



Cycle-Tempo



Examples

Release

5

*A program for thermodynamic modeling
and optimization of energy conversion systems*



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Introduction

This part of the manual contains a number of examples of energy conversion systems. For each example a process scheme is given, the input is described and the results are discussed briefly.

The user may open the files containing the examples to see the input specifications and the calculations results. These examples are installed in the subdirectory EXAMPLES of the directory containing the Cycle-Tempo program files.

The user subroutines used in some examples are supplied with the program too, and are stored in the EXAMPLES directory mentioned above. To run the examples with user subroutines, see the procedure outlined in part “User subroutines” of the manual.

The examples vary from simple to advanced systems. The table shown on the next page lists the examples, and the files containing them.



Example	Variants	File
Simple water/stream cycle	▪ Base calculation	Ex1.GUI
	▪ Base calculation with user subroutine	Ex1A.GUI
	▪ Optimization calculation	Ex1B.GUI
Conventional water/steam cycle	▪ Design calculation	Ex2.GUI
	▪ Off-design calculation (80%)	Ex2A.GUI
	▪ Off-design calculation (80%), condenser model used differently	Ex2B.GUI
Gas turbine model		Ex3.GUI
STAG-unit with back-pressure steam turbine		Ex4.GUI
STAG-unit Free University Amsterdam	▪ Design calculation	Ex5.GUI
	▪ Off-design calculation with maximum electricity production	Ex5A.GUI
	▪ Off-design calculation with maximum heat production	Ex5B.GUI
	▪ Off-design calculation with specified heat production	Ex5C.GUI
District heating cycle	▪ Design calculation with maximum electricity production	Ex6.GUI
	▪ Design calculation with maximum heat production	Ex6A.GUI
	▪ Off-design calculation with maximum electricity production	Ex6B.GUI
	▪ Off-design calculation with maximum heat production	Ex6C.GUI
	▪ Off-design calculation with average heat production	Ex6D.GUI

Chapter 1

A simple water/steam cycle

1.1 Introduction

To introduce the Cycle-Tempo program, using a simple water/steam cycle an example of program input and output is given. The same example is used in the 'Technical Notes' to explain the basic principles of the program. A diagram of the water/steam cycle is shown in Figure 1-1. Only the most important results of the output of the calculation are shown.

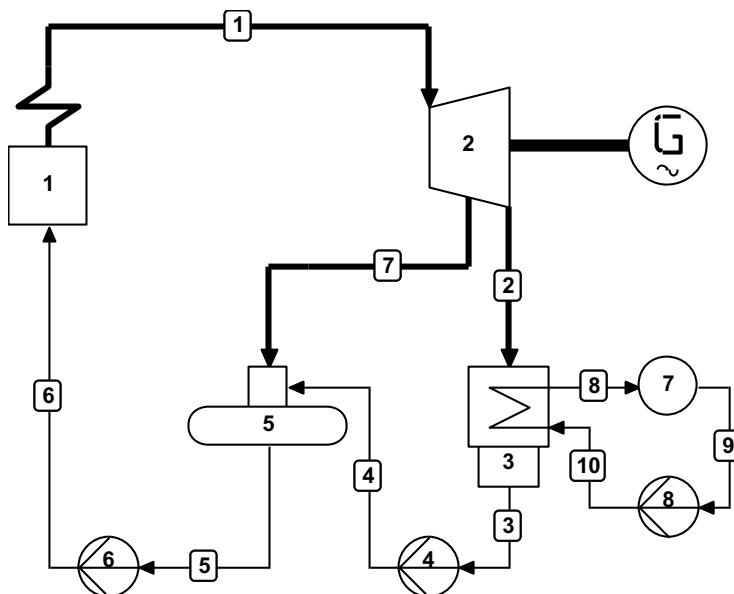
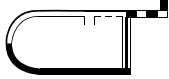


Figure 1-1: a simple water/steam cycle



1.2 Base calculation

1.2.1 System description

In boiler 1 superheated steam is generated. This steam expands in steam turbine 2, which drives a generator. The steam is then condensed into water in condenser 3. The cooling water circuit is modelled by sink 7 and cooling water pump 8. The condensate from the condenser is pumped by condensate pump 4 to the deaerator 5, where it is vented from the turbine using extraction steam. Then the feedwater is pumped by feedwater pump 6 to the boiler.

1.2.2 Starting points for the calculation

In the system the following starting points are specified:

Apparatuses

Boiler 1:	outlet pressure	70 bars
	outlet temperature	400 °C
Turbine 2:	η_i (ETHAI)	0.8
Condensor 3:	outlet pressure condensate	0.027 bars
	pressure drop steam side	0 bar
	pressure drop cooling water	1 bar
	temperature rise cooling water	5 K
Pump 4:	η_i (ETHAI)	0.75
Deaerator 5:	outlet pressure	3 bars
	pressure drop	0 bar
Pump 6:	outlet pressure	80 bars
	η_i (ETHAI)	0.75
Source/sink 7:	outlet temperature cooling water	15 °C
	outlet pressure cooling water	1 bar
Pump 8:	outlet pressure	4 bars
	η_i (ETHAI)	0.75

Pipes

pipe 1: pressure drop	3 bars
pipe 4: pressure drop	2 bars

For the pipes these data are specified as extra conditions.

Pipe 7 is connected to the top of the deaerator, as the pipe contains extraction steam for the deaerator (see also the input description of the deaerator in part “Reference Guide” of the manual).

As production function it is specified that the generator must supply 100 MW_e.

For the efficiency of the generator $\eta_{\text{gen.}} = 0.97$ (ETAGEN) is specified.

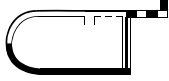
1.2.3 Results of the calculation

In the table “System efficiencies” a summary is given of the system efficiency, the primary energy absorbed by the boiler, the power produced by the generator, the electricity consumption of the pumps and the net supplied power. It should be mentioned that the pump efficiencies shown at “Aux. power consumption” are default values assigned by the program (for input data for pumps, see part “Reference Guide” of the manual).

The table “Energy balance” shows the amounts of energy exchanged with the environment for apparatuses and pipes. The power supplied by the steam in the turbine is presented here (POWER divided by ETAGEN).

The table “Data for all pipes” contains a.o. mass flows, pressures, temperatures and enthalpies of all pipes in the system. Many starting points can be seen here, such as the temperature of the steam from the boiler (inlet of pipe 1), and the outlet pressure of pump 6 (inlet of pipe 6). The cooling water pressure drop in the condenser is the difference between the pressure at the outlet of pipe 10 and the pressure at the inlet of pipe 8.

Losses in pipes are given in table “Losses in pipes”. The pressure loss defined for pipe 1 (specified as an extra condition) can be found here also.



The energy and exergy flows in the pipes are presented in table “Energy and exergy flows”. Each energy or exergy flow is split-up into its thermo-mechanical and chemical parts. Exergy losses and exergy efficiencies for apparatuses in the system can be found in table “Exergy values in the system”.

Calculated and specified efficiencies of devices connected to a shaft (turbines, pumps, compressors, motors and generators) are given in the tables “Rotating equipment” and “Motors and generators”. Data for heat exchangers can be found in table “Heat exchanging equipment”. In this case, only a condenser is present.

All input and output data are also written in the output file, which may be viewed via the “Text Output” option. At the end of the output file design data for turbines, compressors and heat exchangers are presented, which are also written in the file EX1.PLD.

1.3 Calculation with user subroutine

1.3.1 Introduction

In this example for turbine 2 in the user subroutine APSUB a new value for the isentropic efficiency is calculated, which is dependent on the mass flow. For this the number of the turbine is specified via Calculation | User Subroutines. The user-defined subroutine APSUB, which is linked to the user executable, will be called for this apparatus (see also Figure 1-2).

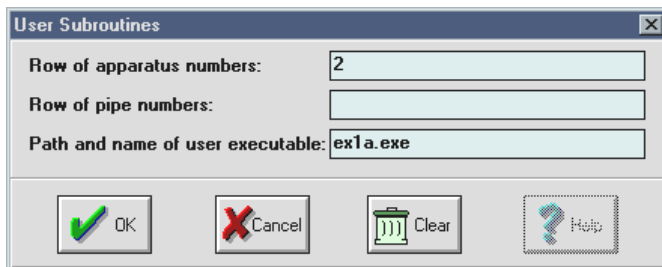
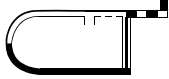


Figure 1-2: input dialog for data of user subroutines

The input remains the same. The example is stored in EX1A.GUI. The user-defined version of subroutine APSUB, of which the content is shown in the next paragraph, is stored in file EX1_APSU.FOR.



1.3.2 Source text of the user subroutine

```
*-----*
*   User subroutine APSUB, meant for example exla   *
*   *                                               *
*   For the turbine with NO=2 the mass flow dependent isentropic *
*   efficiency is calculated.                       *
*-----*

      SUBROUTINE APSUB( NLIN, ILIN, IAP, ITYP, ITM, DATA, NP, IPIPE, FM, H,
*                   P, T, X, W, C, RESULT, COMP, CONC, AVERG, IDX, NCOMP,
*                   APNAME, NUMGEN, RDATGE)

      IMPLICIT DOUBLE PRECISION (A-H, O-Z)

      DOUBLE PRECISION DATA(50), FM(NP), H(NP, 3), P(NP, 3), T(NP, 2),
*                   X(NP, 2), W(NP), C(NP), CONC(NLIN+7, NCOMP),
*                   AVERG(NLIN+7), RDATGE(NUMGEN, 6)
      INTEGER        IPIPE(NP), ILIN(NLIN), IDX(NCOMP)
      CHARACTER      COMP(NCOMP)*8, APNAME*12

      PARAMETER ( A = 0.0001 )
      SAVE ETHAIB

      IF (IAP.EQ.2) THEN

*         Save basic efficiency
          IF (ITM.EQ.0) ETHAIB = DATA(7)

*         Write basic efficiency to standard message file (UNIT = 13)
          WRITE (13,1) DATA(7)

*         Calculate isentropic efficiency depending on massflow
          FLOW = FM(IPIPE(1))
          RESULT = ETHAIB + A*FLOW

*         Write the results of each iteration to standard message file
          WRITE (13,2) FLOW, RESULT

      ENDIF

1  FORMAT (' Ethai(old) = ', F8.5)
2  FORMAT (' Massflow = ', F8.3, '      Ethai = ', F8.5)
      END
```

1.4 Optimization

1.4.1 Introduction

This example concerns the same system, but extended with the use of the optimization procedure of Cycle-Tempo. This example is stored in file EX1B.GUI. The bleeder pressure of turbine 2 (specified by POUT of apparatus 5) will be adjusted by the procedure in such a way that the system efficiency will be maximized. When the optimum value of the bleeder pressure is found, a standard Cycle-Tempo calculation is performed with this value.

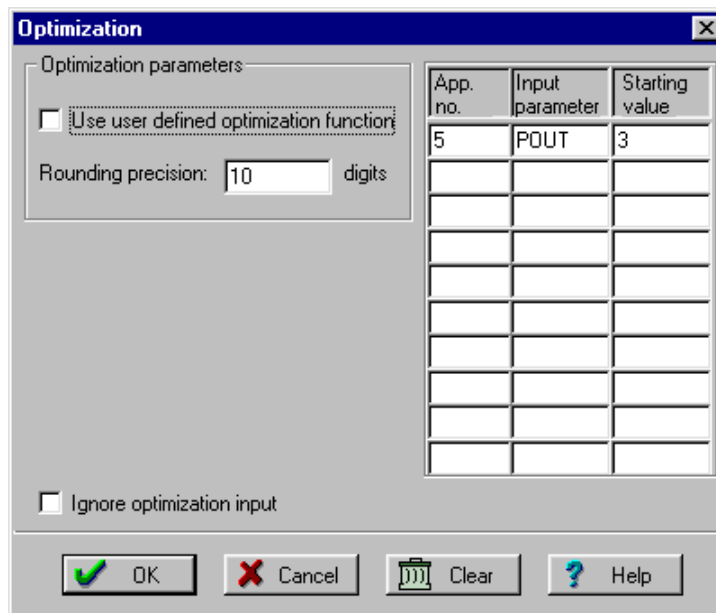
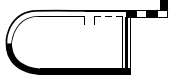


Figure 1-3: input dialog for optimization data

1.4.2 Changes in the input

To activate the optimization procedure a correct set of optimization data should be specified via Calculation | Optimization. The following data are given (see also Figure 1-3):

1. the apparatus number of which the variable has to be optimized: App. No. : 5;
2. the name of the variable to be optimized: POUT;



3. starting value from which the optimization procedure starts: Starting value = 3 (bars);
4. the number of digits to which the system efficiency is to be rounded in the optimization procedure: Rounding precision = 10.

1.4.3 Results of the calculation

In the output file (viewed via View | Text Output) the progress of the optimization procedure is presented. After 19 optimization iterations the value of the bleeder pressure is found at which the system efficiency reaches a maximum. The corresponding values of the bleeder pressure and the system efficiency are given:

1. bleeder pressure: VARIABLE(1): 3.8534375 (bars);
2. system efficiency: OPTIM. FUNCTIONVALUE: 0.3062971649

The optimization progress shows that the system efficiency has risen from 0.3062137331 at a bleeder pressure of 3 bars, to 0.3062971649 at a bleeder pressure of 3.8534375 bars. The gain in efficiency is therefore 0.0000834.

With the new value of the bleeder pressure the normal calculation is performed. The results are stored in the tables as usual.

Chapter 2

Conventional water/steam cycle

2.1 Introduction

This example relates to the water/steam cycle of a large conventional electricity power station with a design power of 600 MW_e.

For this water/steam cycle a scheme is defined (see Figure 2-1) for both design and off-design cases, although each case is stored in a separate file. It will be explained how a design calculation can be transformed into an off-design calculation, using design data generated by the design calculation.

Calculation results are set out for the design calculation and calculations with 80% part load. The calculations demonstrate in particular a few options of the models for steam turbines, flashed heaters and condensers.

2.2 System description

In the water/steam cycle shown in Figure 2-1 steam of 180 bars and 530 °C expands in the high pressure turbine 2 to a pressure of around 38 bars. The steam is reheated (4) to 530 °C and expanded further in two sections of the intermediate-pressure turbine and in six sections of the low-pressure turbine. The number of parallel sections is not shown, but is specified in the TUCODE for apparatus 5. When using the General Electric procedure in Cycle-Tempo the intermediate- and low-pressure turbines are represented by one turbine (5).

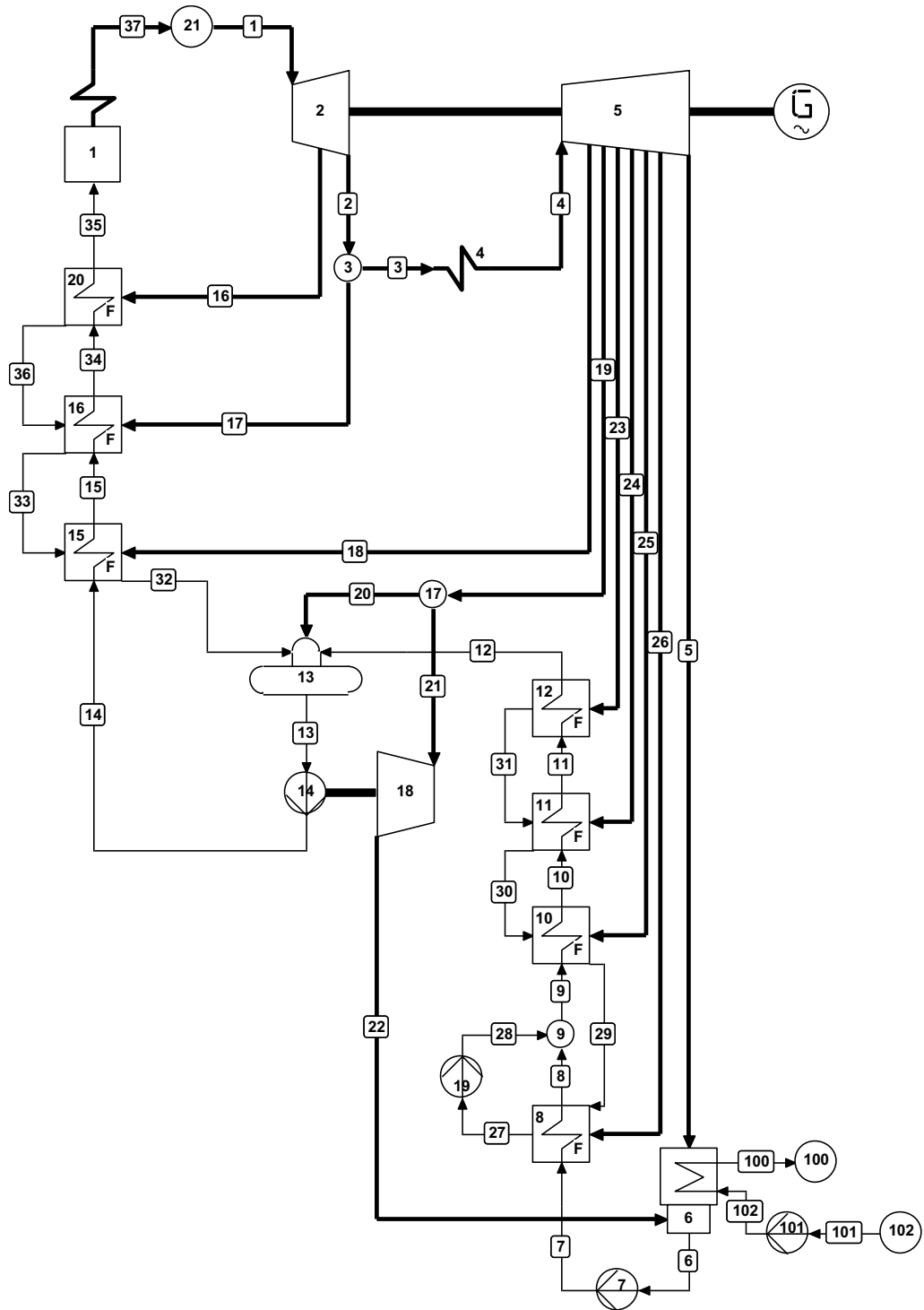
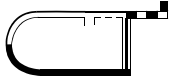


Figure 2-1: conventional water/steam cycle

The steam then condenses in condenser 6 at a pressure of 0.027 bar. The condensate is preheated in eight preheaters: five low-pressure and three high pressure preheaters. The last low-pressure preheater (13) is a deaerator, in fact a mixing preheater. The other low-pressure preheaters (8, 10, 11 and 15) are surface preheaters. The surface preheaters, with the exception of the first preheater, are equipped with a subcooler. The condensate of these four preheaters is collected, via any preceding preheaters, in the first preheater (with the lowest extraction pressure). From here the auxiliary condensate pump 19/8 pumps this condensate to the main condensate line. The three high-pressure preheaters (15, 16 and 20) are surface preheaters, each equipped with a desuperheater and a subcooler. The condensate from this is fed into the deaerator, via any preceding preheaters, with the main condensate stream. Feed pump 14 pumps the feedwater of the deaerator to boiler 1. The feed pump is driven by auxiliary turbine 18. This turbine, like the deaerator, is supplied with extraction steam from the outlet of the intermediate-pressure turbine. In the auxiliary turbine the steam is expanded to a pressure of 0.027 bar. The main and auxiliary condensate pumps (7 and 14) are driven by electro motors.

Apparatus 21 is a dummy. This apparatus is added in order to show separately the pressure drop of the live steam (4% of inlet pressure HP-turbine) (pipe 37) and the throttling losses of the control valves (3% of inlet pressure HP-turbine). This apparatus does in fact form the inlet of turbine 2. The throttling losses of 3% are specified as pressure drop in pipe 1.

2.3 Starting points for the calculation

The most important starting points are:

Boiler 1

pressure drop: 30% PIN HP
(specified as extra condition for pipe 35, see below)

Reheater 4

pressure drop: 10% PIN IP
(specified as extra condition for pipe 4, see below)

Main condensate pump 7

isentropic efficiency: ETHAI = 0.78
 efficiency drive: ETHAE = 0.88

Auxiliary condensate pump 19

isentropic efficiency: ETHAI = 0.78
 efficiency drive: ETHAE = 0.86

Low-pressure preheater 8

temperature difference at high-temperature side: DELTH = 3 K
 condensate at saturation: SATCOD = 0
 pressure drop primary side: DELP1 = 0.5 bar

Low-pressure preheaters 10, 11 and 12

temperature difference at high-temperature side: DELTH = 3 K
 temperature difference at low-temperature side: DELTL = 5 K
 pressure drop primary side: DELP1 = 0.5 bar

Deaerator 13

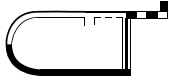
steam pressure: PIN = 5.192 bar

High-pressure preheaters 15, 16 and 20

temperature difference at high-temperature side: DELTH = 0 K
 temperature difference at low-temperature side: DELTL = 7 K
 pressure drop primary side: DELP1 = 1 bar

Steam pressures in the preheaters

preheater 8: PIN2 = 0.082 bar
 preheater 10: PIN2 = 0.236 bar
 preheater 11: PIN2 = 0.676 bar
 preheater 12: PIN2 = 1.895 bar
 preheater 15: PIN2 = 13.919 bar
 preheater 20: PIN2 = 71.472 bar



Generator

design power (production function):	POWER = 600 MWe
capacity:	GENMVA = 750 MVA
power factor:	COSPFI = 0.85

Pipes

Pressure losses

live steam pipe 37: (extra condition:	4% PIN HP DELPR = 0.0384 (= 0.04/1.04))
pipe 1: (extra condition:	3% DELPR = 0.03)
extraction steam pipe 17: (extra condition:	3% DELPR = 0.03)
extraction steam pipes 16, 18, 19, 23-26: (extra condition:	6% DELPR = 0.06)
boiler feed water pipe 35: (extra condition:	30% PIN HP DELPR = 0.2239 (= 0.3/1.34))
reheated steam pipe 4: (extra condition:	10% PIN HP DELPR = 0.0909 (= 0.1/1.01))

As a reference for the calculation of exergies the environment according to the definition of Baehr at 15°C is used.

2.4 Off-design calculations

2.4.1 Changes in the input

For an off-design calculation the input data, as set up for the design calculation, must be altered in a number of points. These alterations may, for example, relate to a more detailed specification of the design of apparatuses or of the control of the system.

In this example the input data are altered for an off-design calculation representing a 80% part load situation.

For the design calculation the assumption has been made of a power to be produced by the steam turbine of 600 MWe. The specification for this is:

- Production functions for turbines 2 and 5: POWER = 600

In the case of an off-design calculation for 80% part load one must then specify:

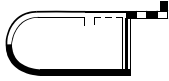
- Production functions for turbines 2 and 5: POWER = 480

For the conversion of the design case into the required off-design case the changes described below must also be made.

For the turbine

For the turbine the design data as generated by the design calculation, must be added to the input. These data can be imported from the appropriate PLD-file using the “Paste” button which will appear after pressing the “Off-design input data” button in the input window of the apparatus. The following design data will be copied in the different fields:

- the volume flow at the inlet of the relevant turbine (DESVOL);
- the numbers of the pipes connected to the turbine, first of all the outlet pipes from low to high pressure, then the inlet pipe (LBLEED);



- the pressures in these pipes in the design situation, specified in the same order, hence from low to high pressure (PBLEED);
- the specific volumes in these pipes in the design situation, specified in the same order (VBLEED);
- the mass flows in these pipes in the design situation, specified in the same order (MBLEED).

If GDCODE = 1 (no governing stage present), which is the case for turbines 2 and 5, the turbine-inlet pressure can be calculated with Traupel's formula. The extraction pressures of the turbine can also be calculated with Traupel's formula. A condition for this is that the pressure at the outlet of the turbine is known. The outlet pressure of turbine 5 is established by the condenser pressure, see the data for the condenser. The outlet pressure of turbine 2 is established by the inlet pressure of turbine 5, see above.

This results in the following changes for the input:

- As the turbine inlet pressure of turbine 2 is calculated, for apparatus 21 the pressure PIN = 180 may no longer be specified. This pressure would otherwise be transmitted to the turbine inlet.
- As the turbine inlet pressure of turbine 5 is calculated, PIN = 34.02 may no longer be specified for the turbine.
- As the extraction pressure of turbine 2 is calculated, PIN2 = 71.472 of flashed heater 20 may no longer be specified. This pressure would otherwise, taking into account the pressure drop in pipe 16, be transmitted to the extraction of turbine 2.
- As the extraction pressures of turbine 5 are calculated, PIN2 = 13.919 of flashed heater 15 may no longer be specified. This pressure would otherwise, taking into account the pressure drop in pipe 18, be transmitted to the extraction of turbine 5.
- For the same reason PIN2 may no longer be specified for the preheaters 8, 10, 11 and 12. The same goes for PIN specified for deaerator 13.

For the condenser

The design data for the condenser can be imported as off-design input data in the input window in the same way as described for the turbine. For a condenser for which geometry data have been specified, this will be:

- the size of the heat exchanging area (m^2)

In the design calculation the cooling water temperatures at the inlet and outlet of the condenser are specified. The quantity of cooling water necessary to absorb the heat of condensation is then to be calculated from the energy balance. If this option is used, the Energy Equation CODE (EEQCOD) must be set to 1.

In the off-design calculation the heating of the cooling water is kept the same as that of the design calculation. The energy balance is again used to calculate the quantity of cooling water necessary to absorb the now smaller quantity of heat of condensation. In the off-design calculation in addition to the energy balance the heat transfer equation is now available for calculating an unknown temperature. This equation is used to calculate the condensation temperature in the condenser. See also part “Reference Guide” of the manual.

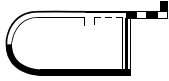
In the input the following then changes:

- the condenser pressure $POUT2 = 0.027$ may no longer be specified, as this pressure is directly coupled with the condensation temperature. This temperature is determined in the off-design calculation using the heat transfer equation.

For the preheaters

For the preheaters in the system flashed heaters (type 5) are used. For a heat exchanger apparatus of type 5 the $U \cdot A$ value can usually not be defined because of the phase change of the extraction steam. For this reason for a heat exchanger of apparatus type 5 the $U \cdot A$ value is not stored in the file containing design data. For the off-design calculation of a flashed heater only the value of either $DSMAS1$ or $DSMAS2$ must be specified. The value of $DSMAS1$ as calculated in the design calculation can be imported via the “Paste” button as described above.

In the input data for the preheaters no further changes need to be made. For apparatus type 5 with an off-design calculation the inlet and outlet temperature differences, which are



specified in the input (DELTH and DELTL), are altered. This alteration is a function of the mass flow ratio off-design, as indicated in Figure 2-3 in paragraph 2.6 of part “Reference Guide” of the manual.

2.4.2 Results of the 80% part load calculation

For the calculated part load situation the results are set out below. The quantity of boiler feedwater/steam, circulating in the system, falls as the demand for power drops. The consequences of this on the various apparatuses are set out below.

Changes for the turbine

For the turbines the extraction pressures and inlet pressure are calculated using Traupel’s formula. Table 2-1 indicates how the pressures and mass flows change in the 80% part load situation for the high-pressure turbine 2.

Table 2-1: Design/off-design calculation high-pressure turbine 2

Turbine 2	Pipe no	Mass flow [kg/s]		Pressure [bar]	
		Design	Off-design	Design	Off-design
inlet	37	519.1	401.0	180.0	141.7
extraction	16	51.6	35.4	76.0	60.5
outlet	2	467.5	365.6	37.4	29.7

For the intermediate- and low-pressure turbine 5 the data are set out in Table 2-2.

Table 2-2: Design/off-design calculation intermediate- and low-pressure turbine 5

Turbine 5	Pipe no.	Mass flow [kg/s]		Pressure [bar]	
		Design	Off-design	Design	Off-design
inlet	4	418.6	331.0	34.02	27.00
extraction	18	25.8	19.5	14.81	11.81
	19	46.5	31.8	5.523	4.480
	23	18.6	14.1	2.016	1.641
	24	15.5	11.9	0.719	0.587
	25	12.7	9.7	0.251	0.205
	26	9.6	6.7	0.087	0.073
outlet	5	289.9	237.4	0.027	0.026

The extraction pressures fall because of the reduced mass flow through the turbines.

Changes for the condenser

The quantity of heat which must be transferred in the condenser is proportionate to the quantity of steam passing through the condenser. The heat of condensation is virtually independent of the steam pressure/steam temperature. The heating of the cooling water is kept the same as that in the design situation. From the energy balance it then follows that less cooling water, primary medium, is necessary to absorb the heat of condensation. If the quantity of cooling water falls, the turbulence in the flow is reduced, as a result of which the overall heat transfer coefficient of the condenser deteriorates. This fall is however less than the fall in the quantity of heat to be transferred. From the heat transfer equation it then also follows that the log mean temperature difference must fall. The off-design value of T_{cond} is then also lower than the design value. Table 2-3 sets out the results for the design and off-design calculation.

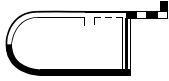


Table 2-3: Design/off-design calculation condenser

Condenser 6	Pipe no.	Mass flow [kg/s]		Temperature [°C]	
		Design	Off-design	Design	Off-design
primary in:	102	22787	18409	12.00	12.00
	out: 100	22787	18409	19.36	19.36
secondary in:	5	289.90	237.36	22.36	21.75
	22	24.23	15.02	22.36	21.75
	out: 6	314.13	252.39	22.36	21.75

Changes for the preheaters

In the preheaters the inlet and outlet temperature differences specified are altered. The off-design values of these differences are dependent on the design values of ΔT_H and ΔT_L specified. In addition the difference is a function of the ratio between the off-design and the design value of the mass flow. In Table 2-4 the data and results for the preheaters are presented. The temperature difference ΔT_L for apparatus 8 is not specified (see input data) and is thus also not altered.

Table 2-4: Design/off-design calculation preheaters (d = design, od = off-design)

Apparatus-number	Pipe, primary in/out	$\Phi_{m,prim,d}$ [kg/s]	$\Phi_{m,prim,od}$ [kg/s]	$\Delta T_{H,d}$ [K]	$\Delta T_{H,od}$ [K]	$\Delta T_{L,d}$ [K]	$\Delta T_{L,od}$ [K]
8	7/8	314.18	252.39	3	2.14	-	-
10	9/10	370.58	294.70	3	2.11	5	3.76
11	10/11	370.58	294.70	3	2.11	5	3.76
12	11/12	370.58	294.70	3	2.11	5	3.76
15	14/15	519.16	401.04	0	-0.51	7	5.10
16	15/34	519.16	401.04	0	-0.51	7	5.10
20	34/35	519.16	401.04	0	-0.51	7	5.10

2.4.3 Other use of the condenser model

In the example shown above the heating of the cooling water is established by specifying the outlet temperature of the cooling water. The energy equation is then used to calculate the mass flow of cooling water (EEQCOD = 1).

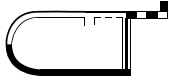
It is however also possible to prescribe the mass flow of cooling water, where then the outlet temperature of the cooling water is calculated (EEQCOD = 2). This can be used when as the load falls the cooling water mass flow remains unchanged. For source 102 the mass flow is then specified, as calculated for the design situation (DELM = -22787).

In this case too the heat transfer equation is used to calculate the condensation temperature/ condensation pressure.

The results of this calculation (see Table 2-5) differ in a number of points from those of the preceding calculation. The quantity of heat of condensation falls in proportion to the mass flow of steam. From the energy balance it then follows that as the cooling water mass flow does not change, the outlet temperature of the cooling water falls with respect to the design situation. The overall heat transfer coefficient remains more or less unchanged. From the heat transfer equation it then follows that the log mean temperature difference falls. This means that the condensation temperature falls more than in the preceding calculation (see Table 2-5 below).

Table 2-5: Results other use condenser model

Condenser 6	Pipe no.	Mass flow [kg/s]		Temperature [°C]	
		Design	Off-design	Design	Off-design
primary	in: 102	22787	22787	12.00	12.00
	out: 100	22787	22787	19.36	17.94
secondary	in: 5	289.90	236.85	22.36	20.38
	22	24.23	14.83	22.36	20.38
	out: 6	314.13	251.68	22.36	20.38



The lower condenser pressure also affects the mass flow through the turbine and the extraction pressures. As a result this produces different temperatures and pressures virtually throughout the system.

The higher fall of the condenser temperature is also shown in Figure 2-2.

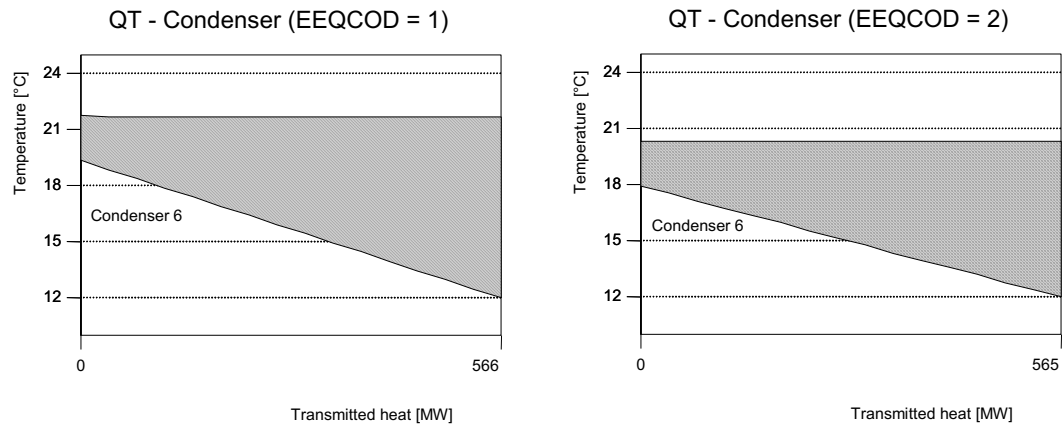


Figure 2-2: Q, T – diagram condenser

Chapter 3

Gas turbine model

3.1 Introduction

To introduce cycles with gas mixtures into the program Cycle-Tempo a gas turbine is modelled as a third example. After this chapter a simple example of a STAG unit and a more advanced model of a district heating cycle with part load calculations are shown. The diagram of the of the gas turbine is shown in Figure 3-1.

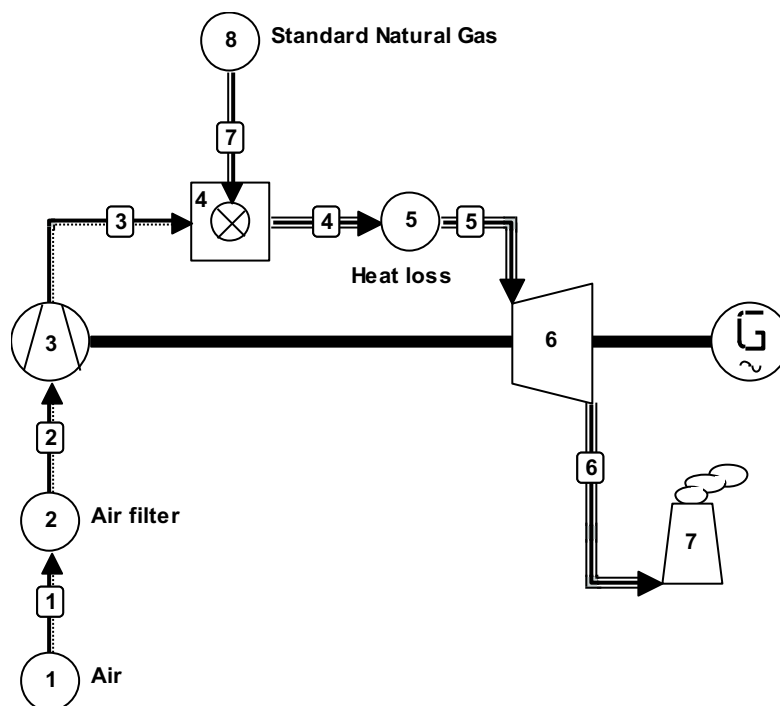
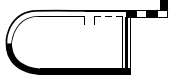


Figure 3-1: Gas turbine model



3.2 System description

Via source 1 air is fed into the system. The air is drawn-in via air filter 2 by compressor 3 and then led to combustor 4. Via source 8 the fuel, natural gas, is brought into the combustor. After the combustor the flue gases pass via sink 5, in which heat losses are taken into account, to turbine 6. In the turbine the flue gases expand to atmospheric pressure. The flue gases exit in stack 7.

3.3 Starting points for the calculation

Air supply 1

Air pressure: $POUT = 1.013$ bars
air temperature: $TOUT = 15$ °C
mass flow air: $DELM = -500$ kg/s

Air filter 2

pressure drop: $DELP = 0.01$ bar

Compressor 3

outlet pressure: $POUT = 13.7$ bars
isentropic efficiency: $ETHAI = 0.87$
mechanical efficiency: $ETHAM = 0.999$

Combustor 4

pressure drop: $DELP = 0.27$ bar
reaction pressure: $PREACT = 13.5$ bars
reaction temperature: $TREACT = 1100$ °C
estimate oxidant/fuel ratio: $ESTOFR = 25$
energy equation code: $EEQCOD = 1$ (calculate mass flow)

Cooler 5

energy loss: $DELE = 2000$ kW
estimate mass flow: $ESTMAS = 600$ kg/s

Turbine 6

inlet temperature:	TIN = 1100 °C
isentropic efficiency:	ETHAI = 0.86
mechanical efficiency:	ETHAM = 0.999
no governing stage:	GDCODE = 1

Stack 7

atmospheric pressure:	PIN = 1.013 bars
-----------------------	------------------

Fuel supply 8

outlet pressure:	POUT = 15 bars
outlet temperature:	TOUT = 15 °C
heating value fuel:	LHV = 37999 kJ/kg

Generator

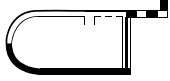
efficiency:	ETAGEN = 0.98
-------------	---------------

Pipe data

Pipe 1:	medium: standard air
Pipe 7:	medium: standard Slochteren natural gas

3.4 Results of the calculation

In table “Composition of fluids” the composition of the fuel (composition number 2), the air (composition number 1) and the flue gases (composition number 3) are presented. Table “Data for all pipes” contains a.o. the calculated mass flows of the fuel and the flue gases. The compressor and turbine powers can be seen in table “Energy balance”, whereas the net gas turbine power is shown in table “System efficiencies”.



Chapter 4

STAG-unit with back-pressure steam turbine

4.1 Introduction

In this chapter a simple example of a STAG unit is shown. This example contains three ‘independent’ cycles, namely a gas turbine cycle, a water/steam cycle and a district heating cycle.

The diagram is shown in Figure 4-1.

4.2 System description

In this example natural gas is fed to the gas turbine via source 1 and air via source 2. The gas turbine consists of compressor 3, combustor 4 and turbine 5. The flue gases from the turbine are passed to stack 6 via surface heat exchangers 18, 17, 14 and 32.

The flue gases are used to heat the water/steam in the water/steam cycle to an inlet temperature for steam turbine 10 of 450 °C. The steam then expands in the turbine and the heat is transferred via condenser 33 to the district heating system. Via pump 11, deaerator 12 and pump 13 the steam condensed to water is passed back to the waste heat boiler (heat exchangers 18, 17, 14 and drum 15) where steam is again produced.

The water from the district heating system is heated in condenser 33 and in an extra heat exchanger 32 in the gas turbine cycle. The heated district heating system is represented by heat sink 30.

The electric power is supplied by generators which are driven by the gas and steam turbine.

(see compressor 3/8)

isentropic efficiency: ETHAI = 0.85
inlet pressure turbine 5/3: PIN = 8.9 bars
outlet temperature turbine 5/3: TOUT = 530 °C
power: SURPMW = 30 MW (specified for the shaft)

Steam turbine 10

inlet pressure: PIN = 40 bars
inlet temperature: TIN = 450 °C

Pump 11

isentropic efficiency: ETHAI = 0.85

Deaerator 12

inlet pressure: PIN = 2 bars

Pump 13

isentropic efficiency: ETHAI = 0.85

Heat exchanger 14

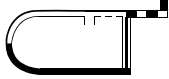
pressure drop primary side: DELP1 = 3 bars
pressure drop secondary side: DELP2 = 0 bar
temperature difference at
high-temperature side: DELTH = 20 K

Pump 16

isentropic efficiency: ETHAI = 0.85

Heat exchanger 17

pressure drop primary side: DELP1 = 1 bar
pressure drop secondary side: DELP2 = 0.005 bar
temperature difference at
low-temperature side: DELTL = 16 K



Heat exchanger 18

pressure drop primary side: DELP1 = 2 bars
pressure drop secondary side: DELP2 = 0.005 bar

District heating system 30

outlet pressure: POUT = 2 bars
outlet temperature: TOUT = 70 °C

Pump 31

outlet pressure: POUT = 15.0 bars
isentropic efficiency: ETHAI = 0.85

Heat exchanger 32

pressure drop primary side: DELP1 = 2 bars
pressure drop secondary side: DELP2 = 0.005 bar
outlet temperature secondary side: TOUT2 = 100 °C

Condenser 33

pressure drop primary side: DELP1 = 1 bar
pressure drop secondary side: DELP2 = 0 bar
outlet temperature primary side: TOUT1 = 110 °C
temperature difference at
high-temperature side: DELTH = 3 K

Generators

efficiency (both): ETAGEN = 0.975

Pipe data

Pipe 13: pressure drop DELP = 0.1 bar

Chapter 5

STAG installation Free University Amsterdam

5.1 Introduction

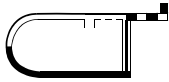
This example relates to the modelling of the combined heat and power installation of the Free University (VU) of Amsterdam, also called a mini STAG. The installation supplies heat and electricity to the University hospital. For this example, use is made of user subroutines, which also shows how the user can define his own namelists. The calculation diagram is shown in Figure 5-1.

5.2 System description

The diagram in Figure 5-1 comprises the following four cycles:

Cycle	Apparatuses	Pipes
gas turbine cycle	1- 18	1- 18
steam system	51- 62	51- 69
heating grid	101-104	101-104
cooling cycle	151-152	151-153

The combined heat and power installation is equipped with the Ruston Typhoon gas turbine, which is modelled mainly by apparatus 3 (air compressor), 5 (combustors), 7 (first stage of the expander) and 9 (second stage of the expander). The pressure drop at the inlet of the compressor is rediscounted in apparatus 2. Valve 4 separates a certain quantity of air to cool the blades of the first stage of the expander and for disc cooling. This cooling air participates in the expansion process of the second expander stage (9). The fuel for the gas turbine is Slochteren natural gas (source 16). In order to limit the NO_x production to 65 g NO_x/GJ



water injection is used. In the modelling this quantity of water (from source 17) is previously mixed with the fuel at junction 18, as only two incoming lines can be connected to apparatus type 13. In apparatus 6 radiation losses are taken into account. After the expander the heat in the flue gases is transferred to the steam system (heat exchangers 11, 12 and 13) and to the heating network (heat exchanger 14).

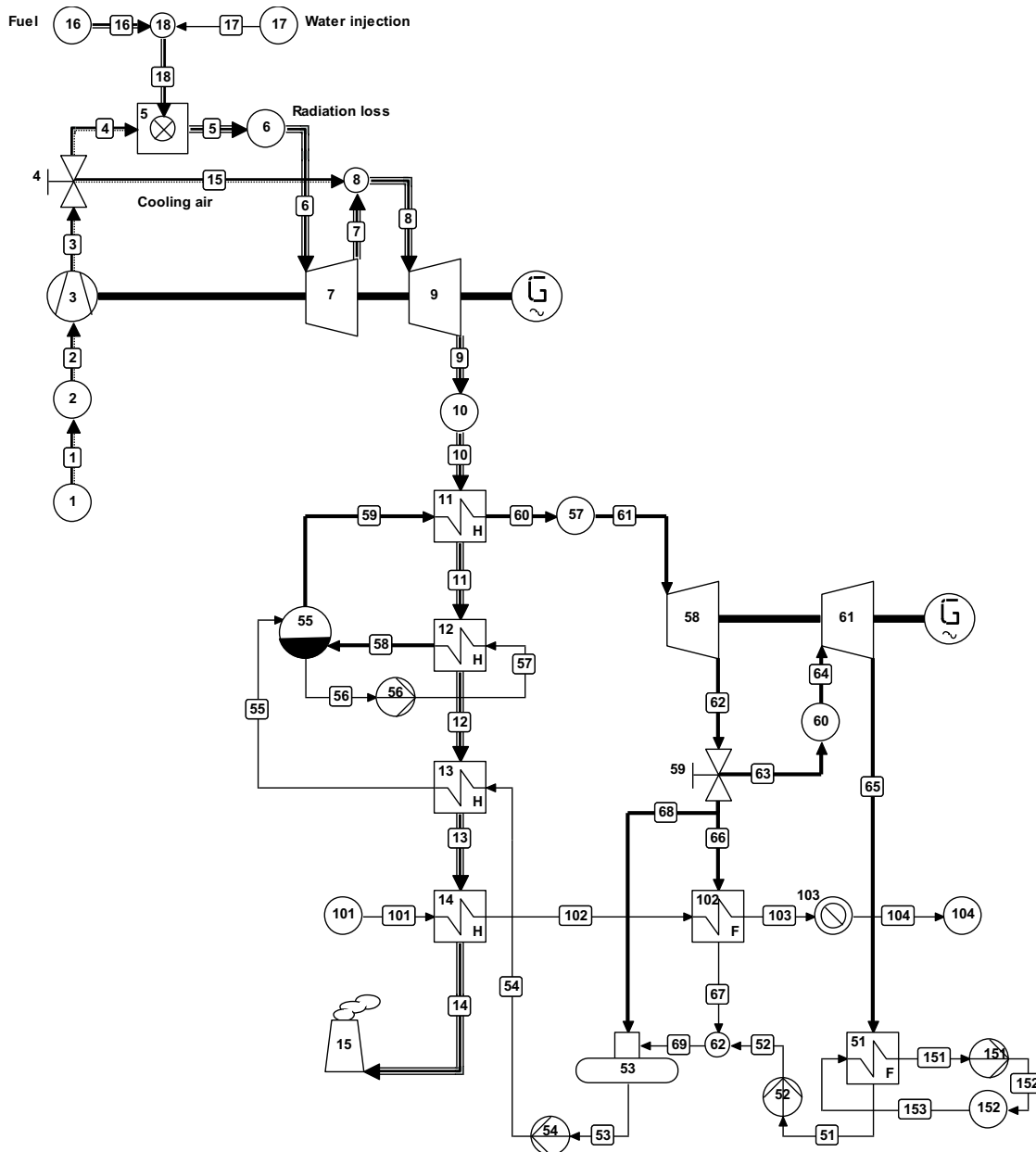
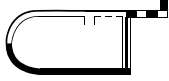


Figure 5-1: STAG installation Free University (VU) Amsterdam

The steam system supplies electricity and any heat for the heating network. The condensate from the condenser (51) is compressed by condensate pump 52 and pumped to the deaerator (53). The condensate is mixed with the condensate from the building heating heat exchanger 102. In the deaerator there is a steam rake, which is supplied with steam from the HP-steam turbine. Under normal conditions no steam is required for deaeration. This is possible by setting the pressure in the deaerator such that the incoming condensate (line 69) boils on its own. This pressure to be set is hence dependent on the mixture ratio between and the temperatures of the condensate streams from the condenser and the building heating heat exchanger. In the user subroutine the outlet pressure of the deaerator is set at the saturation pressure corresponding to the mixing temperature in line 69. It then follows from the calculation that the steam requirement for deaeration (line 68) is equal to zero. The steam rake comes into operation on its own where under emergency conditions the gas turbine has to be fired with oil and the inlet temperature of the feedwater in the exhaust gas boiler must be more than 100 °C.

The feedwater from the deaerator is transported by the feed pump 54 to the exhaust gas boiler. In the economizer 13 the feedwater is heated. The apparatus 12, 55 and 56 represent the evaporator with forced circulation. The steam from the drum (55) is superheated in the superheater 11. Pressure and heat losses of the live steam are rediscounted in apparatus 57. The steam expands in the HP-turbine 58 (a back pressure turbine), which drives a generator. Where there is no demand for heat, the steam can expand further in the LP-turbine 61. This LP-turbine is mounted on the same shaft as the HP-turbine. A carrier coupling makes it possible for the LP-turbine not to be in operation and the steam from the HP-turbine can hence be fully available for heat production. It is also possible for part of the steam from the HP-turbine to be used for heat production, and for the rest to expand in the LP-turbine. The heating network is modelled in apparatus 101 to 104. The conditions in line 104 are the same as in line 101, so that the heat given off to heat sink 103 represents the total heat transfer in apparatus 14 and 102. Apparatus 103 can be understood as the heated system (the buildings).



5.3 Summary of calculations

In principle a distinction can be made between five situations:

1. Design situation: The gas turbine runs at full load, and the installation supplies the maximum electricity production.
2. The gas turbine runs at full load, and the installation supplies the maximum heat production.
3. The gas turbine runs at part load, and the installation supplies the maximum electricity production at this gas turbine load.
4. The gas turbine runs at part load, and the installation supplies the maximum heat production at this gas turbine load.
5. The gas turbine runs at part load, and the installation meets a certain heat demand.

The files representing situations 1 (example 5), 3 (example 5a), 4 (example 5b) and 5 (example 5c) are supplied with the program. The part load percentage of the gas turbine for examples 5a and 5b is 75%. For example 5c the part load percentage and the heat demand are established at 95% and 1500 kW_{th} respectively.

Heat exchangers 11, 12, 13 and 51 as well as turbines 7, 9, 58 and 61 are designed for situation 1. For the other situations the design data which the program generates for this apparatuses can be used as input for off-design calculations.

Heat exchangers 14 and 102 must be designed for situation 2.

For the other three situations the design data from situations 1 and 2 can be used as input.

The user specifies the gas turbine load by the namelist CHP (= Combined Heat and Power), which is read in by user subroutine APSUB (see 5.5). In example 5c the user specifies in the same namelist how large the heat production must be. One should however take into account here that this heat production cannot be larger than the heat production in situation 4 for the same gas turbine load.

5.4 Starting points for the calculations

Here the most important starting points will suffice.

1. Design calculation (example 5)

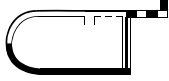
Gas turbine

The data relating to the gas turbine were supplied by the Dutch representative of Gowings Continental from Maasdam. For the gas turbine the following data are used, whereby in each case an indication is given of where this is processed in the input dataset:

- ambient temperature 15 °C (TOUT for 1/10, 16/10 and 17/10)
- inlet loss 0.01 bar (DELP for 2/10)
- compression ratio 12.8:1 (processed in POUT for 3/29)
- outlet temperature compressor 367 °C (TOUT for 3/29)
- water/fuel ratio 1:2 kg/kg (DELM = -0.5, PIPE = 16 for 17/10)
- inlet temperature expander 1053 °C (TIN for 7/3)
- outlet temperature expander 508 °C (TOUT for 9/3)
- outlet loss 0.02 bar (evenly distributed via DELP1 for 11/6, 12/12, 13/6 and 14/6)
- flue gas flow rate 17,1 kg/s (converted to air flow rate DELM for 1/10)
- efficiency gear box 0.985 } both efficiencies processed in
- efficiency generator 0.97 } ETAGEN of generator 1
- fuel: Slochteren natural gas (specified for pipe 16)
- electric power 3995 kWe (follows from the calculation)

In the first instance a calculation is output, where the DELM of 1/10 is not specified and for 15/10 DELM = 17.1 is specified. From this follows an air flow rate of around 16.57 kg/s.

For part load the fuel flow rate is controlled, and the air flow rate is constant. For this reason in the calculations the constant air flow rate is also assumed, specified for 1/10.



Steam system

The most important data for the steam system are:

- pressure live steam 24.1 bars (processed in turbine inlet pressure PIN for 58/3)
- temperature live steam 287 °C (TOUT1 for 11/6)
- pressure drop live steam 0.3 bar (DELP for 57/10)
- heat loss live steam 1 kJ/kg (DELH for 57/10)
- isentropic efficiency back pressure turbine 0.665 (ETHAI for 58/3)
- outlet pressure back pressure turbine 3.6 bar (processed in PIN2 for 102/5)
- isentropic efficiency condensation turbine 0.55 (ETHAI for 61/3)
- condenser pressure 0.2 bar (processed in PINL as extra condition for pipe 65)
- pinch-point temperature difference 10 °C (DELTL for 12/12)
- approach temperature difference around 5 °C (processed in TOUT1 for 13/6)

For the heat exchangers in the heating network (14/6 and 102/5) the outlet temperatures of the heating water (TOUT1) are specified. Via valve 59/14 it is specified that no steam goes to heat exchanger 102/5. From the energy balance over apparatus 102/5 a mass flow of heating water is then calculated equal to zero. From the energy balance over apparatus 14/6 it then naturally follows that the inlet temperature of the flue gases is equal to the outlet temperature.

The specification for valve 59/14 thus ensures that the situation of maximum electricity production is calculated.

2. Maximum heat production (gas turbine at full load)

No calculation example of this situation is included. The differences with regard to the design calculation are:

- for the apparatus 11/6, 12/12, 13/6, 51/5, 58/3 and 61/3 the design data (which are generated by the design calculation) are included as off-design input data for the relevant apparatuses)
- outlet temperature heating water building heating boiler bundle 97 °C (TOUT1 for 14/6)
- outlet temperature heating water building heating heat exchanger 134 °C (TOUT1 for 14/6)
- return temperature heating water 90 °C (TOUT for 101/10)
- no steam through condensation turbine (RMASS1 = 0.0 for 59/14)

3. Maximum electricity production (gas turbine on part load) (example 5a)

The design data for heat exchangers 14/16 and 102/5, generated by running situation 2, are imported as off-design input data. The design data for expanders 7/3 and 9/3 are imported also. The design data for apparatuses 14/6 and 102/5 are not added, as the installation in this situation does not produce any heat. This results in 8 apparatuses operating in off-design mode. A little information on the use of the part load data is appropriate here.

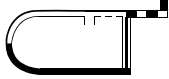
The design data of heat exchangers can be used by the program to calculate a temperature or a temperature difference in off-design situations.

Where the design data of a heat exchanger contains the UA-value for the design situation, then the procedure is that the user omits a temperature specification for the relevant apparatus. This results in the following:

11/6	TOUT1 omitted
12/12	DELTL omitted
13/6	TOUT1 omitted

Where the design data of a heat exchanger contains no UA-value (this is the case for apparatuses of type 5, which is used here for modelling the condenser), then the program alters the temperature difference (DELTH or DELTL) specified for the design situation on the basis of the actual mass flow rates. This design value of the temperature difference must remain as input for the relevant apparatus. To be able to process this procedure, the DELT1 for 51/5 in the design calculation is replaced by a DELTH = 5.0 K corresponding fully to this. Also for the off-design situation for apparatus 51/5 the input parameter DELTH is then kept. The program will use this value to determine a DELTH valid for the present situation (see Figure 2-3 in paragraph 2.6 of part “Reference Guide” of the manual).

The design data of the expanders and turbines in the system can be used by the program to calculate new inlet and extraction pressures on the basis of Traupel’s formula. This means that the user ensures that these pressures are not already established by input data. Also the user must check how far in the design calculation other specified input data must be omitted. The outlet temperature of the gas turbine for example is not constant; this means that in concrete terms for 9/3 TOUT must be omitted. As the air flow rate is constant, it is assumed that the isentropic efficiencies of the compressor and the two expanders are constant. These



isentropic efficiencies are, where they are not specified, calculated by the program in the design situation (see example 5). For this example this results in the following:

3/29	POUT omitted TOUT replaced by ETHAI
7/3	TIN omitted (see below)
9/3	PIN omitted TOUT replaced by ETHAI
58/3	PIN omitted

Finally it can also be indicated that valve 59/14 ensures that no steam goes in the direction of the building heating heat exchanger 102/5.

The user can specify in namelist &CHP what the part load percentage of the gas turbine must be. This part load percentage is converted to a mass flow of fuel. The fuel flow is therefore specified for 16/10. As the fuel stream is now known, for 5/13 the EEQCOD must be changed to 2 to calculate the outlet temperature. In order to leave this outlet temperature unknown, TIN must be omitted for 7/3. 5.5 deals with namelist &CHP.

4. Maximum heat production (gas turbine on part load) (example 5b)

The input dataset for example 5b shows a lot of similarity with the one dealt with above. The design data for heat exchangers 14/6 and 102/5 are added. The input data for these apparatuses are altered as follows:

14/6	TOUT1 omitted
102/5	Unchanged; the specified DELTL (valid for situation 2) is altered by the program.

Valve 59/14 now blocks the steam supply to the condensation turbine 61/3.

The user can again specify in namelist &CHP the part load percentage of the gas turbine.

5. Specified heat demand (gas turbine on part load) (example 5c)

The input data for example 5c correspond very much to that in example 5b. In addition to the part load percentage the user can specify how large the heat demand which the installation must meet is in kW_{th} . This heat demand does in fact determine the mass flow in the heating network. For this reason for apparatus 101/10 the mass flow is specified which is then altered in accordance with the heat demand specified. The quantity of steam necessary for this heat demand can be calculated from the energy balance of apparatus 102/5. The rest of the steam will expand in the condensation turbine. It is then no longer necessary to determine the mass flow by means of valve 59, but to let it be calculated by the program itself. Valve 59/14 is therefore replaced by fork 59/9. The translation of the specified heat demand into the heating water flow rate is further explained in 5.5.

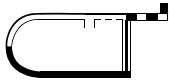
5.5 Use of user subroutine APSUB

For modelling the behaviour of the combined heat and power/power installation the facilities of the routine APSUB is used. In short the use of this routine can be described in this example:

- determination and control of the mass flow of fuel so that the specified part load percentage is reached;
- determination and control of the deaerator pressure so that no steam supply is necessary;
- determination and control of the water flow rate in the heating network so that the specified heat demand is met.

Subroutine APSUB is called for each apparatus which is specified at Calculation | User Subroutines in the row of apparatus numbers, in the order of the input.

In the declarations there is a namelist declaration. The namelist &CHP may contain the variables GTLOAD (the part load percentage of the gas turbine) and HEATPR (the heat production of the combined heat and power installation). The user adds this namelist to the input dataset. The namelist is read in by subroutine APSUB in the zeroth main iteration of the calculation process.



For the input variables GTLOAD and HEATPR a few mass flows specified in the input dataset must be altered. This relates to the specified mass flows for fuel for 16/10 and heating water for 101/10. The program does not have the facility to alter the input parameter DELM of apparatus type 10 where this is specified as an absolute mass flow. It is possible to alter the mass flow where this is specified relative to another mass flow in the system. As the air flow rate is constant at all times, the mass flows to be altered may then be related to the air flow rate, which enters the system via source 1/10.

As the mass flows to be altered are related to the mass flow of air in line 1, this air flow rate must be established in a variable. In the first main iteration the mass flow for the outlet line of apparatus 1 is allocated to the variable FMAIR (kg/s). As subroutine APSUB is then left and only later called for the next apparatus specified at Calculation | User Subroutines, variable FMAIR must be saved. This is the function of the SAVE statement: variable FMAIR is declared in the SAVE statement, so that in a subsequent call of subroutine APSUB the numerical value of variable FMAIR is still known. This also applies for the other variables, which are specified in the SAVE statement.

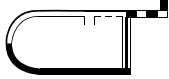
For part load operation of the gas turbine, the fuel input is controlled; the air delivery of the compressor is constant (there is no control by means of the adjustable blades). The part load operation relates to situations 3, 4 and 5 (see 5.3). For these situations the mass flow of fuel is specified for apparatus 16/10. The purpose is to adjust the specified mass flow such that the electric power of the gas turbine corresponds to the specified part load percentage (electric power of the gas turbine expressed in percent of the full load power). The mass flow of fuel is for 16/10 specified relative to the mass flow in line 1. This then offers the possibility of adjusting this input data in subroutine APSUB.

In general with part load operation the efficiency of a gas turbine will fall. This means that with a part load percentage of X% the fuel input will be more than X% of the fuel input at full load. It has hence been found that in this case for a part load percentage of 75% the fuel input will be around 78% of the fuel input at full load. A second point is determined in the same way for a full load percentage of 95%. A straight line is drawn through the points found for the relation between the fuel flow rate and the part load percentage, which is programmed in APSUB. In this way within certain limits the fuel flow can be determined using the part load percentage GTLOAD. For apparatus type 10 the parameter DELM is in memory place 8 in array DATA (see part “User Subroutines” of the manual). In the statement starting with DATA (8) the adjustment of DELM is thus carried out.

The value of the mass flow of fuel applicable in the present main iteration is both written to screen and to the output file.

For apparatus 53 the input parameter POUT must be altered. This parameter is stored in DATA (2) for apparatus type 7. First one finds out what the internal line number is for line 69 (see calculation diagram). This internal line number is stored under the name LCOND. In a simple way in array T the temperature in line 69 can be found. This is written both on the screen and in the output file. Routine PTXW (see part "User Subroutines" of the manual) is used to determine the saturation pressure of water PSAT at this temperature. This value is then allocated to DATA (2) or POUT of apparatus 53. In addition a stability criterion is built in here. The deaerator pressure is minimal, the condenser pressure (0.2 bar). During the iteration process negative temperatures can be calculated. Where this is the case, the deaerator pressure is not calculated but set at 0.2 bar. The saturation pressure is again shown on the screen and in the output file.

The heat production of the installation is equal to the heat which disappears in sink 103/10. The incoming enthalpy (line 103) is allocated to variable HMAX, the outgoing enthalpy (line 104) to variable HMIN. Where now the specified heat production HEATPR is divided by the difference between HMAX and HMIN, the water flow rate in the heating network is obtained. For apparatus 101 parameter DELM is altered in this way. The role of variable FMAIR in this has already been explained above.



5.6 Source text of user subroutine APSUB

```
*-----*
*   User subroutine APSUB, meant for ex5-ex5c   *
*                                               *
*   Mini-STAG VU-Amsterdam                   *
*-----*

      SUBROUTINE APSUB( NLIN, ILIN, IAP, ITYP, ITM, DATA, NP, IPIPE, FM, H,
*                   P, T, X, W, C, RESULT, COMP, CONC, AVERG, IDX, NCOMP,
*                   APNAME, NUMGEN, RDATGE)

      IMPLICIT DOUBLE PRECISION (A-H, O-Z)

      DOUBLE PRECISION DATA(50), FM(NP), H(NP, 3), P(NP, 3), T(NP, 2),
*                   X(NP, 2), W(NP), C(NP), CONC(NLIN+7, NCOMP),
*                   AVERG(NLIN+7), RDATGE(NUMGEN, 6)
      INTEGER        IPIPE(NP), ILIN(NLIN), IDX(NCOMP)
      CHARACTER      COMP(NCOMP)*8, APNAME*12

      NAMELIST /CHP/ GTLOAD, HEATPR
      SAVE GTLOAD, HEATPR, FMAIR, LCOND, LIN, LOU, HMAX, HMIN

*-----*
*   Read namelist &CHP only in iteration number zero   *
*-----*

      IF (ITM.EQ.0) THEN

          REWIND 9
          READ(9, CHP, END=1)
1      CONTINUE

*   Save air mass flow of apparatus 1
      IF (IAP.EQ.1) FMAIR=FM(1)

      ENDIF
```



```
*-----*
* Calculate fuel mass flow ( DELM <==> DATA(8) ) of source 16 *
*-----*

      IF (IAP.EQ.16) THEN

          DATA(8)=-1.0*(0.007472484*GTLOAD+0.25267991)*0.3682923/FMAIR

* Write results to standard message file
      WRITE(13,100) IAP
      WRITE(13,110) -1.0*DATA(8)*FMAIR

110   FORMAT(' Current fuel mass flow is          ',F6.2,' kg/s')

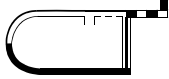
      ENDIF

*-----*
* Equalize the deaerator outlet pressure ( POUT <==> DATA(2) ) *
* to the saturation pressure at prevailing temperature *
*-----*

      IF (IAP.EQ.53) THEN

* Find internal number of deaerator outlet pipe
      IF (ITM.EQ.0) THEN
          DO 10 I=1,NP
              IF (ILIN(IPIPE(I)).EQ.69) LCOND=I
10          CONTINUE
      ENDIF

* Calculate saturation pressure
      IF (T(LCOND,1).GT.0.0) THEN
          W = 1
          PSAT = PTXW(T(LCOND,1),X,W,C,LCOND)
          DATA(2)=PSAT
      ELSE
          PSAT=0.2
          DATA(2)=PSAT
      ENDIF
```



```
*      Write results to standard message file
      WRITE(13,100) IAP
      WRITE(13,120) T(LCOND,1)
      WRITE(13,121) PSAT

120    FORMAT(' Current condensate temperature is ',F6.2,' °C')
121    FORMAT(' Current saturation pressure is   ',F6.2,' bar')

      ENDIF

*-----*
*      Calculate mass flow ( DELM <==> DATA(8) ) of source 101      *
*-----*

      IF (IAP.EQ.101) THEN

          DATA(8)=-1.0*(HEATPR/(HMAX-HMIN))/FMAIR

*      Write results to standard message file
      WRITE(13,100) IAP
      WRITE(13,130) -1.0*DATA(8)*FMAIR

130    FORMAT(' Current mass flow heating grid is ',F6.2,' kg/s')

      ENDIF
```

```
*-----*
*   Calculate enthalpies at in- and outlet of apparatus 103   *
*-----*

      IF ((IAP.EQ.103).AND.(ITM.LE.2)) THEN

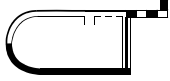
*       Find internal number of inlet and outlet pipe of apparatus 103
      DO 20 I=1,NP
          IF (ILIN(IPIPE(I)).EQ.103) LIN=I
          IF (ILIN(IPIPE(I)).EQ.104) LOUT=I
20      CONTINUE

          HMAX=H(LIN,2)
          HMIN=H(LOUT,1)

      ENDIF

100 FORMAT(/ ' ----- Apparatus',I4,' -----')

*-----*
*       END
*-----*
```



Chapter 6

District heating cycle

6.1 Introduction

As the sixth example an advanced system of district heating has been chosen. This example shows how a complicated scheme with several cycles influencing each other can be modelled. In addition an example is given of the use of user subroutines. The scheme is set out in Figure 6-1.

6.2 System description

Figure 2.6 shows the calculation scheme for a STAG installation with steam extraction for a district heating network. The scheme consists of four closed and one open cycle. The pipes and apparatuses are numbered per cycle by starting each cycle at a hundred.

In the open cycle (apparatuses 1 to 12 and pipes 1 to 11) natural gas is burned in combustor 4/13. The hot flue gases drive turbine 7 and are then used to provide energy to the water/steam cycle via heat exchangers 8 to 10.

In the water/steam cycle (apparatuses 101 to 116, pipes 101 to 122) the steam produced in heat exchangers 8 to 10 and superheated drives turbine 101. The steam is then divided via junction 102 between turbine 104 and heat exchanger 115, which supplies energy to the district heating cycle. After turbine 104 the steam is condensed in heat exchanger 105, by the cooling water cycle (apparatuses 301 to 302, pipes 301 to 303). In junction 107 the two mass flows come back together. The water is preheated in heat exchanger 108, after which it goes to the deaerator and is pumped on to heat exchanger 10.

The district heating cycle (apparatuses 401 to 403, pipes 401 to 405) receives its energy from the water/steam and the hot water cycle (apparatuses 201 to 204, pipes 201 to 206). The latter gets its energy from the flue gases via heat exchanger 11.

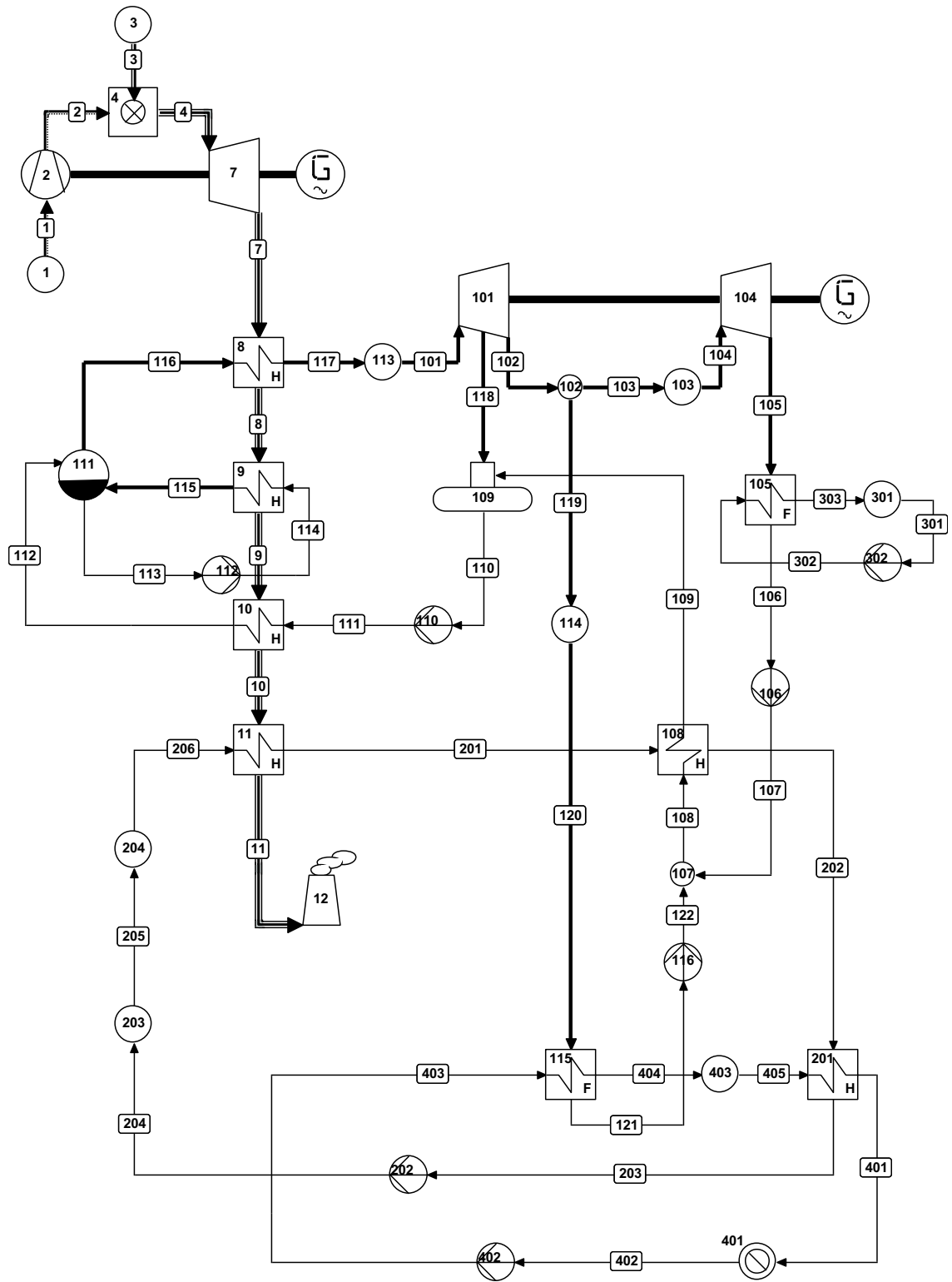
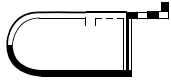


Figure 6-1: District heating cycle

6.3 Summary of calculations

For this system various calculations are performed.

Design calculations

Calculation 1 *Maximum electricity production*

In heat exchanger 115 no energy is given off and all the steam goes through turbine 104.

Calculation 2 *Maximum heat production*

Turbine 104 only gets a minimum quantity of steam, and also as much energy as possible goes to heat exchanger 115.

From these two design calculations the input data for the off-design calculations are formulated.

Off-design calculations

Calculation 3 *Off-design maximum electricity production*

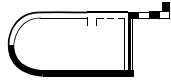
In fact the same calculation as calculation 1, but with relevant apparatuses operating in off-design mode.

Calculation 4 *Off-design maximum heat production*

In fact the same calculation as calculation 2, but with relevant apparatuses operating in off-design mode.

Calculation 5 *Off-design, average heat production*

The heat production is determined from the average of the heat given off in the district heating network from calculations 3 and 4.



6.4 Starting points for the calculation

For each calculation the energy supply is kept the same. This is done via a production function, in which the power which the generator of turbine 7 supplies, is set at 100 MW (specified for the shaft as Surplus Power).

The steam from the HP-turbine (101) can be divided via junction 102 between turbine 104 and the district heating network (sink 401) by heat exchanger (115). Water is supplied to this district heating network at a temperature of 75 °C. This returns at a temperature of 55 °C. The pressures in the cycle are also given, hence by specifying the quantity of energy to be discharged in apparatus 401, the mass flow in the district heating network is determined.

Variables which relate to conditions outside the system (inlet temperatures and pressures of air, fuel and cooling water and the outlet pressure of the stack (sink 12)) are also the same for each calculation.

To make the individual calculations possible the type of some apparatus is changed for a particular calculation:

- Apparatus 203 is a dummy sink in the design calculations, but a valve in the off-design calculations (type 14), in order to determine the mass flow.
- Apparatus 11 is type 12 in the design calculations, but of type 6 in the off-design calculations, as for the latter the outlet temperature of the stack is not given, but in place of it the mass flow of the hot water cycle is determined.
- Apparatus 102 is a valve in calculations 2 and 4, maximum heat production, (type 14), in order to determine the mass flow to turbine 104. In the other calculations it is a junction of type 9.
- Heat exchanger 201 uses $EEQCOD = 1$ in calculations 1 and 3, since in these cases the heat supply of apparatus 201 determines the mass flow in the district heating network; in the other calculations it uses $EEQCOD = 2$.

In the off-design calculations the mass flow in the cooling water cycle is specified and the variables which must be altered to achieve this: 'IN' of apparatus 301. In the data for sink 301 an initial estimate must be specified for TIN. The accuracy of this estimate can be very finely adjusted and the wrong estimate can sometimes result in the failure of a calculation.

6.5 Use of temperature estimates

In all the calculations apart from calculation 1 dummies of apparatus type 10 are used for temperature estimates to enable the program to start the iterative calculation process.

Initial estimates for the temperatures in pipes 404 and 206 are specified in the input window of apparatus 403 and 204, respectively. For apparatus 403 parameter ESTTIN is specified, for apparatus 204 parameter ESTTOU.

The passing-on of temperatures is necessary for the calculation. In order for example to calculate the temperature of pipe 404 that of pipe 405 must be known. But, to calculate the temperature of pipe 405, that of pipe 202 must be known. The circle continues via pipes 108, 122 and 121, and returns for pipe 404, for which the temperature must be known. The dummies are used to supply the program with initial estimates, which will be automatically overwritten by the calculated values during the iteration process.

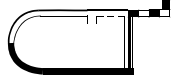
The following temperature estimates are specified:

Calculation 2: ESTTIN at apparatus 403 to pass the calculated temperature of pipe 405 to pipe 404 in succeeding iterations.

Calculation 3: ESTTOU at apparatus 204 to pass the calculated temperature of pipe 205 to pipe 206 in succeeding iterations.

Calculation 4 and 5: ESTTIN at apparatus 403 to pass the calculated temperature of pipe 405 to pipe 404 in succeeding iterations.

ESTTOU at apparatus 204 to pass the calculated temperature of pipe 205 to pipe 206 in succeeding iterations.



6.6 Results of the calculations

As the full calculation results are outside the bounds of this manual, there follows below just a few calculation results.

Temperature estimates:

Pipes 404 and 405 have in fact obtained the same ultimate temperature in calculations 2, 4 and 5, in which the temperature of pipe 405 is transmitted to pipe 404. For pipes 206 and 205 in calculations 3, 4 and 5 the same applies.

An extra condition is specified for pipe 301. There it is stated that the inlet temperature (TIN) of apparatus 301 (APNO = 301) must be adjusted to a prescribed mass flow (MASFLO). Hence a sort of iterative procedure is set in operation. If we look in calculations 4 and 5 at the ultimate mass flow in the cooling water cycle, then we see that this does not differ much from the mass flow specified.

In the situation of maximum heat production virtually no steam goes to the low-pressure turbine. Some steam supply is however necessary to remove the friction heat developed in the low-pressure turbine. The steam velocity at the outlet of the turbine is however so low that this falls outside the validity range of the relations used and the program warns of this. The enthalpy calculated from extrapolation is probably too low. As this is also a relatively small mass flow, this error has a negligible effect on the total heat balance. Only the heat discharge in the condenser will, in percentage terms, have a large error; in addition the results are very useful.