COURSE : REFRIGERATION AND AIR CONDITIONING (6ME5-11)

B. Tech. VI Semester 2019-2020

UNIT-2: GAS CYCLE REFRIGERATION

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Objectives of the lesson

The objectives of this lesson are to make you understand the:

- Reversed Carnot cycle & its limitations
- Reversed Brayton cycle: Ideal & Actual
- Necessity of aircraft refrigeration and various aircraft refrigeration cycles, namely, Simple system, Bootstrap system, Regenerative system and Reduced ambient
- Concept of Dry Air Rated Temperature

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

- Describe various air cycle refrigeration systems
- State the assumptions made in the analyses of air cycle systems
- Show the cycles on T-s diagrams
- Perform various cycle calculations
- State the significance of Dry Air Rated Temperature

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.

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

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.,

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

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 $P_1 v_1^{\gamma} = P_2 v_2^{\gamma}$ 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:

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

where γ is the coefficient of isentropic expansion given by: $\gamma = \left(\frac{c_p}{c_v} \right)$

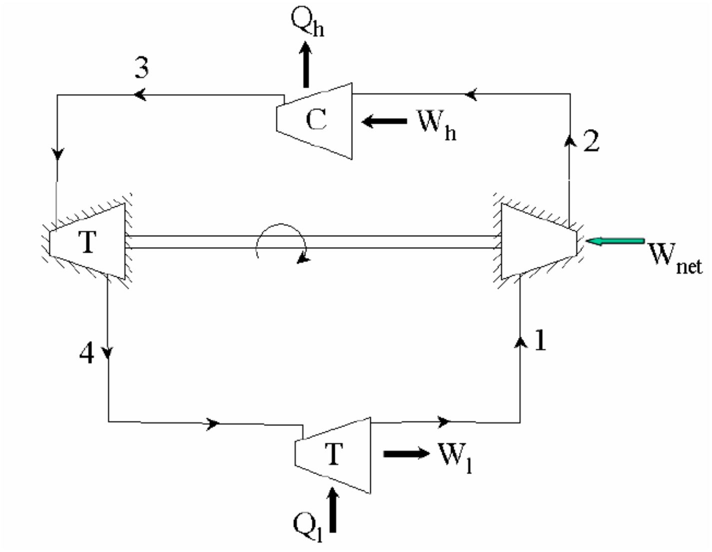
Isentropic expansion of the gas can also be carried out in a steady flow in a turbine which gives a net work output. Neglecting potential and kinetic energy changes, the work output of the turbine is given by:

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

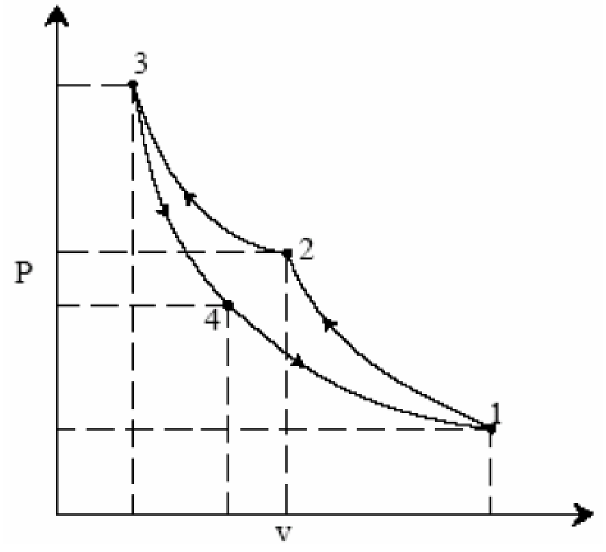
Reversed Carnot cycle employing a gas

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



(a)



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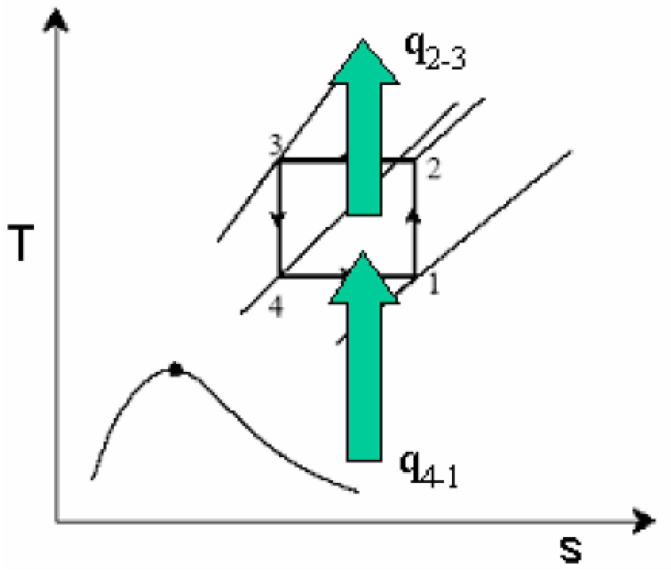


(c)

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

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

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

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

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, w_{3-4} exactly matches the work of isentropic compression w_{1-2}. 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 (T_l) and heat rejection (T_h) temperatures only.

Limitations of Carnot cycle

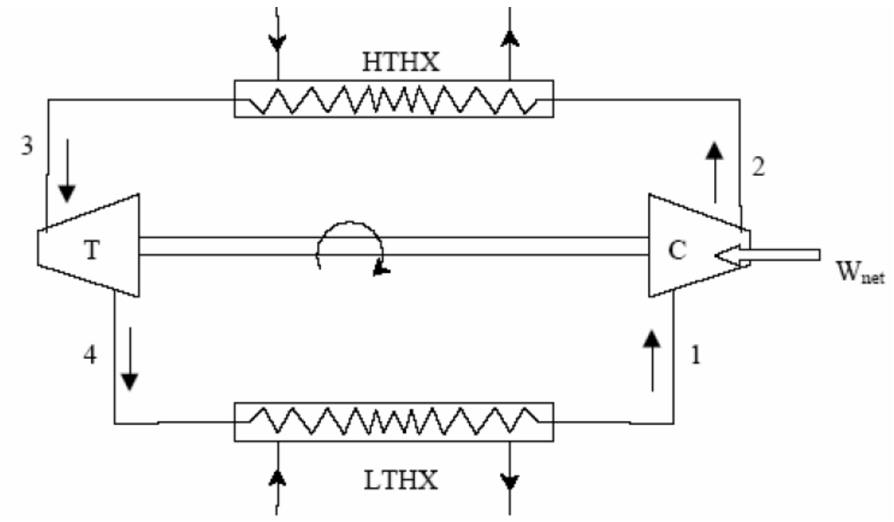
Carnot cycle is an idealization and it suffers from several practical limitations.

- 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.

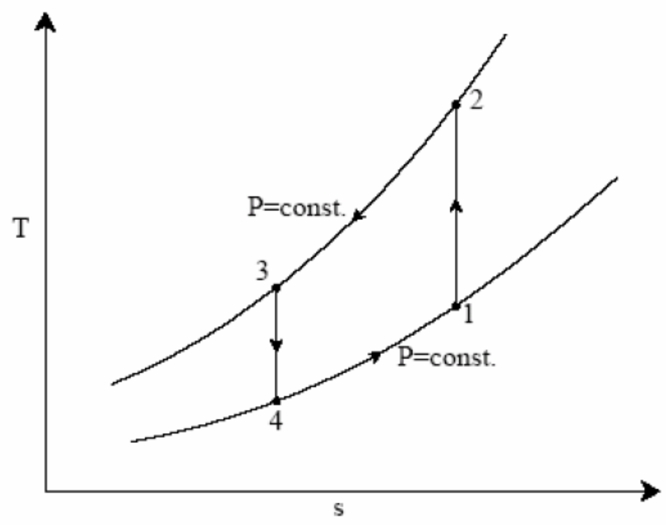
Ideal reversed Brayton cycle

This is an important cycle that might 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 (a) and (b) shows the schematic of a closed, reverse Brayton cycle and also the cycle on T-s diagram. 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



(a)



(b)

Analysis of ideal reversed Brayton cycle

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:

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

$$s_2 = s_1$$

$$\text{and } T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = T_1\, r_p^{\frac{\gamma-1}{\gamma}} \qquad \text{where } r_p = (P_2/P_1) = \text{pressure ratio}$$

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:

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

$$s_2 - s_3 = c_p \ln \frac{T_2}{T_3}$$

$$P_2 = P_3$$

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:

$$W_{3-4} = \dot{m}(h_3 - h_4) = \dot{m}\, c_p (T_3 - T_4)$$

$$s_3 = s_4$$

$$\text{and } T_3 = T_4 \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = T_4\, r_p^{\frac{\gamma-1}{\gamma}} \qquad \text{where } r_p = (P_3/P_4) = \text{pressure ratio}$$

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:

$$Q_{4-1} = \dot{m}(h_1 - h_4) = \dot{m}\, c_p (T_1 - T_4)$$

$$s_4 - s_1 = c_p \ln \frac{T_4}{T_1}$$

$$P_4 = P_1$$

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:

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

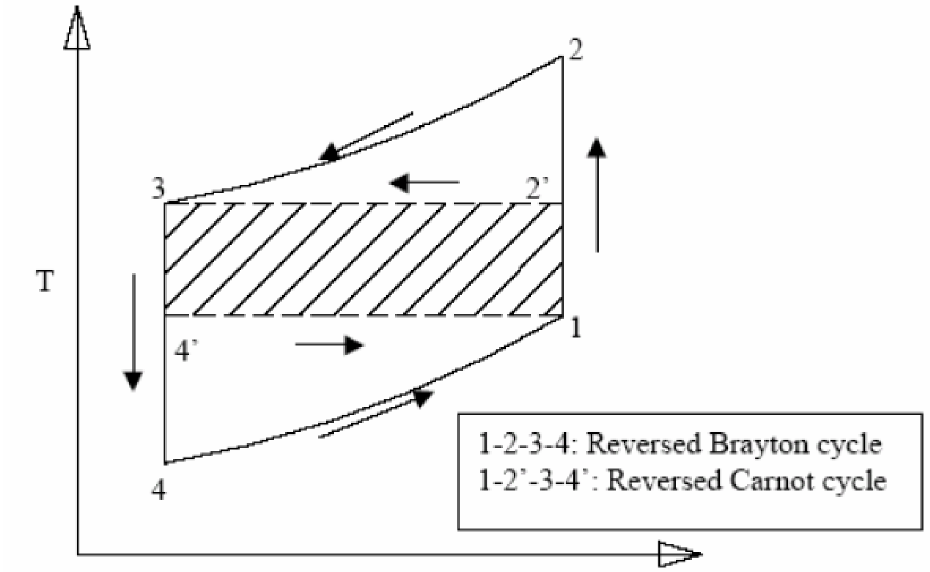
Using the relation between temperatures and pressures, the COP can also be written as:

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

Observations: Carnot cycle Vs. Brayton cycle

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 (Figure), that is :

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

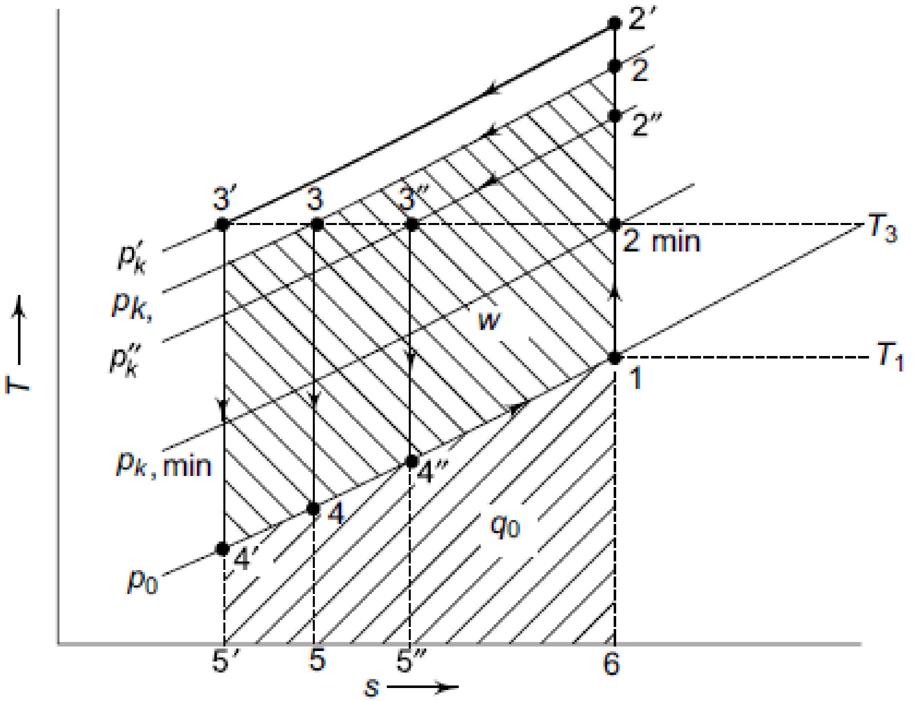


Figure

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

Variation of COP of air cycle with pressure ratio



Effect of discharge pressure p_k on the performance of the gas cycle

p_k/p_0	1	2	3	4	5	6
$\mathcal{E}$	∞	4.56	2.71	2.05	1.72	1.5

It may be noted that the pressures p_k and p_0, and hence the pressure ratio r, have limitations on account of the operating temperatures, viz., (i) T_1 as the highest refrigeration temperature, and (ii) T_3 as the lowest ambient temperature.

Point 1 on the diagram is fixed by the temperature T_1 and also the pressure p_0 which is generally equal to the surrounding atmospheric air pressure. Point 3 is fixed because of the limitations of the ambient temperature T_3 to which the gas can be cooled. Pressure p_k can, however, be varied within wide limits, starting from $p_{k,\ \min}$ onwards as shown in Fig. With the compressor discharge pressure equal to $p_{k,\min}$, the refrigerating capacity of the machine is zero. The air is alternately compressed and expanded between points $2_{\min}$ and 1. The net work is also zero and hence the COP is indeterminate. However, as the pressure p_k is increased, although the refrigerating effect (area under the curve 4-1) and hence the capacity of the refrigerating machine increases, the work of the cycle also increases. For example, when the discharge pressure is pk, the refrigerating effect is 1-4-5-6 and the net work is 1-2-3-4. When the discharge pressure is increased to pk , the increase in the refrigerating effect is 4-4-5-5 and that in the net work is 2-2-3-4-4-3-2. If is evident that the increase in work is much more than the increase in the refrigerating effect. As a result, the COP decreases with increasing pk. We find that a compression ratio of 3 to 4 in a single stage is reasonable.

Actual reversed Brayton cycle

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

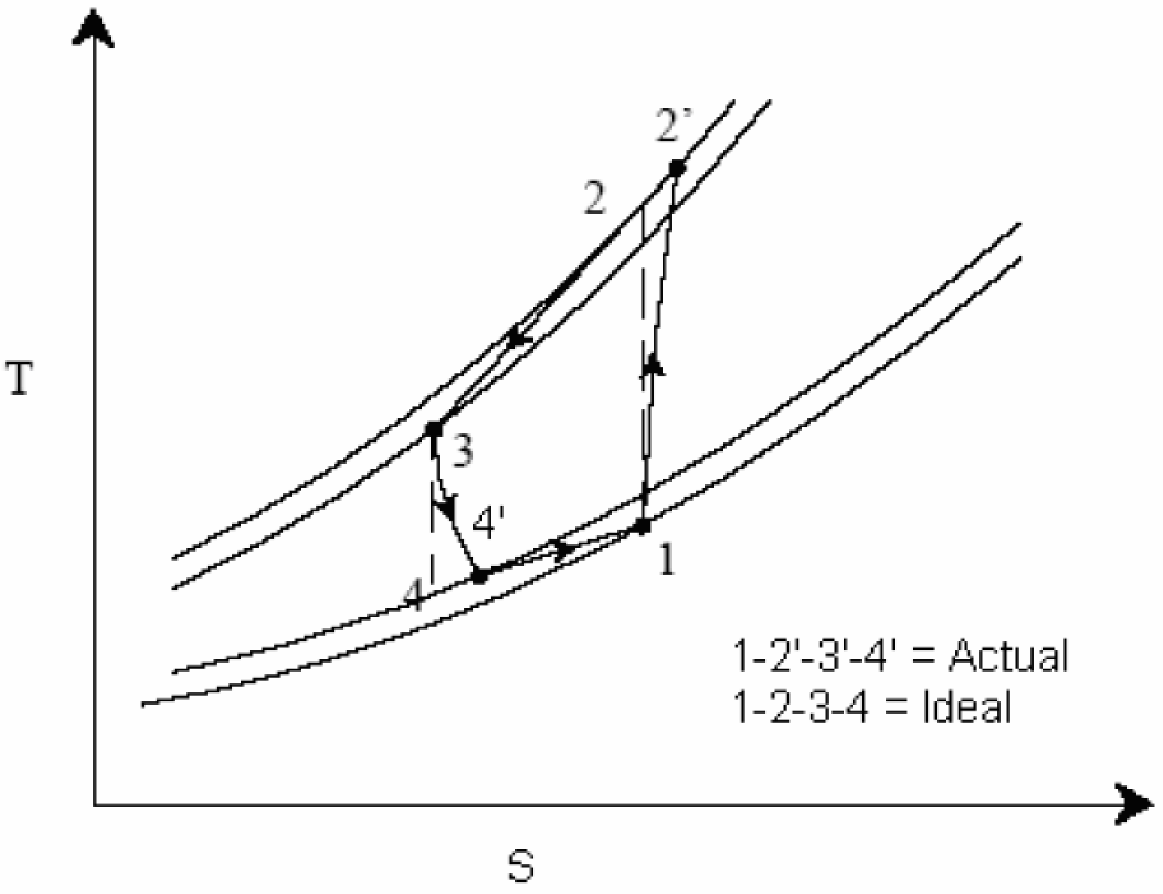
i. Non-isentropic compression and expansion processes

ii. Pressure drops in cold and hot heat exchangers

Figure 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:

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

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



Actual reversed Brayton cycle

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:

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

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

The actual net work input, $w_{net,act}$ 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 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.

Open Vs. Dense air cycle system

- The air cycle can work as an open cycle or as a closed cycle system.
- A *closed air cycle system* or a *dense air machine* has many thermodynamic advantages.
- It can work at a suction pressure $p0$ higher than the atmospheric. This reduces the volumes handled by the compressor and the expander. Also, the operating pressure ratio $pk/p0$ can be reduced, resulting in a higher coefficient of performance.
- In an *open air-cycle system* the air after expansion is directly led to the conditioned space. It is, therefore, necessary to expand air to one atmosphere pressure. This requires larger volumes to be handled.
- The open cycle system, has another advantage over the closed cycle system, in respect that it does not require a heat exchanger for the refrigeration process. This saves the weight and cost of the equipment.
- It, however, has one disadvantage, viz., when the air drawn from the refrigerated space is humid, it might produce fog and ice at the end of the expansion process and clog the line. A drier in the circuit is required in such a case.

Bell-Coleman refrigeration cycle

Let T_1, T_2, T_3 and T_4 be absolute temperatures at points 1, 2, 3 and 4.

Then heat abstracted from cold chamber or heat received by the system from the surrounding/kg of air

$$q_A = C_p(T_1 - T_4) \text{ per kg of air}$$

Also, heat taken away in the cooler or heat rejected from the system to the surroundings/kg of air

$$q_R = C_p(T_2 - T_3)$$

Work done *on* the system = Heat rejected by the system – Heat received by the system. The compression and expansion processes are reversible adiabatic process and involve no heat transfer.

$$w = C_p(T_2 - T_3) - C_p(T_1 - T_4) \text{ per kg of air}$$

Thus

$$\text{C.O.P.} = \frac{q_A}{w} = \frac{C_p(T_1 - T_4)}{C_p(T_2 - T_3) - C_p(T_1 - T_4)}$$

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

The ratio of expansion in the expansion cylinder is the same as the compression ratio in the compression cylinder in order to maintain the same upper and lower limits of pressure.

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{(\gamma - 1)/\gamma} = \left(\frac{p_3}{p_4}\right)^{(\gamma - 1)/\gamma} = \frac{T_3}{T_4}$$

it may be noted that with increase in pressure ratio the refrigeration effect increases or the plant capacity in *TR* increases, but the cycle work increases more rapidly than the refrigerating effect. This results in fall in COP with increasing pressure ratio. The rate of decrease in COP for pressure ratio more than 3 is much less as compared to that for less than 3. A reasonable value of pressure ratio for single stage compression is from 3 to 4.

or

$$T_2 = \frac{T_1 T_3}{T_4}$$

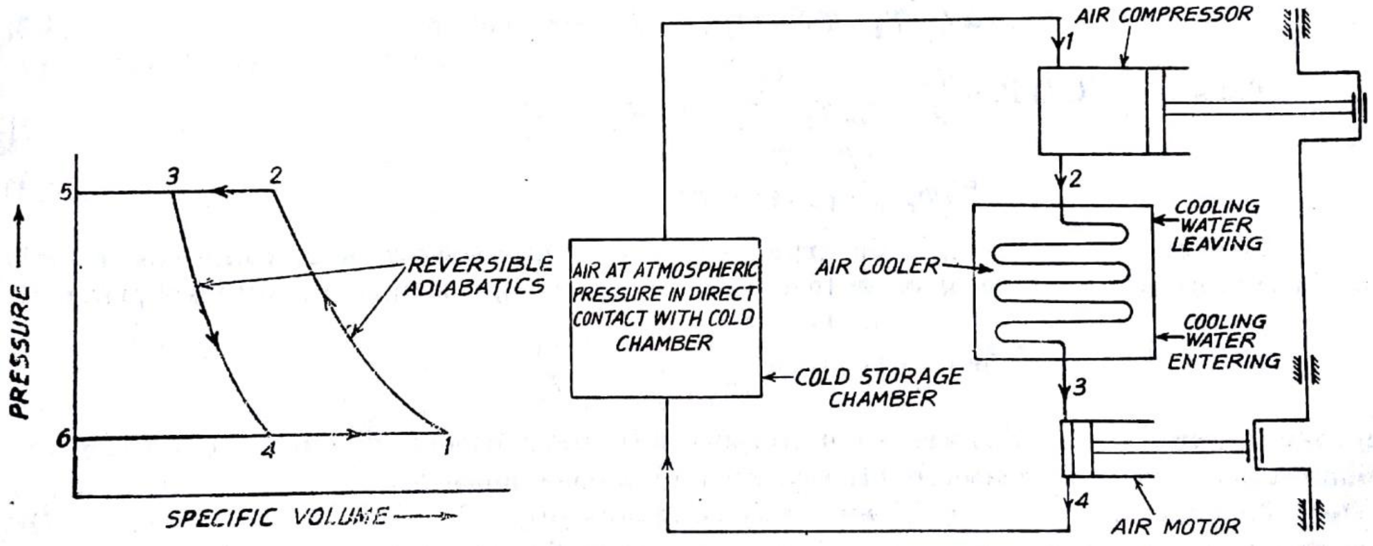
$$\text{C.O.P.} = \frac{T_1 - T_4}{\left(\frac{T_1 T_3}{T_4} - T_3\right) - (T_1 - T_4)} = \frac{T_1 - T_4}{(T_3/T_4)(T_1 - T_4) - (T_1 - T_4)}$$

$$= \frac{1}{(T_3/T_4) - 1} = \frac{T_4}{T_3 - T_4}$$

$$= \frac{1}{(p_2/p_1)^{(\gamma - 1)/\gamma} - 1}$$

$$= \frac{1}{r_p^{(\gamma - 1)/\gamma} - 1}$$

where $r_p = \frac{p_2}{p_1} = \frac{p_3}{p_4}$ = compression ratio or expansion ratio.

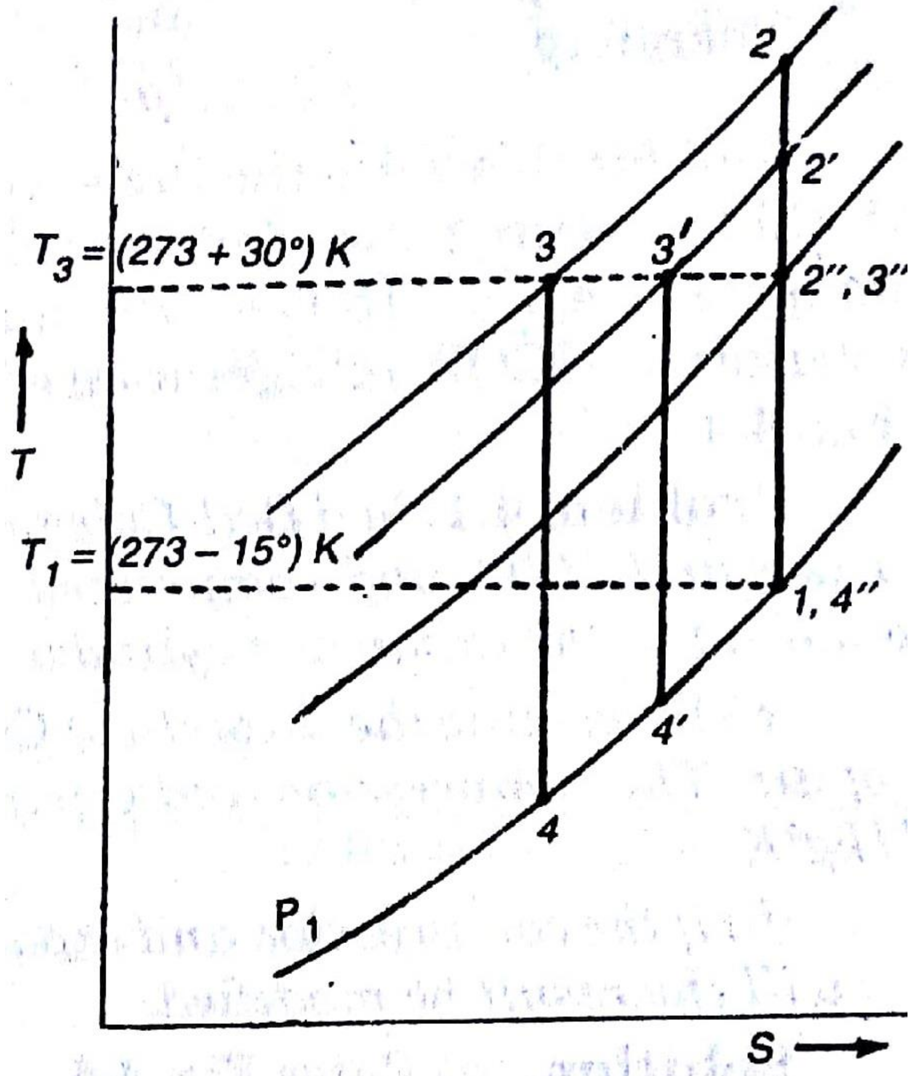


This open air system working on Bell-Coleman cycle has some disadvantages. The air comes in contact with the cold chamber or stuff to be preserved and collects moisture from such articles kept for preservation. This moisture freezes during expansion and there is likelihood of the valves getting choked.

Thus the same Bell-Coleman refrigerator may be worked with closed air system. The air may pass through the coils and abstract heat from cold chamber without coming directly in contact. In this system, the bulky compressor and expander is also avoided by passing dense air through them, *i.e.* suction to the compressor is at higher pressure than the atmospheric. This system is also called Dense Air System.

It may be noted that T_1 is the temperature fixed by refrigeration application. Let us assume it to be $-15°C$ as maximum. Also T_3 is the temperature fixed by the coolant available to which heat is to be rejected. Let us assume it to be $+30°C$. With these limitations, the maximum pressure ratio required would be

$$\left(\frac{p_2''}{p_1}\right)^{(\gamma-1)/\gamma} = \frac{T_2''}{T_1} = \frac{T_3''}{T_1} = \frac{303}{258}$$

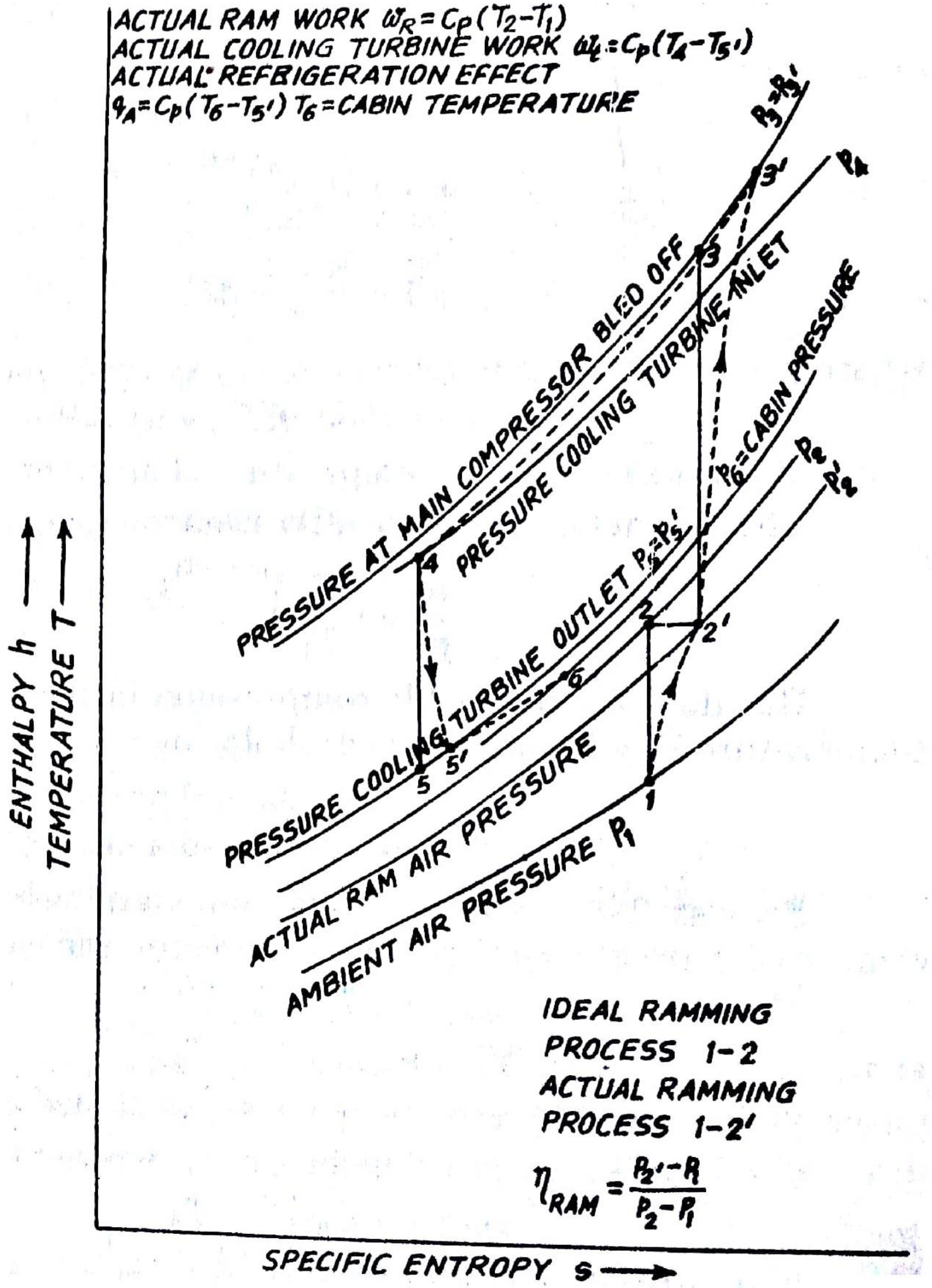


Ramming action in aircrafts

Process 1-2′ is the ramming process. Due to the plane moving at high speed, the air rams against the compressor intake system. Action here is that of diffuser. In the ramming process the total energy, or stagnation enthalpy remains unchanged, if the process is assumed adiabatic, while pressure of the air increases. Ideal ramming action would proceed along isentropic compression path such as 1-2, from pressure p_1 (ambient) and temperature T_1 (ambient) to pressure p_2 and temperature T_2. But due to irreversibilities, the actual state is indicated by point 2′ which shows increase in entropy and decrease in pressure with enthalpy $h_2 = h_2'$ and temperature $T_2 = T_2'$ remaining constant. The kinetic energy of outside air relative to air plane is expressed as

$$\text{K.E.} = \frac{V^2}{2}$$

where V is the velocity of air relative to plane or the velocity of air plane in m/sec.



Compressor intake section

Compressor intake section

Thus
$$h_2 = h_2' = h_1 + \frac{V^2}{2}$$
$$C_pT_2 = C_pT_2' = C_pT_1 + \frac{V^2}{2} \quad \text{...(4.36)}$$
$$T_2 = T_2' = T_1 + \frac{V^2}{2\,C_p}$$
$$\frac{T_2'}{T_1} = 1 + \frac{V^2}{2\,C_p\,T_1}$$
Also
$$C_p = \frac{\gamma}{\gamma - 1} R$$

Therefore
$$\frac{T_2'}{T_1} = 1 + \frac{V^2}{2[\gamma/(\gamma-1)]\,RT_1} = 1 + \frac{V^2}{2[\gamma/(\gamma-1)]\,RT_1} = 1 + \frac{(\gamma-1)\,V^2}{2\gamma\,RT_1}$$
where R is in Nm/kg °K or J/kg °K
$$= 1 + \frac{\gamma-1}{2}\frac{V^2}{a^2}$$
$$= 1 + \frac{\gamma-1}{2}M^2$$
where a is the local sonic velocity at intake conditions, *i.e.* ambient air conditions and M is the Mach number of the flight.
$$a = \sqrt{\gamma RT_1} \text{ where } R \text{ is in Nm/kg °K}$$
If R is expressed in kJ/kg °K
$$\frac{T_2'}{T_1} = 1 + \frac{(\gamma-1)\,V^2}{2\,\gamma RT_1 \times 1000}$$
$$= 1 + \frac{(\gamma-1)\,V^2}{2a^2}$$
$$= 1 + \frac{(\gamma-1)}{2}M^2$$
where a is the local sonic velocity at intake condition and M is the Mach number of flight.
$$a = \sqrt{1000\,\gamma RT_1}; \text{ where } R \text{ is in kJ/kg °K}$$
T_2' is the stagnation temperature of ambient air moving at velocity V m/sec.

The stagnation pressure after isentropic compression p_2 is given by
$$\frac{p_2}{p_1} = \left(\frac{T_2}{T_1}\right)^{\gamma/(\gamma-1)}$$
But due to irreversible compression in the ram, the pressure reached is p_2' at same temperature $T_2 = T_2'$ and same enthalpy $h_2 = h_2'$ due to ram efficiency
$$\eta_{Ram} = \frac{\text{Actual pressure rise}}{\text{Reversible adiabatic pressure rise}} = \frac{p_2' - p_1}{p_2 - p_1}$$
Process 2′-3 is the reversible adiabatic compression process and 2′-3′ is the actual compression process in the main compressor and work of compression is given by
$$w_c = C_p(T_3' - T_2')$$
and
$$W_c = m_b\,C_p(T_3' - T_2')$$
where W_c is the compressor work for m_b kg of bled air form the main compressor.

Similarly as explained before, heat rejected in pre-cooler and heat exchanger
$$Q_R = m_b\,C_p(T_3' - T_4)$$
Also cooling turbine work is given by $W_t = m_b\,C_p(T_4 - T_5')$

Also refrigeration load or heat absorbed is given by $Q_A = m_b\,C_p(T_6 - T_5')$

where T_6 is the cabin or cockpit temperature and T_5' is the cooling turbine exit temperature.

Important assumptions. (*i*) Unless otherwise stated the work done by the cooling turbine is normally used up for running the fan to blow in air through the heat exchanger. The work is not credited for refrigeration plant.

(*ii*) The kinetic energy converted to pressure energy in ram and its subsequent participation by way of expansion to give work output is also ignored in view of the fact that air must leave the cabin also at high velocity into the ambient atmosphere.

Advantages of air refrigeration system in aircrafts

The main considerations involved in an aircraft application in order of importance are weight, space and operating power. Though the power per ton of refrigeration is considerably more for air-cycle refrigeration than for a vapour-compression system. The advantages of air cycle with regard to its application in aircraft refrigeration are as follows:

- Small amounts of leakages are tolerable with air as the refrigerant.
- Air cycle in its simplest form as an open system requires only one heat exchanger.
- Availability of the refrigerant in mid air is also an important consideration.
- Cabin pressurization and air conditioning can be combined into one operation.
- Initial compression of the air is obtained by the *ram effect*, viz., conversion of the high kinetic energy of the ambient air relative to the aircraft into enthalpy and hence pressure rise. The power for this, however, is derived from the engine of the aircraft since the process in the ram causes a drag on the engine.
- Air is cheap, safe, non-toxic and non-flammable. Leakage of air is not a problem
- Cold air can directly be used for cooling thus eliminating the low temperature heat exchanger (open systems) leading to lower weight.
- 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.
- Design of the complete system is much simpler due to low pressures. Maintenance required is also less.

Necessity of aircraft air-conditioning

- Energy release from occupants, human beings even at rest releases about 300 kJ/h per person.

- Large internal heat generation due to control devices and electrical equipment etc. Medium size aircraft requires 10 to 15 kW of energy for the control devices.

- Ramming action causes the increase in incoming air temperature. Aeroplane moving with a speed of 300 m/s in an environment having Pa = 0.3 bar, Ta = -10°C, ramming action with 90% efficiency will give a temperature rise of the order of 40°C.

- There is heat transfer to the cabin because of direct solar radiation through glass panes and through the aeroplane body facing the sun.

- Heat generation due to dissipation of fast moving air relative to airplane (skin friction) caused by the fast moving aircraft (about 300 m/s).

- Aircraft while cruising at an altitude At high altitudes of 8 to 12 km, 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.

Types of aircraft air refrigeration systems

1. Simple air refrigeration system

2. Simple air refrigeration with evaporative cooling

3. Boot strap air refrigeration system

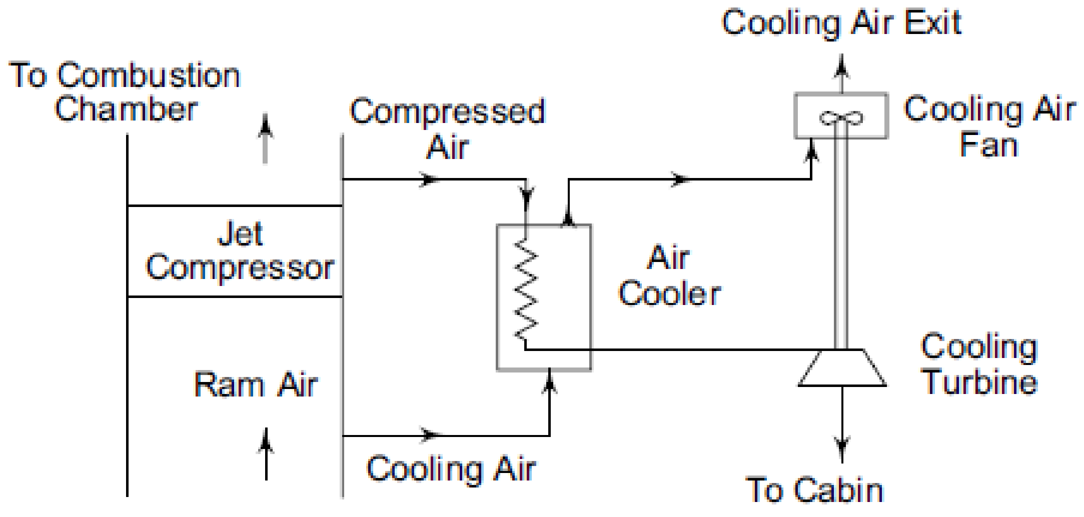
4. Boot strap air refrigeration with evaporative cooling system

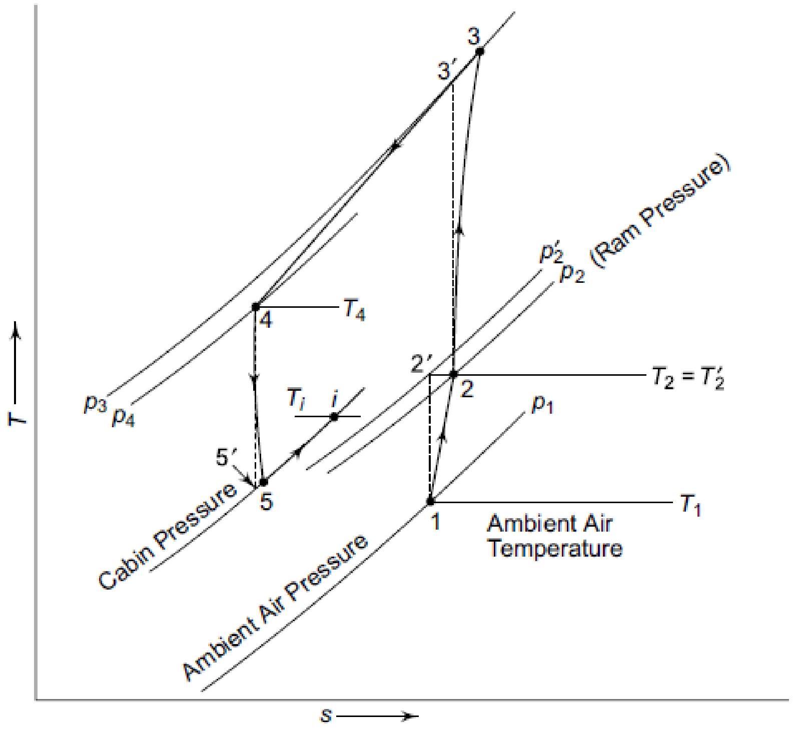
5. Regenerative air refrigeration system

6. Reduced ambient air refrigeration system

Simple aircraft refrigeration cycle with ram compression

Figure shows the schematic of an open system simple aircraft refrigeration system and the operating cycle on T-s diagram. 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.





Simple aircraft refrigeration cycle with ram compression

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 $(a = \sqrt{\gamma R T_1})$ i.e.,

$$M = \frac{C}{a} = \frac{C}{\sqrt{\gamma R T_1}}$$

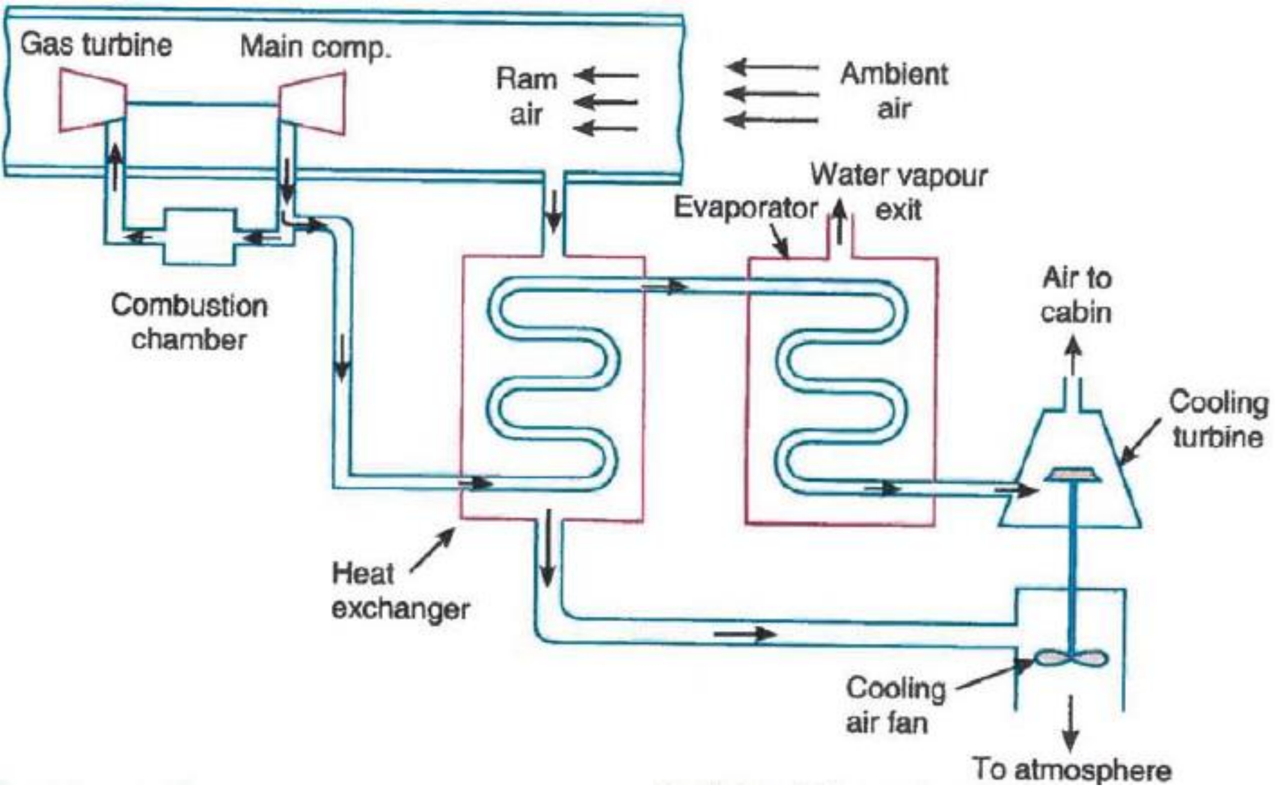
Due to irreversibilities, the actual pressure at the end of ramming will be less than the pressure resulting from isentropic compression. The ratio of actual pressure rise to the isentropic pressure rise is called as ram efficiency, η^{Ram}, i.e.,

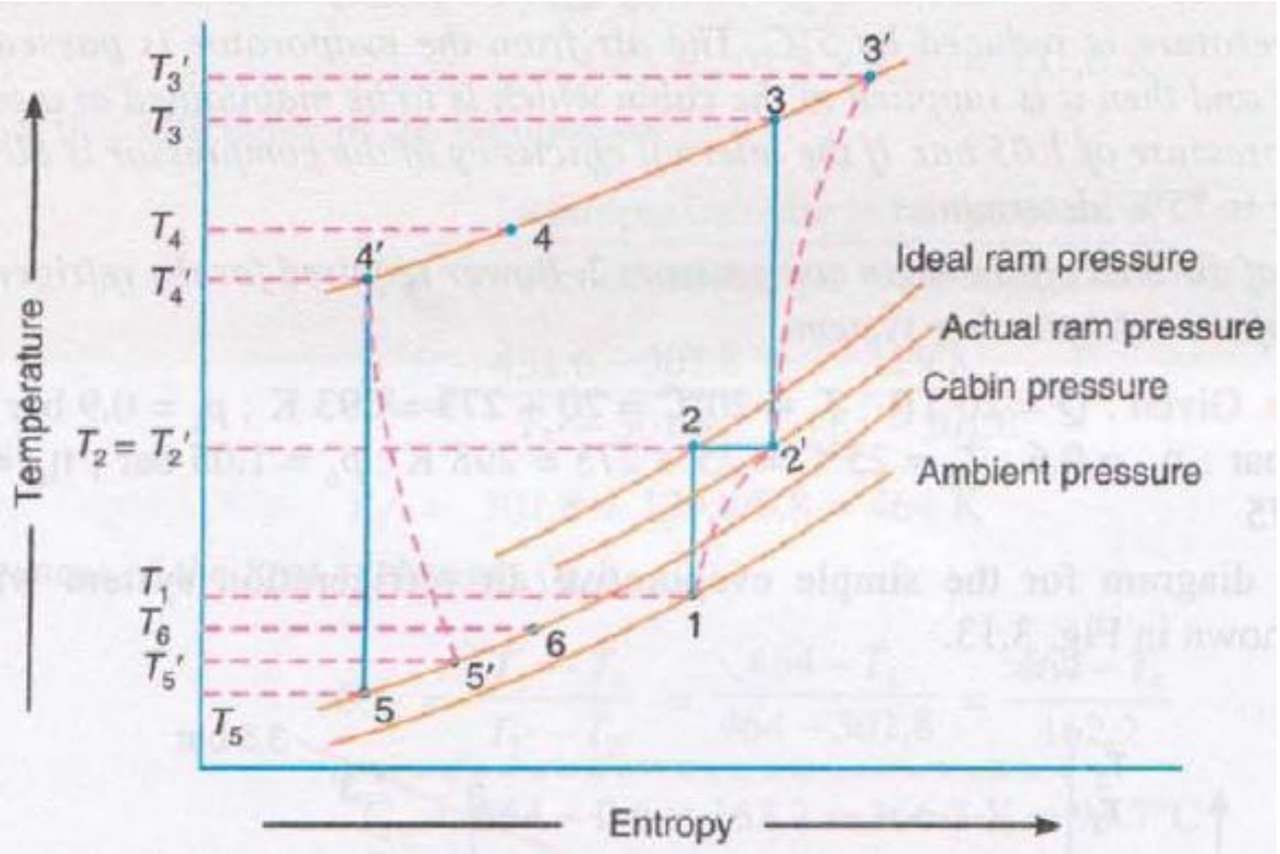
$$\eta_{Ram} = \frac{(P_2 - P_1)}{(P_{2'} - P_1)}$$

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

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

Simple aircraft refrigeration cycle with evaporative cooling





The initial mass of evaporant (m_e) required to be carried for the given flight time is given

$$m_e = \frac{Q_e . t}{h_{fg}}$$

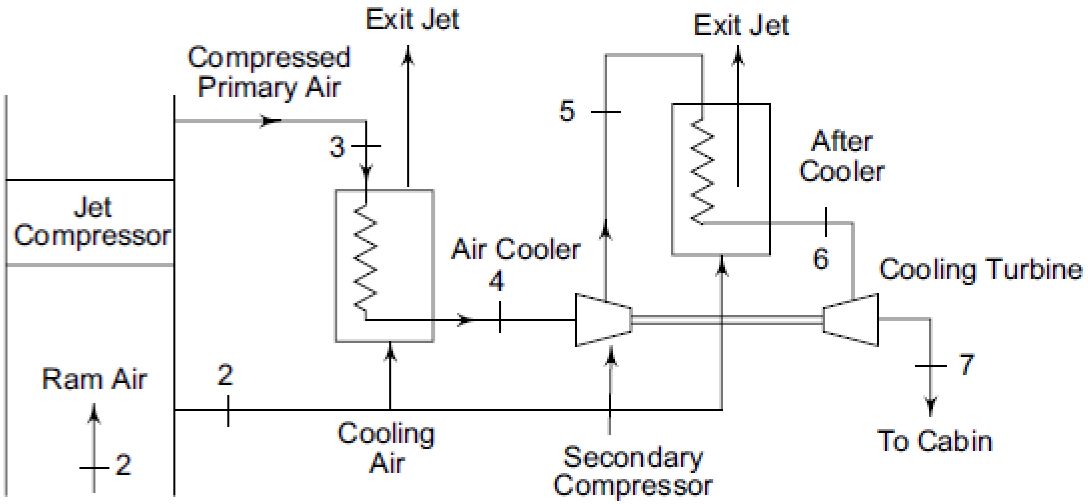
Q_e = Heat to be removed in evaporation in kJ/min,

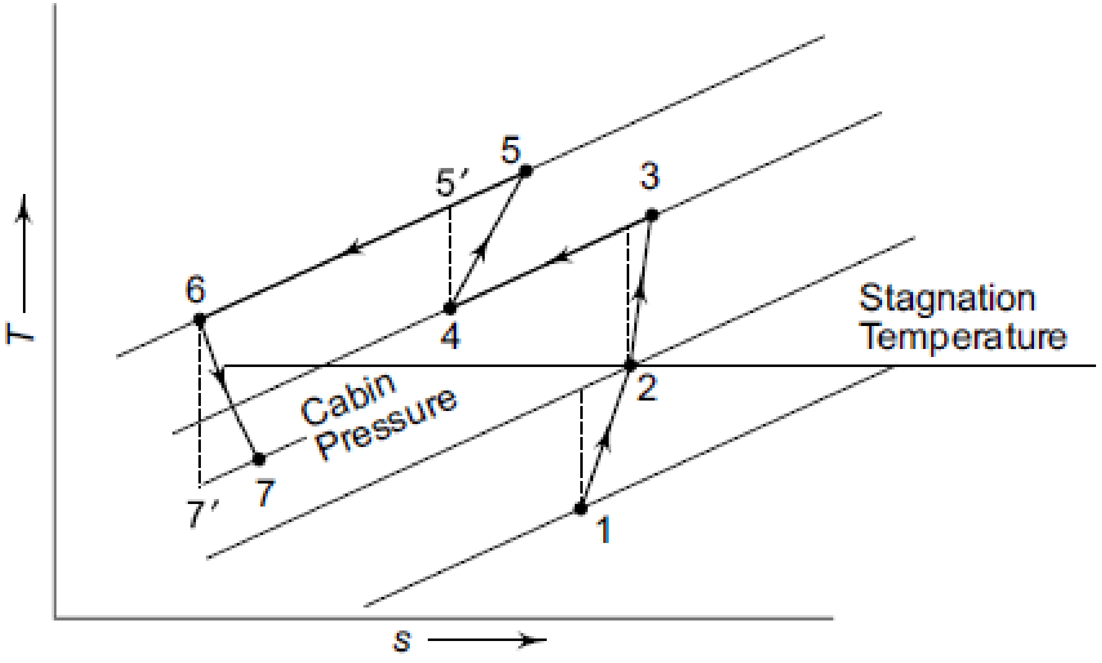
t = Flight time in minutes, and

h_{fg} = Latent heat of vaporisation of evaporant in kJ/kg.

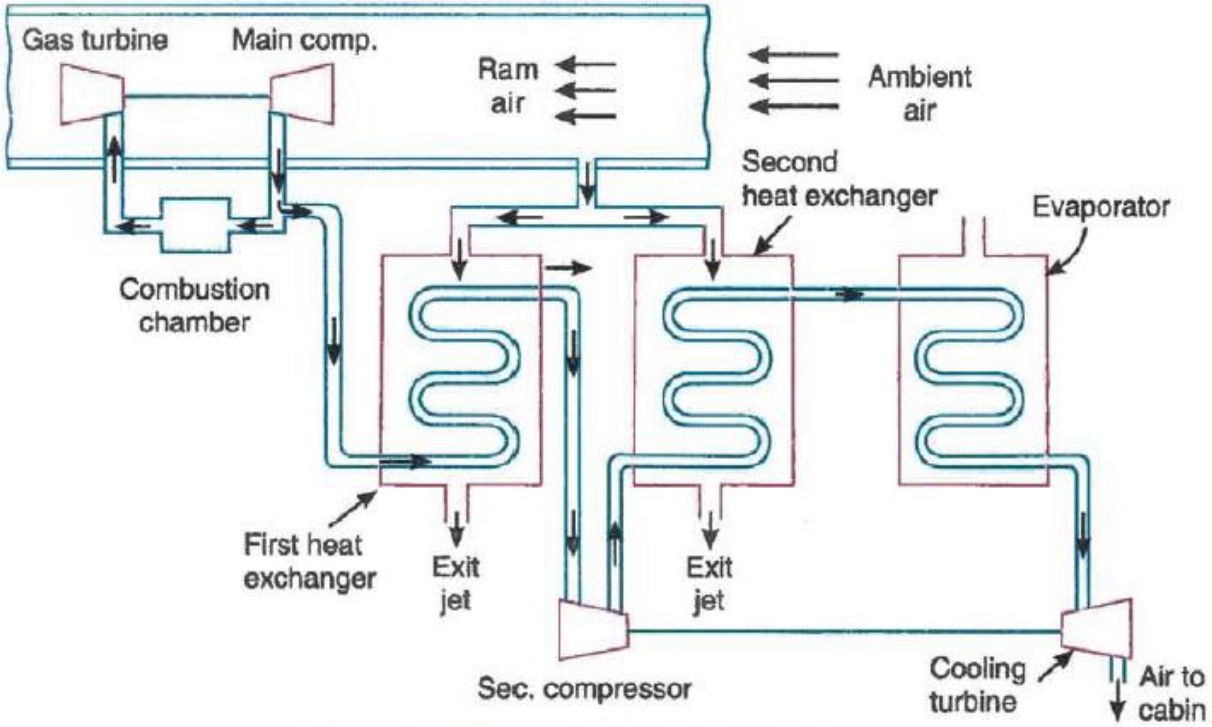
Bootstrap air refrigeration system

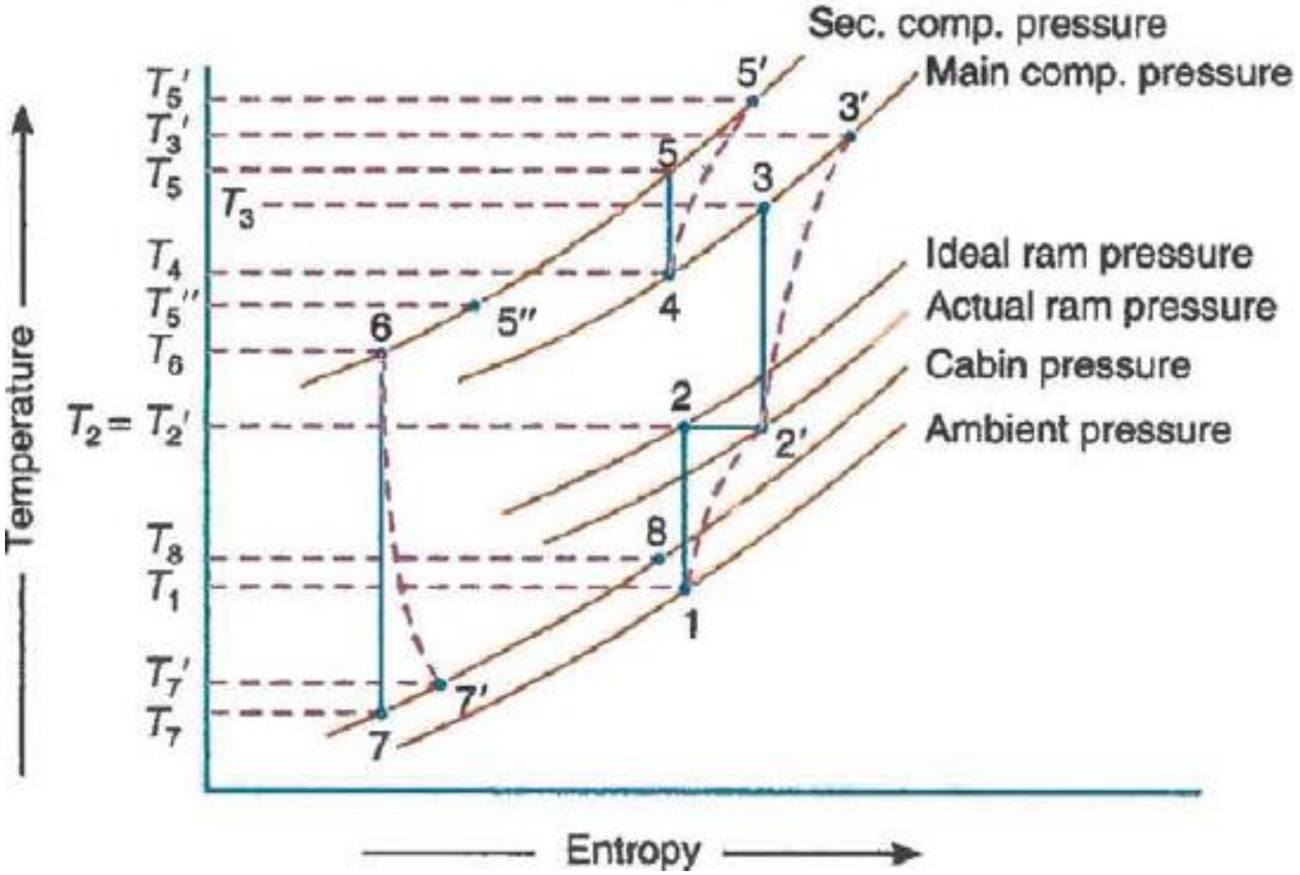
Bootstrap system is a modification of the simple system. It consists of two heat exchangers (air cooler and aftercooler), in stead of one air cooler of the simple system. It also incorporates a secondary compressor, which is driven by the turbine of the cooling system. This system is suitable for high 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. 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 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.





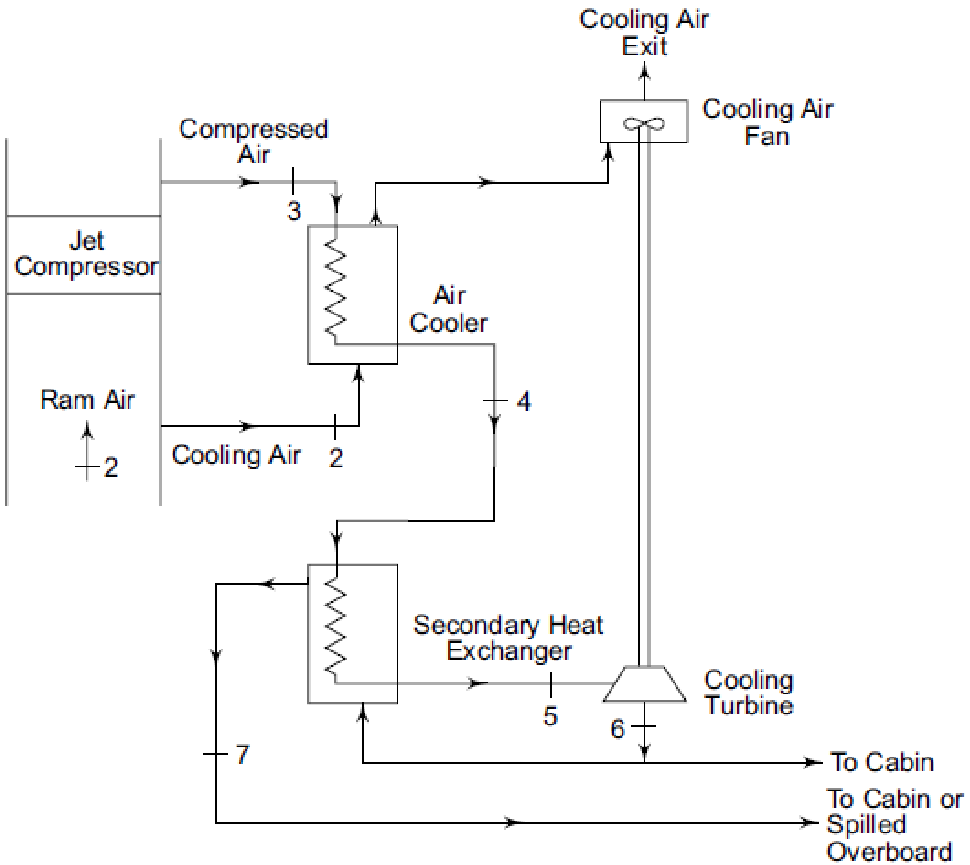
Bootstrap aircraft refrigeration cycle with evaporative cooling

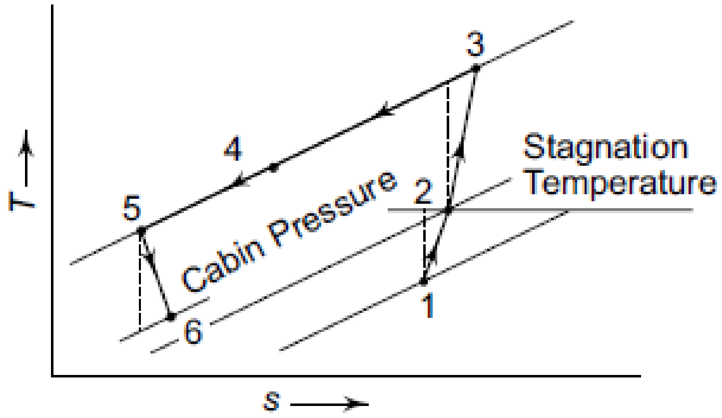




Regenerative air refrigeration system

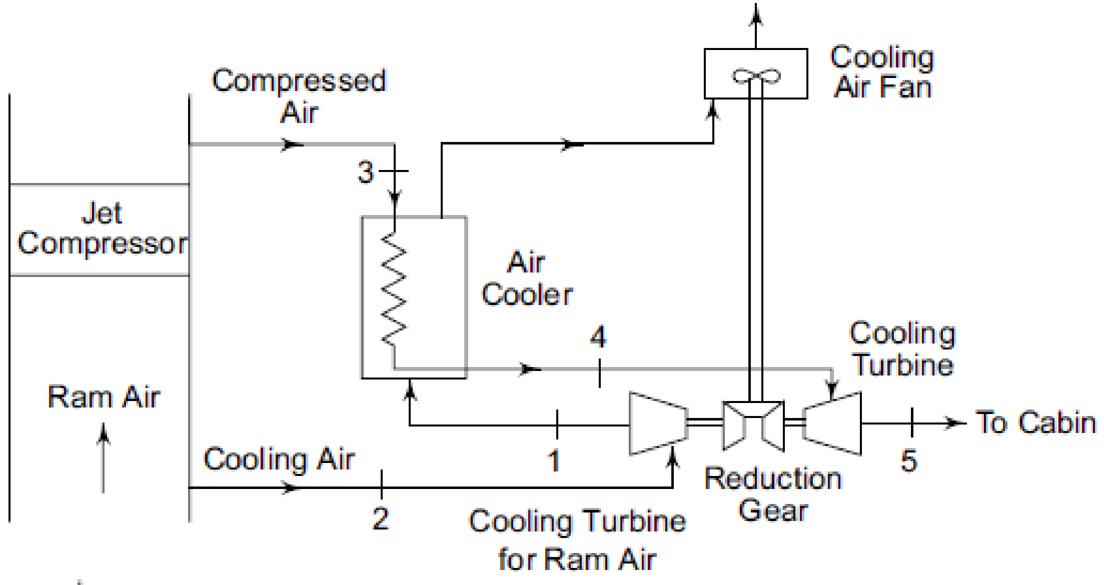
- The regenerative system also has two heat exchangers, but does not require ram air for cooling the air in the second heat exchanger.
- It is a modification of the simple system with the addition of a secondary heat exchanger in which the air from the primary heat exchanger is further cooled with a portion of the refrigerated air bled after expansion in the turbine.
- It provides lower turbine discharge temperatures, but at the expense of some weight and complications.
- It is used for ground cooling as well as high speed aircrafts.

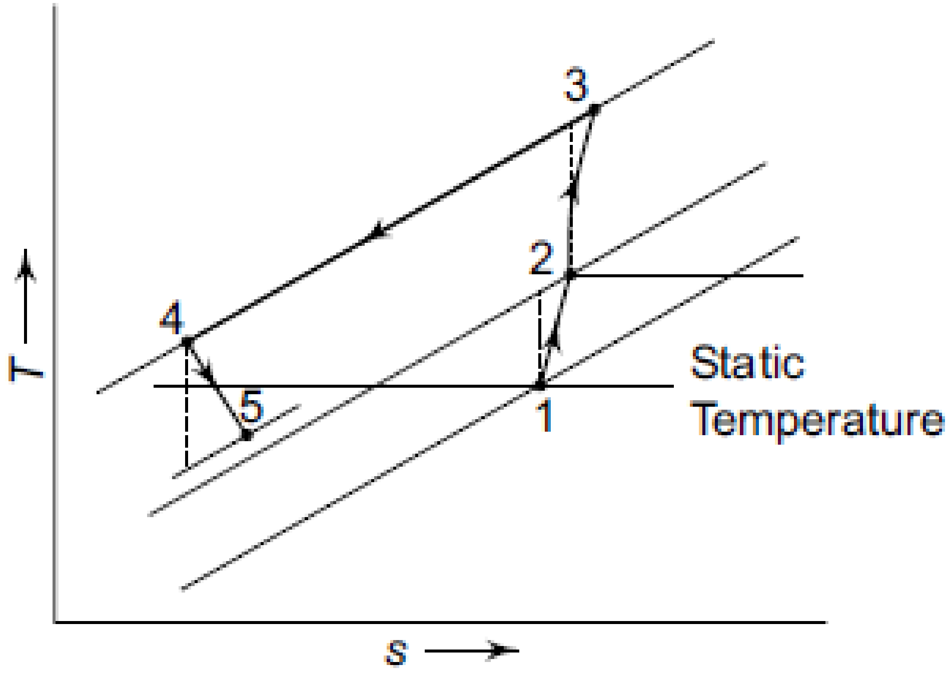




Reduced ambient air refrigeration system

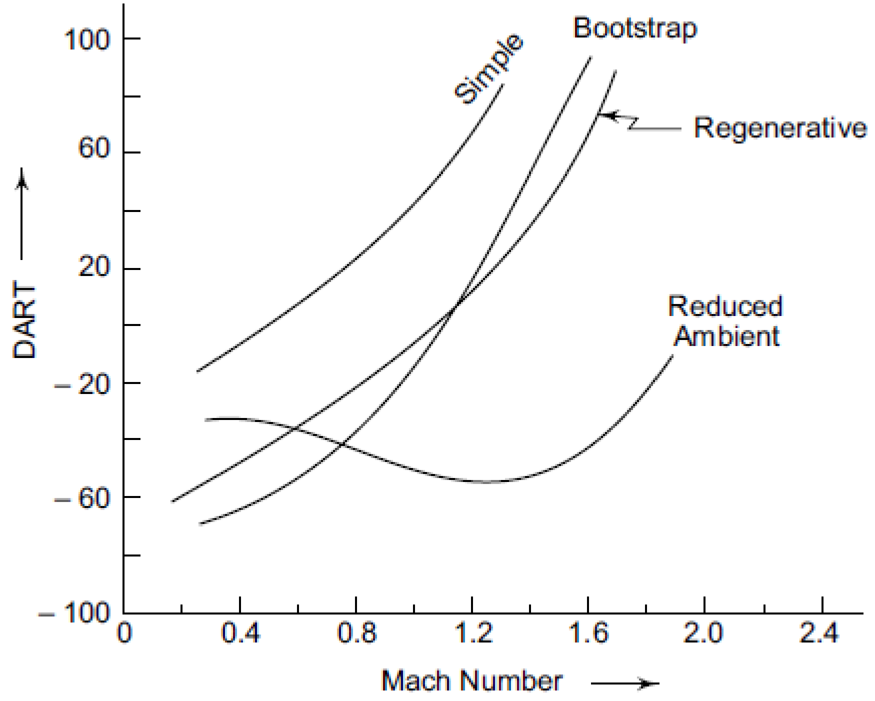
- In the reduced ambient system there are two expansion turbines-one in the cabin air stream and the other in the cooling air streams.
- Both turbines are connected to the shaft driving the fan which absorbs all the power.
- The turbine for the ram air operates from the pressure ratio made available by the ram air pressure.
- The cooling turbine reduces the temperature of cooling air to level of static temperature of ambient air.
- Thus, primary compressed air can be cooled to, say T4 below the stagnation temperature T2 and a little above the static temperature T1.
- It is used in Supersonic aircraft and Rockets.





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: $\dot{Q} = \dot{m} c_p (T_i - T_{DART})$



Comparison of DART vs. Mach number variation for common aircraft refrigeration systems

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:

- DART increases monotonically with Mach number for all the systems except the reduced ambient system
- The simple system is adequate at low Mach numbers
- At high Mach numbers, i.e. above a speed of 1000 kmph, either bootstrap system or regenerative system should be used
- Reduced ambient temperature system is best suited for very high Mach number, supersonic aircrafts and rockets.