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CO_2 Heat Pumps in the Drying Industry: Identifying the Break-Even Point Between Transcritical and Supercritical Operation

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Abstract

In the drying industry, energy-efficient heat recovery systems are essential for optimizing thermal processes and reducing operational costs. High-temperature heat pumps, particularly CO_2-based systems, offer promising potential due to their favourable thermodynamic properties and environmental benefits. Drying processes typically operate at temperatures ranging from 60°C to 220°C, with hot air commonly used as the medium for heat transfer. Transcritical and supercritical CO_2 heat pumps each have unique advantages depending on the temperature and pressure conditions of the process.

This research investigates the performance characteristics of trans-critical and supercritical CO_2 heat pumps in industrial drying applications, while identifying the break-even point between these two configurations, focusing on key parameters such as system efficiency and operational conditions. Thermodynamic modelling and performance simulations are employed to assess how variations in source temperature, system design, and operating conditions affect the choice between transcritical and Supercritical operation.

Understanding this break-even point is crucial for optimizing CO_2 heat pump systems for drying processes, ensuring that the system operates at peak efficiency while meeting the thermal demands of industrial drying. By examining these configurations in-depth, the study aims to contribute valuable insights into the potential for CO_2 heat pumps to enhance energy efficiency, reduce environmental impact, and support more sustainable practices in the drying industry.

Keywords: *CO_2 Heat Pump; Transcritical CO_2; Supercritical CO_2, Industrial drying*

1. Introduction

Drying is an energy-intensive processes essential in various industrial applications, including food processing, pharmaceutical, and chemical industries, where the removal of moisture is crucial to enhance shelf life, facilitate further processing, or achieve desired product characteristics (Van 't Land, 2011). The versatility of drying technologies is evident, as numerous methods, such as spray drying, fluidized bed drying, and belt drying, are employed depending on the specific requirements of the product being processed. In most cases, these drying methods necessitate the use of hot air, which must be supplied at specific temperatures to ensure efficient moisture removal without adversely affecting the product quality (Schuck, 2002).

Spray drying is a process that is extensively used in the food industry as food products tend to be heat sensitive. This process is employed in the manufacture of dried food products, including milk and whey powders, as well as tea and coffee extracts. The technique has also been employed for the encapsulation of active materials within a protective matrix (Gharsallaoui et al., 2007; Barbosa-C´anovas, 2005). The temperature of the hot air utilized is contingent on the volume of air that is circulated through the drying chamber and the quantity of liquid sprayed (Fleming, 1921). In order to ensure effective drying and encapsulation, the desired temperature of the hot air is usually in the range of 150-220 °C (Gharsallaoui et al., 2007).

The generation of hot air for these drying processes is commonly achieved by transferring the heat generated by fossil fuel burners, or steam boilers to ambient air using a heat exchanger (Van ’t Land, 2011). However, in recent years, there has been a growing interest in alternative heating methods, such as the use of electric heaters or heat pumps, because of the requirement for sustainability. State of the art electric heaters are already in use, while high temperature heat pumps are slowly entering the market (Zühlsdorf et al., 2023). Despite this fact, heat pumps are able to compete as they are capable of providing high temperature heat using only a fraction of the electricity consumed by an electric heater by utilizing industrial waste heat. This helps further enhance system efficiency and reduce overall energy consumption (Arpagaus et al., 2018).

A number of different refrigerants are available for utilization in heat pumps. Hydrocarbons such as propane and butane, which are environmentally safe but also flammable, can be utilized in a heat pump (Arpagaus et al., 2018). Notable natural refrigerants, such as ammonia, are also compatible with heat pumps. However, when considering the applications in the food industry, Ammonia might pose significant challenges due to its toxic properties. Other natural refrigerants such as water, air and CO_2 can also be used, which are non-toxic, non-flammable, and abundant, making them an environmentally friendly alternative.

The utilization of carbon dioxide (CO_2) as a refrigerant has been a prevalent practice for several decades within the domain of refrigeration, encompassing subcritical and transcritical cycles. As shown in Fig. 1(a), a subcritical cycle remains below the critical point (31.1 °C, 73.7 bars). In a transcritical cycle, as shown in Fig. 1(b), CO_2 is compressed above the critical point into the supercritical zone, where heat rejection takes place by sensible cooling of the superheated CO_2 using a gas cooler. This is in contrast to the transient cooling in a condenser of a subcritical cycle (Austin et al., 2011). For a High Temperature Heat Pump (HTHP), on the other hand, transcritical and supercritical cycles should be employed because of the required temperatures at the Heat sink. The P-h diagram for a supercritical cycle can be seen Fig. 1(c). It has been found that both cycles are capable of providing a large temperature glide on the heat sink side, which is required for drying processes (Adamson et al., 2022; Aga Vipluv, 2016).

This growing interest in alternative heating technologies raises an important research question: How do the transcritical and supercritical CO_2 heat pumps compare for the purposes of providing hot air for drying, when the available waste heat is in the range of 50-70 °C? This question forms the basis for further investigation, as industries seek to balance energy efficiency, environmental sustainability, and product quality in their drying processes.

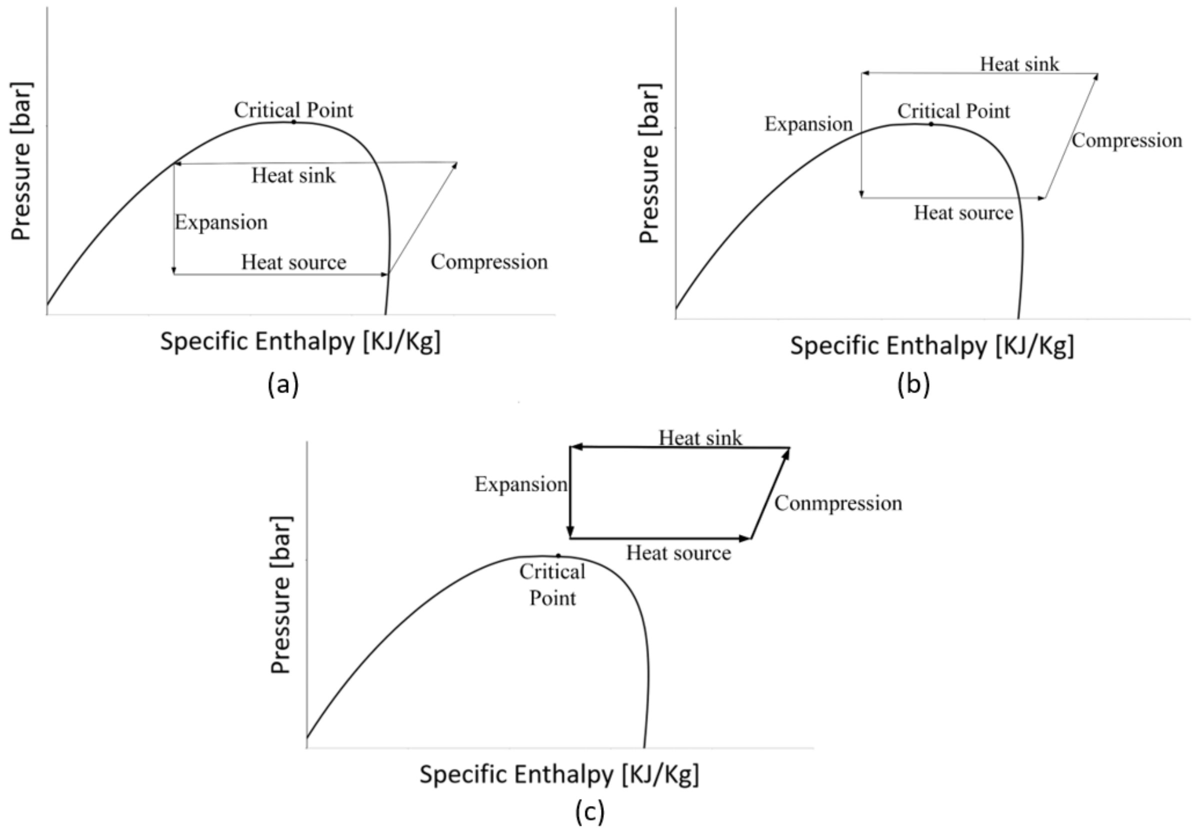

Fig. 1: p-H diagrams showing (a) subcritical cycle, (b) transcritical cycle and (c) supercritical cycle

2. Thermodynamic modelling

The modelling of a HTHP with Carbon dioxide as the refrigerant has been carried out in the software EBSILON®Professional. This software is used for the steady state, thermodynamic simulations of closed loop processes. The different cycle configurations that have been considered here are shown and described in the following sub-sections.

The HTHP can be divided into three sub-systems based on the fluid, namely the heat pump loop, heat source and the heat sink. The reservoir from which the heat is extracted and upgraded is called the heat source, which in this case is industrial waste heat (Zühlsdorf et al., 2023). The heat source medium can either be water or air depending on the availability of waste heat. The parameter fixed here is the temperature of CO_2 in the heat pump loop after the Low Temperature Heat Exchanger (LTHX). The temperature is set as 50 °C, which equates to heat source temperature in the range of 50-70 °C depending on the heat transfer medium, availability of waste heat and heat exchanger design. At the other end is a reservoir to which the upgraded heat is supplied, known as a heat sink. The heat sink medium is considered to be air. This air enters the High Temperature Heat Exchanger (HTHX) with a temperature of 30 °C and a mass flow rate of 2 Kg/s. Using the HTHP, this air heats up to a temperature of 160 °C. The pressure of CO_2 prior to the compressor is constant, though it differs depending on whether the cycle is transcritical or supercritical. In the transcritical cycle configurations, the pressure at which CO_2 enters the compressors is equivalent to 50 bars. Whereas in the supercritical cycle configurations, the pressure at the same point is 80 bar. For the sake of compression, these boundary conditions remain constant for all considered configurations and carried out simulations.

2.1 Heat pump configurations

For both transcritical and supercritical heat pumps, multiple configurations have been modelled and simulated. Each of the modelled configuration are shown in the form of simplified schematics in the following sections.

2.1.1 Transcritical heat pump configurations

Fig. 2 shows the different heat pump configurations, that are considered in the scope of this study for a transcritical heat pump. Both the configurations have two heat exchangers, a compressor and an expander, similar to any standard heat pump. However, both these configurations employ different expansion mechanisms. In Fig. 2(a) a turbine is used the purpose of expansion, while in Fig. 2(b) an expansion valve does the same.
The HTHX is the heat exchanger placed between the outlet of the compressor and the inlet of the expander, on the high-pressure side of the HTHP. This heat exchanger connects the HTHP to the heat sink, where the heat is dissipated and the CO_2 is cooled down to temperatures below or near the critical point. LTHX on the other hand is the heat exchanger placed between the outlet of the expander and the inlet of the compressor, on the low-pressure side of the HTHP. So, the low pressure and low temperature refrigerant flows through this heat exchanger to absorb heat from the heat source. This preheats the refrigerant before it is compressed again, reducing the required temperature lift.

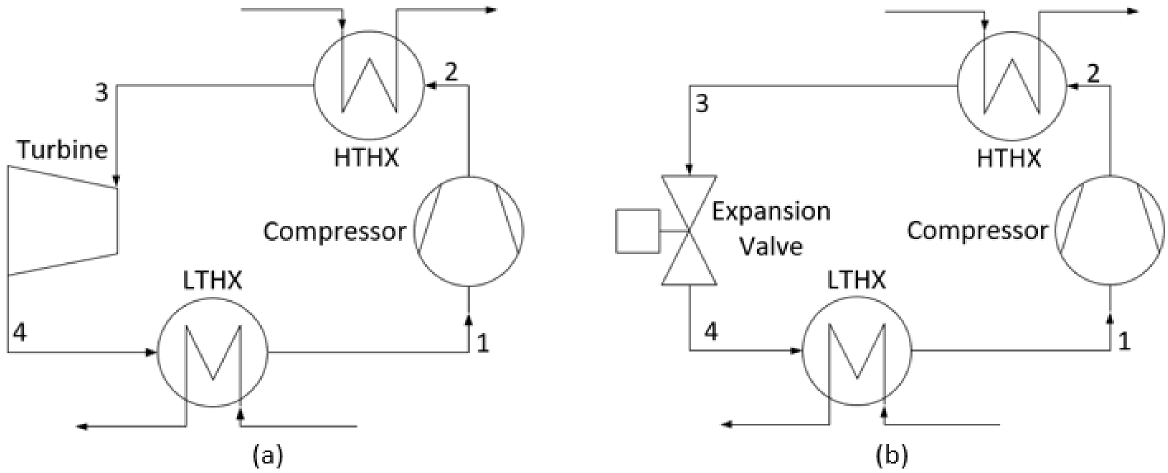

Fig. 2: Transcritical Heat pump configurations

2.1.2 Supercritical heat pump configurations

Fig. 3 presents the schematics of the different heat pump configurations, that are considered within the scope of this study for a supercritical heat pump. A total of four different configurations have been modelled and simulated. In addition to the configurations mentioned in section 2.1.1, two more configurations with three heat exchangers instead of two are considered for supercritical heat pumps. An additional Internal Heat Exchanger (IHX) is considered here in addition to the already mentioned HTHX and LTHX. In contrast to the transcritical cycles, the CO_2 following the HTHX in a supercritical cycle is kept well above the critical point. Hence, there is excess heat available, which the IHX can transfer from the high-pressure to the low-pressure side of the HTHP. Hence, an IHX is useful here, but redundant in a transcritical cycle.

As illustrated in Fig. 3(a) and Fig. 3(b), the schematics depict turbines utilized for the expansion of refrigerant, with and without the incorporation of an IHX, respectively. In a similar manner, the schematics in Fig. 3(c) and Fig. 3(d) illustrate an alternative configuration, wherein an expansion valve is used instead of a turbine.

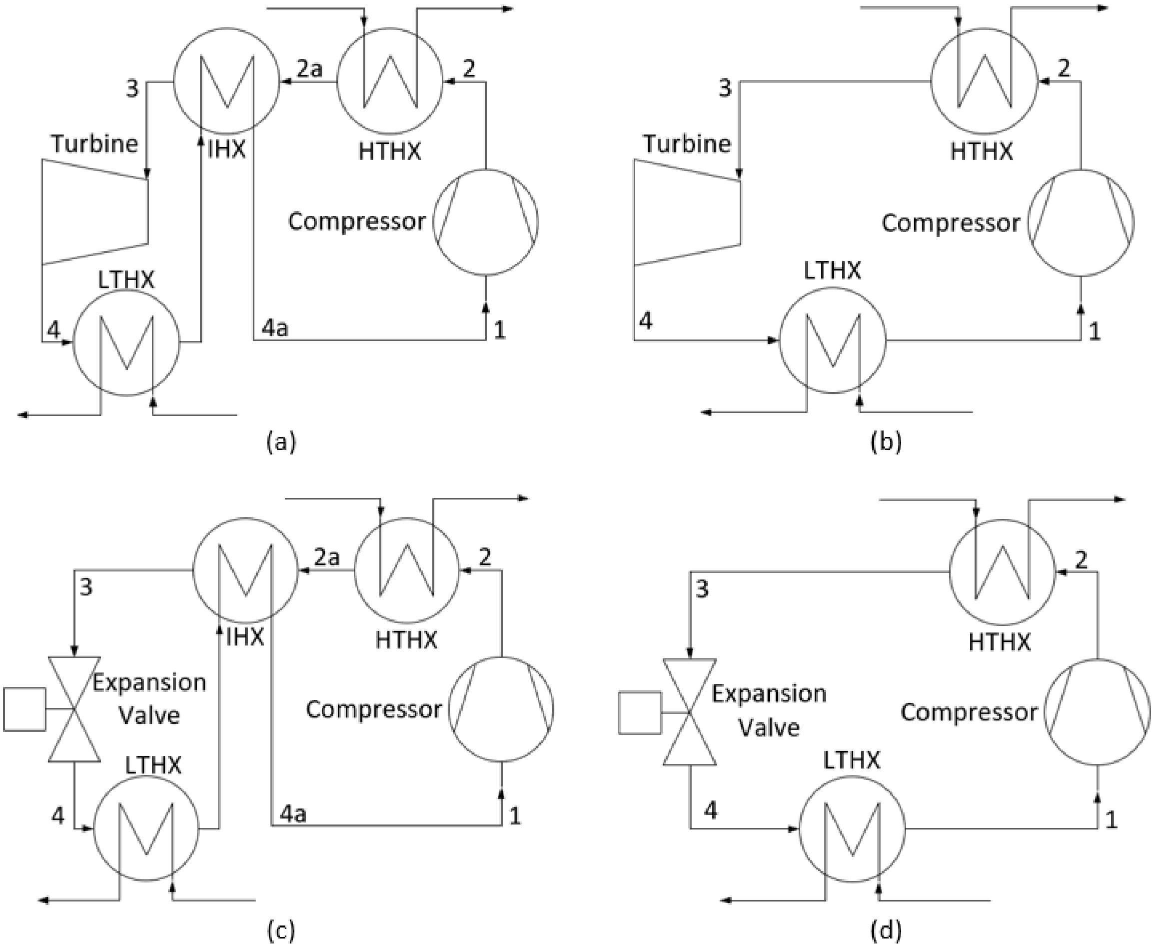

Fig. 3: Supercritical Heat pump configurations

2.2 *Performance indicator*

The main performance indicator used to compare the two different types of heat pump cycles and the multiple configurations is the Coefficient of Performance (COP). The COP is the ratio of the heat dissipated from the HTHX and the net electrical power consumed by the heat pump. This can be calculated as shown in Equation 1.

$$COP = \frac{\dot{Q}_{HTHX}}{P_{el}} \tag{1}$$

Where $\dot{Q}_{HTHX}$ is the amount of heat dissipated from the heat pump through the HTHX, which can be calculated as shown in the equation 2.

$$\dot{Q}_{HTHX} = \dot{m} \cdot c_p \cdot \Delta T \tag{2}$$

Where $\dot{m}$ is the mass flow rate flowing through the heat exchanger, c_p is the specific heat capacity of the fluid at constant pressure and ΔT is the difference in temperature of the fluid between the inlet and outlet of the heat exchanger. As the $\dot{m}$ and the ΔT for the process air is kept constant here, the Q_{HTHX} will remain constant for all cases.

P_{el} is the net electrical power consumed by the HTHP to generate Q_{HTHX}. In the configurations, where an expansion valve is employed, the net electrical power is the electricity consumed by the compressor (P_{Comp}). On the other hand, the net electric power for configurations with a turbine can be calculated as shown in Equation 3 by calculating the difference between the electric power consumed and generated.

$$P_{el} = P_{Comp} - P_{Turb} \tag{3}$$

The second parameter used to indicate the performance of a HTHP and to compare the different configurations is the pressure ratio (PR). The pressure ratio is defined as the ratio of the pressures after and before the compressor. This can be calculated using the Equation 4.

$$PR = \frac{p_1}{p_2} \tag{1}$$

Where p_1 is here the pressure of CO_2 before the compressor and p_2 is the pressure after the compressor. The pressure of the CO_2 after the compressor (p_2) is not a fixed value like the p_1 but has been optimized individually for each case for the best possible COP. Hence, the pressure ratios for the different configurations can vary.

3. Results and discussion

The configurations mentioned in section 2.1 have been simulated thermodynamically and the results of these simulations are summarized below in Table 1. It shows the mass flow rate of CO_2 required in the heat pump loop, the pressure ratio provided by the compressor and the COP of the heat pump for each configuration. These three parameters are calculated through simulations and are not an input parameter. The mass flow rate and the pressure after the compressor (p_2) are optimized for the maximum possible COP, considering the input parameters mentioned in section 2.

Table 1: Simulation results

Fig. No.	Configuration	$\dot{m}$ [kg/s]	p_2 [bar]	Compressor PR (-)	COP
2a	tCO_2 with turbine	1.16	174.83	3.50	3.60
2b	tCO_2 with exp. valve	0.91	198.74	3.98	3.12
3a	sCO_2 with turbine + IHX	1.42	220.52	2.76	3.89
3b	sCO_2 with turbine	1.43	311.40	3.89	3.73
3c	sCO_2 with exp. Valve + IHX	1.42	229.46	2.87	2.80
3d	sCO_2 with exp. valve	1.23	327.09	4.10	2.65

As evident from the Table 1, the mass flow rate of the refrigerant required for optimal COP is lower in the transcritical loop compared to that in the supercritical loop. Although the heat delivered by the HTHX on the sink side is constant, the CO_2 temperatures entering and exiting the HTHX are different across configurations. Hence, with different ΔT the quantity of CO_2 ($\dot{m}$) required to deliver the same amount of heat is different. While mass flow rates differ between transcritical configurations, they remain comparable across supercritical setups. Additionally, turbine configurations exhibit slightly higher mass flow rates than expansion valve configurations to deliver the same heat. This is because, the optimized parameter is the COP and the electrical power provided by the turbines help increase the COP.
The subsequent parameters presented in Table 1 are the compressor outlet pressure (p_2) and the pressure ratio. The data presented in the table indicates that the pressure ratio in a heat pump equipped with an expansion valve exceeds that of a heat pump with a turbine. A potential explanation for this phenomenon lies in the fact that the mass flow rate of CO_2 in the heat pump

loop is different. On the other hand, for supercritical heat pump configurations, it is also evident that the pressure required at the outlet of the compressor (p_2) is notably high in the absence of an IHX. The IHX facilitates the transfer of excess heat from the high-pressure to the low-pressure side of the heat pump. This process of pre-heating involves transferring excess heat from the high-pressure refrigerant exiting the HTHX to the low-pressure refrigerant entering the compressors, thereby reducing the compressor work required to achieve the desired temperature elevation. This is why, the compressor pressure and the temperature of the refrigerant before the compressor are inversely proportional here. The most unfavorable outcome is observable in the supercritical configuration with an expansion valve. The pressure ratio and the p_2 value are the highest in this scenario and are marked in orange in Table 1.
As illustrated in Table 1, an overview of the COPs for each of the considered configurations is provided. It is evident that configuration 3a has the highest COP, which is indicated by the color green. The configuration 3b is the second highest COP, which is noteworthy in its own right. Nevertheless, the pressure ratio required to achieve this COP renders this system less practical. The compression of CO_2 from 80 bars to pressures ranging from 300 to 350 bars, and the maintenance thereof, can be potentially costly, so the integration of an IHX, as shown in Fig. 3(a), should be preferred. It is also notable that configurations 2a and 2b exhibit COP values in excess of 3. However, it is evident that employing a turbine results in a substantial enhancement of the COPs. While a turbine for the transcritical cycle is theoretically preferable, it can be challenging to design a turbine that can operate within the 2-phase zone.

4. Conclusions

A review of the extant literature indicates that the majority of research and development in the field has been focused on transcritical heat pumps, with comparatively limited research activity in the area of supercritical heat pumps. Despite the fact that supercritical cycles provide thermodynamic benefits by using a gas cooler and a gas heater instead of a condenser and evaporator, reducing the exergy losses caused by phase change. This study aims to provide a comparative analysis of the transcritical and supercritical cycles for a CO_2 heat pump, with a particular focus on their application in the domain of drying. This objective is pursued by conducting closed-loop thermodynamic simulations, which are utilized to analyze the required compressor pressure ratio and COP values for the various cycle configurations. Table 1 provides a synopsis of the findings for each considered configuration. The findings suggest that the supercritical CO_2 heat pump is the optimal choice when the available waste heat has a temperature of 50 °C or higher. This is mainly due to the fact that sCO_2 heat pumps are able to achieve higher COPs while efficiently utilizing the sensible heat in drying applications. With regard to cycle configurations, it is evident that the incorporation of the IHX and turbine can significantly enhance the COP of supercritical cycles, as demonstrated in configuration 3a. This finding underscores the necessity for further research in the domain of supercritical CO_2 HTHPs.

Nomenclature

Symbols		**Units**	**Abbreviations**	
h	Specific enthalpy	[J/kg]	CO_2	Carbon Dioxide
$\dot{m}$	Mass flow rate	[kg/s]	COP	Coefficient of Performance
p	Pressure	[bar]	GWP	Global Warming Potential
P	Power	[W]	HTHP	High Temperature Heat Pump
$\dot{Q}$	Heat rate	[W]	HTHX	High Temperature Heat Exchanger

T	Temperature	[°C]	IHX	Internal Heat Exchanger
ΔT	Temperature difference	[K]	LTHX	Low Temperature Heat Exchanger
Subscripts			ODP	Ozone Depletion Potential
Comp	Compressor		PR	Pressure Ratio
el	Electric		sCO_2	Supercritical Carbon Dioxide
Turb	Turbine		tCO_2	Transcritical Carbon Dioxide

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