

Lesson

9

Air cycle refrigeration systems

The specific objectives of the lesson:

This lesson discusses various gas cycle refrigeration systems based on air, namely:

1. Reverse Carnot cycle & its limitations (*Section 9.4*)
2. Reverse Brayton cycle – Ideal & Actual (*Section 9.5*)
3. Aircraft refrigeration cycles, namely Simple system, Bootstrap system, Regenerative system, etc. (*Section 9.6*)

At the end of the lesson the student should be able to:

1. Describe various air cycle refrigeration systems (*Section 9.1-9.6*)
2. State the assumptions made in the analyses of air cycle systems (*Section 9.2*)
3. Show the cycles on T-s diagrams (*Section 9.4-9.6*)
4. Perform various cycle calculations (*Section 9.3-9.6*)
5. State the significance of Dry Air Rated Temperature (*Section 9.6*)

9.1. Introduction

Air cycle refrigeration systems belong to the general class of gas cycle refrigeration systems, in which a gas is used as the working fluid. The gas does not undergo any phase change during the cycle, consequently, all the internal heat transfer processes are sensible heat transfer processes. Gas cycle refrigeration systems find applications in air craft cabin cooling and also in the liquefaction of various gases. In the present chapter gas cycle refrigeration systems based on air are discussed.

9.2. Air Standard Cycle analysis

Air cycle refrigeration system analysis is considerably simplified if one makes the following assumptions:

- i. The working fluid is a fixed mass of air that behaves as an ideal gas
- ii. The cycle is assumed to be a closed loop cycle with all inlet and exhaust processes of open loop cycles being replaced by heat transfer processes to or from the environment
- iii. All the processes within the cycle are reversible, i.e., the cycle is internally reversible
- iv. The specific heat of air remains constant throughout the cycle

An analysis with the above assumptions is called as cold Air Standard Cycle (ASC) analysis. This analysis yields reasonably accurate results for most of the cycles and processes encountered in air cycle refrigeration systems. However, the analysis fails when one considers a cycle consisting of a throttling process, as the temperature drop during throttling is zero for an ideal gas, whereas the actual cycles depend exclusively on the real gas behavior to produce refrigeration during throttling.

9.3. Basic concepts

The temperature of an ideal gas can be reduced either by making the gas to do work in an isentropic process or by sensible heat exchange with a cooler environment. When the gas does adiabatic work in a closed system by say, expanding against a piston, its internal energy drops. Since the internal energy of the ideal gas depends only on its temperature, the temperature of the gas also drops during the process, i.e.,

$$W = m(u_1 - u_2) = mc_v(T_1 - T_2) \quad (9.1)$$

where m is the mass of the gas, u_1 and u_2 are the initial and final internal energies of the gas, T_1 and T_2 are the initial and final temperatures and c_v is the specific heat at constant volume. If the expansion is reversible and adiabatic, by using the ideal gas equation $Pv = RT$ and the equation for isentropic process $P_1v_1^\gamma = P_2v_2^\gamma$ the final temperature (T_2) is related to the initial temperature (T_1) and initial and final pressures (P_1 and P_2) by the equation:

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} \quad (9.2)$$

where γ is the coefficient of isentropic expansion given by:

$$\gamma = \left(\frac{c_p}{c_v} \right) \quad (9.3)$$

Isentropic expansion of the gas can also be carried out in a steady flow in a turbine which gives a net work output. Neglecting potential and kinetic energy changes, the work output of the turbine is given by:

$$W = \dot{m}(h_1 - h_2) = \dot{m}c_p(T_1 - T_2) \quad (9.4)$$

The final temperature is related to the initial temperature and initial and final pressures by Eq. (9.2).

9.4. Reversed Carnot cycle employing a gas

Reversed Carnot cycle is an ideal refrigeration cycle for constant temperature external heat source and heat sinks. Figure 9.1(a) shows the schematic of a reversed Carnot refrigeration system using a gas as the working fluid along with the cycle diagram on T-s and P-v coordinates. As shown, the cycle consists of the following four processes:

Process 1-2: Reversible, adiabatic compression in a compressor

Process 2-3: Reversible, isothermal heat rejection in a compressor

Process 3-4: Reversible, adiabatic expansion in a turbine

Process 4-1: Reversible, isothermal heat absorption in a turbine

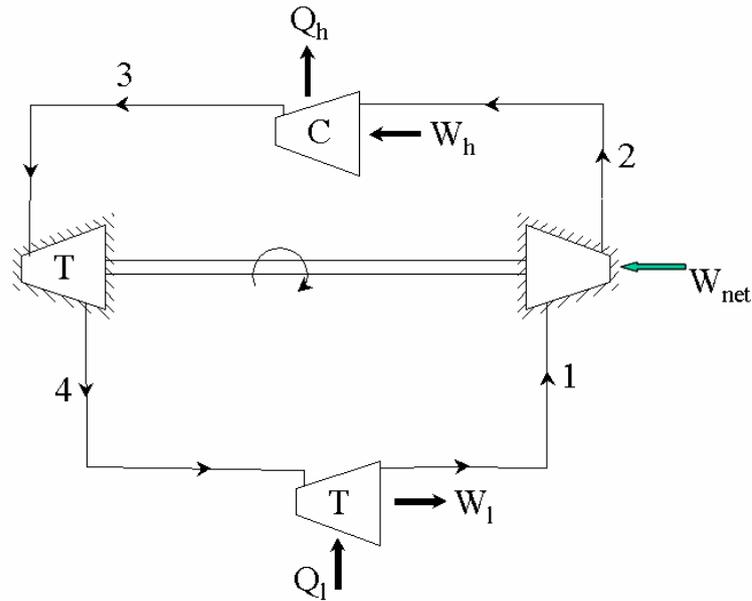


Fig. 9.1(a). Schematic of a reverse Carnot refrigeration system

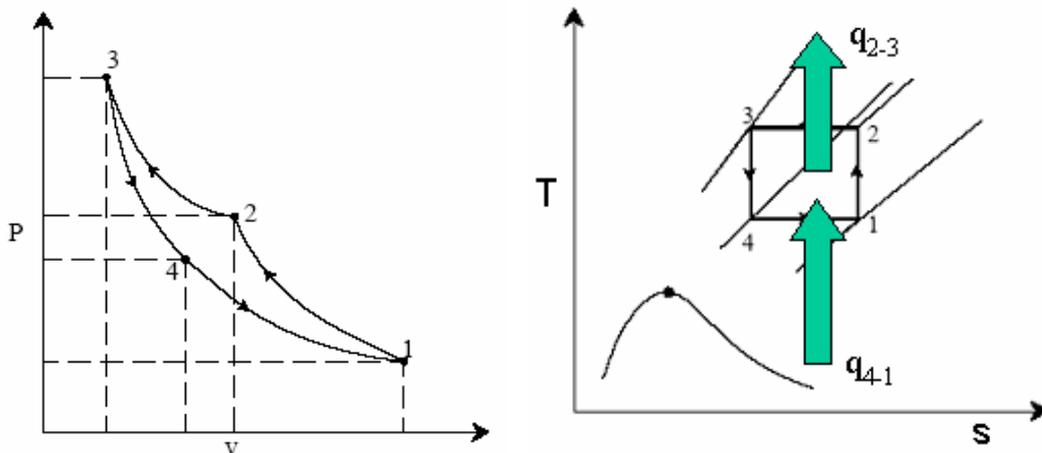


Fig. 9.1(b). Reverse Carnot refrigeration system in P - v and T - s coordinates

The heat transferred during isothermal processes 2-3 and 4-1 are given by:

$$q_{2-3} = \int_2^3 T \cdot ds = T_h (s_3 - s_2) \quad (9.5a)$$

$$q_{4-1} = \int_4^1 T \cdot ds = T_l (s_1 - s_4) \quad (9.5b)$$

$$s_1 = s_2 \quad \text{and} \quad s_3 = s_4, \quad \text{hence} \quad s_2 - s_3 = s_1 - s_4 \quad (9.6)$$

Applying first law of thermodynamics to the closed cycle,

$$\oint \delta q = (q_{4-1} + q_{2-3}) = \oint \delta w = (w_{2-3} - w_{4-1}) = -w_{\text{net}} \quad (9.7)$$

the work of isentropic expansion, w_{3-4} exactly matches the work of isentropic compression w_{1-2} .

the COP of the Carnot system is given by:

$$\text{COP}_{\text{Carnot}} = \left| \frac{q_{4-1}}{w_{\text{net}}} \right| = \left(\frac{T_1}{T_h - T_1} \right) \quad (9.8)$$

Thus the COP of the Carnot system depends only on the refrigeration (T_1) and heat rejection (T_h) temperatures only.

Limitations of Carnot cycle:

Carnot cycle is an idealization and it suffers from several practical limitations. One of the main difficulties with Carnot cycle employing a gas is the difficulty of achieving isothermal heat transfer during processes 2-3 and 4-1. For a gas to have heat transfer isothermally, it is essential to carry out work transfer from or to the system when heat is transferred to the system (process 4-1) or from the system (process 2-3). This is difficult to achieve in practice. In addition, the volumetric refrigeration capacity of the Carnot system is very small leading to large compressor displacement, which gives rise to large frictional effects. All actual processes are irreversible, hence completely reversible cycles are idealizations only.

9.5. Ideal reverse Brayton cycle

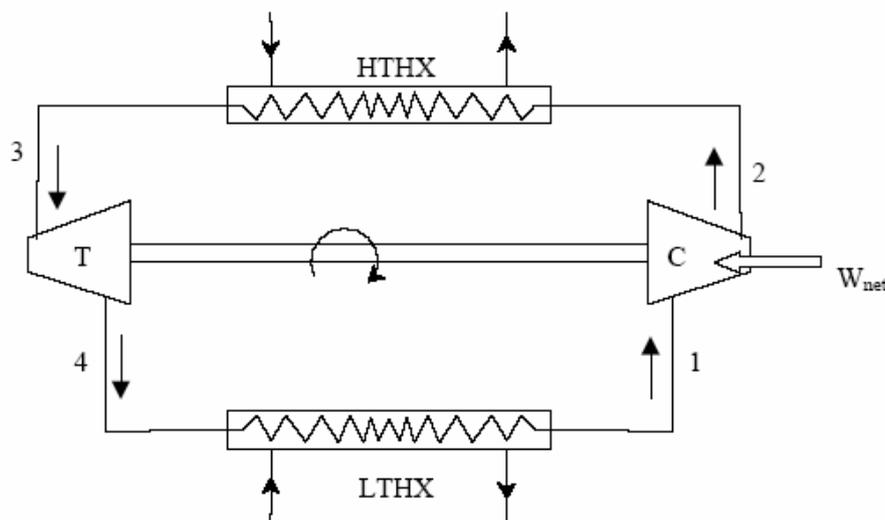


Fig. 9.2(a). Schematic of a closed reverse Brayton cycle

This is an important cycle frequently employed in gas cycle refrigeration systems. This may be thought of as a modification of reversed Carnot cycle, as the two isothermal processes of Carnot cycle are replaced by two isobaric heat transfer processes. This cycle is also called as Joule or Bell-Coleman cycle. Figure 9.2(a) and (b) shows the schematic of a closed, reverse Brayton cycle and also the cycle on T-s

diagram. As shown in the figure, the ideal cycle consists of the following four processes:

Process 1-2: Reversible, adiabatic compression in a compressor

Process 2-3: Reversible, isobaric heat rejection in a heat exchanger

Process 3-4: Reversible, adiabatic expansion in a turbine

Process 4-1: Reversible, isobaric heat absorption in a heat exchanger

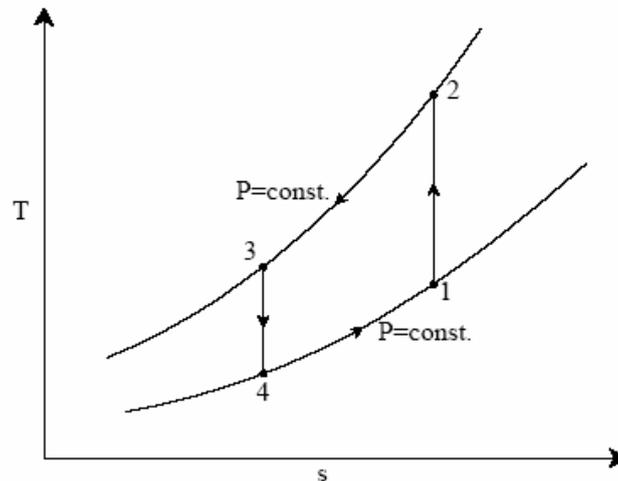


Fig. 9.2(b). Reverse Brayton cycle in T - s plane

Process 1-2: Gas at low pressure is compressed isentropically from state 1 to state 2. Applying steady flow energy equation and neglecting changes in kinetic and potential energy, we can write:

$$W_{1-2} = \dot{m}(h_2 - h_1) = \dot{m} c_p (T_2 - T_1)$$

$$s_2 = s_1 \quad (9.9)$$

$$\text{and } T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} = T_1 r_p^{\frac{\gamma-1}{\gamma}}$$

where $r_p = (P_2/P_1)$ = pressure ratio

Process 2-3: Hot and high pressure gas flows through a heat exchanger and rejects heat sensibly and isobarically to a heat sink. The enthalpy and temperature of the gas drop during the process due to heat exchange, no work transfer takes place and the entropy of the gas decreases. Again applying steady flow energy equation and second $T ds$ equation:

$$Q_{2-3} = \dot{m}(h_2 - h_3) = \dot{m} c_p (T_2 - T_3)$$

$$s_2 - s_3 = c_p \ln \frac{T_2}{T_3} \quad (9.10)$$

$$P_2 = P_3$$

Process 3-4: High pressure gas from the heat exchanger flows through a turbine, undergoes isentropic expansion and delivers net work output. The temperature of the gas drops during the process from T_3 to T_4 . From steady flow energy equation:

$$\begin{aligned} W_{3-4} &= \dot{m}(h_3 - h_4) = \dot{m} c_p (T_3 - T_4) \\ s_3 &= s_4 \\ \text{and } T_3 &= T_4 \left(\frac{P_3}{P_4} \right)^{\frac{\gamma-1}{\gamma}} = T_4 r_p^{\frac{\gamma-1}{\gamma}} \end{aligned} \quad (9.11)$$

where $r_p = (P_3/P_4) =$ pressure ratio

Process 4-1: Cold and low pressure gas from turbine flows through the low temperature heat exchanger and extracts heat sensibly and isobarically from a heat source, providing a useful refrigeration effect. The enthalpy and temperature of the gas rise during the process due to heat exchange, no work transfer takes place and the entropy of the gas increases. Again applying steady flow energy equation and second T ds equation:

$$\begin{aligned} Q_{4-1} &= \dot{m}(h_1 - h_4) = \dot{m} c_p (T_1 - T_4) \\ s_4 - s_1 &= c_p \ln \frac{T_4}{T_1} \\ P_4 &= P_1 \end{aligned} \quad (9.12)$$

From the above equations, it can be easily shown that:

$$\left(\frac{T_2}{T_1} \right) = \left(\frac{T_3}{T_4} \right) \quad (9.13)$$

Applying 1st law of thermodynamics to the entire cycle:

$$\oint \delta q = (q_{4-1} - q_{2-3}) = \oint \delta w = (w_{3-4} - w_{1-2}) = -w_{\text{net}} \quad (9.14)$$

The COP of the reverse Brayton cycle is given by:

$$\text{COP} = \left| \frac{q_{4-1}}{w_{\text{net}}} \right| = \left(\frac{(T_1 - T_4)}{(T_2 - T_1) - (T_3 - T_4)} \right) \quad (9.15)$$

using the relation between temperatures and pressures, the COP can also be written as:

$$\text{COP} = \left(\frac{(T_1 - T_4)}{(T_2 - T_1) - (T_3 - T_4)} \right) = \left(\frac{T_4}{T_3 - T_4} \right) = \left(\frac{(T_1 - T_4)}{(T_1 - T_4)(r_p^{\frac{\gamma-1}{\gamma}} - 1)} \right) = (r_p^{\frac{\gamma-1}{\gamma}} - 1)^{-1} \quad (9.16)$$

From the above expression for COP, the following observations can be made:

- a) For fixed heat rejection temperature (T_3) and fixed refrigeration temperature (T_1), the COP of reverse Brayton cycle is always lower than the COP of reverse Carnot cycle (Fig. 9.3), that is

$$\text{COP}_{\text{Brayton}} = \left(\frac{T_4}{T_3 - T_4} \right) < \text{COP}_{\text{Carnot}} = \left(\frac{T_1}{T_3 - T_1} \right)$$

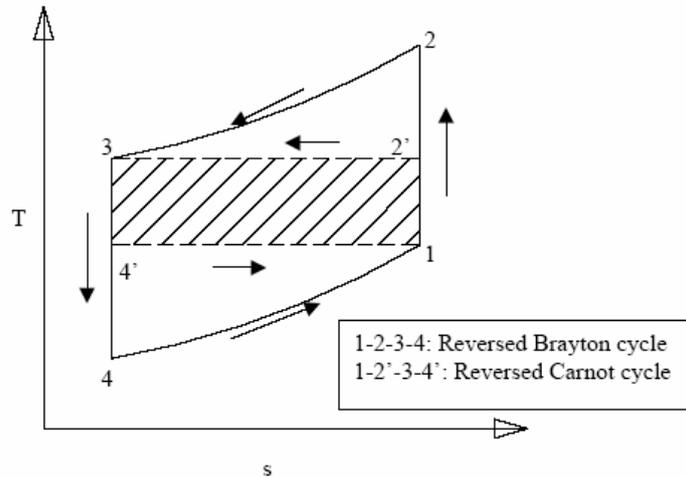


Fig. 9.3. Comparison of reverse Carnot and reverse Brayton cycle in T - s plane

- b) COP of Brayton cycle approaches COP of Carnot cycle as T_1 approaches T_4 (thin cycle), however, the specific refrigeration effect [$c_p(T_1 - T_4)$] also reduces simultaneously.
- c) COP of reverse Brayton cycle decreases as the pressure ratio r_p increases

Actual reverse Brayton cycle:

The actual reverse Brayton cycle differs from the ideal cycle due to:

- i. Non-isentropic compression and expansion processes
- ii. Pressure drops in cold and hot heat exchangers

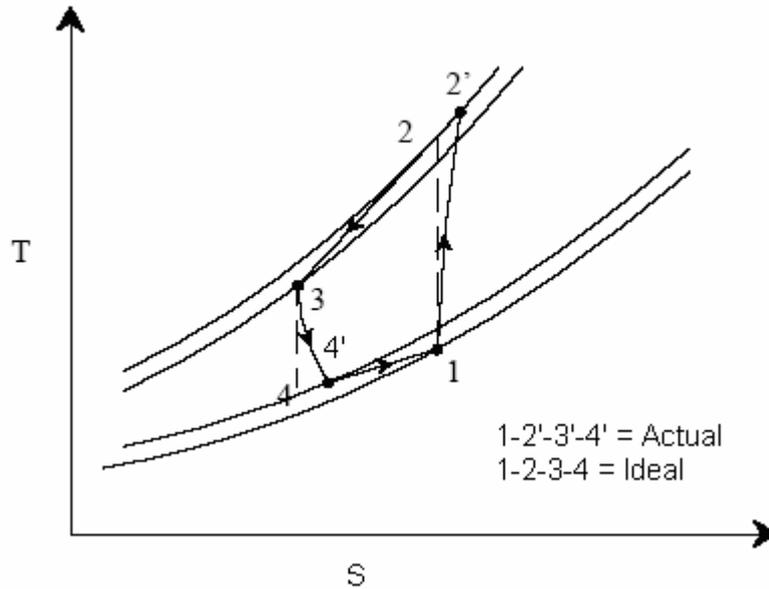


Fig. 9.4. Comparison of ideal and actual Brayton cycles T-s plane

Figure 9.4 shows the ideal and actual cycles on T-s diagram. Due to these irreversibilities, the compressor work input increases and turbine work output reduces. The actual work transfer rates of compressor and turbine are then given by:

$$W_{1-2,act} = \frac{W_{1-2,isen}}{\eta_{c,isen}} \quad (9.17)$$

$$W_{3-4,act} = \eta_{t,isen} W_{3-4,isen} \quad (9.18)$$

where $\eta_{c,isen}$ and $\eta_{t,isen}$ are the isentropic efficiencies of compressor and turbine, respectively. In the absence of pressure drops, these are defined as:

$$\eta_{c,isen} = \frac{(h_2 - h_1)}{(h_{2'} - h_1)} = \frac{(T_2 - T_1)}{(T_{2'} - T_1)} \quad (9.20)$$

$$\eta_{t,isen} = \frac{(h_3 - h_4)}{(h_3 - h_{4'})} = \frac{(T_3 - T_4)}{(T_3 - T_4')} \quad (9.21)$$

The actual net work input, $w_{net,act}$ is given by:

$$W_{net,act} = W_{1-2,act} - W_{3-4,act} \quad (9.22)$$

thus the net work input increases due to increase in compressor work input and reduction in turbine work output. The refrigeration effect also reduces due to the irreversibilities. As a result, the COP of actual reverse Brayton cycles will be considerably lower than the ideal cycles. Design of efficient compressors and turbines plays a major role in improving the COP of the system.

In practice, reverse Brayton cycles can be open or closed. In open systems, cold air at the exit of the turbine flows into a room or cabin (cold space), and air to the

compressor is taken from the cold space. In such a case, the low side pressure will be atmospheric. In closed systems, the same gas (air) flows through the cycle in a closed manner. In such cases it is possible to have low side pressures greater than atmospheric. These systems are known as *dense air systems*. Dense air systems are advantageous as it is possible to reduce the volume of air handled by the compressor and turbine at high pressures. Efficiency will also be high due to smaller pressure ratios. It is also possible to use gases other than air (e.g. helium) in closed systems.

9.6. Aircraft cooling systems

In an aircraft, cooling systems are required to keep the cabin temperatures at a comfortable level. Even though the outside temperatures are very low at high altitudes, still cooling of cabin is required due to:

- i. Large internal heat generation due to occupants, equipment etc.
- ii. Heat generation due to skin friction caused by the fast moving aircraft
- iii. At high altitudes, the outside pressure will be sub-atmospheric. When air at this low pressure is compressed and supplied to the cabin at pressures close to atmospheric, the temperature increases significantly. For example, when outside air at a pressure of 0.2 bar and temperature of 223 K (at 10000 m altitude) is compressed to 1 bar, its temperature increases to about 353 K. If the cabin is maintained at 0.8 bar, the temperature will be about 332 K. This effect is called as ram effect. This effect adds heat to the cabin, which needs to be taken out by the cooling system.
- iv. Solar radiation

For low speed aircraft flying at low altitudes, cooling system may not be required, however, for high speed aircraft flying at high altitudes, a cooling system is a must.

Even though the COP of air cycle refrigeration is very low compared to vapour compression refrigeration systems, it is still found to be most suitable for aircraft refrigeration systems as:

- i. Air is cheap, safe, non-toxic and non-flammable. Leakage of air is not a problem
- ii. Cold air can directly be used for cooling thus eliminating the low temperature heat exchanger (open systems) leading to lower weight
- iii. The aircraft engine already consists of a high speed turbo-compressor, hence separate compressor for cooling system is not required. This reduces the weight per kW cooling considerably. Typically, less than 50% of an equivalent vapour compression system
- iv. Design of the complete system is much simpler due to low pressures. Maintenance required is also less.

9.6.1. Simple aircraft refrigeration cycle:

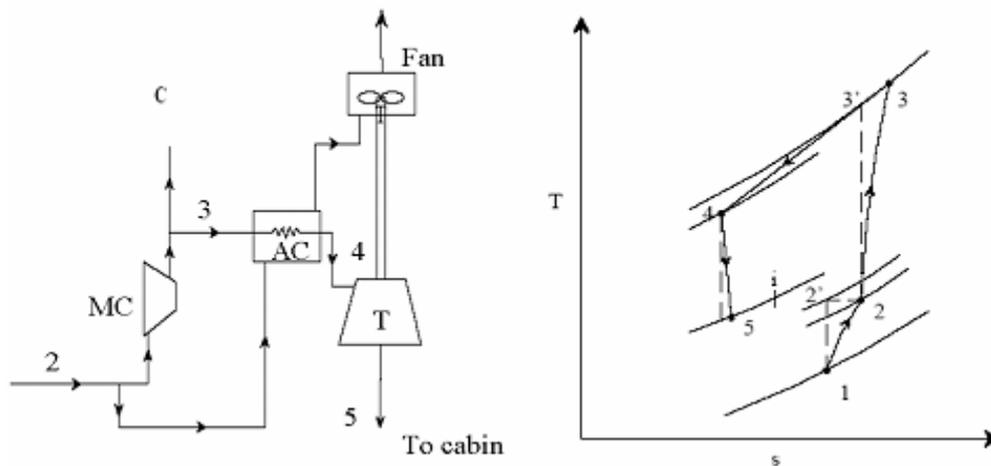


Fig. 9.5. Schematic of a simple aircraft refrigeration cycle

Figure 9.5 shows the schematic of a simple aircraft refrigeration system and the operating cycle on T-s diagram. This is an open system. As shown in the T-s diagram, the outside low pressure and low temperature air (state 1) is compressed due to ram effect to ram pressure (state 2). During this process its temperature increases from 1 to 2. This air is compressed in the main compressor to state 3, and is cooled to state 4 in the air cooler. Its pressure is reduced to cabin pressure in the turbine (state 5), as a result its temperature drops from 4 to 5. The cold air at state 5 is supplied to the cabin. It picks up heat as it flows through the cabin providing useful cooling effect. The power output of the turbine is used to drive the fan, which maintains the required air flow over the air cooler. This simple system is good for ground cooling (when the aircraft is not moving) as fan can continue to maintain airflow over the air cooler.

By applying steady flow energy equation to the ramming process, the temperature rise at the end of the ram effect can be shown to be:

$$\frac{T_2}{T_1} = 1 + \frac{\gamma - 1}{2} M^2 \quad (9.23)$$

where M is the Mach number, which is the ratio of velocity of the aircraft (C) to the sonic velocity a ($a = \sqrt{\gamma RT_1}$), i.e.,

$$M = \frac{C}{a} = \frac{C}{\sqrt{\gamma RT_1}} \quad (9.24)$$

Due to irreversibilities, the actual pressure at the end of ramming will be less than the pressure resulting from isentropic compression. The ratio of actual pressure rise to the isentropic pressure rise is called as ram efficiency, η_{Ram} , i.e.,

$$\eta_{\text{Ram}} = \frac{(P_2 - P_1)}{(P_{2'} - P_1)} \quad (9.25)$$

The refrigeration capacity of the simple aircraft cycle discussed, \dot{Q} is given by:

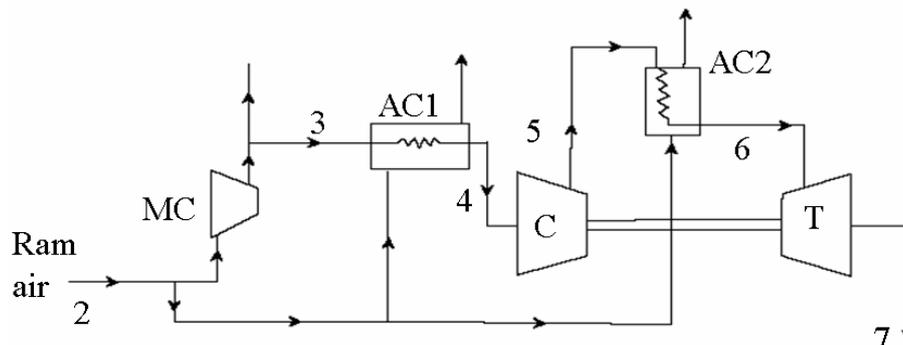
$$\dot{Q} = \dot{m} c_p (T_i - T_5) \quad (9.26)$$

where \dot{m} is the mass flow rate of air through the turbine.

9.6.2. Bootstrap system:

Figure 9.6 shows the schematic of a bootstrap system, which is a modification of the simple system. As shown in the figure, this system consists of two heat exchangers (air cooler and aftercooler), instead of one air cooler of the simple system. It also incorporates a secondary compressor, which is driven by the turbine of the cooling system. This system is suitable for high speed aircraft, where in the velocity of the aircraft provides the necessary airflow for the heat exchangers, as a result a separate fan is not required. As shown in the cycle diagram, ambient air state 1 is pressurized to state 2 due to the ram effect. This air is further compressed to state 3 in the main compressor. The air is then cooled to state 4 in the air cooler. The heat rejected in the air cooler is absorbed by the ram air at state 2. The air from the air cooler is further compressed from state 4 to state 5 in the secondary compressor. It is then cooled to state 6 in the after cooler, expanded to cabin pressure in the cooling turbine and is supplied to the cabin at a low temperature T_7 . Since the system does not consist of a separate fan for driving the air through the heat exchangers, it is not suitable for ground cooling. However, in general ground cooling is normally done by an external air conditioning system as it is not efficient to run the aircraft engine just to provide cooling when it is grounded.

Other modifications over the simple system are: regenerative system and reduced ambient system. In a regenerative system, a part of the cold air from the cooling turbine is used for precooling the air entering the turbine. As a result much lower temperatures are obtained at the exit of the cooling turbine, however, this is at the expense of additional weight and design complexity. The cooling turbine drives a fan similar to the simple system. The regenerative system is good for both ground cooling as well as high speed aircrafts. The reduced ambient system is well-suited for supersonic aircrafts and rockets.



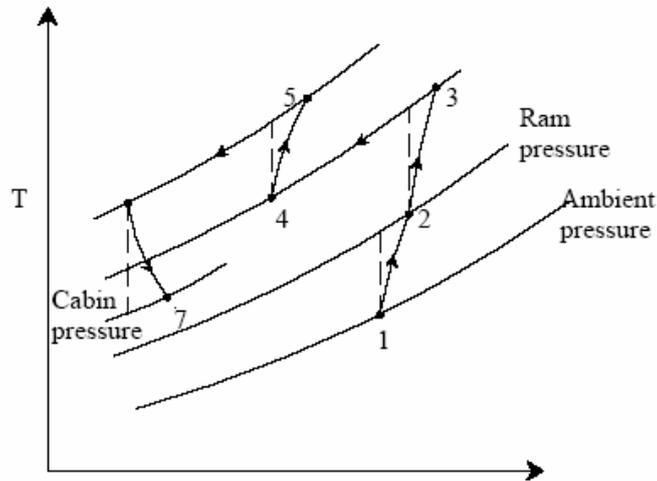


Fig. 9.6. Schematic of a bootstrap system

Dry Air Rated Temperature (DART):

The concept of Dry Air Rated Temperature is used to compare different aircraft refrigeration cycles. Dry Air Rated Temperature is defined as the temperature of the air at the exit of the cooling turbine in the absence of moisture condensation. For condensation not to occur during expansion in turbine, the dew point temperature and hence moisture content of the air should be very low, i.e., the air should be very dry. The aircraft refrigeration systems are rated based on the mass flow rate of air at the design DART. The cooling capacity is then given by:

$$\dot{Q} = \dot{m} c_p (T_i - T_{\text{DART}}) \quad (9.27)$$

where \dot{m} is the mass flow rate of air, T_{DART} and T_i are the dry air rated temperature and cabin temperature, respectively.

A comparison between different aircraft refrigeration systems based on DART at different Mach numbers shows that:

- i. DART increases monotonically with Mach number for all the systems except the reduced ambient system
- ii. The simple system is adequate at low Mach numbers
- iii. At high Mach numbers either bootstrap system or regenerative system should be used
- iv. Reduced ambient temperature system is best suited for very high Mach number, supersonic aircrafts

Questions:

1. A refrigerator working on Bell-Coleman cycle (Reverse brayton cycle) operates between 1 bar and 10 bar. Air is drawn from cold chamber at -10°C . Air coming out of compressor is cooled to 50°C before entering the expansion cylinder. Polytropic law $P.V^{1.3} = \text{constant}$ is followed during expansion and compression. Find theoretical C.O.P of the origin. Take $\gamma = 1.4$ and $C_p = 1.00 \text{ kJ/kg } ^{\circ}\text{C}$ for air. ([Solution](#))

2. An air refrigerator working on the principle of Bell-Coleman cycle. The air into the compressor is at 1 atm at -10°C . It is compressed to 10 atm and cooled to 40°C at the same pressure. It is then expanded to 1 atm and discharged to take cooling load. The air circulation is 1 kg/s.

The isentropic efficiency of the compressor = 80%

The isentropic efficiency of the expander = 90%

Find the following:

- i) Refrigeration capacity of the system
- ii) C.O.P of the system

Take $\gamma = 1.4$, $C_p = 1.00 \text{ kJ/kg } ^{\circ}\text{C}$ ([Solution](#))

3. A Carnot refrigerator extracts 150 kJ of heat per minute from a space which is maintained at -20°C and is discharged to atmosphere at 45°C . Find the work required to run the unit. ([Solution](#))

4. A cold storage plant is required to store 50 tons of fish.

The temperature at which fish was supplied = 35°C

Storage temperature of fish = -10°C

C_p of fish above freezing point = $2.94 \text{ kJ/kg } ^{\circ}\text{C}$

C_p of fish below freezing point = $1.26 \text{ kJ/kg } ^{\circ}\text{C}$

Freezing point of fish = -5°C

Latent heat of fish = 250 kJ/kg

If the cooling is achieved within half of a day, find:

- a) Capacity of the refrigerating plant
- b) Carnot COP
- c) If actual $\text{COP} = \frac{\text{Carnot COP}}{2.5}$ find the power required to run the plant.

([Solution](#))

5. A boot strap cooling system of 10 tons is used in an aeroplane. The temperature and pressure conditions of atmosphere are 20°C and 0.9 atm. The pressure of air is increased from 0.9 atm to 1.1 atm due to ramming. The pressures of air leaving the main and auxiliary compressor are 3 atm and 4 atm respectively. Isentropic efficiency of compressors and turbine are 0.85 and 0.8 respectively. 50% of the total heat of air leaving the main compressor is removed in the first heat exchanger and 30% of their

total heat of air leaving the auxiliary compressor is removed in the second heat exchanger using removed air. Find:

- a) Power required to take cabin load
- b) COP of the system

The cabin pressure is 1.02 atm and temperature of air leaving the cabin should be greater than 25°C. Assume ramming action to be isentropic. ([Solution](#))

6. A simple air cooled system is used for an aeroplane to take a load of 10 tons. Atmospheric temperature and pressure is 25°C and 0.9 atm respectively. Due to ramming the pressure of air is increased from 0.9 atm, to 1 atm. The pressure of air leaving the main compressor is 3.5 atm and its 50% heat is removed in the air-cooled heat exchanger and then it is passed through a evaporator for future cooling. The temperature of air is reduced by 10°C in the evaporator. Lastly the air is passed through cooling turbine and is supplied to the cooling cabin where the pressure is 1.03 atm. Assuming isentropic efficiency of the compressor and turbine are 75% and 70%, find

- a) Power required to take the load in the cooling cabin
- b) COP of the system.

The temperature of air leaving the cabin should not exceed 25°C. ([Solution](#))

7. True and False

1. COP of a Carnot system depends only on the refrigeration and heat rejection temperatures only. ([Answer](#))
2. As heat transfer from a gas can be done isothermally, Carnot cycle is easy to implement practically. ([Answer](#))
3. For a fixed heat rejection and refrigeration temperature, the COP of a brayton cycle is lower than COP of reverse Carnot cycle. ([Answer](#))
4. Efficiency of dense air systems are low as operating pressures are higher ([Answer](#))
5. DART is the temperature of the air at the exit of the cooling turbine. ([Answer](#))
6. A Simple system is adequate to handle high Mach numbers. ([Answer](#))