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USING CARBON DIOXIDE IN A TRANSCRITICAL VAPOR COMPRESSION REFRIGERATION CYCLE

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ABSTRACT

For most of the last fifty years, refrigeration and heat pump systems have been based on cycles using a fluorocarbon as the working fluid. However, with the discovery that fluorocarbons may have a profoundly detrimental effect on the earth's atmosphere, it has become necessary to find a suitable replacement for a fluorocarbon based refrigeration cycle. Such a replacement must perform comparably to current refrigerants, be economically feasible, and significantly reduce the possibility of a negative environmental impact compared to current refrigerants. A review of the literature on the first vapor compression refrigeration cycles indicate that carbon dioxide was used as the working fluid prior to and after the turn of the century. Moreover, recent literature has suggested that carbon dioxide will work for certain air conditioning applications. Not only is carbon dioxide inexpensive and plentiful, but is a substance naturally occurring in large quantities in the earth's atmosphere. Furthermore, initial investigation of the heat transfer properties of carbon dioxide and subsequent modeling of a transcritical carbon dioxide vapor compression cycle suggest potential performance comparable to that of existing refrigeration systems. This paper will present a detailed literature review of carbon dioxide's role in refrigeration cycles, what has been investigated so far to find a suitable application for carbon dioxide as a refrigerant, and what further research needs to be done to implement a feasible cycle.

INTRODUCTION TO CARBON DIOXIDE

Background

Carbon dioxide's role in refrigeration stretches back well over 100 years and likely had the largest impact of any fluid on early food refrigeration and human occupied space air conditioning. Though the Evans-Perkins process upon which modern refrigerators and air conditioners are based, was developed in 1834 [Lorentzen 1994], it wasn't until 1866 when the American Thaddeus S.C. Lowe first harnessed carbon dioxide for ice production [Thevenot 1979]. Following a period of further development, the first documented carbon dioxide compressor was built by Windhausen in 1880 [Goosman 1927]. Other landmark uses of carbon dioxide refrigeration include the first marine installation a carbon dioxide plant by J & E Hall in 1890 [Thevenot 1979] and the first continuous production of carbon dioxide refrigerating equipment in the United States by Kroeschell Bros. of Chicago in 1897 [Goosman 1927]. Not only were carbon dioxide based machines growing in popularity in the late 1800s, but improvements were continually being made on the basic cycle (almost from the incipience of initial development!). J & E Hall demonstrated that the efficiency of the vapor compression process could be improved through the use of two stage compression [Thevenot 1979]. In 1905, Voorhees developed what is now known as the multiple effect cycle, which involves a separation of liquid and vapor at an intermediate stage in the expansion process. According to Voorhees, for heat absorption temperature of 0°F (-17.78°C) and a heat rejection temperature of 90°F (32.22°C)"the least possible gain in refrigeration will be 127% and a saving of 38% of the power required per unit refrigeration will result" [Voorhees 1905]. The advent of refrigeration and air conditioning in the late 1800s had an enormous impact on thousands of industries. At the time, carbon dioxide was not the only refrigerant used. Ammonia and sulfur dioxide based machines enjoyed a level of popularity, but generally only in industrial applications due to their toxicity and/or flammability. None such hazards are inherent in carbon dioxide. For that reason food-related industries and places of human occupancy (theaters, hospitals, restaurants, etc.) used carbon dioxide based refrigerators/air conditioners almost exclusively [Pettersen 1994]. Though they used technology ancient by today's standards, carbon dioxide machinery functioned satisfactorily and generally pleased those who purchased it. However, engineers' pursuit to improve refrigeration efficiency led to the development of fluorocarbons in the 1930s [Lorentzen 1994]. Based on the fact that cycles which employed fluorocarbons had substantially lower heat rejection pressures than those of carbon dioxide (requiring less compressor work, which in turn led to better efficiency) and was considered to be non-toxic to humans and harmless to the environment, fluorocarbons quickly replaced carbon dioxide as the refrigerating fluid of choice. As a result, carbon dioxide based machinery was phased out by the 1950s and has not been mass produced since.

A Recent Revival

Though not currently used in any production refrigerating or air conditioning equipment, carbon dioxide has been the subject of recent research as a refrigerant, and significant contributions to the field have resulted. In particular, Lorentzen and Pettersen [Lorentzen & Pettersen 1993] constructed a laboratory prototype carbon dioxide automobile air conditioner which performed as well as a standard R-12 auto air conditioner. In addition, Lorentzen has recommended other possible future uses for carbon dioxide, such as large heat pumps and commercial cooling [Lorentzen 1994]. In each proposed area of investigation, Lorentzen proposes the use of a transcritical vapor compression cycle. Since the typical environmental temperature of a summer day may reach 35°C or more, a supercritical heat rejection profile profile is required, due to the fact that the critical temperature of carbon dioxide is 30.85°C. The work of Lorentzen and Pettersen also suggests that though there are shortcomings of carbon dioxide as a refrigerant compared to the fluorocarbons (which caused carbon dioxide to be replaced in the first place), there are also many overlooked thermo-physical advantages of carbon dioxide. The initial investigations that Lorentzen and Pettersen have done show that the proper exploitation of these advantages with respect to cycle design may prove useful in not only automobile air conditioning, but in other areas as well.

Carbon Dioxide's 'New' Use

The intentions of governments worldwide to prohibit the use of CFCs (chlorofluorocarbons) and HCFCs (Hydrochlorofluorocarbons) has forced refrigeration engineers to search for new fluids to perform the cooling of enclosures such that there is no degradation of performance compared to that of the proven CFC and HCFC technology. A popular solution, and one that is currently being adopted as a U.S. standard is the employment of a fluid which is changed slightly from the chemical make-up of CFCs and HCFCs. In particular, HFCs (Hydrofluorocarbons) are presently used in newly produced refrigerating/air conditioning systems. HFCs are substances which are not reactive with ozone but retain thermal and transport properties similar to CFCs and HCFCs. Hence this new class of fluids may be used with machinery already designed for CFCs and HCFCs with minor modifications. A long term view of this series of regulations and replacement refrigerants would seem to indicate that manufactured substances (fluorocarbon-based, in particular) which are used as refrigerants are only temporary solutions to the long term need of refrigeration. In almost all cases, it is impossible to foresee the long-term environmental impact of human-manufactured fluids. The only way to be sure that a refrigerant is not harmful to the world environment is to use a substance which is naturally occurring in the earth's atmosphere. Carbon dioxide is a substance which meets the above criteria and also has other advantages as a refrigerant [Lorentzen 1994]:

1. It has a background of successful use as a refrigerant.
2. It is non-flammable and non-toxic.

3. It is cheap, plentiful, and there is no need for recovery.

4. It is compatible with normal lubricants and construction materials.

5. The volumetric refrigerating capacity of carbon dioxide is about 5 times that of R-22.

With respect to the current phase-out of CFCs and HCFCs, it seems necessary to thoroughly investigate the possibility of a revival of carbon dioxide as the refrigerant.

THERMODYNAMIC MODELING

Computer Modeling

Two computer based models of different carbon dioxide cycles were developed at the Ray W. Herrick Laboratories at Purdue University. Figure 1 reveals the thermodynamic processes for both cycles in an lnP-h diagram. Process path 1-2-3-4h indicates a transcritical simple carbon dioxide refrigeration cycle with compression (1-2), isobaric heat rejection (2-3), adiabatic expansion (3-4h), and isobaric evaporation (4h-1). Cycle process path 1-2-3-4t shows a carbon dioxide cycle where the expansion process is replaced by an expansion work output process. The process features compression (1-2), isobaric heat rejection (2-3), expansion with work recovery (3-4t) and isobaric evaporation (4t-1). The dotted lines 1-2s and 3-4s in Fig. 1 represent isentropic compression and expansion, respectively, and are presented in the figure as reference processes. In addition to the carbon dioxide models, a simple R-22 based refrigeration cycle was modeled as a reference to which the carbon dioxide cycles are compared. The property values of both carbon dioxide cycles will be obtained from an equation of state FORTRAN program and the R-22 cycle property values will be found from EES (Engineering Equation Solver) [Klein & Alvarado 1995]. As a means to assess the performance of each individual cycle, the cycle Coefficient of Performance (COP) was calculated. The COP is defined as:

$$COP = \frac{q_{evap}}{w_{in}} \tag{1}$$

q_{evap} Heat absorbed by the refrigerant in the evaporator
w_{in} Total work input to the compressor

To model the compressor and the expansion work recovery device, the isentropic and mechanical efficiencies were taken into account when calculating the values associated with work. Denoting η_{is} as the isentropic efficiency and η_m as the mechanical efficiency, the efficiencies for each device are defined as follows:

For the compressor: $\eta_{is,c} = \frac{h_{2s}-h_1}{h_2-h_1}$ $\eta_{m,c} = \frac{h_2-h_1}{w_{in,act}}$

For the expansion turbine: $\eta_{is,t} = \frac{h_3-h_4}{h_3-h_{4s}}$ $\eta_{m,t} = \frac{w_{out,act}}{h_3-h_4}$

where 'h' represents the specific enthalpy at each state.

Assumptions

In order to simplify the model processes, the following assumptions were made:

- Energy losses due to compressor and turbine inefficiencies are assumed to be heat rejected from the device to the environment.
- Changes in kinetic and potential energies are assumed negligible.
- Cycles operate at steady state.
- Refrigerants exit the evaporator as saturated vapor. R-22 exits the condenser as saturated liquid.

Conditions

In order to define the equipment capabilities and environmental factors the following conditions were assumed:

- The device efficiencies were calculated as follows:

$$\eta_{is,c} = 0.815 + 0.022(\frac{P_2}{P_1}) - 0.0041((\frac{P_2}{P_1})^2) + 0.0001((\frac{P_2}{P_1})^3)$$
$$\eta_{is,t} = 0.8$$
$$\eta_{m,c} = 0.9$$
$$\eta_{m,t} = 0.98$$

- The heat sink for all cycles is air at a constant temperature of 35°C. As a result, the condensation temperature of R-22 is assumed to be 50°C, while the outlet temperature of the carbon dioxide gas cooler is assumed to be 40°C.
- The space to be cooled ranges from -35°C to 10°C in 5°C increments, hence the evaporation temperatures range from -40°C to 5°C for both carbon dioxide and R-22.

Optimum Heat Rejection Pressures

By virtue of the established conditions, both heat source and heat sink temperatures were specified for each carbon dioxide cycle. Furthermore, since the refrigerant during evaporation was modeled as a two-phase mixture, the refrigerant maintained a constant pressure throughout the evaporation process. The final parameter required to define the carbon dioxide cycles was the heat rejection pressure. Optimal pressures were determined for each evaporator fluid temperature based on the COP, defined in equation 1. For each evaporator fluid temperature, the COP was calculated for a range of heat rejection pressures (80-120 bar) at 0.5 bar increments. Within this range of COP values, a maximum COP value existed corresponding to a single 'optimum' heat rejection pressure. Maximum COP values and optimum pressures were identified for all evaporation temperatures.

Irreversibility

In order to identify locations of energy loss within the carbon dioxide cycles, an analysis with respect to the Second Law of Thermodynamics was made on each component in the cycle. The irreversibility of the individual components was identified throughout the range of evaporation temperatures. This indicated which components are the largest factor in reducing cycle efficiency, hence indicating which areas to focus on improving with respect to cycle performance. Defining the specific irreversibility of a process operating between states 1 and 2 as:

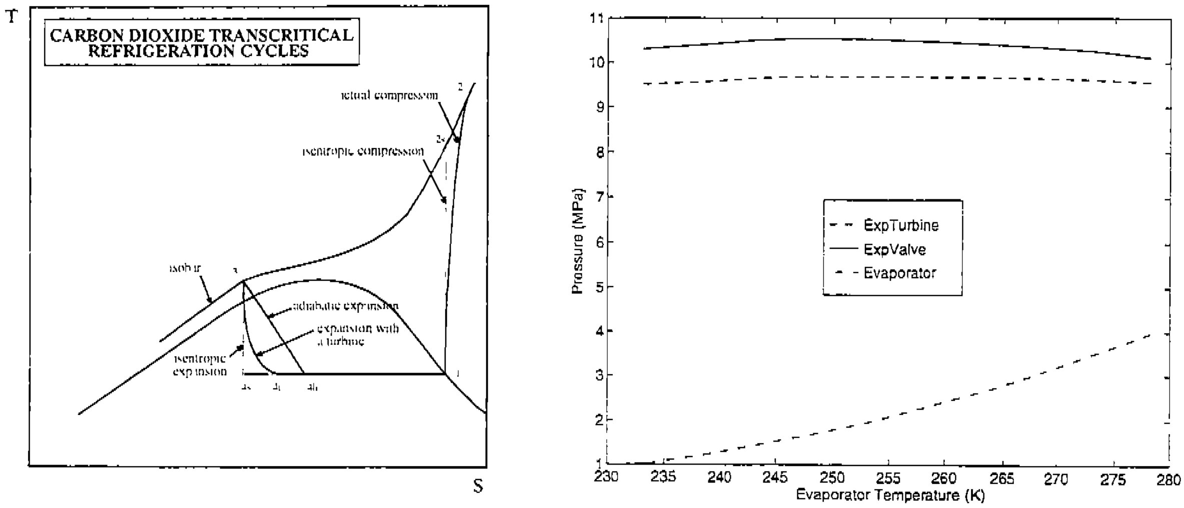
$$i_{12} = w_{act} - w_{rev} \tag{2}$$

where

$$w_{rev} = (h_2 - h_1) - To(s_2 - s_1) - q(1 - \frac{To}{Tj}) \tag{3}$$

$$w_{rev} = (h_2 - h_1) - q \tag{4}$$

w_{act}	actual specific work required to drive the cycle
w_{rev}	specific work done on or by an internally reversible process
To	ambient temperature
Tj	temperature at which the process rejects heat
q	specific heat absorbed by the process from a heat reservoir
h	specific enthalpy
s	specific entropy



(a) (b)

Figure 1: (a) T-s diagram of Carbon Dioxide Processes (b) Optimum Heat Rejection Pressures and Evaporation Pressure vs. Evaporation Temperature

The irreversibility of each component was calculated and divided by the sum of total cycle irreversibility to get the percentage of cycle irreversibility corresponding to each component.

RESULTS

The optimum heat rejection pressures and evaporation pressures for each evaporation temperature for the carbon dioxide cycles are shown in Figure 1(b). The plot indicates that the cycles' optimum heat rejection pressures do not vary widely (about 10.2-10.5 MPa for cycle with expansion work recovery, 9.5-9.7 MPa for the cycle with an expansion valve). The heat rejection pressure for R-22 at a condensing temperature of 50°C is 1.93 MPa. As can be seen by Figure 1(b), heat rejection pressures of carbon dioxide are roughly 5 times the heat rejection pressure of R-22. Due to carbon dioxide's large volumetric heat capacity (with respect to R-22), the tube diameters of a carbon dioxide based apparatus could be drastically decreased, while at the same time the tube wall thickness could be increased to accommodate the high pressures (up to 10.6 MPa). Overall, the device would still weigh less than an R-22 device of the same capacity. Another feature of the operating pressures of a carbon dioxide cycle is the lower pressure ratio with respect to an R-22 cycle. The pressure ratios of carbon dioxide vary from 2.5 to 10.0 (expansion valve cycle) and 2.4 to 9.5 (cycle with expansion work recovery) with decreasing evaporation temperatures, while the pressure ratios of R-22 vary from 2.9 to 14.9 over the same range of evaporation temperatures. Since the isentropic efficiency of a compressor is a function of the pressure ratio, a compressor properly designed for a carbon dioxide cycle could be more efficient than one designed for an R-22 cycle.

The COP values for each carbon dioxide cycle and the R-22 cycle along with the corresponding 2nd law effectiveness for each is shown in Figure 2. The 2nd law effectiveness of the carbon dioxide cycle employing an expansion valve matches the effectiveness of the R-22 cycle only at the lowest evaporator temperature. Throughout the other evaporator temperatures the effectiveness of the carbon dioxide cycle employing an expansion valve averages 70% of the effectiveness of the R-22 cycle. However, the cycle with expansion work recovery performs better than the R-22 cycle for six of the ten specified evaporator temperatures, as shown in Figure 2(b). In addition, the effectiveness of the cycle employing expansion work recovery is no less than 96% of that of the R-22 cycle for the remainder of the evaporator temperatures. Figure 2(b) also reveals that the expansion work recovery maintains an effectiveness above 0.3 for the lowest evaporator temperatures, while

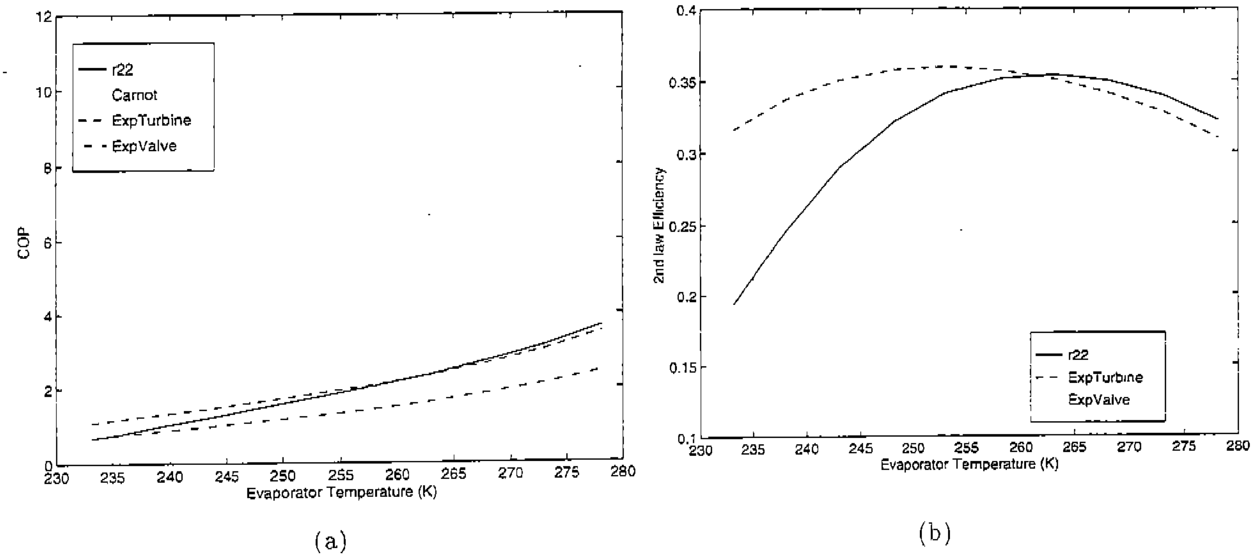


Figure 2: (a) COP vs. Evaporation Temperatures (b) 2nd Law Effectiveness vs. Evaporation Temperature

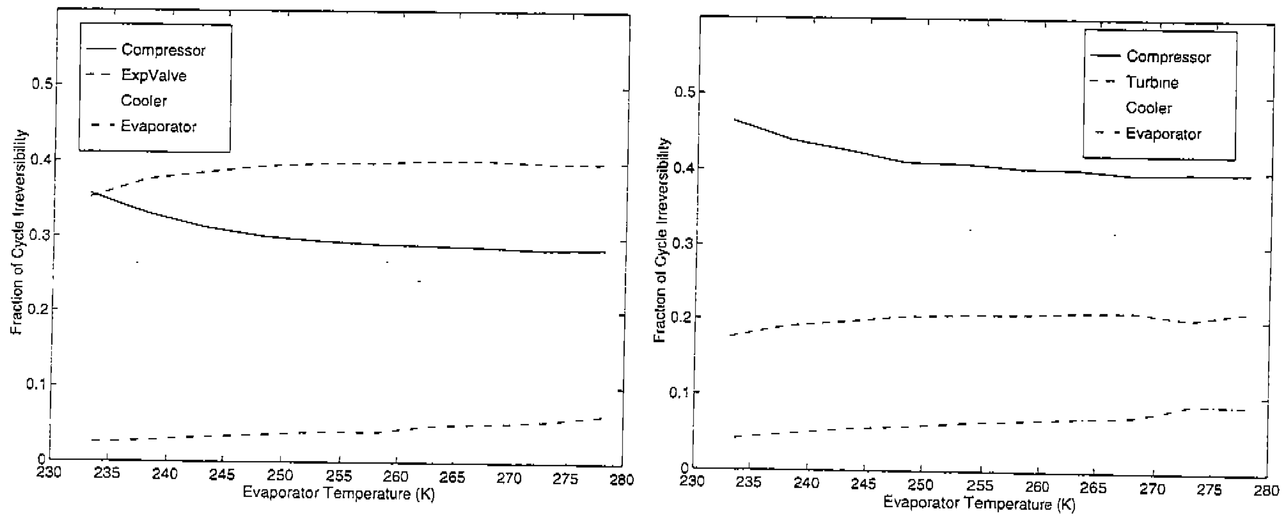
there is a significant decline in effectiveness of the R-22 cycle as evaporator the temperature decreases. The carbon dioxide cycle performs better than the R-22 cycle due to the applicability of work expansion recovery. It can make practical use of work recovery due to the fact that the inlet of the turbine is a supercritical fluid, whereas the R-22 would be a saturated or subcooled liquid. Since the amount of work recovered in the carbon dioxide cycle is a function of pressure ratio, a larger pressure ratio results in more work recovered. Hence, the carbon dioxide cycle with expansion work recovery maintains its effectiveness above 0.3 even at the lowest evaporation temperatures.

The component irreversibility percentages for the two carbon dioxide cycles are shown in Figure 3. In both cases, the evaporator accounts for roughly 5% of the cycle irreversibility throughout the range of evaporator temperatures, while the gas cooler accounts for 26% and 31% of the total cycle irreversibility for the expansion valve cycle model and the expansion work recovery cycle respectively. Though the irreversibility of the heat transfer components between the two carbon dioxide cycles are similar, the irreversibility of the expansion and compression components vary dramatically. For the cycle with an expansion valve, the expansion process produces the highest percentage of irreversibility, approximately 38% of the total cycle irreversibility. After substituting a work recovery turbine for the expansion valve, the expansion process produced the second lowest percentage of cycle irreversibility, averaging only 19% over the range of evaporator temperatures. Furthermore, the total cycle irreversibility of the cycle employing an expansion work recovery turbine is only 68% (±1% throughout the range of evaporator temperatures) that of the total cycle irreversibility of the cycle using an expansion valve.

CONCLUSIONS

Though the models produced for this research are based on theoretical conditions, and the results are somewhat generalized, some specific conclusions can be made with regard to the use of carbon dioxide as a refrigerant:

- A refrigeration cycle based on R-22 subject to the aforementioned conditions will perform better than a carbon dioxide refrigeration cycle that uses a valve for expansion. However, a carbon dioxide cycle that employs an expansion turbine subject to the aforementioned conditions performs as good or better than the R-22 baseline cycle for the specified range of evaporation temperatures.



(a) (b)

Figure 3: (a) Component Irreversibility Percentages: Basic Cycle (b) Component Irreversibility Percentages: Turbine Cycle

- Optimum heat rejection pressures for carbon dioxide refrigeration cycles equipped with both an expansion valve and an expansion work recovery device stay within ±5% of the average heat rejection pressure over the range of evaporation temperatures.

- The expansion valve is the component with the largest percentage of total irreversibility in the simple carbon dioxide cycle. Replacing the expansion valve with an expansion work recovery turbine reduces the process's contribution to total cycle irreversibility by half.

Future Work

In addition to specific conclusions drawn from the models performance, this research also suggests directions for future research:

- Turbines that are driven by supercritical and two-phase fluids do exist; investigating and optimizing the design parameters of these turbines for the given application in a carbon dioxide cycle would be a challenging topic for future research.

- Assuming the work recovery device is practical, the 2nd law analysis shows that the next largest source of irreversibility in the cycle is the compressor. To reduce this irreversibility and further improve the compression process, multi-stage compression as proposed by Linde or multiple-effect compression as proposed by Voorhees should be investigated in more detail.

- In order to design a gas cooler which maximizes the heat transfer characteristics of supercritical carbon dioxide, the transport properties in the supercritical region must be found. At this time, no literature contains the data necessary to properly design such a heat exchanger.

- Carbon dioxide has thermal and transport properties which are dramatically different than any fluorocarbon based fluid. While not suitable for all refrigerating/air conditioning applications, carbon dioxide's differences compared to the fluorocarbons could be used to a refrigeration engineer's advantage in designing with respect to a particular application. There has not yet been a thorough investigation of the range of applications which could take full advantage of the unique traits of carbon dioxide, such as the large volumetric heat capacity and the characteristic temperature heat rejection profile.

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