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THE CO_2 TRANSCRITICAL POWER CYCLE FOR LOW GRADE HEAT RECOVERY-DISCUSSION ON TEMPERATURE PROFILES IN SYSTEM HEAT EXCHANGERS

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ABSTRACT
Carbon dioxide transcritical power cycle has many advantages in low-grade heat source recovery compared to conventional systems with other working fluids. This is mainly due to the supercritical CO_2's temperature profile can match the heat source temperature profile better than other pure working fluids and its heat transfer performance is better than the fluid mixtures, which enables a better cycle efficiency. Moreover, the specific heat of supercritical CO_2 will have sharp variations in the region close to its critical point, which will create a concave shape temperature profile in the heat exchanger that used for recovering heat from low-grade heat sources. This brings more advantage to carbon dioxide transcritical power systems in low-grade heat recovery.

This study discusses the advantage of carbon dioxide power system in low-grade heat source recovery by taking this effect into account. A basic carbon dioxide transcritical power system with an Internal Heat Exchanger (IHX)[1] is employed for the analysis and the system performance is also compared with a basic Organic Rankin Cycle (ORC).

Software Engineering Equation Solver[2] (EES) and Refprop 7.0[3] are used for the cycle efficiency and working fluid properties calculations.

Key words: pinching, specific heat (CP), internal heat exchanger (IHX), efficiency

[1] Sometimes also called regenerator in power systems
[2] Engineering Equation Solver: http://www.fchart.com/ees/ees.shtml
[3] Refprop 7.0: http://www.nist.gov/srd/nist23.htm

INTRODUCTION

Most ongoing research projects in the field of low-grade heat source utilization are focusing on so-called Organic Rankine Cycles (ORC) and Kalina cycles (binary fluids and fluid mixtures) [1-8].However, the drawbacks of these cycles are numerous: for ORC, working fluids such as R113 and R123 are expensive and have strong climate impacts [9]. Furthermore, the constant boiling temperature in a typical ORC will create so-called "pinching"[4] in the heat exchanger and thus limit the heat exchanger and cycles' performances. For Kalina cycles, the heat transfer characteristics are always poorer for the fluid mixtures than for the pure working fluids. Moreover, ammonia, which is one of the main working fluids in Kalina cycles, is highly toxic and corrosive [10].

Compared to these working fluids, carbon dioxide (CO_2) has many advantages to be used as a working fluid in power cycles. Carbon dioxide is inexpensive, non-explosive, non-flammable and abundant in the nature. Besides, it has no ozone depleting potential (ODP) and negligible global warming potential (GWP). As a pure working fluid, it also has better heat transfer characteristics than fluid mixtures. Furthermore, the carbon dioxide system is also more compact than the systems using other working fluids, owing to its relatively high working pressure and specific power density. Due to the characteristics of carbon dioxide's critical point (7.38 Mpa / 1070.38psi, 31.1°C / 87.98°F), a carbon dioxide power cycle will work as a transcritical cycle a Brayton cycle, depending upon whether the cycle is partly or totally located in the supercritical region.

[4] Pinching is the minimum temperature difference inside a heat exchanger , which limited the heat exchanger performance

The CO_2 power cycle was firstly proposed by Sulzer Bros in 1948 and then researchers from several countries, such as Soviet Union and United States, were involved, mainly for the applications in nuclear power plants. After the interest in 1960s', the research on such a cycle, however, perished for many years mainly for the reasons as limited amount of suitable (e.g. nuclear) heat sources, limited knowledge in suitable compact heat exchangers or insufficient experience in suitable expansion machines [11]. However, since the late of 1990's, the research on carbon dioxide power cycle has regained its interest. Since then, several research institutes have been involved in such a research [12-13]. Nevertheless, all of these investigations are focusing on a carbon dioxide power cycle with a nuclear reactor as a heat source, thus the cycle is working with high-grade heat source (up to 800 °C) and the research on such a cycle in low-grade heat source utilizations are relatively limited [14-17].

Supercritical carbon dioxide shows an inclined temperature profile, which matches better the temperature profile of the low-grade heat source in the heat exchanger than other working fluids used in conventional ORCs. Thanks to its low critical temperature, it is easier for carbon dioxide to work in the supercritical region than other conventional working fluids, when utilizing the energy in low-grade heat sources. Furthermore, since the heating process of supercritical CO_2 takes place in the supercritical region, the complexities such as flow maldistribution that involved by phase changing can therefore be avoided. Meanwhile, the thermophysical properties of supercritical CO_2 will have sharp variations in the region close to its critical point, which will significantly influence the heat transfer features in the heat exchangers [18-21]. It is important to take this effect into account, when one evaluating the performances of CO_2 transcritical power cycles and its heat exchangers

DESCRIPTION OF THE SYSTEM AND THE CYCLE

The schematic layout of a carbon dioxide power system is shown in Fig 1. A carbon dioxide power system is mainly composed of four main components: a pump, a gas heater, an expansion machine5 and a condenser (gas cooler for Brayton cycle). To improve the cycle efficiency, an IHX can also be integrated in the system.

[5] Turbine or expander coupled to a generator to generate electric power

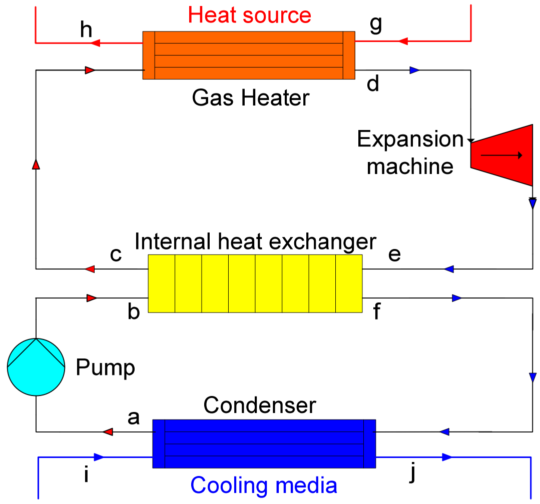


Fig. 1 Schematic of carbon dioxide power system

Due to the low available temperature of low-grade heat sources, CO_2 transcritical power cycle will be more promising to be adopted than CO_2 Brayton cycle, due to its relative low system pressure and high expansion ratio. Therefore, current study is focusing on the CO_2 transcritical power cycle only and its schematic T-S chart is shown in Fig. 2.

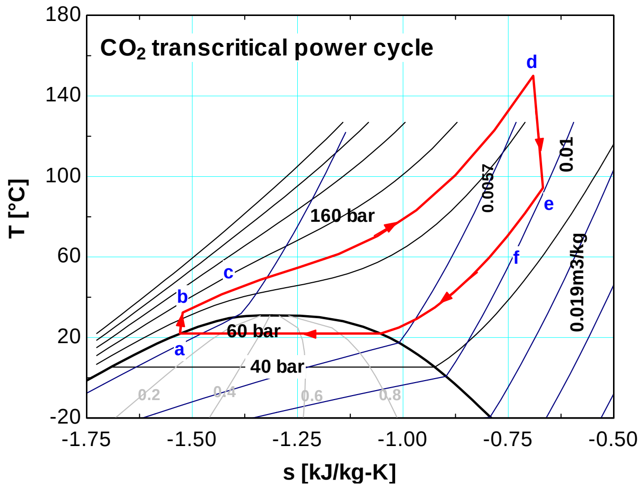


Fig. 2 Carbon dioxide transcritical power cycle T-S chart

THE SPECIFIC HEAT VARIATIONS IN CARBON DIOXIDE TRANSCRITICAL POWER CYCLE

The thermophysical properties of supercritical CO_2 will have sharp variations in the region close to its critical point, which is also the working region of the transcritical carbon dioxide power cycle's heat recovering process. Therefore, the thermophysical properties of supercritical CO_2 needs to be carefully examined, when one analyzing the CO_2 transcritical power cycles, due to their significant influences on the performances of both gas heater and the IHX in the heat transfer processes. The specific heat (Cp), which is the main factor that influences the supercritical CO_2's temperature profile in both the heat exchangers, is plotted as a function of the temperature at different pressures in the following figure (Fig.3)

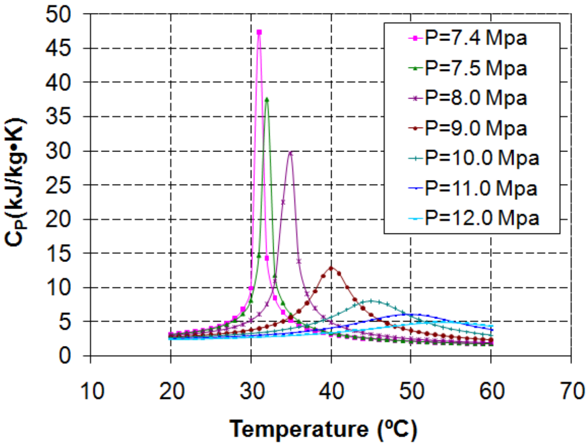


Fig. 3 Specific heat of supercritical CO_2 vs. temperature at different pressures

It can be noticed from the figure that the specific heat of the supercritical CO_2 has more obvious changing, when the pressure gets close to its critical pressure. Furthermore, it may also be noticed that the temperature corresponding to the peak value of specific heat is increasing with increasing pressure.

The variation of carbon dioxide's specific heat at the expansion outlet is however relatively modest. The specific heat of CO_2 at the expansion outlet (at different pressures) is plotted as a function of temperature in Fig. 4.

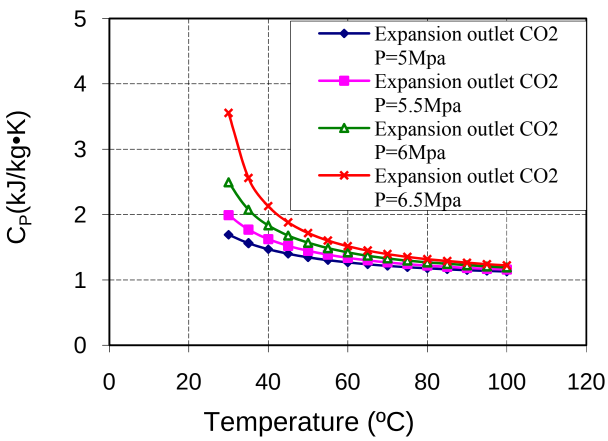


Fig. 4 Specific heat of exhaust gas and expansion outlet carbon dioxide (notice the scale difference from Fig. 3)

From Fig.3 and Fig.4, it may be seen that the specific heat value of the expansion outlet carbon dioxide is almost "constant", compared to the dramatic changing of the supercritical CO_2's specific heat value near its critical point. Due to this difference in the changing of specific heat values, the shape of the temperature profiles in the heat exchangers are greatly influenced, which should be carefully examined when evaluating the performances of the heat exchangers for carbon dioxide transcritical power cycles.

THE INFLUENCE OF THE SUPERCRITICAL CO_2'S SPECIFIC HEAT VARIATION ON THE HEAT EXCHANGER PERFORMANCE

A basic carbon dioxide power cycle can be adopted to analyze the influence of the sharp variations of supercritical CO_2's specific heat on the performance of supercritical CO_2 power cycles. Several assumptions are needed to analyze the cycle performances, such as compression and expansion efficiencies etc. The research on CO_2 pumps is relatively limited compared with the research on CO_2 compressors. Considering the fact that pump efficiencies are normally higher than compressors, due to the smaller volume change during a pumping process than a gas compressing process, the pump's efficiency is assumed to be 0.8 in the current study [22]. For CO_2 expansion machines, the research is mainly on CO_2 expanders for transcritical refrigeration cycles instead of for power cycles [23-25].The efficiency of CO_2 expander is related to many factors such as the type of expanders and the leakage. For current cycle calculations, the expansion process efficiency is assumed to be 0.7 based on research by Nickl et al. [25].The cycle operation conditions are listed in table 1. Furthermore, the temperatures and pressures of vehicles' exhaust gases are adopted as an example of low-grade heat source in the basic cycle's analysis. The data for exhaust gases are listed in table 2 (the mass flow value is based on data from a typical tested truck diesel engine at 75% load).

Table 1Carbon dioxide transcritical power cycle operating conditions

Items	Value	Unit
Gas heater pressure	80-160	bar
Condenser pressure	60	bar
Expansion inlet temperature	Related to the heat source temp.	°C
Condensing temperature	21.98	°C
Pump efficiency	0.8	
Expansion efficiency	0.7	

**The Cp values of carbon dioxide and the exhaust gas are calculated in Refprop 7 and EES.*

Table 2 Heat source (exhaust gas) data

Items	Value	Unit
Exhaust gas mass flow	0.4	kg/s
Exhaust gas inlet temperature	150	°C

Based on the definition of heat exchanger effectiveness (Equation 1), a gas heater with 90% effectiveness is used to harvest the energy from the low-grade heat source for the proposed basic cycle [27].

$$\varepsilon = \frac{T_{h,i} - T_{h,o}}{T_{h.i} - T_{c.i}} \qquad \text{Equation 1}$$

By potting the T-Δh chart of the gas heater, one may notice that the temperature profile of the supercritical CO2 shows an obvious concave shape along the gas heater, which enables small temperature differences at both ends of the gas heater (Fig.5). Meanwhile, due to the big "gap" in the middle of the gas heater, the temperature difference inside the gas heater is much larger than what appears at its ends. Thanks to this temperature profile, the smallest temperature difference in the heat exchanger will appear at one of the gas heater ends, thus the so-called "pinching" can be avoided inside the gas heater. Moreover, the temperate difference, which is the "driving force" for heat transfer to take place, is much larger inside the gas heater than at its ends. As a result, the heat exchanger area (volume) for the gas heater will be greatly reduced. Therefore, the heat exchanger size will be much smaller for carbon dioxide transcritical power cycle than the power cycle with conventional working fluids (e.g. organic working fluids), if the same temperature difference should be achieved at the heat exchanger ends.

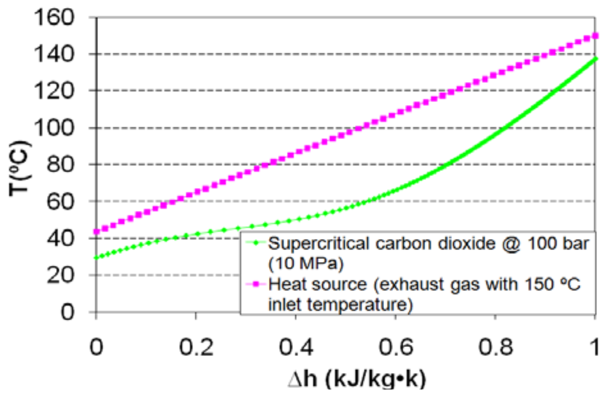


Fig. 5 T-Δh chart for the main heat exchanger (gas heater) of carbon dioxide transcritical power cycle CO_2 transcritical power cycle (mCO2=0.15 kg/s, m exhaust gas=0.4kg/s, without IHX, Gas heater effectiveness=0.9)

In Fig. 6, the temperature profiles of supercritical CO_2 in the gas heater are also plotted for different gas heater pressures under the same operating conditions as above. The result clearly shows that the lower the gas heater pressure is, the more obviously the concave shape of the temperature profile will be. As mentioned above, the concave shape temperature profile will influence the heat exchanger size (volume). For lower gas heater pressures the heat exchanger size will thus become smaller accordingly, if the same temperature difference is sought at the heat exchanger ends.

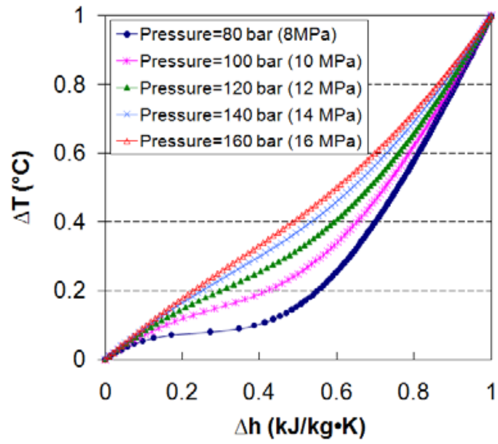


Fig. 6 T-Δh chart for the main heat exchanger of carbon dioxide transcritical power cycle (m_{CO2}=0.15 kg/s, m exhaust gas=0.4 kg/s, without IHX, Gas heater effectiveness=0.9)

Based on the definition of thermal efficiency (Equation 2), the thermal efficiency of the above CO_2 power cycle is plotted for different gas heater pressures and different expansion inlet temperatures (Fig.7).

$$\eta_{th} = \frac{W_{net}}{Q_{hs}} = \frac{W_{\exp} - W_{pump}}{Q_{hs}} \qquad \text{Equation 2}$$

It can be found from the figure that there is an optimum gas heater pressure for a certain cycle working condition. Additionally, it is also showed that for a certain condensing pressure, the optimum gas heater pressure is decreasing with decreasing heat source temperature. Therefore, it shows that the advantages of carbon dioxide transcritical power cycle are more obvious with low-grade heat sources than with high-grade ones, especially if the pressure limitations for the material and the components are considered.

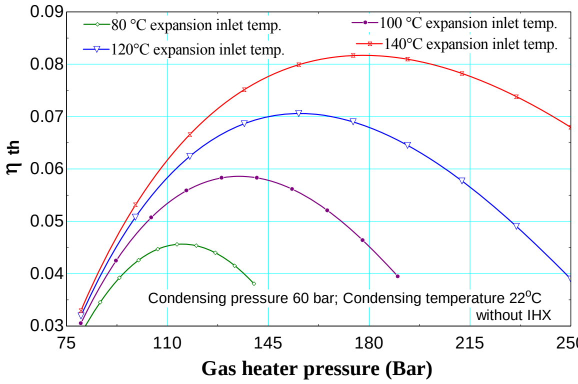


Fig. 7 CO_2 transcritical power cycle thermal efficiency vs. high side pressure (gas heater pressure) at different expansion inlet temperatures (basic cycle without IHX)

INFLUENCE OF INTERNAL HEAT EXCHANGER ON THE CYCLE PERFORMANCE

As shown previously in Fig.2 that CO_2 at the expansion outlet still holds a high energy content (temperature), which can be further recovered to preheat the supercritical CO_2 before it enters the gas heater if an IHX can be integrated into the system. Boewe et al. showed in their research that a CO_2 IHX's effectiveness can reach up to 90% according to different design configurations, which involves a trade-off between the effectiveness of the IHX and the suction-side pressure drop required to achieve it [26]. A counter flow IHX with 90% effectiveness is thus chosen in the current study to show the influence of an IHX on the cycle performance. Employ the same basic cycle that analyzed before, the cycle thermal efficiencies with and without IHX have been calculated for different gas heater pressures. The results show that an IHX can improve the cycle thermal efficiency by up to about 50% (the cycle thermal efficiencies are 6.8 % and 9.2 % respectively for 120 bar gas heater pressure as an example, Fig.8).

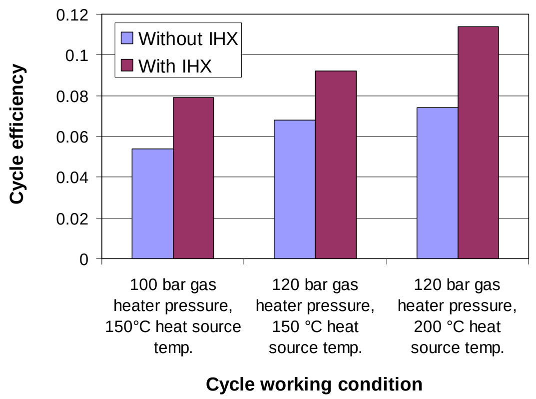


Fig. 8 Efficiency comparison for the cycles that with and without IHX

Besides the advantage of increasing cycle efficiency, inserting an IHX will bring more benefits to the cycle. A Cp-ΔH chart is plotted for the integrated total heat exchanger length, which includes both IHX and the MHX (gas heater[6]) to show the specific heat variations of all the working fluids (i.e. supercritical CO_2, expansion outlet carbon dioxide and the exhaust gas) along the heat exchangers for a certain cycle working condition (Fig.9). It can be noticed from the figure that for the IHX part, the Cp of the incoming supercritical CO_2 (point b) is higher than the Cp of the outgoing expansion outlet carbon dioxide (point f) at one end of the IHX, and this difference is increasing rapidly along the IHX until it reaches a high value at the other end, where the supercritical CO_2 flows out (point c) and the expansion outlet carbon dioxide comes in (point e).

For the gas heater part, a reverse trend is shown. After being preheated in the IHX, supercritical CO_2 enters the gas heater (point c) with a much higher Cp than the outgoing heat source (exhaust gas, point h), then the difference is decreasing rapidly along the gas heater until it reaches a very small value at the other end of the gas heater, where is the inlet of the heat source (exhaust gas, point g) and the outlet of the supercritical CO_2 (point d).

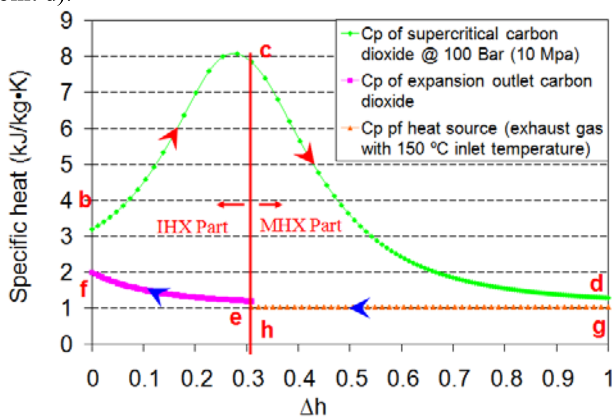


Fig. 9 Cp —Δh chart for supercritical CO_2, expansion outlet carbon dioxide and heat source for the integrated total heat exchanger length

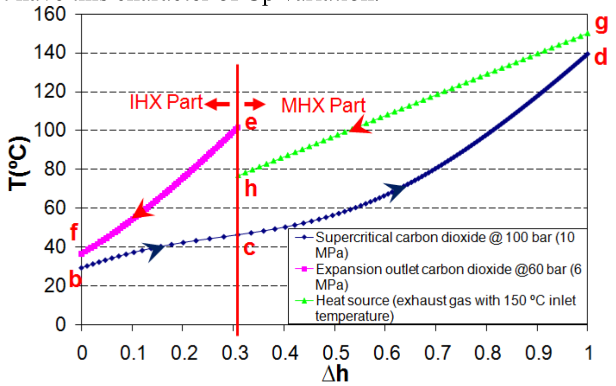
Due to these big differences in specific heats, the temperate profiles in both MHX and IHX will be influenced. T-Δh charts for the integrated total heat exchanger length (MHX and IHX) with different CO_2 mass flow rates are plotted in the following figure to show this influence (Fig.10 a and b)[7]. From the figures, one can see that except a short increase when supercritical CO_2 firstly enters the IHX (point b), the temperature profile of supercritical CO_2 in the IHX is fairly flat due to the sharp increase of its Cp (point b-point c). When it

[6] All the heat exchangers analyzed in this paper are referring to counter flow heat exchangers.

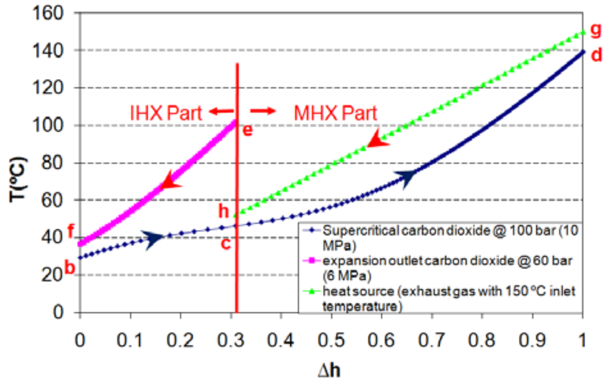
[7] The state points are corresponding to points in figure 2

flows further through the gas heater (point c–point d), the temperature of supercritical CO_2 will increase rapidly due to the sharp decrease of its Cp value (see Fig.9).

This characteristic provides the carbon dioxide transcritical power cycle with more advantages in low-grade heat source recovery than power cycles using other working fluids. Due to the sharp increase of its Cp value in the IHX, the supercritical CO_2 can recover a substantial amount of energy from the expansion outlet (point f to point r in (Fig. 10) without an obvious increase in its temperature (point e). Therefore, after recovering a significant amount of energy from expansion outlet, the supercritical CO_2 has still a huge potential to further recover the energy from low-grade heat source when it enters the gas heater. Therefore, the specific energy content of supercritical CO_2 will be higher and consequently a lower mass flow can be achieved compared to the working fluids that do not have this character of Cp variation.



(a)



(b)

Fig. 10 Integrated heat exchanger's T--ΔH chart of carbon dioxide transcritical power cycle with different mass flow rates of supercritical CO_2 (a): m_{CO2}=0.1 kg/s, m exhaust gas=0.4 kg/s, IHX effectiveness=0.9, MHX effectiveness =0.9 (b): m_{CO2}=0.2 kg/s, m exhaust gas=0.4 kg/s, IHX effectiveness=0.9, MHX effectiveness =0.9

The benefit from supercritical CO_2's Cp variation for the carbon dioxide transcritical power cycle in low-grade heat source recovery can be showed more obviously, if a comparison to other working fluids is made. A T-Δh chart of the integrated total heat exchanger length for an ORC (without superheating) using R123 as a working fluid is plotted for the same heat source condition that used to CO_2 power cycle analysis (Fig. 11). For the R123 ORC, part of the working fluid heating process is taking place in the subcritical region as boiling (i.e. constant temperature) and does not have Cp variation during the heating process as supercritical CO_2 does. It can be seen from the figure that R123 will have more temperature lift in the IHX than supercritical CO_2, due to its relatively linear Cp change. Furthermore, due to its boiling process, pinching will also appear in the MHX, which leads to the consequence that only a very limited amount of energy in the heat source can be utilized.

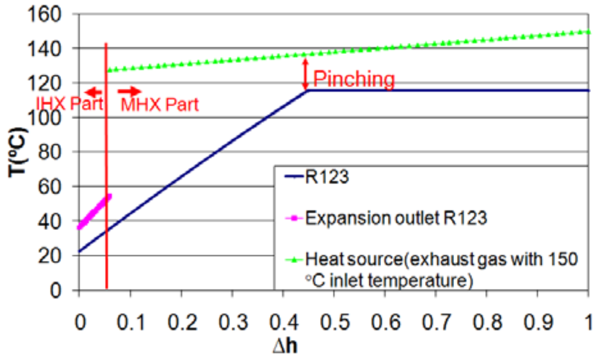


Fig. 11 Integrated heat exchanger's T-Δh chart of R123 ORC. m_{R123}=0.15 kg/s, $m_{exhaust\ gas}$=0.4 kg/s, IHX effectiveness =0.9, MHX effectiveness =0.9 (the process in the figure does not include the superheating of vapor)

CONCLUSION

In the currently study, the sharp Cp variation of supercritical CO_2 in the region close to its critical point is examined. The influence of this variation on the performances of the heat exchangers that used in the carbon dioxide transcritical power systems and the performance of carbon dioxide transcritical power cycle in low-grade heat source utilization are studied and discussed. Furthermore, the influence on the cycle performance by inserting an internal heat exchange is also investigated.

By examining a basic carbon dioxide transcritical power cycle, it is found that due to the dramatic change of the thermophysical properties (Cp) of supercritical CO_2 in the region close to its critical point, the temperature profiles will show a concave shape in the CO_2 system gas heater. Moreover, it is also showed that the concavity will be more obvious at lower gas heater pressures than at higher ones. This enables the system's gas heater to achieve a small temperature difference on both ends and at the same time maintain a small heat exchanger size. Furthermore, the Cp variation also enables supercritical CO_2 to recover the energy both in expansion outlet carbon dioxide and in low-grade heat sources sufficiently.

By examining the carbon dioxide transcritical power cycle's optimum working pressure, it is found that there is an optimum gas heater pressure for a certain cycle working condition and the optimum pressure value is decreasing with the decreasing heat source temperature.

By studying the Internal Heat Exchanger's influence on the basic carbon dioxide transcritical power cycle, it is found that inserting an Internal Heat Exchanger can improve the cycle efficiency by up to about 50% for certain cycle working conditions.

NOMENCLATURE

CO_2	Carbon Dioxide	-
C_P	Specific Heat	kJ/kg·K
EES	Engineering Equation Solver	-
Eff.	Heat exchanger effectiveness	-
GWP	Global Warming Potential	-
IHX	Internal Heat Exchanger	-
MHX	Main Heat Exchanger	
ORC	Organic Rankine Cycle	-
ODP	Ozone Depleting Potential	
Q	heat	kW
Temp.	Temperature	ºC
W	Power	kW

Greek Letters

η_{th}	Cycle thermal efficiency	
Δh	Enthalpy difference	kJ/kg·K
ε	Effectiveness	%

Subscript

a-f	Points of the cycle route
g-j	Points of heat source and heat sink condition
c	Cold
exp	Expansion
h	Hot
hs	Heat source
i	In
o	Out

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