Air Cycle Refrigeration Systems

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The specific objectives of the lesson

This lesson discusses various gas cycle refrigeration systems based on air, namely:
1. Reverse Carnot cycle & its limitations
2. Reverse Brayton cycle – Ideal & Actual
3. Aircraft refrigeration cycles, namely Simple system, Bootstrap system, Regenerative system, etc.

1.1. 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.

1.2. 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 inlets 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 suspicions is called as frosty Air Standard Cycle (ASC) investigation. This analysis yields sensibly exact results for a large portion of the cycles and methodologies experienced in air cycle refrigeration frameworks. Then again, the examination fizzles when one considers a cycle comprising of a throttling procedure, as the temperature drop amid throttling is zero for a perfect gas, while the real cycles depend only on the genuine gas conduct to create refrigeration amid throttling.

1.3. Basic concepts

The temperature of a perfect gas can be decreased either by making the gas to do work in an isentropic methodology or by sensible warmth trade with a cooler domain. At the point when the gas does adiabatic work in a shut framework by say, growing against a cylinder, its inner vitality drops. Since the inner vitality of the perfect gas depends just on its temperature, the temperature of the gas additionally drops amid the procedure, i.e.,

$$W = m(u1 - u2) = mcv(T1 - T2) \quad (1.1)$$

Where m is the mass of the gas, u1 and u2 are the initial and final internal energies of the gas, T1 and T2 are the initial and final temperatures and cv is the specific heat at constant volume. If the expansion is reversible and adiabatic, by using the ideal gas equation and the equation for isentropic process Pv = RT P1v1 γ =P2v2 γ , the final temperature (T2) is related to the initial temperature (T1) and initial and final pressures (P1 and P2) by the equation:

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \quad (1.2)$$

Where γ is the coefficient of isentropic expansion given by:

$$\gamma = \left(\frac{c_p}{c_v}\right) \quad (1.3)$$

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:

$$W = m(h_1 - h_2) = mcp\,(T1\ T2) \quad (1.4)$$

The final temperature is related to the initial temperature and initial and final pressures by Eq. (1.2).

1.4. Reversed Carnot cycle employing a gas

Reversed Carnot cycle is an ideal refrigeration cycle for constant temperature external heat source and heat sinks. Figure 1.1(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

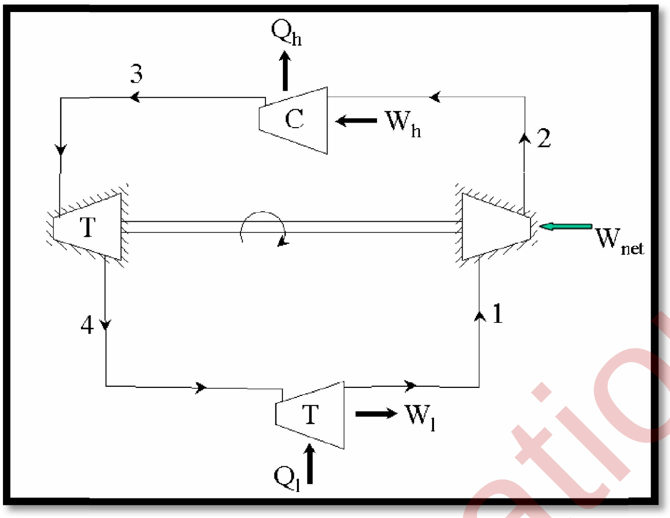


Fig. 1.1(a). Schematic of a reverse Carnot refrigeration system

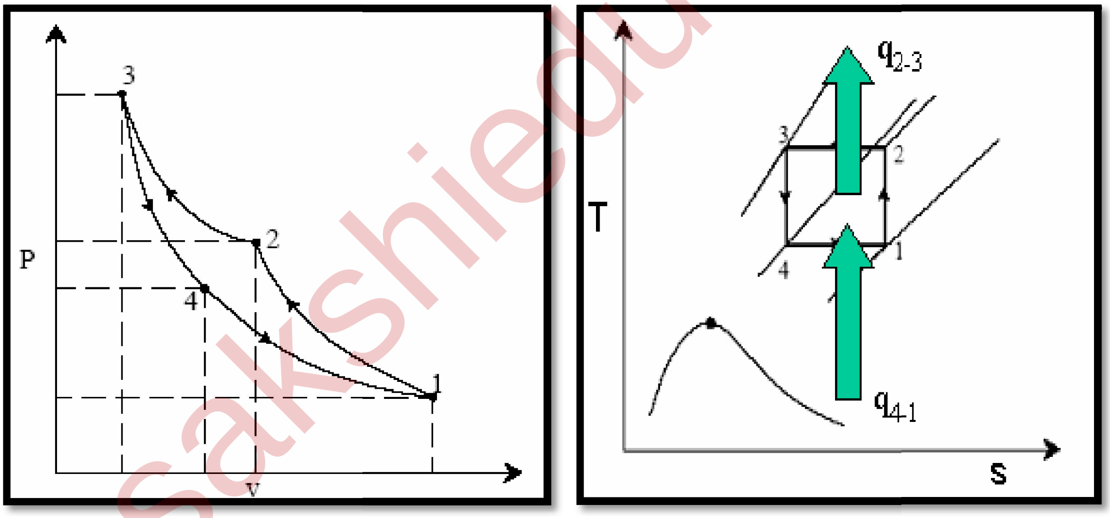


Fig.**1.1 (b)**. Reverse Carnot refrigeration system in P-v and T-s coordinates

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

$$q_{2-3} = \int_2^3 T.ds = T_h(s_3 - s_2)$$

$$q_{4-1} = \int_4^1 T.ds = T_l(s_1 - s_4)$$

$$s_1 = s_2 \text{ and } s_3 = s_4, \text{ hence } s_2 - s_3 = s_1 - s_4$$

Applying first law of thermodynamics to the closed cycle,

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

The work of isentropic expansion, w3-4 exactly matches the work of isentropic compression w1-2. The COP of the Carnot system is given by:

$$COP_{Carnot} = \left|\frac{q_{4-1}}{w_{net}}\right| = \left(\frac{T_l}{T_h - T_l}\right)$$

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

Limitations of Carnot cycle:

Carnot cycle is an idealization and it experiences a few functional restrictions. One of the primary troubles with Carnot cycle utilizing a gas is the trouble of accomplishing isothermal warmth exchange amid methodologies 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.

1.5. Ideal Reverses Brayton cycle

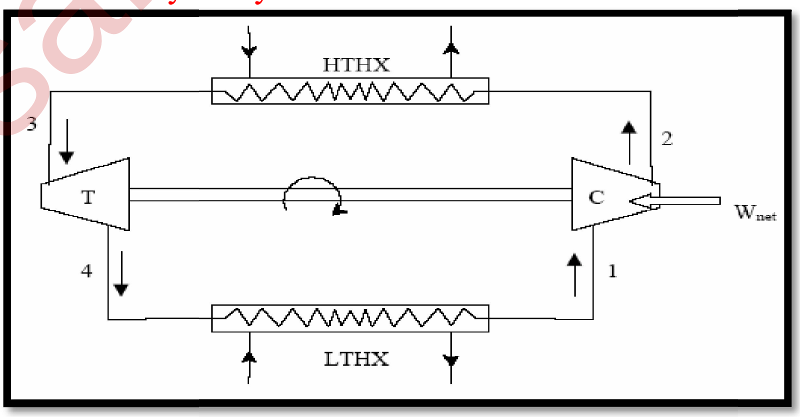


Fig.1.2 (a). Schematic of a closed reverse Brayton cycle

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 1.2(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

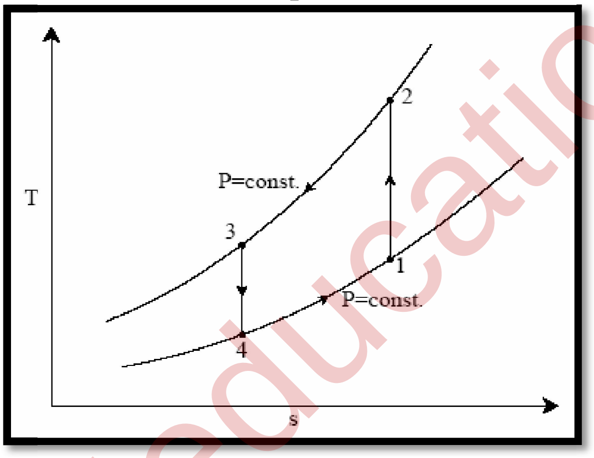


Fig. 1.2 (b). Reverse Brayton cycle in T-s plane

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:

$$W_{1-2} = \dot{m}(h_2 - h_1) = \dot{m} c_p (T_2 - T_1)$$
$$s_2 = s_1$$
$$\text{and } T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = T_1 r_p^{\frac{\gamma-1}{\gamma}}$$

Where rp = (P2/P1) = pressure ratio

Process 2-3: Hot and high weight gas moves through a warmth exchanger and rejects warm sensibly and isobarially to a warmth sink. The enthalpy and temperature of the gas drop amid the methodology because of warmth trade, no work exchange happens and the entropy of the gas diminishes. Again applying relentless stream vitality mathematical statement and second

$$Q_{2-3} = \dot{m}(h_2 - h_3) = \dot{m} c_p (T_2 - T_3)$$
$$s_2 - s_3 = c_p \ln \frac{T_2}{T_3}$$
$$P_2 = P_3$$

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 T3 to T4. From steady flow energy equation:

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

Where rp = (P3/P4) = pressure ratio

Process 4-1: Cold and low pressure gas from turbine courses through the low temperature heat exchanger and concentrates warm sensibly and isobarically from a warmth source, giving a helpful refrigeration impact. 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:

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

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

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

Applying 1st law of thermodynamics to the entire cycle:

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

The COP of the reverse Brayton cycle is given by:

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

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

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

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

a) For fixed heat rejection temperature (T3) and fixed refrigeration temperature (T1), the COP of reverse Brayton cycle is always lower than the COP of reverse Carnot cycle (Fig. 1.3), that is

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

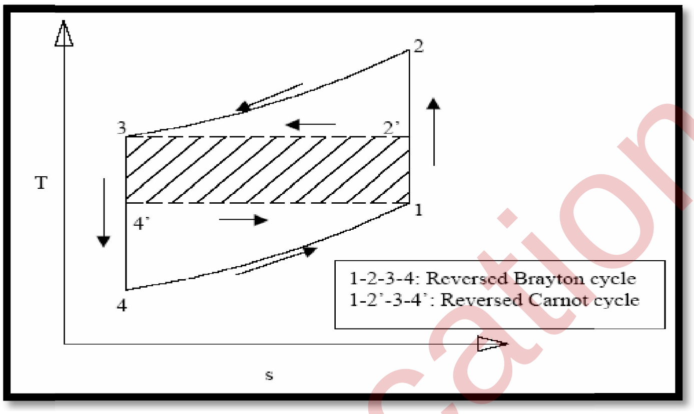


Fig. 1.3. Comparison of reverse Carnot and reverse Brayton cycle in T-s plan*e*

b) COP of Brayton cycle approaches COP of Carnot cycle as T1 approaches T4 (thin cycle), however, the specific refrigeration effect [cp(T1-T4)] also reduces simultaneously.

c) COP of reverse Brayton cycle decreases as the pressure ratio rp 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

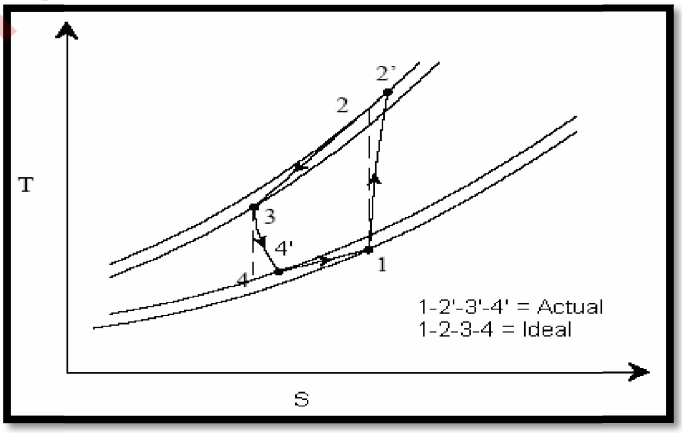


Fig. 1.4. Comparison of ideal and actual Brayton cycles T-s plane

Figure 1.4 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:

$$W_{1-2,act} = \frac{W_{1-2,isen}}{\eta_{c,isen}}$$

$$W_{3-4,act} = \eta_{t,isen} W_{3-4,isen}$$

Where η c,isen and η t,isen are the isentropic efficiencies of compressor and turbine, respectively. In the absence of pressure drops, these are defined as:

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

$$\eta_{t,isen} = \frac{(h_{3'} - h_{4'})}{(h_3 - h_4)} = \frac{(T_{3'} - T_{4'})}{(T_3 - T_4)}$$

The actual net work input, wnet,act is given by:

$$W_{net,act} = W_{1-2,act} - W_{3-4,act}$$

thus the net work input increases due to increase in compressor work input and reduction in turbine work output. The refrigeration impact additionally lessens because of the irreversibilities. Accordingly, the COP of real inverse Brayton cycles will be significantly lower than the ideal cycles. Configuration of proficient compressors and turbines assumes a significant part in enhancing the COP of the framework. By and by, opposite Brayton cycles can be open or shut. In open frameworks, icy air at the way out of the turbine streams into a room or lodge (cold space), and air to the compressor is taken from the frosty space. In such a case, the low side weight will be environmental. In closed systems, the same gas (wind currents through the cycle in a closed way. In such cases it is conceivable to have low side weights more prominent than air. These frameworks are known as thick air frameworks. Thick air frameworks are invaluable as it is conceivable to diminish the volume of air took care of by the compressor and turbine at high weights. Productivity will likewise be high because of littler weight proportions. It is likewise conceivable to utilize gasses other than air (e.g. helium) in closed systems.

1.6. Aircraft cooling systems

In an aircraft, cooling systems are required to keep the cabin temperatures at a comfortable level. Despite the fact that the outside temperatures are low at high elevations, as yet cooling of lodge is needed because of:

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.

iv. Solar radiation.

When air at this low weight is packed and supplied to the lodge at weights near to air, the temperature increments altogether. Case in point, when outside air at a weight of 0.2 bar and temperature of 223 K (at 10000 m elevation) is compacted to 1 bar, its temperature increments to around 353 K. On the off chance that the lodge is kept up at 0.8 bar, the temperature will be around 332 K. This impact is called as ram impact. This impact adds warmth to the lodge, which needs to be taken out by the cooling systems.

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 vapours compression system.

iv. Design of the complete system is much simpler due to low pressures. Maintenance required is also less.

1.6.1. Simple aircraft refrigeration cycle:

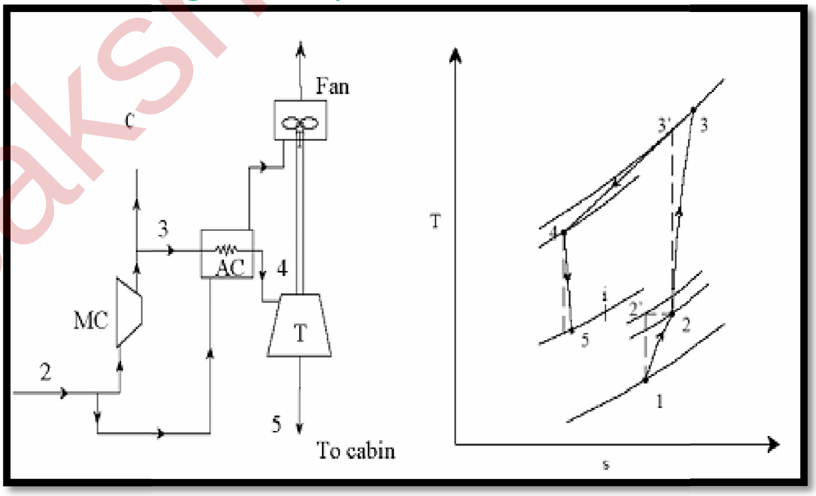


Fig. 1.5. Schematic of a simple aircraft refrigeration cycle

Figure 1.5 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 gets warm as it courses through the lodge giving helpful cooling impact. The force yield of the turbine is utilized to drive the fan, which keeps up the obliged wind stream over the air cooler. This basic framework is useful for ground cooling (when the airplane is not moving) as fan can keep on maintaining wind stream 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:

$$\frac{T_{2'}}{T_1}=1+\frac{\gamma-1}{2}M^2$$

where M is the Mach number, which is the ratio of velocity of the aircraft (C) to the sonic velocity a (a =$\sqrt{\gamma}$ RT1), i.e.,

$$M=\frac{C}{a}=\frac{C}{\sqrt{\gamma RT_1}}$$

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.,

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

The refrigeration capacity of the simple aircraft cycle discussed, Q is given by:

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

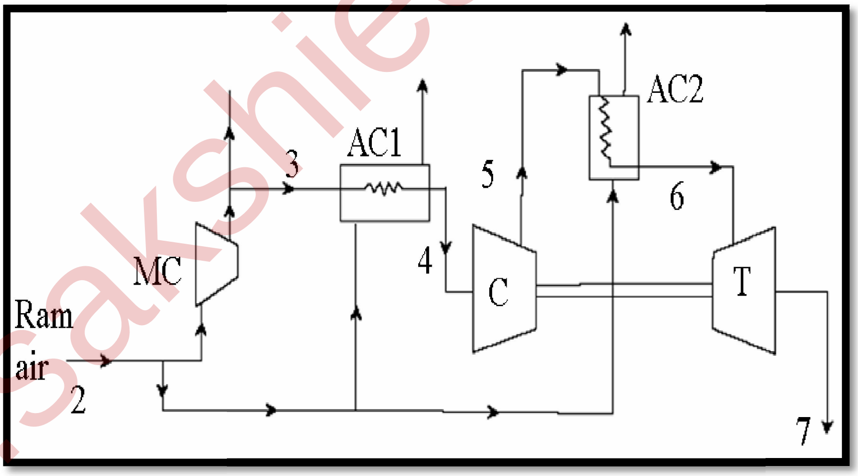
where m is the mass flow rate of air through the turbine.

1.6.2. Bootstrap system:

Figure 1.6 demonstrates the schematic of a bootstrap framework, which is a change of the straightforward framework. As shown in the figure, this framework comprises of two heat exchangers (air cooler and after cooler), rather than one air cooler of the straightforward framework. It likewise joins an auxiliary compressor, which is driven by the turbine of the cooling systems. This system is suitable for rapid airplane, where in the speed of the air ship gives the fundamental wind current to the warmth exchangers, accordingly a different fan is not needed.

As shown in the cycle outline, encompassing air express 1 is pressurized to express 2 because of the ram impact. 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 T7. 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 pre cooling 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.



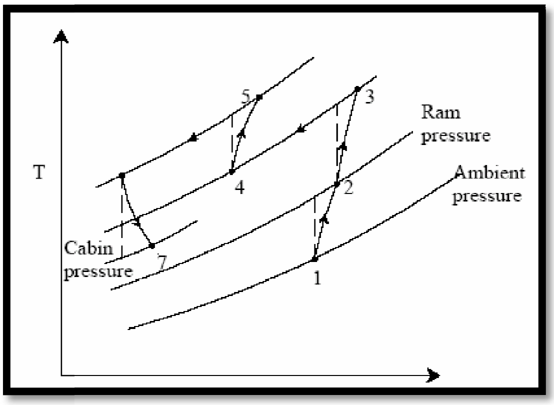


Fig. 1.6. Schematic of a bootstrap system

1.7 Dry Air Rated Temperature (DART):

The concept of Dry Air Rated Temperature is utilized to think about diverse airplane refrigeration cycles. Dry Air Rated Temperature is characterized as the temperature of the air at the way out of the cooling turbine without dampness build-up. For condensation not to happen during expansion in turbine, the dew point temperature and thus dampness substance of the air ought to be low, i.e., the air ought to be extremely dry. The flying machine refrigeration frameworks are evaluated in view of the mass stream rate of air at the outline DART. The cooling limit is then given by:

$$Q= mcp\ (Ti - TDART\ )$$

Where .m is the mass flow rate of air, TDART and Ti 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