sCO_2 Cooling Performance in Turbomachinery

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Abstract
Current designs for sCO_2 power cycles use indirect heating to achieve turbine inlet temperatures of 775°C, which requires the use of advanced nickel alloys in the turbine. Because cycle efficiency is closely tied to the turbine inlet temperature, higher turbine inlet temperatures will be needed in the near future. In fact, temperatures as high as 1,200°C are already being achieved using oxy-combustion technology that is expected to be used in sCO_2 power cycles. To withstand higher inlet temperatures, the simplest solution is to integrate internal cooling into the sCO_2 turbine similar to conventional gas turbines in power generation and propulsion applications. Because the cooling fluid would be sCO_2 instead of air, the real gas effects must be included in the cooling design. In this paper, the real gas effects of sCO_2 are discussed using a cooling effectiveness chart, which is commonly employed to evaluate cooling designs in air-cooled gas turbines.

Introduction
Supercritical Carbon Dioxide (sCO_2) power cycles can provide higher cycle efficiencies than state-of-the-art air-Brayton cycles. However, turbomachinery technology for sCO_2 is still catching up to the air-Brayton cycle, which is operating at large commercial scales and firing temperatures above 1,300°C. To make sCO_2 power cycles competitive at commercial scale, air-Brayton turbomachinery technologies must be applied to sCO_2 turbomachinery while accommodating the challenges of sCO_2.

Currently, sCO_2 turbomachinery is being used for indirect-fired cycles with turbine inlet temperatures limited to 775°C, due to limits in the high-temperature recuperator material. Increasing the turbine inlet temperature will result in higher cycle efficiencies, but requires a direct-fired cycle, such as an oxy-fuel combustion cycle. Therefore, sCO_2 turbomachinery technology development is needed to design an expansion turbine that can operate at oxy-fuel combustion temperatures (1,200°C) while limiting the turbine exhaust temperature to current limits of high-temperature recuperators (775°C). Increasing the maximum temperature for a sCO_2 power cycle from 775°C to 1,200°C will increase the Carnot efficiency by 2%. While Carnot efficiency gains are not a guarantee of actual cycle efficiency, it can be expected that careful thermal management and cycle design can achieve a proportionate increase in actual cycle efficiency over the current state-of-the-art conditions. Therefore, an efficiency increase of a few percentage points will be helpful to achieve a significant reduction in the cost of electricity

One approach to managing the higher turbine inlet temperature is to use low-temperature sCO_2 in cooling passages to reduce material temperatures, giving the stressed components higher strength. This could also enable a higher-pressure ratio in the turbine, which would lower the turbine exhaust temperature. Careful design of the thermal management system would allow the recuperation of heat gained by the cool sCO_2 for use in the cycle. This approach would provide an improvement to current cooling technologies that do not utilize coolant flows for cycle recuperation.

Variation of Thermophysical Fluid Properties
Supercritical CO_2 has very high heat capacity near the critical point, which allows a lot of energy to be added to sCO_2 near the critical point without resulting in a high temperature rise. This is one feature of sCO_2 that makes it attractive for a power cycle because sCO_2 can be compressed without a large change in fluid temperature, resulting in a more efficient compression process. In regards to hot-section cooling, the high heat capacity of sCO_2 near the critical point can be useful to improve the effectiveness of

cooling designs, as shown in Figure 1. In this way, less cooling mass flow would be required to extract heat from the metal in the hot-section.

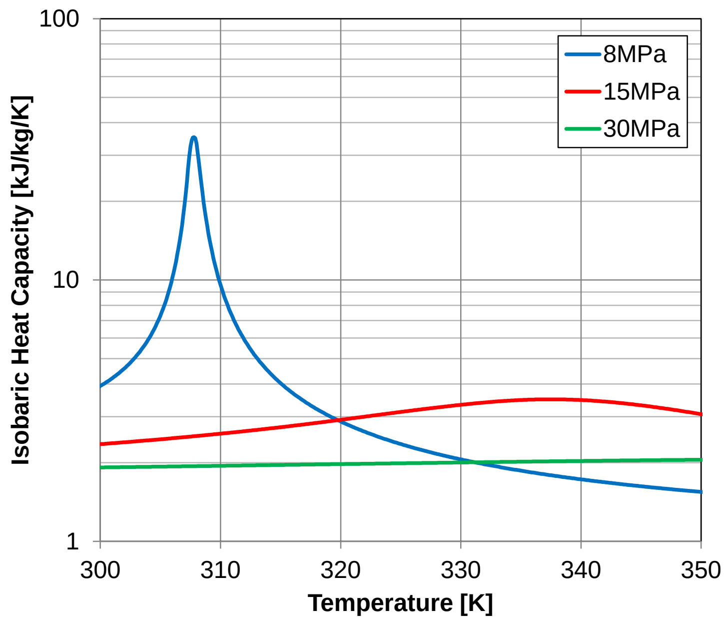

Figure 1.Variation of heat capacity of CO_2

Cooling Effectiveness and Heat Load Parameter

A cooling effectiveness chart is a common method to evaluate cooling designs in turbomachinery [1,2]. In this way, the external metal temperature of the component and the outlet temperature of the coolant can be used to evaluate the cooling design. The equations of cooling effectiveness (ϕ), heat load parameter (HLP), and thermal efficiency (η_{th}) are listed below. These equations are derived by a one-dimensional heat balance across a flat plate. In the case of turbomachinery, the thickness of an airfoil wall can be approximated as the flat plate. In the derivation, the following assumptions are made:

- Metal temperature (T_m) refers to the temperature on the gas path side of the plate
- Thermal conduction resistance of the plate is negligible compared to the convective resistances
- The gas path temperature (T_g) is assumed to be constant
- The heat capacity of the fluid is identical between the coolant and the hot gas path

$$\phi = \frac{T_g - T_m}{T_g - T_{c,in}} \qquad HLP = \frac{T_g - T_m}{T_{c,out} - T_{c,in}} \qquad \eta_{th} = \frac{T_{c,out} - T_{c,in}}{T_m - T_{c,in}}$$

Cooling effectiveness ***Heat Load Parameter*** ***Thermal Efficiency***

Utilizing the equations, Figure 2 can be developed to indicate that increasing the heat load capacity (HLP) and the thermal efficiency of the cooling design will result in a high cool effectiveness. As the cooling effectiveness increases, the metal temperature approaches the coolant outlet temperature ($T_{c,o}$). The significance of this chart is that cooling designs can be evaluated easily by plotting the ϕ and HLP values. This can be useful during conceptual design to determine the cooling technology required to reduce the component metal temperature within structural requirements. For example, a serpentine cooling passage and an impingement cooling design may produce the same cooling effectiveness. However, the serpentine passage will have a higher thermal efficiency and require less cooling air (lower

HLP) than the impingement cooling design. The serpentine cooling design would appear on the left side of Figure 2 while the impingement design would appear on the right side.

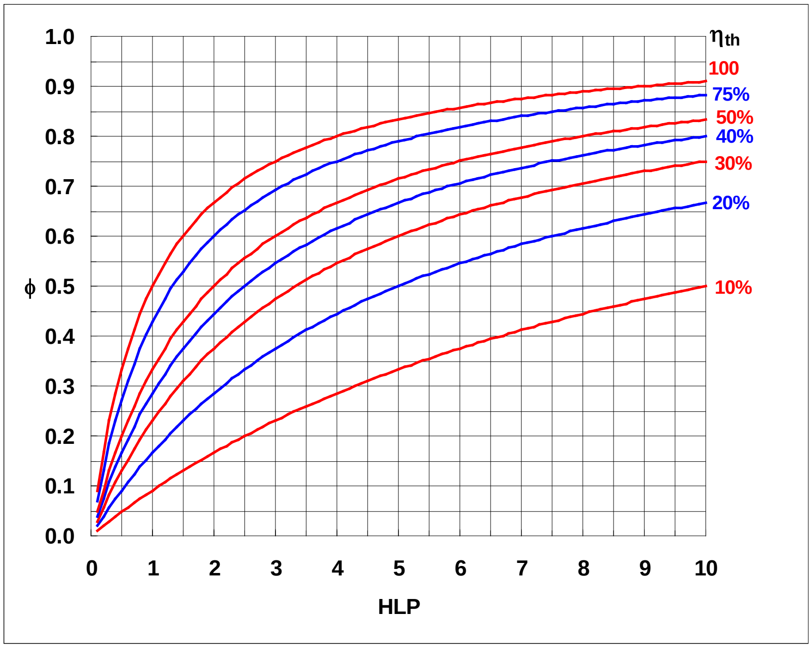

Figure 2.Cooling effectiveness (ϕ) and heat load parameter (HLP) trends for constant specific heat

In regards to sCO_2 cooling systems, the cooling effectiveness chart requires modification because the heat capacity of the hot gas path and the cooling fluid are not identical. In fact, the heat capacity of the cooling fluid could vary significantly along the cooling passage. Therefore, the simplified equations must be adjusted to utilize enthalpy (h) values instead of approximating enthalpy as the product of specific heat and temperature (C_pT). This is especially important if the cooling fluid is being supplied near the critical point and not at a higher temperature, such as from a re-compressor in the sCO_2 power cycle.

To generate the cooling effectiveness chart for sCO_2, the cooling fluid outlet temperature should be calculated from the fluid enthalpy. To be specific, thermal efficiency lines are generated by defining a gas path temperature, cooling inlet temperature, operating pressure, ϕ value, and thermal efficiency. In this way, the HLP value can be calculated and compared between an air gas turbine and an sCO_2 expander. In both example cases, the gas path and cooling inlet temperature are defined as 1,200°C and 50°C, respectively. The pressure of the coolant supply is 30MPa for the sCO_2 expander case while the coolant supply pressure is 1.5MPa for the air gas turbine case.

In calculating the cooling effectiveness values, the sCO_2 cooling curve at 40% thermal efficiency is below the air cooling curve at the same efficiency, as shown in Figure 3. This could be thought to indicate that for the same heat load parameter, a higher metal temperature would be achieved in the sCO_2 design. However, the high heat capacity of the sCO_2 fluid should be considered, which suggests that a higher HLP value is obtained because the sCO_2 has a lower coolant outlet temperature, as shown in Figure 4. This indicates that a higher thermal efficiency could be obtained with sCO_2 than air because less sCO_2 cooling flow would be needed. This is shown in Figure 3 where the 50% thermal efficiency curve for sCO_2 is nearly the same as 40% thermal efficiency for air. Furthermore, the benefit of sCO_2 as a cooling fluid diminishes as the thermal efficiency increases. Because internal convective cooling is generally in the 40% efficiency range, internal cooling with sCO_2 could be expected to be more efficient than air.

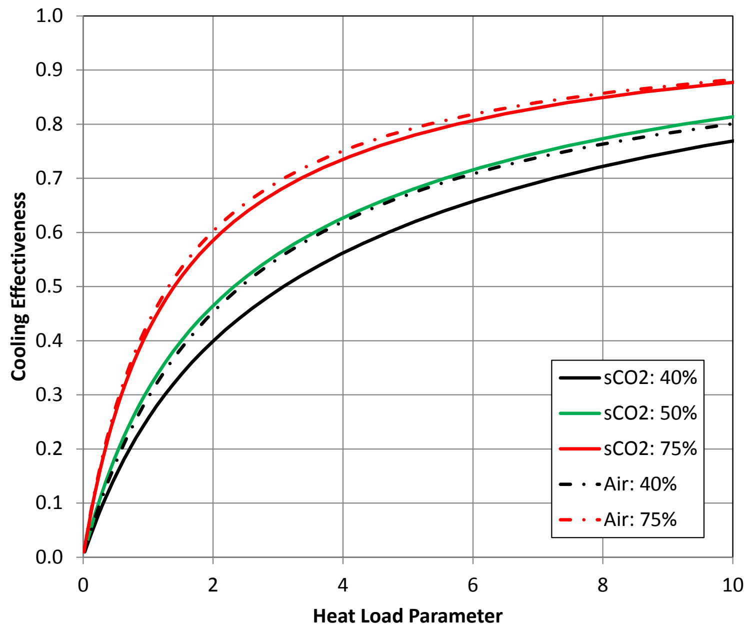

Figure 3.Cooling effectiveness differences in Air and sCO_2.

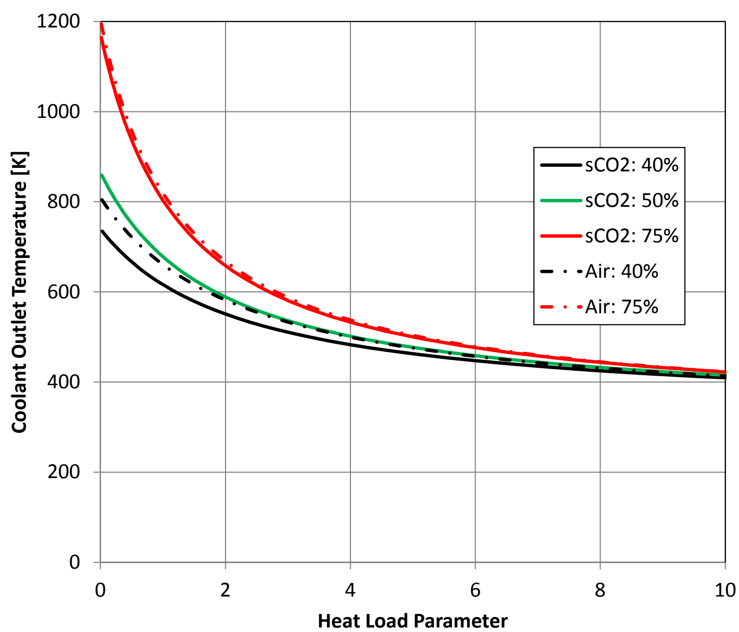

Figure 4.The sCO2 fluid provides better cooling than air because it has a lower coolant outlet temperature

Due to the variation of sCO_2 properties, the coolant supply pressure provides a method to modify the effectiveness of the cooling design. For example, reducing the coolant supply pressure from 30MPa to 15MPa results in a significant effect on the metal temperature (cooling effectiveness), as shown in Figure 5. Similar to what has been discussed previously, a lower cooling effectiveness is observed for the lower coolant pressure because the coolant has a greater thermal capacity, as evident by the lower coolant outlet temperature. As coolant pressure is reduced from 15MPa, less thermal capacity is available for the 50°C (323K) coolant, which results in increased coolant outlet temperature, as shown in Figure 1.

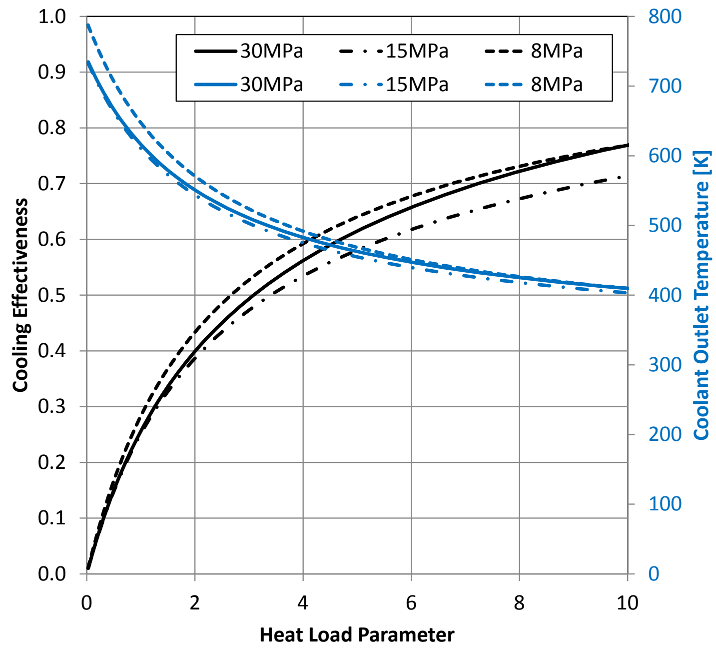

Figure 5. Reducing cooling flow pressure can have a significant impact on overall cooling effectiveness for sCO_2

Summary and Conclusions

In air-cooled gas turbines, cooling technology is commonly evaluated using cooling effectiveness and HLP. When expanding this approach to sCO_2-cooled turbomachinery, real gas effects of CO_2 must be considered. Real gas effects are quantified in the thermal capacity of sCO_2, where the coolant supply pressure and temperature have a significant effect on the cooling effectiveness and thermal efficiency of a given cooling technology.

In comparing cooling effectiveness curves of sCO_2 to air at similar coolant and gas path temperatures, the use of sCO_2 results in a lower cooling effectiveness. This is because of the higher thermal capacity of sCO_2, which results in a lower coolant outlet temperature than air. The significance of a lower coolant outlet temperature indicates that the sCO_2 cooling design could benefit from additional cooling passages in the component to increase the thermal efficiency of the cooling design. Increasing thermal efficiency with sCO_2 could be done by increasing the number of serpentine passes inside a component, such as a blade. Because of the high density of sCO_2, this may be possible because low pressure drop could be achievable.

The pressure of the coolant flow is significant in adjusting the cooling performance. If conventional cooling designs are implemented with sCO_2, the coolant supply pressure will be required to be greater than the gas path pressure to allow the sCO_2 to purge from the component. However, sCO_2 cooling designs could utilize a closed-circuit and recover the heat gained from the coolant. In this way, coolant pressure could be selected for increasing cooling performance instead of maintaining positive purge.

As sCO_2 turbomachinery continues to develop for higher operating temperatures, the natural progression of turbine cooling will apply conventional cooling designs with sCO_2 as the cooling fluid. In doing so, real gas effects will need to be considered. Like the power cycle itself, real gas effects of CO_2 should be used to enhance the cooling performance whenever possible.

Grant Musgrove is Senior Research Engineer and Aero/Mechanical Coordinator in the Machinery Program at Southwest Research Institute. He currently conducts applied research for heat exchanger and turbomachinery applications in oil and gas, power generation, and propulsion applications. Mr. Musgrove has been involved with the technical development of both compressors and heat exchangers for sCO_2 applications. Mr. Musgrove earned his B.S. and M.S. in Mechanical Engineering from Oklahoma State University and from The Pennsylvania State University.