CARBON DIOXIDE COOLING AND POWER COMBINED CYCLE FOR MOBILE APPLICATIONS

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ABSTRACT

A carbon dioxide cooling and power combined system, which can provide A/C and produce power at the same time, is proposed in this paper. The power part of the system can utilize the energy in automobile exhaust gases to produce electricity for the A/C's compressor, thus both increases the cooling COP and decreases the vehicle's fuel consumption by reducing the compressor's energy demand. Moreover, different factors that influence the cycle's optimum working condition are also discussed.

The software Engineering Equations Solver (EES) is used to model and to analysis the system performance. Results show that the proposed combined system can achieve a COP of 3.18 for the cooling part and 12.6% efficiency for the power part under a typical working condition. After transferring the energy gained from the cycle to its compressor, the new COP will be 4.45 and the improvement of COP will be around 40% accordingly.

1. INTRODUCTION

The most commonly used refrigerant for nowadays automotive A/C systems is the hydrofluorocarbon (HFC) refrigerant R-134a, which has a Global Warming Potential (GWP) 1300 times that of CO_2. Ever since the release of Kyoto protocol, many countries have been planning seek means to limit the use of this kind of synthetic refrigerants by adding tax, legislations etc. In the European Union (EU), the legislation has already been proposed to forbid the usage of HFC refrigerants with a GWP > 150 in new vehicle automotive A/C systems from 2011.

Compared to the synthetic refrigerants, Carbon dioxide (CO_2) as an environmentally benign natural refrigerant has many advantages. It is inexpensive, non-explosive, non-flammable and abundant in the nature. Further, due to its relative high working pressure, the carbon dioxide refrigeration system is more compact than other refrigeration systems as well. With the trend of abandoning the use of HFC refrigerants, it has becoming more and more popular. Besides the advantages as a refrigerant, CO_2 also has its advantage in utilizing the energy in waste heat with gradient temperature (e.g. vehicles' exhaust gases). This is mainly due to the character of its temperature profile in the supercritical region that can provide a better matching to the heat source than other working fluids (organic working fluid, fluid mixtures etc). Thus the so-called pinching problem, which limits the cycle performance in waste heat recovery, will not be encountered inside the CO_2 counter current heat exchanger as what happened to other working fluids (Y.Chen et al., 2005). The pinching is schematically illustrated in figure 1.

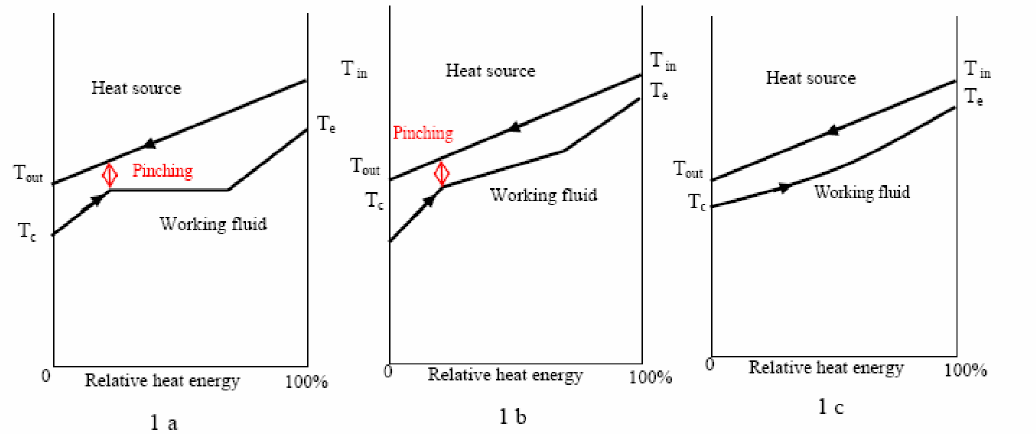

Figure 1. Schematic illustration of the heat transfer between the low-grade heat source and the working fluid in a counter flow heat exchanger. (1a) pure fluid; (1b) zeotropic fluid mixtures; (1c) carbon dioxide

Based on the reasons mentioned above, the authors proposed a CO_2 cooling and power combined cycle (system) for automotive applications. The power part of the combined cycle will utilize the energy in the engine's waste heat to produce power and this power can be either transferred to the cycle cooling part to reduce the energy consumption of its compressor or be used to provide electricity for the vehicle. The power output from the combined cycle power part can be called "free" energy that produced by the cycle, due to the fact that it is gained from the engine's waste heat (i.e. exhaust gas). Further, if this power is used to reduce the energy consumption of the cooling part's compressor, the combined cycle can achieve the required cooling capacity with less energy demand for the compression work thus saving the fuel and lowering the emissions. Consequently, the new COP of the combined cycle cooling part can be defined by the following equation.

$$COP_{new} = \frac{Q_{cooling}}{W_{new}} = \frac{Q_{cooling}}{W_{basic} - W_{output}} \tag{1}$$

Where $Q_{cooling}$ is the required cooling capacity of the cooling system, W_{new} is the new compression work after taking away the energy gained by the combined cycle's power part, W_{basic} is the original compression work of the cooling cycle, W_{output} is the work output from the combined cycle's power part. I.e. the "free" energy gained from the engine exhaust gas.

From eq. (1) one can see that the proposed cycle (system) will increase the cooling COP by reducing its compressor's energy consumption with the "free" energy gained from the engine's waste heat.

2. DESCRIPTION OF THE PROPOSED SYSTEM AND THE CYCLE

The proposed CO_2 cooling and power combined system is mainly composed of six parts, namely: evaporator, compressor, gas heater, expander, gas cooler and throttling valve. The system layout and the corresponding cycle's T-S chart are showed respectively as follows (Figure 2 & Figure 3).

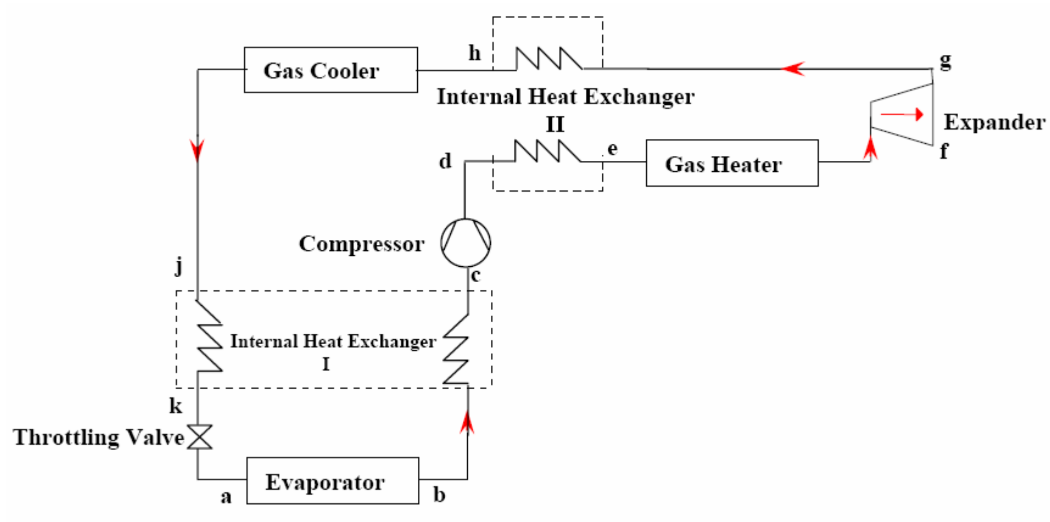

Figure2. Carbon dioxide cooling and power combined system schematic layout

After absorbing heat from the compartment room (a-b) in the evaporator, the working fluid will be further heated in the internal heat exchanger I (IHX I) until it becomes slightly superheated (b-c). The superheated carbon dioxide vapour will then be compressed by a compressor to a supercritical pressure (c-d), where supercritical carbon dioxide absorbs the heat firstly from the expansion outlet carbon dioxide in IHX II (d-e) and then from the engine exhaust gas in a gas heater (e-f). After that, the supercritical carbon dioxide will be expanded in an expander (f-g) and then cooled by IHX II (g-h), a gas cooler (h-j) and IHX I (j-k) in turn. Finally, it flows through a throttling valve and enters the evaporator (k-a). In the power part of the combined cycle (d-f-g-i), carbon dioxide will absorb the heat in the engine exhaust gas (e-f) and convert it into useful energy (i.e. electricity). Further, this energy will be transferred to the cooling part of the combined cycle to reduce the compressor's power consumption (c-d).

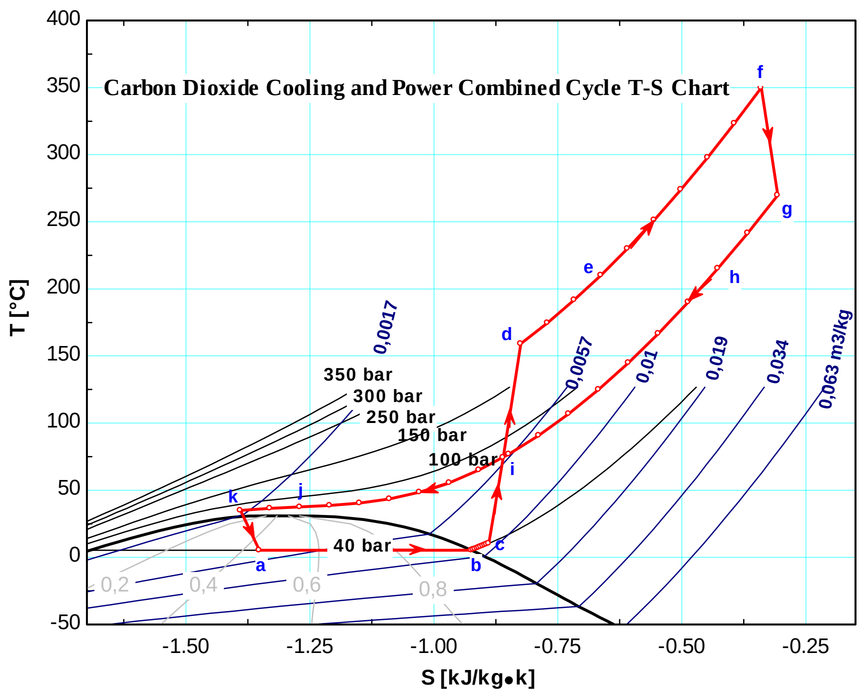

Figure3. Carbon dioxide cooling and power combined cycle T-S chart

A basic combined cycle under typical working condition can be employed here as an example for the basic cycle analysis. For the cooling part, the working condition is selected according the most commonly used working condition in others research and in CO_2 automobile A/C prototype testing (Man-Hoe K. et al., 2004). The evaporation pressure is selected to 40 bar and the evaporation temperature will be 5.3 ºC accordingly. The compressor's isentropic efficiency is assumed to 75% according to the research done by Andrey R. and Chi-Chuan W. (2001). Further, 5 ºC superheat at evaporator outlet is assumed as a fixed value to ensure there is no moisture at the compressor inlet. Ongoing researches on carbon dioxide power cycles are mainly focusing on utilizing such a cycle in combination with nuclear reactors as the heat source to produce electricity, therefore the cycle works with very high gas heater pressure and high expansion inlet temperature (V. Dostal et al., 2004; E.G. Feher, 1967). However, in this application these extreme pressures are avoided and the gas heater pressure is selected as 200 bar for the power part of the combined cycle. The gas temperature at the engine exhaust gas manifold can reach as high as 500 ºC, nevertheless the expansion inlet temperature is assumed as 350 ºC by taking into account the heat exchanger size, component material, etc. The research on carbon dioxide expansion machines in power cycles is very limited, except using CO_2 expanders in transcritical refrigeration cycle to replace the throttling valve to increase the cycle's COP. The expansion efficiency is assumed to be 80% based on findings by several authors (J. Nickl et al, 2003; S.T. Zha et al, 2003; D. E. Boewe et al, 2003). The gas cooler pressure is selected to 85 bar, which is the approximate optimum gas cooler pressure for the cooling part of the combined cycle under the current working conditions and the gas cooler outlet temperature is assumed to 35 ºC. The Internal Heat Exchanger (IHX) effectiveness is assumed to be 90% according to the D. E. Boewe et al's research (2001).

A calculation program is made in EES and the calculation results show that such a cycle under the described working condition will achieve a COP of 3.18 for the cooling part and 12.6% efficiency for the power part. After transferring the energy gained from the power part to the cooling part, the new COP of the cooling part, as defined in equation 1, will be 4.45 and the enhancement of COP will be around 40% accordingly. The cycle working conditions are listed in the following table.

Table 1. Basic combined cycle working condition

Items	Value	Unit
Evaporator pressure	40	bar
Evaporation temperature	5.3	ºC
Cooling capacity	10	KW
Superheat after Evaporator	5 (fixed value)	K
Gas heater pressure	200	bar
Expansion inlet temperature	350	ºC
Gas cooler pressure	85	bar
Gas cooler outlet temperature	35	ºC
Compression eff.	75%	-
Expansion eff.	80%	-

3. DISCUSSION

The carbon dioxide transcritical cooling cycle will have an optimum gas cooler pressure for a certain working condition. The cooling part COP of the combined cycle is plotted against different gas cooler pressures while keeping other cycle working conditions constant as proposed above for the basic cycle (Figure 4). From the figure, one can see that at the optimum gas cooler pressure, the improvement of cooling cycle's COP is tremendous (e.g. around 40% as mentioned above in the basic cycle analysis) and the enhancement of COP is different for different gas heater pressures. It is also found that the optimum gas cooler pressure for the combined cycle's cooling part (i.e. to achieve the highest COP_{new}, after transferring the power gained form the cycle's power part to the cooling part) will remain the same as that for the basic transcritical cooling cycle (around 85 bar for the proposed working condition).

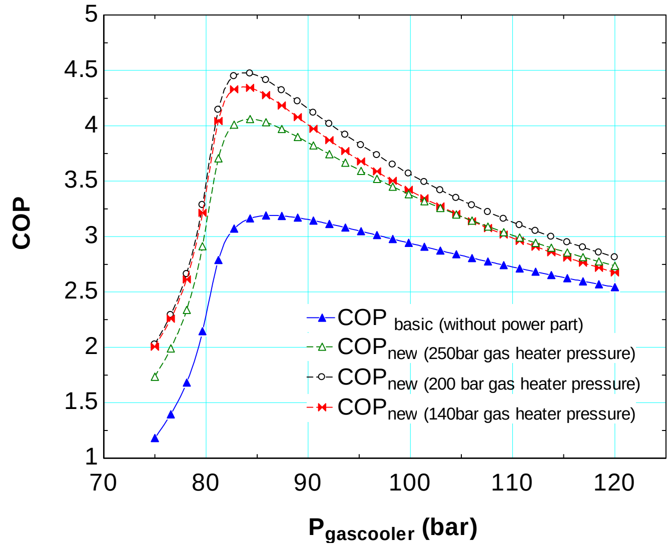

Figure 4. The COP of cooling part of the combined cycle vs. different gas heater pressure

Keeping the optimum gas cooler pressure constant, the combined cycle's cooling part COP is plotted against different gas heater pressures for different expansion inlet temperatures (Figure 5). It is found that there is an optimum gas heater pressure for the combined cycle's new COP for certain expansion inlet temperature (i.e. the gas heater pressure for combined cycle power part to achieve the highest W_{output} for a certain expansion inlet temp.). Further, the optimum gas heater pressure is increasing with the increase of expansion inlet temperature.

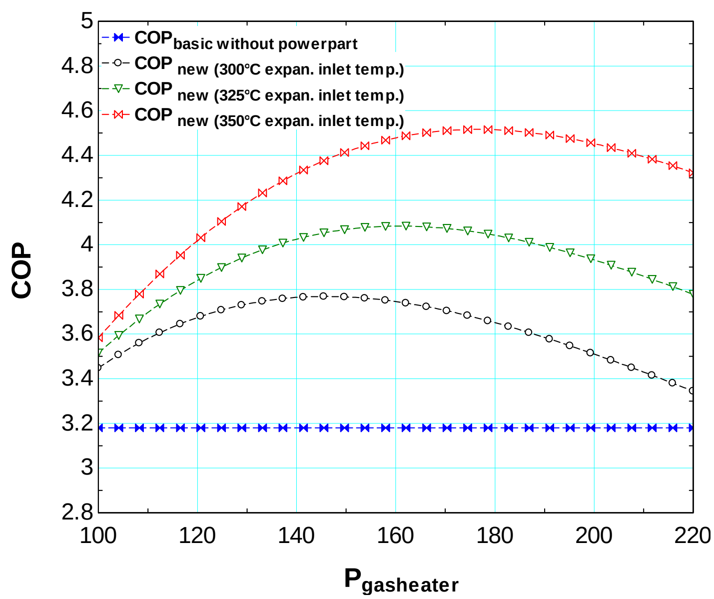

Figure 5. The COP of cooling part of the combined cycle vs. different gas heater pressure

Further, by plotting both the combined cycle's cooling part COP (both COP_{basic} and COP_{new}) and the power part efficiency against different gas heater pressures, it is noticed that for a certain expansion inlet temperature, the optimum gas heater pressure for the combined cycle's power part is different from the one for the cooling part (Figure 6). Moreover, the optimum gas heater pressure for the combined cycle's power part with IHX is lower than that for the power part without IHX. Further, it is also noticed that the IHX has a crucial influence for the power part efficiency.

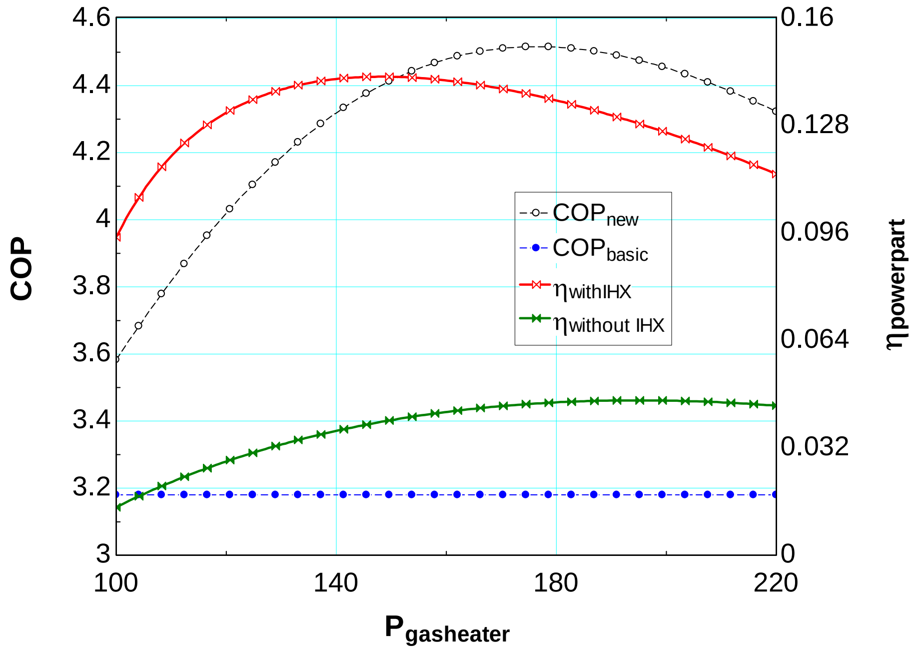

Figure 6. Combined cycle cooling part COP and power part efficiency vs. different gas heater pressure at 350ºC expansion inlet temperature.

The reason why the optimum gas heater pressure for the combined cycle's power part is different from the one for the cooling part can be explained by Figure 7. From Figure 7a, one can see that for a certain expansion inlet temperature, the power output (W_{output}) from the combined cycle's power part is increasing first and then decreasing with the increasing gas heater pressure. Thus, the combined cycle cooling part's new compression work, which is the value of original cooling part's compression work minus the work gained by the cycle power part (i.e. W_{basic}-W_{output}), is decreasing first and then increasing. Since the required cooling capacity does not change ($Q_{cooling}$), the combined cycle cooling part's highest COP (COP_{new}), which determined by the ratio between the cooling capacity ($Q_{cooling}$) and the new compression work (W_{new}), appears when the power part has its highest work output. For the efficiency of the power part with IHX, the required heat input to reach the preset expansion inlet temperature is increasing with increasing gas heater pressure while the work output from the cycle keeping the same trend as mentioned before (Figure 7b). The cycle's efficiency will achieve its maximum value when the ratio between the cycle work output and the required heat input reaches its maximum. However, the gas heater pressure for the combined cycle power part to achieve its highest work output is different from the one for it to achieve the highest ratio between its work output and the requirement heat input. Therefore, the optimum gas heater pressure for the combined cycle's power part to achieve the highest efficiency is different from the one for its cooling part to achieve the highest COP_{new}. Nevertheless, since the combined cycle's power part is utilizing the engine's waste heat to produce power (free power), the power part's efficiency is not as important as the efficiency for the traditional power cycle, which utilizing the fossil fuels as heat source. Therefore, the gas heater pressure for the highest COP should be employed for the combined cycle. If the power gained from the combined cycle's power part is used for other purposes (e.g. producing electricity for the vehicle, etc.), the cycle's gas heater pressure should be however reconsidered by considering the power part efficiency as well.

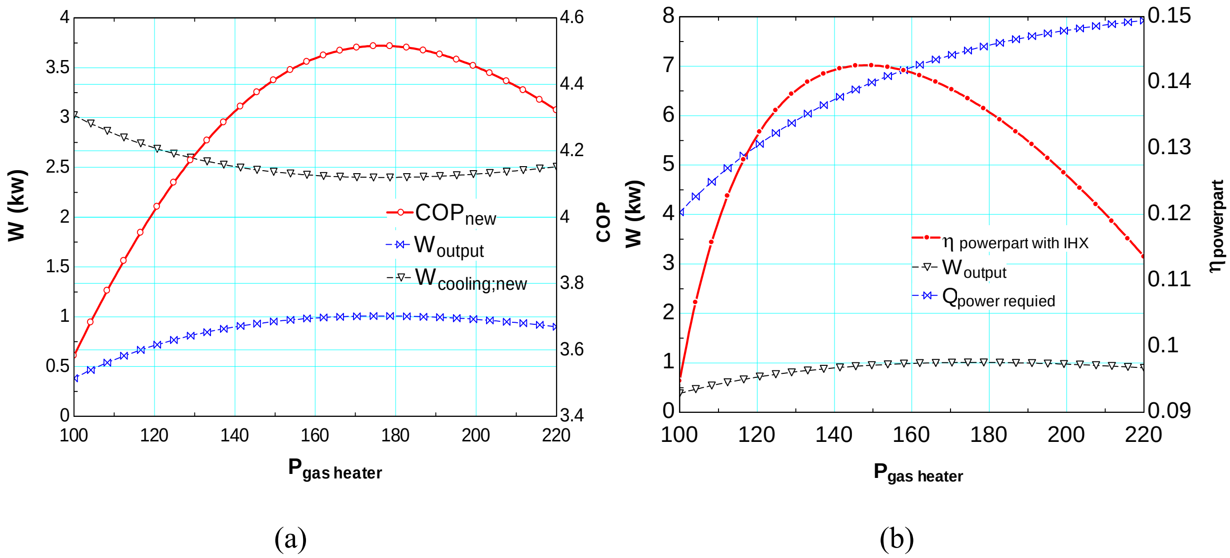

Figure 7. (a) COP $_{new}$ and W vs. different gas heater pressure at 350ºC expansion inlet temperature. (b) Efficiency and W vs. different gas heater pressure at 350ºC expansion inlet temperature

CONCLUSION

A novel carbon dioxide cooling and power combined system has been proposed for automobile application. The power part of the combined system can utilize the energy in the engine's exhaust gas to produce power for the cooling part's compressor; therefore, such a system can both achieve a higher cooling COP than the traditional CO_2 transcritical cooling systems and reduce the vehicle fuel consumption by reducing the energy demand of cooling part's compressor.

By analyzing a basic cycle with the typical working condition, it is found that there is an optimum gas cooler pressure as well as an optimum gas heater pressure for the combined cycle to achieve the highest COP $_{new}$ for a certain cycle operating condition. Further, for a certain evaporating pressure and temperature, the optimum gas heater pressure is increasing with the increasing expansion inlet temperature. It is also found that the optimum gas heater pressure for the combined cycle's power part to achieve the highest efficiency is different from the one for the combined cycle's cooling part to achieve the highest COP. However, since the combined cycle power part is utilizing the engine waste heat to produce power for the cooling part compressor, the optimum gas cooler pressure for the cooling part should be employed for the combined cycle. Nevertheless, if the energy gained from the cycle power part is used for other purpose (e.g. producing electricity for the vehicle); the difference in optimum gas heater pressure for power part and cooling part should be considered when selecting the right gas heater pressure for the combined cycle.

NOMENCLATURE

IHX	Internal Heat Exchanger	(-)
GWP	Global Warming Potential	(-)
COP	Coefficient of Performance	(-)
Eff.	Efficiency	(-)
η	Cycle efficiency	(-)
Q	Required Cooling capacity	(KW)
W	Work	(KW)

Subscripts

a-k	Cycle working points
output	The power output the combined cycle power part
cooling	Combined cycle cooling part
power	Combined cycle power part
new	The new cooling COP of the combined cycle

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