VTU-NPTEL-NMEICT
Project Progress Report

The Project on Development of Remaining Three Quadrants to NPTEL Phase-I under grant in aid NMEICT, MHRD, New Delhi

Subject Matter Expert Details

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Module VI

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1) http://www.youtube.com/watch?v=E6_jw841vKE
2) http://www.youtube.com/watch?v=ITCu7gNMicc
3) http://bin95.com/swf/air-compressor-review.swf
4) http://www.machinerylubrication.com/Read/488/compressor-lubricants
5) http://www.slideshare.net/julsaez/compressor-basis-10733805
6) http://www.egpet.net/library/reciprocating-compressor-compressor-animation-2-video_06b782a66.html
7) http://www.brighthubengineering.com/hvac/51688-principle-of-working-of-refrigeration-reciprocating-compressors/
8) http://www.training-classes.com/programs/05/73/57384_air_compressor_training.php

Videos

http://www.4shared.com/video/Fyu2XA3O/YouTube_-_Reciprocating_Compre.htm

http://www.yourepeat.com/watch/?v=hGACRR_FETs

http://youviddy.com/video/wkuXz2YrwPs/reciprocating-compressor-an-introduction-to-vibration.html

http://www.metacafe.com/watch/3374356/simple_reciprocating_pump/

http://wn.com/rotary_vs_reciprocating_air_compressors

http://www.savevid.com/video/reciprocating-commercial-electrolux-and-tecumseh-compressors.html

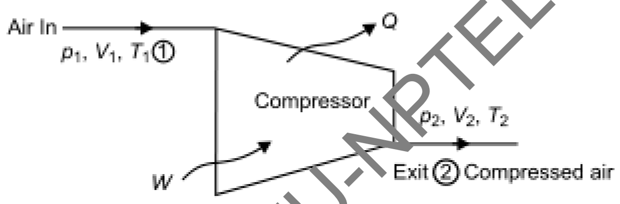
http://www.videvo.com/yvideo.php?i=N0JLN1ZKcWuRpRDhZT1k&how-a-two-stage-air-pressor-works

http://www.videvo.com/yvideo.php?i=NEdWOFA2cWuRpbmpOaEE&final-pressor-animation

ILLUSTRATIONS

INTRODUCTION TO COMPRESSOR:

Compressors are work absorbing devices which are used for increasing pressure of fluid at the expense of work done on fluid. The compressors used for compressing air are called air compressors. Compressors are invariably used for all applications requiring high pressure air. Some of popular applications of compressor are, for driving pneumatic tools and air operated equipments, spray painting, compressed air engine, supercharging in internal combustion engines, material handling (for transfer of material), surface cleaning, refrigeration and air conditioning, chemical industry etc. Compressors are supplied with low pressure air (or any fluid) at inlet which comes out as high pressure air (or any fluid) at outlet. Work required for increasing pressure of air is available from the prime mover driving the compressor. Generally, electric motor, internal combustion engine or steam engine, turbine etc. are used as prime movers. Compressors are similar to fans and blowers but differ in terms of pressure ratios. Fan is said to have pressure ratio up to 1.1 and blowers have pressure ratio between 1.1 and 4 while compressors have pressure ratios more than 4.



CLASSIFICATION OF COMPRESSORS:

Compressors can be classified in the following different ways.

(a) Based on principle of operation: Based on the principle of operation compressors can be classified as,

(i) Positive displacement compressors

(ii) Non-positive displacement compressors

In positive displacement compressors the compression is realized by displacement of solid boundary and preventing fluid by solid boundary from flowing back in the direction of pressure gradient. Due to solid wall displacement these are capable of providing quite large pressure ratios. Positive displacement compressors can be further classified based on the type of mechanism used for compression. These can be Reciprocating and Rotary Compressor

(i) Reciprocating type positive displacement compressors

(ii) Rotary type positive displacement compressors

Reciprocating compressors generally, employ piston-cylinder arrangement where displacement of piston in cylinder causes rise in pressure. Reciprocating compressors are capable of giving large pressure ratios but the mass handling capacity is limited or small. Reciprocating compressors may also be single acting compressor or double acting compressor. Single acting compressor has one delivery stroke per revolution while in double acting there are two delivery strokes per revolution of crank shaft. Rotary compressors employing positive displacement have a rotary part whose boundary causes positive displacement of fluid and thereby compression. Rotary compressors of this type are available in the names as given below;

(i) Roots blower

(ii) Vaned type compressors

Rotary compressors of above type are capable of running at higher speed and can handle large mass flow rate than reciprocating compressors of positive displacement type. Non-positive displacement compressors, also called as steady flow compressors use dynamic action of solid boundary for realizing pressure rise. Here fluid is not contained in definite volume and subsequent volume reduction does not occur as in case of positive displacement compressors. Non-positive displacement compressor may be of 'axial flow type' or 'centrifugal type' depending upon type of flow in compressor.

(b) Based on number of stages: Compressors may also be classified on the basis of number of stages. Generally, the number of stages depends upon the maximum delivery pressure. Compressors can be single stage or multistage. Normally maximum compression ratio of 5 is realized in single stage compressors. For compression ratio more than 5 the multi-stage Compressors are used. Typical values of maximum delivery pressures generally available from different types of compressor are,

(i) Single stage compressor, for delivery pressure up to 5 bar

(ii) Two stage compressor, for delivery pressure between 5 and 35 bar

(iii) Three stage compressor, for delivery pressure between 35 and 85 bar

(iv) Four stage compressor, for delivery pressure more than 85 bar

(c) Based on capacity of compressors: Compressors can also be classified depending upon the capacity of compressor or air delivered per unit time. Typical values of capacity for different compressors are given as;

(i) Low capacity compressors, having air delivery capacity of 0.15 m^3/s or less

(ii) Medium capacity compressors, having air delivery capacity between 0.15 and 5 m^3/s.

(iii) High capacity compressors, having air delivery capacity more than 5 m^3/s.

(d) Based on highest pressure developed: Depending upon the maximum pressure available from compressor they can be classified as low pressure, medium pressure, high pressure and super high pressure compressors. Typical values of maximum pressure developed for different compressors are as under;

(i) Low pressure compressor, having maximum pressure up to 1 bar

(ii) Medium pressure compressor, having maximum pressure from 1 to 8 bar

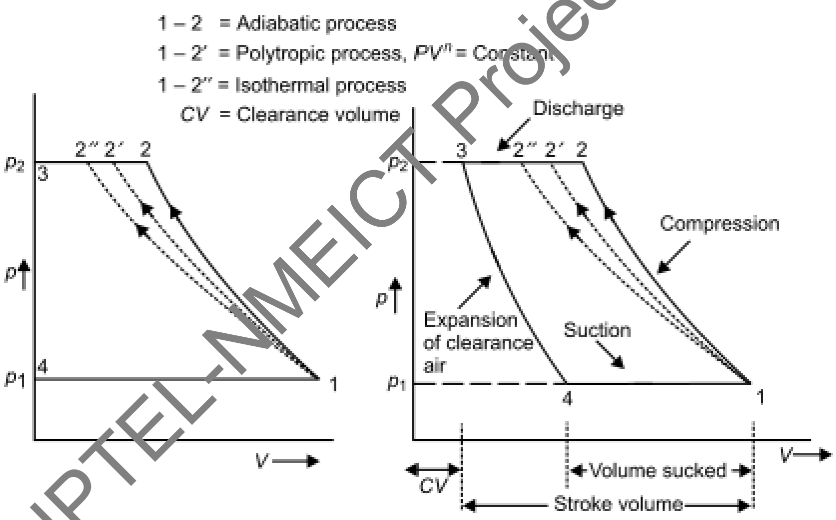
(iii) High pressure compressor, having maximum pressure from 8 to 10 bar

(iv) Super high pressure compressor, having maximum pressure more than 10 bar.

THERMODYNAMIC ANALYSIS ON COMPRESSOR:

Compression of air in compressor may be carried out following number of thermodynamic processes such as isothermal compression, polytropic compression or adiabatic compression. Figure shows the thermodynamic cycle involved in compression. Theoretical cycle is shown neglecting clearance volume but in actual cycle clearance volume can not be negligible. Clearance volume is necessary in order to prevent collision of piston with cylinder head, accommodating valve mechanism etc. Compression process is shown by process 1–2, 1–2', 1–2" following adiabatic, polytropic and isothermal processes.

On p-V diagram process 4–1 shows the suction process followed by compression during 1–2 and discharge through compressor is shown by process 2–3. Air enters compressor at pressure p_1 and is compressed up to p_2. Compression work requirement can be estimated from the area below the each compression process. Area on p–V diagram shows that work requirement shall be minimum with isothermal process 1–2'. Work requirement is maximum with process 1–2 i.e. adiabatic process. As a designer one shall be interested in a compressor having minimum compression work requirement. Therefore, ideally compression should occur isothermally for minimum work input. In practice it is not possible to have isothermal compression because constancy of temperature during compression cannot be realized. Generally, compressors run at substantially high speed while isothermal compression requires compressor to run at very slow speed so that heat evolved during compression is dissipated out and temperature remains constant. Actually due to high speed running of compressor the compression process may be assumed to be near adiabatic or polytropic process following law of compression as $PV^n = C$ with value of 'n' varying between 1.25 and 1.35 for air. Compression process following three processes is also shown on T-s diagram. It is thus obvious that actual compression process should be compared with isothermal compression process. A mathematical parameter called isothermal efficiency is defined for quantifying the degree of deviation of actual compression process from ideal compression process. Isothermal efficiency is defined by the ratio of isothermal work and actual indicated work in reciprocating compressor.



Isothermal efficiency = Isothermal work/Actual indicated work

Practically, compression process is attempted to be closed to isothermal process by air/water cooling, spraying cold water during compression process. In case of multi-stage compression process the compression in different stages is accompanied by intercooling in between the stages. Mathematically, for the compression work following polytropic process, $PV^n = C$. Assuming negligible clearance volume the cycle work done,

W c = Area on p-V diagram

$$= \left[P_2V_2 + \left(\frac{P_2V_2 - P_1V_1}{n-1}\right)\right] - P_1V_1$$

$$= \left(\frac{n}{n-1}\right)[P_2V_2 - P_1V_1]$$

$$= \left(\frac{n}{n-1}\right)P_1V_1\left[\frac{P_2V_2}{P_1V_1} - 1\right]$$

$$w_c = \left(\frac{n}{n-1}\right)P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right]$$

$$w_c = \left(\frac{n}{n-1}\right)mRT_1\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right]$$

$$w_c = \left(\frac{n}{n-1}\right)mR[T_2 - T_1]$$

In case of compressor having isothermal compression process, n = 1, i.e. $P_1 V_1 = P_1 V_2$

$W_{c,\,iso} = P_2 V_2 + P_1 V_1 \ln r - P_1 V_1$

$W_{c,\,iso} = P_1 V_1 \ln r$, where $r = \frac{V_1}{V_2}$

In case of compressor having adiabatic compression process, n = γ

$$w_{c\ adiabatic} = \left(\frac{\gamma}{\gamma-1}\right)mR[T_2 - T_1]$$

$$w_{c\ adiabatic} = C_p m[T_2 - T_1]$$

$$w_{c\ adiabatic} = [h_2 - h_1]$$

$$\eta_{iso} = \frac{P_1V_1 \ln r}{\left(\frac{n}{n-1}\right)P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right]}$$

The isothermal efficiency of a compressor should be close to 100% which means that actual compression should occur following a process close to isothermal process.

Considering clearance volume: With clearance volume the cycle is represented on Fig. The work done for compression of air polytropically can be given by the area enclosed in cycle 1–2–3–4. Clearance volume in compressors varies from 1.5% to 35% depending upon type of compressor.

$$w_{c\ with\ CV} = \left(\frac{n}{n-1}\right)P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right] - \left(\frac{n}{n-1}\right)P_4V_4\left[\left(\frac{P_3}{P_4}\right)^{\left(\frac{n-1}{n}\right)} - 1\right]$$

Here $P_1 = P_4$, $P_2 = P_3$

$$w_{c\ with\ CV} = \left(\frac{n}{n-1}\right) P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right] - \left(\frac{n}{n-1}\right) P_1 V_4 \left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right]$$

$$w_{c\ with\ CV} = \left(\frac{n}{n-1}\right) P_1 \left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right] (V_1 - V_4)$$

For single acting compressor running with N rpm, power input required, assuming clearance volume.

$$Power\ required\ = \left\{\left(\frac{n}{n-1}\right) P_1 \left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right] (V_1 - V_4)\right\} \times N$$

for double acting compressor,

$$Power\ required\ = \left\{\left(\frac{n}{n-1}\right) P_1 \left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right] (V_1 - V_4)\right\} \times 2N$$

Volumetric efficiency: Volumetric efficiency of compressor is the measure of the deviation from volume handling capacity of compressor. Mathematically, the volumetric efficiency is given by the ratio of actual volume of air sucked and swept volume of cylinder. Ideally the volume of air sucked should be equal to the swept volume of cylinder, but it is not so in actual case. Practically the volumetric efficiency lies between 60 and 90%. Volumetric efficiency can be overall volumetric efficiency and absolute volumetric efficiency as given below:

$$Overall\ volumetric\ efficiency\ = \frac{Volumeof\ freeairsuckedintocylinder}{Sweptvolumeof\ LP\ cylinder}$$

OR

$$\begin{gathered}Volumetric\ efficiency\\ referred\ to\ free\ air\ conditions = \frac{Volumeof\ freeairsuckedintocylinder}{Sweptvolumeof\ LP\ cylinder}\end{gathered}$$

QUADRANT-3

Wikis:

1 http://petrowiki.org/Reciprocating_compressor

2) http://en.wikipedia.org/wiki/Gas_compressor

3) http://demonstrations.wolfram.com/ReciprocatingCompressorWithAnIntercooler/
4) http://en.wikipedia.org/wiki/Reciprocating_compressor
5) http://en.wikipedia.org/wiki/Air_compressor
6) http://www.ask.com/question/what-is-a-reciprocating-compressor
7) http://petrowiki.org/Reciprocating_compressor
8) http://cair.wikia.com/wiki/Compressor
9) http://www.roymech.co.uk/Related/Thermos/Thermos_Air_com_mot.html
10) http://www.authorstream.com/Presentation/venumanu2008-1709112-compressors/

Open Contents:

***Applied Thermodynamics* by R. K. Rajput**

***Applied Thermodynamics* for Engineering Technologists by Eastop**

***Applied Thermodynamics* by B. K. Venkanna, B. V. S**

Basic and *Applied Thermodynamics* by Nag

***Applied Thermodynamics* by D. S. Kumar**

A *textbook* of *applied thermodynamics*, steam and thermal ... by S. K. Kulshrestha

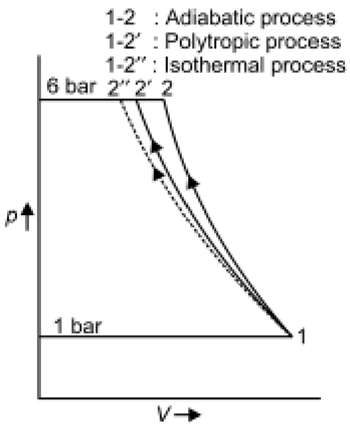
Applied thermodynamics by Anthony Edward John Hayes

QUADRANT-4

Problems

1) A reciprocating air compressor has cylinder with 24 cm bore and 36 cm stroke. Compressor admits air at 1 bar, 17°C and compresses it up to 6 bar. Compressor runs at 120 rpm. Considering compressor to be single acting and single stage determine mean effective pressure and the horse power required to run compressor when it compresses following the isothermal process and polytropic process with index of 1.3. Also find isothermal efficiency when compression is of polytropic and adiabatic type.

Solution:



Compression ratio $= \frac{P_2}{P_1} = 6 = r$

From cylinder dimensions the stroke volume $= \frac{\pi}{4}$ X $(0.24)^2$X (0.36)

$= 0.01628\ m^3$

Volume of air compressed per minute = 0.01628 X 120= 1.954 m^3/min

Let us neglect clearance volume.

Work done in isothermal process

$$W_{iso} = P_1 V_1 \ln r$$

Mean effective pressure in isothermal process

$$mep_{iso} = P_1 V_1 \ln r / V_1$$

$$= P_1 \ln r$$

$$= 1 \text{ X } 10^2 \ln 6 = 179.18 \text{ kPa}$$

Work done in polytropic process with index $n = 1.3$, *i.e.* $PV^{1.3} = C$

$$W_{poly} = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

Mean effective pressure in polytropic process,

$$mep_{poly} = \frac{\frac{n}{n-1} P_1 V_1}{v_1} \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$mep_{poly} = \frac{n}{n-1} P_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$mep_{poly} = \frac{1.3}{1.3-1} \times 1 \times 10^2 \times \left[(6)^{\frac{1.3-1}{1.3}} - 1\right]$$

$mep_{poly} =$ 221.89 kPa

Work done in adiabatic process, $W_{adiabatic} = \gamma/(\gamma-1)\ P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{-1}{}} - 1\right]$

Mean effective pressure in adiabatic process, $mep_{adiabatic} = \frac{W_{adiabatic}}{v1}$

$mep_{adiabatic} = \gamma/(\gamma\text{-}1)\ P_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{-1}{}} - 1\right] = \frac{1.4}{1.4-1} \times 1 \times 10^2 \times \left[(6)^{\frac{1.4-1}{1.4}} - 1\right]$

$mep_{adiabatic} =$ 233.98 kPa

Horse power required for isothermal process ,

$HP_{iso} = \frac{mep_{\,iso} \text{ x Volumeperminute}}{0.7457\text{x } 60}$ (As 1 hp = 0.7457 kW)

$HP_{iso} = \frac{179.18\text{x}1.954}{0.7457\text{x } 60}$, HP_{iso} = 7.825 hp

Horse power required for polytropic process,

$HP_{poly} = \frac{mep_{\,poly} \text{ x Volumeperminute}}{0.7457\text{x } 60}$, HP_{poly} = 9.69 hp

Horse power required for adiabatic process,

$HP_{adiabatic} = \frac{mep_{\,adiabatic} \text{ x Volumeperminute}}{0.7457\text{x } 60} = \frac{233.98\text{x}1.954}{0.7457\text{x } 60}$ = 10.22 hp

Isothermal efficiency = Isothermal process power required/ Actual power required

Isothermal efficiency$_{poly}$ $= \frac{HP_{iso}}{HP_{poly}}$ = 0.8075 or 80.75%

Isothermal efficiency$_{adiabatic}$ $= \frac{HP_{adabatic}}{HP_{poly}}$ = 0.7657 or 76.57%

2) A single stage single acting reciprocating air compressor has air entering at 1 bar, 20°C and compression occurs following polytropic process with index 1.2 upto the delivery pressure of 12 bar. The compressor runs at the speed of 240 rpm and has L/D ratio of 1.8. The compressor has mechanical efficiency of 0.88. Determine the isothermal efficiency

and cylinder dimensions. Also find out the rating of drive required to run the compressor which admits 1 m3 of air perminute.

Solution:

Using perfect gas equation the mass of air delivered per minute can be obtained as,

$$m = \frac{PV}{RT} = \frac{(1\times10^2\times1)}{(0.287\times293)}$$

= 1.189 kg/min

Compression process follows $PV^{1.2}$ = constt.

Temperature at the end of compression;

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

$$T_2 = 293\left(\frac{12}{1}\right)^{\frac{1.2-1}{1.2}}$$

T_2 =443.33 K

Work required during compression process $W=\left(\frac{n}{n-1}\right)\times mR(T_2-T_1)$

$=\left(\frac{1.2}{1.2-1}\right)\times 1.189\times 0.287(433.33-293)$

W = 307.79 kJ/min = 307.79/(60×0.7457)hp

W = 6.88 hp

Capacity of drive required to run compressor = 6.88/0.88= 7.82 hp

Isothermal work required for same compression,

$$W_{iso} = m\,R\,T_1\,ln\left(\frac{P_2}{P_1}\right)$$

= 1.189×0.287×293× $ln\left(\frac{12}{1}\right)$ = W_{iso} = 248.45 kJ/min

Isothermal efficiency = Isothermal work /Actual work = 248.45 / 307.79 = 0.8072

Volume of air entering per cycle = 1/240= 4.167×10^{-3} m^3/cycle

Volume of cylinder = $4.167\times10^{-3} = (\pi/4)D^2L$

$= 4.167\times10^{-3} = (\pi/4) \times D \times 1.8D$

Bore, D = 0.1434 m or 14.34 cm

Stroke length L = 1.8 D = 1.8 ×14.34 = 25.812 cm

3) A reciprocating compressor of single stage and double acting type is running at 200 rpm with mechanical efficiency of 85%. Air flows into compressor at the rate of 5 m3/min measured at atmospheric condition of 1.02 bar, 27°C. Compressor has compressed air leaving at 8 bar with compression following polytropic process with index of 1.3. Compressor has clearance volume of 5% of stroke volume. During suction of air from atmosphere into compressor its temperature rises by 10°C. There occurs pressure loss of 0.03 bar during suction and pressure loss of 0.05 bar during discharge passage through valves. Determine the dimensions of cylinder, volumetric efficiency and power input required to drive the compressor if stroke to bore ratio is 1.5.

Solution:

Considering the losses at suction and discharge, the actual pressure at suction and delivery shall be as under.

Atmospheric pressure, P_a = 1.02 bar, T_a = 273 + 27 = 300 K, V_a = 5 m^3/min

Pressure at suction, P_1 = 1.02 – 0.03 = 0.99 bar

T_1 = 300 + 10 = 310 K

Pressure at delivery, P_2 = 8 + 0.05 = 8.05 bar

Volume corresponding to suction condition of $P1$, $T1$,

$$V_1 = \frac{P_a\ T_1\ V_a}{P_1\ Ta} = (1.02\times310\times 5)/0.99\times300 = 5.32 \text{ m3/min}$$

$$W = \frac{n}{n-1}\ P_1\,V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$W = \frac{1.3}{1.3-1} \times 0.99 \times 10^2 \times 5.32\ \left[\left(\frac{8.05}{0.99}\right)^{\frac{0.3}{1.3}} - 1\right] = 23.66 \text{ kW or } 31.73 \text{ hp}$$

Power input required = 31.7 / 0.85= 37.33 hp

Volumetric efficiency, $\eta_{VOL} = \frac{T_a\, P_1}{P_a\ \ T_1}\left[1 + C - C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]$ $C = 0.05$,

$$\eta_{VOL} = \frac{0.99 \times 300}{1.02 \times 310}\left[1 + 0.05 - 0.05\left(\frac{8.03}{0.99}\right)^{\frac{1}{1.3}}\right]$$

= 0.7508 or 75.08%

Stroke volume per cycle = 5/ (2 × 200) = 0.0125 m^3/cycle

Actual stroke volume taking care of volumetric efficiency = 0.0125/0.7508 = 0.0167 m^3/cycle

Stroke volume = 00.0167 = $(\pi/4)D^2 L$

=00.0167 = $(\pi/4)D^2\ 1.5D$

D = 0.2420 m or 24.20 cm

Stroke L = 1.5 D = 36.3 cm

4) A reciprocating air compressor has four stage compression with 2 m 3 /min of air being delivered at 150 bar when initial pressure and temperature are 1 bar, 27°C. Compression occur polytropically following polytropic index of 1.25 in four stages with perfect intercooling between stages. For the optimum intercooling conditions determine the intermediate pressures and the work required for driving compressor.

Solution:

Here there is four stage compression with perfect intercooling at optimum intercooling conditions.

So optimum stage pressure ratio = $(150)^{1/4}$ = 3.499 = 3.5

Intermediate pressure shall be as follows:

Between Ist and IInd stage = 3.5 bar

Between IInd and IIIrd stage = 12.25 bar

Between IIIrd and IVth stage = 42.87 bar

Intermediate pressure: 3.5 bar, 12.25 bar, 42.87 bar.

Since it is perfect intercooling so temperature at inlet of each stage will be 300K.

So temperature at the end of fourth stage,

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

$$T = 300(3.5)^{\frac{0.25}{1.25}}$$

T = 385.42 K

Mass of air, kg/min, $m = \frac{PV}{RT} = \frac{(150\times10^2\times2)}{(0.287\times385.42)} = 271.21$ kg/min

Work required for driving compressor,

$$W = \frac{n}{n-1}\, mRT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \times 4$$

$$W = \frac{1.25}{1.25-1} 271.21 \times 0.287 \times 300 \left[(3.5)^{\frac{1.25-1}{1.25}} - 1\right] \times 4$$

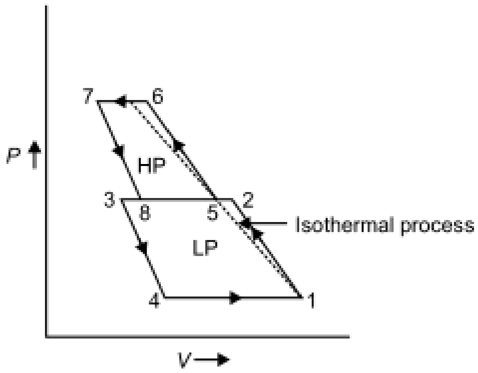
=132978.04 kJ/min or 2972.11 hp Work input = 2972.11 hp

5) In a two stage reciprocating air compressor running at 200 rpm the air is admitted at 1 bar, 17°C and discharged at 25 bar. At low pressure stage suction conditions the rate of air flow is 4 kg/minute. The low pressure cylinder and high pressure cylinders have clearance volumes of 4% and 5% of respective cylinder stroke volumes. The index for compression and expansion processes in two stages are sameas 1.25. Considering an optimum and perfect intercooling in between two stages determine the power required, isothermal efficiency, free air delivered, heat transferred in each cylinder and the cylinder volumes

Solution:

For the optimum intercooling the pressure ratio in each stage = $\sqrt{\frac{25}{1}}$ = 5

$$\frac{P_2}{P_1} = \frac{P_6}{P_5} = 5$$



Perfect intercooling indicates, T 1 = T 5 = 273 + 17 = 290 K

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} = 400.12 \text{ K}$$

$$T_6 = T_5 \left(\frac{P_6}{P_5}\right)^{\frac{n-1}{n}} = 400.12 \text{ K}$$

Actual compression work requirement, W = W HP + W LP

$$\frac{n}{n-1}\, mRT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \times 2$$

$$= \frac{1.25}{1.25-1}\, 4 \times 0.287 \times 290 \left[(5)^{\frac{1.25-1}{1.25}} - 1\right] \times 2$$

W = 1264.19 kJ/min or 28.25 hp

Work requirement if the process is isothermal compression,

$$W_{iso} = m\, R\, T_1\, ln\left(\frac{P_6}{P_1}\right)$$

$$W_{iso} = 4 \times\ 0.287 \times 290\, ln(25)$$

W_{iso} = 1071.63 kJ/min

Isothermal efficiency = $\frac{W_{iso}}{W}$ = 0.8477 or 84.77%

Free air delivered = $\frac{mRT_1}{P_1} = \frac{(4\times0.287\times290)}{(1\times10^2)}$ = 3.33 m^3 /min

Heat transferred in HP cylinder = Heat transferred in LP cylinder = Q

(Due to optimum and perfect intercooling)

Q = $\left(\frac{W}{2}\right) - mC_P(T_2 - T_1)$

Q = $\left(\frac{1264.19}{2}\right) - 4 \times 1.0032 \times (400.12 - 290)$

Q = 190.21 kJ/min

Volumetric efficiency, $\eta_{VOL} = \frac{T_a\, P_1}{P_a\ T_1}\left[1 + C - C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]$

Here the ambient conditions and suction conditions are same so expression gets modified as,

$$\eta_{VOL} = \left[1 + C - C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]$$

Volumetric efficiency of HP,

$$\eta_{VOL\ HP} = \left[1 + C_{HP} - C_{HP}\left(\frac{P_6}{P_5}\right)^{\frac{1}{n}}\right]$$

$C_{HP} = 0.04$ $= 1 + 0.04 - 0.04\,(5)^{1/1.25}$

$\eta_{VOL\ HP} = 0.895$ or 89.5%

Volumetric efficiency of LP,

$\eta_{VOL\ LP} = \left[1 + C_{LP} - C_{LP}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]$ $C_{LP} = 0.05$

$= 1 + 0.05 - 0.05\,(5)^{1/1.25} = 0.8688$ or 86.88%

$$Stroke\ volume\ of\ HP\ cylinder = \frac{Freeairdelivery}{Pressureratio \times speed \times\ \eta_{VOL\ HP}}$$

$V_{s\,HP} = \frac{3.33}{5\times200\times0.895} = 3.721 \times 10^{-3}\ m^3$

Clearance volume, $V_{c,\,HP} = 0.05 \times 3.721 \times 10^{-3} = 1.861 \times 10^{-4}\ m^3$

Total HP cylinder volume, $V_{HP} = V_{s,\,HP} + V_{c,\,HP} = 3.907\times10^{-3}\ m^3$

$V_{c,\,HP}$ = Clearance volume of HP

Stroke volume of LP cylinder =Free air delivery / (Speed× $\eta_{VOL\ LP}$)

$= 3.33/(200\times0.8688) = V_{s,\,LP} = 0.01916\ m^3$

Clearance volume, $V_{c,\,LP} = 0.04 \times V_{s,\,LP} = 7.664\times10^{-4}\ m^3$

Total LP cylinder volume, $V_{LP} = V_{s,\,LP} + V_{c,\,LP} = 0.019926\ m^3$

6) A two stage double acting reciprocating air compressor running at 200 rpm has air entering at 1 bar, 25°C. The low pressure stage discharges air at optimum intercooling pressure into intercooler after which it enters at 2.9 bar, 25°C into high pressure stage. Compressed air leaves HP stage at 9 bar. The LP cylinder and HP cylinder have same stroke lengths and equal clearance volumes of 5% of respective cylinder swept volumes. Bore of LP cylinder is 30 cm and stroke is 40 cm. Index of compression for both stages may be taken as 1.2.

Determine, (i) the heat rejected in intercooler, (ii) the bore of HP cylinder, (iii) the hp required to drive the HP cylinder.

SOLUTION:

Optimum intercooling pressure = $\sqrt{9}$ = 3 bar

LP stage pressure ratio = HP stage pressure ratio = 3

From the given dimensions of LP cylinder, the volume of LP cylinder, in m^3 /min

$V_{LP} = \frac{\pi}{4} X\ 0.30\ X\ 0.40 \times 200 \times 2$

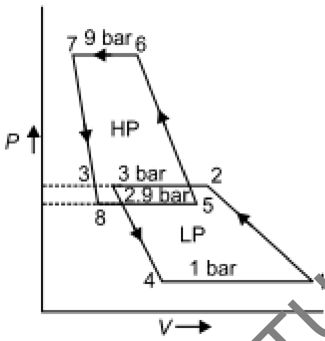
$V_{LP} = 11.31\ m^3$ /min

Volumetric efficiency of LP compressor, here ambient and suction conditions are same,

$$\eta_{VOL\ LP} = \left[1 + C_{LP} - C_{LP}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]$$

$$\eta_{VOL\ LP} = \left[1 + 0.05 - 0.05\left(\frac{3}{1}\right)^{\frac{1}{n}}\right]$$

= 0.9251 or 92.51%



Volume of air inhaled in LP stage = $V_{LP} \times \eta_{VOL,LP}$

= 11.31 × 0.9251

= 10.46 m^3 /min

Mass of air per minute, $m = \frac{P_1 V_1}{T_1 R} = \frac{(1\times10^2\times10.46)}{(0.287\times298)}$

$= 12.23\ kg/min$

Temperature after compression in LP stage,

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

$$T_2 = 298\left(\frac{3}{1}\right)^{\frac{1.2-1}{1.2}}$$

T_2 = 357.88 K

Volume of air going into HP cylinder $V_1 = \frac{mRT_5}{P_5}$, After intercooling, T_5 = 298 K, P_5 = 2.9 bar,

$$= \frac{12.23 \times 0.287 \times 298}{2.9}$$

V_5 = 3.61 m 3 /min

Since the clearance volume fraction and pressure ratio for both HP and LP stages are same so the
volumetric efficiency of HP stage referred to LP stage suction condition shall be same

$$\eta_{VOL\ LP} = \eta_{VOL\ HP} = 0.9251$$

Hence, the volume of HP cylinder/min= $\frac{V_5}{\eta_{VOL\ HP}} = \frac{3.61}{0.9251} = 3.902$ m 3 /min

Let bore of HP cylinder be D_{HP}

$3.902 = (\pi/4) \times D_{HP}^2 \times 0.40 \times 2 \times 200$

D_{HP} = 0.1762 m or 17.62 cm

Heat rejected in intercooler, Q = m C_p ($T_2 - T_5$)

= 12.23 ×1.0032 × (357.88 – 298)

= 734.68 kJ/min

In HP stage, $T_6 = T_5 \left(\frac{P_6}{P_5}\right)^{\frac{n-1}{n}}$

$$T_6 = 298\left(\frac{9}{2.9}\right)^{\frac{1.2-1}{1.2}}$$

T 6 = 359.91 K

Work input required for HP stage, $W_{HP} = \frac{n}{n-1} mR(T_6 - T_5)$

$$\frac{1.2}{1.2-1} \times 12.23 \times 0.287 \times (359.9 - 298)$$

W $_{HP}$ = 1303.62 kJ/min

or W $_{HP}$ = 29.14 hp

7) During an experiment on reciprocating air compressor the following observations are being taken; Barometer reading = 75.6 cm Hg, Manometer reading across orifice = 13 cm Hg. Atmospheric temperature = 25°C. Diameter of orifice = 15 mm. Coefficient of discharge across the orifice = 0.65 Take density of Hg = 0.0135951 kg/cm 3 Determine the volume of free air handled by compressor in m 3 /min.

Solution:

Cross-sectional area of orifice, A = $\frac{\pi}{4}\times(15\times10^{-3})^2 = 1.77\times10^{-4}$ m^2

Atmospheric pressure = $75.6\times0.0135951\times9.81\times10^4\times10^{-3}$= 100.83 kPa

Specific volume of air per kg at atmospheric conditions, $v = \frac{RT}{P} = \frac{0.287\times298}{100.83} = 0.848$ m 3 / kg

Density of air = 1/v = 1.18 kg/m^3

Pressure difference across orifice = $13\times0.0135951\times9.81\times10^4\times10^{-3}$ = 17.34 kPa

Height of air column for pressure difference across orifice.

$\rho_a \times h_a \times g = 17.34\times10^3$

$\rho_a = 1.18$ kg/m^3

$\therefore h_a = 1497.95$ m

Free air delivery = $C_d \times A\times\sqrt{2gha}$

$= 0.65\times1.77\times10^{-4}\times\sqrt{2\times9.81\times1497.95}$

= 0.01972 m^3/s or 1.183 m^3/min

. Free air delivery = 1.183 m^3/min

8) During a trial on single acting single stage compression the following observations are made;

Dimensions of cylinder: 10 cm bore and 8 cm stroke.

Speed of rotation: 500 rpm. Barometer reading: 76 cm Hg

Atmospheric temperature: 27°C

Delivery air temperature = 130°C

Free air delivery = 15 m 3 /hr

Spring balance of dynamometer type (electric motor) reading: 10 kg

Radius of arm of spring balance: 30 cm

Take mechanical efficiency = 0.90.

Determine the volumetric efficiency, shaft output per m 3 of free air per minute.

Solution:

Free air delivery = 15 m^3/hr = 0.25 m^3/min

Volume of cylinder = $\frac{\pi}{4}\times(0.10)^2\times(0.08) = 6.28\times10^{-4}$ m^3

Volumetric efficiency = $\frac{15/60}{6.28\times10^{-4}\times500}$ = 0.7962 or 79.62%

Shaft output = $\frac{2\ \pi NT}{60}$

Shaft output =$\frac{2\ \times\pi\times500\times10\times9.81\times0.30\times10^{-3}}{60}$

= 15.41 kJ/s or 20.66 hp

Shaft output per m 3 of free air per minute = 20.66 /0.25 = 82.64 hp per m^3 of free air per minute.

9) Determine the minimum number of stages required in an air compressor which admits air at 1bar, 27°C and delivers at 180 bar. The maximum discharge temperature at any stage is limited to 150°C. Consider the index for polytropic compression as 1.25 and perfect and optimum intercooling in between the stages. Neglect the effect of clearance.

Solution:

Let there be 'i' number of stages. So the overall pressure ratio considering inlet state as P_a and T_a

and delivery state pressure as P_i

$$\frac{P_i}{P_a} = \frac{P_1}{P_a}\times\frac{P_2}{P_1}\times\frac{P_3}{P_2}\times\ ....\times\frac{P_i}{P_{i-1}}$$

When perfect and optimum intercooling is considered then pressure ratio in each stage will be same.

$$\frac{P_1}{P_a} = \frac{P_2}{P_1} = \frac{P_3}{P_2} = \cdots = \frac{P_i}{P_{i-1}} = r$$

$$\frac{P_i}{P_a} = \ r^i$$

for any stage, say second stage, T_1 = 273 + 27 = 300 K

and T_2 = 273 + 150 = 423 K

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{\frac{n}{n-1}}, \frac{P_i}{P_a} = (r)^i = \left(\frac{T_2}{T_1}\right)^{\frac{in}{n-1}}$$

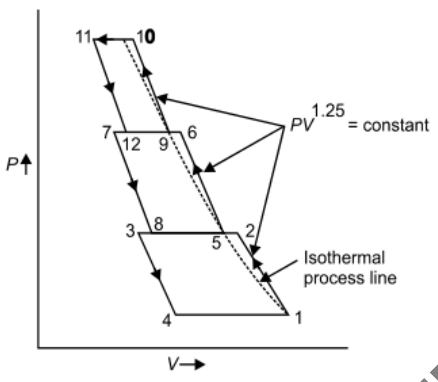
$$\frac{180}{1} = \left(\frac{423}{300}\right)^{\frac{i*1.25}{1.25-1}}$$

Taking log for solving,

$$\ln 180 = \frac{i * 1.25}{1.25 - 1} \times ln\frac{423}{300}$$

Solving, i = 3.022 say **3 stages**

10) In a triple stage reciprocating compressor of single acting type the air enters at 1 bar, 27°C. The compressor has low pressure cylinder with bore of 30 cm and stroke of 20 cm. Clearance volume of LP cylinder is 4% of the swept volume. The final discharge from compressor takes place at 20 bar. The expansion and compression index may be taken uniformly as 1.25 for all the stages. The intercooling between the stages may be considered to be at optimum intercooling pressure and perfect intercooling. Determine, the interstage pressures, effective swept volume of low pressure cylinder, temperature and volume of air delivered in each stroke and the work done per kg of air.



Solution:

Here P_1 = 1 bar, T_1 = 300 K, C = 0.04, P_{10} = 20 bar, n = 1.25, See Fig.

For optimum and perfect intercooling,

$$\frac{P_2}{P_1} = \frac{P_6}{P_{10}} = \frac{P_{10}}{P_6} = \left(\frac{20}{1}\right)^{\frac{1}{3}}$$

$$= 2.714$$

P_2 = 2.714 bar, $T_5 = T_1$ = 300 K

P_6 = 7.366 bar $T_9 = T_1$ = 300 K

Volumetric efficiency of LP stage,

$$\eta_{VOL\ LP} = \left[1 + C_{LP} - C_{LP}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]$$

$$\eta_{VOL\ LP} = \left[1 + 0.04 - 0.04(2.714)^{\frac{1}{1.25}}\right]$$

$$= 0.9511 \text{ or } 95.11\%$$

LP swept volume, $V_1 - V_3 = \frac{\pi}{4} \times (D)^2 \times (L)$

$$= \frac{\pi}{4} \times (0.30)^2 \times (0.20) = 0.01414 \text{ m}^3$$

Effective swept volume of LP cylinder, $V_1 - V_4 = \eta_{VOL\ LP} \times V_1 - V_3$

$$= 0.9511 \times 0.01414 = 0.01345 \text{ m}^3$$

Temperature of air delivered, $T_{10} = T_9 \times \left(\frac{P_{10}}{P_6}\right)^{\frac{0.25}{1.25}}$

$$= 300 \times (2.714)^{\frac{0.25}{1.25}}$$

$$= 366.31 \text{ K}$$

For the compression process of air as perfect gas;

$$\frac{P_1 \times (V_1 - V_4)}{T_1} = \frac{P_{10} \times (V_{10} - V_{11})}{T_{10}}$$

$$(V_{10} - V_{11}) = \frac{P_1 \times (V_1 - V_4) \times T_{10}}{T_1 P_{10}}$$

$$= \frac{0.01345 \times 1 \times 10^2 \times 366.31}{300 \times 20 \times 10^2}$$

Volume of air delivered = $V_{10} - V_{11} = 8.2115 \times 10^{-4}$ m^3

Total Work done per kg air,

$$W = \frac{n}{n-1}\, mRT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \times 3$$

$$W = \frac{1.25}{1.25-1} \times 0.287 \times 300T_1 \left[(2.714)^{\frac{1.25-1}{1.25}} - 1\right] \times 3$$

$$= 285.44 \text{ kJ/kg of air}$$

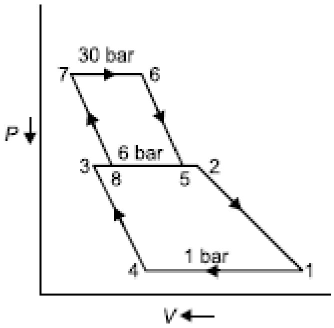
11) A two stage reciprocating air compressor has air being admitted at 1 bar, 27°C and delivered at 30 bar, 150°C with interstage pressure of 6 bar and intercooling up to 35°C. Compressor delivers at the rate of 2 kg/s. Clearance volumes of LP and HP cylinders are 5%

and 7% of stroke volume respectively. The index of compression and expansion are same throughout. Determine the swept volume of both cylinders in m^3/min, amount of cooling required in intercooler and total power required. Also estimate the amount of cooling required in each cylinder.

Solution:

Given: P_1 = 1 bar, T_1 = 300 K, P_2 = 6 bar, P_6 = 30 bar,

T_6 = 273 + 150 = 423 K, T_5 = 273 + 35 = 308 K, C_{LP} = 0.05, C_{HP} = 0.07, m = 2 kg/s



For process 5–6, $P_2 = P_5$

$$\frac{P_6}{P_5} = \left(\frac{T_6}{T_5}\right)^{\frac{n}{n-1}}$$

$$\frac{30}{6} = \left(\frac{423}{308}\right)^{\frac{n}{n-1}}$$

Taking log of both sides, $\ln(5) = \frac{n}{n-1}\ln(1.3734)$

solving we get, n = 1.245

Volumetric efficiency of LP cylinder,

$$\eta_{VOL\ LP} = \left[1 + C_{LP} - C_{LP}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]$$

$$\eta_{VOL\ LP} = \left[1 + 0.05 - 0.05\left(\frac{6}{1}\right)^{\frac{1}{1.245}}\right]$$

= 0.8391 or 83.91%

Volumetric efficiency of HP cylinder,

$$\eta_{VOL\ HP} = \left[1 + C_{HP} - C_{HP}\left(\frac{P_2}{P_1}\right)^{\frac{1}{1.245}}\right]$$

$$\eta_{VOL\ HP} = \left[1 + 0.07 - 0.07\left(\frac{30}{6}\right)^{\frac{1}{1.245}}\right]$$

$$= 0.815 \text{ or } 81.50\%$$

For suction of LP cylinder $P_1 \times (V_1 - V_4) = mRT_1$

$$(V_1 - V_4) = \frac{2 \times 0.287 \times 300}{1 \times 10^2}$$

$$= 1.722 \text{ m}^3/\text{s or } 103.32 \text{ m}^3/\text{min}$$

$$\eta_{VOL\ LP} = \frac{V_1 - V_4}{V_1 - V_3}$$

$V_1 - V_3 = 103.32/0.8391 = 123.13\text{m}^3/\text{min}$ =Swept volume of LP cylinder

For suction of HP cylinder $P_2 \times (V_5 - V_8) = mRT_5$

$$(V_5 - V_8) = \frac{2 \times 0.287 \times 308}{6 \times 10^2}$$

$$= 0.2946 \text{ m}^3/\text{s or } 17.676 \text{ m}^3/\text{min}$$

$$\eta_{VOL\ HP} = \frac{V_5 - V_8}{V_6 - V_7}$$

$V_6 - V_7$ =17.676/0.815 = 21.69 m^3/min =Swept volume of LP cylinder

For compression in LP stage

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

$$T_2 = 300_1\left(\frac{6}{1}\right)^{\frac{1.245-1}{1.245}}$$

$$T2 = 426.83 \text{ K}$$

Cooling required in intercooler, $Q_{IC} = m \times C_P \times (T_2 - T_5)$

= 2×1.0032 × (426.83 – 308)

Q_{IC} = 238.42 kJ/s Heat picked in intercooler = 238.42 Kw

Work input required = $W_{LP} + W_{HP}$

$$= \frac{n}{n-1}\, mRT_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] + = \frac{n}{n-1}\, mRT_5\left[\left(\frac{P_6}{P_5}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{n}{n-1}\, mR\left\{T_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] + T_5\left[\left(\frac{P_6}{P_5}\right)^{\frac{n-1}{n}} - 1\right]\right\}$$

$$= \frac{1.245}{1.245-1}\, mR\left\{300\left[\left(\frac{6}{1}\right)^{\frac{1.245-1}{1.245}} - 1\right] + 308\left[\left(\frac{30}{6}\right)^{\frac{1.245-1}{1.245}} - 1\right]\right\}$$

Total work required = 704.71 kW

Heat transferred in LP cylinder = Amount of cooling required in LP cylinder

$$Q_{LP} = m \times \frac{-n}{n-1} \times C_v \times (T_2 - T_1)$$

$$= 2 \times \frac{1.4 - 1.245}{1.245 - 1} \times 0.72 \times (426.83 - 300)$$

= 115.55 kJ/s   Amount of cooling required in LP cylinder = 115.55 kW

Heat transferred in HP cylinder = Amount of cooling required in HP cylinder

$$Q_{HP} = m \times \frac{-n}{n-1} \times C_v \times (T_6 - T_5)$$

$= 2 \times \frac{1.4-1.245}{1.245-1} \times 0.72 \times (423 - 308) = 104.77$ kJ/s

Amount of cooling required in HP cylinder = 104.77 kW

Frequently asked Questions

1) Classify the compressors.

2) Discuss the applications of compressed air to highlight the significance of compressors.

3) Obtain the volumetric efficiency of single stage reciprocating compressor with clearance volume and without clearance volume.

4) Discuss the effects of clearance upon the performance of reciprocating compressor.

5) Define isothermal efficiency. Also discuss its significance.

6) What do you understand by multistage compression? What are its' merits over single stage compression?

7) What is the optimum pressure ratio for perfect intercooling in between two stages of compression? The inlet and outlet pressures may be taken as *P*1 and *P*3.

8) A single stage single cylinder reciprocating compressor has 60 m3/hr air entering at 1.013 bar, 15°C and air leaves at 7 bar. Compression follows polytropic process with index of 1.35.

Considering negligible clearance determine mass of air delivered per minute, delivery temperature, indicated power and isothermal efficiency.

[**ANS:**1.225 kg/min, 202.37°C, 4.23 kW, 77.1%]

9) A reciprocating compressor of single stage and double acting type has free air delivered at 14 m3/min measured at 1.013 bar, 288 K. Pressure and temperature at suction are 0.95 bar and 305K. The cylinder has clearance volume of 5% of swept volume. The air is delivered at pressure of 7 bar and expansion and compression follow the common index of 1.3. Determine the indicated power required and volumetric efficiency with respect to free air delivery.

[**ANS:**63.55 kW, 72.4%]

10) A single stage double acting reciprocating compressor delivers 14 m3/min measured at suction states of 1 bar and 20°C. Compressor runs at 300 rpm and air is delivered after compression with compression ratio of 7. Compressor has clearance volume of 5% of swept volume and compression follows polytropic process with index 1.3. Determine the swept volume of cylinder and indicated power in hp.

[**ANS:**0.028 m3, 76.86 hp]

11) A single stage single acting reciprocating air compressor handles 0.5 m3/min of free air measured at 1 bar. Compressor delivers air at 6.5 bar while running at 450 rpm. The volumetric efficiency is 0.75, isothermal efficiency is 0.76 and mechanical efficiency is 0.80. Determine indicated mean effective pressure and power required to drive the compressor. [**ANS:**0.185 MPa, 3.44 hp]

12) A reciprocating compressor has two stages with inlet air going into LP stage at 1 bar, 16°C and at the rate of 12 m3/min. Air is finally delivered at 7 bar and there is perfect intercooling at optimum pressure between the stages. The index for compression is 1.25 and compressor runs at 600 rpm. Neglecting clearance volume determine intermediate pressure, total volume of each cylinder and total work required.

[**ANS:**2.645 bar, 0.02 m3, 0.0075 m3, 57.6 hp]

13) A two stage reciprocating air compressor delivers 4.2 kg of free air per min at 1.01325 bar and 15°C. The suction conditions are 0.95 bar, 22°C. Compressor delivers air at 13 bar. Compression throughout occurs following $PV1.25 = C$. There is optimum and perfect intercooling between the two stages. Mechanical efficiency is 0.75. Neglecting clearance volume determine

(*i*) the heat transfer in intercooler per second.

(*ii*) the capacity of electric motor.

(*iii*) the % saving in work if two stage intercooling is compared with single stage compressor between same limits.

[**ANS:**7.6 kJ/s, 44.65 hp, 13%]

Self Answered Question & Answer

1) A single stage single acting reciprocating air compressor handles 0.5 m3/min of free air measured at 1 bar. Compressor delivers air at 6.5 bar while running at 450 rpm. The volumetric efficiency is 0.75, isothermal efficiency is 0.76 and mechanical efficiency is 0.80. Determine indicated mean effective pressure and power required to drive the compressor. [**ANS:**0.185 MPa, 3.44 hp]

2) A single stage single acting reciprocating air compressor compresses air by a ratio of 7. The polytropic index of both compression and expansion is 1.35. The clearance volume is 6.2% of cylinder volume. For volumetric efficiency of 0.8 and stroke to bore ratio of 1.3 determine the dimensions of cylinder.

[**ANS:**14.67 cm and 19.08 cm]

3) A single stage single acting reciprocating air compressor runs with air entering at 1 bar and leaving at 7 bar following *PV*1.3 = constant. Free air delivery is 5.6 m3/minute and mean piston speed is 150 m/min. Take stroke to bore ratio of 1.3 and clearance volume to be $1/15^{th}$ of swept volume per stroke. The suction pressure and temperature are equal to atmospheric air pressure and temperature. Determine volumetric efficiency, speed of rotation, stroke and bore. Take mean piston speed = 2 stroke rpm.

[**ANS:**76.88%, 164 rpm, 45.7 cm, 35.1 cm]

4) A reciprocating compressor of single acting type has air entering at 1.013 bar, 15°C and leaving at 8 bar. Compressor is driven by electric motor of 30.84 hp and the mechanical efficiency is 0.87. The clearance volume is 7% of swept volume and the bore is equal to stroke. The compression and expansion follow *PV*1.3 = constant. Determine (*i*) free air delivered in m3/min, (*ii*) volumetric efficiency, and (*iii*) cylinder dimensions.

[**ANS:**4.47 m3/min, 72.68%, *L* = *D* = 29.7 cm]

5) A single acting reciprocating air compressor has two stages with the optimum and perfect intercooling in between. Compressor has air sucked at 1 bar and at the rate of 2.4 m3/min when measured at 1.013 bar, 288 K. Compressor delivers air at 70 bar. Temperature at the end of suction stroke is 32°C. The compression and expansion follows polytropic process *PV*1.25 = *C* uniformly. The clearance volume is 3% of swept volume in each HP and LP cylinder. Compressor runs at 750 rpm. If the mechanical efficiency is 0.85 then determine the power of drive required, swept volumes of each cylinder, % saving in power as compared to single stage compression within limits.

[**ANS:**35.8 hp, 3963 cm3, 473 cm3, 20.89%]

Test Your Skills

1) For reciprocating air compressor the law of compression desired is isothermal and that may be possible by

a) Very low speed b) very high speed c) any speed d) none of the above

2) Work input to thye air compressor with ‘n’ as index of compression

a) Increases with increase in value of ‘n’ b) decreases with increase in value of ‘n’ c) remain same for all value of ‘n’ d) first increase and then decrease with increase in value of ‘n’

3) The Clarence volume in reciprocating compressor is provided to

a) To reduce the work done b) to increase the volumetric efficiency c) to accommodate valves d) to create the turbulence

4) Suction pressure being atmospheric, increase in delivery pressure with fixed clearance volume

a) Increase in volumetric efficiency b) decrease in volumetric efficiency c) does not change in volumetric efficiency d) first increase and then decreases in volumetric efficiency

5) For the same overall pressure ratio, the leakage of air past the piston for multi satge compression as compared to single stage compression is,

a) More b) less c) constant d) may be more or less

6) In reciprocating air compressor the method of controlling the quantity of air delivered is done by

a) Throttle control b) blow-off control c) Clarence control d) all of the above

7) With increase in clearance volume, the ideal work of the compressing 1 kg of air

a) Increases b) decreases c) remain same d) first increase and then decreases

Answers: 1)-a, 2-a, 3)-c, 4)-b, 5)-b, 6)-d, 7)-c