The Associated Reversible Transformation Method for the Analysis and Diagnosis of a CO_2 Refrigerating Cycle

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Abstract:

In this paper, a carbon dioxide refrigerating cycle is used to illustrate an application of the 'Associated Reversible Transformation' (A.R.T.) Method. It can be used to describe each real thermodynamic transformation of the working fluid inside a thermodynamic cycle, similarly to the well-known description of ideal gas adiabatic compression by means of a polytropic transformation. Once a complete set of independent parameters of the carbon dioxide refrigerating cycle has been identified through the A.R.T. formulation of processes and constraints, numerical simulation techniques may be employed to identify the origin of the malfunction and useful information can be obtained by a comparison with the approach based on pure exergy balances. The application of A.R.T. approach to the energy diagnosis of a carbon dioxide refrigerating cycle shows that the actual origin of the malfunction can be clearly identified in all cases.

Keywords:

Energy diagnosis, Internal irreversibility, Refrigeration, CO_2.

1. Introduction

A general description of a thermodynamic cycle (whatever kind and number of transformations) as a function of a set of independent parameters characterizing thermodynamic transformations, as well as of constraints imposed on the cycle by plant physical nature, lay-out and controls, has been proposed in [1]. In that paper an 'Associated Reversible Transformation' (A.R.T.) is used to describe each real thermodynamic transformation of working fluid, similarly to the well-known description of ideal gas adiabatic compression by means of a polytropic transformation.

In this paper, a carbon dioxide refrigerating cycle is used to illustrate an application of A.R.T. In fact CO_2 is one of the emerging refrigerants for commercial refrigeration, a market which represents around 20% of the use of refrigerants and where the use of natural refrigerants is especially addressed. In Europe this is mostly due to the "F-gas regulation" which calls for a phase down schedule for HFCs [2]. In particular, as of 1st January 2022 placing on the market is prohibited for the most widely used HFCs (GWP higher than 150) used in multipack centralised refrigeration systems for commercial use, with a rated capacity of 40 kW or more. From this point of view carbon dioxide is preferred because of its negligible environmental impact when compared to synthetic refrigerants. However, due to its thermodynamic properties (especially critical temperature, about 31 °C), transcritical refrigeration cycles are to be considered for supermarket application in mild climates, because outdoor air is commonly used as a sink for heat rejection. [3, 4, 5]. A thorough comparison between CO_2 and other refrigerants in terms of energy efficiency has to be performed on an annual basis, since the high operating pressure of CO_2 allows setting lower condensing conditions in winter time, thus reaching higher energy efficiency. Therefore, most efforts are being spent in the investigation of transcritical operation which occurs when outdoor temperature raises over 25 °C, because in such conditions specific plant solutions are needed to improve the energetic and exergetic efficiency. In this paper the basic transcritical cycle is considered, as an example of application of the A.R.T. approach.

Once a complete set of independent parameters of the considered carbon dioxide refrigerating cycle has been identified through A.R.T. formulation of processes and constraints, numerical simulation techniques may be employed to perform perturbations analysis of whatever energetic magnitude around a workable condition. In the paper, the A.R.T. approach is used to perform the Energy Diagnosis of a simple refrigerating cycle which is supposed to be operating at off-design conditions, because of the deterioration in the performance of a component, or because of an increase of the ambient temperature, or for both. In the Energy Diagnosis the perturbed components (and/or external constraints) are not known *in advance*, but they have to be identified on the basis of some kind of analysis, starting from information about the current operating condition, supplied by the plant monitoring system. The comparison between exergy losses in the design and in the current operating conditions is a widely recognized methodology in literature (see for instance [6]).

By applying the A.R.T. approach, it is possible to obtain which part of whole exergy loss of the perturbed plant is related to each perturbed thermodynamic transformation of the cycle (trajectory) and to each perturbed external constraint, so that internal and external causes of additional irreversibility can be easily identified and separated.

A numerical approximation of the relation expressing additional exergy losses vs. A.R.T. coefficients is used in the paper to identify the origin of the additional irreversibility with respect to the design condition (often named *malfunction*) and a comparison is made with the approach based on pure exergy balances.

2. Associated Reversible Transformations (A.R.T.) in brief

The basic idea is to describe each real (irreversible) thermodynamic transformation of a generic thermodynamic cycle by means of an Associated Reversible Transformation (A.R.T.) of the same working fluid, following the same trajectory in the thermodynamic plane (h, s). It can be easily demonstrated [1] that, if the following dimensionless coefficients are supposed to be constant along a generic A.R.T., and the starting point of each transformation (or any other point) is known, they identify the A.R.T. pattern in the thermodynamic plane:

$$\pi_w \equiv \frac{T ds_{gen}}{\delta w_{rev}} = \text{const} \quad , \quad \pi_q \equiv \frac{T ds_{gen}}{\delta q_{rev}} = \text{const} \qquad (1)$$

$$h_{real} = h_{rev} \equiv h \qquad , \quad s_{real} = s_{rev} \equiv s$$

π_w and π_q may be named A.R.T. coefficients. They and their ratio ρ defined as $\rho = \pi_q/\pi_w$ have the following properties:

No work exchange process $\Rightarrow$ $\delta w_{rev} = T ds_{gen} \Rightarrow \pi_q = \rho; \pi_w = 1$ (2)

Adiabatic process $\Rightarrow$ $\delta q_{rev} = T ds_{gen} \Rightarrow \pi_q = 1; \pi_w = \dfrac{1}{\rho} = 1 - \dfrac{\delta w}{\delta w_{rev}}$ (3)

Throttling process $\Rightarrow$ $\pi_w = \pi_q = 1$ (4)

In practical applications, when a thermodynamic cycle is considered, made of *n* transformations, some of the starting points of each transformation are not known in advance, but some additional constraints contribute to determining the cycle together with thermodynamic transformation trajectories. It can be easily inferred [1] that *2n* further independent variables related to constraints have to be stated, in addition to the *n* independent variables $\{\rho_{12},.., \rho_{n1}\}$ related to the *n* transformations, in order to completely define the cycle path. If A.R.T. are used for describing the constraints (related, for instance, to ambient condition, control system or performance parameters) as well as to the trajectories, the *2n* further independent variables can be a set of A.R.T. coefficient ratios ρ and thermodynamic states (e.g. *h*) of some auxiliary points: $\{\rho_{A1},.., \rho_{An}, h_{A1},.., h_{An}\}$.

3. Energy Diagnosis with A.R.T.

Let us consider a functional decay in a component of a refrigeration system, working on the basis of a thermodynamic cycle, for instance in the compressor: as a well known consequence, additional losses may arise in all control volumes of the other components when the system is still working at nominal load. The value of the additional losses can be calculated on the basis of conventional parameters (the lowest value of η_{is} or η_{pol}), as well as of A.R.T. coefficients (the lowest value of π_w).

Some difficulties arise if the malfunctioning component is not known in advance, but it has to be identified on the basis of information supplied by the plant monitoring system (Energy Diagnosis). In fact this information allows the calculation of the thermodynamic states of all working fluids and of exergy losses in all components, but the picture often shows additional losses in all control volumes, not only in the actual malfunctioning one.

Through A.R.T. formulation of cycles, the analytical relation expressing additional exergy losses vs. a known set of independent parameters can be calculated, or simulation techniques can be used to obtain a numerical approximation of its first order Taylor expansion. For instance, if one extensive condition like the net cooling power output is fixed, additional exergy losses may be evaluated as:

$$\Delta W_{lost} = \underbrace{{}^{t}\left[\frac{\partial W_{lost}}{\partial \boldsymbol{\pi}_q}\right]\Delta\boldsymbol{\pi}_q + {}^{t}\left[\frac{\partial W_{lost}}{\partial \boldsymbol{\pi}_w}\right]\Delta\boldsymbol{\pi}_w}_{trajectories} + \underbrace{{}^{t}\left[\frac{\partial W_{lost}}{\partial \boldsymbol{\rho}_A}\right]\Delta\boldsymbol{\rho}_A + {}^{t}\left[\frac{\partial W_{lost}}{\partial \mathbf{h}_A}\right]\Delta\mathbf{h}_A}_{constraints} \tag{5}$$

In Eq. 5, each thermodynamic transformation of the cycle is described by means of its two A.R.T. coefficients (π_q e π_w), while each constraint is also described by auxiliary A. R. Transformations, identified by the A.R.T. coefficient ratio and the state of an auxiliary point. It is worth noting that constraints like a prescribed temperature or pressure can be introduced by means of A.R.T. coefficients as well as of conventional thermodynamic parameters, provided that the number of conditions imposed by the constraints on the points of the cycle is the same [1].

In Eq. 5, partial derivatives can be obtained through symbolic manipulation or simulation techniques, while the variation in all ART coefficients can be calculated starting from data supplied by the plant monitoring system, utilizing the relations presented in section 2. Therefore it becomes possible to relate, at least in principle, each perturbed trajectory in the state plane and each perturbed constraint to the associated variation in whole plant exergy losses. This information makes easier to identify actual malfunctioning components, or improper operation of the control system, that is the object of Diagnosis.

4. A Simple Carbon Dioxide Refrigerating Cycle

A vapour compression refrigerating cycle is usually made of a compressor, a condenser, an expansion device (throttling valve) and an evaporator. Heat is taken from the cold source through the evaporator, where the refrigerant evaporates, while heat is released at the hot sink at the condenser, where the refrigerant changes state from superheated vapour to saturated liquid. In a transcritical refrigerating cycle the refrigerant does not experience condensation, because both temperature and pressure at the compressor outlet are above the critical values. Thus the condenser is replaced by a *gas cooler*, and the layout of the system is represented in Figure 1.

Carbon dioxide at the exit of the gas cooler is in supercritical high density vapour state. It is then throttled to the evaporating pressure and evaporated.

In the refrigerating cycle considered in this paper the evaporating temperature is set at 0 °C, for food refrigeration purposes. The system is assumed to reject heat at outdoor air, entering the gas cooler at 20 °C and exiting at 50 °C, in nominal operating conditions.

Pressure level of carbon dioxide at the gas cooler is a crucial parameter for the performance of the system. In fact, due to the shape of isotherms in the near supercritical region, it strongly affects the performance of the system. Various correlations are available to optimise the gas cooler pressure as a function of the fluid temperature at the exit of the gas cooler itself. In this case the temperature of carbon dioxide at the exit of the gas cooler is set at 30 °C, and its pressure results 7. 86 MPa following the correlation proposed by [7]. Given the evaporating and gas cooling pressures, the pressure ratio is 2.29. The compression isentropic efficiency is assumed to be 0.75 [7].

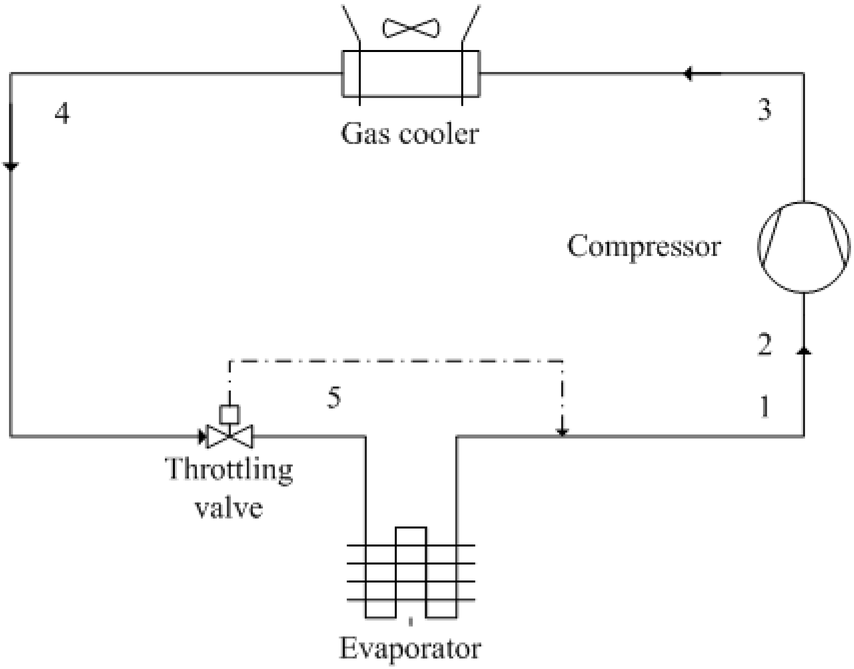

Fig. 1. Layout of a basic transcritical refrigeration system

Following the above mentioned assumptions, the refrigerating cycle is depicted in a pressure-enthalpy diagram in Figure 2. By imposing the global thermal transmittance in the gas cooler, KA= 12 kW/K; and the mass flow rate of the cooling air in the gas cooler, m_{air}=6.5 m^3/s, the unit gives a nominal cooling power equal to Q_{ev}=141 kW and a COP=3.09, corresponding to a work expenditure in the compressor equal to W_{comp}= 45.6 kW.

4.1 – A.R.T. Description of the Cycle

The basic transcritical refrigeration cycle is made up by the following thermodynamic transformations (in parentheses, performance values for design operating conditions):

- 1-2: quasi-isobaric heating, with pressure losses (δ_{p12}=0.5 bar; not appreciated in Fig.2);
- 2-3: adiabatic compression, with irreversibility (η_{is}=0.75);
- 3-4: quasi-isobaric cooling, with pressure losses (δ_{p34}=3.0 bar);
- 4-5: isenthalpic throttling;
- 5-1: quasi-isobaric evaporation, with pressure losses (δ_{p12}=1.0 bar)

At the same time, the following constraints affect the points of the cycle:

- Point 1: saturated vapour (it can be regarded as an effect of the liquid separator in the evaporator) and fixed temperature (it can be regarded as an effect of the control on the CO_2 mass flow rate);
- Point 2: fixed δT_{12} to prevent liquid from reaching the compressor;
- Point 3: fixed pressure as a function of T_4 ($P_3 = -1.15 + 2.66\ T_4$) to maximize the COP.
- Point 4: T_4 has to be regarded as, fixed from the refrigeration cycle point of view; in practice it is the consequence of the global thermal transmittance of the gas cooler and of the mass flow and temperature of the air used as cooling medium. In design operating conditions, T_4= 30°C.

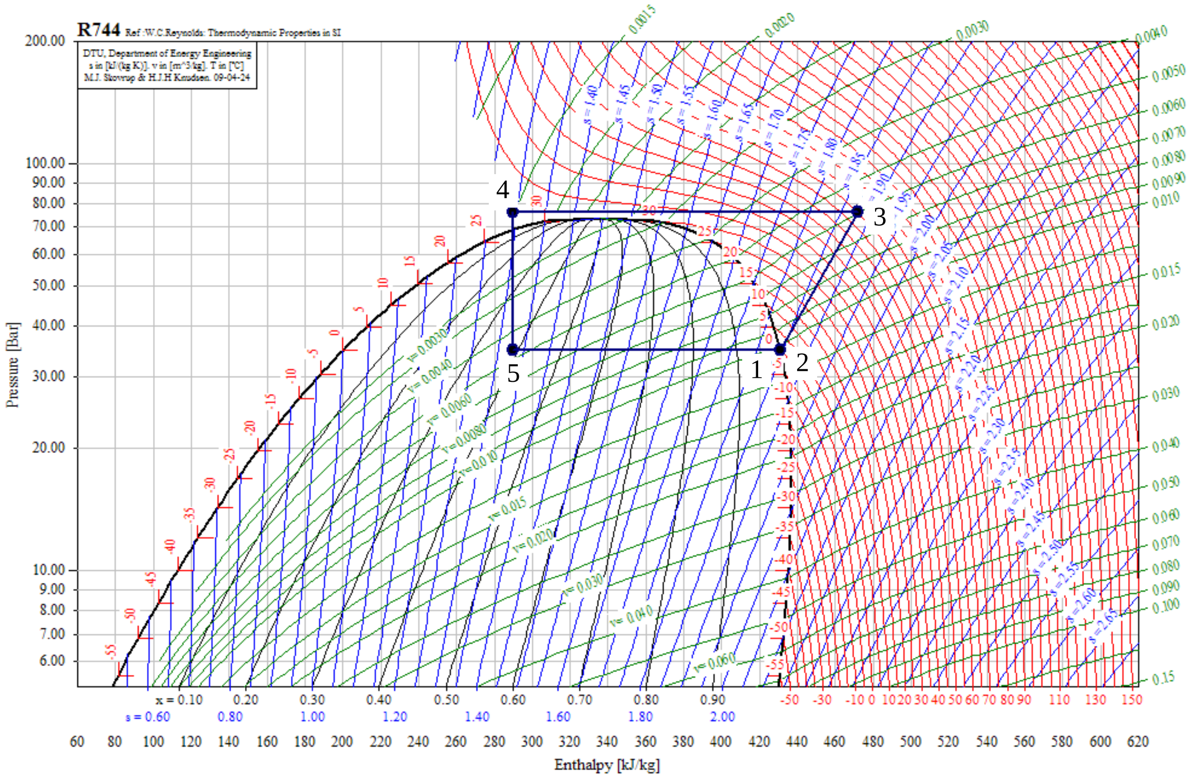

Fig. 2. Representation of the refrigerating cycle in a (p, h) diagram

By combining these five transformations and five constraints, the transcritical refrigeration cycle is completely defined and only one extensive variable, like the CO_2 mass flow rate, or the cooling power, is enough to determine all energy flows and the COP.

The A.R.T. coefficients completely describing the five transformations in design operating conditions are shown in table 1.

Table 1. A.R.T. coefficients describing the five transformations at nominal conditions.

π_{w12} =	1.000	π_{q12} =	0.05175	ρ_{12} =	0.05175
π_{w23} =	-0.2829	π_{q23} =	1.000	ρ_{23} =	-3.5350
π_{w34} =	1.000	π_{q34} =	-0.0053	ρ_{34} =	-0.0053
π_{w45} =	1.000	π_{q45} =	1.000	ρ_{45} =	1.000
π_{w51} =	1.000	π_{q51} =	0.0052	ρ_{51} =	00052

5. Results and Discussion

To show the application of the A.R.T. approach to the Energy Diagnosis of a refrigerating cycle, let us introduce four scenarios where components and/or external conditions of the refrigerating cycle previously described are affected by some modification of the actual operating conditions with respect to the design one, keeping constant the cooling power at the nominal value of 141 kW:

a) A reduction of compressor isentropic efficiency, $\Delta\eta_{is23}$ = -2%;

b) An increase in the ambient temperature, ΔT_{air}= +5°C;

c) A reduction of global thermal transmittance in the gas cooler, ΔKA= -0.5 kW/K;

d) The three perturbations *a)*, *b)* and *c)* at the same time.

Each scenario can be regarded as a test: the actual cause of the additional losses is known in advance, but the Energy Diagnosis has been developed without taking this information into account. On the contrary, only the information about the current state of the cycle has been used and compared with the state prescribed by the design condition. The expectation is that a good diagnostic methodology allows the actual cause of the additional losses to be clearly identified, showing that 100% of those losses are due to the perturbations introduced in the scenario considered, not to the malfunction of any other not-perturbed component, that would bring to a diagnostic error.

In practice, obtaining a perfect diagnostic indication is not realistic and some approximation has to be accepted. Nevertheless, the object of a perfect diagnostic indication can be approached to a different extent.

For each scenario, a different cycle has been obtained and the whole plant malfunctioning conditions have been calculated on that basis. Then, a numerical approximation of Eq. 5 is applied, to obtain which part of whole system exergy losses were related to each perturbed thermodynamic transformation (trajectory) and each perturbed constraint. The results presented in Table 2 show that the actual origin of the malfunction is clearly identified in all case by the A.R.T. approximation (column named ART approx).

In the same tables a comparison is made with the evaluation of pure additional exergy losses in each control volume (for exergy losses evaluation, see for instance [6, 8]. Contributions smaller than 1% are not shown, while the nominal losses are repeated in all scenarios to improve readability. The A.R.T. approximation for the whole system in scenarios *a* and *b* does not reach 100 % because of minor contributions being neglected.

If the malfunction is a reduction of compressor isentropic efficiency (scenario *a*), the A.R.T. approach allows the identification of the malfunctioning component (the one where transformation 2-3 takes place). If the design value of the isentropic efficiency were restored, the exergy losses of the whole system would go back to the nominal value, i.e. the additional exergy losses would be nullified; this means that the quantification of the intervention of restoring the design conditions in the component identified as the causes of the additional losses, would imply an exergy saving of 0,987 kW, which is predicted by the A.R.T. approach within an approximation of about 5% only.

Similar results can be obtained from the evaluation of exergy loss variation, with respect to the nominal condition; in fact in this case the losses allocated on the compressor are much closer to the actual value (in principle, all additional losses should be allocated on the compressor, that is known in advance as the origin of the performance decay), but a loss variation of about +/- 7% is also allocated on other components, which are not responsible for the performance decay of the whole system [9-11].

It is worth noting that the A.R.T. approach does not allocate any cause of malfunction on the throttling process, consistently with the idea that a pure dissipative component cannot work worst (or better) by itself, so that an increase (or a reduction) in the exergy destruction inside its control volume has necessarily to be regarded as the consequence of the behaviour of other, different, components.

Table 2. Results obtained for the four scenarios considered (losses in kW).

Scenario *a*:	Nominal losses (transformation Fig.2)		Actual losses	Exergy loss variations		ART Approx. (Eq. 5)
Total system	35.33		36.31	0.987	(100%)	*0.941 (95%)*
Super Heater	1.19	(1-2)	1.19	-0.002	-	-
Compressor	9.52	(2-3)	10.51	0.988	(100%)	0.941 (95%)
Gas cooler	14.15	(3-4)	14.22	0.067	(7%)	-
Throttling	9.97	(4-5)	9.91	-0.066	(-7%)	-
Evaporator	0.49	(5-1)	0.49	-0.001	-	-
Constraints	-		-	0	-	-
Scenario *b*:	**Nominal losses (transformation Fig.2)**		**Actual losses**	**Exergy loss variations**		**ART Approx. (Eq. 5)**
Total system	35.33		44.28	8.955	(100%)	*8,647 (97%)*
Super Heater	1.19	(1-2)	1.46	0.272	(3%)	-
Compressor	9.52	(2-3)	12.11	2.590	(29%)	0.158 (2%)
Gas cooler	14.15	(3-4)	16.80	2.651	(30%)	-
Throttling	9.97	(4-5)	13.34	3.366	(38%)	-
Evaporator	0.49	(5-1)	0.56	0.076	-	-
Constraints	-		-	$(T_{thr}) = 0$	-	8.489 (95%)
Scenario *c*:	**Nominal losses (transformation Fig.2)**		**Actual losses**	**Exergy loss variations**		**ART Approx. (Eq. 5)**
Total system	35.33		36.74	1.420	(100%)	*1.42 (100%)*
Super Heater	1.19	(1-2)	1.20	0.010	-	-
Compressor	9.52	(2-3)	9.81	0.290	(20%)	0.02 (1%)
Gas cooler	14.15	(3-4)	14.89	0.733	(52%)	-
Throttling	9.97	(4-5)	10.35	0.376	(26%)	-
Evaporator	0.49	(5-1)	0.50	0.010	-	-
Constraints	-		-	$(T_{thr}) = 0$	-	1.41 (99%)
Scenario *d* = *a*+*b*+*c*:	**Nominal losses (transformation Fig.2)**		**Actual losses**	**Exergy loss variations**		**ART Approx. (Eq. 5)**
Total system	35.33		47.34	12.010	(100%)	*12,05 (100%)*
Super Heater	1.19	(1-2)	1.47	0.280	(2%)	-
Compressor	9.52	(2-3)	13.75	4.220	(35%)	1,085 (9%)
Gas cooler	14.15	(3-4)	17.80	3.640	(30%)	-
Throttling	9.97	(4-5)	13.75	3.780	(31%)	-
Evaporator	0.49	(5-1)	0.57	0.080	-	-
Constraints	-		-	$(T_{thr}) = 0$	-	10.97 (91%)

A reduction of global thermal transmittance in the gas cooler and an increase in the ambient temperature are both regarded, in the A.R.T. approach, as perturbation of the external constraint represented by T_{thr} (scenarios *b* and *c*). In fact they both are limiting factor for the object of discharging heat from the compressed CO_2, to the external environment. The A.R.T. approach allocates all the additional losses on this constraint, within an approximation of about 5%, or less. On the other hand, the exergy analysis does not allow a clear identification of the causes of malfunction, and the additional losses appear spread on all components (scenario b, with increasing ambient temperature), or are allocated on the gas cooler in an amount of about 50% (scenario c, with decreasing global thermal transmittance).

The picture is even more complicated in the scenario *d*, where three components (throttling, compressor and gas cooler) show about the same contribution to the additional exergy losses, while the A.R.T. approach allocates less than 10% of them on the compressor, being the remaining 90% related to the external constraints affecting the gas cooler; by comparing scenario d and a, it can be inferred that this proportion is very close to the actual exergy saving that could be obtained by restoring the design condition separately for the compressor and for the T_{thr}. For separating the two contributions affecting T_{thr} and then identifying additional losses caused only by fouling of the gas cooler (reduction of its global thermal transmittance), a first order Taylor expansion of T_{thr} versus T_{air} and KA can be used, provided that the later two variables can be regarded as independent one another.

6. Conclusions

On the basis of A.R.T., a general description of a thermodynamic cycle may be obtained (whatever kind and number of transformations), as a function of a set of independent parameters characterizing thermodynamic transformations, as well as of constraints imposed on the cycle by plant physical nature, ambient conditions and controls.

Once a complete set of independent parameters has been identified, analytical expressions, or numerical evaluation of energy interactions, entropy generation or exergy losses may be obtained, in principle, as a function of those independent parameters.

If the object is the energy diagnosis of a simple refrigerating cycle, the A.R.T. approach allows, with good approximation, the identification of the malfunctioning component (the one where the perturbed thermodynamic transformation takes place, or which is affected by any perturbed external constraints) and the quantification of the exergy saving that could be obtained by restoring the initial conditions. This information is the rational basis for the evaluation of the economic convenience of actually restore the design condition in one or more components.

It is worth noting that the A.R.T. approach does not allocate any cause of malfunction on the throttling process, consistently with the idea that a pure dissipative component cannot work worst (or better) by itself, so that an increase (or a reduction) in the exergy destruction inside its control volume has necessarily to be regarded as the consequence of the behaviour of other components. On the other hand, the exergy analysis alone does not generally allow a clear identification of the causes of malfunction, and the additional losses appear spread on various components, or are allocated on the actual malfunctioning one only in a limited amount.

Nomenclature

f generic functions;

h specific enthalpy;

KA global thermal transmittance kW/°C;

m mass flow rate;

n	number of transformations;
Q	heat flow rate;
s	specific entropy;
T	temperature;
W	work;
W_{lost}	exergy losses;

Greek symbols

β	pressure ratio;
Δ	variation with respect of nominal conditions;
η	efficiency;
π_w, π_q	ART coefficients;
ρ	ART coefficients ratio;

Subscripts and superscripts

12	related to transformation from 1 to 2;
°	reference conditions;
A	auxiliary
air	cooling air in the gas cooler
comp	compression;
i,j	generic indices;
gen	generic
pol	polytropics;
rev	reversible;
thr	throttling
Tot	total system.

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