature							
Heat to superheat, kJ/kg_{steam}	927.5	871.3	821.7	782.5	748.1	721.2	696.8
furnace dimensioned for constant 918 °C outlet temperature							

Nikkanen (92) reports no decreased superheating in Ahlstrom Äänekoski boiler. The reason for this is the decreased fouling of heat transfer surfaces.

8.4.2 *Design criteria*

Because the total heat flow to steam increases, the total heat flow from the flue gases increases. The temperature drop required of the flue gas will increase. This means that either the design furnace outlet temperature must increase or boiler bank inlet temperature must decrease.

Decreasing the boiler bank inlet temperature is not practical as the available temperature difference is reduced and the average heat transfer coefficient is decreased. Substantially more superheating surface would be needed with low boiler bank inlet temperature.

The solution is to increase the furnace exit temperature by decreasing the furnace height. According to McCann (93) conventional recovery boiler design temperature entering superheater is 930 ... 955 °C. If similar superheating as with 65 % dry solids is required the design furnace exit temperature must clearly be higher than the conventional.

The increase in the furnace exit temperature is usually opposed because of fear of increased corrosion. In modern high solids recovery boilers the superheater corrosion rates are much reduced, because the combustion and air distribution are in better control. Nikkanen (92) reports decreased superheater corrosion with increased dry solids.

8.5 Boiler bank design

Recovery boiler boiler bank or evaporator design is mainly dependent on the convective heat transfer. Flue gas flows vertically downwards along finned tube banks. The high flue gas speed promotes significant convective heat transfer and the flue gas temperature is low so the radiative heat transfer is very small.

The effect of the black liquor dry solids to the modern vertical boiler bank performance is shown in *Table 7*. The chosen boiler bank design inlet temperature is low when compared to values presented by McCann (93) but corresponds to recent trend for new boilers.

Table 7. Effect of dry solids to boiler bank performance

Liquor dry solids, %	60	65	70	75	80	85	90
Flue gas flow, kg/s	217.9	214.1	209.6	206.3	203.4	200.8	198.5
Boiler bank inlet, °C	560.0	560.0	560.0	560.0	560.0	560.0	560.0
c_p flue gas, kJ/kg°C	1.391	1.372	1.355	1.340	1.324	1.314	1.302
Steam flow, kg/s	122.8	126.6	129.9	132.8	135.3	137.5	139.4
Boiler bank exit, °C	404.9	403.6	402.6	401.3	400.0	398.9	397.8
constant surface area and flow area							
Change in area, m²/kgds	+8.6	0	-8.6	-15.8	-21.6	-27.4	-31.7
exit temperature is 400 °C and flue gas design velocity is constant							

The increase in the black liquor dry solids content reduces the flue gas mass flow and the flue gas heat capacity. The heat flux from the flue gas to boiler bank decreases with increasing dry solids. The total steam production and so evaporation increases. The increase in the evaporation capacity of furnace is much larger than the decrease of boiler bank evaporation.

If the boiler bank dimensions are fixed, the flue gas exit temperature decreases with increasing dry solids. The effect is small, flue gas temperature decreases only 3.6 °C, with dry solids increase from 65 to 80 %. High dry solids has not been reported to affect the boiler bank performance. In practice the effect is lower than the typical measurement accuracy.

The design heat transfer area decreases with increasing dry solids content. Less surface is needed to transfer less heat. The required heat transfer area decreases significantly, 21.6 m^2/kgds if the design is based on average flue gas side velocity (constant mass flow per unit area).

8.6 Economizer design

Recovery boiler economizer heat transfer is almost solely dependent on the convective heat transfer. Typically the flue gas temperature is so low that the radiative heat transfer is negligible.

The effect of the black liquor dry solids to the vertical economizer performance is shown in *Table 8*. The economizer design flue gas exit temperature is typical for Scandinavian units.

Table 8. Effect of dry solids to economizer performance

Liquor dry solids, %	60	65	70	75	80	85	90
Flue gas flow, kg/s	217.9	214.1	209.6	206.3	203.4	200.8	198.5
Economizer inlet flue gas, °C	400.0	400.0	400.0	400.0	400.0	400.0	400.0
Economizer inlet feedwater, °C	120.0	120.0	120.0	120.0	120.0	120.0	120.0
c_p flue gas, kJ/kg°C	1.391	1.372	1.355	1.340	1.324	1.314	1.302
Feedwater flow, kg/s	126.8	130.6	133.9	136.8	139.3	141.5	143.4
Economizer exit flue gas, °C similar economizer	158.7	156.0	153.5	151.6	150.0	148.5	145.4
Change in area, m^2/kgds exit temperature is 150 °C and constant velocity	+34.6	0	−31.7	−60.5	−86.4	−109.4	−129.6

The increase in the black liquor dry solids content reduces the flue gas mass flow and the flue gas heat capacity. The heat from the flue gas decreases. If the economizer dimensions do not change, the flue gas exit temperature decreases with increasing dry solids content. The flue gas exit temperature decreases 6 °C, with dry solids increase from 65 to 80 %.

The vertical economizer is usually dimensioned so that average flue gas flow velocity is below maximum design velocity. This velocity is based on the manufactures experience on fouling speeds. Increasing black liquor dry solids decreases flue gas flow and the required cross flow area reduces.

Increasing black liquor dry solids decreases the heat transfer surface area required, if the design is based on flue gas velocity. The decrease is 86.4 m^2/kgds, with dry solids increase from 65 to 80 %.

The same flue gas exit temperature with the same economizer flue gas side flow velocity can be achieved with shorter economizer, when the black liquor dry solids content increases. Continuous tubes without buttwelds are often required to decrease the probability of economizer leaks. The economizer tube manufacturing process limits the maximum tube length. The size of recovery boiler which can be built with just two economizers increases with increasing black liquor dry solids.

9. DISCUSSION

The requirements for the the boiler proposal creation and criteria for computer programs for the proposal phase of steam generator thermal design have been presented. The criteria described were used to create a program to calculate offdesign performance values for steam generators.

A method of representing recovery boiler process in vector form was presented. It was shown that these vectors can be solved without iteration. By proper use of the derived method of calculating multiple heat transfer surfaces simultaneously, the complex, nonlinear heat transfer, mass and energy balance converged quickly.

This quick convergence is attributed to the fact that heat flows were used as the torn iteration parameters as opposed to the current practice of using mass flows and temperatures.

A new method to handle pressure loss calculations with linearization was presented. This method enabled less time to be spent calculating pressure losses.

The derived vector representation of the steam generator was used to calculate off-design operation parameters for one new recovery boiler type.

The 3000 tds/d example recovery boiler was used to study recovery boiler part load operation. It was concluded that a close approximation of heat flows to surface elements can be calculated with a right exponent function. For accurate representation the coefficients for the exponents must be separately evaluated for each surface.

The exponential method is suited to scale temperature raises for prediction of fouling in the recovery boilers.

The same 3000 tds/d example recovery boiler was used to study the effect of the black liquor dry solids increase on recovery boiler dimensioning.

To produce the same amount of steam the recovery boiler should be dimensioned for 6.9 % higher solids flow with the 65 % dry solids liquor than with the 80 % dry solids liquor. The steam generating efficiency would then be 7.4 % lower with 65 % dry solids than with 80 % dry solids.

For a similar size furnace the firing of the 80 % dry solids liquor produces lower hearth heat release rate than the 65 % dry solids liquor at the same steam flow. The furnace outlet temperatures show that a capacity increase with a firing rate increase produces higher loadings than a capacity increase with a dry solids increase.

The economizers, boiler banks and furnaces can be dimensioned smaller if the black liquor dry solids content increases.

The main problem with the increasing black liquor dry solids content is the decrease in the heat available to superheat. If constant furnace exit temperatures is required, the needed superheating surface increases. Because of the smaller flue gas mass flow and heat capacity, there is less heat in the flue gases for superheating.

If possible the furnace exit temperature should be increased by decreasing the furnace height. If similar superheating at 80 % dry solids as with 65 % dry solids is required the design furnace exit temperature should be higher than the conventional 930 ... 950 °C. The increase in the furnace exit temperature is usually opposed because of fear of increased superheater corrosion.

When computer aided design is established, the next logical step is to try to include price functions into the design structure in order to make possible instant price comparisons with alternative designs. The main obstacle is the difficulty in creating functions that predict price changes accurately. Price differences are created because the different number and type of welds, changes in the number of bends, tube material and wall thickness affect the manufacturing hours.Typically manufacturing information is very case sensitive.

There remains the great challenge of integrated, multilevel design. As there are computers capable of different speeds of data processing, commonly divided as micros, minis and mainframes, there are types of design work for each size of machine. Introduction of networks to connect different types of computers brings forth the possibility to use common databases.

With continuous development and upgrading of all types of computing equipment and development of information processing systems, the programs that will be created to handle design tasks will grow more complex and enable more design variations to be checked.

The creation of new graphically-oriented operating systems will enable the user to change from using numbers and letters to visual representations of design data. This will reduce the mistakes and further speed up the design phase.

REFERENCES

Ahmavaara, Outi, Kauppinen, Kari, Lamberg, Jukka and **Utriainen, Jari**, 1987, TATO, Final report of proposal phase operations study. Ahlstrom Boiler Works, Internal study. 367 p.

Adams, Terry N., 1987, Effect of high dry solids firing on recovery boiler performance. Ahlstrom Technical Seminar, Pine Isle, GA, October 8. 13 p.

Adams, Terry N. and **Frederick, James William**, 1988, Kraft recovery boiler physical and chemical processes. American Paper Institute. 256 p.

Adibhatla, Sridhar, 1981, Parametric optimization and configuration optimization of engineering systems. Ph.D. thesis, University of Kentucky, Lexington. 239 p.

Asikainen, Ari, 1983, Development of a calculation model for pulverized peat fired boilers. (in Finnish) M.Sc. thesis, Helsinki University of Technology, Department of Mechanical Engineering, Espoo. 82 p.

Churchill, S. W., 1972, Friction-factor equation spans all fluid flow regimes. *Chemical Engineering*, November 7. pp. 91 - 92.

Demidovic, B. P. and **Maron, I. A.**, 1976, Computational mathematics. MIR Publishers, Moscow. 691 p.

Doležal, Richard, 1967, Large boiler furnaces. Elsevier publishing company, Amsterdam. 394 p.

El-Masri, M. A., 1986, CASCAN - an interactive code for thermal analysis of gas turbine systems. *Computer-aided engineering of energy systems. Vol. 1 - Optimization.* The American Society of Mechanical Engineers, New York. pp. 115 - 124.

Fogelholm, Carl-Johan, 1989, PROSIM, program reference. 15 p.

Greig, D. M., 1980, Optimisation. Longman, London. 179 p.

Gundersen, Truls, 1982, Decomposition of large scale chemical engineering systems. Ph.D. thesis, The Technical University of Norway, Chemical Engineering Laboratory, Trondheim. 234 p.

Gundersen, Truls and **Herzberg, Treje**, 1984, Process flowsheeting. *Use of modelling and simulation in energy research*, Schalien, Randolf von, ed., Åbo Akademi, Department of Chemical Engineering. pp. 13 - 56.

Hagström, Jarkko, 1991, Optimization of black liquor combustion with different dry solids levels in a modernized boiler. (in Finnish) M.Sc. thesis, University of Oulu. 109 p.

Harrison, B. K., 1992, Computational inefficiencies in sequential modular flowsheeting. *Computers in Chemical Engineering*, Vol. 16, No. 7. pp. 637 - 639.

Haynes, Jim B., Adams, Terry N. and **Edwards, Lou L.**, 1988, Recovery boiler thermal performance. TAPPI Proceedings, *Engineering Conference*, Book 2. pp. 355 - 367.

He, B. X. and **Foster, A. R.**, 1986, Functional analysis of thermal power cycle. *Computer-aided engineering of energy systems. Vol. 2 - Analysis and simulation.* The American Society of Mechanical Engineers, New York. pp. 9 - 17.

Horton, Robert R. and **Vakkilainen, Esa K.**, 1993, Comparison of simulation results and field measurements of an operating recovery boiler. To be presented in TAPPI *Engineering Conference.* 14 p.

Hooper, William B., 1981, The two-K method predicts head losses in pipe fittings. *Chemical Engineering*, August 24. pp. 96 - 100.

Jones, Andrew K., 1989, A model of kraft recovery furnace. Ph.D. thesis, The Institute of Paper Chemistry, Appleton, Wisconsin. 173 p.

Jones, Andrew K. and **Anderson, Michael J.**, 1992, High solids firing at Arkansas Kraft. *78th Annual meeting, Technical Section, CPPA.* pp. A39 - A49.

Juslin, Kai, Tuuri, Sami and **Hautamaa, Jukka**, 1992, Dynamic simulation of a recovery boiler using the APROS simulation program. TAPPI Proceedings, *International Chemical Recovery Conference, Seattle*, Book 2. pp. 293 - 300.

Jutila, Esa, Pantsar, Ossi and **Uronen, Paavo**, 1978, Computer control of a recovery boiler. *Pulp and Paper Canada*, Vol. 79, No. 4. pp. 61 - 65.

Jutila, Esa, Uronen, Paavo, Huovinen, Niilo and **Peltola, Hannu**, 1981, Improved recovery boiler control system reduces energy, costs. *Pulp & Paper*, Vol. 55, July. pp. 133 - 38.

Kaijaluoto, Sakari, 1984, Process optimization by flowsheet simulation. Technical Research Centre of Finland, Publications 20, Espoo. 87 p.

Karvinen, Reijo, Hyöty, Paavo and **Siiskonen, Pekka**, 1991, The effect of dry solids content on recovery boiler furnace behavior. *Tappi Journal*, December pp. 171 - 177.

Kiiskilä, Erkki, Lääveri, Anu, Nikkanen, Samuli and **Vakkilainen, Esa**, 1992, Possibilities for new black liquor processes in the pulping industry. *Power Production from Biomass, JALO - seminar Espoo.* VTT, Laboratory of Fuel and Process Technology. 10 p.

Kittredge, C. P. and **Rowley, D. S.**, 1974, Resistance coefficients for laminar and turbulent flow through 1/2 inch valves and fittings. Transactions of the ASME, *Journal of Engineering for Power*, Vol. 79. pp. 1579 - 66.

Lundborg, Sten T., Brooks, Todd R. and **Edwards, Louis L.**, 1992, Diagnosing recovery boilers and establishing and evaluating retrofit concepts using boiler simulation. TAPPI Proceedings, *International Chemical Recovery Conference, Seattle*, Book 1. pp. 113 - 120.

McCann, Derek, 1993, A review of recovery boilers process design. *Pulp & Paper Canada*, Vol. 94, No. 4. pp. 17 - 22.

Nikkanen, Samuli, 1992, Advanced high dry solids combustion. *Ahlstrom Recovery Processes.* Varkaus. pp. 35 - 51.

Nishio, Masatoshi, 1977, Computer aided synthesis of steam and power plants for chemical complexes. Ph.D. thesis, The University of Western Ontario, London. 330 p.

Perez, E., 1991, A Computer method for thermal power cycle calculation. Transactions of ASME, *Journal of Engineering for Gas Turbines and Power*. Vol. 113, April. pp. 184 - 189.

Perregaard, J., **Pedersen, B. S.** and **Gani, R.**, 1992, Steady state and dynamic simulation of complex chemical processes. *Transactions of the Institution of Chemical Engineers, Part A*, Vol. 70, Number A2. pp. 99 - 109.

Purola, Veli Matti, 1982, Recovery boiler fouling and sootblowing control. (in Finnish) M.Sc. thesis, University of Oulu. 94 p.

Raiko, Markku, 1982, Composition and use of nonlinear simulation models for once through boilers. (in Finnish) Licentiate thesis, Helsinki University of Technology, Department of Mechanical Engineering. 56 p.

Rahtu, Jari, 1990, Heat transfer in the recovery boiler superheater section. (in Finnish) M.Sc. thesis, Lappeenranta University of Technology, Department of Energy. 92 p.

Reese, Carl and **McHugh, Robert**, 1987, PCALC3 users guide. Pyropower Corporation, San Diego, California. 109 p.

Robinson, G. F. and **Shiue, Y. L.**, 1987, Recent developments in mathematical modelling of transient operation of steam generators and pulverizers. *International technical conference, Twentieth meeting of Sulzer monotube boiler licensees*, Hartford, Connecticut, Paper 4. 12 p.

Rosen, M. A. and **Scott, D. S**, 1986, Comparison of energy and exergy efficiencies for cogeneration systems. *Computer-aided engineering of energy systems. Vol. 2 - Analysis and simulation*. The American Society of Mechanical Engineers, New York. pp. 41 - 47.

Ryham, Rolf, 1990, High solids evaporation of kraft black liquor using heat treatment. *TAPPI Engineering Conference, Westin Seattle, Seattle Washington, September 24 - 27*, Proceedings, Book 2, TAPPI Press. pp. 677 - 681.

Ryti, Henrik, 1969, Heat transfer. (in Finnish) *Technical handbook*, Part 4, K. J. Gummerus, Jyväskylä, pp. 595 - 600.

Sandner, Theodor, 1983, Verfaren zur berechnung von thermodynamischen kreisprozessen in dampfkraftanlagen. *VGB Kraftwerkstechnik* 63, Heft 1, Januar. pp. 1-7.

Sas, Marc, 1984, Discussion of "The Colebrook equation utilized in the Hardy-Cross method". *Journal of Pipelines*, No. 4. pp. 311 - 314.

Schalien, Randolf von, 1985, Engineering thermodynamics and modelling.Åbo Akademi, Department of Chemical Engineering. 157 p.

Schmidt Ernst, ed., 1982, Properties of water and steam in SI-units. Springer-Verlag, Berlin. 194 p.

Sciubba, E. and **Su, T. M.**, 1986, Second law analysis of the steam turbine power cycle: a parametric study. *Computer–aided engineering of energy systems. Vol. 3 - Second law analysis and modelling*. The American Society of Mechanical Engineers, New York. pp. 151 - 165.

Shiang, Niann Tsair, 1986, Mathematical modelling and simulation of recovery furnaces. Ph.D. thesis, University of Idaho, Chemical Engineering Department, Moscow, Idaho. 280 p.

Somerton, C. W., et al., 1987, RANKINE: a computer software package for the analysis and design of steam power generating units. Transactions of the ASME, *Journal of Engineering for Power*, Vol. 109, April. pp. 222 -227.

Sonnenschein, H., 1982, A modular optimizing calculation method of power station energy balance and plant efficiency. Transactions of the ASME, *Journal of Engineering for Power*, Vol. 104, April. pp. 255 - 259.

Stecco, S. S., **Gusso, R.** and **Galletti, A.**, 1986, Energy-exergy analysis of an actual regenerative cogenerating gas turbine plant. *Computer–aided engineering of energy systems. Vol. 3 - Second law analysis and modelling*. The American Society of Mechanical Engineers, New York. pp. 69 - 76.

Talonpoika, Timo, 1985, WATLIB, Property library for water and steam, Users manual. (in Finnish) Lappeen Prosessiohjelmointi, Lappeenranta. 22 p.

Tolvanen, Antti, 1982, Recovery boiler load deviation. (in Finnish) M.Sc thesis, Lappeenranta University of Technology. 109 p.

Vakkilainen, Esa, 1986, Optimization of main variables in a combined cycle plant. (in Finnish) Licentiate thesis, Department of Energy Technology, Lappeenranta University of Technology, Lappeenranta. 90 p.

Vakkilainen, Esa and **Vihavainen, Esa**, 1992, Long term fouling of recovery boiler surfaces. TAPPI Proceedings, *International Chemical Recovery Conference*, Seattle, Book 1. pp. 255 - 261.

Vakkilainen, Esa, **Hautamaa, Jukka**, **Anttonen, Timo** and **Nikkanen, Samuli**, 1991, Flows in the upper region of the recovery boilers. *Advances in Chemical Recovery, AIChE Meeting, November 17 - 22*, Los Angeles. 15 p.

Valero, A., et al., 1986, GAUDEAMO: a system for energetic/exergetic optimization of coal power plants. *Computer-aided engineering of energy systems. Vol. 1 - Optimization*. The American Society of Mechanical Engineers, New York. pp. 43 - 49.

VDI-Wärmeatlas, Berechnungsblätter für den Wärmeübergang. 1984, 4. auflage, Verein Deutcher Ingenieure, VDI-Verlag GmbH, Düsseldorf.

Westerlund, Tapio, 1984, Simulation of power plants and heat exchanger networks. (in Swedish) *Use of modelling and simulation in energy research*. Schalien, Randolf von, ed., Åbo Akademi, Department of Chemical Engineering, Turku. pp. 111 - 139.

Winter, P., 1990, Perspectives on computer aided process engineering. *Transactions of the Institution of Chemical Engineers, Part A*, Vol. 68, September. pp. 403 - 406.

CRITERIA FOR THE DESIGN PHASE PROGRAMS

While the designer works with the steam generator design there are several different types of operations. Configuration creation is used for building and modifying different configurations of steam generators to suit the specific needs of the current task. Dimensioning is used for specifying the shape parameters of all heat transfer surfaces to best fit required operating conditions. Calculation is used for inspecting how the steam generator works under different operating conditions.

In the configuration creation the designer specifies the configuration of heat transfer surfaces in the current of steam generator. To define the steam generator the type and number of heat exchangers must be given. Next, the flow directions in the heat exchanger surfaces and all the interconnections must be specified. For heat transfer calculations the designer must define the suitable heat transfer function. Usually it is enough to pick or slightly modify one configuration already created.

Configuration creation must include routines to add heat exchanger subvolume types, erase heat exchanger subvolume types, add mass exchange subvolume types, erase mass exchange subvolume types, change the way the heat transfer surfaces are connected, i.e. flow directions and change the heat transfer calculation routine.

In dimensioning the designer defines the various shape parameters of each heat transfer surface. This means specifying parameters like tube spacings, diameters, wall thicknesses, materials, number of rows etc. When changing parameters, results of those changes should be calculated.

Dimensioning must include routines for altering surface size and shape parameters, specifying heat transfer equation parameters (many correction parameters are functions of shape parameters), design load performance evaluation and offdesign performance evaluation.

While calculating, the designer should see and be able to examine the steam generator under design. He calculates how the steam generator will function under determined working conditions.

Calculation must include routines for design load performance evaluation and offdesign load performance evaluation.

If we add routines to visualize these tasks this helps the computer - user interaction. It helps the designer to 'see' what he is doing. This is done by routines to show different parameters and variables on screen such as temperatures, pressures and massflows and graphical output which includes view of each individual heat transfer surface and an overall view of the steam generator.

Requirements for the design phase programs

During the conducted study, Ahmavaara et. al. 87, the four most important qualifications required of the proposal phase programs were

1. Clear user - program interaction; the program should be easy to use, self-explanatory and give considerable output.
2. Versatility; the steam generator types we design today are not necessarily the types we will design tomorrow, but the current program should work also tomorrow.
3. Modularity; since during the lifetime of a program about two thirds of the work is directed to maintenance, modularity means ease of change.
4. repeatability; results obtained should be obtainable also tomorrow, all main variables and parameters must be recorded.

To fulfill these goals five new requirements were used in the offdesign program created.

1. Interactive use. The program runs step by step all the time controlled by the user. This means that; intermediate results must be shown, direction of calculations must be alterable and many pauses are inserted in the program. Most of the current steam generator design programs run only in the batch mode.
2. Capability to produce various kinds of documents at any stage of program operation, with which proper functioning of the program can be ascertained. The type of documents must be such that their information content is high and they can be used later. The current design phase programs frequently produce only one standard type of output.
3. Ability to 'spot out' false input and be able to continue when getting one. Very seldom is there any emphasis placed on developing routines solely to check input values.
4. Recalculations can be performed. Use of false assumptions that lead to impossible output should be correctable. As error condition is reached it should be possible to go back to the last correct stage. Usually, proposal phase programs cannot store their current state for restart.
5. Ability to produce output in several languages. When output is based on filling stored sheets with appropriate numbers, output language can be changed by choosing different output forms. Necessity to have multiple language output is often overlooked.

Modularity and hierarchy

To make changes in the program as easy as possible, it should be made modular and heat transfer equation independent.

Modularity means that the program is made of many subroutines arranged into hierarchical order.

Heat transfer equation independent modelling means that instead of source code which does predetermined types of calculations heat exchanger by heat exchanger, equations describing the heat transfer process are separate from dimension data. The steam generator is described as consisting of some tens of heat transfer surface models. Each heat transfer surface model describes some type of heat exchanger surface. Typical models are furnace model, vertical economizer model, panel superheater model etc. For each surface model any valid heat transfer model can be used. In fact for most surface models several alternative heat transfer models are defined.

Modularity and heat transfer equation independence lead to an easily alterable program.

Another type of organizing program is to use hierarchy. Developing programs is easier as the work can be divided into smaller separate tasks and more complete understanding of program functioning is possible. Making changes is easier as the scope of change is smaller. Structure of program is clearer, so no unnecessary calculations are done. The user's creativity and experience speeds up design phase and ameliorates results.

Unfortunately hierarchy also means that the way the user is supposed to be working is fixed and a truly flexible program is very difficult to make.

File organization

The information stored by the proposal phase programs can be divided into three separate types. The first type describes configuration, the second describes dimensions and the third describes operation. If configuration parameters and dimension parameters are kept separately it is easier to create a new steam generator design by scaling from the old design of an almost similar boiler. To keep operation variables separate is also advantageous, because such variables change from load to load.

During execution of these programs data files are created and continuously updated. Changing a file content occurs when the designer saves current modifications. Conscious updating by the user is a better alternative than frequent machine-made additions and updates.

With existing data files the program can be restarted if further modifications or recalculations are needed. These data files serve also as a safety copy of the work done.

Data files can also be used later to create lay-out and even manufacturing drawings when the proposal phase leads to the actual sale.

User interface

User interfaces should be menu-oriented. Each time the designer completes a subwork a list of choices is flashed to the computer screen from which he must choose the next subwork to be performed. If the program is menu-driven, then it is usually interactive i.e. the user directs the execution of the program.

Giving the user the control over the execution of the program enables immediate reactions to false operation or wrong input data. If immediate action is taken, then less recalculations have to be done. An experienced designer works quicker that way.

Visual representation of data

Visual pictures of dimensioning data and model interconnections on the screen help the designer to understand what he has done so far. Seeing what you do makes the user connection friendly and speeds up the program execution. For reference purposes hardcopies of those pictures made by plotters and printers are needed.

The types of visual information created during the proposal phase are two or three dimensional sketches of the boiler often from all of the main three axes, sketches of individual heat transfer surfaces including main numeric data at proper locations and flowsheets of air, flue gas and steam / water circulation.

Display of visual property and parameter data on the screen helps the user to determine what should be done to finish the current task. As much as possible of the data calculated should be made available for the user to see before he proceeds to the next step.

Data necessary to complete the present subtask should be viewed on the current screen. The designer should be able to view additional data through subroutines.

Output sheets are used to display calculation results. They are needed to record results obtained during the dimensioning phase. Output sheets are also included into the proposal text.

Such output sheets can be divided into heat transfer surface dimensioning sheets, fuel data sheets and technical data summary sheets.

PROCESS UNIT HEAT TRANSFER MODELS

Each offdesign program process unit is a process model of some part of the steam generator. The process units can define the heat exchange (heat transfer process unit) or mass flow changes (mass exchange process unit).

Generic process unit model (3.4.2) can be used to model heat transfer between the heat transfer surface elements and the heat source, gas or another element. Heat transfer process unit models are used to define the heat transfer coefficients, U for each type of heat transfer surface.

The heat transfer calculation procedures used in these process units were developed during this work. For each offdesign process model a PC design program was first developed to test and check the validity of the chosen approach.

REFU model

REFU represents an Ahlstrom type recovery boiler furnace where the flue gas transfers heat to the boiling water inside membrane walls. The heat transfer can be modelled with one-dimensional or constant temperature models. The flue gas temperature profile is also calculated if one-dimensional model is used.

REFU can also be used to calculate other boiler furnaces when a simple model is appropriate in predicting the outlet temperature.

There are four different heat transfer model possibilities to be used with REFU.

RACA model

RACA represents a heat transfer surface where flue gas makes a 90° turn. The heat is exchanged mainly by radiation between the flue gas and the wall surfaces.

RACA is used to model convection cages in recovery boilers. Radiation to adjacent surfaces is accounted for.

RADI model

RADI is used to model the superheaters and screen tubes in recovery boilers.

RADI represents a pendant heat transfer surface where flue gas transfers heat to water / steam flow and cooled walls. The main heat transfer surface should be inline with transversal pitch much greater than the longitudinal pitch.

The effects of radiation from the furnace and the cage are accounted for in RADI.

VEBA model

VEBA is used to model the vertical and horizontal boiler banks for single drum recovery boilers.

VEBA represents a heat transfer surface where the flue gas transfers heat to the boiling water inside finned tubes.

BOBA model

BOBA is used to model boiler banks in conventional two drum recovery boilers. The boiler banks for the Pyroflow or coal fired conventional boilers are also modelled with BOBA.

BOBA represents a tube bank heat transfer surface where the flue gas transfers heat to the boiling water and cooled walls. The main heat transfer surface consists of tubes arranged crossflow with the flue gas flow.

There are four different heat transfer model possibilities to use with BOBA.

SEKO model

SEKO is used to model vertical economizers for recovery boilers. The SEKO model can be used to model also other types of the vertical economizers.

SEKO represents a vertical panel tube heat transfer surface where the flue gas transfers heat to water flow inside the finned tubes.

There are two different heat transfer model possibilities to use with SEKO.

CHAM model

CHAM represents a Pyroflow boiler furnace. The heat is transferred from solid / gas suspension to the water cooled walls and several types of superheater surfaces. The CHAM model also includes the cyclone.

The heat transfer model in CHAM is one-dimensional. It can be used to model successfully various fuels including lignite, anthracite, brown coal, oil shale, bark, peat, bagasse, wood chips and oil.

CAGE model

CAGE is used to model convection cages in oil-fired, Pyroflow and coal-fired boilers. Radiative heat transfer between adjacent surfaces and the cage is accounted for.

CAGE represents a heat transfer case where the flue gas makes a 90° turn. The heat is exchanged mainly by radiation between the flue gas and the wall surfaces.

There can be hanger tubes extending through CAGE.

SURF model

SURF is used to model superheaters, reheaters or economizers in boiler backpasses. It is a typical steam generator surface and can also be used for many other models.

SURF represents a crossflow tube bank heat exchanger where flue gas transfers heat to water / steam flow, hanger tubes and cooled walls.

There are eight different heat transfer functions to be used with SURF representing both staggered and inline flow.

For cases with no hanger tubes or when the walls are not cooled, the heat transfer is calculated only to the existing heat transfer surfaces.

PANE model

PANE is used to model superheaters in conventional coal and oil-fired boilers.

PANE represents a heat transfer surface where flue gas transfers heat to superheat steam flow. The main heat transfer surface can be either panel or tube bank type.

AIRH model

AIRH is used to model the primary and secondary air heaters in Pyroflow or conventional pulverized coal fired boilers.

AIRH represents either a vertical or a horizontal tube heat transfer surface where the flue gas transfers heat to cold air. The heat transfer surface consists of tubes arranged as crossflow bundles with respect to either the flue gas flow or the air flow.

RIPP model

RIPP represents any type of extended surface heat transfer model.

The primary use for RIPP is modelling airheaters for recovery and Pyroflow boilers. RIPP is the main process unit for modelling the gas turbine waste heat recovery boilers.

FIXD model

FIXD is used to study the effect of different heat transfer coefficients on steam generator performance. We can also use FIXD to study the effect of increasing or decreasing the heat transfer surface.

FIXD represents any type of heat transfer surface. The user enters the heat transfer coefficient for FIXD.

Sweet water condenser model

The sweet water condenser model is used to model the attemperating heat exchanger. The saturated steam from the drum is condensed to attemperating flow by the economizer water flow.

The use of this type attemperating is typical in recovery boilers.

CALCULATION OF THERMOPHYSICAL PROPERTIES OF MIXTURE OF GASES

1 General treatment of property data

There is ample literature on the properties of individual gases see for example Sychev (87a, 87b, 87c), Reynolds (79) and Maddox (83a, 83b).

The usual procedure is to calculate first the property value for each gaseous component i, i = 1, ... m with mole fraction x_i for given temperature T and pressure p. The property value g of the gaseous mixture of m gases is expressed as a function of the calculated component properties

$$g = g(f(T,p,x_1), f(T,p,x_2), \dots, f(T,p,x_m)) \qquad (C1)$$

Usually it is possible to express the property g as polynomial function of T with a linear pressure correction for a boiler flue gas temperature range of 250 - 1500 K and a pressure range of 0.1 - 6.0 MPa.

$$g = \sum_{i=1}^{n} a_i T^{i-1} + e(p) \qquad (C2)$$

e(p) represents the linear pressure correction.

When the number of the flue gas components is high, the number of arithmetic operations is high and the property data evaluation subroutines are slow.

Typically the flue gas composition is fixed, but the pressures and the temperatures vary. For quick determination of physical properties a similar polynome for the mixture of gases can be used as is used to calculate the same property of the individual gas. For each transport property the coefficients of a polynome of temperature T and pressure p are calculated.

1 Density

The density ρ_i of a pure fluid i at low reduced pressure and high reduced temperature can be expressed as a function of temperature T and pressure p according to Schunck (83).

$$\rho_i = \frac{p}{R_i T} = \frac{1}{v_i} \qquad (C3)$$

The usual procedure for mixture of gases is to use the inverse of the molar average of the specific volumes of each gas see foe example Maddox (83a).

$$\rho = \frac{\sum_{i=1}^{m} x_i M_i \rho_i}{\sum_{i=1}^{m} x_i M_i} = \frac{\sum_{i=1}^{m} x_i M_i \rho_i}{M} \qquad (C4)$$

For a mixture of gases we can then substitute for given T and p.

$$\rho = \sum_{i=1}^{m} \frac{x_i p}{R_i TM} \qquad (C5)$$

For quick determination of density for the mixture of gases we can rearrange p and T to get

$$\rho = \frac{p}{T} \sum_{i=1}^{m} \frac{x_i}{R_i M} \qquad (C6)$$

which then can be expressed as

$$\rho = \frac{p}{R\ T} \qquad (C7)$$

2 Enthalpy

The specific enthalpy of pure fluid i at low reduced pressure and high reduced temperature can be expressed as a polynomial function of temperature T as in VDI (84).

$$h_i = \sum_{j=1}^{n} a_{ij} T^{j-1} \qquad (C8)$$

The typical procedure to calculate the enthalpy of a gaseous mixture is to sum the individual enthalpies as the weighted average

$$h = \frac{\sum_{i=1}^{m} x_i M_i h_i}{\sum_{i=1}^{m} x_i M_i} = \frac{\sum_{i=1}^{m} x_i M_i h_i}{M} \qquad (C9)$$

This can be further expressed as

$$h = \left(\sum_{i=1}^{m} x_i M_i \sum_{j=1}^{n} a_{ij} T^{j-1}\right)/M \qquad (C10)$$

We can then rearrange the expression and calculate new coefficients

$$a_j = \left(\sum_{i=1}^{m} x_i M_i a_{ij}\right)/M \qquad (C11)$$

We can then express specific enthalpy with these new coefficients as

$$h = \sum_{j=1}^{n} a_j T^{j-1} \qquad (C12)$$

3 Heat capacity

The specific heat capacity of pure fluid i at low reduced pressures and high reduced temperatures can be expressed with molar enthalpy as

$$c_{pi} = \frac{d}{dT} h_i(T) \qquad (\text{C13})$$

We can substitute the specific enthalpy function **B8** and do the derivation and obtain a similar polynomial function of temperature T, as the one for specific enthalpy

$$c_{pi} = \sum_{j=1}^{n-1} j\, a_{ij+1} T^{j-1} \qquad (\text{C14})$$

The usual procedure to calculate the specific heat capacity of gaseous mixture is to sum the individual heat capacities as the weighted average

$$c_p = \left(\sum_{i=1}^{m} x_i M_i c_{pi} \right) / M \qquad (\text{C15})$$

This expression can be simplified using new coefficients

$$\mathbf{b}_j = \left(\sum_{i=1}^{m} x_i M_i\, j\, a_{ij+1} \right) / M \qquad (\text{C16})$$

We can then express specific heat capacity with these coefficients as

$$c_p = \sum_{j=1}^{n-1} \mathbf{b}_j T^{j-1} \qquad (\text{C17})$$

5 Thermal conductivity

The thermal conductivity of pure fluid i at low reduced pressure and high reduced temperature can be expressed as a polynomial function of temperature T with pressure correction as expressed by Stelzer (84)

$$\lambda_i = \sum_{j=1}^{n} d_{ij} T^{j-1} + d_{in+1} p \qquad (C18)$$

The usual procedure to calculate the thermal conductivity of a gaseous mixture is to sum the individual conductivities as the weighted average of molar weights raised to one third

$$\lambda = \frac{\sum_{i=1}^{m} x_i \lambda_i M_i^{1/3}}{\sum_{i=1}^{m} x_i M_i^{1/3}} \qquad (C19)$$

This can be further expressed as

$$\lambda = \frac{\sum_{i=1}^{m} x_i \left(\sum_{j=1}^{n} d_{ij} T^{j-1} + d_{in+1} p \right) M_i^{1/3}}{\sum_{i=1}^{m} x_i M_i^{1/3}} \qquad (C20)$$

We can then calculate new coefficients as we did with enthalpies

$$d_j = \frac{\sum_{i=1}^{m} x_i d_{ij} M_i^{1/3}}{\sum_{i=1}^{m} x_i M_i^{1/3}} \qquad (C21)$$

and express conductivity with these coefficients as

$$\lambda = \sum_{j=1}^{n} d_j T^{j-1} + d_{n+1} p \qquad (C22)$$

6 Viscocity

The dynamic viscosity of pure fluid i at low reduced pressure and high reduced temperature can be expressed as a polynomial function of temperature T with pressure correction as

$$\eta_i = \sum_{j=1}^{n} c_{ij} T^{j-1} + c_{in+} p \qquad (C23)$$

The usual procedure to calculate the viscosity of the gaseous mixture is to sum the individual conductivities as the weighted average of square roots of molar weights see for example Talonpoika (89).

$$\eta = \frac{\sum_{i=1}^{m} x_i \eta_i M_i^{1/2}}{\sum_{i=1}^{m} x_i M_i^{1/2}} \qquad (C24)$$

This can be further expressed as

$$\eta = \frac{\sum_{i=1}^{m} x_i \left(\sum_{j=1}^{n} c_{ij} T^{j-1} + c_{in+1} p \right) M_i^{1/2}}{\sum_{i=1}^{m} x_i M_i^{1/2}} \qquad (C25)$$

We can then calculate new coefficients as we did with enthalpies

$$c_j = \frac{\sum_{i=1}^{m} x_i c_{ij} M_i^{1/2}}{\sum_{i=1}^{m} x_i M_i^{1/2}} \qquad (C26)$$

and express the viscosity of the gaseous mixture with these coefficients as

$$\eta = \sum_{j=1}^{n} c_j T^{j-1} + c_{n+1} p \qquad (C27)$$

REFERENCES FOR APPENDIX C

Maddox, R., N., 1983a, Properties of mixtures of fluids. in Heat Exchanger Design Handbook, Part 5. Physical properties. Hemisphere Publishing Corporation. pp. 5.2.1-1 - 5.2.5-4.

Maddox, R., N., 1983b, Properties of superheated gases. in Heat Exchanger Design Handbook, Part 5. Physical properties. Hemisphere Publishing Corporation. pp. 5.5.2-1 - 5.5.2-11.

Reynolds, William, C., 1979, Thermodynamic properties in SI. Department of Mechanical Engineering, Stanford University. 173 p.

Schunck, M., 1983, Properties of pure fluids. in Heat Exchanger Design Handbook, Part 5. Physical properties. Hemisphere Publishing Corporation. pp. 5.1.1-1 - 5.1.5-5.

Stelzer, Friedrich, J., 1984, Physical property algorithms. Karl Thiemig Ag, Munich. 502 p.

Sychev, V. V., et. al., 1987a, Thermodynamic Properties of Nitrogen. National Standard reference Data Service of the USSR, A Series of Property Tables. Hemisphere Publishing Corporation. 341 p.

Sychev, V. V., et. al., 1987b, Thermodynamic Properties of Methane. National Standard reference Data Service of the USSR, A Series of Property Tables. Hemisphere Publishing Corporation. 341 p.

Sychev, V. V., et. al., 1987c, Thermodynamic Properties of Oxygen. National Standard reference Data Service of the USSR, A Series of Property Tables. Hemisphere Publishing Corporation. 307 p.

Talonpoika, Timo, 1989 Calculation of transport and radiation properties of flue gases. (in Finnish) Lappeenranta University of Technology, Department of Energy Technology, EN C-48. 45 p.

VDI-Wärmeatlas, Berechnungsblätter für den Wärmeübergang. 1984, 4. auflage, Verein Deutcher Ingenieure, VDI-Verlag GmbH, Düsseldorf. pp. Da 1 - Da 36.

3000 TDS/D RECOVERY BOILER HEAT AND MASS BALANCE DATA

TABLE 1, Main design data for the example boiler

I	LOAD, STEAM AND FEEDWATER		
	Design firing rate, tds/d		3000
	Main steam pressure, bar		90
	Main steam temperature, °C		490
	Feedwater pressure, bar		120
	Feedwater temperature, °C		120
II	AIR VALUES		
	Primary air percentage, %		35
	Primary air temperature, °C		120
	Secondary air percentage, %		50
	Secondary air temperature, °C		120
	Tertiary air percentage, %		15
	Tertiary air temperature, °C		50
	Ambient air temperature, °C		30
	Air humidity, g/kg dry air		13
III	FLUE GAS VALUES		
	O_2 in dry fluegas, vol-%		3
	SO_2 content in dry fluegas, ppm		20
	Fluegas temperature after economizer, °C		150
IV	BLACK LIQUOR VALUES		
	Dry liquor analysis, wt-%	C	37.6
		Na	19.9
		S	4.8
		O_2	32.9
		H_2	3.5
		K_2	1.0
		Others	0.2
	Higher heating value of dry solids, MJ/kgds		15
	Dry solids content before mixing tank, wt-%		60-90
	Liquor temperature before mixing tank, °C		115
	Make-up salt flow, g/kgds		0
V	MISCELLANEOUS VALUES		
	Chemical loss to stack, g/kgds		0.5
	Reduction, %		97
	Sootblowing steam flow, %		1.5
	Continuous blowdown, %		0.2
	Heat balance datum temperature, °C		0

TABLE 2, Material balance for the example boiler

Liquor dry solids, %	60	65	70	75	80	85	90
Air flow, m^3n/kgds	3.981	3.981	3.981	3.981	3.981	3.981	3.981
Flue gas flow, m^3n(dry)/kgds	3.825	3.825	3.825	3.825	3.825	3.825	3.825
Air flow, kg/s	177.4	177.4	177.4	177.4	177.4	177.4	177.4
Flue gas flow, kg/s	221.0	216.9	212.8	209.5	206.7	204.2	201.9
Flue gas volumetric composition	(sootblowing steam included)						
CO_2, %	12.16	12.53	12.88	13.20	13.49	13.75	14.0
H_2O, %	26.35	24.04	21.96	20.06	18.31	16.71	15.23
SO_2, %	0.00	0.00	0.00	0.00	0.00	0.00	0.00
N_2, %	59.28	61.14	62.81	64.35	65.75	67 4	68.24
O_2, %	2.21	2.29	2.34	2.40	2.45	2.50	2.54
O_2 dry gas, %	3	3	3	3	3	3	3
CO_2 dry gas, %	16.5	16.5	16.5	16.5	16.5	16.5	16.5
SO_2 dry gas (volume), ppm	19.6	19.6	19.6	19.6	19.6	19.6	19.6
SO_2 dry gas (mole), ppm	20.0	20.0	20.0	20.0	20.0	20.0	20.0
SO_2 dry gas, mg/m^3n	57.3	57.3	57.3	57.3	57.3	57.3	57.3
Density of gas, kg/m^3n	1.224	1.238	1.250	1.261	1.271	1.280	1.289

TABLE 3, Energy balance for the example boiler

Liquor dry solids, %	60	65	70	75	80	85	90
HHV of black liquor, kJ/kgds	15000	15000	15000	15000	15000	15000	15000
Heat in black liquor, kJ/kgds	543.8	502.0	466.2	435.0	407.9	383.9	362.6
Sensible and preheat							
Heat in air, kJ/kgds	566.0	566.0	566.0	566.0	566.0	566.0	566.0
Total heat in fuel, kJ/kgds	16110	16068	16032	16001	15974	15950	15929
Chemical losses, kJ/kgds	-1434	-1434	-1434	-1434	-1434	-1434	-1434
Heat in smelt, kJ/kgds	-613	-613	-613	-613	-613	-613	-613
Heat in furnace, kJ/kgds	14063	14022	13986	13955	13927	13903	13882
Heat in H_2O, kJ/kgds	-2456	-2135	-1861	-1622	-1414	-1230	-1066
Heat in dry flue gases, kJ/kgds	-1112	-1076	-1045	-1018	-995	-974	-956
Miscellaneous losses, kJ/kgds	-290	-290	-290	-290	-290	-290	-290
Unburned combustible							
Radiation and convection							
Other losses and margin							
Heat available to steam, kJ/kgds	10205	10520	10790	11024	11229	11409	11570
Sootblowing steam, kJ/kgds	-119	-119	-119	-119	-119	-119	-119
Continuous blowdown, kJ/kgds	-6.4	-6.6	-6.8	-6.9	-7.0	-7.2	-7.3
Total heat to steam, kJ/kgds	10080	10394	10664	10898	11102	11283	11443
Steam flow, kg/kgds	3.537	3.648	3.742	3.824	3.896	3.959	4.016
Efficiency, %	62.6	64.7	66.5	68.1	69.5	70.7	71.8