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A NEW METHOD FOR MODELLING OFF-DESIGN PERFORMANCE OF sCO2 HEAT EXCHANGERS WITHOUT SPECIFYING DETAILED GEOMETRY

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ABSTRACT

The advantage to using supercritical carbon dioxide (sCO2) in a power cycle (whether Rankine or Brayton) over other fluids is justified on the basis of cycle performance simulations. These simulations are influenced by the assumptions and resolution of turbomachinery and heat exchanger models. For turbomachinery, isentropic or polytropic efficiencies can be used for design conditions whereas generalized performance maps are used for off-design analysis. For heat exchangers, either the logarithmic mean temperature approach or the ε-NTU methodology can be used to obtain a first-order estimate of the required global heat transfer coefficient at the cycle design operating point for unknown heat exchanger geometry. However, neither of these methods allow the straight forward calculation of off-design heat exchanger performance. The work described in this paper discusses the development of a tool to evaluate off-design heat exchanger performance without specifying heat exchanger geometry. Special attention is given to the application of this tool to heat exchangers found in sCO2 power cycles. To that end, common assumptions to the treatment of overall heat transfer coefficient during heat exchanger design will be discussed using quantitative examples for sCO2 applications. In the examples,

the accuracy of modeling methods will be illustrated using both on-design and off-design operation of an sCO2 heat exchanger.

INTRODUCTION

FUNDAMENTALS OF HEAT EXCHANGER MODELLING

Heat transfer plays a crucial role in supercritical carbon dioxide power cycles and it is critical to achieving high cycle efficiencies. This is because of the highly recuperative nature of these cycles and the need to reduce the fluid temperature during the compression process. The large amount of recuperation in these cycles comes from the following properties of the cycle and working fluid:

- Compression near the supercritical point. Compression in a semi-incompressible state brings about minimum energy consumption and hence temperature rise. The consequence is peak efficiency at a very low compressor delivery temperature.
- Low specific heat ratio at turbine inlet, on the order of $k = 1.20 - 1.25$ for common turbine inlet temperatures and pressure. This low value of k brings about a very low isentropic temperature drop across the turbine, in the range from 120-140ºC depending on turbine inlet temperature and pressure ratio.

When all this is taken into account, there is a chance to recuperate up to roughly 80% of the total temperature rise from compressor delivery to turbine inlet by merely recuperating heat from the cycle itself (i.e., only 20% of the total temperature rise coming from external heat input)[1]. This can be informally termed as recuperative potential of the supercritical carbon dioxide cycle:

$$\psi_{rec} = \frac{T_{T,out} - T_{C,out}}{T_{T,in} - T_{C,out}}$$

The heat exchanged in the recuperator is subjected to a very large temperature change (up to 400ºC or more depending on cycle configuration and turbine inlet temperature).

While the recuperator is key to reducing the external heat input to the cycle, the cooler plays a fundamental role in reducing temperature at the compressor inlet and consequently overall compression work. This heat exchanger is located downstream of the recuperative heat exchanger, on the low pressure side, and might incorporate a condensing section (for liquid compression sections). Regardless of the working fluid state at compressor inlet, the most challenging aspect of the coolers is the very low outlet temperature on the cycle side, which poses the following design challenges:

- Very small temperature difference between the hot and cold fluids (pinch point) for which large exchange areas are mandatory.
- High volumetric flow rates on the coolant side.
- Dissimilar thermo-physical properties of coolant and working fluid and, for the latter, marked variation of these properties in a narrow temperature range (risk that the minimum temperature difference is found internally rather than at one end of the heat exchanger).

The variation of properties is nevertheless common to recuperator and cooler and represents one fundamental difficulty when modeling heat exchanger equipment in supercritical CO_2 cycles. This singularity is the non-linear cooling/heating line in a temperature vs. heat exchanged (TQ) plot whose main effect is discussed below.

Two modeling approaches are typically used during the first stages of cycle design (i.e., thermodynamic modeling):

[1] Note that these figures apply to an ideal simple cycle comprising compressor, heater, turbine and cooler. For more complex cycles, the recuperative potential might change in either way.

- Flange to flange (also lumped volume). Only the inlet/outlet temperatures on both sides are relevant inasmuch as temperature variations of the hot and cold fluids are linear in a TQ diagram. This approach is appropriate if the fluids exhibit linear properties with temperature.
- Discretized approach whereby a number of internal temperature nodes complement the inlet/outlet values of temperature on both sides. In practice, this approach transforms the original HX into a number of HX in series, each one of which can be modeled individually. This type of model is appropriate when any of the fluids (hot/cold) exhibits nonlinear properties with respect to temperature and its main disadvantages are the longer computational time and iterative calculation scheme. The number of internal nodes required for such a calculation depends on the temperature derivative of the properties of interest ($\partial\varphi/\partial T$).

Generally, heat transfer in heat exchangers is modeled with a variant of the following general equation:

$$Q = U \cdot A \cdot f(\Delta T)$$

Where U is the overall heat transfer coefficient. This coefficient and its associated area are made up of the various parts of the thermal circuit that connect the hot and cold streams of the heat exchanger. These terms are often expressed as a combination of the convective heat transfer on the hot and cold sides and conduction across the wall separating both fluids such as in the following equation[2].

$$\frac{1}{UA} = \left.\frac{1}{hA}\right|_{hot} + R_{wall} + \left.\frac{1}{hA}\right|_{cold}$$

When designing a heat exchanger, the on-design performance of the heat exchanger is only a function of the overall UA term. Thus, after the on-design flow conditions are established and the required overall UA computed, the hot and cold geometries can be specified in such a way as to ensure that the overall UA term is met. Because of this, on-design heat exchanger performance is not a function of the individual terms combined to form the overall UA term, but only the resultant value of the UA term itself.

The off-design performance analysis of a given heat exchanger is based on the same principles but a different approach is used. The individual components that make up the definition of the overall UA term are evaluated first using the off-design flow conditions and then combined to form the overall UA term, thus enabling the calculation of the heat exchanger performance.

As a consequence of this, there are many different heat exchangers designs that would produce the same overall UA term and thus the same on-design performance. When evaluated at an off-design condition however, each of these heat exchangers could potentially have very different performance. This is because their individual components, i.e. hot and cold geometry, could react differently to the same set of off-design flow conditions. These terms would then combine to form different values of the off-design UA term and thus produce different off-design heat exchanger performance.

In order to gain insight into the off-design performance of heat exchangers before specifying heat exchanger geometry, we define the following relationship:

$$hA_{ratio} = \frac{hA_{hot}}{hA_{cold}}$$

The hA_{ratio} is essentially a term that defines how the on-design UA term is divided into its constituent parts. For example, if a heat exchanger has an hA_{ratio} that is very high, this indicates that the majority of the resistance to heat flow comes from the cold side of the heat exchanger. Thus, during off-design operation, the heat exchanger will be more sensitive to the off-design flow conditions on the cold side of the heat exchanger.

This paper will show that if the value of hA_{ratio} is specified then the off-design performance of the heat exchanger can be estimated without detailed geometric design. By varying the value of $hA_{ratio}$the

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[2] This is the simplest case and other contributions to the overall thermal resistance can be found in practice.

investigator can explore potential off-design heat exchanger performance while doing initial cycle analysis. The investigator can easily see the possible range of potential off-design performance and then specify the hA_{ratio} when creating heat exchanger requirements; in the subsequent design phase, this would act as additional boundary condition (constraint) to the process. This will ensure that the production heat exchanger will have the desired off-design performance as simulated during cycle analysis. This method also allows one to predict the off-design performance of an existing heat exchanger without knowing its detailed geometry if the value of hA_{ratio} is known or can be estimated.

COMMON SCIENTIFIC/INPRACTICE AND SCOPE OF WORK

Off-design operation is usually defined by a change in the boundary conditions of the heat exchanger. Often, this change affects the mass flow rate on one or both sides of the heat exchanger as well as the inlet temperatures and presures. Situations where mass flow rates remain constant and inlet pressures and temperatures vary are not common in power generation applications. Let us think about steam condensers, feedwater heaters, steam generators, gas turbine recuperators; all these equipment are subjected to mass flow rate variations which bring about subsequent variations of Reynolds and Nusselt numbers. The overall effect of these changes is a change in the constituents of the UA term which will change in a somewhat unpredictable manner.

The inherent difficulty of anticipating the UA term variations in a heat exchanger of unknown geometry is commonly encountered when doing preliminary cycle analysis (such would be the case of a tender developing basic engineering for a bid). This is usually solved by either of the following methods:

- UA is assumed to be constant.
- If the performance of the heat exchanger is defined by terminal temperature difference, this is assumed to be constant.
- If the performance of the heat exchanger is defined by effectiveness, this is assumed constant or, as much, variable according to a generalized $\varepsilon - NTU$ wherein the UA term in NTU is constant.
- A generic, draft heat exchanger detailed geometry is created that meets the on-design performance requirements. This heat exchanger is then evaluated at the desired off-design condition.

Of particular note is the use of a generic or draft heat exchanger. The use of this method may seem straightforward but it brings with it many complications. It might happen that an HX design code is not available; such would be the case of a scientific approach by thermodynamic-focused researchers who are not familiar with heat exchanger technology. Or possibly the type of HX to be used has not been decided yet, as happens in the early stages of industrial development. The biggest issue with this method is that by creating a draft geometry, the investigator is in effect fixing the off-design performance of the heat exchanger. This is problematic as the detailed heat exchanger design, or even heat exchanger type, may vary markedly from the assumed draft geometry. Also there may be other off-design scenarios that are more favorable than those predicted by the draft geometry that would be hidden from the investigator as the draft geometry method predicts only a single value for off-design performance.

In the previously described context, this paper presents a method to calculate off-design heat exchanger performance without specifying detailed geometry. This allows the investigator to explore the possible off-design space for the specified on-design heat exchanger performance. The results from using this method are then compared to experimental results from an sCO2 heat exchanger to show validity as well as a developmental PCHE design code to show applicability to an sCO2 heat exchanger relevant to the sCO2 community

METHODOLOGY

The calculation method presented in this paper for predicting off-design heat exchanger performance without specifying heat exchanger geometry can be broken into two basic steps. First, the performance of the heat exchanger is evaluated at an on-design condition. Second, the results from the first step are scaled based on the desired off-design condition to estimate the off-design performance of the heat exchanger.

CALCULATION OF ON-DESIGN PERFORMANCE

For calculation of on-design heat exchanger performance, the following boundary conditions are needed: stream fluids and massflow values for both sides of the heat exchanger, inlet temperatures and pressures as well as outlet pressures for both sides of the heat exchanger, and one outlet temperature to calculate overall heat exchanger duty. Along with these values, the desired value of hA_{ratio} is also specified at this point.

With these data in hand, the overall heat exchanger duty is calculated from the resulting stream enthalpies and massflows. For this and all other steps, NIST REFPROP is used as the fluid property source. The heat exchanger is then broken up into an arbitrary number of divisions with the assumption that each division performs the same duty. With this assumption and by defining the pressure drop through the heat exchanger as linear, the thermodynamic state is fixed for each division of the exchanger. Mean fluid properties are then calculated for each division including temperature, density, thermal conductivity, Prandtl number, and viscosity.

In each division, the following equation is then used to calculate the division UA term:

$$Q = UA * \Delta T$$

In this equation the values for Q and ΔT calculated previously are used to solve for UA in each division. Using the hA_{ratio} defined previously, the calculated UA term is divided into its constituent hA terms using the following equations and ignoring wall resistance:

$$Given: hA_{ratio} = \frac{hA_{hot}}{hA_{cold}} \ and \ \frac{1}{UA} = \frac{1}{hA_{hot}} + \frac{1}{hA_{cold}}$$

$$Substituting: \frac{1}{UA} = \frac{1}{hA_{hot}} + \frac{1}{hA_{hot}/hA_{ratio}}$$

These equations can then be solved for $hA_{hot}$and hA_{cold}. With mean fluid properties and values for hA for each division in both hot and cold sides, the calculation of off-design heat exchanger performance can begin.

CALCULATION OF OFF-DESIGN PERFORMANCE

In order to begin the off-design performance calculation, the following parameters defining the desired off-design heat exchanger operating point are needed: new stream fluids and massflows for both sides of the heat exchanger and new inlet temperatures and pressures for both sides of the heat exchanger.

Using the same number of divisions as was used in the on-design performance calculation step, initial guesses for the temperature and pressure profiles are created. In order to calculate the actual temperature and pressure profiles through the exchanger, an iterative scheme that relies on an energy balance between the hot and cold sides of each division is used.

In this scheme, an iteration is defined as one pass through each heat exchanger division. First, for each division, mean fluid properties are calculated. Similar to the on-design calculation, in each division, the following equation will be used, but in this case it is used to calculate the division duty from which new downstream temperatures for the next iteration will be calculated.

$$Q_{off-design} = UA_{off-design} * \Delta T_{off-design}$$

The division $UA_{off-design}$ term is estimated by scaling the on-design division hA terms for both the hot and cold sides using the following scaling law:

$$hA_{off-design} = hA_{on-design} * \left[\frac{\lambda_{off-design}}{\lambda_{on-design}}\right] * \left[\frac{Re_{off-design}}{Re_{on-design}}\right]^x * \left[\frac{Pr_{off-design}}{Pr_{on-design}}\right]^y$$

Because the off-design flow conditions on the hot and cold sides of the heat exchanger are independent, the hA term for each division is scaled separately for each side of the heat exchanger. To aid in the calculation, the Reynolds number is defined in terms of stream massflow and the characteristic length is assumed to be constant. This allows the scaling law to be written as:

$$hA_{off-design} = hA_{on-design} * \left[\frac{\lambda_{off-design}}{\lambda_{on-design}}\right] * \left[\frac{\dot{m}_{off-design}/\mu_{off-design}}{\dot{m}_{off-design}/\mu_{on-design}}\right]^x * \left[\frac{Pr_{off-design}}{Pr_{on-design}}\right]^y$$

The values for x and y can be defined by the user and can be specified separately for the hot and cold sides. In all the results outlined in this paper, the Reynolds number and Prandtl number exponents suggested by the Dittus-Boelter pipe flow Nusselt number correlation were used.

The hot and cold division hA terms are then combined to form the overall division off-design UA term as shown in the following equation:

$$\frac{1}{UA_{off-design}} = \frac{1}{hA^{hot}_{off-design}} + \frac{1}{hA^{cold}_{off-design}}$$

The off-design UA term is then used to calculate the division duty which is the same for both sides of the heat exchanger. With the division duty, the downstream enthalpy of both the hot and cold sides can be computed. When combined with the downstream pressures, the enthalpies can be used to calculate the downstream temperatures and all other required fluid properties.

In order to estimate the off-design division pressure drop, a similar scaling approach is taken using the following scaling law for both the hot and cold sides of the division:

$$\Delta P_{off-design} = \Delta P_{on-design} * \frac{\dot{m}^2{}_{off-design}/\rho_{off-design}}{\dot{m}^2{}_{on-design}/\rho_{on-design}}$$

Using the results from this equation the pressure can be updated in each division during each iteration.

This process continues from division to division until the iteration is complete. A new iteration is then begun. The iteration process terminates once the changes in the outlet temperatures form iteration to iteration falls below a certain tolerance.

RESULTS

In order to evaluate the performance of our method, the results from this method are compared first to a set of experimental off-design sCO2 heat exchanger data and then to the results from a heat exchanger design code which incorporates detailed geometric design.

EXPERIMENTAL DATA FROM BMPC

In support of their ongoing research into sCO2 power cycles, Fourspring and Nehrbauer from Bechtel Marine Propulsion Corporation recently published details on an sCO2 heat exchanger which included experimental off-design performance data (Fourspring & Nehrbauer, 2015). In their paper, the authors describe the heat exchanger as a low-fined, shell and tube type with water on the tube side and sCO2 on the shell side. The heat exchanger was tested at a variety of conditions derived from the design conditions found in the following table.

	$\dot{m}\left[\frac{kg}{s}\right]$	T_{in} [ºC]	$\bar{p}$ [bar]	Δp [bar]	Q [kW]
sCO2	1.36	58.89	95.15	<2.07	100
Water	1.41	18.33	4.83	<2.76	100

Table 1. Design conditions used by BMPC (Fourspring & Nehrbauer, 2015)

Due to the proprietary nature of these data, the complete test matrix with all of the off-design flow conditions cannot be published here. It is sufficient to note that the heat exchanger was tested at a variety of off-design conditions mainly by varying the CO_2 massflow, inlet pressure, and inlet temperature. The CO_2 outlet temperatures from several of these off-design cases are shown in Figure 1. Along with the experimental results, the predicted off-design performance using our method for various values of $hA_{ratio}$is also shown.

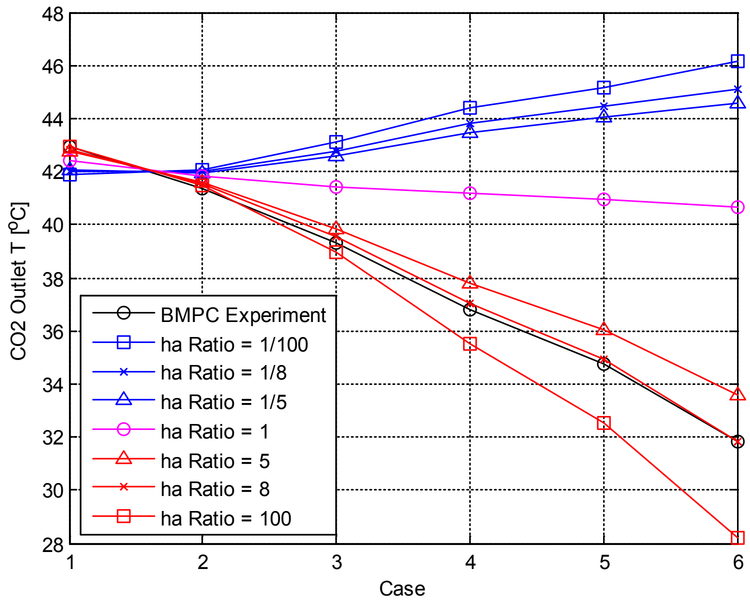

Figure 1: CO2 outlet temperature for several off-design cases of the sCO2/water heat exchanger from BMPC along with predicted performance using our method for various values for hA_{ratio}.

As can be seen in the figure, the off-design performance is well predicted for these cases using an hA_{ratio} of 8. The dependence of the results on hA_{ratio} is also shown in Figure 2. In this figure, the outlet temperature for case 6 from Figure 1 is shown as a function of hA_{ratio}.

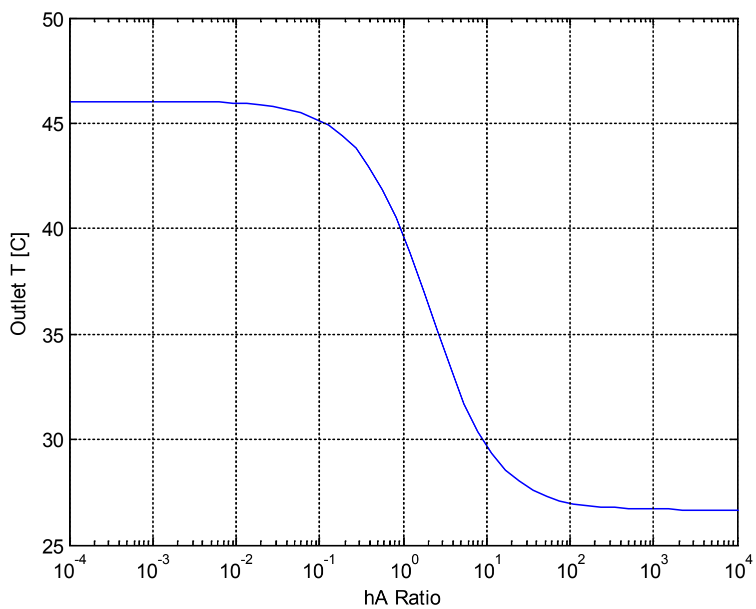

Figure 2: Hot CO2 outlet temperature from our method for case 6 as a function of hA_{ratio}.

Two main conclusions are drawn from Figure 1:

- The off-design performance prediction is very sensitive to hA_{ratio}. The outlet temperature can vary as much as 20ºC depending on the chosen value of hA_{ratio} is used.
- When the appropriate $hA_{ratio}$is selected ($hA_{ratio} = 8$), the experimental and predicted outlet temperatures match very well.

This is confirmed by Figure 2 where the dependence of outlet temperature for a single case on hA_{ratio} is shown and is evident from the figure that the outlet temperature varies between two extremes. A high value of hA_{ratio} would indicate a heat exchanger with a very high value of hA_{hot} relative to the value of hA_{cold}. In this type of heat exchanger, the resistance provided by the hot term to the overall heat transfer coefficient would be low enough that the heat exchanger would be effectively dominated by the value of hA_{cold} and its associated resistance. The opposite is true for a heat exchanger with a very low value of hA_{ratio}. Thus, a chart like the one shown in Figure 2 represents the entire range of possible off-design performance values for the off-design conditions described by case 6. Each different value of hA_{ratio} could be thought of as representing a different heat exchanger configuration that would yield the same on-design performance but different off-design performance as shown in the figure.

Using Figure 1 as a guide, an hA_{ratio} of 8 was chosen as the most appropriate for the design condition of BMPC heat exchanger. When using our method as intended during the cycle analysis phase, this type of calibration would not be necessary. The BMPC data only represents one possible set of off-design results for the collection of heat exchangers that could be designed to meet the required on-design operating conditions. During the cycle analysis phase, the entire possible range of off-design performance could be explored by varying the value of hA_{ratio}.

To be useful, our method needs to be able to predict off-design heat exchanger performance at a wide range of flow conditions using a single value of hA_{ratio}. To verify this, the CO_2 outlet temperatures for all of the off-design points presented in the BMPC study were computed using an hA_{ratio} of 8. In Figure 3, these values are plotted against the experimental off-design CO_2 outlet temperature for each corresponding case. As can be seen in the figure, overall, the experimental and predicted CO_2 outlet temperatures match very well. It should be noted that all of these off-design performance values were calculated using the design conditions specified in Table 1 and an hA_{ratio} of 8.

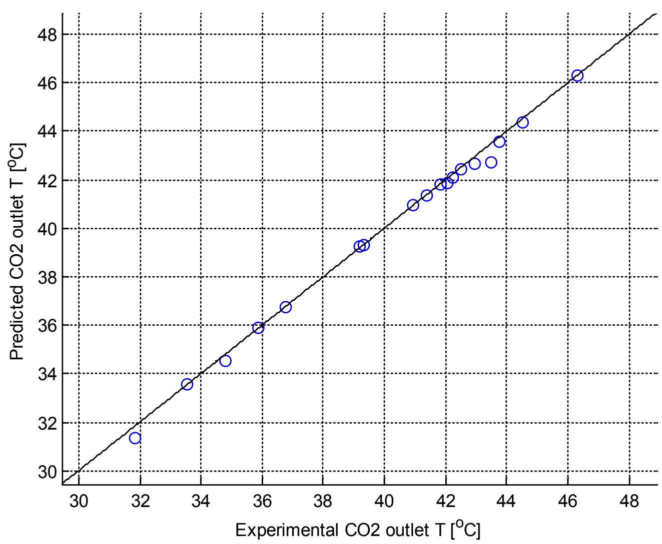

Figure 3: Experimental CO2 outlet temperatures from the BMPC study along with predicted CO2 outlet temperature using out method for an hA_{ratio} of 8.

In Figure 3 the same test condition and value of hA_{ratio} was used to predict the sCO2 outlet conditions of all the other conditions in the test matrix. In order to test the flexibility of our method, each of the conditions in the test matrix was also used in an attempt to predict the sCO2 outlet temperature of every other condition in the test matrix. The results from this study are shown in Figure 4.

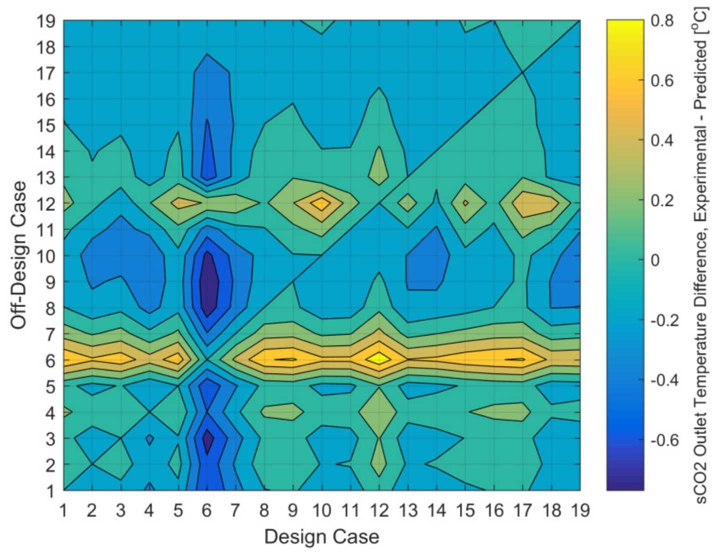

Figure 4: Contours of sCO2 outlet temperature difference (experimental – predicted) for different combinations of on and off-design conditions.

In this figure, the x-axis represents the condition used as the design case while the y-axis shows the condition in the test matrix used as the off-design case for prediction. For each of the design conditions, a new value for hA_{ratio} was chosen that minimized the RMS error of the experimental versus predicted sCO2 outlet temperature. These new values of hA_{ratio} tailored to each condition in the test matrix ranged between 2 to 15. As can be seen in the figure, some cases are more difficult to predict than others, such as cases 6 and 12. Interestingly these cases also make relatively poor design conditions for use in predicting the sCO2 outlet temperature resulting from the other conditions in the test matrix. That being said, the overall match between experimental and predicted sCO2 outlet temperature is very good, with the maximum error observed being 0.8 ^oC for any prediction.

PCHE DESIGN CODE PREDICITIONS FOR AN sCO2 RECUPERATOR

In this section the methodology described in the previous section is now applied to an sCO2-sCO2 heat exchanger using an in-house PCHE design tool for comparison. The model is based on the following common approach (Nellis & Klein, 2009): the heat exchanger is divided in N divisions, each one characterized by the same heat duty and ensuring that the temperature rise/drop on each side is small enough for the $\varepsilon - NTU$ method to be applicable in each division[3]. This allows to calculate the length of each division (and hence the entire heat exchanger) once the heat transfer balance is resolved.

One of the interesting features of the code is flexibility. It enables several cross sectional geometries (semi-cylindrical, rectangular) with dissimilar dimensions on each side, and different streamline approaches are also possible (straight, zig-zag, wavy). Flexibility applies to heat transfer correlations as well and several of these are built into the code. It is worth noting though that the code calculates the thermal and hydraulic performance and does not evaluate the thermo-mechanical performance of the PCHE. It is hence possible that the configurations obtained with the in-house code have some structural problems.

[3]The $\varepsilon - NTU$method is based on the assumption that the fluid's properties are constant. Hence, it cannot be applied in a flange to flange simulation or with too few internal divisions, due to the strong real-gas-behavior of sCO2.

In order to evaluate the hA_{ratio} methodology, the recuperator of a well-known sCO2 cycle is evaluated using this method as well as the detailed geometry approach. To this end, the recuperator of a standard Allam cycle is selected (Allam, y otros, 2013), with the main operating conditions summarized in Table 2. It is worth noting that the working fluid is assumed pure CO_2 in lieu of the gas mixture flowing in the Allam cycle. This enables easier property evaluation although it brings about some deviations in the final results. The conclusions of the analysis regarding the validity of the scaling methodology should not be affected by this change.

	T_{in} [ºC]	p_{in} [bar]	T_{out} [ºC]	p_{out} [bar]	$\dot{m}$[kg/s]
sCO2 (hot side)	776.9	30.86	102.9	30.05	290
sCO2 (cold side)	81.9	297.62	624.6	297.48	290

Table 2. Operating conditions of the selected recuperator.

The default geometry considered in the application of the PCHE design code to the operating conditions in Table 2 is presented in Figure 5. This geometry uses a counter-flow layout with semi-cylindrical, zig-zag channels and channel dimensions taking values in the range of standard industrial practice (Dostal, Driscoll, & Hejzlar, March, 2004), (Heatric, 2015).

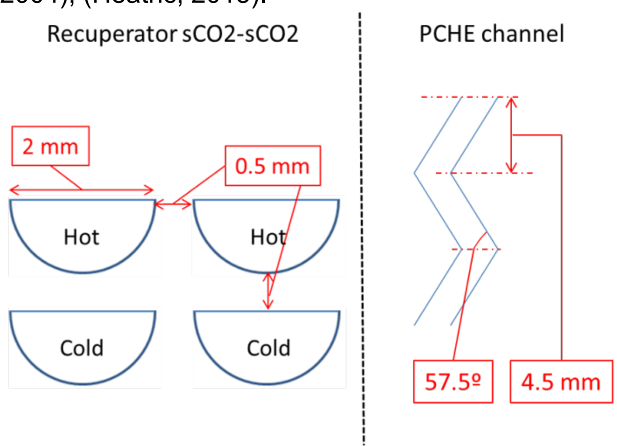

Figure 5: Basic geometry of reference PCHE.

The off-design case studies are summarized in Table 3 along with the outlet temperatures and pressures predicted by the scaling method. The inlet temperatures and pressures remain constant (boundary conditions) whilst the mass flow rates on both sides decrease from the rated value down to 25% of the design condition. It should be noted that as both sides of the heat exchanger experienced the same changes in massflow rate, the results are not very sensitive to hA_{ratio}.

		PCHE model				Simplified approach			
$\dot{m}_{CO2}$ $[kg/s]$	Q $[kW]$	$T_{out,h}$ [ºC]	$T_{out,c}$ [ºC]	$P_{out,h}$ $[bar]$	$P_{out,c}$ $[bar]$	$T_{out,h}$ [ºC]	$T_{out,c}$ [ºC]	$P_{out,h}$ $[bar]$	$P_{out,c}$ $[bar]$
58.0	44599	102.9	624.6	30.05	297.48	102.9	624.5	30.05	297.48
55.1	42417	102.1	625.3	30.12	297.49	102.3	625.1	30.14	297.49
52.2	40230	101.3	625.9	30.19	297.50	101.6	625.7	30.21	297.51
49.3	38038	100.5	626.6	30.25	297.51	100.9	626.3	30.28	297.52
46.4	35841	99.6	627.3	30.31	297.51	100.2	626.9	30.35	297.53
43.5	33639	98.8	628.0	30.37	297.52	99.4	627.5	30.41	297.54
40.6	31432	98.0	628.7	30.42	297.53	98.6	628.2	30.48	297.55
37.7	29220	97.2	629.4	30.48	297.54	97.8	628.9	30.53	297.56
34.8	27002	96.4	630.0	30.53	297.55	96.9	629.6	30.58	297.57
31.9	24778	95.6	630.7	30.57	297.55	96.0	630.4	30.63	297.58
29,0	22549	94.9	631.3	30.61	297.56	95.1	631.2	30.67	297.59
26.1	20313	94.2	631.9	30.65	297.57	94.1	632.0	30.71	297.59
23.2	18071	93.6	632.4	30.69	297.58	93.0	632.9	30.74	297.60
20.3	15822	93.1	632.8	30.72	297.58	91.9	633.8	30.77	297.60
17.4	13565	93.0	632.9	30.75	297.59	90.7	634.8	30.79	297.61
14.5	11297	93.5	632.5	30.78	297.59	89.4	635.9	30.81	297.61

Table 3. Part load performance comparison of a sCO2 recuperator using the simplified and detailed geometry approaches.

Graphical results are presented in Figure 6 and Figure 7 for the outlet temperatures on both sides of the recuperator. A very good match between the PCHE model and the simplified approach is observed for loads down to 50%. The deviation is rather small for loads between 40% and 50% and below this there is an evident mismatch between both models.

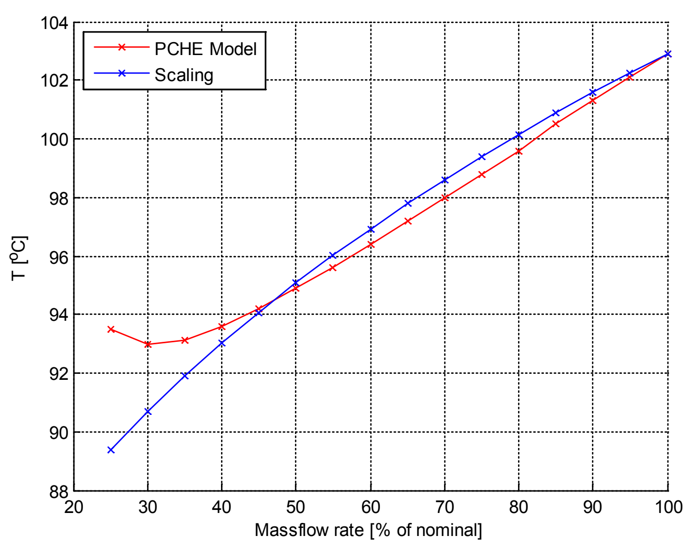

Figure 6: Predicted outlet temperature from the hot (low pressure) side.

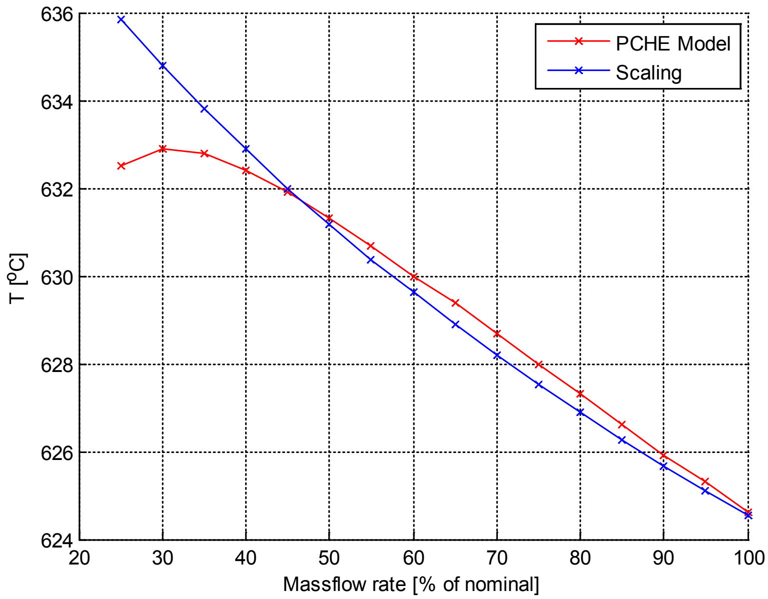

Figure 7: Predicted outlet temperature from the cold (high pressure) side.

A closer look into the results of the PCHE model reveals that the root cause for the deviation observed at loads lower than 40% is the flow becoming laminar. In effect, at very low loads the velocity of the flow decreases substantially and so does Reynolds number. At about 40% load, local Reynolds numbers decrease to values lower than 3000 which is below the assumed critical value (Nellis & Klein, 2009). This is easily observed in Figure 8 where Re on the hot and cold sides is plotted for 25% and 100% load.

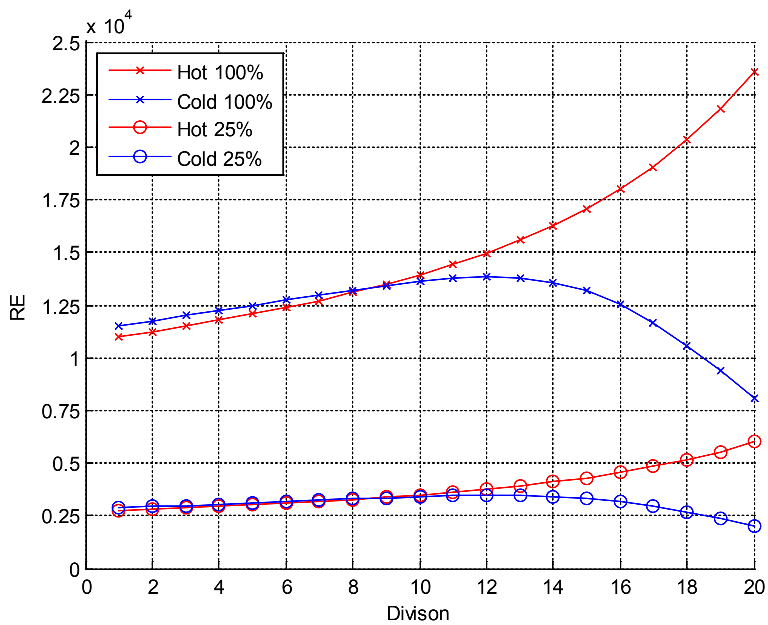

Figure 8: Average Reynolds number on both sides as predicted by the PCHE model at various load settings. In this figure, the hot side of the heat exchanger flows from left to right while the cold side of the heat exchanger flows right to left.

At full load, the Reynolds number is higher than 10^4 in all locations in the heat exchanger, in particular on the hot side given the lower density brought about by a lower pressure and higher temperature. Nevertheless, at 25% load, low values of Re in the entry region of both sides bring about laminar flow locally, hence lower heat transfer coefficients (h) and higher thermal resistances ($1/hA$). These hinder heat transfer locally and bring about a higher temperature gradient at both ends of the heat exchanger, as observed in Figure 6 and Figure 7.

It should be noted that the larger temperature difference at both ends of the equipment is not due to laminar flow on both sides but just to low heat transfer coefficients on one side. In other words, the pattern exhibited on both sides is an increasing Re from inlet to roughly 50% of the HX length which later falls to values below critical. These latter values are not likely to bring about laminar flow as the flow is influenced by the upstream conditions.

A similar effect is found at the other end. Our scaling method is not sensitive to this behavior as it relies on energy conservation equations and a constant scaling law without any knowledge of the geometry. As discussed previously our scaling method, makes use of exponents based on commonly used heat transfer correlations for turbulent pipe flow. Nevertheless, this correlation is for internal forced turbulent flow only and hence it does not apply there were laminar flow is found (or to external flows).

It would be difficult for our scaling method to predict where laminar flow occurs as it is not aware of the absolute value of Reynolds number, only its scale relative to the on-design conditions. The corresponding scaling process is thus affected by this error in the load region where the flow is laminar on either side of the heat exchanger[4]. It would be possible for the scaling law to be modified to assume constant laminar flow, but for the same reasons as outlined above, if the flow were to become turbulent, similar discrepancies in outlet temperature would arise.

CONCLUSIONS & FUTURE WORK

The introduction of this paper presented the inherent difficulties found by cycle analysts when assessing the off-design performance of power cycles, especially when considering sCO2 due to the highly recuperative nature of these cycles. For off-design heat exchanger performance calculations, the investigator was actually left with few options, mainly two: either producing a draft geometry of a heat exchanger which could be used for off-design modelling, or considering constant UA or temperature difference. Both of these methods carry with them assumptions that detract from their accuracy as discussed previously.

In order to overcome these difficulties, this paper has outlined a new method to calculate heat exchanger off-design performance without detailed geometry design. It is conceptually (computationally) simple and relies on a combination of energy conservation and hA corrections for Reynolds, Prandtl and thermal conductivity variations (thermal resistance scaling). A comparison against experimental sCO2 heat exchanger performance results as well as against the results from a numerical design tool for the thermal design of sCO2 heat exchangers have confirmed that the approach is valid and can be used to predict off-design performance of a heat exchanger of unknown geometry, as well as investigate possible off-design performance scenarios.

From a practical point of view, the interest of this method must not be overlooked. Thanks to this simple, geometry-independent and accurate approach, the investigator analyzing potential thermodynamic cycles can not only estimate off-design heat exchanger performance without knowing anything about its actual geometry, but evaluate all possible off-design scenarios. The results from this type of evaluation can then be used to specify the off-design performance of a heat exchanger by setting the desired value of hA_{ratio} when doing detailed heat exchanger design.

Various steps are foreseen to continue developing this concept. These can be summarized as follows:

- To explore means to account for laminar flow occurrences at low loads, hence extending the range of application of the methodology.

[4]More information about correlations to evaluate Nu in laminar flow can be found in (Kakac, et al., 1987),

- To explore geometrical or mechanical limitations to the range of hA_{ratio}, providing typical values for various applications and heat exchanger types.
- To evaluate the impact of the inaccuracies brought about by standard assumptions (constant UA and draft geometry) on cycle performance and to assess how this can be reduced by the proposed approach.
- Implementing a method to account for heat exchanger thermal mass to allow for time accurate calculation of heat exchanger off-design performance.

NOMENCLATURE (except if noted in the text)

k	=	Ratio of specific heats (c_p/c_v)
ψ_{rec}	=	Recuperative potential
$T_{T,in}$	=	Turbine inlet temperature
$T_{T,out}$	=	Turbine exhaust temperature
$T_{C,in}$	=	Compressor inlet temperature
$T_{C,out}$	=	Compressor delivery temperature
φ	=	Thermo-physical property of the working fluid
Q	=	Heat exchanged
U	=	Overall heat transfer coefficient
A	=	Heat transfer area
ΔT	=	Representative temperature difference
$\dot{m}$	=	Massflow
ρ	=	Fluid density
μ	=	Fluid viscosity
λ	=	Fluid thermal conductivity
Re	=	Fluid Reynolds number
Pr	=	Fluid Prandtl number
Nu	=	Fluid Nusselt number

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