DEVELOPMENT OF A MODEL FOR THE SIMULATION OF ORGANIC RANKINE CYCLES BASED ON GROUP CONTRIBUTION TECHNIQUES

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ABSTRACT

Many industrial sectors and applications are characterized by the availability of low enthalpy thermal sources with temperatures lower than 400 °C, such as the ones deriving from both industrial processes (e.g. combustion products from gas turbines and internal combustion engines, technological processes and cooling systems) and renewable sources (e.g. solar and geothermal energy).

The usual systems for the conversion of thermal energy into mechanical and/or electrical energy work due to the high temperature difference available between the source (i.e., combustion products) and the sink (i.e., the ambient).

The Organic Rankine Cycle (ORC) is a promising process for conversion of heat at low and medium temperature to electricity. An ORC system works like a Clausius–Rankine steam power plant but uses an organic working fluid instead of water. A certain challenge is the choice of the organic working fluid and of the particular design of the cycle. The process should have high thermal efficiency and allow a high coefficient of utilization of the available heat source. Moreover, the working fluid should fulfill safety criteria, it should be environmentally friendly, and allow low cost for the power plant. An important aspect for the choice of the working fluid is also the temperature of the available heat source, which can range from low (about 100 °C) to medium temperatures (about 350 °C).

In this paper, a model for the simulation of Organic Rankine Cycles is presented. The model is based on thermodynamics tables for the calculation of fluid properties and the Lee-Kesler method for the calculation of specific heat. Six commonly used working fluids (propane, butane, benzene, toluene, R134a and R123) are considered. Both saturated and superheated cycles are evaluated. A sensitivity analysis of the main process parameters is performed. Finally, the model is applied to a micro gas turbine/ORC combined cycle.

NOMENCLATURE

С	specific heat
С	condenser
C_n	real gas heat capacity
$C_n^{r_0}$	ideal gas heat capacity
E^{r}	evaporator
EC	electric generator
GWP	global warming potential
h	enthalpy
т	mass flow
MGT	micro gas turbine
MW	molecular mass
n	number
ODP	ozone depletion potential
ORC	Organic Rankine Cycle
р	pressure
Р	power, pump
PP	pinch point
R	regenerator, generic result
S	entropy
Т	temperature, turbine
TIT	turbine inlet temperature
v	specific volume
V	volume
ΔC_p	residual heat capacity
ω	acentric factor
η	efficiency
Δ	coefficient in Eq. (1)
A, B, C, I	D coefficients in Eq. (2)

Subscripts and superscripts

L L	
critical	
current model	
cooling water	
electrical	
heat source	
	critical current model cooling water electrical heat source

i, i1, i2	intermediate
in	inner flow
is	isentropic
k	condensation
OF	organic fluid
out	outer flow
р	constant pressure
PP	pinch point
pump	pump
r	reduced (i.e. normalized by property at critical point)
ref	reference
t	thermal
turbine	turbine
v	vaporization

INTRODUCTION

Many studies on the Organic Rankine Cycle as energy system for the conversion of low enthalpy heat flux have been presented in the past. Some of these studies focus on the whole energy system [1-4] and its integration with other systems [5] and others on the working fluid [6-8].

In [1], a traditional ORC module (composed of heat exchangers, pump and turbine) has been developed and tested. The module is able to operate using low-temperature heat at 81 °C, but the obtained efficiency of this system is very low (about 4 %). The maximum achievable efficiency for a heat engine operating between the boiler saturation temperature of 81 °C and condenser saturation temperature of 38 °C is 12 % (Carnot efficiency). The low isentropic efficiency of the turbine (49.8 %) and fluid imperfections reduce this efficiency further. To improve the viability of the system tested in [1], authors suggested that the efficiency needs to be increased. They proposed (i) to increase the isentropic efficiency of the turbine and the reduction of the heat losses around the system by better insulation and (ii) to increase the boiler-condenser saturation temperature difference, thereby increasing the Carnot efficiency for the cycle.

In [2], three complete prototypes of a high-speed ORC have been made: two prototypes for medium temperature waste heat using R114 as the working fluid and one prototype for hightemperature waste heat using toluene as the working fluid.

In [3, 4], the thermosyphon Rankine engine is designed, manufactured and tested for power generation using solar or other available low grade heat sources. This engine is composed of a closed vertical cylinder that functions as an evaporator, an insulated section and a condenser. A turbine is placed in the upper end between the insulated section and the condenser section and a plate is installed to separate the high pressure region in the evaporator from the low pressure region in the condenser. Conversion of thermal energy to kinetic energy is achieved through nozzles. The diameter of the turbine is less than the internal diameter of the pipe, in order to provide space for the annular jacket for the collection of the returning condensed liquid. The return of the liquid is assisted by gravity through the difference of elevation of the liquid in the evaporator and the condenser section. The static head produced by the column of liquid in the return tube is the driving force against the pressure difference between the evaporator and condenser.

In [5], the hypothesis of a combined cycle composed of a low pressure ratio recuperated gas cycle with an Organic Rankine Cycle (ORC) is studied.

In [6], the authors used the BACKONE equation of state to calculate the thermodynamic properties of different working fluids for production of electricity from low temperature heat sources as, e.g., geothermal heat with a temperature of about 100 °C. Examples of these plants are the plants in Altheim, Austria, with a power production of 1 MW [9, 10] and in Neustadt-Glewe, Germany, with a power production of 0.2 MW [11]. Both plants initially used n-perfluoropentane as the working fluid. This substance, however, causes several problems. It is not environmentally friendly because of a very high global warming potential and a high atmospheric lifetime [12]. Moreover, as shown in [6], it has relatively low thermal efficiency in a simple cycle without the internal heat exchanger and the degree of utilization of the available heat source is limited by a high pinch point temperature. The highest cycle efficiency values (up to 14 %) are obtained for the high boiling substances with an overhanging saturated vapor line in the subcritical processes with an internal heat exchanger.

Authors in [7] address the feasibility of ozone-neutral low grade heat recovery to produce power by using an ORC module with a scroll expander. The low grade heat sources taken into consideration were an industrial exhaust and solar radiation. It has been shown that R-30 and HFE-7000 are viable working fluids for a low grade waste heat recovery, based on the boiling point at atmospheric pressure, ozone depletion potential, global warming potential, and safety issues. The theoretical ORC model based on HFE-7000 showed promising results of 11 % thermal efficiency for a 10 kW work output. The paper concluded that component research and economic considerations suggest that this system would be very attractive compared to an equivalent photovoltaic system.

In [8], a simple and compact power generator driven by waste heat is developed. Hot water at a temperature of 80 °C to 90 °C is assumed as the heat source. A feasibility study on the characteristics of a low temperature power cycle (evaporated at 77 °C, condensed at 42 °C) was conducted. As a result, TFE (Trifluoroethanol CF₃CH₂OH), R123, F245fa were selected as suitable fluids to optimize the cycle efficiency. Experimental validation of the power generator in which TFE was adopted as a working fluid was also conducted. A radial turbine was adopted as an expander, and was newly designed using an inverse design method, whereby the 3-D blade geometry for specified blade loading distribution was numerically obtained. The experimental results of the expander turbine performance, using TFE as a working fluid, were in agreement with CFD results.

In this paper, a model for the simulation of Organic Rankine Cycles is presented. The model is based on thermodynamic tables and correlations. Six commonly used working fluids (propane, butane, benzene, toluene, R134a and R123) are considered in saturated and superheated cycles. A sensitivity analysis of the main process parameters and validation are performed. Finally, the model is applied to a micro gas turbine/ORC combined cycle.

MODEL DEVELOPMENT.

The system. Plants based on ORC are usually composed of three main circuits:

• the first circuit (heat source circuit in Fig. 1) makes it possible to transfer heat from the heat source to the ORC unit. The kind of fluid, circulating in this circuit, depends the application of ORC units. Usually, the fluid is diathermal oil or pressurized water, but there are applications with direct heat transfer from hot gases;

• with the second circuit (main circuit in Fig. 1) the ORC unit operates as a completely closed process. The pressurized organic working fluid is vaporized (Rankine cycle), or superheated (Hirn Cycle), in the evaporator by heat transfer from the first circuit. The fluid then expands in a turbine which is directly connected to a generator. The low enthalpy head in the turbine allows the use of a regenerator to increase the cycle efficiency. Then, after the expanded working fluid passes through a regenerator, it enters into the condenser;

• the third circuit (cooling water circuit in Fig. 1) is the condensing water circuit. The low grade heat extracted from the ORC can be used for CHP applications (e.g. domestic heating).

Thermodynamic cycles. Depending on the source temperature, saturated cycle and slightly superheated cycle are considered.

The properties of the fluids considered make it possible to avoid condensation inside the turbine even without high superheated fluids. In fact, the fluids usually chosen for these applications have a positive slope of saturated–vapor line (i.e., the so-called "dry fluids"). The pressure of vaporization is much lower than a traditional steam cycle. For a given turbine inlet temperature, saturated cycles (Fig. 2) are always preferred to the superheated cycle [13-15], therefore it is advisable to use a fluid characterized by a critical temperature slightly higher than the vaporization temperature of the considered cycle.

The main limitation is imposed by condensation pressure, since it is not practicable to fall below 0.03 bar [16]. For high temperature heat sources, a saturated cycle requires a working fluid with a very high critical temperature which in general implies a very low condensation pressure. In such cases, a slightly superheated cycle should be adopted (Fig. 3).

Heat exchangers. The external heat sources and sinks are considered non-isothermal. Regenerator, evaporator and condenser are considered as surface counter-flow heat exchangers.

Regenerator. The use of high molecular mass organic fluids as working fluids implies an expansion in the turbine with a small enthalpy head, which corresponds to a small change in terms of temperature. Then, it is possible to use superheated steam leaving the turbine to heat the compressed fluid from the pump through the regenerator. This implies a reduction in thermal power which is necessary to provide the evaporation of the organic fluid. The minimum difference of temperature (pinch point) is located at the inlet liquid side (Fig. 4).



Figure 1 - Plant scheme



Figure 2 – Saturated cycle (fluid: Toluene)



Figure 3 – Superheated cycle (fluid: Toluene)

Evaporator. The liquid from the regenerator enters into the evaporator. In Fig. 5 the temperature profiles are reported. The presence of the isothermal section corresponding to vaporization, suggests a division of evaporator as follows: (i) a first stage, which works as an economizer, where the organic fluid is heated towards the evaporating temperature, (ii) a second stage for the isothermal vaporization and (iii) a potential third stage where the superheating of the organic fluid takes place.

Condenser. Unlike the traditional water steam cycle, the working fluid in ORC enters the condenser as superheated steam. In Fig. 6 the temperature profile are reported. The condenser consists of two different stages: (i) a first stage where desuperheating occurs with temperature variation, and (ii) a second stage where isothermal condensation take place, with enthalpy change only. The pinch point is placed corresponding to the phase change.

Pump and turbine. The compression and expansion within the pump and turbine respectively are considered nonisentropic and the end point of the real transformation is calculated by means of the definition of isentropic efficiency.

Thermodynamic properties. The thermodynamic properties of the considered organic fluids are calculated from literature p-v-T tables in the case of saturated liquid or vapor, while for superheated vapor the enthalpy is calculated by starting from the isobaric point on the saturated vapor curve and by using the specific heat at constant pressure.



Figure 4 – Temperature profile of regenerator



Figure 5 – Temperature profile of evaporator for (a) superheated cycle and (b) saturated cycle



Figure 6 – Temperature profile of condenser

The specific heat at constant pressure is obtained when the molecular structure of the fluid is known and literature correlations are used. This approach allows the model to be apply to each fluid whenever the p-v-T tables and molecule structure are known.

In this study, two different empirical methods to calculate heat capacity of ideal gas C_p^{0} , and therefore specific heat at constant pressure, are used. Both of the methods are based on the assumption that each functional group contributes to the overall specific heat in an additive manner.

The two methods taken into consideration are:

• Joback method [17], which is a polynomial method where the ideal gas heat capacity can be calculated by means of Eq. (1)

$$C_{p}^{0}(T) = \left(\sum_{j} n_{j} \Delta_{j,a} - 37.93\right) + \left(\sum_{j} n_{j} \Delta_{j,b} + 0.210\right) T + \left(\sum_{j} n_{j} \Delta_{j,c} - 3.91 \cdot 10^{-4}\right) T^{2} + \left(\sum_{j} n_{j} \Delta_{j,d} + 2.06 \cdot 10^{-7}\right) T^{3}$$
(1)

where n_j is the number of groups of *j*-th type and Δ_j contributions are for the *j*-th atomic or molecular group. The temperature T is the absolute temperature.

• Thinh Method [18], which is an exponential method where the heat capacity can be calculated by means of Eq. (2)

$$C_{\rm p}^{0}(T) = \sum_{\rm j} n_{\rm j} \Big[A_{\rm j} + B_{\rm j,1} \exp\left(-C_{\rm j,1}/T^{D_{\rm j,1}}\right) - B_{\rm 2} \exp\left(-C_{\rm j,2}/T^{D_{\rm j,2}}\right) \Big]$$
(2)

where n_j is the number of groups of *j*-th type and A_j , B_j , C_j and D_j coefficients refer to the *j*-th atomic or molecular group. Also in this case the temperature T is the absolute temperature.

The two methods are used alternatively depending on the type of fluid considered: the Joback method is used for halogenated hydrocarbons (e.g. refrigerants) while the Thinh method is most suitable for hydrocarbons in general (e.g. alkanes, aromatic hydrocarbons, etc.).

The heat capacity of the real gas is related to the value for the ideal gas state, at the same temperature by means of this equation:

$$C_{\rm p} = C_{\rm p}^0 + \Delta C_{\rm p} \tag{3}$$

where $\Delta C_{\rm p}$ is a residual heat capacity.

In order to calculate ΔC_p the Lee-Kesler method [19] is used. According to this method the residual heat capacity is given by

$$\Delta C_{\rm p} = \left(\Delta C_{\rm p}\right)^{(0)} + \omega \left(\Delta C_{\rm p}\right)^{(1)} \tag{4}$$

where $(\Delta C_p)^{(0)}$ is the simple fluid contribution and $(\Delta C_p)^{(1)}$ is the deviation function, which are given as a function of the

reduced temperature $T_{\rm r}$ and pressure $P_{\rm r}$, and ω is the acentric factor.

The acentric factor represented the acentricity or nonsphericity of the molecule: for methane, for example, it is very small, for higher molecular weight hydrocarbons, ω increases. It also rises with polarity. Essentially ω is used as a parameter to measure the complexity of a molecule with respect to both the geometry and polarity.

The equations used in the ORC thermodynamic model are presented in Annex A. In the following, the model is analyzed and validated. Then an example of the capabilities of the model is presented by applying it to a MGT-ORC combined cycle.

MODEL ANALYSIS

Working fluid. The fluids that are most appropriate in ORC units are dry fluids, which must have a relatively low temperature and critical pressure. After analysis related to security, availability and environmental compatibility (ODP and GWP), six working fluids were considered: four hydrocarbons and two refrigerants.

The working fluids considered are Propane (C_3H_8), Butane (C_4H_{10}), Benzene (C_6H_6), Toluene (C_7H_8), R134a ($C_2H_2F_4$), R123 ($C_2HCl_2F_3$), which are characterized by properties summerized in Tab. 1. These fluids are chosen since they are commonly used in ORC applications and many studies were presented in literature [13, 15, 20, 21].

Saturated cycle. The behavior of the model is analyzed by starting from the saturated cycle. The effect of the vaporization pressure and temperature on the cycle efficiency are presented in Fig. 7 for the six fluids. The comparison is made for a given condensation temperature of $T_{\rm k} = 300$ K, which corresponds to a different condensation pressure for each fluid. For this analysis and for the analyses presented below it is assumed that pump and turbine efficiency are constant and equal to 0.65 and 0.85 respectively.

As expected, by increasing the vaporization temperature the cycle efficiency increases. It can be highlighted that different fluids covers different ranges of temperature, and the envelope of the curves define a general trend. Efficiency also increases by increasing the vaporization pressure. Also in this case, each fluid covers a different range of pressure according to its critical properties.

Table 1 - Molecular mass and critical points of working fluids

		MW [kg/kmol]	<i>T</i> _c [K]	p _c [bar]	v _c [m ³ /kmol]
Propane	alkane	44.1	369.8	42.5	0.203
Butane	alkane	58.1	425.16	38.0	0.255
Benzene	aromatic	78.1	562.2	49.0	0.212
Toluene	aromatic	92.1	591.8	41.1	0.316
R134a	halogenated	102.0	374.2	40.6	0.198
R123	halogenated	152.9	456.9	36.7	0.287

The effect of the condensation pressure and temperature on the cycle efficiency are presented in Fig. 8 for the six fluids for the best efficiency vaporization conditions found in the previous analysis. Also in this case, the behavior of the model agrees with literature: by decreasing condensation temperature and pressure, the efficiency of the cycle increases.



Figure 6 - Liquid-vapor saturation curves of working fluids



Figure 7 – Cycle efficiency versus vaporization temperature and pressure for various working fluids



Figure 8 – Cycle efficiency versus condensation temperature and pressure for various working fluids

Superheated cycle. The effect of the superheating temperature on the cycle efficiency for two values of vaporization pressure (i.e. $p_r < 0.5$ and $p_r > 0.5$) are presented in Fig. 9. By increasing the superheating temperature the efficiency of the cycle increases both for low and high vaporization pressure.

MODEL VALIDATION

To evaluate the validity of the model, the results of the current model are compared with the results of a model presented in literature [21] and the results of a commercial code widely used for energy system modelling and analysis [22]. For the validation, a saturated cycle using toluene as the working fluid is considered. The parameters are reported in Tab. 2 while the results are reported in Tab. 3. It can be highlighted that the model is in agreement with the literature model and the commercial code. The relative error defined as



Figure 9 – Cycle efficiency versus turbine inlet temperature for various working fluids at low vaporized pressure, i.e. $p_r < 0.5$ (above), and at high vaporized pressure, i.e. $p_r > 0.5$, (below).

$$e = \frac{R_{\rm CM} - R_{\rm ref}}{R_{\rm ref}} \tag{5}$$

is in general lower than 5 %, with the exception of the thermal power transferred in the regenerator. This can be due to the fact that the heat exchange in the regenerator is not constrained by an external fluid, and, therefore, its determination has a higher degree of freedom.

MODEL APPLICATION

The model presented is used to calculate the performance of a unit ORC that is positioned as a bottomer of a micro gas turbine (MGT). In particular, exhaust gases from the micro gas turbine are used directly as the heat source for the ORC. The micro gas turbine considered is a Capstone C30 [23]. The fluid used is the Toluene and the saturated cycle is taken into consideration. The ORC unit is characterized by the data resumed in Tab. 4.

The heat exchanger outer gas flow temperature $T_{\text{out,HS}}$ is set to 413 K (i.e. 140 °C), in order to avoid the formation of acid condensation. The analysis with the model gives the results summarized in Tab. 5. It can be highlighted that the MGT-ORC combined cycle allows to increase the electrical power of the

plant from 30 kW to approximately 41 kW and the global efficiency from 26 % [23] to 35 %. This result is in agreement with other analyses presented in literature (e.g. [24]).

CONCLUSIONS

In this paper, a model for the simulation of saturated and superheated organic Rankine cycles was developed. The model is based on thermodynamic tables for the determination of saturated liquids and vapors while it uses thermodynamic correlation to calculate the specific heat. Six commonly used working fluids (propane, butane, benzene, toluene, R134a and R123) were considered, but the structure of the model allows the simulation of any fluid whenever p-v-T tables and molecule structure are known.

A sensitivity analysis on the main process parameters was performed. In the case of the saturated cycle, the model shows an increase in efficiency when the vaporization pressure and temperature are increased. The same behavior is highlighted by decreasing the condensation pressure and temperature. For the superheated cycle the effect of the superheating temperature is evaluated: the efficiency increases when the superheating temperature is increased.

The model is then validated by comparing it with model found in literature and commercial software. The model proved to be capable of correctly simulating an organic Rankine cycle with relative errors generally lower than 5 %.

Finally, the model was applied to a micro gas turbine/ORC combined cycle. The combined cycle allows the plant global efficiency to be increased from 26 % to 35 %.

ACKNOWLEDGMENTS

This work was carried out within the framework of 2008 PRIN national research project funded by MIUR (Italian Ministry of Education, University and Research).

The authors would like to thank Prof. Roberto Bettocchi and Prof. Pier Ruggero Spina for their valued suggestions.

Property	Value
$T_{ m v}$	548 K
$T_{ m k}$	311 K
$T_{ m in,CW}$	288 K
$\eta_{ m is,\ turbine}$	0.85
$\eta_{ m is,\ pump}$	0.65
$\Delta T_{ m PP,E}$	30 K
$\Delta T_{ m PP,R}$	10 K
$\Delta T_{ m PP,C}$	7 K

Table 2 – Operating parameter and components efficiencies

Table 3 – Comparison of results and relative errors

	Current model	Model in [15]	e [%]	TFLOW	e [%]
$P_{\rm e}$ [kW]	349	345	1.2	339	2.9
η [%]	31	31	0.0	30	3.3
$p_{\rm v}$ [bar]	24.2	23.9	1.0	24.3	-0.5
p_k [bar]	0.07	0.07	0.0	0.07	0.0
h_4 - h_{5is} [kJ/kg]	224	221	1.4	221	1.4
$P_{t,R}$ [kW]	234	222	5.4	204	15
$P_{t,E}$ [kW]	1140	1124	1.4	1124	1.4
$P_{t,C}$ [kW]	778	779	-0.1	775	0.4

Table 4 – Data for model application to MGT-ORC combined cycle

Property	Value
$T_{ m in,HS}$	548 K
$T_{\rm out,HS}$	413 K
$m_{ m HS}$	0.31 kg/s
$T_{ m in,CW}$	288 K
$\eta_{ m is,\ turbine}$	0.85
$\eta_{ m is,\ pump}$	0.65
$\Delta T_{ m PP, E}$	10 K
$\Delta T_{ m PP, R}$	10 K
$\Delta T_{ m PP,C}$	10 K

|--|

$P_{\rm e}$ [kW]	10.8
η_{cycle} [%]	24.57
$p_{\rm v}$ [bar]	5.22
$p_{\rm k}$ [bar]	0.05
h_4 - h_{5is} [kJ/kg]	224
$P_{t,R}$ [kW]	4
$P_{t,E}$ [kW]	44
$P_{t,C}[kW]$	34

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ANNEX A

MODEL DESCRIPTION

The main input of model calculation are the condensation temperature T_k and the vaporization temperature T_v in the case of saturated cycle. While the condensation temperature T_k , the vaporization pressure p_v and the turbine inlet temperature TIT are model input in the case of superheated cycle. The model calculates by means of thermodynamic tables the properties resumed in Table 6.

Then, by assuming $p_2 = p_{3'}$

$$h_2 = \frac{v_1(p_2 - p_1)}{\eta_{\text{is,pump}}} + h_1$$
(6)

It is assumed negligible the variation of the temperature due to pumping, i.e. $T_2 = T_1$.

Then, by assuming $p_{4'} = p_4 = p_3 = p_{3'}$ and $T_{4'} = \text{TIT}$ (this latter only in the case of superheated cycle)

$$s_{4'} = s_4 + \left[c_p(p_4)\right]_{T_4}^{T_4} \ln \frac{T_{4'}}{T_4}$$
(7)

$$h_{4'} = h_4 + \left[c_p(p_4)\right]_{T_4}^{T_4}(T_{4'} - T_4)$$
(8)

According to the definition of isentropic expansion $s_{5\mathrm{is}} = s_{4^{\prime}}$ and then

$$T_{5is} = T_6 e^{\frac{(s_{5is} - s_6)}{[c_p(p_6)]_{T_6}^{F_{5is}}}}$$
(9)

$$h_{5is} = h_{6'} + \left[c_{p} \left(p_{6'} \right) \right]_{T_{6}}^{T_{5is}} \left(T_{5is} - T_{6'} \right)$$
(10)

$$h_{5} = h_{4'} - \eta_{\rm is,turbine} \left(h_{4'} - h_{5\rm is} \right) \tag{11}$$

$$T_{5} = T_{6'} + \frac{(h_{5} - h_{6'})}{[c_{p}(p_{6})]_{T_{6'}}^{T_{5}}}$$
(12)

Then, the temperature inside the heat exchangers are calculated by considering the heat balances at the regenerator,

$$T_6 = T_1 + \Delta T_{\rm PP,R} \tag{13}$$

$$T_{3} = \frac{m_{\rm OF} [c_{\rm p}(p_{\rm 6})]_{T_{\rm 6}}^{T_{\rm 5}}(T_{\rm 5} - T_{\rm 6})}{m_{\rm OF} [c_{\rm p}(p_{\rm 1})]_{T_{\rm 5}}^{T_{\rm 5}}} + T_{\rm 1}$$
(14)

at the evaporator,

$$T_{\rm i1,HS} = T_{\rm 3'} + \Delta T_{\rm PP,E} \tag{15}$$

$$T_{\rm i2,HS} = \frac{m_{\rm OF}(h_4 - h_{\rm 3'})}{m_{\rm HS}c_{\rm p,HS}} + T_{\rm i1,HS}$$
(16)

at the economizer,

$$T_{\text{out,HS}} = -\frac{m_{\text{OF}} \left[c_{\text{p}} \left(p_{3} \right)_{T_{3}}^{T_{3}} \left(T_{3} - T_{3} \right) \right]}{m_{\text{HS}} c_{\text{p,HS}}} + T_{\text{i1,HS}}$$
(17)

at the superheater

$$T_{\rm in,HS} = T_{\rm i2,HS} - \frac{m_{\rm OF} \left[c_p \left(p_{4^{\circ}} \right) \right]_{T_4}^{T_4} \left(T_{4^{\circ}} - T_4 \right)}{m_{\rm HS} c_{\rm p,HS}}$$
(18)

and at the condenser

$$T_{\rm i,CW} = T_{\rm in,CW} + \Delta T_{\rm PP,C} \tag{19}$$

$$m_{\rm CW} = \frac{m_{\rm OF} (h_{6'} - h_1)}{c_{\rm CW} (T_{\rm i,CW} - T_{\rm in,CW})}$$
(20)

$$T_{\text{outCW}} = \frac{m_{\text{OF}} \left[c_{\text{p}} \left(p_{6'} \right) \right]_{T_{6'}}^{T_{6}} \left(T_{6} - T_{6'} \right)}{m_{\text{CW}} c_{\text{CW}}} + T_{\text{i,CW}}$$
(21)

Table 6 – Property calculated by means of tables

Input	Output
$T_1 = T_k$ $x_1 = 0$	$p_1 = p_k$ h_1 v_1
$T_{3'} = T_{v}$ $x_{3'} = 0$	$p_{3'} = p_v^{(\dagger)}$ $h_{3'}$
$p_{3'} = p_v$ $x_{3'} = 0$	$T_{3} = T_{v}^{(\ddagger)} h_{3'}$
$T_4 = T_v$ $x_4 = 1$	$egin{array}{c} h_4 \ s_4 \end{array}$
$T_{6'} = T_k$ $x_{6'} = 1$	$p_{6'} = p_k$ $h_{6'}$ $s_{6'}$

^(†) saturated cycle ^(‡) superheated cycle