MECHANICAL ENGINEER'S MANUAL

By : ENGR. YURI G. MELLIZA

DEFINITION OF TERMS

Mass: It is the absolute quantity of matter in it.

m – mass, kg

Conversion

1 kg = 2.205 lbs

1 Metric Ton = 1,000 kg

1 Ton = 2000 Lbs

Velocity: It is the distance per unit time.

$$v = \frac{d}{t} \quad \frac{m}{sec}$$

Where

d – distance, m

t – time, sec

Conversion

1 meter = 3.28 ft = 100 cm = 1000 mm

1 ft. = 12 inches

Acceleration: Rate of change of velocity with respect to time.

$$a = \frac{dv}{dt} \quad \frac{m}{sec^2}$$

Gravitational Acceleration: It is the acceleration due to gravity.

At standard condition or Sea level condition:

g = 9.81 m/sec^2 = 32.2 ft/sec^2

Force: It is the mass multiplied by the acceleration.

$$F = ma \ \ N\,(Newton)$$

$$F = \frac{ma}{1000} \ \ KN\,(Kilo\ Newton)$$

Newton: It is the force required to accelerate 1 kg mass at the rate of 1 m/sec per second.

Where:

m – mass in kg

a – acceleration in m/sec^2

Weight: It is the forge due to gravity.

$$W = mg \ \ N$$

$$W = \frac{mg}{1000} \ \ KN$$

where

$$g \text{ - gravitational acceleration, } \frac{m}{sec^2}$$

$$g = 9.81 \frac{m}{sec^2} \rightarrow (\text{Standard gravitational acceleration})$$

Force of Attraction: From Newton's Law of Gravitation the force of attraction between two masses m_1 and m_2 is given by the equation:

$$F_g = \frac{Gm_1m_2}{r^2} \quad \text{Newton}$$

Where:

m_1 and m_2 - masses in kg
r - distance apart in meters
G - Gravitational constant in N-m^2/kg^2

$$G = 6.670 \text{ x } 10^{-11} \frac{N-m^2}{kg^2}$$

PROPERTIES OF FLUIDS

Density (ρ) - it is the mass per unit volume.

$$\rho = \frac{m}{V} \ \frac{kg}{m^3}$$

Where:

m - mass, kg
V - volume , m^3

Conversion:

$$1\frac{lb}{ft^3} = 16.0185 \ \frac{kg}{m^3}$$

$$1\frac{kg}{cm^3} = 62{,}427.9606\frac{lb}{ft^3}$$

$$1\frac{g}{liter} = 1\frac{kg}{m^3}$$

Specific volume (υ) - it is the volume per unit mass or the reciprocal of its density.

$$\upsilon = \frac{V}{m} \ \frac{m^3}{kg}$$

$$\upsilon = \frac{1}{\rho} \ \frac{m^3}{kg}$$

Specific weight (γ) - it is the weight per unit volume.

$$\gamma = \frac{W}{V} \ \frac{KN}{m^3}$$

$$\gamma = \frac{mg}{1000V} \ \frac{KN}{m^3}$$

$$\gamma = \frac{\rho g}{1000} \ \frac{KN}{m^3}$$

$$\rho = \frac{1000\gamma}{g} \ \frac{kg}{m^3}$$

Specific Gravity Or Relative Density (S):

FOR LIQUIDS: Its specific gravity or relative density is equal to the ratio of its density to that of water at standard temperature and pressure.

$$S_L = \frac{\rho}{\rho_w} = \frac{\gamma_L}{\gamma_w}$$

At standard condition:

ρ_{water} = 1000 kg/m^3

γ_{water} = 9.81 KN/m^3

FOR GASES: Its specific gravity or relative density is equal to the ratio of its density to that of either air or hydrogen at some specified temperature and pressure

$$PV = mRT$$
$$\rho = \frac{P}{RT} \frac{kg}{m^3}$$
$$R = \frac{8.3143}{M} \frac{KJ}{kg - °K}$$

Where:

ρ_{ah} - density of either air or hydrogen at some value of P and T.

Equation of State for Gases

Conversion

$$°C = \frac{°F - 32}{1.8}$$
$$°F = 1.8(°C) + 32$$

Where:

P – absolute pressure in KPa

V – volume in m3

m – mass in kg

R – Gas constant in KJ/kg-K

T – absolute temperature in K.

M – molecular weight of gas, kg/kgmol

Specific gravities of some substances at 0°C

Substance	SG
Water	1.0
Blood	1.05
Seawater	1.025
Gasoline	0.7
Ethyl alcohol	0.79
Mercury	13.6
Wood	0.3–0.9
Gold	19.2
Bones	1.7–2.0
Ice	0.92
Air (at 1 atm)	0.0013

Temperature: It is the measure of the intensity of heat.

$$S_G = \frac{\rho_G}{\rho_{AH}}$$

FAHRENHEIT SCALE

Freezing Point = 32°F

Boiling Point = 212°F

CENTIGRADE SCALE

Freezing Point = 0°C

Boiling Point = 100°C

ABSOLUTE SCALE

°R = F + 460

K = °C + 273

Pressure: - is defined as the normal component of a force per unit area.

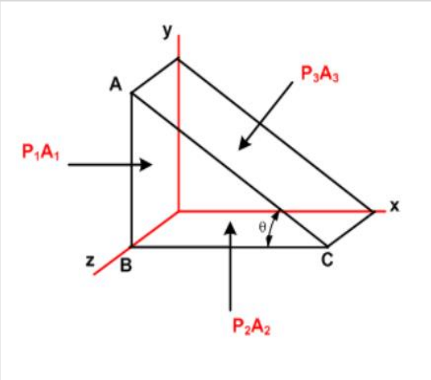
$$P = \frac{F}{A} \; \frac{KN}{m^2} \; \text{or} \; KPa \, (\text{Kilo Pascal})$$

$$1 \frac{KN}{m^2} = 1 \; KPa$$

If a force dF acts on infinitesimal area dA the intensity of pressure P is

$$P = \frac{dF}{dA}$$

PASCAL`S LAW
At any point in a homogeneous fluid at rest the pressures are the same in all directions.



$$\Sigma Fx = 0$$
$$P_1A_1 - P_3A_3 \sin\theta = 0$$
$$P_1A_1 = P_3A_3 \sin\theta \rightarrow eq.1$$
From Figure
$$\sin\theta = \frac{A_1}{A_3};$$
$$A_1 = A_3 \sin\theta \quad \rightarrow eq.2$$
Equation 2 to equation 1
$$P_1 = P_3$$

$$\Sigma Fy = 0$$
$$P_2A_2 - P_3A_3 \cos\theta$$
$$P_2A_2 = P_3A_3 \cos\theta \quad \rightarrow eq.3$$
$$\cos\theta = \frac{A_2}{A_3}$$
$$A_2 = A_3 \cos\theta \quad \rightarrow eq.4$$
Equation 4 to equation 3
$$P_2 = P_3$$
therefore
$$P_1 = P_2 = P_3$$

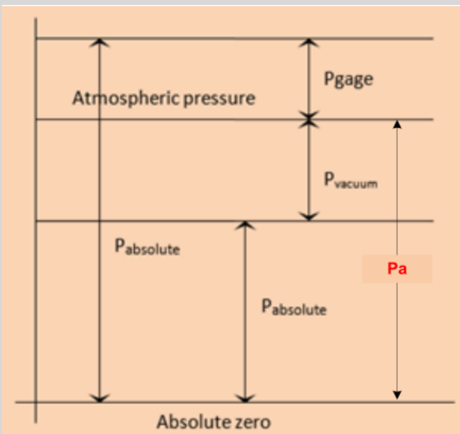
Atmospheric Pressure (Pa): It is the average pressure exerted by the atmosphere.

At sea level (Standard Condition)

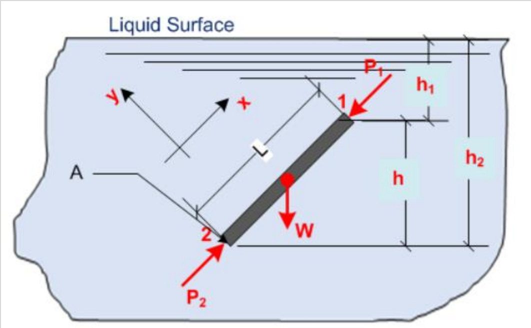
Pa	=	101.325	KPa
	=	0.101325	MPa
	=	760	mm Hg
	=	10.33	m of H_2O
	=	1.033	Kg/cm^2
	=	14.7	Lb/in^2
	=	29.921	in Hg
	=	33.88	Ft of H_2O
1Bar	=	100	KPa
1MPa	=	1000	KPa

Absolute And Gage Pressure
Absolute Pressure - is the pressure measured referred to absolute zero and using absolute zero as the base.
Gage Pressure - is the pressure measured referred to atmospheric pressure and using atmospheric pressure as the base.

$$P_{abs} = P_a + P_{Gage}$$
$$P_{abs} = P_a - P_{vacuum}$$



VARIATION OF PRESSURE WITH DEPTH OR ELEVATION



By applying equations for equilibrium

$$
\begin{aligned}
&\sum Fx = 0\\
&P_2A - P_1A - W\cos\theta = 0\\
&P_2A - P_1A = W\cos\theta\\
&(P_2 - P_1)A = W\cos\theta\\
&\text{But}\\
&W = \gamma V\\
&V = AL\\
&(P_2 - P_1)A = \gamma AL\cos\theta\\
&\text{from Figure}\\
&\cos\theta = \frac{h}{L}\\
&h = L\cos\theta\\
&(P_2 - P_1)A = -\gamma Ah\\
&(P_2 - P_1) = -\gamma h
\end{aligned}
$$

General Equation

$$dP = -\gamma dh$$

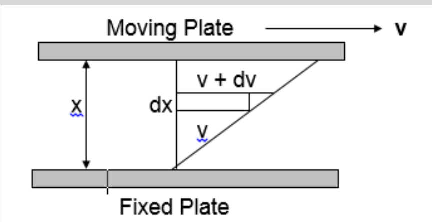
If the specific weight γ or density ρ is constant

$$\Delta P = -\gamma \Delta h$$

Note: Negative sign is used because pressure decreases with increasing elevation and increases with decreasing elevation.
h is positive if measured upward
h is negative if measured downward

Where:
P – Pressure in KPa
γ - specific weight in KN/m^3
h – elevation in meters

Viscosity: It is the property of a fluid that determines the amount of its resistance to shearing stress.



Assumptions:

1. Fluid particles in contact with the moving surface moves with the same velocity of that surface.
2. The rate of change of velocity dv/dx is constant in the direction perpendicular to the direction of motion.
3. The shearing is directly proportional to the rate of change of velocity.

let S - shearing stress in Pa or N/m^2

$$S \alpha \frac{dv}{dx}$$

$$S = \mu \frac{dv}{dx} = \mu \frac{v}{x}$$

$$\frac{dv}{dx} = \frac{v}{x}$$

$$\mu = S\left(\frac{x}{v}\right) \text{Pa - sec or } \frac{\text{N - sec}}{\text{m}^2}$$

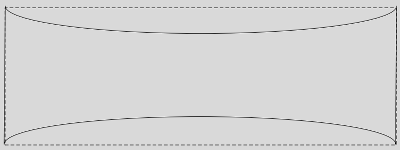
where:
μ - absolute or dynamic viscosity in Pa – sec or N – sec/m^2
S - shearing stress Pa
x - distance apart, m
v - velocity , m/sec

Kinematic Viscosity: It is the ratio of the absolute or dynamic viscosity to the mass density.

$$\nu = \frac{\mu}{\rho} \frac{m^2}{sec}$$

Elasticity: If a pressure is applied to a fluid, it contracts; if the pressure is released, it expands, the elasticity of a fluid is related to the amount of deformation(expansion or contraction) for a given pressure change. Quantitatively, the degree of elasticity is equal to:

$$Ev = -\frac{dP}{dV/V}$$



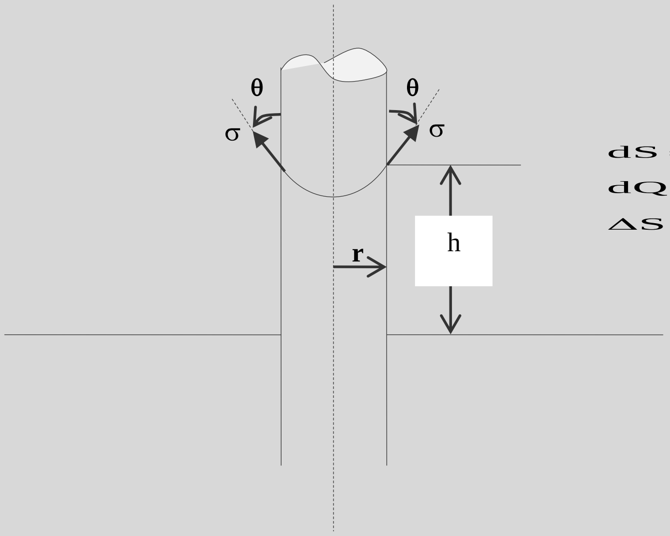
Where negative sign is used because dV/V is negative for a positive dP.

$$Ev = \frac{dP}{d\rho/\rho} \text{; because } -\frac{dV}{V} = d\rho/\rho$$

Where:

Ev - bulk modulus of elasticity
dV - is the incremental volume change
V - is the original volume
dP - is the incremental pressure change

Surface Tension: Capillarity



$$dS = \frac{dQ}{T}$$
$$dQ = TdS$$
$$\Delta S = \int \frac{dQ}{T}$$

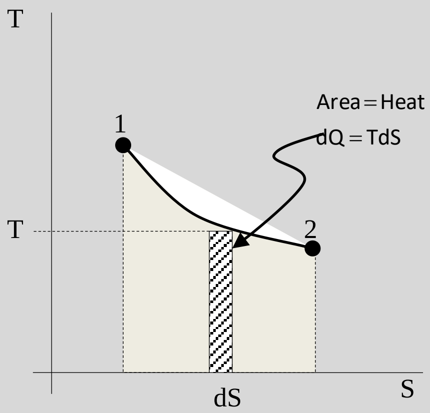
Where:

σ - surface tension, N/m
γ - specific weight of liquid, N/m3
r – radius, m
h – capillary rise, m

Table 1.Surface Tension of Water

°C	σ
0	0.0756
10	0.0742
20	0.0728
30	0.0712
40	0.0696
60	0.0662
80	0.0626
100	0.0589

Entropy (S): It is a property of a fluid that determines the amount of its randomness and disorder of a substance. If during a process an amount of heat is taken at certain instant and is divided by the absolute temperature at which it is taken the result is called the “Change of Entropy”

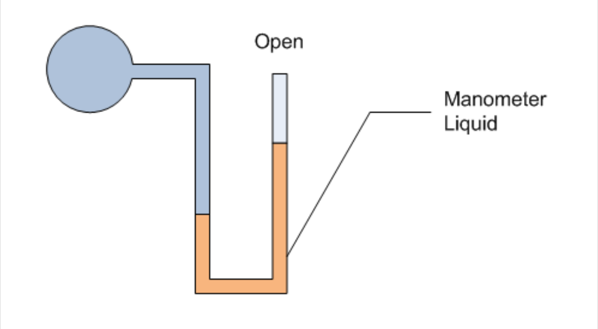


$$h = \frac{2\sigma\cos\theta}{\gamma r}$$

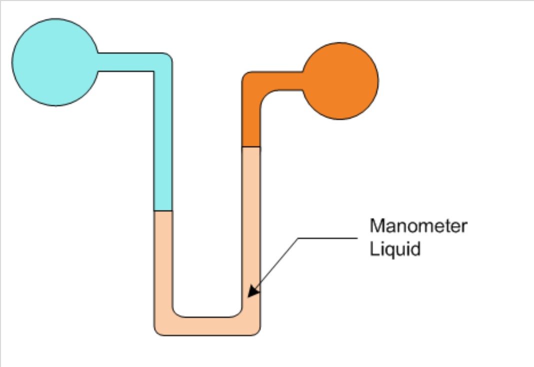
MANOMETERS

Manometer is an instrument used in measuring gage pressure in length of some liquid column.
1. *Open Type Manometer* : It has an atmospheric surface and is capable in measuring gage pressure.
2. *Differential Type Manometer* : It has no atmospheric surface and is capable in measuring differences of pressure.

Open Type

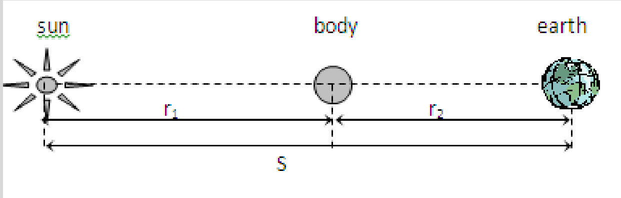


Differential Type



SAMPLE PROBLEMS

Problem No. 1 (Force of Attraction)
How far from the earth must a body be along a line toward the sun so that the gravitational pull of the sun balances that of the earth? Earth to sun distance is 9.3 x 107 mi; mass of sun is 3.24 x105 times mass of earth. (1.63 x 105 mi.)



Given
$S = 9.3 \times 10^7$ miles
$m_s = 3.24 \times 10^5 m_e$
m_b - mass of body
m_e - mass of sun
$S = r_1 + r_2$
$F_{g1} = F_{g2}$

$$F_{g1} = \frac{Gm_s m_b}{r_1^2}$$

$$F_{g2} = \frac{Gm_b m_e}{r_2^2}$$

$$F_{g1} = F_{g2}$$

$$\frac{Gm_s m_B}{r_1^2} = \frac{Gm_E m_B}{r_2^2}$$

$$\frac{(3.24 \times 10^5 m_E)}{r_1^2} = \frac{m_E}{r_2^2}$$

$$\frac{(3.24 \times 10^5)}{r_1^2} = \frac{1}{r_2^2}$$

$$S = r_1 + r_2$$

$$r_1 = (S - r_2)$$

$$\frac{(3.24 \times 10^5)}{(S - r_2)^2} = \frac{1}{r_2^2}$$

$$(S - r_2)^2 = (3.24 \times 10^5) r_2^2$$

$$(S - r_2) = r_2 \sqrt{(3.24 \times 10^5)}$$

$$r_2 = \frac{S}{1 + \sqrt{(3.24 \times 10^5)}} = 1.63 \times 10^5 \text{ miles}$$

Problem No. 2 (Measuring Temperature)

If the °F scale is twice the C scale, what will be the corresponding reading in each scale? (160° ; 320°)

$$°C = \frac{°F - 32}{1.8}$$

$$°F = 1.8°C + 32$$

$$°F = 2°C$$

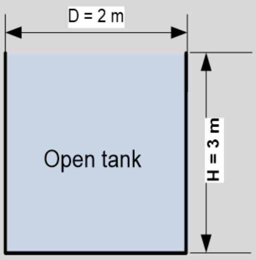
$$2°C = 1.8°C + 32$$

$$°C = \frac{32}{2 - 1.8} = 160°$$

$$°F = 320°$$

Problem No. 3 (Density, Specific gravity)

A cylindrical tank 2 m diameter, 3 m high is full of oil. If the specific gravity of oil is 0.9, what is the mass of oil in the tank? (8482.3 kg)



$$V_{cylinder} = \frac{\pi}{4} D^2 H = \frac{\pi}{4}(2^2)3 = 3\pi \text{ m}^3$$

$$S = \frac{\rho}{\rho_{water}}$$

$$0.9 = \frac{\rho}{1000}$$

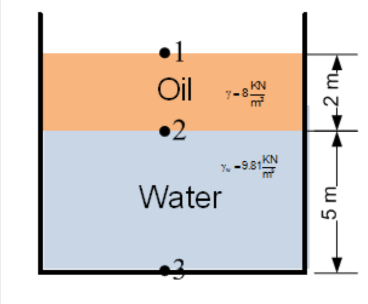
$$\rho = 900 \frac{kg}{m^3}$$

$$\rho = \frac{m}{V}$$

$$m = 900(3\pi) = 8482.3 \text{ kg}$$

Problem No. 5 (Variation in Pressure)

An open tank contains 5 m of water covered with 2 m of oil (γ = 8 KN/m3). Find the pressure at the interface and at the bottom of the tank.



$$P_2 = 0 + 8(2) = 16\ \text{KPa}$$

$$P_3 = 0 + 16 + 9.81(5) = 65.05\ \text{KPa}$$

Problem No. 6 (Force)

10 liters of an incompressible liquid exert a force of 20 N at the earth's surface. What force would 2.3 Liters of this liquid exert on the surface of the moon? The gravitational acceleration on the surface of the moon is 1.67 m/sec^2.

$$F = ma$$

$$\rho = \frac{m}{V}$$

$$m = \rho V = \rho(10.0)\left(\frac{1\,m^3}{1000\,L}\right)kg$$

$$20 = \frac{\rho(10.0)}{1000}(9.81)\,N$$

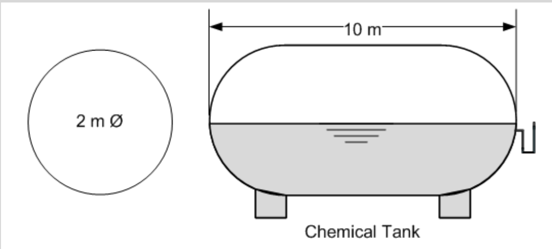
$$\rho = 203.874\frac{kg}{m^3}$$

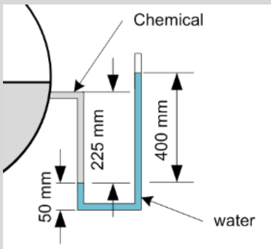
On the surface of the moon

$$F = 203.874\left(\frac{2.3}{1000}\right)(1.67) = 0.783\,N$$

Problem No. 7 (Manometer)

An open manometer is used to measure the pressure in the tank. The tank is half filled with 50,000 kg of a liquid chemical that is not miscible in water. The manometer tube is filled with liquid chemical. What is the pressure in the tank relative to atmospheric pressure?





$$V = \left[\frac{4}{3}\pi r^3 + \pi r^2(6)\right]\frac{1}{2} = \left[\frac{2}{3}\pi r^3 + \pi r^2(3)\right] = \pi r^2\left[\frac{2}{3}r + (3)\right] = 54.45\,m^3$$

$$\rho = \frac{m}{V} = \frac{50{,}000}{54.45} = 918.2\frac{kg}{m^3}$$

$$\gamma = \frac{\rho g}{1000} = \frac{918.2(9.81)}{1000} = 9.008\frac{KN}{m^3}$$

$$P + 9.008(0.225) - 0.350(9.81) = 0$$

$$P = 1.4067\,KPa$$

$$Pa_{bs} = 101.325 + 1.4067 = 102.73\,KPa$$

Problem No. 8 (Temperature)
If the temperature inside a furnace is 700 K, what is the corresponding reading in °F? (800.6)
Solution:

$t = 700 - 273 = 427°C$
$°F = (427)(1.8) + 32$
$°F = 800.6°F$

Problem No. 9 (Absolute Pressure)
The suction pressure of a pump reads 540 mm Hg vacuum. What is the absolute pressure in KPa? (29.33)
Solution:

$P = 760 - 540 = 220$ mm Hg absolute

$$P = \frac{220}{760}(101.325) = 29.33 \text{ mm Hg}$$

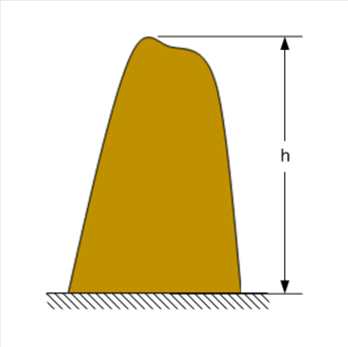
Problem No. 10 (Pressure Variation)
A storage tank contains oil with a specific gravity of 0.88 and depth of 20 m. What is the hydrostatic pressure at the bottom of the tank in kg/cm2.(1.7)
Solution:

Using: g = 9.81 m/sec2

$$P = 0 + \frac{0.88(1000)(9.81)(20)}{1000} = 172.656 \text{ KPa}$$

P = 1.7 kg/cm2

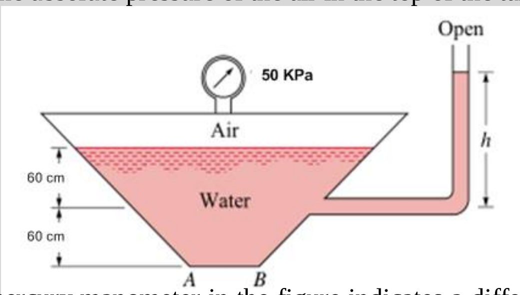
Problem No. 11 (Variation in Pressure Measuring Altitude of Mountain)
A hiker carrying a barometer that measures 101.3 KPa at the base of the mountain. The barometer reads 85 KPa at the top of the mountain. The average air density is 1.21 kg/m3. Determine the height of the mountain.



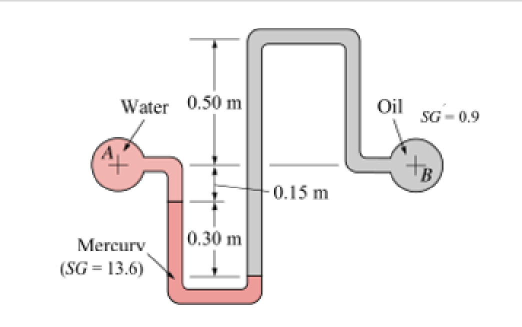
$$dP = -\gamma dh$$
$$P_2 - P_1 = -\gamma(h_2 - h_1)$$
$$h_2 - h_1 = h$$
$$h = \frac{(P_1 - P_2)1000}{\rho g}$$
$$h = \frac{(101.3 - 85)1000}{1.21(9.81)}$$
$$h = 1373 \text{ m}$$

MANOMETER PROBLEMS

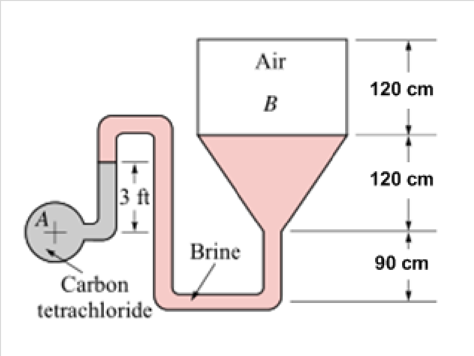
1. The closed tank in the figure is filled with water. The pressure gage on the tank reads 48 KPa. Determine
   a. The height h in mm in the open water column
   b. The gage pressure acting on the bottom of the tank surface AB
   c. The absolute pressure of the air in the top of the tank if the local atmospheric pressure is 101 KPa absolute.



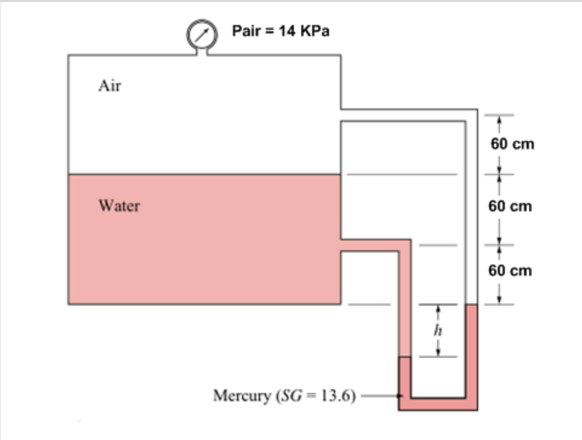
2. The mercury manometer in the figure indicates a differential reading of 30 m when the pressure in pipe A is 30 mm Hg vacuum. Determine the pressure in pipe B.



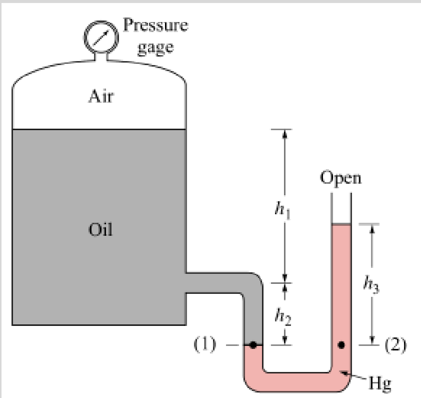
3. In the figure pipe A contains carbon tetrachloride (S = 1.60) and the closed storage tank B contains a salt brine (S = 1.15). Determine the air pressure in tank B in KPa if the gage pressure in pipe A is 1.75 kg/cm^2.



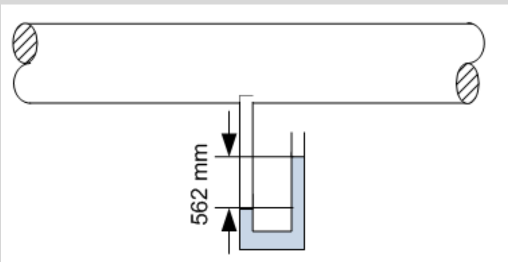
4. A U tube mercury manometer is connected to a closed pressurized tank as shown. If the air pressure is 14 KPa, determine the differential reading h. The specific weight of the air is negligible.



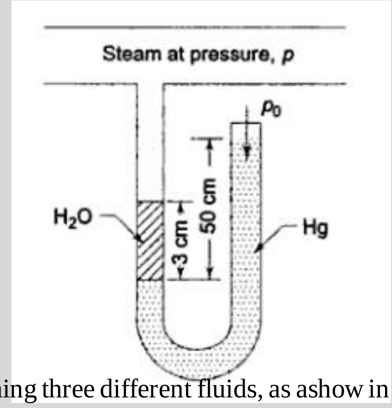
5. A closed tank contains compressed air and oil (S = 0.90) as shown in the figure.a U – tube manometer using mercury (S = 13.6) is connected to the tank as shown. For column heights h1 = 90 cm; h2 = 15 cm and h3 = 22 cm, determine the pressure reading of the gage.



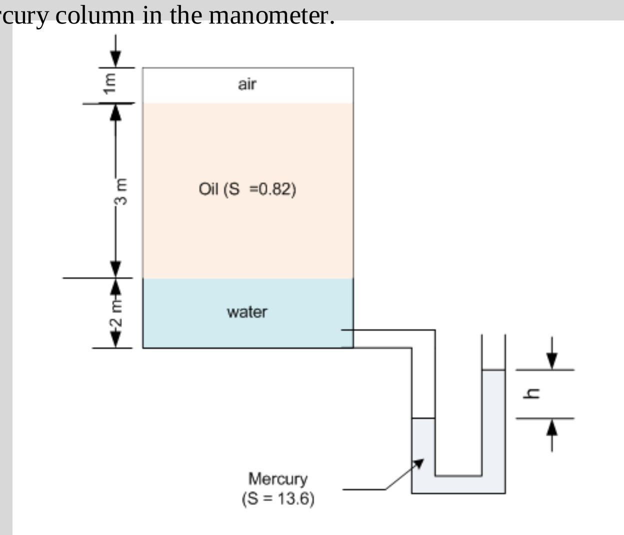
6. The pressure of gas in a pipeline is measured with a mercury manometer having one limb open to the atmosphere. If the difference in the height of mercury in the limbs is 562 mm, calculate the absolute gas pressure. The barometer reads 761 mm Hg, the acceleration due to gravity is 9.79 m/sec^2 and S_{Hg} = 13.64.



7. A turbine is supplied with steam at a gauge pressure of 1.4 MPa, after expansion in the turbine the steam flows into a condenser which is maintained at a vacuum of 710 mm Hg. The barometric pressure is 772mm Hg. Express the inlet and exhaust pressure in kg/cm^2 . Take the S of mercury is 13.6.
8. The pressure of steam flowing in a pipe line is measured with a mercury manometer. Some steam condenses in to water. /estimate the steam pressure in KPa. Take the density of mercury as 13,600 kg/m^3, the barometer reading as 76.1 cm Hg and g = 9.806 m/sec^2.

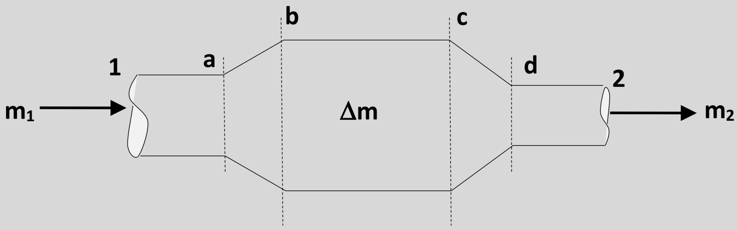


9. A manometer is attached to a tank containing three different fluids, as ashow in the figure below. What will be the difference in elevation h of th mercury column in the manometer.



LAW OF CONSERVATION OF MASS

Mass is indestructible. In applying this law we must except nuclear processes during which mass is converted into energy.



The verbal Form of the law is:

Mass Entering - Mass Leaving = Change of Mass Stored in the system

In equation form:

$$m_1 - m_2 = \Delta m$$

For a Closed System, a system of fixed mass, no equation is necessary. But for a steady-state, steady flow (SSSF) system (OPEN SYSTEM) Δm = 0. Therefore:

$$m_1 - m_2 = 0\text{; therefore}$$
$$m_1 = m_2$$

CONTINUITY EQUATION:

For one dimensional flow, the mass flow rate, m in kg/sec is equal to

$$m = \rho A v = \frac{Av}{\upsilon}$$

From $m_1 = m_2 = m$

$$\rho_1 A_1 v_1 = \rho_2 A_2 v_2 = \rho A v$$
$$\frac{A_1 v_1}{\upsilon_1} = \frac{A_2 v_2}{\upsilon_2} = \frac{Av}{\upsilon}$$

Where:

m – mass flow rate in kg/sec
A - cross sectional area in m^2
v - velocity in m/sec
υ - specific volume in m^3/kg
ρ - density in kg/m^3

Problem No. 1
A counter flowing heat exchanger is used to cool air at 540 K, 400 KPa to 360 K by using 0.05 kg/sec supply of water at 20°C, 200 KPa. The airflow is 0.5 kg/sec in a 10 cm diameter pipe. Find the inlet velocity and the water exit temperature.

Problem No.2

A water line with an internal radius of 6.5×10^{-3} m is connected to a shower head that has 12 holes. The speed of the water in the line is 1.2 m/s. (a) What is the volume flow rate in the line? (b) At what speed does the water leave one of the holes (effective hole radius = 4.6×10^{-4} m) in the head?

Problem No. 3
Water flows through a pipe of radius 8.0 cm with a speed of 10.0 m/s. It then enters a smaller pipe of radius 3.0 cm. What is the speed of the water as it flows through the smaller pipe? Assume that the water is incompressible.

MCQ PROBLEMS

1. If a liquid enters a pipe of diameter d with a velocity v, what will it's velocity at the exit if the diameter reduces to 0.5d?
   a) v
   b) 0.5v
   c) 2v
   d) 4v

2. The continuity equation is based on the principle of
   a) conservation of mass
   b) conservation of momentum
   c) conservation of energy
   d) conservation of force

3. Two pipes of diameters d1 and d2 converge to form a pipe of diameter d. If the liquid flows with a velocity of v_1 and v_2 in the two pipes, what will be the flow velocity in the third pipe? (d)

a) $\frac{d_1 v_1 + d_2 v_2}{d}$

b) $\frac{d^2 (v_1 + v_2)}{d}$

c) $\frac{(d_1 + d_2)^2 (v_1 + v_2)}{d^2}$

d) $\frac{d_1^2 v_1 + d_2^2 v_2}{d^2}$

4. Two pipes of diameters d1 and d2 converge to form a pipe of diameter 2d. If the liquid flows with a velocity of v1 and v2 in the two pipes, what will be the flow velocity in the third pipe?

a) v1 + v2
b) v1 + v2/2
c) v1 + v2/4
d) 2(v1 + v2)

5. Two pipes, each of diameter d, converge to form a pipe of diameter D. What should be the relation between d and D such that the flow velocity in the third pipe becomes double of that in each of the two pipes?

a) D = d
b) D = 2d
c) D = 3d
d) D = 4d

6. Two pipes, each of diameter d, converge to form a pipe of diameter D. What should be the relation between d and D such that the velocity in the third pipe becomes half of that in each of the two pipes?

a) D = d/2
b) D = d/3
c) D = d/4
d) D = d/5

FORMS OF ENERGY

Work: It is the force multiplied by the displacement in the direction of the force.

$$W = \int_1^2 F \cdot dx$$

By convention:

-W - indicates that work is done on the system
+W - indicates that work is done by the system.

Heat: It is a form of energy that crosses a system's boundary, because of a temperature difference between the system and the surrounding.

Q - Heat

By convention:

+Q - indicates that heat is added to the system
-Q - indicates that heat is rejected from the system.

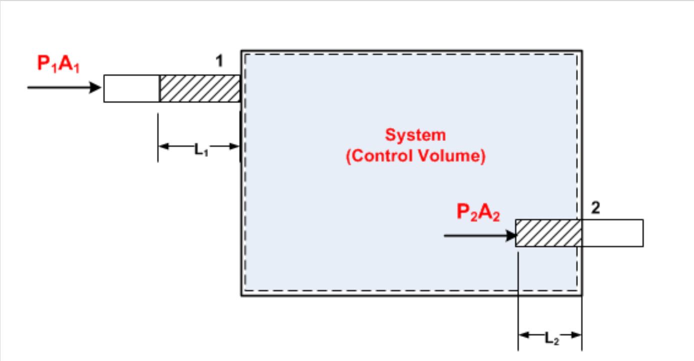
Internal Energy: It is the energy acquired due to the overall molecular interaction, or the total energy that a molecule has.

U = mu KJ

Where

U - total internal energy KJ, KW
u - specific internal energy KJ/kg
ΔU- change of internal energy
m - mass kg, kg/sec

Flow Enegery Or Flow Work: It is the energy required in pushing a fluid usually into the system or out from the system.



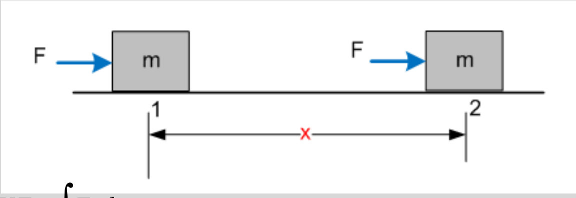
Ef – Flow work or Flow energy

$$E_{f1} = F_1L_1 = P_1A_1L_1 = P_1V_1$$
$$E_{f2} = F_2L_2 = P_2A_2L_2 = P_2V_2$$
$$\Delta Ef = E_{f2} - E_{f1}$$
$$\Delta Ef = P_2V_2 - P_1V_1$$
$$\Delta Ef = \Delta(PV)$$
$$\Delta(PV) = P_2V_2 - P_1V_1 \quad KJ; KW$$
$$\Delta(P\upsilon) = P_2\upsilon_2 - P_1\upsilon_1 \quad \frac{KJ}{kg}$$

$P - KPa$

$$V = m^3; \frac{m^3}{sec}$$
$$\upsilon = \frac{m^3}{kg}$$
$$\upsilon = \frac{1}{\rho}$$

Kinetic Energy: It is the energy acquired due to the motion of a body or a system.



$$\Delta KE = \int F \cdot dx$$

$$F = \frac{ma}{1000}$$

$$a = \frac{dv}{dt}$$

$$\Delta KE = \frac{m}{1000}\int_1^2 \frac{dv}{dt} \bullet dx$$

$$\Delta KE = \frac{m}{1000}\int_1^2 dv \bullet \frac{dx}{dt}$$

$$\frac{dx}{dt} = v$$

$$\Delta KE = \frac{m}{1000}\int_1^2 v \bullet dv$$

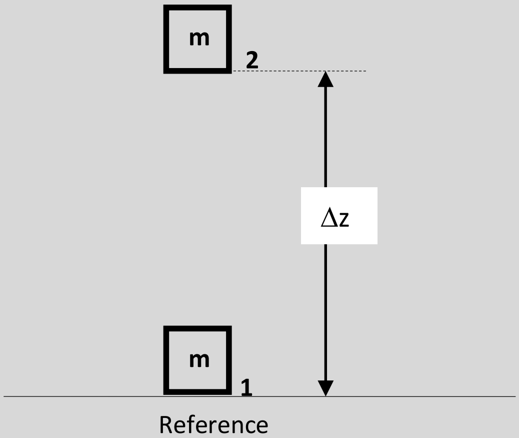
$$\Delta KE = \frac{m}{1000}\left[\frac{v^2}{2}\right]_1^2$$

$$\Delta KE = \frac{m\left(v_2{}^2 - v_1{}^2\right)}{2(1000)} KJ; KW$$

$$\Delta KE = \frac{\left(v_2{}^2 - v_1{}^2\right)}{2(1000)} \; \frac{KJ}{kg}$$

$$m - kg; kg/sec$$

Potential Energy: It is the energy required by virtue of its configuration or elevation.



$$\Delta PE = \int W \bullet dz$$

$$W = \frac{mg}{1000}$$

$$\Delta PE = \frac{mg}{1000}\int_1^2 dz$$

$$\Delta PE = \frac{mg(z_2 - z_1)}{1000} \; KJ; KW$$

$$\Delta PE = \frac{g(z_2 - z_1)}{1000} \; \frac{KJ}{kg}$$

$$m - kg; kg/sec$$

Where:

Z - elevation in m, (+) if above datum and (-) if below datum
g - gravitational acceleration, m/sec^2

Enthalpy (h): It is the sum of the Internal Energy and the Flow Energy.

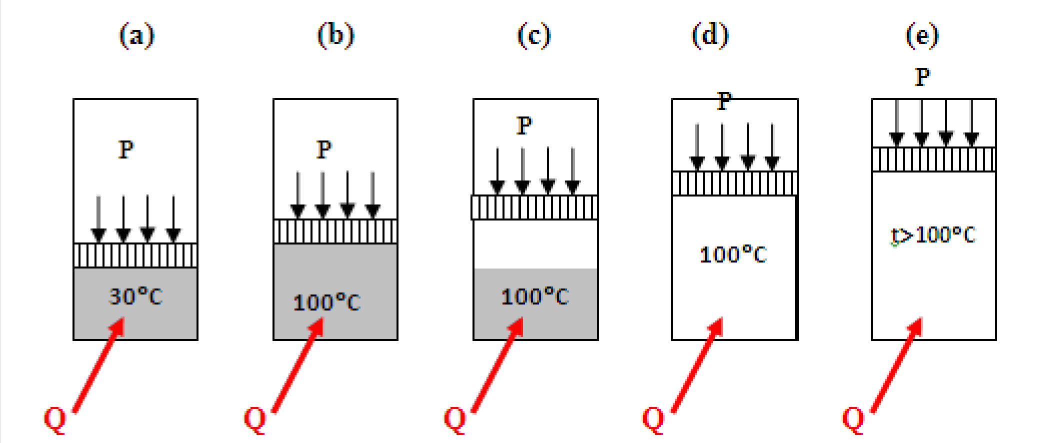
$$h = U + PV \; KJ$$

$$\Delta h = \Delta U + \Delta(PV) \; KJ$$

$$\Delta h = \Delta u + \Delta(P\upsilon) \frac{KJ}{kg}$$

PROPERTIES OF PURE SUBSTANCE

A pure substance is one that is uniform and invariable in chemical composition. A pure substance can exist in more than one phase, but its chemical composition must be the same in each phase. For example, if liquid water and water vapor form a system with two phases, the system can be regarded as a pure substance because each phase has the same composition. A uniform mixture of gases can be regarded as a pure substance provided it remains a gas and does not react chemically.

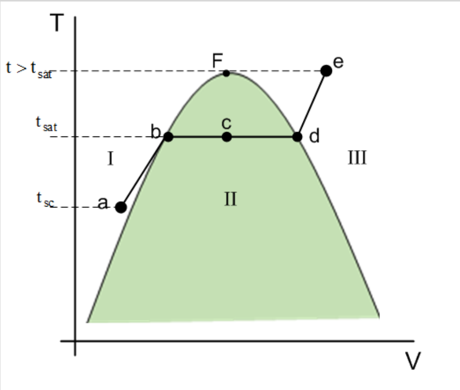


- a - sub-cooled liquid
- b - saturated liquid
- c - saturated mixture
- d - saturated vapor
- e - superheated vapor

Considering that the system is heated at constant pressure where *P = 101.325 KPa*, the *100°C* is the saturation temperature corresponding to 101.325 KPa, and 101.325 KPa pressure is the saturation pressure at 100°C.

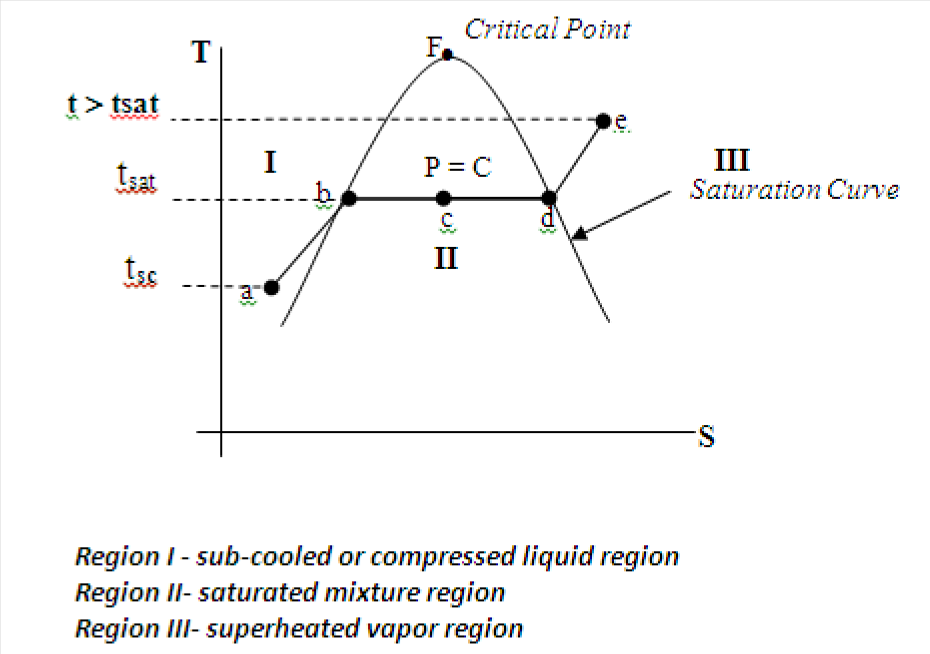
Saturation Temperature (tsat) - is the highest temperature at a given pressure in which vaporization takes place.
Saturation Pressure (Psat) - is the pressure corresponding to the temperature.
Sub-cooled Liquid - is one whose temperature is less than the saturation temperature corresponding to the pressure.
Compressed Liquid - is one whose pressure is greater than the saturation pressure corresponding to the temperature.
Saturated Mixture - a mixture of liquid and vapor at the saturation temperature.
Superheated Vapor - a vapor whose temperature is greater than the saturation temperature.

Temperature - Specific volume Diagram (T-υ diagram)



Region I – Sub-cooled Liquid Compressed Liquid region
Region II -Saturated Mixture region
Region III - Superheated Vapor region
F (critical point)- at the critical point the temperature and pressure is unique.
For Steam: At Critical Point, P = 22.09 MPa; t = 374.136°C

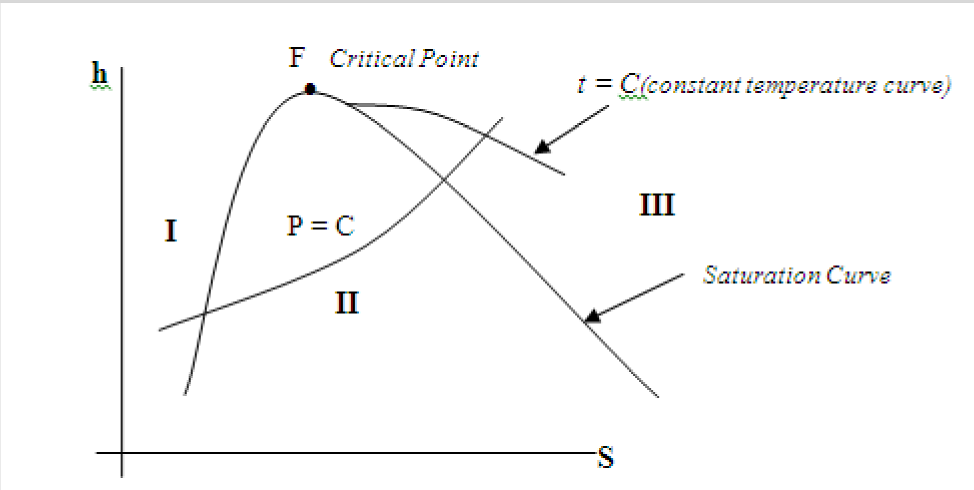
Temperature-Entropy Diagram (T-S Diagram)



Entropy (S): Is that property that determines the randomness and disorder of a substance. If during a process, an amount of heat is taken and is divided by the absolute temperature at which it is taken the result is called the "Change of Entropy".

$$dS = \frac{dQ}{T} \qquad \Delta S = \int \frac{dQ}{T}$$

Enthalpy-Entropy Diagram (h versus S Diagram or Mollier Chart)



The properties h,S,U,and υ at saturated liquid, saturated vapor, sub-cooled or compressed liquid and superheated vapor condition, can be determined using the Steam Table.
For the properties at the saturated mixture condition, its properties is equal to

$$r = r_f + x(r_{fg})$$
$$r_{fg} = r_g - r_f$$

where r stands for any property, such as h, S, U,and υ, where subscript f refers to saturated liquid condition and fg refers to the difference in property between saturated vapor and saturated liquid and x is called the quality.

QUALITY

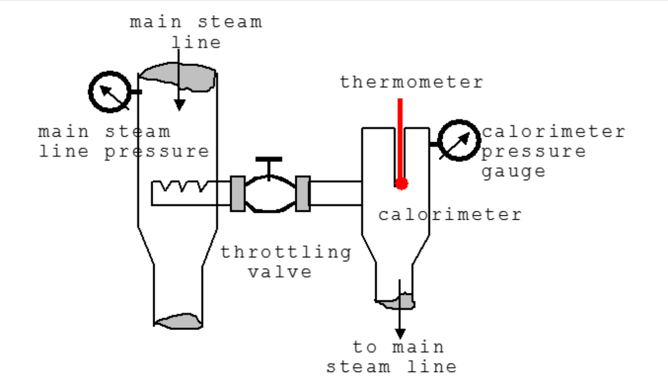
$$x = \frac{m_{vapor}}{m_{vapor} + m_{liquid}} = \frac{m_v}{m_v + m_l}$$

Where:

m - mass
v - refers to vapor
l - refers to liquid

Note: For sub-cooled liquid, its properties are *approximately* equal to the properties at saturated liquid which corresponds to the sub-cooled temperature.

Throttling Calorimeter:
An apparatus that is used to determine the quality of a desuperheated steam flowing in a steam line.



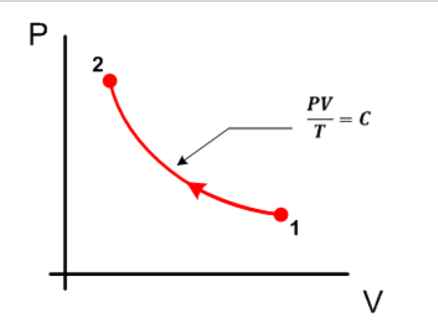
A throttling process is a Steady State, Steady Flow Process in which Q = 0; W = 0; ΔKE = 0; ΔPE = 0 but the Enthalpy (h) remains constant (h = C).
From the figure, steam from the main steam line expands in the calorimeter to the calorimeter pressure and temperature at h = constant.. A throttling calorimeter is an instrument used to determine the quality of steam.

IDEAL OR PERFECT GAS

Ideal Gas: A hypothetical gas that obeys the gas laws perfectly at all temperatures and pressures.

1. CHARACTERISTIC EQUATION

$$PV = mRT$$
$$P\upsilon = RT$$
$$\upsilon = \frac{RT}{P}$$
$$\rho = \frac{P}{RT}$$
$$\frac{PV}{T} = C \quad \text{or} \quad \frac{P\upsilon}{T} = C$$



Where:

P - absolute pressure in KPa
V - volume in m^3
m -mass in kg
R -Gas constant in KJ/kg-°K
T - absolute temperature in °K
υ - specific volume in m^3/kg
ρ - density in kg/m^3

2. GAS CONSTANT

$$R = \frac{\bar{R}}{M} = \frac{8.3143}{M} \frac{KJ}{kg - K}$$

$\bar{R}$ = 8.3143 KJ/kgm-K
$\bar{R}$ - universal gas constant, KJ/kgm-°K
M - molecular weight, kg/kg mol

$$M = \frac{m}{n} \frac{kg}{kg_{mol}}$$

m – mass, kg
n - no. of moles

3. BOYLE`S LAW (T = C) Robert Boyle (1627-1691) :If the temperature of a certain quantity of gas is held constant, the volume V is inversely proportional to the absolute pressure P, during a quasi-static change of state.

$$V \alpha \frac{1}{P} \text{ or } V = C\frac{1}{P}$$

$$PV = C \text{ or}$$

$$P_1 V_1 = P_2 V_2 = C$$

4. CHARLE`S LAW (P = C and V = C) Jacques Charles (1746-1823) and Joseph Louis Gay-Lussac (1778-1850)

At constant pressure (P = C), the volume V of a certain quantity of gas is directly proportional to the absolute temperature T, during a quasi static change of state.

$$V \propto T \text{ or } V = CT$$

$$\frac{V}{T} = C \text{ or } \frac{V_1}{T_1} = \frac{V_2}{T_2}$$

At constant volume (V = C), the pressure P of a certain quantity of gas is directly proportional to the absolute temperature T, during a quasi-static change of state.

$$P \propto T \text{ or } P = CT$$

$$\frac{P}{T} = C$$

or

$$\frac{P_1}{T_1} = \frac{P_2}{T_2}$$

*5. AVOGADRO`S LAW: Amedeo Avogadro (1776-1856) :*All gases at the same temperature and pressure, under the action of a given value of g, have the same number of molecules per unit of volume. From which it follows that the the specific weight is directly proportional to its molecular weight.

At the same temperature and pressure for all gases

$$\frac{n_1}{V_1} = \frac{n_2}{V_2} = \frac{n}{V}$$

or

$$\frac{\gamma_1}{\gamma_2} = \frac{M_1}{M_2} = \frac{R_2}{R_1}$$

6. SPECIFIC HEATS

Specific Heat At Constant Pressure (Cp)

$$Q = mC_p(\Delta T) = m\Delta h$$
$$\Delta h = h_2 - h_1$$
From
$$h = u + P\upsilon$$
$$dh = du + Pd\upsilon + \upsilon dP$$
but
$$dQ = du + Pd\upsilon$$
$$dh = dQ + \upsilon dP$$
At $P = C$ (Constant Pressure)
$$dP = 0$$
$$dh = dQ$$
$$Q = \Delta h$$

considering m
$$Q = m(\Delta h) = \Delta H$$
From $C = \frac{dQ}{dT}$
$$C_p = \frac{dQ}{dT} = \frac{dh}{dT} \rightarrow (C_p \text{ - Specific heat at } P = C)$$
$$dQ = C_p dT$$
$$Q = C_p(\Delta T) = C_p(\Delta t)$$
$$Q = \Delta H = mC_p(\Delta T) = mC_p(\Delta t)$$

Specific Heat At Constant Volume (Cv)

$$Q = mC_V\Delta T = m\Delta u$$
$$\Delta U = m(u_2 - u_1)$$

From
$$C = \frac{dQ}{dT}$$
$$C_v = \frac{dQ}{dT} = \frac{du}{dT} \rightarrow (C_v - \text{specific heat at } V = C)$$
$$dQ = C_v(dT)$$
$$Q = C_v(\Delta T) = C_v(\Delta t)$$
$$Q = \Delta U = mC_v(\Delta T) = mC_v(\Delta t)$$

From
$$dQ = du + Pd\upsilon$$
at $\upsilon = C; d\upsilon = 0$

therefore
$$dQ = du$$
$$Q = \Delta u = (u_2 - u_1)$$
$$Q = \Delta U = m(u_2 - u_1)$$

From
$$h = u + P\upsilon$$
$$P\upsilon = RT$$
$$h = u + RT$$
$$dh = du + Rdt$$
$$C_p dT = C_v dT + Rdt$$
dividing the equation by dT
$$C_p = C_v + R$$

7. RATIO OF SPECIFIC HEATS

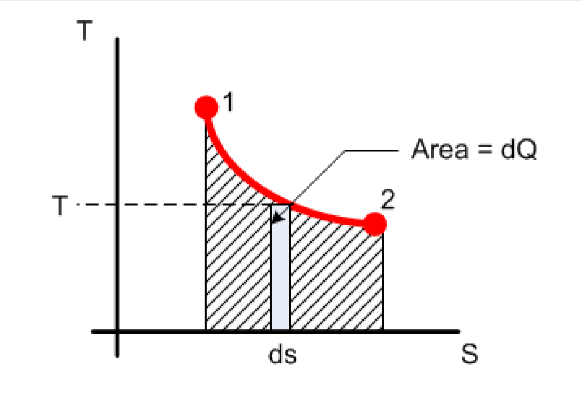
$$k = \frac{C_p}{C_V} = \frac{dh}{du} = \frac{\Delta H}{\Delta U}$$
$$C_p = kC_v$$
$$C_v = \frac{C_p}{k}$$
$$C_p = C_v + R$$
$$kC_v = C_v + R$$
$$C_v = \frac{R}{k-1}$$
$$C_p = C_v + R$$
$$C_p = \frac{C_p}{k} + R$$
$$R = \frac{Rk}{k-1}$$

8. ENTROPY CHANGE (ΔS)

Entropy is that property of a substance that determines the amount of randomness and disorder of a substance. If during a process, an amount of heat is taken and is by divided by the absolute temperature at which it is taken, the result is called the "Entropy Change"

ENTROPY CHANGE.

$$dS = \frac{dQ}{T}$$
$$dQ = T(dS)$$
by integration
$$\Delta S = \int_1^2 \frac{dQ}{T}$$



From:
$$h = U + P\upsilon$$
$$h = U + RT$$
$$dQ = dU + Pd\upsilon$$
$$P = \frac{RT}{\upsilon} \text{ and } dQ = TdS$$
$$TdS = C_v dT + \frac{RT}{\upsilon} d\upsilon$$
dividing by T
$$dS = C_v \frac{dT}{T} + R\frac{d\upsilon}{\upsilon}$$
By integration
$$\Delta S = C_v \ln\frac{T_2}{T_1} + R\ln\frac{\upsilon_2}{\upsilon_1}$$
From
$$dh = dQ + \upsilon dP$$
$$dQ = dh - \frac{RT}{P} dP$$
$$TdS = C_p dT - RT\frac{dP}{P}$$
dividing by T
$$dS = C_p \frac{dT}{T} - R\frac{dP}{P}$$
by integration
$$\Delta S = C_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

Actual – Gas equation of State

In actual gases the molecular collision are inelastic; at high densities in particular there are intermolecular forces that the simplified equation of the state do not account for. There are many gas equations of state that attempt to correct for the non - ideal behavior of gases. The disadvantage of all methods is that the equations are more complex and require the use of experimental coefficients.

a. Van der Waals Equation

$$\left(P + \frac{a}{\bar{\upsilon}^2}\right)\left(\bar{\upsilon} - b\right) = \bar{R}T$$

where:

a and b a coefficients that compensate for the nonideal behavior of the gas.

$$\bar{\upsilon} \text{ in } \frac{m^3}{kg_{mol}}$$

b. Beattie-Bridgeman Equation

$$\left(P+\frac{A}{\bar{\upsilon}^{2}}\right)\frac{\bar{\upsilon}^{2}}{(1-\varepsilon)(\bar{\upsilon}+B)}=\bar{R}T$$

where :

$$A=A_0\left(1-\frac{a}{\bar{\upsilon}}\right)$$

$$B=B_0\left(1-\frac{b}{\bar{\upsilon}}\right)$$

$$\varepsilon=\frac{c}{\bar{\upsilon}T^3}$$

Compressibility Factor

$$\frac{P\upsilon}{RT}=1$$

For nonideal behavior of gases

$$\frac{P\upsilon}{RT}=Z$$

Where: Z – compressibility factor

IDEAL GAS MIXTURE

Mixture of gases are common in many applications. Our most common example is air - mainly consisting of nitrogen, oxygen and water vapor - as moist air. A combustion gas with nitrogen, water vapor and carbon dioxide is an other example.

- *Total Mass Of A Mixture*

$$m=\sum m_i$$

- MASS FRACTION

$$x_i=\frac{m_i}{m}$$

- *Total Moles Of A Mixture*

$$n=\sum n_i$$

- *Mole Fraction*

$$y_i=\frac{n_i}{n}$$

- *Equation Of State*

Mass Basis

For the Mixture

$$PV=mRT$$

Fort the Components

$$P_iV_i=m_iR_iT_i$$

Mole Basis

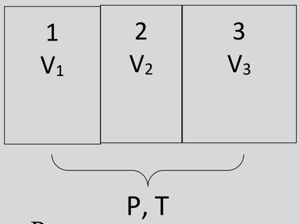
For the Mixture

$$PV=n\bar{R}T$$

Fort the Components

$$P_iV_i=n_i\bar{R}T_i$$

- *Amagat's Law*
  The total volume of a mixture of gases is equal to the sum of the volume occupied by each component at the mixture pressure P and temperature T.



$P = P_1 = P_2 = P_3$
$T = T_1 = T_2 = T_3$

$$n = n_1 + n_2 + n_3$$

$$n = \frac{PV}{\bar{R}T} \ ; \ n_1 = \frac{PV_1}{\bar{R}T} \ ; \ n_2 = \frac{PV_2}{\bar{R}T} \ ; \ n_3 = \frac{PV_3}{\bar{R}T}$$

$$\frac{PV}{\bar{R}T} = \frac{PV_1}{\bar{R}T} + \frac{PV_2}{\bar{R}T} + \frac{PV_3}{\bar{R}T}$$

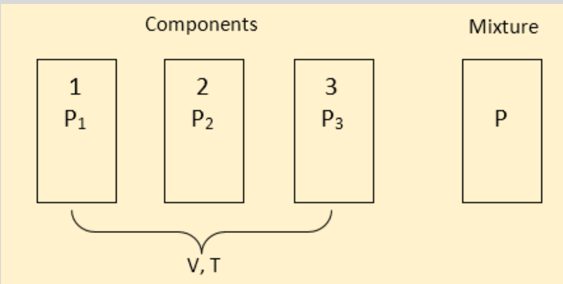
$$\left[\frac{PV}{\bar{R}T} = \frac{PV_1}{\bar{R}T} + \frac{PV_2}{\bar{R}T} + \frac{PV_3}{\bar{R}T}\right]\left(\frac{\bar{R}T}{P}\right)$$

$$V = V_1 + V_2 + V_3$$

$$V = \sum V_i$$

$$y_i = \frac{n_i}{n} = \frac{V_i}{V}$$

- *Dalton's LAW*
  The total pressure of a mixture of gases P is equal to the sum of the partial pressure that each gas would exert at the mixture volume V and temperature T.



$$V = V_1 = V_2 = V_3$$

$$T = T_1 = T_2 = T_3$$

$$n = n_1 + n_2 + n_3$$

$$n = \frac{PV}{\bar{R}T} \ ; \ n_1 = \frac{P_1V}{\bar{R}T} \ ; \ n_2 = \frac{P_2V}{\bar{R}T} \ ; \ n_3 = \frac{P_3V}{\bar{R}T}$$

$$\frac{PV}{\bar{R}T} = \frac{P_1V}{\bar{R}T} + \frac{P_2V}{\bar{R}T} + \frac{P_3V}{\bar{R}T}$$

$$\left[\frac{PV}{\bar{R}T} = \frac{P_1V}{\bar{R}T} + \frac{P_2V}{\bar{R}T} + \frac{P_3V}{\bar{R}T}\right]\left(\frac{\bar{R}T}{V}\right)$$

$$P = P_1 + P_2 + P_3$$

$$P = \sum P_i$$

$$y_i = \frac{n_i}{n} = \frac{P_i}{P}$$

- *Molecular Weight Of A Mixture (M)*

$$M = \sum y_i M_i$$

$$M = \frac{\bar{R}}{R} = \frac{8.3143}{R} \quad \frac{KJ}{kg - K}$$

- *Gas Constant (R)*

$$R = \sum x_i R_i$$

$$R = \frac{\bar{R}}{M} = \frac{8.3143}{M} \quad \frac{KJ}{kg - K}$$

- *Specific Heat Of A Mixture*

At Constant Volume

$$C_v = \sum x_i C_{vi}$$

At Constant Pressure

$$C_P = \sum x_i C_{Pi}$$

$$C_p = C_v + R$$

- *Ratio Of Specific Heat*

$$k = \frac{C_P}{C_V}$$

$$C_P = \frac{Rk}{k-1}$$

$$C_V = \frac{R}{k-1}$$

- *Gravimetric And Volumetric Analysis*

Gravimetric analysis gives the mass fractions of the components in the mixture.
Volumetric analysis gives the volumetric or molal fractions of the components in the mixture.

CONVERSION

$$x_i = \frac{y_i M_i}{\sum y_i M_i} = \frac{y_i M_i}{M}$$

$$yi = \frac{\frac{x_i}{M_i}}{\sum \frac{x_i}{M_i}}$$

SAMPLE PROBLEMS

1. Determine the molecular weight of a gas if its specific heats at constant pressure and volume are Cp = 2.286 kJ/kg K and Cv = 1.768 kJ/kg K.

$$Cp = Cv + R$$
$$R = Cp - Cv$$
$$R = \frac{8.3143}{M}$$
$$\frac{8.3143}{M} = Cp - Cv$$
$$M = \frac{8.3143}{Cp - Cv} = 16.05\,\frac{kg}{kgm}$$

2. A perfect gas at pressure of 750 kPa and 600 K is expanded to 2 bar pressure. Determine final temperature of gas if initial and final volume of gas are 0.2 m^3 and 0.5 m^3 respectively.

$$P1 = 750\,KPa$$
$$T1 = 600\,K$$
$$V1 = 0.2\,m^3$$
$$P_2 = 2\,Bar = 200\,KPa$$
$$V_2 = 0.5\,m^3$$
$$T_2 =$$
$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$
$$T_2 = 400\,K$$

3. A vessel of 5 m^3 capacity contains air at 100 kPa and temperature of 300K. Some air is removed from vessel so as to reduce pressure and temperature to 50 kPa and 7ºC respectively. Find the amount of air removed and volume of this mass of air at initial states of air. Take R = 287 J/kg.K for air.(Answer: 2.696 kg; 2.32 m^3)

4. A cylindrical vessel of 1 m diameter and 4 m length has hydrogen gas at pressure of 100 kPa and 27ºC. Determine the amount of heat to be supplied so as to increase gas pressure to 125 kPa. For hydrogen take Cp = 14.307 kJ/kg.K, Cv = 10.183 kJ/kg K. (Answer: Q = 194 KJ)

5. A spherical balloon of 5 m diameter is filled with Hydrogen at 27ºC and atmospheric pressure of 1.013 bar. It is supposed to lift some load if the surrounding air is at 17ºC. Estimate the maximum load that can be lifted. (Answer: m = 74.344 kg)

6. A pump draws air from large air vessel of 20 m^3 at the rate of 0.25 m^3 /min. If air is initially at atmospheric pressure and temperature inside receiver remains constant then determine time required to reduce the receiver pressure to 1/4 of its original value. (t = 110.9 minutes)

in this case T is constant and V is constant, and mass and pressure changes with time t

$$PV = mRT$$

$$V\left(\frac{dP}{dt}\right) = RT\left(\frac{dm}{dt}\right)$$

$\frac{dm}{dt}$ – is the mass change wrt time t

$\frac{dm}{dt} = -\frac{Pv}{RT}$ (- sign is used because mass decreases with time

$$V\left(\frac{dP}{dt}\right) = RT\left(-\frac{Pv}{RT}\right)$$

$$V\left(\frac{dP}{dt}\right) = -Pv$$

$$\frac{dt(v)}{V} = -\frac{dP}{P}$$

$$dt = -\frac{V}{v}\left(\frac{dP}{P}\right)$$

by integratio n

$$t = -\frac{V}{v}\left(\ln\frac{P_2}{P_1}\right)$$

$$P_2 = \frac{P_1}{4}$$

$$\frac{P_2}{P_1} = \frac{1}{4}$$

$$t = -\frac{20}{0.25}\left(\ln\frac{1}{4}\right) = 110.9 \text{ minutes}$$

7. A vessel of volume 0.2 m^3 contains nitrogen at 101.3 KPa and 15ºC. If 0.2 kg of nitrogen is now pumped into the vessel, calculate the new pressure when the vessel has returned to its initial temperature. For nitrogen: M = 28 and k = 1.399. (187 KPa)

8. A certain perfect gas of mass 0.01 kg occupies a volume of 0.003 m^3 at a pressure of 700 KPa and a temperature of 131ºC. The gas is allowed to expand until the pressure is 100 KPa and the final volume is 0.02 m^3. Calculate:
   a) the molecular weight of the gas (16)
   b) the final temperature (111.5ºC)

9. A perfect gas has a molecular weight of 26 kg/kgmol and a value of k = 1.26. Calculate the heat rejected
   a) when 1 kg of the gas in contained in a rigid vessel at 300 KPa and 315ºC, and is then cooled until the pressure falls to 150 KPa. (-361 KJ)

   b) when 1 kg/sec mass flow rate of the gas enter a pipeline at 280ºC and flows steadily to the end of the pipe where the temperature is 20ºC. Neglect changes in kinetic and otential energies.(-403 KW)

10. Calculate the internal energy and enthalpy of 1 kg of air occupying 0.05 m^3 2000 KPa. If the internal energy is increased by 120 KJ as the air is compressed to 5000 KPa, calculate the new volume occupied by 1 kg of the air.
    For air: R = 0.287 KJ/kg-ºK and k = 1.4.(250.1 KJ/kg; 350.1 KJ/kg; 0.0296 m^3)

11. When a certain perfect gas is heated at constant pressure from 15ºC to 95ºC, the heat required is 1136 KJ/kg. When the same gas is heated at constant volume between the same temperatures the heat required is 808 KJ/kg. Calculate Cp, Cv, k, and M of the gas. (14.2 KJ/kg; 10.1 KJ/kg; 1.405; 4.1 and 2.208)

12. A quantity of a certain perfect gas is compressed from an initial state of 0.085 m^3, 100 KPa to a final state of 0.034 m^3, 390 KPa. the Cv = 0.724 KJ/kg-ºC and Cp = 1.020 KJ/kg-ºC. The observed temperature rise is 146ºK. Calculate R, the mass present, and ΔU of the gas.(0.296 KJ/kg-K; 0.11 kg; 11.63 KJ)

13. A mass of 0.05 kg of air is heated at constant pressure of 200 KPa until the volume occupied is 0.0658 m^3. Calculate the heat supplied, the work and the change in entropy for the process if the initial temperature is 130ºC. (Q = 25.83 KJ; W = 7.38 KJ)

14. A 1 kg of nitrogen is compressed reversibly and isothermally from 101 KPa, 20ºC to 420 KPa. Calculate the nonflow work and the heat flow during the process assuming nitrogen to be a perfect gas. (Q = W = 124 kJ/KG)

15. Air at 102 KPa, 22ºC, initially occupying a cylinder volume of 0.015 m^3 is compressed isentropically by a piston to a pressure of 680 KPa. Calculate the final temperature, the final volume, the work done on the mass of air in the cylinder.
(234.3 ºC; .00387 m^3; 2.76 KJ)
16. 1 kg of air is compressed from 110 KPa, 27 ºC in a polytropic process where n = 1.3 until the final pressure is 660 KPa.
Calculate:
    a) $\int PdV$
    b) - $\int VdP$
    c) ΔS

17. There are 1.36 kg of air at 138 KPa stirred with internal paddles in an insulated rigid container, whose volume is 0.142 m^3 until the pressure becomes 689.5 KPa. Determine the work input and ΔPV. (196.2 KJ; 78.3 KJ)

18. During an isentropic process of 1.36 kg/sec of air, the temperature increases from 4.44ºC to 115.6 ºC. for a nonflow process and for a steady flow process ($\Delta KE = 0$ and $\Delta PE = 0$) Find:
    a) U in KW
    b) H in KW
    c) W in KW
    d) ΔS in KW/ºK
    e) Q in KW

19. A certain perfect gas is compressed reversibly from 100 KPa, 17 ºC to a pressure of 500 KPa in a perfectly thermally insulated cylinder, the final temperature being 77 ºC. The work done on the gas during the compression is 45 KJ/kg. Calculate, k , Cv, R and M of the gas.(1.132; 0.75 KJ/kg-ºK; 0.099 KJ/kg-ºK; 84)

20. 1 kg of air at 102 KPa, 20 ºC is compressed reversibly according to a law $PV^{1.3} = C$ to a pressure of 550 KPa. Calculate the work done on the air and the heat supplied during the compression. (133.46 KJ/kg; -33.3 KJ/kg)

21. Oxygen (M = 32) is compressed polytropically in a cylinder from 105 KPa, 15ºC to 420 KPa in such a way that one third of the work input is rejected as heat to the cylinder walls. Calculate the final temperature of the oxygen. Assume oxygen to be perfect gas and take Cv = 0.649 KJ/kg-K. (113 ºC)

22. Air at 690 KPa, 260ºC is throttled to 550 KPa before expanding through the nozzle to a pressure of 110 KPa. Assuming that the air flows reversibly in steady flow through the nozzle and that no heat is rejected, calculate the velocity of the air at exit from the nozzle when the inlet velocity is 100 m/sec. (636 m/sec)

23. Air at 40ºC enters a mixing chamber at a rate of 225 kg/sec where it mixes with air at 15ºC entering at a rate of 540 kg/sec. Calculate The temperature of the air leaving the chamber, assuming steady flow conditions. Assume that the heat loss is negligible. (22.4ºC)

24. A gaseous mixture composed of 25 kg of N_2, 3.6 kg of H_2, and 60 kg of CO_2 is at 200 KPa, 50°C. Find the respective partial pressures and compute the volume of each component at its own partial pressure and 50°C.
    Given: m_{N2} = 25 kg ; m_{H2} = 3.6 kg ; m_{CO2} = 60 kg
    m = 25 + 3.6 + 60 = 88.6 kg
    P = 200 KPa ; T = 323 K

25. Assume 2 kg of O_2 are mixed with 3 kg of an unknown gas. The resulting mixture occupies a volume of 1.2 m^3 at 276 KPa and 65°C. Determine
    a) R and M of the unknown gas constituent
    b) the volumetric analysis
    c) the partial pressures
    Given; m_{O2} = 2 kg; m_x = 3 kg
    V = 1.2 m^3 ; P = 276 KPa; T = 338 K
    a)
    m = 5 kg
    x_{O2} = 0.40 ; x_x = 0.60
    $PV = mRT$
    R = 0.1361 KJ/kg-K
    R = .40(0.26) + 0.60(R_x)
    R_x = 0.535 KJ/kg-K
    M_x = 15.54 kg/kg_m
    b) $\sum \frac{x_i}{M_i} = \frac{.40}{32} + \frac{.60}{15.54} = 0.0511$
    y_{O2} = 0.245 ; y_x = 0.755
    c) P_{O2} = .245(276) = 67.62 KPa ;

$P_x = 0.755(276) = 208.38$ KPa

26. Assume 28 m^3 of a gaseous mixture whose gravimetric analysis is 20% CO_2, 15% O_2, 65% N_2, are at 103.4 KPa and 150°C. Find
    a) the volumetric analysis
    b) the respective partial pressures
    c) R and M
    d) the moles of mixture and of each constituent
    e) the heat transferred with no change in pressure to reduce the temperature to 75°C
    f) the volume the mixture occupies after the cooling

27. Consider 2 kg of CO and 1 kg of CH4 at 32°C that are in a 0.6 m3 rigid drum. Find:
    a) the mixture pressure P in KPa
    the volumetric analysis
    c) the partial pressures in KPa
    d) the heat to cause a temperature rise of 50°C.

28. A gaseous mixture has the following volumetric analysisO_2, 30%; CO_2, 40% N_2, 30%. Determine
    a) the analysis on a mass basis
    b) the partial pressure of each component if the total pressure is 100 KPa and the temperature is 32°C
    c) the molecular weight and gas constant of the mixture

29. A gaseous mixture has the following analysis on a mass basis, CO_2, 30%; SO_2, 30%; He, 20% and N_2, 20%. For a total pressure and temperature of 101 KPa and 300 K, Determine
    a) the volumetric or molal analysis
    b) the component partial pressure
    c) the mixture gas constant
    d) the mixture specific heats

30. A cubical tank 1 m on a side, contains a mixture of 1.8 kg of nitrogen and 2.8 kg of an unknown gas. The mixture pressure and temperature are 290 KPa and 340 K. Determine
    a) Molecular weight and gas constant of the unknown gas
    b) the volumetric analysis

31. A mixture of ideal gases at 30°C and 200 KPa is composed of 0.20 kg CO_2, 0.75 kg N_2, and 0.05 kg He. Determine the mixture volume.10. An ideal gas with R = 2.077 KJ/kg-K and a constant k= 1.659 undergoes a constant pressure process during which 527.5 KJ are added to 2.27 kg of the gas. The initial temperature is 38°C. Find the ΔS in KJ/K.
    a) 2.687 b) 1.586 c) 8.64 d) 3.861

32.Calculate the volume occupied by 13.6 kg of chlorine at a pressure of 73 mm Hg and a temperature of 21°C. molecular weight of chlorine may be taken as 71. (4.7 m^3)

33.A 3.8 m^3 tank filled with acetylene at 103.4 KPa is supercharged with 1.95 kg of the gas. What would be the final pressure, assuming no change in temperature (16°C) that takes place during the charging period. (155 KPa)

34. A volume of gas having initial entropy of 5317.2 KJ/K is heated at constant temperature of 540°C until the entropy is 8165.7 KJ/K. How much heat is added and how much work is done during the process.

35. An ideal gas with R = 2.077 KJ/kg-K and a constant k= 1.659 undergoes a constant pressure process during which 527.5 KJ are added to 2.27 kg of the gas. The initial temperature is 38°C. Find the ΔS in KJ/K.
    a) 2.687 b) 1.586 c) 8.64 d) 3.861

36. Calculate the volume occupied by 13.6 kg of chlorine at a pressure of 73 mm Hg and a temperature of 21°C. molecular weight of chlorine may be taken as 71. (4.7 m^3)

37. A volume of gas having initial entropy of 5317.2 KJ/K is heated at constant temperature of 540°C until the entropy is 8165.7 KJ/K. How much heat is added and how much work is done during the process.

THE FIRST LAW OF THERMODYNAMICS
(The Law of Conservation of Energy)

"Energy can neither be created nor destroyed but can only be converted from one form to another."

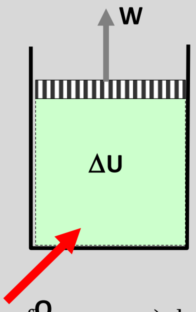
The Verbal form of the Law is:

Energy Entering - Energy Leaving = Change of Energy Stored within the system

In equation Form:

$$E_1 - E_2 = \Delta Es$$

Corollary Laws of the First Law:
First Corollary: Is the application of the Law of Conservation of Energy principle to a Closed System. A system of Fixed mass.



For a Closed system (Nonflow system) the ΔKE and ΔPE are negligible and Δ(pv) doesn't exists.

Q - W = ΔEs

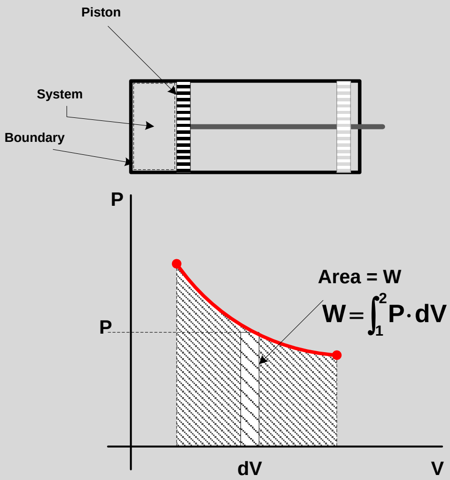
where
ΔEs = ΔU

$$Q - W = \Delta U \rightarrow 1$$
$$Q = \Delta U + W \rightarrow 2$$

By differentiation:

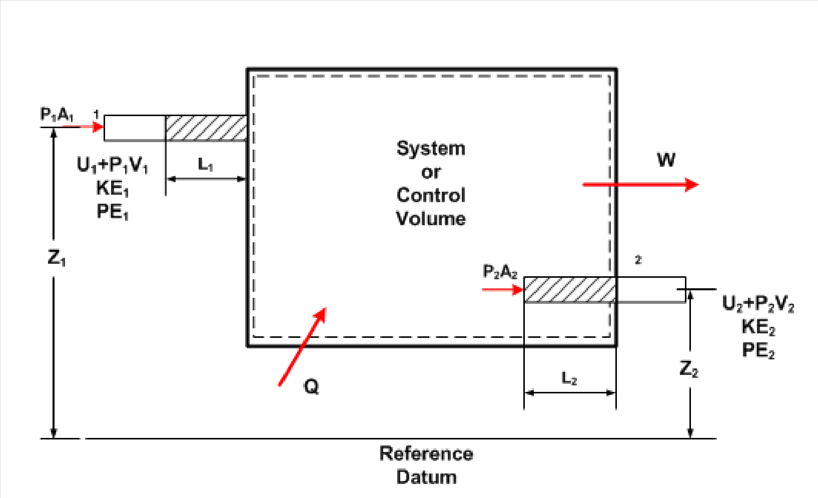
$$dQ = dU + dW \rightarrow 3$$

Work for a Moving Boundary of a Closed System



$$W = \int_1^2 P \cdot dV \rightarrow 4$$
$$dW = P \cdot dV \rightarrow 5$$
$$dQ = dU + dW$$
$$dQ = dU + dW$$
$$dQ = dU + P \cdot dV \rightarrow 6$$

Second Corollary: Is the application of the Law of Conservation of Energy principle to an Open System (Steady-state, Steady Flow System) a system of Fixed space.



For an OPEN SYSTEM (Steady-State, Steady-Flow system)

$\Delta ES = 0$, therefore

Energy Entering = Energy Leaving,
In equation form

$$E_1 - E_2 = 0$$
$$E_1 = E_2$$

$$U_1 + P_1V_1 + KE_1 + PE_1 + Q = U_2 + P_2V_2 + KE_2 + PE_2 + W \rightarrow 1$$
$$Q = U_2 - U_1 + P_2V_2 - U_1 + KE_2 - KE_1 + PE_2 - PE_1 + W \rightarrow 2$$
$$Q = \Delta U + \Delta(PV) + \Delta KE + \Delta PE + W \rightarrow 3$$

Enthalpy (h):It is the sum of internal energy and flow energy.

$$h = U + PV \rightarrow 5$$
$$\Delta h = \Delta U + \Delta(PV) \rightarrow 6$$
$$Q = \Delta h + \Delta KE + \Delta PE + W \rightarrow 7$$
$$W = Q - \Delta h - \Delta KE - \Delta PE \rightarrow 8$$

By differenti ation

$$dh = dU + PdV + VdP \rightarrow 9$$

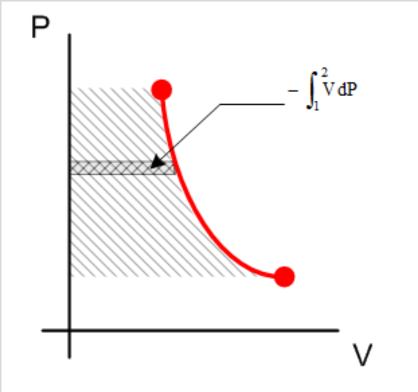
but

$$dU + PdV = dQ$$
$$dh = dQ + VdP$$
$$dW = dQ - dh - dKE - dPE \rightarrow 10$$
$$dW = dQ - dQ - VdP - dKE - dPE$$
$$dW = -VdP - dKE - dPE \rightarrow 11$$
$$W = -\int VdP - dKE - dPE \rightarrow 12$$
$$-\int VdP = dQ - dh \rightarrow 13$$

If ΔKE = 0 and ΔPE = 0 or negligible;

$$W = -\int VdP \rightarrow 14$$

$$W = Q - \Delta h \rightarrow 15$$



ZEROTH LAW OF THERMODYNAMICS

If two bodies are in thermal equilibrium with a third body, they are in thermal equilibrium with each other, and hence their temperatures are equal.

SPECIFIC HEAT: It is the amount of heat required to raise the temperature of a 1 kg mass, 1° K or 1° C.

$$C = \frac{dQ}{dt} = \frac{dQ}{dT} \; \frac{KJ}{kg\text{-}°C} \text{ or } \frac{KJ}{kg\text{-}K}$$

then dQ = Cdt
by integration, where C is constant

$$Q = C(\Delta t) = C(\Delta T)$$
Considerin g m (mass)
$$Q = mC(\Delta t) = mC(\Delta T)$$
where :
$$\Delta t = \Delta T$$
t - temperature in °C
T - absolute temperature in K

Sensible Heat: The amount of heat per unit mass that must be transferred (added or remove) when a substance undergoes a change in temperature without a change in phase.

$$Q = mC(\Delta t) = mC(\Delta T)$$

Where:

m - mass , kg
C - heat capacity or specific heat, KJ/kg-°C or KJ/kg-K
t - temperature in °C
T - temperature in K

HEAT OF TRANSFORMATION (LATENT HEAT): The amount of heat per unit mass that must be transferred when a substance completely undergoes a phase change without a change in temperature.

$$Q = mh$$

A. Heat of Vaporization: Amount of heat that must be added to vaporize a liquid or that must be removed to condense a gas.

$$Q = mh_v$$

Where

h_v- latent heat of vaporization, KJ/kg

B. Heat of Fusion : Amount of heat that must be added to melt a solid or that must be removed to freeze a liquid.

$$Q = mh_F$$

Where

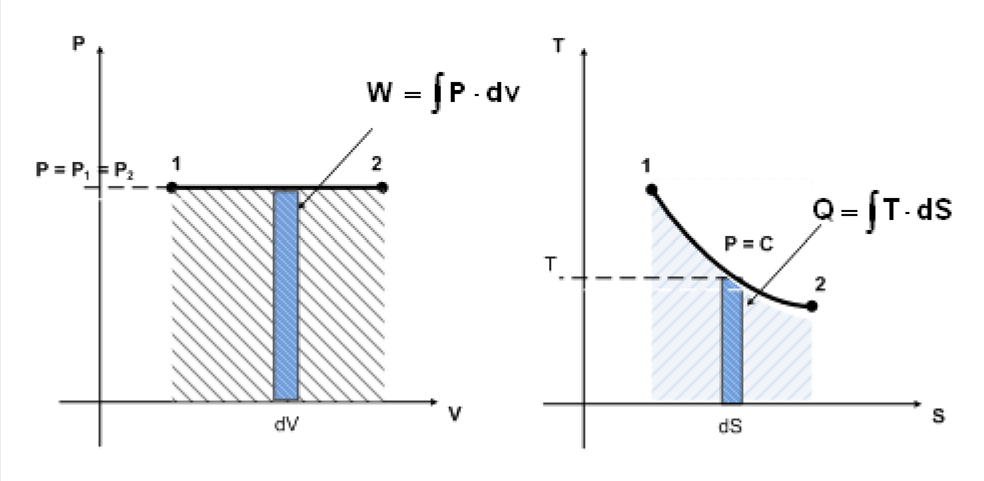
h_F – latent heat of fusion, KJ/kg

Water Equivalent: The water equivalent of a substance is the mass of water that would require the same heat transfer as the mass of that substance to cause the same change of temperature.

$$m_w = \frac{m_s C_{ps}}{C_{pw}}$$

PROCESSES OF FLUIDS

ISOBARIC PROCESS (P = C): An Isobaric Process is an internally reversible Constant Pressure process.



CLOSED SYSTEM

For any substance

$Q = \Delta U + W \rightarrow 1$

$W = \int P \bullet dV$

At $P = C$

$W = P(V_2 - V_1) \rightarrow 2$

$\Delta U = m(U_2 - U_1) \rightarrow 3$

from

$h = U + PV$

$dh = dU + PdV + VdP$

$dP = 0$ at $P = C$

$dU + PdV = dQ$

$dQ = dh$

$Q = \Delta h$

$Q = m(h_2 - h_1) \rightarrow 4$

For Ideal Gas

$\frac{V_1}{T_1} = \frac{V_2}{T_2} \rightarrow 1$

$W = P(V_2 - V_1) = mR(T_2 - T_1) \rightarrow 5$

$\Delta U = mC_v(T_2 - T_1) \rightarrow 3$

$Q = \Delta h = mC_p(T_2 - T_1) \rightarrow 6$

ENTROPY CHANGE

For any substance

$\Delta S = \int \frac{dQ}{T} = \int \frac{dh}{T} = S_2 - S_1 \rightarrow 7$

For Ideal Gas

$dQ = dh = mC_p dT$

$\Delta S = \int \frac{dQ}{T} = \int \frac{dh}{T} = mC_p \int_1^2 \frac{dT}{T}$

$\Delta S = mC_p \ln \frac{T_2}{T_1} \rightarrow 8$

OPEN SYSTEM

$Q = \Delta h + \Delta KE + \Delta PE + W \rightarrow 9$

$W = -\int VdP - \Delta KE - \Delta PE \rightarrow 10$

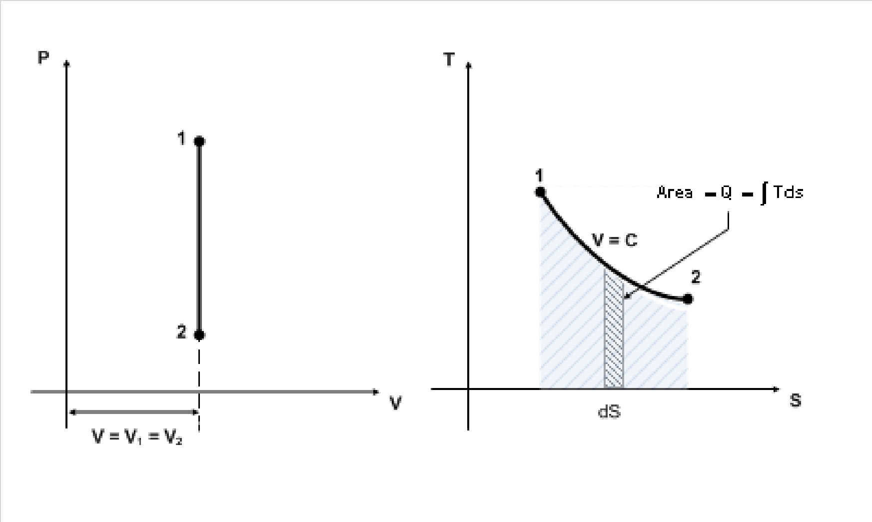
$dP = 0$ at $P = C$ and $Q = \Delta h; -\int V \cdot dP = 0$

$W = -\Delta KE - \Delta PE \rightarrow 11$

If $\Delta KE = 0$ and $\Delta PE = 0$

$W = 0 \rightarrow 12$

ISOMETRIC PROCESS (V = C): An Isometric Process is an internally reversible "Constant Volume" process.



CLOSED SYSTEM

For any substance

$Q = \Delta U + W \rightarrow 1$

$W = \int P \bullet dV$

At $V = C$

$dV = 0$

$W = 0 \rightarrow 2$

$dQ = dU + PdV$

$dV = 0$

$dQ = dU$

$Q = \Delta U$

$Q = m(U_2 - U_1) \rightarrow 3$

For Ideal Gas

$\frac{P_1}{T_1} = \frac{P_2}{T_2} \rightarrow 4$

$Q = \Delta U = mCv(T_2 - T_1) \rightarrow 5$

ENTROPY CHANGE

$\Delta S = \int \frac{dQ}{T} = mC_v \int \frac{dT}{T}$

$\Delta S = mC_v \ln \frac{T_2}{T_1} \rightarrow 6$

OPEN SYSTEM

$Q = \Delta h + \Delta KE + \Delta PE + W \rightarrow 7$

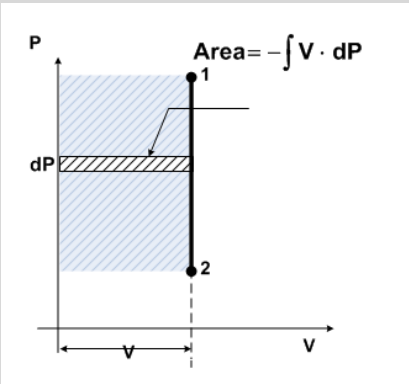
$W = -\int VdP - \Delta KE - \Delta PE \rightarrow 8$

$-\int V \cdot dP = V(P_1 - P_2) \rightarrow 9$

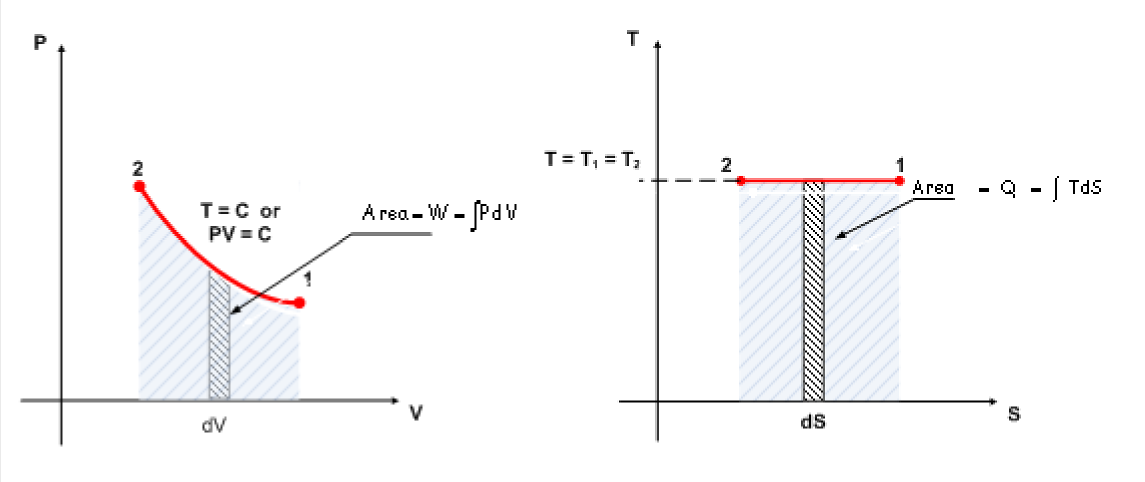
$W = -\Delta KE - \Delta PE \rightarrow 11$

If $\Delta KE = 0$ and $\Delta PE = 0$

$W = 0 \rightarrow 12$



ISOTHERMAL PROCESS (T = C or PV = C): An Isothermal Process is an internally reversible "Constant Temperature" Process



CLOSED SYSTEM

For any substance

$Q = \Delta U + W \rightarrow 1$

$W = \int P \cdot dV \rightarrow 2$

$\Delta U = m(U_2 - U_1) \rightarrow 3$

For Ideal Gas

$PV = C \quad \text{or} \quad P = \dfrac{C}{V}$

$P_1V_1 = P_2V_2 = C$

$\Delta U = mC_v(T_2 - T_1)$

But $T_1 = T_2$

$\Delta U = 0 \rightarrow 4$

$W = \int PdV = C\int \dfrac{dV}{V}$

$W = P_1V_1 \ln \dfrac{V_2}{V_1} \rightarrow 5$

$W = mRT_1 \ln \dfrac{V_2}{V_1} \rightarrow 6$

$\dfrac{V_2}{V_1} = \dfrac{P_1}{P_2}$

$W = mRT_1 \ln \dfrac{P_1}{P_2} \rightarrow 7$

ENTROPY CHANGE

For any substance

$\Delta S = S_2 - S_1 \rightarrow 8$

From

$dQ = Tds$

At $T = C$

$Q = T\Delta S$

$\Delta S = \dfrac{Q}{T} \rightarrow 9$

For ideal gas

$\Delta U = 0$, therefore

$W = Q \rightarrow 10$

OPEN SYSTEM

$Q = \Delta h + \Delta KE + \Delta PE + W \rightarrow 1$

$W = -\int VdP - \Delta KE - \Delta PE \rightarrow 2$

$\Delta h = m(h_2 - h_1) \rightarrow 3$

For ideal gas

$\Delta h = mC_p(T_2 - T_1)$

but $T_1 = T_2$

$\Delta h = 0 \rightarrow 4$

From

$PV = C \quad \text{or} \quad V = \dfrac{C}{P}$

$P_1V_1 = P_2V_2 = C$

$-\int VdP = -C\int \dfrac{dP}{P}$

$-\int VdP = -P_1V_1 \ln \dfrac{P_2}{P_1} \rightarrow 5$

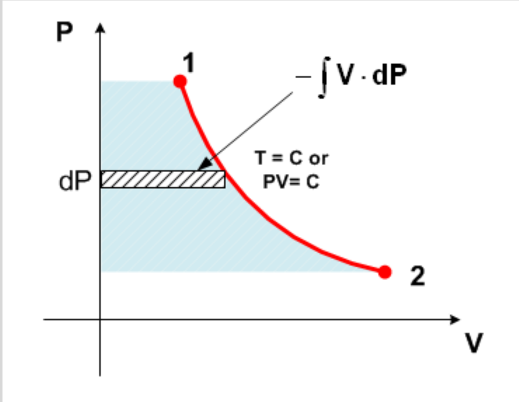
and applying laws of logarithm

$-\int VdP = P_1V_1 \ln \dfrac{P_1}{P_2} = mRT_1 \ln \dfrac{P_1}{P_2} \rightarrow 6$

$-\int VdP = mRT_1 \ln \dfrac{V_2}{V_1} \rightarrow 7$

If $\Delta KE = 0$ and $\Delta PE = 0$

$W = -\int VdP = -P_1V_1 \ln \dfrac{P_2}{P_1} \rightarrow 8$



ISENTROPIC PROCESS (S = C): An Isentropic Process is an internally "reversible adiabatic" process in which the entropy remains constant where S = C (for any substance) or PVk = C (for an ideal or perfect gas)

From

$$dQ = dU + PdV$$
$$dQ = 0, \text{ for adiabatic}$$
$$dU = -PdV \rightarrow 1$$
$$dh = dQ + VdP$$
$$dQ = 0$$
$$dh = VdP \rightarrow 2$$
$$\frac{dh}{dU} = \frac{VdP}{-PdV} \rightarrow 3$$

but

$$\frac{C_p}{C_v} = \frac{dh}{dU} = k, \text{ hence}$$

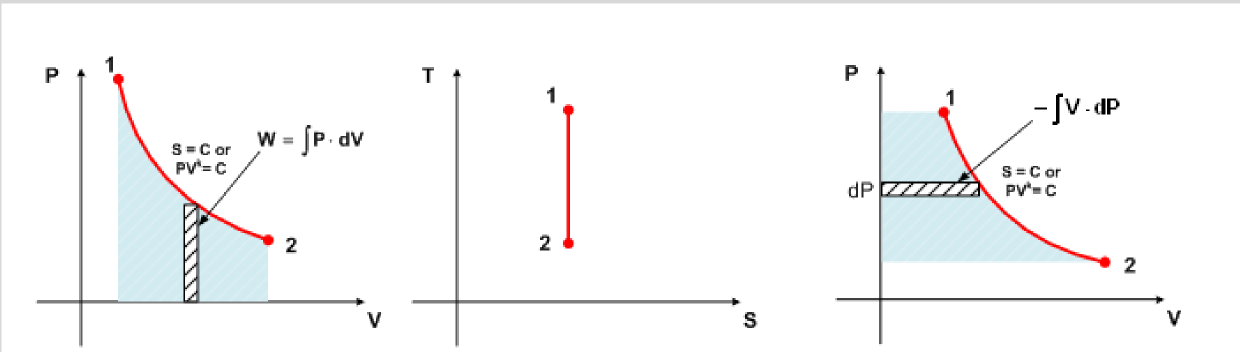
$$k = \frac{-VdP}{PdV}$$
$$\frac{dP}{P} = -k\frac{dV}{V}$$

by integration

$$\int_1^2 \frac{dP}{P} = -k\int_1^2 \frac{dV}{V}$$
$$\ln\frac{P_2}{P_1} = -k\ln\frac{V_2}{V_1} = k\ln\frac{V_1}{V_2} = \ln\left(\frac{V_1}{V_2}\right)^k$$

taking antilog

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^k = \frac{V_1{}^k}{V_2{}^k} \quad \text{or } P_1V_1{}^k = P_2V_2{}^k \rightarrow 1$$



Using

$$PV^k = C \text{ and } \frac{PV}{T} = C$$
$$P_1V_1{}^k = P_2V_2{}^k \text{ and } \frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$
$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = \left(\frac{V_1}{V_2}\right)^{k-1} \rightarrow 2$$

From $PV^k = C$

$$P = \frac{C}{V^k} \text{ and } V = \frac{C^{\frac{1}{k}}}{P^{\frac{1}{k}}} \rightarrow 3$$

By integration

$$\int_1^2 PdV = \frac{P_2V_2 - P_1V_1}{1-k} \rightarrow 4$$
$$-\int_1^2 VdP = \frac{k(P_2V_2 - P_1V_1)}{1-k} \rightarrow 5$$
$$-\int_1^2 VdP = k\int_1^2 PdV \rightarrow 6$$
$$P_1V_1 = mRT_1 \rightarrow 7$$
$$P_2V_2 = mRT_2 \rightarrow 8$$

CLOSED SYSTEM

For any substance

$Q = \Delta U + W$

$Q = 0 \rightarrow 1$

$W = -\Delta U \rightarrow 2$

For Ideal Gas

$W = -\Delta U = -mC_v(T_2 - T_1) \rightarrow 3$

$$W = \int PdV = \frac{P_2V_2 - P_1V_1}{1-k} = \frac{mR(T_2 - T_1)}{1-k} \rightarrow 4$$

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

$$W = \int PdV = \frac{mR(T_2 - T_1)}{1-k} = \frac{mRT1}{1-k}\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right] = \frac{P_1V_1}{1-k}\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right] \rightarrow 5$$

ENTROPY CHANGE

$\Delta S = 0$

OPEN SYSTEM

$Q = \Delta h + \Delta KE + \Delta PE + W$

$W = -\int VdP - \Delta KE - \Delta PE$

$Q = 0 \rightarrow 1$

$W = -\Delta h - \Delta KE - \Delta PE \rightarrow 2$

$-\int VdP = -\Delta h \rightarrow 3$

For Ideal Gas

$\Delta h = mC_p(T_2 - T_1) \rightarrow 4$

If $\Delta KE = 0$ and $\Delta PE = 0$

$W = -\int VdP = -\Delta h$

$-\int VdP = k\int PdV$

$$-\int VdP = \frac{k(P_2V_2 - P_1V_1)}{1-k} = \frac{kmR(T_2 - T_1)}{1-k} = \frac{kmRT1}{1-k}\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right] = \frac{kP_1V_1}{1-k}\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right] \rightarrow 5$$

POLYTROPIC PROCESS ($PV^n = C$): A Polytropic Process is an internally reversible process of an ideal or perfect gas in which $PV^n = C$, where n stands for any constants.

Using

$PV^n = C$ and $\dfrac{PV}{T} = C$

$P_1V_1^{\,n} = P_2V_2^{\,n}$ and $\dfrac{P_1V_1}{T_1} = \dfrac{P_2V_2}{T_2} \rightarrow 1$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} = \left(\frac{V_1}{V_2}\right)^{n-1} \rightarrow 2$$

From $PV^n = C$

$$P = \frac{C}{V^n} \quad \text{and} \quad V = \frac{C^{\frac{1}{n}}}{P^{\frac{1}{n}}} \rightarrow 3$$

By integration

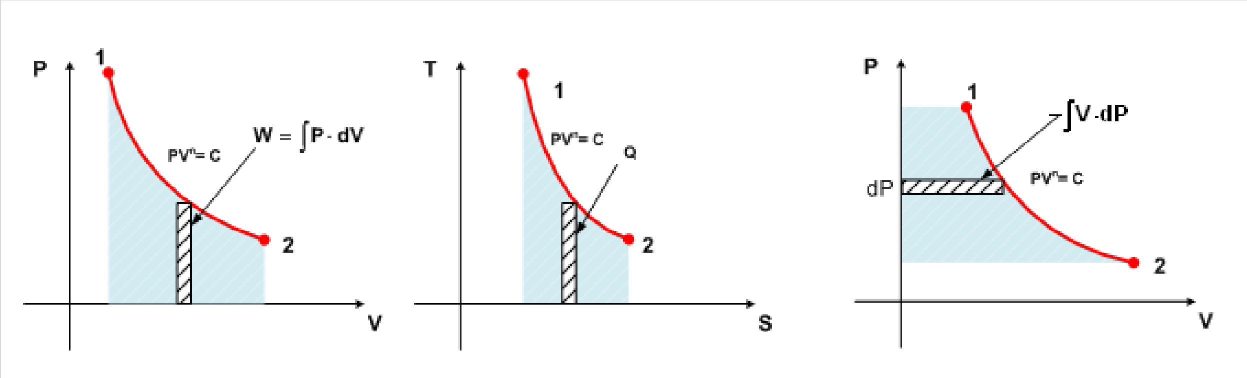
$$\int_1^2 PdV = \frac{P_2V_2 - P_1V_1}{1-n} \rightarrow 4$$

$$-\int_1^2 VdP = \frac{n(P_2V_2 - P_1V_1)}{1-n} \rightarrow 5$$

$$-\int_1^2 VdP = n\int_1^2 PdV \rightarrow 6$$

$P_1V_1 = mRT_1 \rightarrow 7$

$P_2V_2 = mRT_2 \rightarrow 8$



From

$$Q = \Delta U + W \rightarrow 9$$

$$W = \int P d\upsilon = \frac{P_2\upsilon_2 - P_1\upsilon_1}{1-n} = \frac{R(T_2 - T_1)}{1-n}$$

$$dW = \frac{RdT}{1-n} \rightarrow 10$$

$$dQ = dU + dW$$

$$dQ = C_v dT + \frac{RdT}{1-n}$$

$$R = C_p - C_v$$

$$C_p = kC_v$$

$$dQ = C_v dT + \frac{kC_v dT - C_v dT}{1-n}$$

$$dQ = CvdT + CvdT\left(\frac{k-1}{1-n}\right)$$

$$dQ = C_v dT\left(1 + \frac{k-1}{1-n}\right) = C_v dT\left(\frac{1-n+k-1}{1-n}\right)$$

$$dQ = C_v dT\left(\frac{k-n}{1-n}\right) = C_v\left(\frac{k-n}{1-n}\right)dT$$

$$\text{let}:\ C_n = C_v\left(\frac{k-n}{1-n}\right)$$

$$dQ = C_n dT$$

$$Q = C_n(T_2 - T_1) \rightarrow 10$$

Considering m

$$Q = mC_n(T_2 - T_1) \rightarrow 11$$

$$C_n = C_v\left(\frac{k-n}{1-n}\right) \rightarrow \text{Polytropic specific heat}$$

CLOSED SYSTEM

$$Q = \Delta U + W$$

$$Q = \Delta U \rightarrow 12$$

$$Q = mC_n(T_2 - T_1) \rightarrow 13$$

$$\Delta U = -mC_v(T_2 - T_1) \rightarrow 14$$

$$W = \int PdV = \frac{P_2V_2 - P_1V_1}{1-n} = \frac{mR(T_2 - T_1)}{1-n} \rightarrow 15$$

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

$$W = \int PdV = \frac{mR(T_2 - T_1)}{1-n} = \frac{mRT1}{1-n}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] = \frac{P_1V_1}{1-n}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \rightarrow 16$$

ENTROPY CHANGE

$$\Delta S = \int \frac{dQ}{T} = mC_n \int \frac{dT}{T}$$

$$\Delta S = mC_n \ln \frac{T_2}{T_1}\ \frac{KJ}{K} \rightarrow 17$$

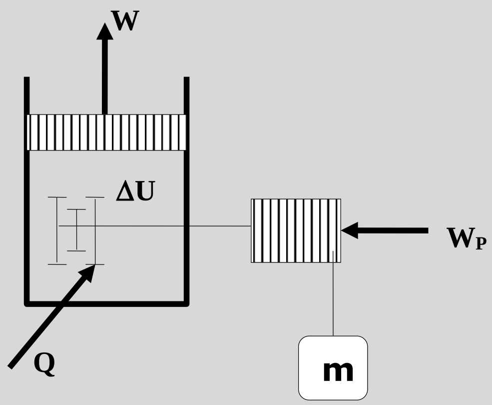
OPEN SYSTEM

$$Q = \Delta h + \Delta KE + \Delta PE + W \rightarrow 18$$
$$W = -\int VdP - \Delta KE - \Delta PE \rightarrow 19$$
$$W = Q - \Delta h - \Delta KE - \Delta PE \rightarrow 20$$
$$h = U + PV$$
$$\Delta h = \Delta U + \Delta(PV)$$
$$\Delta(PV) = \Delta h - \Delta U$$
$$\Delta h = mC_p(T_2 - T_1) \rightarrow 21$$
$$Q = mC_n(T_2 - T_1) \rightarrow 22$$
$$-\int VdP = \frac{n(P_2V_2 - P_1V_1)}{1-n} = \frac{nmR\,(T_2 - T_1)}{1-n} = \frac{nmRT_1}{1-n}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] = \frac{nP_1V_1}{1-n}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$
$$\text{If } \Delta KE = 0 \text{ and } \Delta PE = 0$$
$$W = -\int VdP$$

ISOENTHALPIC PROCESS or THROTTLING PROCESS(h = C): An Isoenthalpic Process is a steady state, steady flow, process in which W = 0, $\Delta KE = 0$, $\Delta PE = 0$, and Q = 0, where the enthalpy h remains constant.

$$h_1 = h_2$$
or
$$h = C$$

IRREVERSIBLE OR PADDLE WORK



$$Q = \Delta U + W - W_P$$

Where:

WP – irreversible or Paddle Work

Problem No. 1

Consider the design of a nozzle in which nitrogen gas flowing in a pipe at 500 KPa, 200°C, and a velocity of 10 m/sec, is to be expand to produce a velocity of 300 m/sec. Determine the exit pressure and cross-sectional area of the nozzle if the mass flow rate is 0.15 kg/sec, and the expansion is reversible and adiabatic. For N_2: R = 0.2968 KJ/kg-K ; k = 1.399; Cp = 1.04 KJ/kg-K

Given

$$P_1 = 500\,\text{KPa}$$

$$T_1 = 200 + 273 = 473\,\text{K}$$

$$v_1 = 10\,\frac{\text{m}}{\text{sec}}$$

$$v_2 = 300\,\frac{\text{m}}{\text{sec}}$$

$$m = 0.15\,\frac{\text{kg}}{\text{sec}}$$

$$\text{Process}: PV^k = C$$

$$R = 0.2968\,\frac{\text{KJ}}{\text{kg - K}}$$

$$k = 1.399$$

Problem No. 2

A counterflowing heat exchanger is used to cool air at 540 K, 400 KPa to 360 K by using 0.05 kg/sec supply of water at 20°C, 200 KPa. The airflow is 0.5 kg/sec in a 10 cm diameter pipe. Find the inlet velocity and the water exit temperature.

Given

$$Ta = 540\,\text{K}$$

$$Pa = 400\,\text{KPa}$$

$$ma = o.5\,\frac{\text{kg}}{\text{sec}}$$

$$mw = 0.05\,\frac{\text{kg}}{\text{sec}}$$

$$tw = 20°\text{C}$$

$$Pw = 200\,\text{KPa}$$

$$d = 0.10\,\text{m (pipe diameter)}$$

Problem No. 3

Consider that 1 kg of air has a decrease in internal energy of 22 KJ while its centigrade temperature is reduced to one-third of the initial temperature during a reversible nonflow constant pressure process. Determine

a) the initial temperature in°C
b) the final temperature in°C
c) the heat in KJ
d) the work in KJ
e) the entropy change in KJ/°K

For air: R = 0.287 KJ/kg-°K; k = 1.4

$$m = 1\text{kg}$$

$$\Delta U = -22\,\text{KJ}$$

$$t_2 = \frac{t_1}{3}$$

$$Q = \Delta U = -22 = 1(0.7175)\left(\frac{t_1}{3} - t_1\right) = 0.7175\left(\frac{-2t_1}{3}\right)$$

$$t_1 = \frac{3(-22)}{-2(0.7175)} = 46°\text{C}$$

$$T_{1=} 319\,\text{K}$$

$$t_2 = 15.33°\text{C}$$

$$T_2 = 288.33\,\text{K}$$

$$Q = -22\text{KJ}$$

$$\Delta S = mC_v \ln\frac{T_2}{T_1} = \text{-0.0726}\frac{\text{KJ}}{\text{K}}$$

Problem No. 4

Nitrogen gas enters a turbine operating at steady state through a 5 cm duct with a velocity of 60 m/sec, a pressure of 345 KPa and a temperature of 550 K. At the exit, the velocity is 0.6 m/sec, the pressure is 138 KPa and the temperature is 389 K. Heat transfer from the surface is 37.2 KJ/kg of nitrogen flowing. Neglecting PE effects and using the ideal gas model, determine the power developed by the turbine in KW. (For N2, M = 28 kg/kg mol ; k = 1.399)

Given

$v_1 = 60 \text{ m/sec}$

$d_1 = 0.05 \text{ m}$

$P_1 = 345 \text{ KPa} \; ; T1 = 550 \text{ K}$

$v_2 = 0.6 \text{ m/sec}$

$P_2 = 138 \text{ KPa} \; ; T2 = 389 \text{ K}$

$Q = \text{-}37.2 \text{ KJ/kg}$

$\Delta PE = 0$

$M = 28 \; ; k = 1.399$

$$R = \frac{8.3143}{28} = 0.297 \frac{KJ}{kg - K}$$

$$\upsilon_1 = \frac{0.297(550)}{345} = 0.47 \frac{m^3}{kg}$$

$$A = \frac{\pi}{4}(0.05)^2 = 0.00196 \text{ m}^2$$

$$m = \frac{Av_1}{\upsilon_1} = \frac{0.00196(60)}{0.47} = 0.250 \text{ kg/sec}$$

$$Cp = \frac{Rk}{k-1} = 1.0414$$

$$Cv = \frac{R}{k-1} = 0.7444$$

$$\Delta h = mCp(T2 - T1) = -41.915 \text{ KW}$$

$$\Delta KE = \frac{m(v_2^{\,2} - v_1^{\,2})}{2(1000)} = -0.450 \text{ KW}$$

$$Q = 0.25(-37.2) = -9.3 \text{ KW}$$

$$W = Q - \Delta h - \Delta KE - \Delta PE = 33.065 \text{ KW}$$

Problem No. 5

Air (R = 0.287 KJ/kg-K; k =1.4) is contained in a cylinder fitted with a frictionless piston. Initially the cylinder contains 500 liters of air at 150 KPa and 20°C. The air is then compressed in a polytropic process until the final pressure is 600 KPa and the temperature is 120°C. Calculate

a. The polytropic exponent n

b. The final volume V 2 in li

c. The work W

d. The heat transfer Q

$V_1 = 0.50 \text{ m}^3 ; P_1 = 150 \text{ KPa} \; ; T_1 = 273 + 20 = 293 \text{ K}$

$P_2 = 600 \text{ KPa} \; ; T_2 = 273 + 120 = 393 \text{ K}$

$$\frac{n-1}{n} = \frac{\ln \frac{T_2}{T_1}}{\ln \frac{P_2}{P_1}}$$

$n = 1.269$

$$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} = 0.168 \text{ m}^3$$

$$W = \frac{(P_2V_2 - P_1V_1)}{1-n} = -96 \text{ KJ}$$

$$Q = mCn(T2 - T1) = -34.94$$

$$Cn = Cv\frac{k-n}{1-n} = -0.349$$

Problem No. 6

Consider 2 kg of CO and 1 kg of CH_4 at 32°C that are in a 0.6 m^3 rigid drum. Find:

a. the mixture pressure P in KPa
b. the volumetric analysis
c. the partial pressures in KPa
d. the heat to cause a temperature rise of 50°C.

Given :

$m_{CO} = 2 \text{ kg}$

$m_{CH4} = 1 \text{ kg}$

$T = 32 + 273 = 305 \text{ K}$

$V = 0.6 \text{ m}^3$

Problem No. 7

A gaseous mixture has the following volumetric analysisO_2, 30%; CO_2, 40% N_2, 30%. Determine

a. the analysis on a mass basis
b. the partial pressure of each component if the total pressure is 100 KPa and the temperature is 32°C
c. the molecular weight and gas constant of the mixture

Given

Volumetric Analysis

$y_{O2} = 30\%$

$y_{CO2} = 40\%$

$y_{N2} = 30\%$

Problem No. 8

A gaseous mixture has the following analysis on a mass basis, CO_2, 30%; SO_2, 30%; He, 20% and N_2, 20%. For a total pressure and temperature of 101 KPa and 300 K, Determine

a. the volumetric or molal analysis
b. the component partial pressure
c. the mixture gas constant
d. the mixture specific heats

Given

Gravimetri c Analysis

$y_{CO2} = 30\%$

$y_{SO2} = 30\%$

$yHe = 20\%$

$yN2 = 20\%$

$P = 101\,KPa$

$T = 300\,K$

Problem No. 9

A cubical tank 1 m on a side, contains a mixture of 1.8 kg of nitrogen and 2.8 kg of an unknown gas. The mixture pressure and temperature are 290 KPa and 340 K. Determine

a. Molecular weight and gas constant of the unknown gas
b. the volumetric analysis

Problem No. 10

A mixture of ideal gases at 30°C and 200 KPa is composed of 0.20 kg CO_2, 0.75 kg N_2, and 0.05 kg He. Determine the mixture volume.

Given

$T = 30 + 273 = 303\,K$

$m_{CO2} = 20\,kg$

$m_{N2} = 0.75\,kg$

$m_{He} = 0.05\,kg$

Problem No. 11

A gaseous mixture composed of 25 kg of N_2, 3.6 kg of H_2, and 60 kg of CO_2 is at 200 KPa, 50°C. Find the respective partial pressures and compute the volume of each component at its own partial pressure and 50°C. Given: m_{N2} = 25 kg ; m_{H2} = 3.6 kg ; m_{CO2} = 60 kg

m = 25 + 3.6 + 60 = 88.6 kg

x_{N2} = 0.282 ; x_{H2} = 0.041 ; x_{CO2} = 0.678

P = 200 KPa ; T = 323 K

$$yi = \frac{\frac{xi}{Mi}}{\Sigma \frac{xi}{Mi}} \quad yi = \frac{Pi}{P}$$

$$\Sigma \frac{xi}{Mi} = \frac{0.282}{28} + \frac{.041}{2} + \frac{.678}{44} = 0.046$$

$$y_{N_2} = 0.219$$

$$y_{H_2} = 0.446$$

$$y_{CO_2} = 0.335$$

$$P_i V_i = m_i R_i T_i$$

$$V_{N_2} = \frac{25(0.297(323)}{43.8} = 54.76 \ m^3$$

$$V_{H_2} = \frac{3.6(4.16)(323)}{89.2} = 54.23 \ m^3$$

$$V_{CO_2} = \frac{60(0.189)(323)}{67} = 54.67 \ m^3$$

P_{N2} = .219(200) = 43.8 KPa

P_{H2} = .446(200) = 89.2 KPa

P_{CO2} = 0.335(200) = 67 KPa

Problem 12

Assume 2 kg of O_2 are mixed with 3 kg of an unknown gas. The resulting mixture occupies a volume of 1.2 m^3 at 276 KPa and 65°C. Determine

a) R and M of the unknown gas constituent

b) the volumetric analysis

c) the partial pressures

Given; m_{O2} = 2 kg; m_x = 3 kg

V = 1.2 m^3 ; P = 276 KPa; T = 338 K

a) m = 5 kg

x_{O2} = 0.40 ; x_x = 0.60

$PV = mRT$

R = 0.1361 KJ/kg-K

R = .40(0.26) + 0.60(R_x)

R_x = 0.535 KJ/kg-K

M_x = 15.54 kg/kg_m

b) $$\Sigma \frac{x_i}{M_i} = \frac{.40}{32} + \frac{.60}{15.54} = 0.0511$$

y_{O2} = 0.245 ; y_x = 0.755

c) P_{O2} = .245(276) = 67.62 KPa ;

P_x = 0.755(276) = 208.38 KPa

Problem 13

A piston cylinder contains air at 600 KPa, 290 K and a volume of 0.01 m^3. A constant pressure process gives 54 KJ of work out. Determine the heat transfer of the process.

Given:

P_1 = P2 = 600 KPa

T_1 = 290 K

V_1 = 0.01 m^3

W = 54 KJ (work out)

$$W = P(V_2 - V_1) = mR(T_2 - T_1)$$

$$P_1 V_1 = mRT_1$$

$$m = \frac{P_1 V_1}{RT_1} = 0.07 \ kg$$

$$T_2 = \frac{W}{mR} + T_1 = 478.4 \, K$$

$$Q = mC_p(T_2 - T_1)$$

$$C_p = \frac{Rk}{k-1} = 1.0045 \frac{KJ}{kg-K}$$

$$Q = 13.24 \, KJ$$

Problem 14

A piston cylinder device, whose piston is resting on a set stops, initially contains 3 kg of air at 200 KPa and 27°C. The mass of the piston is such that a pressure of 400 KPa is required to move it. Heat is now transferred to the air until its volume doubles. Determine the work done by the air and the total heat transferred to the air during this process. Also show the process on a P-V diagram.

Given:

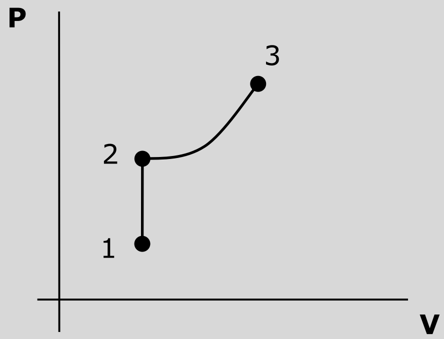
$m = 3$ kg

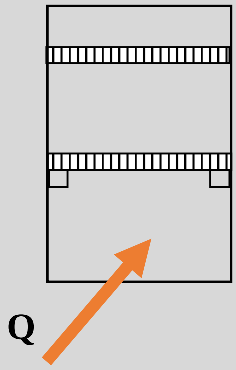
$P_1 = 200$ KPa

$T_1 = 27 + 273 = 300$ K

$P = 400$ KPa

$V_2 = 2V_1$





$$\text{At } V = C$$
$$V_1 = V_2$$
$$\frac{P_2}{P_1} = \frac{T_2}{T_1}$$
$$T_2 = 600\text{ K}$$
$$Q_v = 3(0.7175)(600 - 300) = 645.75\text{ KJ}$$
$$\text{At } P = C$$
$$\frac{V_3}{V_2} = \frac{T_3}{T_2}$$
$$V_3 = 2V_1 = 2.6\text{ m}^3$$
$$\frac{2.6}{1.3} = \frac{T_3}{600}$$
$$T_3 = 1200\text{ K}$$
$$Q_p = mC_p(T_3 - T_2) = 3(1.0045)(1200 - 600)$$
$$Q_p = 1808.1\text{ KJ}$$
$$W = P(V_3 - V_2) = 400(2.6 - 1.3)$$
$$W = 520\text{ KJ}$$
$$Q_T = 645.75 + 1808.1 = 2453.85\text{ KJ}$$

Problem No. 15

During some actual expansion and compression processes in piston cylinder devices, the gases have been observed to satisfy the relationship $PV^n = C$, where n and C are constants. Calculate the work done when a gas expands from a state of 150 KPa and 0.03 m^3 to a final volume of 0.2 m^3 for the case of n = 1.3. Also show the process on the PV diagram.

Given:

$P_1 = 150$ KPa ; $V_1 = 0.03\ m^3$

$V_2 = 0.2\ m^3$

$$PV^n = C$$
$$P_1V_1^n = P_2V_2^n = C$$
$$\text{For a Closed system}$$
$$W = \int P \bullet dV$$
$$P = \frac{C}{V^n}$$
$$\text{By integration}$$
$$W = \int_1^2 PdV = \frac{P_2V_2 - P_1V_1}{1-n}$$
$$P_2 = \frac{P_1V_1^n}{V_2^n} = P_1\left(\frac{V_1}{V_2}\right)^n = 150\left(\frac{0.03}{0.2}\right)^{1.3}$$
$$P_2 = 12.74\text{ KPa}$$
$$W = \int_1^2 PdV = \frac{P_2V_2 - P_1V_1}{1-n} = \frac{12.74(0.2) - 150(0.03)}{1-1.3}$$
$$W = 6.533\text{ KJ}$$

Problem No. 15

1. A. How many kilograms of nitrogen must be mixed with 3.6 kg of CO_2 in order to produce a gaseous mixture that is 50% by volume of each constituent?

   B. For the resulting mixture, determine M and R, and the partial pressure of the N_2 if that of the CO_2 is 138 KPa.

$m_{CO2} = 3.6$ kg ; $y_{CO2} = 0.50$; $y_{N2} = 0.50$

$$x_{CO2} = \frac{y_{CO2}M_{CO_2}}{M};\ x_{N2} = \frac{y_{N2}M_{N2}}{M}; M = \sum y_iM_i$$

$$M = 0.5(44) + 0.50(28) = 22 + 14 = 36M = 0.5(44) + 0.50(28) = 22 + 14 = 36$$

$$x_{CO2} = \frac{22}{36} = 0.61;\ x_{N2} = \frac{14}{36} = 0.39$$

$$x_{CO2} = \frac{m_{CO2}}{m_{CO2} + m_{N2}}$$

$$0.61 = \frac{3.6}{3.6 + m_{N2}}$$

$$m_{N2} = \frac{3.6 - 2.196}{0.61} = 2.3\,\text{kg}$$

$$M = 36$$

$$R = \frac{8.3143}{36} = 0.231\,\text{KJ/kg - °K}$$

$$yi = \frac{Pi}{P}$$

$$y_{CO2} = \frac{138}{P} = 0.50$$

$$P = 276\,\text{KPa}$$

$$P_{N2} = 0.50(276) = 138\ \text{KPa}$$

Problem No. 15

Assume 2 kg of O_2 are mixed with 3 kg of an unknown gas. The resulting mixture occupies a volume of 1.2 m^3 at 276 KPa and 65°C. Determine

a) R and M of the unknown gas constituent

b) the volumetric analysis

c) the partial pressures

Given; $m_{O2} = 2$ kg; $m_x = 3$ kg

$V = 1.2\ m^3$; $P = 276$ KPa; $T = 338$ K

a)

$m = 5$ kg

$x_{O2} = 0.40$; $x_x = 0.60$

$$PV = mRT$$

R = 0.1361 KJ/kg-K

$R = .40(0.26) + 0.60(R_x)$

$R_x = 0.535$ KJ/kg-K

$M_x = 15.54$ kg/kg_m

b)

$$\sum\frac{x_i}{M_i} = \frac{.40}{32} + \frac{.60}{15.54} = 0.0511$$

$y_{O2} = 0.245$; $y_x = 0.755$

c) $P_{O2} = .245(276) = 67.62$ KPa ; $P_x = 0.755(276) = 208.38$ KPa

Problem No. 17

One mole of a gaseous mixture has the following gravimetric analysis: O_2 = 16%, CO_2 = 44%, N_2 = 40%. Find :

a) the molecular weight of the mixture

b) the mass of each constituent,

c) the moles of each constituent in the mixture,

d) R and

e) Partial pressure for P = 207 KPa.

Given: $x_{O2} = 0.16$; $x_{CO2} = .44$; $x_{N2} = 0.40$

$$yi = \frac{\frac{xi}{Mi}}{\sum \frac{xi}{Mi}}$$

$$\sum \frac{xi}{Mi} = \frac{0.16}{32} + \frac{.44}{44} + \frac{.40}{28} = 0.0293$$

$y_{O_2} = 0.171$

$y_{CO_2} = 0.341$

$y_{N_2} = 0.488$

M = .171(32) + .341(44) + .488(28) = 34.14 kg/kgm

$$n = \sum ni$$

$$yi = \frac{ni}{n}$$

n_{O2} = .171 moles; n_{CO2} = .341; n_{N2} = .488

$$M = \frac{m}{n}$$

m_{O2} = .171(32) = 5.472 kg
m_{CO2} =.341(44) = 15.004 kg
m_{N2} = .488(28) = 13.664 kg
m = 5.472 + 15.004 + 13.664 = 34.14 kg

$$R = \frac{8.3143}{M}$$ = 0.244 KJ/kg-K

$$yi = \frac{Pi}{P}$$

P_{O2}= .171(207) = 35.4 KPa
P_{CO2} = .341(207) = 70.6 KPa
P_{N2} = .488(207) = 101.016 KPa

Problem No. 18

A gaseous mixture composed of 25 kg of N_2, 3.6 kg of H_2, and 60 kg of CO_2 is at 200 KPa, 50°C. Find the respective partial pressures and compute the volume of each component at its own partial pressure and 50°C.

Given: m_{N2} = 25 kg ; m_{H2} = 3.6 kg ; m_{CO2} = 60 kg
m = 25 + 3.6 + 60 = 88.6 kg
x_{N2} = 0.282 ; x_{H2} = 0.041 ; x_{CO2} = 0.678
P = 200 KPa ; T = 323 K

$$yi = \frac{\frac{xi}{Mi}}{\sum \frac{xi}{Mi}}$$

$$\sum \frac{xi}{Mi} = \frac{0.282}{28} + \frac{.041}{2} + \frac{.678}{44} = 0.046$$

$y_{N_2} = 0.219$

$y_{H_2} = 0.446$

$y_{CO_2} = 0.335$

$$yi = \frac{Pi}{P}$$

P_{N2} = .219(200) = 43.8 KPa
P_{H2} = .446(200) = 89.2 KPa
P_{CO2} = 0.335(200) = 67 KPa

$$P_i V_i = m_i R_i T_i$$

$$V_{N_2} = \frac{25(0.297(323)}{43.8} = 54.76 \ m^3$$

$$V_{H_2} = \frac{3.6(4.16)(323)}{89.2} = 54.23 \ m^3$$

$$V_{CO_2} = \frac{60(0.189)(323)}{67} = 54.67 \ m^3$$

Problem No. 19

A 0.23 m3 drum contains a gaseous mixture of CO_2 and CH_4 each 50% by mass at P = 689 KPa, 38°C; 1 kg of O_2 are added to the drum with the mixture temperature remaining at 38°C. For the final mixture, find;

a) the gravimetric analysis
b) the volumetric analysis
c) the Cp (For CO_2: k = 1.288; CH_4: k = 1.321; O_2: k = 1.395)
d) the total pressure P

Problem No. 20

Assume 28 m^3 of a gaseous mixture whose gravimetric analysis is 20% CO_2, 15% O_2, 65% N_2, are at 103.4 KPa and 150°C. Find

a) the volumetric analysis
b) the respective partial pressures
c) R and M
d) the moles of mixture and of each constituent
e) the heat transferred with no change in pressure to reduce the temperature to 75°C
f) the volume the mixture occupies after the cooling

Problem 21

Consider 2 kg of CO and 1 kg of CH4 at 32°C that are in a 0.6 m3 rigid drum. Find:

a) the mixture pressure P in KPa
b) the volumetric analysis
c) the partial pressures in KPa
d) the heat to cause a temperature rise of 50°C.

Problem 22

A gaseous mixture has the following volumetric analysisO_2, 30%; CO_2, 40% N_2, 30%. Determine

a) the analysis on a mass basis
b) the partial pressure of each component if the total pressure is 100 KPa and the temperature is 32°C
c) the molecular weight and gas constant of the mixture

Problem 23

A gaseous mixture has the following analysis on a mass basis, CO_2, 30%; SO_2, 30%; He, 20% and N_2, 20%. For a total pressure and temperature of 101 KPa and 300 K, Determine

a) the volumetric or molal analysis
b) the component partial pressure
c) the mixture gas constant
d) the mixture specific heats

Problem 24

A cubical tank 1 m on a side, contains a mixture of 1.8 kg of nitrogen and 2.8 kg of an unknown gas. The mixture pressure and temperature are 290 KPa and 340 K. Determine

a) Molecular weight and gas constant of the unknown gas
b) the volumetric analysis

Problem 25

A mixture of ideal gases at 30°C and 200 KPa is composed of 0.20 kg CO_2, 0.75 kg N_2, and 0.05 kg He. Determine the mixture volume.

FUELS AND COMBUSTION

Fuel **-** a substance composed of chemical elements, which in rapid chemical union with oxygen produced combustion.
Combustion **-** is that rapid chemical union with oxygen of an element, whose exothermic heat of reaction is sufficiently great and whose rate of reaction is sufficiently fast, whereby useful quantities of heat are liberated at elevated temperatures. It is the burning or oxidation of the combustible elements.

TYPES OF FUEL

1) Solid Fuels
   Example: a. coal
   b. charcoal
   c. coke
   d. woods
2) Liquid Fuels (obtained by the distillation of petroleum)
   Example: a. Gasoline
   b. kerosene
   c. diesoline
   d. Fuel oil
   e. alcohol (these are not true hydrocarbons, since it contains oxygen in the molecule)
3) Gaseous Fuels (a mixture of various constituents hydrocarbons, its combustion products do not have sulfur components)
   Example:
   a. Natural Gas (example: methane, ethane, propane)
   b. Coke oven gas -obtained as a byproduct of making coke
   c. Blast furnace gas - a byproduct of melting iron ore
   d. LPG
   e. Producer Gas - fuel used for gas engines
4) Nuclear Fuels
   Example:
   a. Uranium
   b. Plutonium

COMBUSTIBLE ELEMENTS

1. Carbon (C)
2. Hydrogen (H_2)
3. Sulfur (S)

TYPES OF HYDROCARBONS

1) Paraffin - all ends in "ane"
   Formula: C_nH_{2n+2}
   Structure: Chain (saturated)
   Example:
   GAS
   a. Methane(CH_4)
   b. Ethane (C_2H_6)
   LPG
   a. Propane (C_3H_8)
   b. Butane (C_4H_{10})
   c. Pentane (C_5H_{12})

   GASOLINE
   a. n-Heptane (C_7H_{16})
   b. Triptane (C_7H_{16})
   c. Iso- octane (C_8H_{18})

   FUEL OIL
   a. Decane ($C_{10}H_{22}$)
   b. Dodecane ($C_{12}H_{26}$)
   c. Hexadecane ($C_{16}H_{34}$)
   d. Octadecane ($C_{18}H_{38}$)

2) Olefins - ends in "ylene" or "ene"

Formula: C_nH_{2n}

Structure: Chain (unsaturated)

Example:

a. Propene (C_3H_6)
b. Butene (C_4H_8)
c. Hexene (C_6H_{12})
d. Octene (C_8H_{16})

3) DIOLEFIN - ends in "diene"

Formula: C_nH_{2n-2}

Structure: Chain (unsaturated)

Example:

a. Butadiene (C_4H_6)
b. Hexadiene (C_6H_{10})

4) NAPHTHENE - named by adding the prefix "cyclo"

Formula: C_nH_{2n}

Structure: Ring (saturated)

Example:

a. Cyclopentane (C_5H_{10})
b. Cyclohexane (C_6H_{12})

5) AROMATICS - this hydrocarbon includes the;

A. Benzene Series (C_nH_{2n-6})

B. Naphthalene Series (C_nH_{2n-12})

Structure: Ring (unsaturated)

Example:

a. Benzene (C_6H_6)
b. Toluene (C_7H_8)
c. Xylene (C_8H_{10})

6) ALCOHOLS - These are not true hydrocarbon, but sometimes used as fuel in an internal combustion engine. The characteristic feature is that one of the hydrogen atom is replaced by an OH radical.

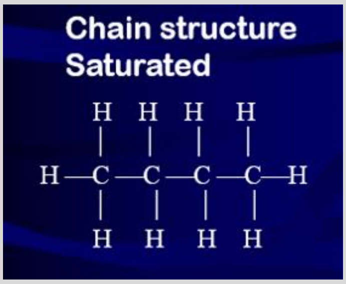
Example:

a. Methanol (CH_4O or CH_3OH)
b. Ethanol (C_2H_6O or C_2H_5OH)

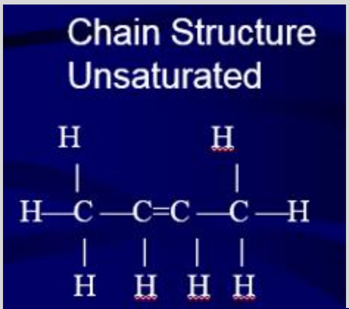
Saturated Hydrocarbon - all the carbon atoms are joined by a single bond.
Unsaturated Hydrocarbon - it has two or more carbon atoms joined by a double or triple bond.
Isomers - two hydrocarbons with the same number of carbon and hydrogen atoms, but at different structure.

STRUCTURE OF C_nH_m

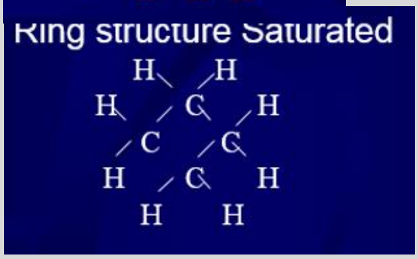
Chain Structure (saturated)



Chain Structure (unsaturated)



c. Ring Structure (saturated)



Family	Formula	Structure	Saturated
Paraffin	C_nH_{2n+2}	Chain	Yes
Olefin	C_nH_{2n}	Chain	No
Diolefin	C_nH_{2n-2}	Chain	No
Naphthene	C_nH_{2n}	Ring	Yes
Aromatic			
Benzene	C_nH_{2n-6}	Ring	No
Naphthalene	C_nH_{2n-12}	Ring	No
Alcohols		Note: Alcohols are not pure hydrocarbon, because one of its hydrogen atom is replace by an OH radical. Sometimes it is used as fuel in an ICE.	
Methanol	CH_3OH		
Ethanol	C_2H_5OH		

THE COMBUSTION CHEMISTRY

A. Oxidation of Carbon

$C + O_2 \rightarrow CO_2$

Mole Basis:

$1 + 1 \rightarrow 1$

Mass Basis:

$1(12) + 1(32) \rightarrow 1(12 + 32)$

$12 + 32 \rightarrow 44$

$3 + 8 \rightarrow 11$

B. Oxidation of Hydrogen

$H_2 + ½O_2 \rightarrow H_2O$

Mole Basis:

$1 + ½ \rightarrow 1$

Mass Basis:

$1(2) + ½(32) \rightarrow 1(2 + 16)$

$2 + 16 \rightarrow 18$

$1 + 8 \rightarrow 9$

C. Oxidation of Sulfur

$S + O_2 \rightarrow SO_2$

Mole Basis:

$1 + 1 \rightarrow 1$

Mass Basis:

$1(32) + 1(32) \rightarrow 1(32+32)$

$32 + 32 \rightarrow 64$

$1 + 1 \rightarrow 2$

Complete Combustion: Occurs when all the combustible elements has been fully oxidized.

Example: $C + O_2 \rightarrow CO_2$

Incomplete Combustion: Occurs when some of the combustible elements have not been fully oxidized and it may result from;

a. Insufficient oxygen
b. Poor mixing of fuel and oxygen
c. the temperature is too low to support combustion.

Result: Soot or black smoke that sometimes pours out from chimney or smokestack.

Example: $C + ½O_2 \rightarrow CO$

Composition of Air

a) Volumetric or Molal analysis

$O_2 = 21\%$

$N_2 = 79\%$

b) Gravimetric Analysis

$O_2 = 23.3\%$

$N_2 = 76.7\%$

$$\frac{\text{moles of } N_2}{\text{mole of } O_2} = \frac{79}{21} = 3.76$$

COMBUSTION WITH AIR

A) Combustion of Carbon with air

$C + O_2 + (3.76)N_2 \rightarrow CO_2 + (3.76)N_2$

Mole Basis

$1 + 1 + 3.76 \rightarrow 1 + 3.76$

Mass Basis

$1(12) + 1(32) + (3.76)(28) \rightarrow 1(44) + (3.76)(28)$

$12 + 32 + 3.76(28) \rightarrow 44 + 3.76(28)$

$3 + 8 + 3.76(7) \rightarrow 11 + 3.76(7)$

$$\frac{\text{kg of air}}{\text{kg of C}} = \frac{8+3.76(7)}{3}$$

$$\frac{\text{kg of air}}{\text{kg of C}} = 11.44$$

B) Combustion of Hydrogen with Air

$H_2 + ½O_2 +½3.76)N_2 \rightarrow H_2O + ½(3.76)N_2$

Mole Basis:

$1 + ½ + ½(3.76) \rightarrow 1 + ½(3.76)$

Mass Basis:

$1(2) + ½(32) + ½(3.76)(28) \rightarrow 1(18) + ½(3.76)(28)$

$2 + 16 + 3.76(14) \rightarrow 18 + 3.76(14)$

$$\frac{\text{kg of air}}{\text{kg of H}} = \frac{16+3.76(14)}{2}$$

$$\frac{\text{kg of air}}{\text{kg of H}} = 34.32$$

C) Combustion of Sulfur with air

$S + O_2 + 3.76N_2 \rightarrow SO_2 + 3.76N_2$

Mole Basis:

$1 + 1 + 3.76 \rightarrow 1 + 3.76$

Mass Basis:

$1(32) + 1(32) + (3.76)(28) \rightarrow 1(64) + (3.76)(28)$

$8 + 8 + (3.76)(7) \rightarrow 16 + 3.76(7)$

$$\frac{\text{kg of air}}{\text{kg of S}} = \frac{8+3.76(7)}{8}$$

$$\frac{\text{kg of air}}{\text{kg of S}} = 4.29$$

THEORETICAL AIR

It is the minimum amount of air required to oxidized the reactants. With theoretical air no O_2 is found in the products.

EXCESS AIR

It is an amount of air in excess of the theoretical air required to influence complete combustion. With excess air, O_2 is present in the products. Excess air is usually expressed as a percentage of the theoretical air. But in actual combustion, although there is an amount of excess air, the presence of CO and other hydrocarbon in the products cannot be avoided.

Example: 25% excess air is the same as 125% theoretical air.

COMBUSTION OF HYDROCARBON FUEL (CnHm)

Combustion of C_nH_m with 100% theoretical air

$C_nH_m + aO_2 + a(3.76)N_2 \rightarrow bCO_2 + cH_2O + a(3.76)N_2$

$$\left(\frac{A}{F}\right)_t = \frac{a(32)+a(3.76)(28)}{12n+m} \quad \frac{\text{kg of air}}{\text{kg of CnHm}}$$

In theoretical combustion, for the combustion of CnHm,

$a = n + 0.25m$

$b = n$

$c = 0.5m$

Combustion with excess air

$C_nH_m + (1+e)aO_2 + (1+e)a(3.76)N_2 \rightarrow bCO_2 + cH_2O + dO_2 + (1+e)a(3.76)N_2$

$$\left(\frac{A}{F}\right)_a = (1+e)\left(\frac{A}{F}\right)_t$$

$$\left(\frac{A}{F}\right)_a = (1+e)\ \frac{a(32)+a(3.76)(28)}{12n+m}\ \ \frac{\text{kg of air}}{\text{kg of CnHm}}$$

$$d = e(n+0.25m)$$

where:
e - excess air in decimal
$(A/F)_t$ - theoretical air-fuel ratio
$(A/F)_a$ - actual air-fuel ratio

Actual or Real world Combustion Process

Combustion with CO in the products due to incomplete combustion (100% theoretical air)

$$CnHm + aO_2 + a(3.76)N_2 \rightarrow bCO_2 + cH_2O + dCO + a(3.76)N_2$$

Combustion with CO in the products due to incomplete combustion (with excess air)

$$CnHm + (1+e)aO_2 + (1+e)a(3.76)N_2 \rightarrow bCO_2 + cH_2O + dCO + fO_2 + (1+e)a(3.76)N_2$$

Typical Real-World Engine Combustion Process:

$$\text{Fuel (CnHm)} + \text{Air } (O_2 \text{ and } N_2) \rightarrow CO_2 + H_2O + O_2 + N_2 + CH(VOC's) + CO + NOx$$

CH(VOC's) - Volatile Organic Compounds
CO - Carbon Monoxide
NOx - Nitrogen Oxides

Note: Values of a, b, c, and d in terms of n and m in theoretical combustion, cannot be applied in actual in an actual combustion process.

Common Combustion Gases

Gases	Molecular Weight
C	12
H	1
H_2	2
O	16
O_2	32
N	14
N_2	28
S	32

COMBUSTION OF SOLID FUELS

Components of Solid Fuels: C, H_2, O_2, N_2, S, and Moisture

A) Combustion with 100% theoretical air

$aC + bH_2 + cO_2 + dN_2 + eS + fH_2O + gO_2 + g(3.76)N_2 \rightarrow hCO_2 + iH_2O + jSO_2 + kN_2$

B) Combustion with excess air x

Where:
x - excess air in decimal

$aC + bH_2 + cO_2 + dN_2 + eS + fH_2O + (1+x)gO_2 + (1+x)g(3.76)N_2 \rightarrow hCO_2 + iH_2O + jSO_2 + lO_2 + mN_2$

The theoretical and actual air-fuel ratio of solid fuels can be computed based on their balance combustion equation above.

$$\left(\frac{A}{F}\right)_t = \frac{32g + 3.76(28)g}{12a + 2b + 32c + 28d + 32e + 18f}$$

$$\left(\frac{A}{F}\right)_a = (1 + x)\left(\frac{A}{F}\right)_t$$

DEW POINT TEMPERATURE

The Dew Point Temperature (t_{dp}) is the saturation temperature corresponding the partial pressure of the water vapor in the mixture (products of combustion).

ULTIMATE ANALYSIS

Ultimate Analysis gives the amount of C, H_2, O_2, N_2, S and moisture in percentages by mass, sometimes the percentage amount of Ash is given.

100% = %C + %H_2 + %O_2 + %N_2 + %S + %H_2O + %Ash

$$\left(\frac{A}{F}\right)_t = 11.44C + 34.32\left(H - \frac{O_2}{8}\right) + 4.29S \;\; \frac{\text{kg of air}}{\text{kg of Fuel}}$$

where: C, H, O and S are in decimals obtained from the Ultimate Analysis

PROXIMATE ANALYSIS

Proximate Analysis gives the percentage amount of Fixed Carbon, Volatiles, Ash and Moisture.

100% = % Fixed Carbon + % Volatiles + % Ash + % Moisture

ORSAT ANALYSIS

Orsat Analysis gives the volumetric or molal analysis of the products of combustion or exhaust gases on a Dry Basis.

100% = % CO_2 + % CO + % O_2 + % SO_2 + % N_2 + % CH + % NO + NO_x

$$e = \frac{100 - \left[4.76 + 11.28\frac{H}{C}\right]CO_2}{\left[4.76 + 14.28\frac{H}{C}\right]CO_2}$$

$$\left(\frac{A}{F}\right)_a = \frac{11.44(1 + e)\left[1 + 3\frac{H}{C}\right]}{\left[1 + \frac{H}{C}\right]} \;\; \frac{\text{kg o air}}{\text{kg of Fuel}}$$

where:
H and C are in % (by mass)
CO_2 is in % by volume from the exhaust gas analysis

MASS FLOW RATE OF FLUE GAS

a) Without considering Ash loss:

$$m_g = m_F\left(\frac{A}{F} + 1\right)$$

b) Considering Ash loss

$$m_g = m_F\left(\frac{A}{F} + 1 - \text{Ash loss}\right)$$

where ash loss in decimal

Properties of Fuels and Lubricants

a) **Viscosity -** a measure of the resistance to flow that a lubricant offers when it is subjected to shear stress.
b) **Absolute Viscosity -** viscosity which is determined by direct measurement of shear resistance.
c) **Kinematics Viscosity -** the ratio of the absolute viscosity to the density
d) **Viscosity Index -** the rate at which viscosity changes with temperature.
e) **Flash Point -** the temperature at which the vapor above a volatile liquid forms a combustible mixture with air.
f) **Fire Point -** The temperature at which oil gives off vapor that burns continuously when ignited.
g) **Pour Point -** the temperature at which oil will no longer pour freely.
h**) Dropping Point -** the temperature at which grease melts.
i) **Condradson Number**(carbon residue) **-** the percentage amount by mass of the carbonaceous residue remaining after destructive distillation.
j) **Octane Number -** a number that provides a measure of the ability of a fuel to resist knocking when it is burnt in a gasoline engine. It is the percentage by volume of iso-octane in a blend with normal heptane that matches the knocking behavior of the fuel.
k) **Cetane Number -** a number that provides a measure of the ignition characteristics of a diesel fuel when it is burnt in a standard diesel engine. It is the percentage of cetane in the standard fuel.

PROPERTIES OF COMMON GASES

GAS	M	R	k	Cp	Cv	k
O_2	32	0.260	1.395	0.918	0.658	1.395
CO_2	44	0.189	1.288	0.845	0.656	1.288
N_2	28	0.297	1.399	1.041	0.744	1.399
C_2H_2	26	0.320	1.232	1.698	1.378	1.232
AIR	28.97	0.287	1.4	1.004	0.717	1.4
ARGON	39.95	0.208	1.666	0.521	0.312	1.666
CO	28	0.297	1.399	1.041	0.744	1.399
Cl2	70.914	0.117	1.324	0.479	0.362	1.324
C_2H_6	30	0.277	1.187	1.759	1.482	1.187
C_2H_4	28	0.297	1.24	1.534	1.237	1.24
He	4	2.079	1.666	5.200	3.121	1.666
H2	2	4.157	1.4	14.550	10.393	1.4
N2H4	32	0.260	1.195	1.592	1.332	1.195
CH_4	16	0.520	1.321	2.138	1.619	1.321
Ne	20.183	0.412	1.666	1.030	0.619	1.666
C_3H_8	44	0.189	1.127	1.677	1.488	1.127
SO_2	64	0.130	1.263	0.624	0.494	1.263
H_2O	18	0.462	1.329	1.866	1.404	1.329
Xe	131.3	0.063	1.666	0.158	0.095	1.666

Problem No. 1
An automotive internal combustion engine burns liquid Octane ($C_3 H_8$) at the rate of 0.005 kg/sec, and uses 20% excess air. The air and fuel enters the engine at 25°C and the combustion products leaves the engine at 900 K. It may be assumed that 90% of the carbon in the fuel burns to form CO_2 and the remaining 10% burns to form CO. Determine

a. The actual air – fuel ratio
b. The kg/s of actual air
c. The M and R of the products

Combustion with 100% theoretical air
Combustion Equation with 20% excess air and 10% of C burns to CO

$$C_3H_8 + 5.82O_2 + 21.88N_2 \rightarrow 2.7CO_2 + 4H_2O + 0.97O_2 + 0.3CO + 21.88N_2$$

Problem No. 2

A hydrocarbon fuel represented by $C_{12}H_{26}$ is used as fuel in an IC engine and requires 25 % excess air for complete combustion. Determine

a. The combustion equation
b. The theoretical air – fuel ratio
c. The actual air – fuel ratio
d. The volumetric and gravimetric analysis of the products
e. The molecular weight M and gas constant R of the products
f. The kg of CO_2 formed per kg of fuel
g. % C and %H in the fuel

Solution

Fuel: $C_{12}H_{26}$

Combustion with 100% theoretical air

$$C_{12}H_{26} + aO_2 + a(3.76)N_2 \rightarrow bCO_2 + cH_2O + a(3.76)N_2$$

Carbon balance

$$12 = b$$

Hydrogen balance

$$26 = 2c$$

$$c = 13$$

Oxygen balance

$$2a = 2b + c$$

$$a = b + \frac{c}{2} = 18.5$$

Combustion with excess air e = 0.25

$$C_{12}H_{26} + (1.25)aO_2 + (1.25)a(3.76)N_2 \rightarrow bCO_2 + cH_2O + dO_2 + (1.25)a(3.76)N_2$$

By oxygen balance

$$1.25(2)a = 2b + c + 2d$$

$$d = \frac{1.25(2)a - 2b - c}{2} = 4.625$$

$$C_{12}H_{26} + 23.125O_2 + 86.95N_2 \rightarrow 12CO_2 + 13H_2O + 4.625O_2 + 86.95N_2$$

Gases	Mi	ni	yi	mi	xi	yiMi	yi(%)	xi(%)
CO_2	44	12	0.103	528	0.158	4.529	10.3	15.8
H_2O	18	13	0.112	234	0.070	2.007	11.2	7.0
O_2	32	4.625	0.040	148	0.044	1.270	4.0	4.4
N_2	28	86.95	0.746	2434.6	0.728	20.884	74.6	72.8
total		116.575	1.00	3344.6	1.00	28.691	100.0	100.0

The combustion equation

$$C_{12}H_{26} + 23.125O_2 + 86.95N_2 \rightarrow 12CO_2 + 13H_2O + 4.625O_2 + 86.95N_2$$

The theoretical A/F ratio

$$\left(\frac{A}{F}\right)_t = \frac{a(32) + a(3.76)(28)}{12(12) + 26} = 14.94\ \frac{\text{kg of air}}{\text{kg of } C_{12}H_{26}}$$

The actual A/F ratio

$$\left(\frac{A}{F}\right)_a = (1+e)\left(\frac{A}{F}\right)_t = 14.94\,(1.25) = 18.674\ \frac{\text{kg of air}}{\text{kg of } C_{12}H_{26}}$$

The Volumetric and gravimetric Analysis

Gases	yi(%)	xi(%)
CO_2	10.3	15.8
H_2O	11.2	7.0
O_2	4.0	4.4
N_2	74.6	72.8
total	100.0	100.0

Molecular weight and Gas constant

$$M = \Sigma yiMi = 28.691 \ \frac{kg}{kg_m}$$

$$R = \frac{\overline{R}}{M} = \Sigma xiRi$$

$$R = \frac{8.3143}{28.691} = 0.2898 \ \frac{KJ}{kg - K}$$

Kg of CO_2 per kg of fuel

$$\frac{kg\ \ of\ CO_2}{kg\ \ of\ C_{12}H_{26}} = \frac{528}{12(12)+26} = 3.106$$

% C and % H in the fuel

$$\%C = \frac{12n}{12n+m}\ x\,100\%$$

$$\%C = \frac{12(12)}{12(12)+26}\ x\,100 = 84.7\%$$

$$\%H = \frac{m}{12n+m}\ x\,100\%$$

$$\%H = \frac{26}{12(12)+26}\ x\,100 = 15.3\%$$

Problem No. 3

A fuel represented by C_7H_{16} is oxidized with 20% excess air and the mass of fuel required for combustion is 50 kg/hr. Determine the mass flow rate of the products in kg/hr.

$$\frac{A}{F} = \frac{137.28(1.20)(7+4)}{(84+4)} = 20.592\frac{kg\,of\ air}{kg\,fuel}$$

$$m_p = m_F(20.592) + m_F$$

$$m_p = 50(21.592) = 1079.6\ \ kg/hr$$

The mass analysis of hydrocarbon fuel A is 88.5% Carbon and 11.5% Hydrogen. Another hydrocarbon fuel B requires 6% more air than fuel A for complete combustion. Calculate the mass analysis of Fuel B.

a) C = 84.8%; H = 15.2% c) C = 74.8%; H = 25.2%

b) C = 15.2% ; H = 84.8% d) C = 25.2% ; H = 74.8%

Solution:

Fuel A: C = 0.885 ; H = 0.115

Fuel B: C = ; H =

$(A/F)_B = 1.06(A/F)_A$

$(A/F)_A = 11.44(0.885) + 34.32(0.115) = 14.0712$ kg/kg

$(A/F)_B = 1.06(14.0712) = 14.9155$ kg/kg

For fuel B: H + C = 1

H = (1 – C)

14.9155 = 11.44C + 34.32(1-C)

$$C = \frac{34.32 - 14.9155}{34.32 - 11.44} = 0.8481$$

H = 0.1519

C = 84.8%
H = 15.2%

The analysis of the natural gas showed the following percentages by volume: C_2H_6 = 9%; CH_4 = 90%; CO_2 = 0.2 % and N_2 = 0.8 %. Find the volume of air required per cu,m. of gas if the gas and air are at temperature of 16°C and a pressure of 101.6 KPa.

a) 10.07 b) 14.9 c) 17.1 d) 19.08

solution: basis 100 moles of fuel)

$9C_2H_6 + 90CH_4 + 0.2CO_2 + 0.8N_2 + aO_2 + a(3.76)N_2 \rightarrow$
$bCO_2 + cH_2O + dN_2$

By Carbon balance:
2(9) + 90 + 0.20 = b
b = 108.20

By Hydrogen Balance:
6(9) + 4(90) = 2c
c = 207

By O_2 balance:
0.2 + a = 108.20 + (207/2)
a = 211.5
$n_a = a(1 + 3.76)$
$n_F = 100$
$n_a/n_F = 10.07$

A diesel engine uses a hydrocarbon fuel represented by $C_{12}H_{26}$ and is burned with 30% excess air. The air and fuel is supplied at 1 atm and 25°C. Determine

a. the actual air-fuel Ratio
b. the m^3 of CO_2 formed per kg of fuel if the product temp. is 400°C and a pressure of 1 atm.
c. The M and R of the Products
d. The M and R of the dry flue gas

Combustion with 100% theoretical air (basis 1 mole of fuel)

$C_{12}H_{26} + aO_2 + a(3.76)N_2 \rightarrow bCO_2 + cH_2O + a(3.76)N_2$
a = n + 0.25m = 18.5
b = n = 12
c = 0.5m = 13

Combustion with 30% excess air

$C_{12}H_{26} + 1.30aO_2 + 1.30a(3.76)N_2 \rightarrow bCO_2 + cH_2O + dO_2 + 1.30a(3.76)N_2$

d = ea = 5.55

$$\left(\frac{A}{F}\right)a = \frac{137.28(1+e)(n+0.25m)}{12n+m} = 19.42\ \frac{\text{kg of air}}{\text{kg of fuel}}$$

$$V_{CO_2} = \frac{n\bar{R}T}{P} = \frac{12(8.3143)(400+273)}{101.325} = 662.7\ \text{m}^3\text{ of }CO_2$$

$$\frac{\text{m}^3\text{ of }CO_2}{\text{kg of Fuel}} = \frac{662.7}{12(12)+26} = 3.9$$

n_P = b + c + d + 1.3(18.5)(3.76) = 120.978

$$M = \frac{1}{n}\sum n_i M = \frac{1}{120.978}[12(44)+13(18)+5.55(32)+90.428(28)]$$

M = 28.7

R = 0.2897

$$n_d = n_p - c = 107.978$$

$$M = \frac{1}{n}\sum n_i M = \frac{1}{107.978}[12(44)+5.55(32)+90.428(28)]$$

M = 29.983

R = 0.2773

A fuel consisting 80% $C_{12}H_{26}$ and 20% $C_{14}H_{30}$ is burned with 30% excess air. The flue gas is at atmospheric pressure. Find the minimum exhaust temperature to avoid condensation. (Answer: 50.5°C; 47.45°C)

Assuming that the given fuel analysis is percentages by mass, converting it to molal analysis;

Solution:

Assuming that the given fuel analysis is percentages by mass, converting it to molal analysis;

$$C_{12}H_{26} = \frac{\frac{.8}{170}}{\frac{.8}{170} + \frac{0.2}{198}} = .823$$

$$C_{14}H_{30} = \frac{\frac{.2}{198}}{\frac{.8}{170} + \frac{0.2}{198}} = 0.177$$

Combustion with 100% theoretical air (basis; 100 moles of fuel)

$$82.3C_{12}H_{26} + 17.7C_{14}H_{30} + aO_2 + a(3.76)N_2 \rightarrow bCO_2 + cH_2O + a(3.76)N_2$$

b = 1235.4 ; c = 1335.4; a = 1903.1

Combustion with 30% excess air

$82.3C_{12}H_{26} + 17.7C_{14}H_{30} + (1.3)aO_2 + (1.3)a(3.76)N_2 \rightarrow$

$bCO_2 + cH_2O + dO_2 + (1.30)a(3.76)N_2$

By O_2 balance

d = 570.93

$n_P = b + c + d + 1.30a(3.76) = 12{,}444.0828$

$y_{H2O} = c/n_P = 0.107$

$P_{H2O} = 0.107(101.325) = 10.9$ KPa

From steam table t_{sat} at 10.9 KPa = 47.45°C

Assuming that the given analysis is molal analysis

$80C_{12}H_{26} + 20C_{14}H_{30} + aO_2 + a(3.76)N_2 \rightarrow bCO_2 + cH_2O + a(3.76)N_2$

b = 1240; c = 1340; a = 1910

with excess air:

$80C_{12}H_{26} + 20C_{14}H_{30} + (1.3)aO_2 + (1.3)a(3.76)N_2 \rightarrow$

$bCO_2 + cH_2O + dO_2 + (1.30)a(3.76)N_2$

d = 573

$n_P = 12489.08$

$y_{H2O} = c/n_P = .1073$

$P_{H2O} = 10.9$ KPa ; tsat = 47.45°C

A fuel has the following volumetric analysis: CH_4 = 68% C_2H_6 = 32%. Assume complete combustion with 15% excess air at 101.325 KPa, 21°C wet bulb and 27°C dry bulb. What is the partial pressure of the water vapor in KPa?

Problem *Natural Gas*

A natural gas fuel showed the following percentages by volume: C_2H_6 = 9%; CH_4 = 90%; CO_2 = 0.2 % and N_2 = 0.8 %. If this gas is used as fuel and is burned with 20% excess air for complete combustion, Determine

The Combustion Equation

The theoretical and actual air fuel ratio

The molecular weight and gas constant of the products

The volume of air required per m^3 of natural gas if the gas and air are at temperature of 16°C and a pressure of 101.6 KPa.

Combustion with 100% theoretical air

$$9C_2H_6 + 90CH_4 + 0.2CO_2 + 0.8N_2 + aO_2 + a(3.76)N_2 \rightarrow bCO_2 + cH_2O + dN_2$$

Carbon Balance

$$2(9) + 90 + 0.2 = b$$

$$b = 108.2$$

Hydrogen Balance

$$9(6) + 90(4) = 2c$$

$$c = 207$$

Oxygen Balance

$$2(0.2) + 2a = 2b + c$$

$$a = \frac{2(108.2) + 207 - 2(0.2)}{2}$$

$$a = 211.5$$

Nitrogen Balance

$$2(0.8) + 2(a)(3.76) = 2d$$

$$d = \frac{2(0.8) + 2(211.5)(3.76)}{2}$$

$$d = 796.04$$

Combustion with excess air

$$9C_2H_6 + 90CH_4 + 0.2CO_2 + 0.8N_2 + (1.20)aO_2 + (1.20)a(3.76)N_2 \rightarrow bCO_2 + cH_2O + eO2 + fN_2$$

By oxygen balance

$$2(0.2) + 2(1.20)(211.5) = 2(108.2) + 207 + 2e$$

$$e = \frac{2(0.2) + 2(1.20)(211.5) - 2(108.2) - 207}{2}$$

$$e = 42.3$$

By Nitrogen balance

$$2(0.8) + 2(1.20)(211.5)(3.76) = 2f$$

$$f = \frac{2(0.8) + 2(1.20)(211.5)(3.76)}{2}$$

$$f = 955.088$$

Combustion Equation

$$9C_2H_6 + 90CH_4 + 0.2CO_2 + 0.8N_2 + 253.8O_2 + 954.288N_2 \rightarrow 108.2CO_2 + 207H_2O + 42.3O2 + 955.088N_2$$

Mass of Fuel

$$9(30) + 90(16) + 0.2(44) + 0.8(28) = 1741.2\,kg$$

Mass of air

$$253.8(32) + 954.288(28) = 34841.664\,kg$$

$$\left(\frac{A}{F}\right)_{actual} = \frac{34841.664}{1741.2} = 20.01\,\frac{kg\ of\ air}{kg\ of\ fuel}$$

$$\left(\frac{A}{F}\right)_{actual} = (1+e)\left(\frac{A}{F}\right)_{theoretical}$$

$$\left(\frac{A}{F}\right)_{theoretical} = \frac{20.01}{1.20} = 16.68\ \frac{kg\ of\ air}{kg\ of\ fuel}$$

Moles of Products

$$108.2+207+42.3+955.088=1312.588$$

$$\sum yiMi = M = \frac{108.2(44)+207(18)+42.3(32)+955.088(28)}{1312.588}$$

$$M = 27.9\ \frac{kg}{kg_{mol}}$$

$$R = \frac{8.3143}{27.9} = 0.298\frac{KJ}{kg\text{-}K}$$

$$PV_a = n_a\bar{R}T$$

$$V_a = \frac{(253.8+954.288)(8.3143)(16+273)}{101.6} = 28571.2\,m^3$$

$$PV_F = n_F\bar{R}T$$

$$V_F = \frac{(9+90+.2+0.8)(8.143)(16+273)}{101.6} = 2365\,m^3$$

$$\frac{V_a}{V_F} = \frac{28571.2}{2365} = 12.08$$

Sample Problem (Known Orsat analysis and Fuel type)

A fuel oil $C_{12}H_{26}$ is used in an internal combustion engine and the Orsat analysis are as follows: CO_2 = 12.8% ; O_2 = 3.5%; CO = 0.2% and N_2 = 83.5%. Determine the actual air-fuel ratio and the percent excess air.

Solution: (Basis 100 moles of dry flue gas)

$$aC_{12}H_{26} + bO_2 + b(3.76)N_2 \rightarrow 12.8CO_2 + cH_2O + 0.2CO + 3.5O_2 + 83.5N_2$$

By C balance

12a = 12.8 + 0.2

a = 1.0833

By N_2 Balance

b(3.76) = 83.5

b = 22.207

By H balance

26a = 2c

c = 26(1.0833)/2

c = 14.083

Dividing the equation by a

$$C_{12}H_{26} + 20.5O_2 + 77.08N_2 \rightarrow 11.816CO_2 + 13H_2O + 0.185CO + 3.23O_2 + 77.08N_2$$

$$\left(\frac{A}{F}\right)_a = \frac{20.5(32)+77.08(28)}{12(12)+26} = 16.56\ \frac{\text{kg of air}}{\text{kg of fuel}}$$

Combustion of $C_{12}H_{26}$ with 100% theoretical air

$$C_{12}H_{26}+aO_2+a(3.76)N_2 \rightarrow bCO_2+cH_2O+a(3.76)N_2$$

$$12 = b$$

$$26 = 2c$$

$$c = 13$$

$$2a = 2b + c$$

$$a = 18.5$$

$$\left(\frac{A}{F}\right)_t = \frac{18.5(32)+(18.5)(3.76)(28}{12(12)+26} = 14.94\ \frac{\text{kg of air}}{\text{kg of } C_{12}H_{26}}$$

$$\left(\frac{A}{F}\right)_a = (1+e)\left(\frac{A}{F}\right)_t$$

$$1+e = \frac{\left(\frac{A}{F}\right)_a}{\left(\frac{A}{F}\right)_t}$$

$$e = \frac{\left(\frac{A}{F}\right)_a}{\left(\frac{A}{F}\right)_t} - 1 = 0.108 = 10.8\%$$

Sample Problem (Unknown Fuel – Known Orsat analysis)

An unknown hydrocarbon is used as fuel in a diesel engine, and after an emission test the orsat analysis shows, CO_2 = 12.5% ; CO = 0.3% ; O_2 = 3.1% ; N_2 = 84.1%.Determine

a. the actual air-fuel ratio
b. the percent excess air
c. the fuel analysis by mass

$C_nH_m + aO_2 + a(3.76)N_2 \rightarrow 12.5CO_2 + bH_2O + 0.3CO + 3.1O_2 + 84.1N_2$

By Carbon balance

n = 12.5 + 0.3
n = 12.8

By Hydrogen balance

m = 2b $\rightarrow$ eq. 1

By Oxygen balance

2a = 2(12.5) + b + 0.3 + 2(3.1) $\rightarrow$ eq. 2

By Nitrogen balance

a(3.76) = 84.1
a = 22.367

substituting a to eq. 2

b = 13.234

substituting b to eq. 1

m = 26.47

$$C_{12.8}H_{26.47} + 22.367O_2 + 84.1N_2 \rightarrow 12.5CO_2 + 13.234H_2O + 0.3CO + 3.1O2 + 84.1N_2$$

$$\left(\frac{A}{F}\right)_a = \frac{22.367(32) + 84.1(28)}{12(12.8) + 26.47} = 17.05 \ \frac{\text{kg of air}}{\text{kg of fuel}}$$

Combustion with 100% theoretical air

n = 12.8 ; m = 26.47

$$C_{12.8}H_{26.47} + aO_2 + a(3.76)N_2 \rightarrow bCO_2 + cH_2O + a(3.76)N_2$$

$$12.8 = b$$

$$26.47 = 2c$$

$$c = 13.235$$

$$2a = 2b + c$$

$$a = 19.4175$$

$$\left(\frac{A}{F}\right)_t = \frac{19.4175(32) + 19.4175(3.76)(28}{12(12.8) + 26.47} = 14.8 \ \frac{\text{kg of air}}{\text{kg of } C_{12}H_{26}}$$

$$\left(\frac{A}{F}\right)_a = (1+e)\left(\frac{A}{F}\right)_t$$

$$e = 0.152 = 15.2\,\%$$

$$\%C = \frac{12n}{12n+m} = \frac{12(12.8)}{12(12.8)+26.47} = 85.3\%$$

$$\%H = \frac{m}{12n+m} = \frac{26.47}{12(12.8)+26.47} = 14.7\%$$

Problem (Combustion of Coal Fuel)

A coal fired steam power plant uses that has the following Ultimate Analysis;

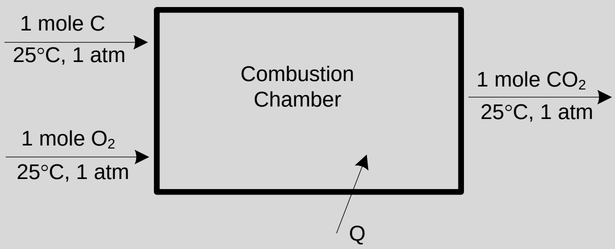
C = 74% ; H_2 = 5% ; O_2 = 6%; S = 1%; N_2 = 1.2%; H_2O = 3.8% and Ash = 9%. If this coal is burned with 30% excess air, determine

a. The combustion equation
b. The actual air – fuel ratio
c. The M and R of the products
d. The Orsat analysis of the products
e. The HHV of coal in KJ/kg

FUEL	Ultimate	Ashless	M	x/M	Ashless	100% theoretical air		With Excess air (x)			PRODUCT ANALYSIS	
	Analysis %	% (By Mass)			% (By Mole)	O2	N2	x (%)	O2	N2		
C	74	81.3	12	6.78	67.5	79.44	298.70	30	103.27	388.3075	WET	ORSAT
H2	5	5.5	2	2.75	27.4	PRODUCTCS		PRODUCTS		M		
O2	6	6.6	32	0.21	2.1	CO2	67.5	CO2	67.47	44	13.2	14.0
N2	1.2	1.3	28	0.05	0.5	H20	29.7	H20	29.66	18	5.8	0
S	1	1.1	32	0.03	0.3	SO2	0.3	SO2	0.34	64	0.1	0.1
H20	3.8	4.2	18	0.23	2.3	N2	299.2	N2	388.8	28	76.2	80.9
Ash	9			10.04	100.0	Total	396.6	O2	23.83	32	4.7	5.0
	100							Total	510.09		100	100
$(A/F)_{theo}$	$(A/F)_{act}$	HHV	M	R								
10.95	14.24	34,339.40	29.75	0.280								

ENTHALPY OF FORMATION

The “Enthalpy of Formation of a compound is the enthalpy at the Arbitrary Reference State (t = 25°C and P = 1 Atm).



Let;
H_R – total enthalpy of Reactants
H_P – total enthalpy of products

From 1st Law

$$Q + H_R = H_P$$

or

$$Q + \sum_R n_i \overline{h_i} = \sum_P n_i \overline{h_i}$$

but the enthalpy of all the reactants (at the reference state) is Zero (for they are all elements)

$$Q = H_P = -393{,}757\,KJ$$

therefore

$$(\bar{h}^\circ f)_{CO_2} = -393{,}757\,\frac{KJ}{kg_{mol}} \rightarrow \text{Enthalpy of formation of } CO_2$$

$$M_{Carbon=}12\,\frac{kg}{kg_m}$$

$$(\bar{h}^\circ f)_{CO_2} = -32{,}813\,\frac{KJ}{kg_C}$$

Note: Negative sign is due to the reaction's being "Exothermic"

For Hydrogen (Vapor)

$$(\bar{h}^\circ f)_{H_2O} = -241{,}971\,\frac{KJ}{kg_{mol}}$$

$$(\bar{h}^\circ f)_{H_2O} = -120{,}985.5\,\frac{KJ}{kg_{H_2}}$$

For Hydrogen (Liquid)

$$(\bar{h}^\circ f)_{H_2O} = -286{,}010\,\frac{KJ}{kg_{mol}}$$

$$(\bar{h}^\circ f)_{H_2O} = -143{,}005\,\frac{KJ}{kg_{H_2}}$$

For Sulfur

$$(\bar{h}^\circ f)_S = -296{,}840\,\frac{KJ}{kg_{mol}}$$

$$(\bar{h}^\circ f)_S = -9{,}276.25\,\frac{KJ}{kg_{Sulfur}}$$

ENTHALPY OF COMBUSTION OR HEATING VALUE

It is the difference between the enthalpies of the products and the reactants at the same temperature T and Pressure P.

$$\bar{h}_{RP} = H_P - H_R \quad \frac{KJ}{kg_{mol}}$$

$$\bar{h}_{\underset{R}{RP}} = \underset{P}{\Sigma} n_j\left[\bar{h}_{\circ f} + (\bar{h}^\circ - \bar{h}^\circ_{298})\right] - \Sigma n_i\left[\bar{h}_{\circ f} + (\bar{h}^\circ - \bar{h}^\circ_{298})\right]_i \quad \frac{KJ}{kg_{mol}}$$

$$\bar{h}_{RP} = \frac{(H_P - H_R)}{M} \quad \frac{KJ}{kg}$$

where

M - molecular weight

Example:

Determine the heating value of $C_{12}H_{26}$ (liquid) when burned with 25% excess air. Air and fuel enters at 25°C and products leaves at 900 K. Assume P =101 KPa.

Combustion Equation

P = 101 KPa

$$C_{12}H_{26} + 23.13O_2 + 86.95N_2 \rightarrow 12CO_2 + 13H_2O + 4.63O_2 + 86.95N_2$$

T = (25 + 273) = 298   T = 900 K

$$C_{12}H_{26} + 23.13O_2 + 86.95N_2 \rightarrow 12CO_2 + 13H_2O + 4.63O_2 + 86.95N_2$$

LOWER HEATING VALUE (H_2O in the products is VAPOR

$$H_R = 1(394{,}199)\frac{KJ}{kg_{mol}}$$

$$H_P = 12(-393{,}757 + 28{,}041) + 13(-241{,}971 + 21{,}924) + 4.63(0 + 19{,}246) + 86.95(0 + 18{,}221)$$

$$H_P = \text{-}5{,}575{,}874.30\frac{KJ}{kg_{mol}}$$

$$LHV = \text{-}5{,}181{,}675.30\frac{KJ}{kg_{mol}}$$

$$LHV = \frac{\text{-}5{,}181{,}675.30}{170} = 30{,}480.44\frac{KJ}{kg_{C_{12}H_{26}}})$$

HIGHER HEATING VALUE (H_2O in the products is LIQUID)

$$H_R = 1(394{,}199)\frac{KJ}{kg_{mol}}$$

$$H_P = 12(-393{,}757 + 28{,}041) + 13(\text{-}286{,}010.0 + 21{,}924) + 4.63(0 + 19{,}246) + 86.95(0 + 18{,}221)$$

$$H_P = \text{-}6{,}148{,}381.30\frac{KJ}{kg_{mol}}$$

$$HHV = \text{-}5{,}754{,}182.30\frac{KJ}{kg_{mol}}$$

$$HHV = \frac{\text{-}5{,}754{,}182.30}{170} = 33{,}848.13\frac{KJ}{kg_{C_{12}H_{26}}}$$

FROM BUREAU OF STANDARD

HEATING VALUE

Heating Value - is the energy released by fuel when it is completely burned and the products of combustion are cooled to the original fuel temperature.
Higher Heating Value (HHV) - is the heating value obtained when the water in the products is liquid.
Lower Heating Value (LHV) - is the heating value obtained when the water in the products is vapor.

For Solid Fuels

$$HHV = 33{,}820C + 144{,}212\left(H_2 - \frac{O_2}{8}\right) + 9{,}304S \ \frac{KJ}{kg}$$

where: C, H_2, O_2, and S are in decimals from the ultimate analysis

For Coal and Oils with the absence of Ultimate Analysis

$$\left(\frac{A}{F}\right)_t = \frac{HHV}{3{,}041} \text{ kg of air/kg of fuel}$$

HHV = 31,405C + 141 647H KJ/kg

For Liquid Fuels

HHV = 43,385 + 93(Be - 10)   KJ/kg
Be - degrees Baume

For Gasoline

HHV = 41,160 + 93 (°API) KJ/kg
LHV = 38,639 + 93 (°API)   KJ/kg

For Kerosene

HHV = 41,943 + 93 (°API) KJ/kg
LHV = 39,035 + 93 (°API)   KJKkg

For Fuel Oils

HHV = 41,130 + 139.6(°API)   KJ/kg
LHV = 38,105 + 139.6(°API)   KJ/kg

API - American Petroleum Institute

For Fuel Oils (From Bureau of Standard Formula)

HHV = 51,716 - 8793.8 $(S)^2$ KJ/kg
LHV = HHV - Q_L KJ/kg
Q_L = 2,442.7($9H_2$)   KJ/kg
H_2 = 0.26 - 0.15(S)   kg of H_2/ kg of fuel

$$S = \frac{141.5}{131.5 + °API}$$

$$S = \frac{140}{130 + °Be}$$

Where:

S @ t = S - 0.0007(t-15.56)
S - specific gravity of fuel oil at 15.56 °C
H_2 - hydrogen content of fuel oil, Kg H/Kg fuel

Q_L - heat required to evaporate and superheat the water vapor formed by the combustion of hydrogen in the fuel,KJ/kg
S @ t - specific gravity of fuel oil at any temperature t
Oxygen Bomb Calorimeter - instrument used in measuring heating value of solid and liquid fuels.
Gas Calorimeter - instrument used for measuring heating value of gaseous fuels.

SECOND LAW OF THERMODYNAMICS

Whenever energy is transferred, the level of energy cannot be conserved and some energy must be permanently reduced to a lower level.
When this is combined with the first law of thermodynamics, the law of energy conservation, the statement becomes:
Whenever energy is transferred, energy must be conserved, but the level of energy cannot be conserved and some energy must be permanently reduced to a lower level.

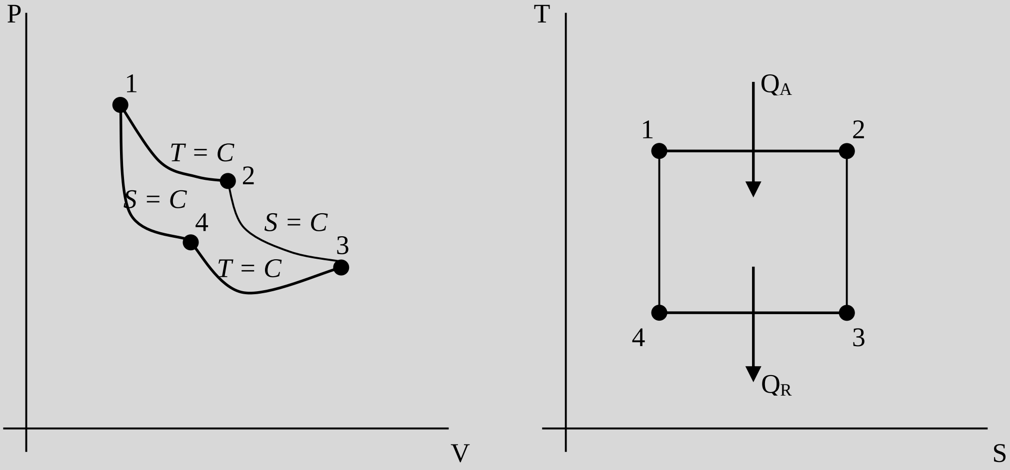
Kelvin-Planck statement of the second law:
No cyclic process is possible whose sole result is the flow of heat from a single heat reservoir and the performance of an equivalent amount of work.
For a system undergoing a cycle: The net heat is equal to the network.

$$\oint dW = \oint dQ \quad \text{or } W = \Sigma Q$$

W - net work
ΣQ - net heat

CARNOT CYCLE

Nicolas Léonard Sadi Carnot (1796-1832), French physicist and military engineer, son of Lazare Nicolas Marguerite Carnot, born in Paris, and educated at the École Polytechnique. In 1824 he described his conception of the perfect engine, the so-called Carnot engine, in which all available energy is utilized. He discovered that heat cannot pass from a colder to a warmer body, and that the efficiency of an engine depends upon the amount of heat it is able to utilize. These discoveries led to the development of the Carnot Cycle, which later became the basis for the second law of thermodynamics.
A. Carnot Engine Cycle
Processes:
1 to 2 - Heat Addition (T = C)
2 to 3 - Expansion (S = C)
3 to 4 - Heat Rejection (T = C)
4 to 1 - Compression (S = C)



Heat Added (T = C)

$$Q_A = T_H(\Delta S) \rightarrow 1$$

Heat Rejected (T = C)

$$Q_R = T_L(\Delta S) \rightarrow 2$$

$$\Delta S = S_2 - S_1 = S_4 - S_3 \quad \rightarrow 3$$

Net Work

$$W = \Sigma Q = Q_A - Q_R \quad \rightarrow 4$$

$$W = (T_H - T_L)(\Delta S) \quad \rightarrow 5$$

Thermal Efficiency

$$e = \frac{W}{Q_A} \times 100\% \rightarrow 6$$

$$e = \frac{Q_A - Q_R}{Q_A} \times 100\% \rightarrow 7$$

$$e = \left[1 - \frac{Q_R}{Q_A} \times\right] 100\% \rightarrow 8$$

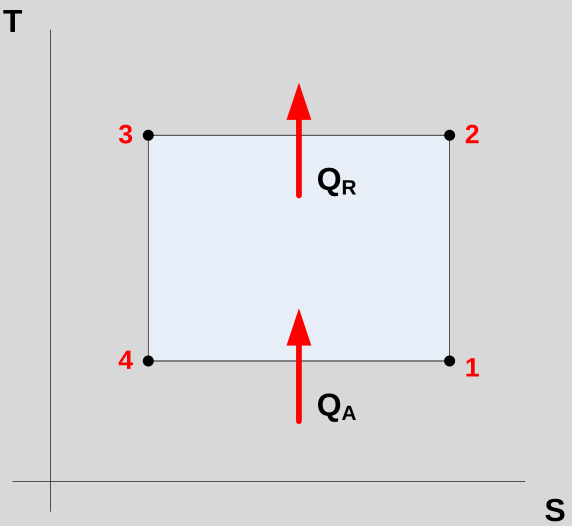
Eq. 5 to and Eq. 1 to Eq. 6

$$e = \frac{T_H - T_L}{T_H} \times 100\% \rightarrow 9$$

B. Carnot Refrigeration Cycle (Reversed Carnot Cycle)

Processes:

1 to 2 - Compression (S =C)
2 to 3 - Heat Rejection (T = C)
3 to 4 - Expansion (S = C)
4 to 1 - Heat Addition (T = C)



Heat Added (T=C)

$$Q_A = T_L(\Delta S) \rightarrow 1$$

Heat Rejected (T=C)

$$Q_R = T_H(\Delta S) \rightarrow 2$$

$$\Delta S = S_1 - S_4 = S_2 - S_3 \rightarrow 3$$

Net Work

$$W = \Sigma Q \rightarrow 4$$

$$W = Q_R - Q_A \rightarrow 5$$

$$W = (T_H - T_L)(\Delta S) \rightarrow 6$$

COEFFICIENT OF PERFORMANCE

$$COP = \frac{Q_A}{W} \rightarrow 6$$

$$COP = \frac{Q_A}{Q_R - Q_A} \rightarrow 7$$

$$COP = \frac{T_L}{T_H - T_L} \rightarrow 8$$

Carnot Heat Pump:
A heat pump uses the same components as the refrigerator but its purpose is to reject heat at high energy level.

Performance Factor

$$PF = \frac{Q_R}{W} \rightarrow 11$$

$$PF = \frac{Q_R}{Q_R - Q_A} \rightarrow 12$$

$$PF = \frac{T_H}{T_H - T_L} \rightarrow 13$$

$$PF = COP + 1 \rightarrow 14$$

TON OF REFRIGERATION
It is the heat equivalent to the melting of 1 ton (2000 lb) of water ice at 0°C into liquid at 0°C in 24 hours.

$$TR = \frac{2000(144)}{24} = 12{,}000\ \frac{BTU}{hr}$$

$$TR = 211\ \frac{KJ}{min}$$

where 144 BTU/lb is the latent heat of fusion

Problem No. 1
A Carnot engine operates between temperature levels of 397°C and 7°C and rejects 20 KJ/min to the environment. The total network output of the engine is used to drive heat pump which is supplied with heat from the environment at 7°C and rejects heat to a home at 40°C. Determine:
a) the network delivered by the engine in KJ/min
b) the heat supplied to the heat pump in KJ/min
c) the overall COP for the combined devices which is defined as the energy rejected to the home divided by the initial energy supplied to the engine.

Problem No. 2
A reversed Carnot cycle is used for refrigeration and rejects 1000 KW of heat at 340 K while receiving heat at 250 K. Determine:
a) the COP
b) the power required in KW
c) the refrigerating capacity in Tons

1. A Carnot engine operating between 775 K and 305 K produces 54 KJ of work. Determine the change of entropy during heat addition. (ΔS_{12}= 0.115 KJ/K)

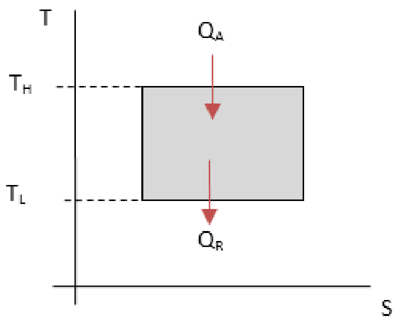
$$e = \frac{T_H - T_L}{T_H} = \frac{775 - 305}{775} = 0.606$$

$$e = \frac{W}{Q_A}$$

$$Q_A = \frac{54}{0.606} = 89\,KJ$$

$$Q_A = T_H \Delta S$$

$$\Delta S = \frac{89}{775} = 0.115\ \frac{KJ}{°K}$$



2. A Carnot engine operates between temperatures of 1000 K and 300 K The engine operates at 2000 RPM and develops 200 KW. The total engine displacement is such that the mean effective pressure is 300 KPa. Determine:

a) the thermal efficiency (71%)

b) the heat added (286 KW)

c) the total engine displacement in m3/cycle (0.02 m3/cycle)

$$e=\frac{1000-300}{1000}=0.70$$

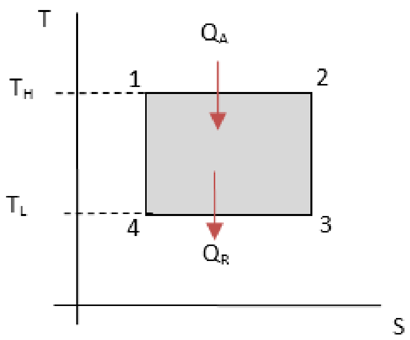
$$W=eQ_A$$

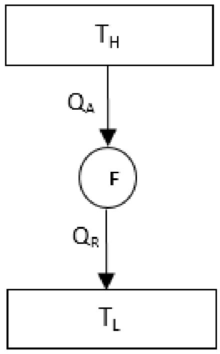
$$Q_A=\frac{200}{0.70}=286\ \ KW$$

$$W=Pm(V_D)$$

$$V_D=\frac{200}{300}=0.67\ \frac{m^3}{sec}x\frac{min}{2000rev}x\frac{60sec}{min}x\frac{rev}{cycle}$$

$$VD=0.02\ m^3 cycle$$





3. A Carnot refrigerator rejects 2500 KJ of heat at 353 K while using 1100 KJ of work. Find

a) the cycle low temperature

b) the COP

c) the heat absorbed or refrigerating capacity

$$W=Q_R-Q_A$$

$$1100=2500-Q_A$$

$$Q_A=1400\ \ KJ$$

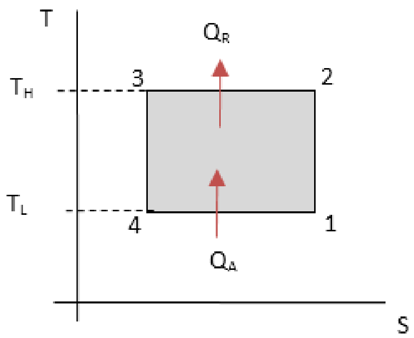
$$COP=\frac{Q_A}{W}=\frac{T_L}{T_H-T_L}$$

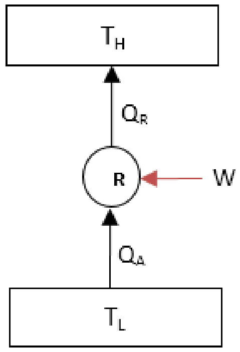
$$\frac{1400}{1100}=\frac{T_L}{353-T_L}$$

$$1100T_L=494{,}200-1400T_L$$

$$T_L=197.7°K$$

$$COP=1.2727$$





SAMPLE PROBLEMS

1. A Carnot engine operating between 775 ˚K and 305˚K produces 54 KJ of work. Determine the change of entropy during heat addition. (ΔS_{12}= -0.12 KJ/K)

2. A Carnot engine operates between temperature reservoirs of 817°C and 25°C and rejects 25 KW to the low temperature reservoir. The Carnot engine drives the compressor of an ideal vapor compression refrigerator, which operates within pressure limits of 190 KPa and 1200 Kpa. The refrigerant is ammonia. Determine the COP and the refrigerant flow rate.(4; 14.64 kg/min)

3. A Carnot engine operates between temperatures of 1000 K and 300°K The engine operates at 200 RPM and develops 200 KW. The total engine displacement is such that the mean effective pressure is 300 Kpa. Determine:

a) the thermal efficiency (71%)

b) the heat added (286 KW)

c) the total engine displacement in m^3/cycle (0.02 m^3/cycle)

4. A Carnot engine operating between 775 ˚K and 305˚K produces 54 KJ of work. Determine the change of entropy during heat addition. (ΔS_{12}= -0.12 KJ/K)

5. A Carnot engine produces 25 KW while operating between temperature limits of 1000 K and 300 K. Determine (a) the heat supplied in KW (b) the heat rejected in KW

6. A Carnot heat engine rejects 230 KJ of heat at 25°C. The net cycle work is 375 KJ. Determine the cycle thermal efficiency and the cycle high temperature.

7. A Carnot refrigerator operates between temperature limits of -5°C and 30°C. The power consumed is 4 KW and the heat absorbed is 30 KJ/kg. Determine; (a) the COP (b) the refrigerant flow rate

8. A non polluting power plant can be constructed using the temperature difference in the ocean. At the surface of the ocean in tropical climates, the average water temperature year round is 30°C. At a depth of 305 m, the temperature is 4.5°C. Determine the maximum thermal efficiency of such a power plant. (8.4%)

9. A heat is used to heat a house in the winter months. When the average outside temperature is 0°C and the indoor temperature is 23°C, the heat loss from the house is 20 KW. Determine the minimum power required to heat the heat pump.

10. A Carnot heat pump is being considered for home heating in a location where the outside temperature may as low as -35°C. The expected COP for the heat pump is 1.50. To what temperature could this unit provide?

INTERNAL COMBUSTION ENGINE CYCLE

- Air Standard Cycle
- Otto Cycle
- Diesel Cycle
- Dual Cycle

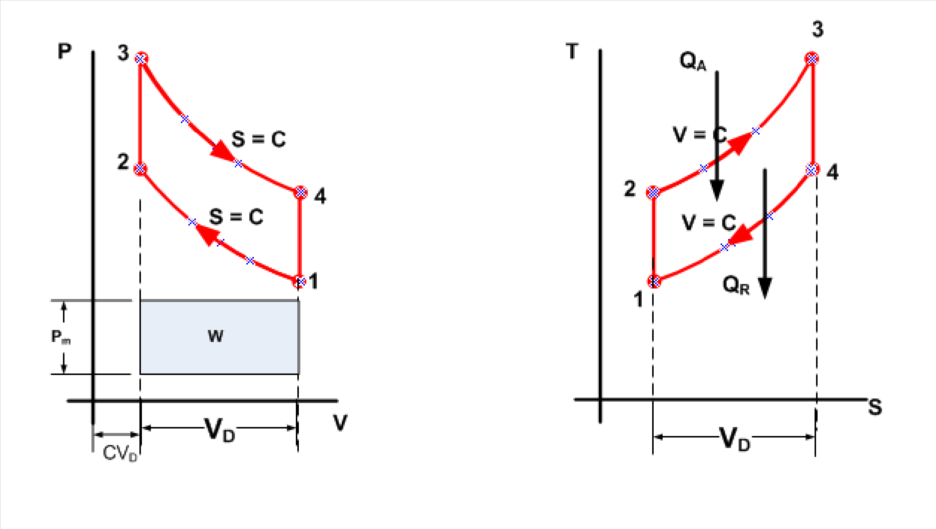
AIR STANDARD OTTO CYCLE

Otto, Nikolaus August
Born:June 10, 1832, Holzhausen, Nassau
Died: Jan. 26, 1891, Cologne

German engineer who developed the four-stroke internal-combustion engine, which offered the first practical alternative to the steam engine as a power source.
Otto built his first gasoline-powered engine in 1861. Three years later he formed a partnership with the German industrialist Eugen Langen, and together they developed an improved engine that won a gold medal at the Paris Exposition of 1867.
In 1876 Otto built an internal-combustion engine utilizing the four-stroke cycle (four strokes of the piston for each ignition). The four-stroke cycle was patented in 1862 by the French engineer Alphonse Beau de Rochas, but since Otto was the first to build an engine based upon this principle, it is commonly known as the Otto cycle. Because of its reliability, its efficiency, and its relative quietness, Otto's engine was an immediate success. More than 30,000 of them were built during the next 10 years, but in 1886 Otto's patent was revoked when Beau de Rochas' earlier patent was brought to light.

Processes

1 to 2 – Isentropic Compression (S = C)
2 to 3 – Constant Volume Heat Addition (V = C)
3 to 4 – Isentropic Expansion (S = C)
4 to 1 – Constant volume Heat Rejection (V = C)



Compression Ratio (r)

$$r = \frac{V_1}{V_2} = \frac{V_4}{V_3} \rightarrow 1$$

$V_1 = V_4$ and $V_2 = V_3$

V_1 - volume at bottom dead center (BDC)

V_2 – volume at top dead center (TDC)(clearance volume)

Displacement Volume (V_D)

$V_D = V_1 - V_2 \quad \rightarrow$ Eq. 2

Percent Clearance (C)

$$C = \frac{V_2}{V_D}$$ $\rightarrow$ Eq. 3

$$r = \frac{1 + C}{C}$$ $\rightarrow$ Eq. 3'

Heat Added (Q_A)

At V = C ; $Q = mCv(\Delta T)$

$Q_A = mC_v(T_3 - T_2) \quad \rightarrow$ Eq. 4

Heat Rejected (Q_R)

$Q_R = mC_v(T_4 - T_1) \quad \rightarrow$ Eq. 5

Net Work (W)

$W = \Sigma Q$

$W = Q_A - Q_R \quad \rightarrow$ Eq. 6

P,V and T Relations

At point 1 to 2 (S = C)

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} = (r)^{k-1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_2 = T_1(r)^{k-1} \rightarrow \text{Eq. 7}$$

At point 2 to 3 (V = C)

$$\frac{T_3}{T_2} = \frac{P_3}{P_2} \rightarrow \text{Eq. 8}$$

At point 3 to 4 (S = C)

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{k-1} = \left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}} = (r)^{k-1}$$

$$T_3 = T_4 (r)^{k-1} \rightarrow \text{Eq.9}$$

At point 4 to 1 (V = C)

$$\frac{T_4}{T_1} = \frac{P_4}{P_1} \rightarrow \text{Eq. 10}$$

Entropy Change

a) ΔS during Heat Addition

$$S_3 - S_2 = mC_v \ln \frac{T_3}{T_2} \rightarrow \text{Eq. 11}$$

b) ΔS during Heat Rejection

$$S_1 - S_4 = mC_v \ln \frac{T_1}{T_4} \rightarrow \text{Eq. 12}$$

$$S_1 - S_4 = -(S_3 - S_2) \rightarrow \text{Eq. 13}$$

Thermal Efficiency

$$e = \frac{W}{Q_A} \text{ x } 100\% \rightarrow \text{Eq. 14}$$

$$e = \frac{Q_A - Q_R}{Q_A} \text{ x } 100\% \rightarrow \text{Eq. 15}$$

$$e = \left[1 - \frac{Q_R}{Q_A}\right] \text{x } 100\% \rightarrow \text{Eq. 16}$$

$$e = \left[1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}\right] \text{x } 100\% \rightarrow \text{Eq.17}$$

$$e = \left[1 - \frac{1}{(r)^{k-1}}\right] \text{x } 100\% \rightarrow \text{Eq. 18}$$

Mean Effective Pressure

$$P_m = \frac{W}{V_D} \text{ KPa} \rightarrow \text{Eq. 19}$$

where:

Pm – mean effective pressure, KPa
W – Net Work KJ, KJ/kg, KW
V_D – Displacement Volume m^3, m^3/kg, m^3/sec

Displacement Volume

$V_D = V_1 - V_2\ m^3 \rightarrow$ Eq. 20

$V_D = \upsilon_1 - \upsilon_2\ m^3/kg \rightarrow$ Eq.21

Note:

For Cold Air Standar: $k = 1.4$

For Hot Air Standard: $k = 1.3$

AIR STANDARD DIESEL CYCLE

Diesel, Rudolf (Christian Karl)

Born: March 18, 1858, Paris, France

Died: September 29, 1913, at sea in the English Channel

German thermal engineer who invented the internal-combustion engine that bears his name. He was also a distinguished connoisseur of the arts, a linguist, and a social theorist.

Diesel, the son of German-born parents, grew up in Paris until the family was deported to England in 1870 following the outbreak of the Franco-German War. From London Diesel was sent to Augsburg, his father's native town, to continue his schooling. There and later at the Technische Hochschule (Technical High School) in Munich he established a brilliant scholastic record in fields of engineering. At Munich he was a protégé of the refrigeration engineer Carl von Linde, whose Paris firm he joined in 1880.

Diesel devoted much of his time to the self-imposed task of developing an internal combustion engine that would approach the theoretical efficiency of the Carnot cycle. For a time he experimented with an expansion engine using ammonia. About 1890, in which year he moved to a new post with the Linde firm in Berlin, he conceived the idea for the diesel engine . He obtained a German development patent in 1892 and the following year published a description of his engine under the title Theorie und Konstruktion eines rationellen Wäremotors (Theory and Construction of a Rational Heat Motor). With support from the Maschinenfabrik Augsburg and the Krupp firms, he produced a series of increasingly successful models, culminating in his demonstration in 1897 of a 25-horsepower, four-stroke, single vertical cylinder compression engine. The high efficiency of Diesel's engine, together with its comparative simplicity of design, made it an immediate commercial success, and royalty fees brought great wealth to its inventor.

Diesel disappeared from the deck of the mail steamer Dresden en route to London and was assumed to have drowned.

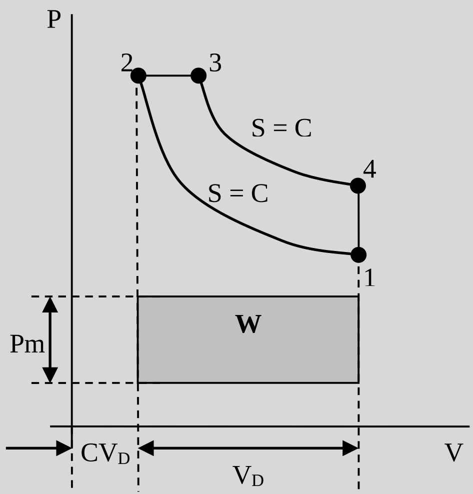
Processes:

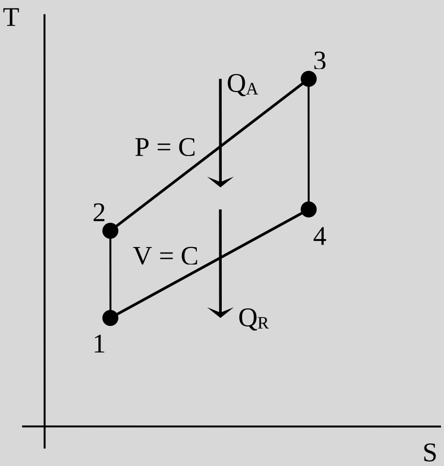
1 to 2 – Isentropic Compression (S = C)

2 to 3 – Constant Pressure Heat Addition (P = C)

3 to 4 – Isentropic Expansion (S = C)

4 to 1 – Constant Volume Heat Rejection (V = C)





Compression Ratio

$$r = \frac{V_1}{V_2} = \frac{V_4}{V_2} \quad \rightarrow 1$$

$$\frac{V_1}{V_2} \neq \frac{V_4}{V_3}$$

Cut-Off Ratio(r_c)

$$r_c = \frac{V_3}{V_2} \quad \rightarrow 2$$

Percent Clearance

$$C = \frac{V_2}{V_D} \quad \rightarrow 3$$

$$r = \frac{1+C}{C} \quad \rightarrow 4$$

Displacement Volume (V_D)

$V_D = V_1 - V_2 \quad \rightarrow$ Eq. 5

Heat Added (Q_A)

At $P = C$; $Q = mC_p(\Delta T)$

$Q_A = mC_p(T_3 - T_2) \quad \rightarrow$ Eq. 6

$Q_A = mkC_v(T_3 - T_2) \quad \rightarrow$ Eq. 7

Heat Rejected (Q_R)

$Q_R = mC_v(T_4 - T_1) \quad \rightarrow$ Eq. 8

Net Work (W)

$W = \Sigma Q$

$W = Q_A - Q_R \quad \rightarrow$ Eq. 9

$W = mkC_v(T_3 - T_2) - mC_v(T_4 - T_1)$

$W = mC_v[k(T_3 - T_2) - (T_4 - T_1)] \rightarrow$ Eq. 10

P,V and T Relations

At point 1 to 2 (S = C)

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} = (r)^{k-1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_2 = T_1(r)^{k-1} \rightarrow \text{Eq. } 11$$

At point 2 to 3 (P = C)

$$\frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c \rightarrow \text{Eq. } 12$$

$$T_3 = T_1(r)^{k-1}(r_c) \rightarrow \text{Eq. } 13$$

At point 3 to 4 (S = C)

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{k-1} = \left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}}$$

$$T_4 = T_1 (r)^{k-1} (r_c) \left(\frac{V_3}{V_4}\right)^{k-1}$$

$$T_4 = T_1 \frac{V_1^{k-1}}{V_2^{k-1}} \frac{V_3}{V_2} \frac{V_3^{k-1}}{V_4^{k-1}}$$

$$T_4 = T_1 (r_c)^k \rightarrow \text{Eq.14}$$

At point 4 to 1 (V = C)

$$\frac{T_4}{T_1} = \frac{P_4}{P_1} \quad \rightarrow \text{ Eq. 15}$$

Entropy Change

a) ΔS during Heat Addition

$$S_3 – S_2 = mC_p \ln \frac{T_3}{T_2} \quad \rightarrow \text{ Eq. 16}$$

b) ΔS during Heat Rejection

$$S_1 - S_4 = mC_v \ln \frac{T_1}{T_4} \quad \rightarrow \text{ Eq. 17}$$

$$S_1 - S_4 = -(S_3 – S_2) \quad \rightarrow \text{ Eq. 18}$$

Thermal Efficiency

$$e = \frac{W}{Q_A} \text{ x } 100\% \quad \rightarrow \text{ Eq. 19}$$

$$e = \frac{Q_A - Q_R}{Q_A} \text{ x } 100\% \quad \rightarrow \text{ Eq. 20}$$

$$e = \left[1 - \frac{Q_R}{Q_A}\right] \text{x } 100\% \quad \rightarrow \text{ Eq. 21}$$

$$e = \left[1 - \frac{(T_4 - T_1)}{k(T_3 - T_2)}\right] \text{x } 100\% \quad \rightarrow \text{ Eq.22}$$

Substituting Eq. 11, Eq. 13 and Eq. 14 to Eq. 22

$$e = \left[1 - \frac{1}{(r)^{k-1}} \left(\frac{r_c^{\,k} - 1}{k(r_c - 1)}\right)\right] \text{ x } 100\% \quad \rightarrow \text{ Eq. 23}$$

Mean Effective Pressure

$$P_m = \frac{W}{V_D} \text{ KPa} \quad \rightarrow \text{ Eq. 24}$$

where:

Pm – mean effective pressure, KPa
W – Net Work KJ, KJ/kg, KW
V_D – Displacement Volume m^3, m^3/kg, m^3/sec

Displacement Volume

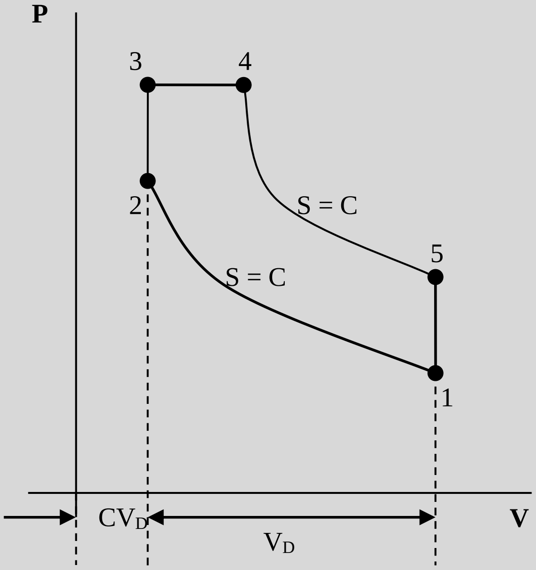
$$V_D = V_1 – V_2 \text{ m}^3 \quad \rightarrow \text{ Eq. 25}$$

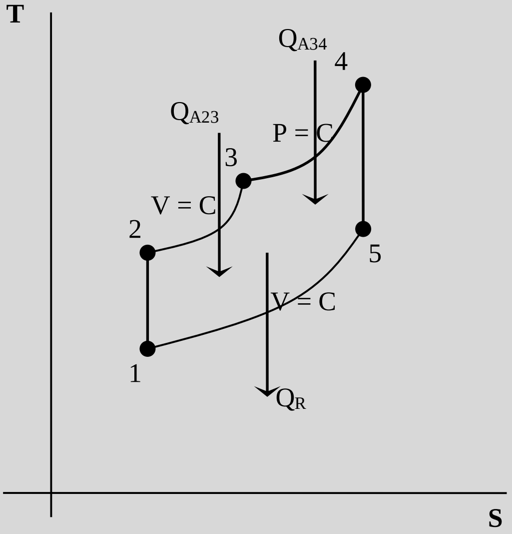
$$V_D = \upsilon_1 - \upsilon_2 \text{ m}^3/\text{kg} \quad \rightarrow \text{ Eq.26}$$

AIR STANDARD DUAL CYCLE

Processes:

1 to 2 – Isentropic Compression (S = C)
2 to 3 – Constant Volume Heat Addition Q_{23} (V = C)
3 to 4 – Constant Pressure Heat Addition Q_{34} (P = C)
4 to 5 – Isentropic Expansion (S = C)
5 to 1 - Constant Volume Heat Rejection (V = C)





Compression Ratio

$$r = \frac{V_1}{V_2} \rightarrow \text{Eq. 1}$$

$V_1 = V_5$ and $V_2 = V_3$

Cut-Off Ratio

$$r_c = \frac{V_4}{V_3} \rightarrow \text{Eq. 2}$$

Pressure Ratio

$$r_p = \frac{P_3}{P_2} \rightarrow \text{Eq. 3}$$

$P_3 = P_4$

Percent Clearance

$$C = \frac{V_2}{V_D} \rightarrow \text{Eq. 4}$$

$$r = \frac{1+C}{C} \rightarrow \text{Eq. 5}$$

Displacement Volume

$V_D = V_1 – V_2$ m^3 $\rightarrow$ Eq. 6
$V_D = \upsilon_1 - \upsilon_2$ $\rightarrow$ Eq. 7

P, V, and T Relations

At point 1 to 2 (S = C)

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} = (r)^{k-1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$T_2 = T_1(r)^{k-1} \rightarrow \text{Eq.} 8$$

At point 2 to 3 (V = C)

$$\frac{T_3}{T_2} = \frac{P_3}{P_2} = r_P \rightarrow \text{Eq. 9}$$

$$T_3 = T_1(r)^{k-1}(r_P) \rightarrow \text{Eq. 10}$$

At point 3 to 4 (P = C)

$$\frac{T_4}{T_3} = \left(\frac{V_4}{V_3}\right) = r_c$$

$$T_4 = T_1(r)^{k-1}(r_P)(r_c) \rightarrow \text{Eq.11}$$

At point 4 to 5 (S = C)

$$\frac{T_5}{T_4} = \left(\frac{V_4}{V_5}\right)^{k-1}$$

$$T_5 = T_1\frac{V_1^{\,k-1}}{V_2^{\,k-1}}(r_P)\frac{V_4}{V_2}\left(\frac{V_4^{\,k-1}}{V_1^{\,k-1}}\right) \rightarrow \text{Eq. 12}$$

$$T_5 = T_1(r_c)^k(r_P) \rightarrow \text{Eq.13}$$

At 5 to 1 (V = C)

$$\frac{T_5}{T_1} = \frac{P_5}{P_1} \rightarrow \text{Eq. 14}$$

Entropy change

a) At 2 to 3 (V = C)

$$S_3 - S_2 = mC_v\ln\frac{T_3}{T_2} \rightarrow \text{Eq. 15}$$

b) At 3 to 4 (P = C)

$$S_4 - S_3 = mC_P\ln\frac{T_4}{T_3} \rightarrow \text{Eq. 16}$$

$$S_4 - S_2 = (S_3 - S_2) + (S_4 - S_3) \rightarrow \text{Eq. 17}$$

c) At 5 to 1 (V = C)

$$S_1 - S_5 = mC_v\ln\frac{T_1}{T_5} \rightarrow \text{Eq. 18}$$

Heat Added

$Q_A = Q_{A23} + Q_{A34} \rightarrow$ Eq. 19

$Q_{A23} = mC_v(T_3 - T_2) \rightarrow$ Eq. 20

$Q_{A34} = mC_p(T_4 - T_3) = mkC_v(T_4 - T_3) \rightarrow$ Eq. 21

$Q_A = mC_v[(T_3 - T_2) + k(T_4 - T_3)] \rightarrow$ Eq. 22

Heat Rejected

$Q_R = mC_v(T_5 - T_1) \rightarrow$ Eq. 23

Net Work

$W = (Q_A - Q_R) \rightarrow$ Eq. 24

$W = mC_v[(T_3 - T_2) + k(T_4 - T_3) - (T_5 - T_1)] \rightarrow$ Eq. 25

Thermal Efficiency

$$e = \frac{W}{Q_A} \times 100\% \rightarrow \text{Eq. 26}$$

$$e = \frac{Q_A - Q_R}{Q_A} \text{ x } 100\% \quad \rightarrow \text{ Eq. 27}$$

$$e = \left[1 - \frac{Q_R}{Q_A}\right] \text{x } 100\% \quad \rightarrow \text{ Eq. 28}$$

$$e = \left[1 - \frac{(T_5 - T_1)}{(T_3 - T_2) + k(T_4 - T_3)}\right] \text{x } 100\% \quad \rightarrow \text{ Eq.29}$$

Substituting Eq. 8, Eq. 10 and Eq. 11and Eq. 13 to Eq. 29

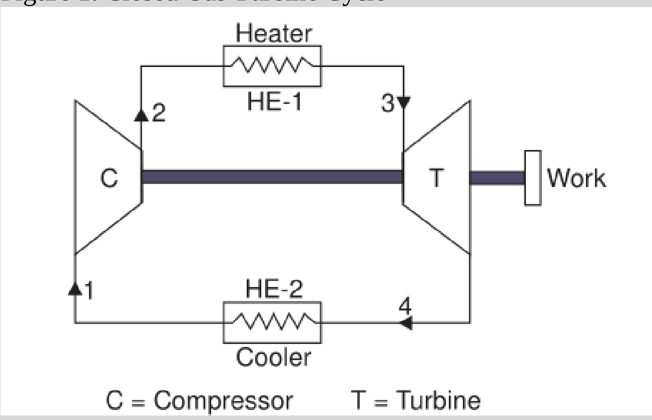
$$e = \left[1 - \frac{1}{(r)^{k-1}}\left(\frac{r_p r_c^{\ k} - 1}{(r_p - 1) + k r_p (r_c - 1)}\right)\right] \text{ x } 100\% \quad \rightarrow \text{ Eq. 30}$$

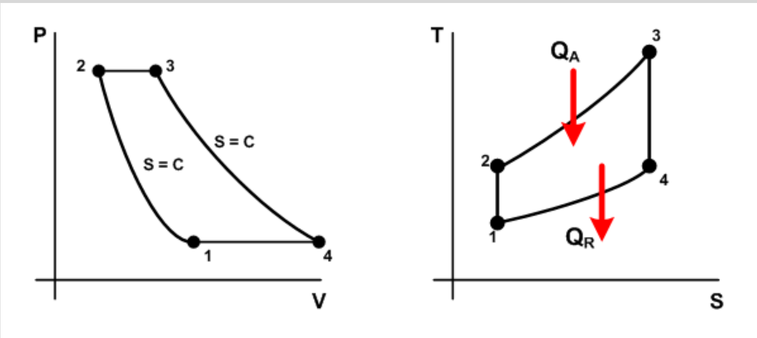
BRAYTON CYCLE

Basic Components
Air Compressor
Heater
Gas Turbine
Cooler

Processes
1 to 2 – Isentropic Compression
2 to 3 – Isobaric Heat Addition
3 to 4 – Isentropic Expansion
4 to 1 – Isobaric Heat Rejection

Figure 1: Closed Gas Turbine Cycle





Pressure Ratio

$$r_p = \frac{P_2}{P_1} = \frac{P_3}{P_4} \rightarrow \text{Eq.1}$$

Compressor Work

$$Wc = mC_p(T_2 - T_1) \rightarrow \text{Eq. 2}$$

$$T_2 = T_1\left(r_p\right)^{\frac{k-1}{k}} \rightarrow \text{Eq.3}$$

$$T_3 = T_4\left(r_p\right)^{\frac{k-1}{k}} \rightarrow \text{Eq.4}$$

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

Heater

$$Q_A = mC_p(T_3 - T_2) \rightarrow \text{Eq.5}$$

Turbine

$$Wt = mC_p(T_3 - T_4) \rightarrow \text{Eq. 6}$$

Cooler

$$Q_R = mC_p(T_4 - T_1) \rightarrow \text{Eq.7}$$

Net Work

$$W = Q_A - Q_R \rightarrow \text{Eq.8}$$

Thermal Efficiency

$$e = \frac{W}{Q_A} \text{x}100\% \rightarrow \text{Eq.9}$$

$$e = \frac{Q_A - Q_R}{Q_A} \text{x}100\% \rightarrow \text{Eq.10}$$

$$e = \left[1 - \frac{Q_R}{Q_A}\right] \text{x}100\% \rightarrow \text{Eq.11}$$

$$e = \left[1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}\right] \text{x}100\% \rightarrow \text{Eq.12}$$

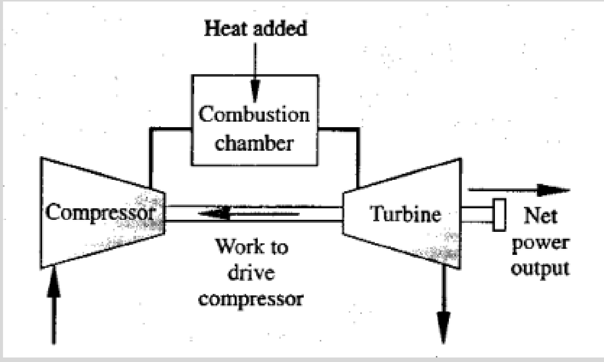
Substituting Eq. 3 and Eq.4 to Eq.12

$$e = \left[1 - \frac{1}{\left(r_p\right)^{\frac{k-1}{k}}}\right] \text{x}100\% \rightarrow \text{Eq.13}$$

Work Ratio

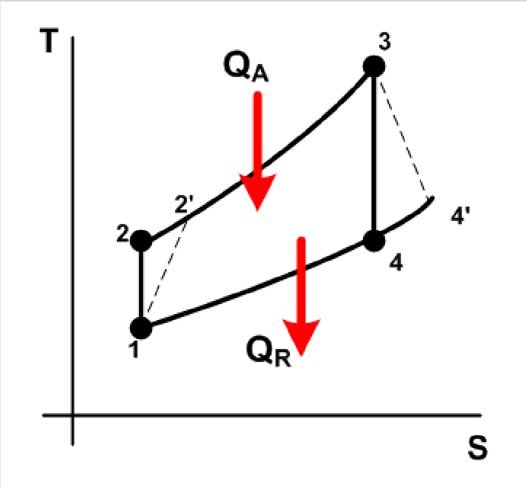
$$WR = \frac{\text{Net Work}}{\text{Turbine Work}}$$

Figure 2: Open Gas Turbine Cycle



$$W = W_{Turbine} - W_{Compressor}$$

ACTUAL BRAYTON CYCLE



Turbine Efficiency

$$\eta_{Turbine} = \frac{\text{Actual Turbine Work}}{\text{Ideal Turbine Work}} x100\%$$

$$W_{t'} = mC_p(T_3 - T_{4'}) \rightarrow \text{Actual Work}$$

$$W_t = mC_p(T_3 - T_4) \rightarrow \text{Ideal Work}$$

$$\eta_{Turbine} = \frac{(T_3 - T_{4'})}{(T_3 - T_4)} x100\%$$

Compressor Efficiency

$$\eta_{Compressor} = \frac{\text{Ideal Compressor Work}}{\text{Actual Compressor Work}} x100\%$$

$$W_c = mC_p(T_2 - T_1) \rightarrow \text{Actual Work}$$

$$W_{c'} = mC_p(T_{2'} - T_1) \rightarrow \text{Ideal Work}$$

$$\eta_{Compressor} = \frac{(T_2 - T_1)}{(T_{2'} - T_1)} x100\%$$

Actual Thermal efficiency

$$e_{t'} = \frac{\text{Net Work}}{\text{Heat added}} x100\%$$

Problem

In the air standard dual cycle, the isentropic compression starts at 100 KPa and 300□K. The compression ratio is 13, the maximum temperature is 2750°K; the maximum pressure is 6894 KPa. Determine:

a. the work in KJ/kg
b. the heat added in KJ/kg
c. the thermal efficiency
d. the mean effective pressure in KPa
e. the entropy change during the heat addition processes
f. the percent clearance c

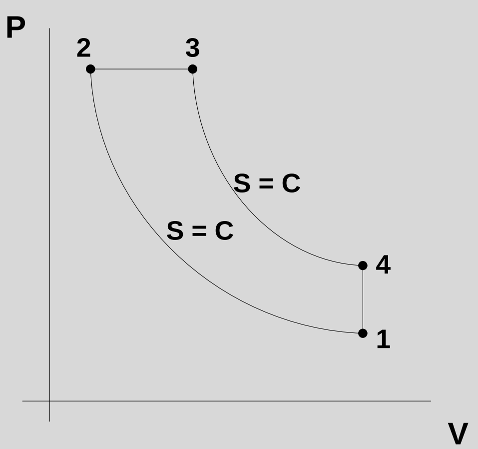
Solution:

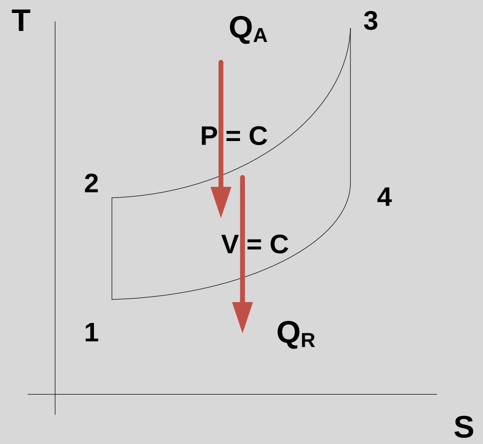
DUAL CYCLE									
P1	100	T1	300	V1	0.861	QA	1705.27	k	1.4
P2	3626.78	T2	836.95	V2	0.0662	QR	665.09	k-1	0.4
P3	6894	T3	1590.92	V3	0.0662	W	1040.18	(k-1)/k	0.286
P4	6894	T4	2750	V4	0.114	e	61.00	R	0.287
P5	408.99	T5	1226.96	V5	0.861	VD	0.795	Cp	1.0045
m	1	c (%)	8.33	r	13	Pm	1308.78	Cv	0.7175
				rc	1.729	(S5 - S1)	-1.011	(S3 - S2)	0.461
				rp	1.90	(S4 - S2)	1.011	(S4 - S3)	0.550

Problem No. 1

In the air standard diesel cycle, the air is compressed isentropically from 26°C and 105 KPa to 3700 KPa. The entropy change during heat rejection is -0.6939 KJ/kg-K. Determine

a) the heat added per kg
b) the thermal efficiency
c) the maximum temperature
d) the temperature at the start of heat rejection





Given:

$T_1 = 26 + 273 = 299$ K
$P_1 = 105$ KPa
$P_2 = P_3 = 3700$ KPa
$S_1 - S_4 = -0.6939$ KJ/kg-K
$S_3 - S_2 = 0.6939$ KJ/kg-K

$$Q_R = C_V(T_4 - T_1)$$
$$Q_R = 349.76 \text{ KJ/kg}$$
$$W = Q_A - Q_R = 477.34 \text{ KJ/kg}$$
$$e = \frac{W}{Q_A} \times 100\% = 57.71\%$$

$$\frac{T_2}{T_1} = \left[\frac{P_2}{P_1}\right]^{\frac{k-1}{k}}$$

$$T_2 = 827.3\,K$$

$$S_3 - S_2 = \Delta S = mC_p \ln \frac{T_3}{T_2}$$

$$0.6939 = (1.0045)\ln \frac{T_3}{827.3}$$

$$e^{\frac{0.6939}{1.0045}} = \frac{T_3}{827.3}$$

$$T_3 = 1650.71\ K$$

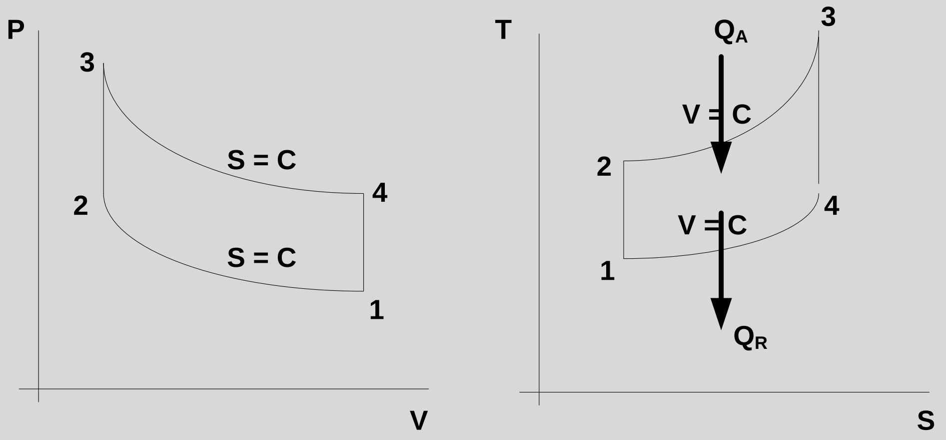
$$Q_A = C_p(T_3 - T_2) = 827.1\ KJ/kg$$

$$S_4 - S_1 = C_v \ln \frac{T_4}{T_1}$$

$$e^{\frac{0.6939}{0.7175}} = \frac{T_4}{299}$$

$$T_4 = 786.47\ K$$

Problem No. 2
For an ideal Otto engine with 17% clearance and an initial pressure of 93 Kpa, determine the pressure at the end of compression. If the pressure at the end of constant volume heating is 3448 KPa, what is the mean effective pressure.



Given:

$c = 0.17$
$P_1 = 93$ KPa
$P_3 = 3448$ KPa

$$r = \frac{1+c}{c} = \frac{1.17}{.17}$$
$$r = 6.88$$
$$r = \frac{V_1}{V_2} = \frac{V_4}{V_3}$$
$$\frac{P_2}{P_1} = r^k$$
$$P_2 = 93(6.88)^{1.4} = 1383.91 \text{ KPa}$$
$$e = \left[1 - \frac{1}{r^{k-1}}\right] = 0.54 = 54\%$$
$$T_2 = T_1(r)^{k-1} = 2.163T_1 \rightarrow \text{eq.1}$$
$$T_3 = T_2\left(\frac{P_3}{P_2}\right) = 5.39T_1 \rightarrow \text{eq.2}$$
$$T_4 = \frac{T_3}{(r)^{k-1}} = T_3(r)^{1-k} = 2.49T_1 \rightarrow \text{eq.3}$$
$$V_D = \upsilon_1 - \upsilon_2 \rightarrow \text{eq.4}$$
$$r = \frac{\upsilon_1}{\upsilon_2}$$
$$\upsilon_1 = r\upsilon_2 \rightarrow \text{eq.5}$$
$$V_D = r\upsilon_2 - \upsilon_2$$
$$V_D = \upsilon_2(r-1) \rightarrow \text{eq.6}$$
$$\upsilon_2 = \frac{RT_2}{P_2} = \frac{0.287(2.163T_1)}{1383.91} = 0.00045T_1 \rightarrow \text{eq.7}$$
$$V_D = 0.00045T_1(6.88-1)$$
$$V_D = 0.0026T_1 \rightarrow \text{eq.8}$$

$$e = \frac{W}{Q_A}$$
$$W = eQ_A \rightarrow \text{eq.9}$$
$$Q_A = C_v(T_3 - T_2)$$
$$Q_A = 0.7175(5.39T_1 - 2.163T_1)$$
$$Q_A = 0.7175T1(5.39 - 2.163)$$
$$Q_A = 2.315T_1 \rightarrow \text{eq.10}$$
$$W = 0.54(2.315T_1)$$
$$W = 1.25T_1 \rightarrow \text{eq.11}$$
$$P_m = \frac{W}{V_D} = \frac{1.25T_1}{0.0026T_1}$$
$$P_m = 480.9 \text{ KPa}$$

Problem No. 3

Air at 300 K and 100 kPa enters into the compressor of a gas turbine engine that operates on a Brayton cycle. The air mass fl ow rate is 5 kg/s and the maximum air temperature in the engine is 1200 K. If the pressure ratio of the cycle is rp = 4,

a. Find the efficiency of the engine, assuming ideal compression and expansion.

b. Find the net power

c. What are the exit temperatures from the turbine and the compressor if $\eta_{Compressor} = \eta_{Turbine} = 60.95\%$

given

$$T1 = 300\,K$$

$$P1 = 100\,KPa$$

$$T3 = 1200\,K$$

$$rp = 4$$

$$rp = \frac{P2}{P1} = \frac{P3}{P4}$$

Problem No. 4

An air standard diesel cycle has a compression ratio of 18 and the heat transferred is 1800 KJ/kg. At the beginning of the compression process the pressure is 100 KPa and the temperature is 15°C. For 1kg of air, determine

The entropy change during isobaric heat addition

The temperature at the start of heat rejection

The work in KJ

The thermal efficiency

The mean effective pressure in KPa

For Air

k = 1.4

R = 0.287 KJ/kg-K

$$T_2 = T_1(r)^{k-1} = 915°K$$

$$P_2 = P_1(r)^k = 100(18)^{1.4} = 5719.8\ \text{KPa}$$

$$Q_A = mC_p(T_3 - T_2)$$

$$T_3 = \frac{Q_A}{mC_p} + T_2 = 2707.1°K$$

$$S_3 - S_2 = mC_p \ln\frac{T_3}{T_2} = 1.0894\ \frac{KJ}{K}$$

$$S_1 - S_4 = -(S_3 - S_2) = mC_v \ln\frac{T_1}{T_4}$$

$$T_4 = 1314.61°K$$

$$Q_R = mC_v(T_4 - T_1) = 736.6\ KJ$$

$$e = \frac{Q_A - Q_R}{Q_A} = 59.1\%$$

$$W = Q_A - Q_R = 1063.4\ KJ$$

$$Pm = \frac{W}{V_D}$$

$$V_D = V_1 - V_2$$

$$V_1 = \frac{mRT_1}{P_1}\ ;\ V_2 = \frac{mRT_2}{P_2}$$

$$V_D = 0.781\ m^3$$

$$P_m = 1362.2\ KPa$$

Problem No. 5

An ideal dual combustion cycle operates on 0.454 kg of air. At the beginning of compression air is at 97 KPa, 316°K. Let r_p = 1.5, r_c= 1.6 and r = 11. Determine

the percent clearance (C = 10%)

the heat added, heat rejected and the net cycle work

the thermal efficiency

$$r = \frac{1+C}{C}$$

a)

$$C = \frac{1}{r-1} = 10\,\%$$

b) $Q_A = Q_{A1} + Q_{A2}$

$Q_{A1} = mCv(T_3 – T_2)$

$$T_2 = T_1(r)^{k-1} = 825\ °K$$

$$P_2 = P_1(r)^k = 2{,}784\ KPa$$

$$T_3 = T_2(r_p) = 1238^\circ K$$

Q_{A1} = 0.454(0.7175)(1238 – 825) = 135 KJ

$Q_{A2} = mCp(T_4 – T_3)$

$$r_c = \frac{T_4}{T_3} = 1.6$$

$$T_4 = 1.6T_3$$

$$T_4 = 1981°K$$

Q_{A2} = 0.454(1.0045(1981 – 1238) = 339 KJ

$Q = Q_{A1} + Q_{A2}$ = 474 KJ

$$e = \left[1 - \frac{1}{(r)^{k-1}}\left(\frac{r_P {r_c}^k - 1}{(r_P - 1) + kr_P(r_c - 1)}\right)\right] \ x\ 100\%$$

e = 58.7 %

Problem No. 6

An air standard Otto cycle has a compression ratio of 8 and has air conditions at the beginning of compression of 100 KPa and 25°C. The heat added is 1400 KJ/kg. Determine:

the thermal efficiency (56.5%)

the mean effective pressure (1057 KPa)

the percent clearance c (14.3%)

Given:

r = 8; P1 = 100 KPa ; T1 = 298 °K; Q_A = 1400 KJ/kg

Solution:

a)
$$e = \left[1 - \frac{1}{(r)^{k-1}}\right] x\,100\%$$

e = 56.5%

b *)W = eQ_A = 0.565(1400) = 791 KJ/kg*

$$Pm = \frac{W}{V_D}$$

$$V_D = \upsilon_1 - \upsilon_2$$

$$V_D = \upsilon_1\left(1 - \frac{1}{r}\right)$$

$$V_D = \frac{RT_1}{P_1}\left(1 - \frac{1}{r}\right) = 0.748\ m^3/kg$$

Pm = 1057 KPa

c)

$$r = \frac{1+C}{C}$$

$$C = \frac{1}{r-1}$$

C = 14.3%

Problem No. 7

An engine operates on the air standard Otto cycle. The cycle work is 900 KJ/kg, the maximum cycle temperature is 3000°C and the temperature at the end of isentropic compression is 600°C. Determine the engine's compression ratio and the thermal efficiency.(r = 6.35; e = 52.26%)

Given: $W = 900\ KJ/kg;\ T_3 = 3273°K;\ T_2 = 873°K$

$Q_A = C_v(T_3 - T_2) = 0.7175(3273 - 873) = 1722\ KJ/kg$

$e = W/Q_A$

$e = 52.26\%$

$$r = \left(\frac{1}{1-e}\right)^{\frac{1}{k-1}}$$

$r = 6.36$

Problem No. 8

An air standard diesel cycle has a compression ratio of 20 and a cut-off ratio of 3. Inlet pressure and temperature are 100 KPa and 27°C. Determine:

the heat added in KJ/kg (1998 KJ/kg)

the net work in KJ/kg (1178 KJ/kg)

the thermal efficiency (61%)

Given: $r = 20;\quad r_c = 3\quad;\ P_1 = 100\ KPa\ ;\ T_1 = 300°C$

Solution:

$$r = \frac{V_1}{V_2}$$

$$r^k = \frac{P_2}{P_1}$$

$$P_2 = 100(20)^{1.4} = 6229\ KPa$$

$$P_2 = P_3 = 6229\ KPa$$

$$\frac{T_2}{T_1} = (r)^{k-1}$$

$$T_2 = 300(20)^{1.4-1} = 994.3^\circ K$$

$$\frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c$$

$$T_3 = 994.3(3) = 2983^\circ K$$

$$\upsilon_3 = \frac{RT_3}{P_3} = \frac{0.287(2983)}{6229} = 0.14\ m^3$$

$$\upsilon_1 = \upsilon_4 = \frac{RT_1}{P_1} = \frac{0.287(300)}{100} = 0.861\ m^3$$

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{k-1}$$

$$T_4 = 2983\left(\frac{0.14}{0.861}\right)^{1.4-1} = 1442.45^\circ K$$

$Q_A = C_p(T_3 - T_2) = 1.0045(2983 - 994.3)$

$Q_A = 1997.65\ KJ/kg$

$Q_R = C_v(T_4 - T_1) = 0.7175(1442.45 - 300)$

$Q_R = 820\ KJ/kg$

$W = Q_A - Q_R = (1997.65 - 820)$

$W = 1177.65\ KJ/kg$

$$e = \frac{W}{Q_A}\ x\,100\%$$

$$e = \frac{1177.65}{1997.65}\ x\,100\%$$

$$e = 59\ \%$$

$$e = \left[1 - \frac{1}{(r)^{k-1}}\left(\frac{r_c{}^k - 1}{k(r_c - 1)}\right)\right]$$

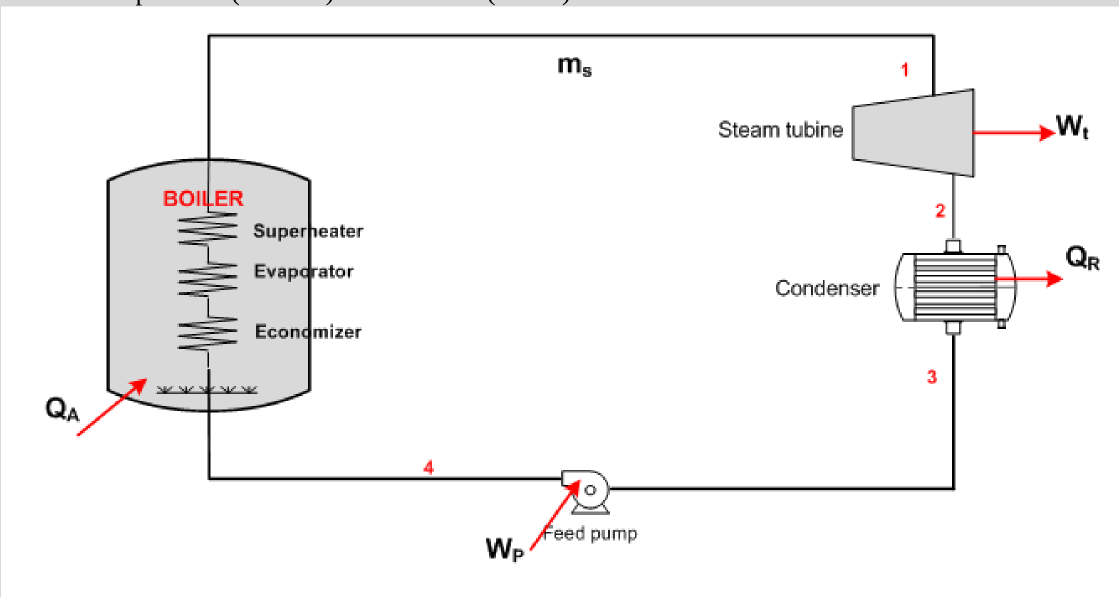
$$e = \left[1 - \frac{1}{(20)^{1.4-1}}\left(\frac{(3)^{1.4}-1}{1.4(3-1)}\right)\right] \text{ x } 100\%$$

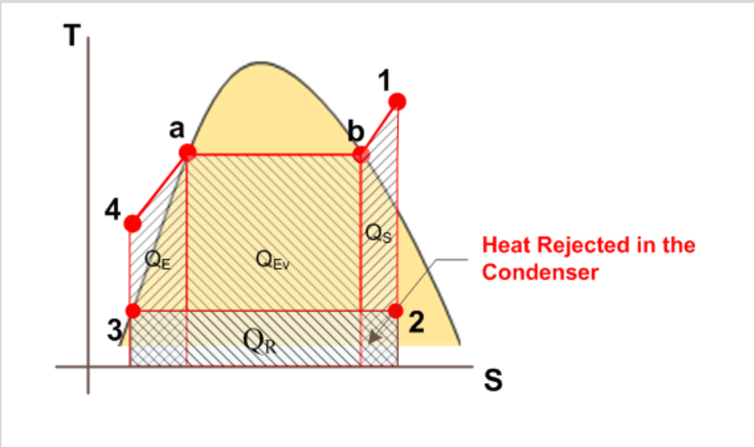
e = 60.6 %

VAPOR POWER CYCLE

Rankine Cycle

Processes:
1 to 2 - Constant entropy (Isentropic) expansion (Turbine)
2 to 3 - Constant pressure (Isobaric) heat rejection (Condenser)
3 to 4 - Constant entropy (Isentropic) compression (Pump)
4 to 1 - Constant pressure (Isobaric) heat addition (Boiler)





$Q_A = Q_E + Q_{Ev} + Q_s$

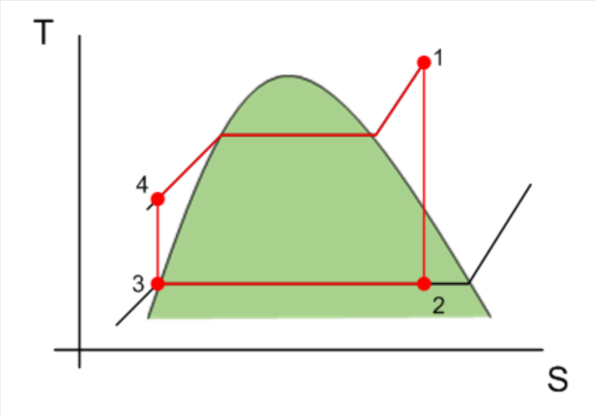
Q_E - heat required by the Economizer

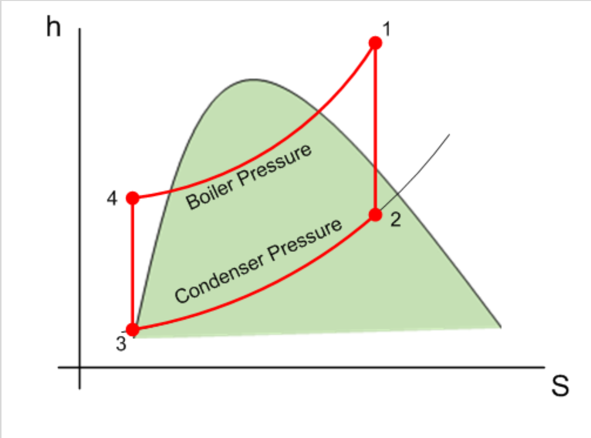
Q_{Ev} - heat required by the Evaporator

Q_s - heat required by the Superheater

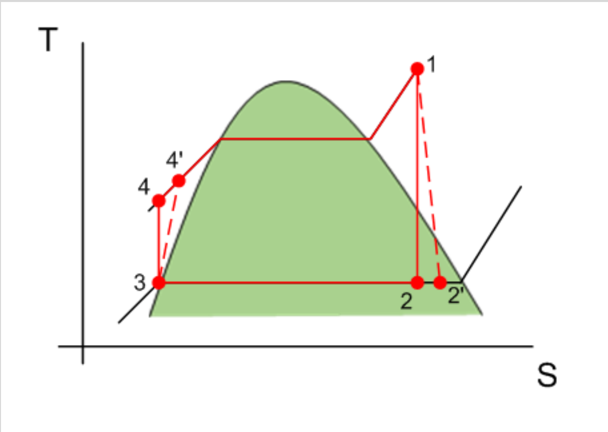
Q_R - heat rejected by steam in the condenser

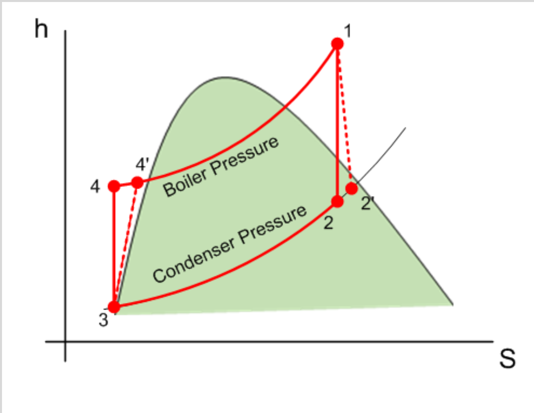
Ideal Cycle





Actual Cycle





A. System: Turbine

Ideal Cycle

$W_t = (h_1 - h_2)$KJ/kg

$W_t = m_s(h_1 - h_2)$ KW

Actual Cycle

$W_{t'} = (h_1 - h_{2'})$KJ/kg

$W_{t'} = m_s(h_1 - h_{2'})$ KW

where:

m_s is the steam flow rate in kg/sec

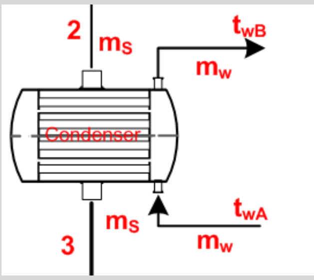
W_t - ideal turbine work

$W_{t'}$ - actual turbine work

Turbine Efficiency

$$e_{Turbine} = \frac{W_{t'}}{W_t} \times 100\%$$

B. System: Condenser



m_w - cooling water flow rate that enters and leaves the condenser in kg/sec

Ideal Cycle

$Q_R = (h_2 - h_3)$ KJ/kg

$Q_R = m_s(h_2 - h_3)$ KW

$Q_R = m_w\, C_{pw}\, (t_{wB}-t_{wA})$KW

Actual Cycle

$Q_R = (h_{2'}-h_3)$ KJ/kg
$Q_R = m_s(h_{2'}-h_3)$ KW
$Q_R = m_w\, C_{pw}\,(t_{wB}-t_{wA})$KW
where:Q_R - heat rejected in the condenser
m_w - cooling water flow rate in kg/sec
C_{pw} - specific heat of cooling water KJ/kg-°C or KJ/kg-°K
C_{pw} = 4.187 KJ/kg-°C or KJ/kg-°K (average value)
t_{wA} - inlet cooling water temperature,°C
t_{wB} - outlet cooling water temperature,°C

C. System: Pump

Ideal Cycle
$W_P = (h_4-h_3)$ KJ/kg
$W_P = m_s(h_4-h_3)$ KW

Actual Cycle
$W_{P'} = (h_{4'}-h_3)$ KJ/kg
$W_{P'} = m_s(h_{4'}-h_3)$ KW

Pump Efficiency

$$e_{pump} = \frac{W_p}{W_{p'}} \text{ x } 100\%$$

D. System: Boiler or Steam Generator

Ideal Cycle
$Q_A =(h_1-h_4)$ KJ/kg
$Q_A =m_s(h_1-h_4)$ KW

Actual Cycle
$Q_A =(h_1-h_{4'})$ KJ/kg
$Q_A = m_s(h_1-h_{4'})$ KW

Boiler or Steam Generator Efficiency

$$e_{Boiler} = \frac{Q_A}{Q_s} \text{ x } 100\%$$

Q_A = Heat absorbed by boiler or steam generator, KW
Q_S = Heat supplied to boiler or steam generator, KW
$Q_A = m_s(h_1-h_4)$ KW (for the ideal cycle)
$Q_A = m_s(h_1-h_{4'})$ KW (for the actual cycle)

If the heat supplied to boiler came from the burning of fuel or products of combustion, the heat supplied Q_S is:
$Q_S = m_f$(HV) KW
where: m_f - is the fuel consumption in kg/sec
HV - heating value of fuel in KJ/kg

Heating Value (HV) - is the energy released by fuel when it is completely burned and the products of combustion are cooled to the original fuel temperature.

STEAM RATE

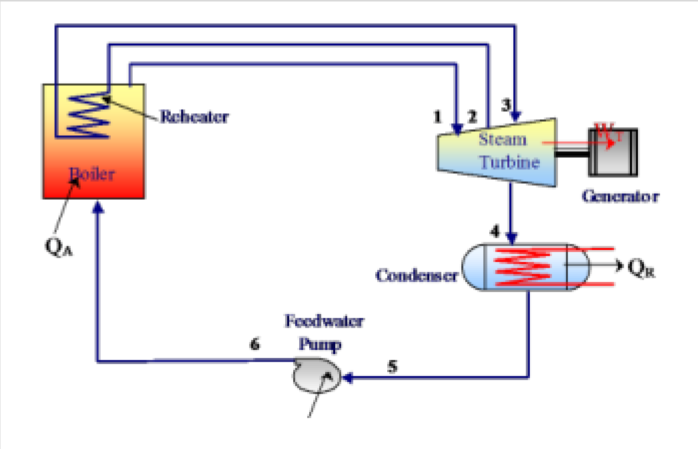
$$SR = \frac{\text{Steam Flow Rate}}{\text{Power (KW) Produced}} \; \frac{\text{kg}}{\text{KW - hr}}$$

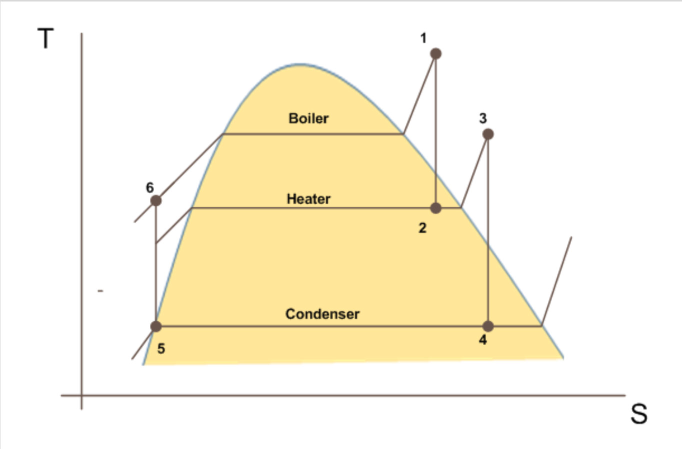
HEAT RATE

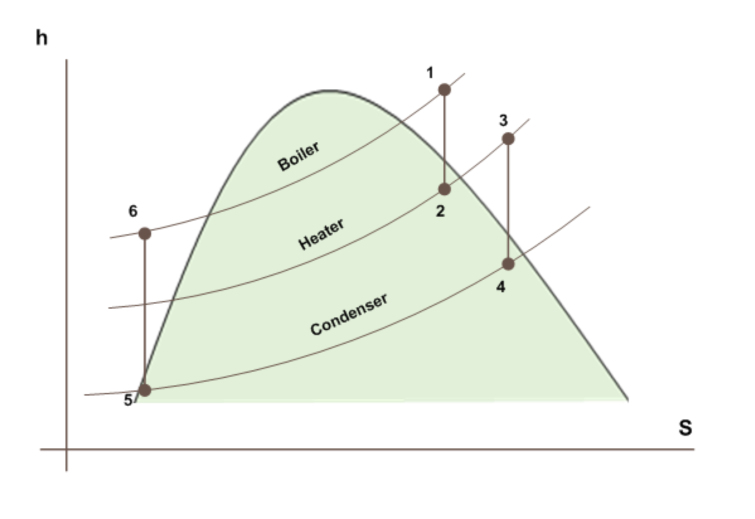
$$HR = \frac{\text{Heat Supplied}}{\text{Power (KW) Produced}} \; \frac{\text{KJ}}{\text{KW - hr}}$$

Reheat Cycle

For a steam power plant operating on a reheat cycle, after partial expansion of the steam in the turbine, the steam flows back to a heat exchanger called the "re-heater" to heat the steam almost the same to its initial temperature.

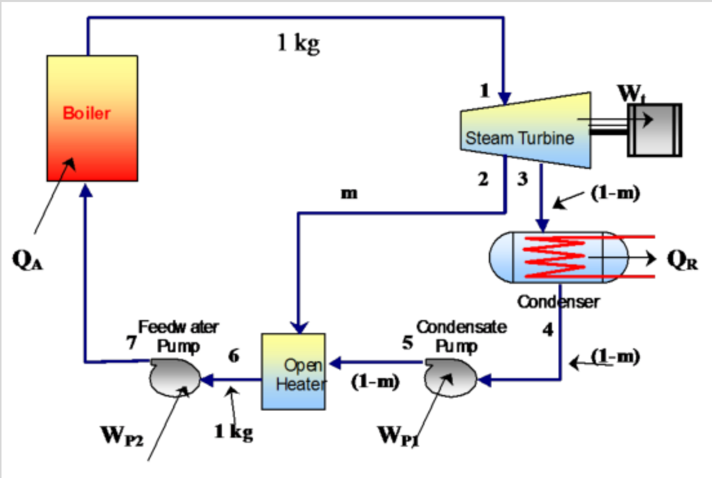


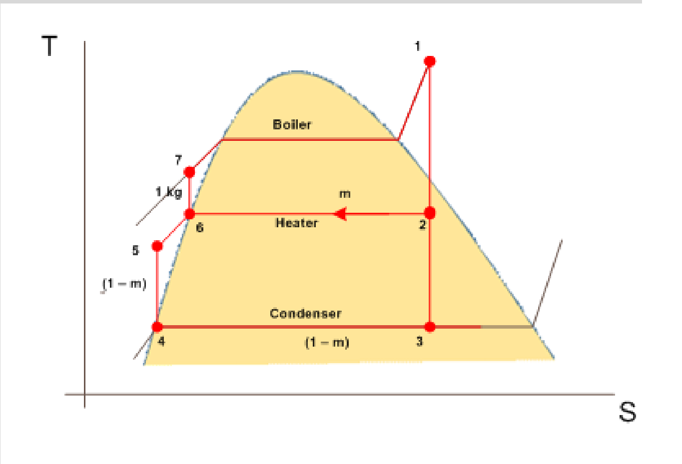


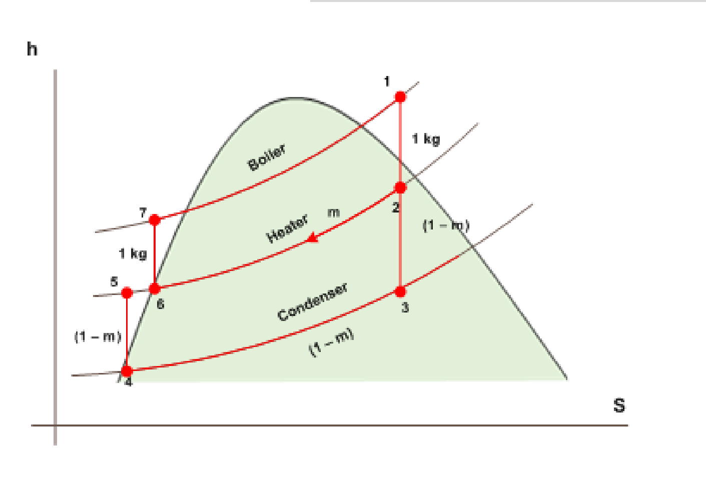


Regenerative Cycle

For a regenerative cycle steam power plant some of the steam were taken from the turbine before it reached the condenser and used to heat the feed-water with no extra energy input and the latent heat of vaporization would not be lost from the system in the condenser. Ideally the water leaving the feed-water heater is saturated liquid at the heater pressure and the heater is a direct-contact type. If the system uses a shell-and-tube heat exchanger the extracted steam doesn't mixed with the feed-water and the drains is send back to the previous heat exchanger, also a separate drain pump may be employed.





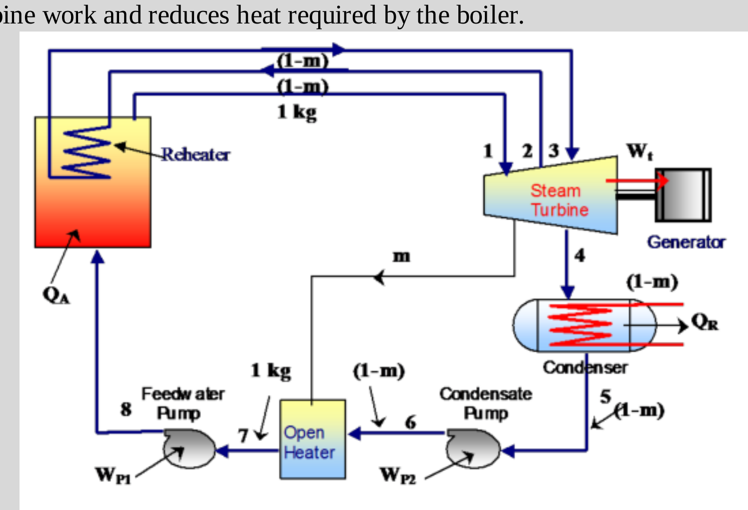


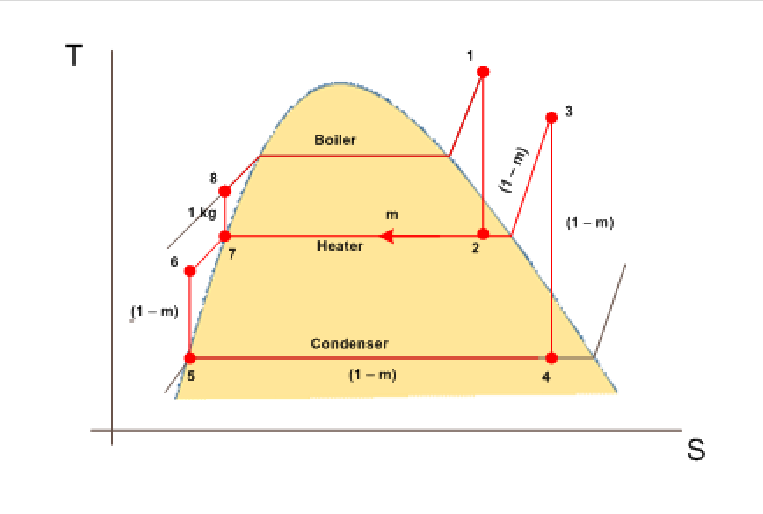
m - fraction of steam bleed for feed-water heating, kg of bled steam/kg of throttled steam

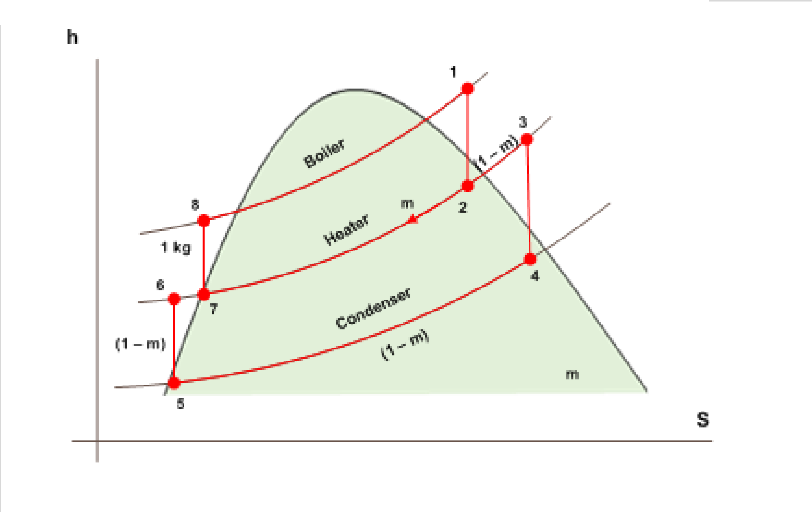
m_s - steam flow rate, kg/sec

Reheat-regenerative Cycle

For a reheat-regenerative cycle, further increase in thermal efficiency will occur because this cycle uses the combined effect of reheating the steam after partial expansion and extracting some part of the circulating steam for feed-water heating, which result to an increase in turbine work and reduces heat required by the boiler.







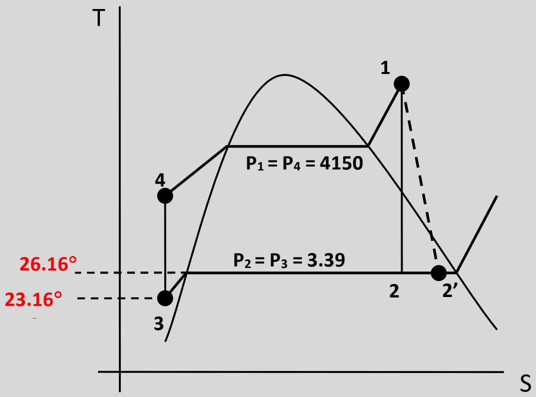
Steam Generator (boiler) – an integrated assembly of several components which converts water into steam at a pre-determined pressure and temperature.
Steam turbine – is a device used to convert the kinetic energy of the substance into mechanical energy/work.
A.C. Generator – an electrical device used to convert the mechanical energy into electrical energy.
Condenser – a heat exchanger used to condense the exhaust steam from the turbine for recycling.
Pump – a mechanical device used to raise the condensate pressure to that of steam generator pressure.
Heat exchangers – are auxiliary equipment used to assist in steam plant operation.
Super heater (closed) – use to superheat the saturated steam from steam drum.
De-superheater (closed or open) –to de-superheat the superheated steam.to final state.
Reheater (closed) - used to reheat the steam from turbine after expansion.
Condenser (closed or open) – used to condemns the exhaust stream.
Feedwater heater (closed or open) – used to preheat the condensate/feedwater.
Economizer (closed) – used to convert the sub-cooled liquid to saturated liquid.
Air pre-heater (closed) - used to preheat the air for combustion.

Example no. 1

A 15,000 KW turbine has a guaranteed steam rate of 5.4 kg/KW-hr at rated load. Steam enters the turbine at 4.15 MPa and 400°C, with exhaust at 25.4 mm Hg absolute. Condensate is sub-cooled 3°C, radiation and frictional losses in the turbine is estimated to be 3% of the turbine output. Calculate:

a) the enthalpy of exhaust steam
b) the total heat removed by the condenser in KJ/hr
c) the thermal efficiency
d) m^3/min of condenser cooling water based on 6°C temperature rise

P_1 = 4150 KPa
$P_2 = P_3$ = 25.4 mm Hg absolute
$P_2 = P_3$ = 3.39 KPa (tsat at 3.39 KPa = 26.16°C)
At P_1 = 4150 KPa and t_1 = 400°C
h_1= 3211.5 KJ/kg ; S_1 = 6.7526 Kj/kg-K
h_3 = h at 3.39 KPa and t_3 = (26.16 – 3) = 23.16°C
h_3 = 97.016°C ; S_3 = 0.3408 KJ/kg-K
h_4 = h at 4150 KPa and S4 = 0.3408 KJ/kg-K
h_4 = 101.18 KJ/kg

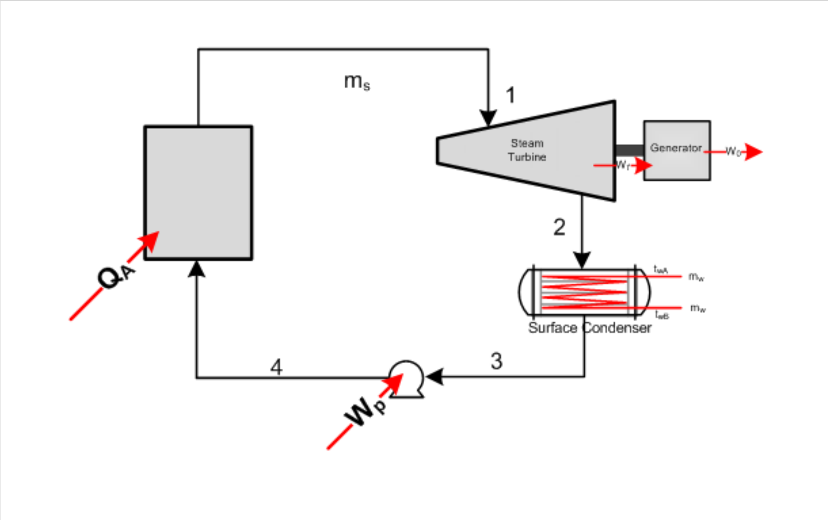


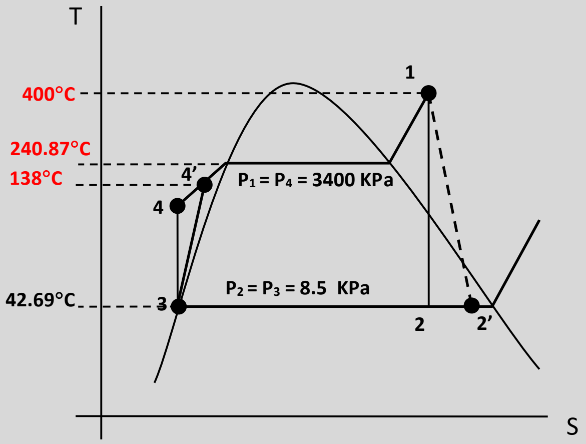
$$SR = \frac{\text{Steam flow rate}}{\text{KW produced}}$$
$$m_s = 5.4(15{,}000) = 81{,}000 \text{ kg/hr}$$
$$m_s = 22.5 \text{ kg/sec}$$
$$W_{t'} = 15{,}000(1-.03) = 14{,}550 \text{ KW}$$
$$W_{t'} = m_s\left(h_1 - h_{2'}\right)$$
$$h_{2'} = h_1 - \frac{W_{T'}}{m_s} = 2564.83 \text{ KJ/kg}$$
$$Q_R = m_s(h_{2'} - h_3) = 22.5(2564.83 - 97.016) = 55{,}525.82 \text{ KW}$$
$$Q_R = m_w(C_{pw})(t_{wB} - t_{wA})$$
$$55{,}525.82 = m_w(4.187)(6)$$
$$m_w = 2{,}210.3 \text{ kg/sec}$$
$$Q_w \text{ - volume flow rate of water in } m^3/\text{min}$$
$$Q_w = 2210.3\,\text{kg/sec}\left(\frac{\text{m3}}{1000\,\text{kg}}\right)(60) = 132.6 \ \frac{m^3}{\text{min}}$$
$$W_P = m_s(h_4 - h_3) = 93.7 \text{ KW}$$
$$Q_A = m_s(h_1 - h_4) = 69{,}982.2 \text{ KW}$$
$$e = \frac{W_{T'} - W_p}{Q_A} \text{ x } 100\%$$
$$e = 20.7\%$$

ExampleNo. 2

A 2500 KW steam turbo generator set has a combined steam rate of 7 kg/KW-hr with a throttle steam at 3.4 MPa (tsat = 240.87°C) and 400°C and a condenser pressure 8.5 KPa absolute(tsat = 42.69°C). Feedwater enters the boiler at 138°C. The steam generator supplying the steam is coal fired, with a coal heating value of 24,000 KJ/kg. Overall steam generator efficiency is 74%. Assuming no losses, Determine:

a) the Rankine cycle efficiency
b) the brake thermal efficiency if generator efficiency is 94%
c) the combined thermal efficiency
d) the fuel consumption in kg/hr
e) the enthalpy of exhaust steam if brake power is equal to actual turbine work





At 3400 KPa and 400°C
h_1 = 3224.6 KJ/kg
S_1 = 6.8620 KJ/kg-K
At 8.5 KPa and $S_2 = S_1$ = 6.8620 KJ/kg-K
h_2 = 2153.4 KJ/kg ; x_2 = 82.302%
At 8.5 KPa; saturated liquid,
h_3 = 178.63 KJ/kg ; S_3 = 0.6078 KJ/kg-K
At $S_3 = S_4$ to 3400 KPa
h_4 = 182.04 KJ/kg
At 3400 KPa and 138°C
$h_{4'}$ = 582.22 KJ/kg

$$e_B = \frac{2,659.6}{17,496.84} = x100\%$$

$$e_B = 15.2\%$$

$$e_{combined} = \frac{2,500}{17,496.84} x100\% = 14.3\%$$

$$m_F = \frac{Q_S}{HV} = \frac{17,496.84}{24,000}(3600) = 2624.526 \frac{kg}{hr}$$

$$W_{t'} = W_B = ms(h1 - h2')$$

$$h_{2'} = 3224.6 - \frac{2,659.6}{4.9} = 2681.824 \frac{KJ}{kg}$$

$$m_s = 7(2500) = 17{,}500\ \frac{kg}{hr} = 4.9\ \frac{kg}{sec}$$

$$e = \frac{(h1 - h2) - (h4 - h3)}{(h1 - h4)} x100\%$$

$$e = 35.07\%$$

$$e_g = \frac{W_o}{W_B}\ x\,100\%$$

$$0.94 = \frac{2500}{W_B}$$

$$W_B = \frac{2500}{0.94} = 2{,}659.6\,KW$$

$$e_B = \frac{W_B}{Q_S} x100\%$$

$$Q_s = m_F(HV)$$

$$e_{Boiler} = \frac{Q_A}{Q_S} x100\%$$

$$Q_A = m_s(h_1 - h_{4'}) = 4.9(3224.6 - 582.22)$$

$$Q_A = 12{,}947.662\,KW$$

$$Q_S = \frac{12{,}947.662}{0.74} = 17{,}496.84\,KW$$

Example No. 3

An ideal steam power plant operating on single stage reheat cycle has steam conditions at throttle of 7.6 MPa and 500°C and expands to the turbine to 2 MPa after which the steam is withdrawn and is reheated in the reheater to 450°C and re-expands again in the turbine to the condenser at a pressure of 7 KPa. For a steam flow rate of 10 kg/sec. determine

The ideal turbine work in KW

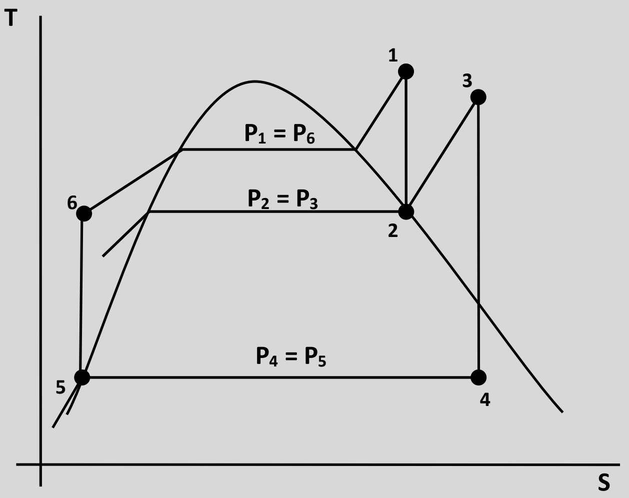
The ideal pump work in KW

The net cycle work in KW

The ideal thermal efficiency

The ideal steam rate in kg/KW-hr

The heat rate in KJ/KW-hr



Steam Properties

At P_1 = 7600 KPa and t_1 = 500°C

h_1 = 3402.7 KJ/kg ; S_1 = 6.7482 KJ/kg-K

At $S_1 = S_2$ to P_2 = 2000 KPa

h_2 = 3012.7 KJ/kg

At P_3 = 2000 KPa and t_3 = 450°C

h_3 = 3357.9 KJ/kg ; S_3 = 7.2900 KJ/kg-K

At $S_3 = S_4$ to P_4 = 7 KPa

h_4 = 2264.0 KJ/kg ; x_4 = 87.2311%

At $P_5 = P_4$ saturated liquid

h_5 = 163.33 KJ/kg; S_5 = 0.5590 KJ/kg-K

At $S_5 = S_6$ to $P_6 = P_1$

a. $W_t = m_s[(h_1 - h_2) + (h_3 - h_4)]$ KW

b. $W_p = m_s(h_6 - h_5)$ KW

c. $W = W_t - W_p$ KW

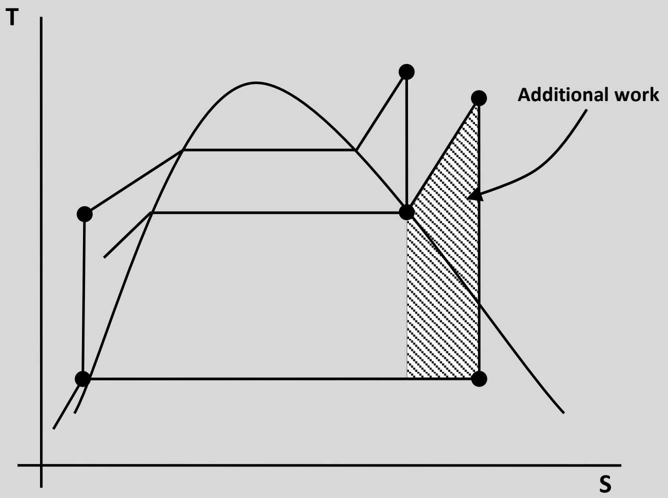
d. $e = \frac{W}{Q_A}\ x\ 100\%$

$Q_A = m_s(h_1 - h_6)$ KW

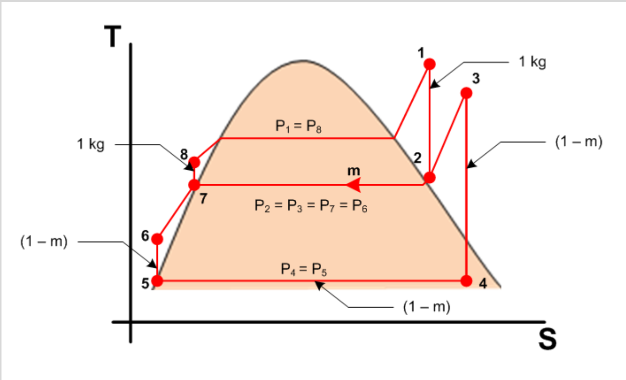
e. $SR = \frac{3600 m_s}{W}\ \frac{kg}{KW\text{-}HR}$

f. $HR = \frac{3600 Q_A}{W}\ \frac{KJ}{KW\text{-}hr}$

h_6 = 170.97 KJ/kg
m_S = 10 kg/sec

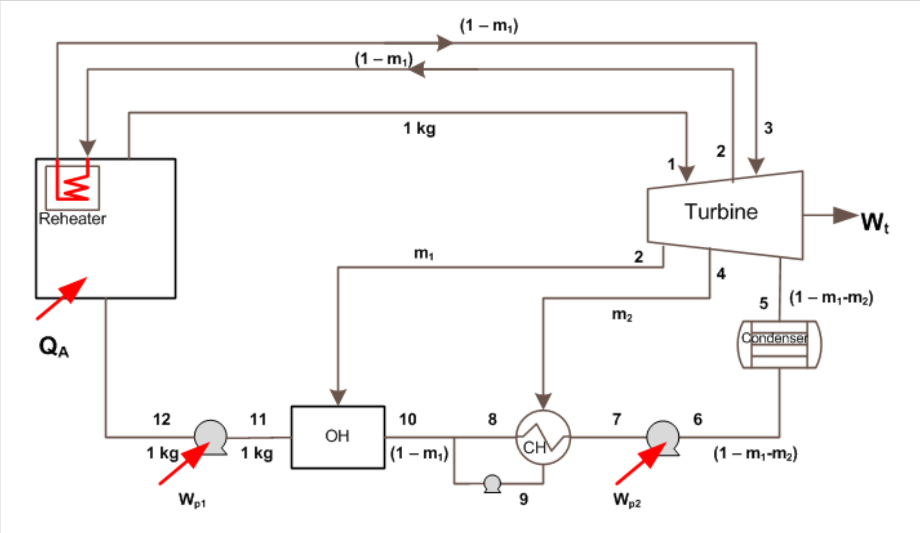


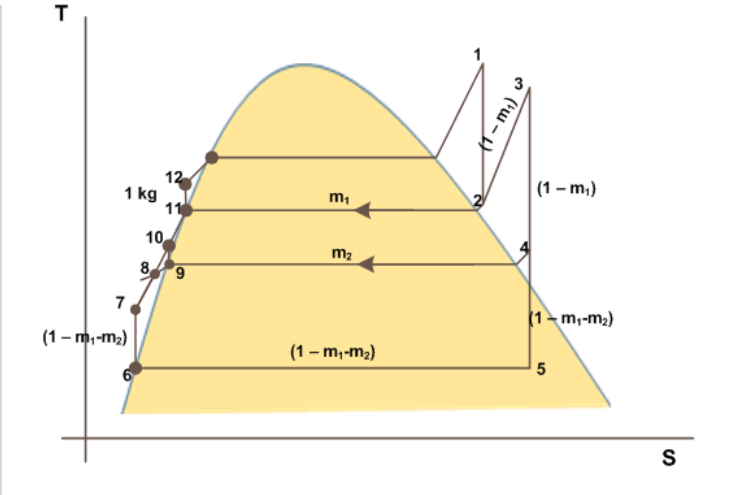
Example No. 4
A steam power plant operates on an ideal reheat– regenerative Rankine cycle and has a net power output of 80 MW. Steam enters the high-pressure turbine at 10 MPa and 550°C and leaves at 0.8 MPa. Some steam is extracted at this pressure to heat the feedwater in an open feed-water heater. The rest of the steam is reheated to 500°C and is expanded in the low-pressure turbine to the condenser pressure of 10 KPa. Show the cycle on a T-s diagram with respect to saturation lines, and determine (a) the mass flow rate of steam through the boiler and (b) the thermal efficiency of the cycle.



Example No. 5
A steam power plant is operating on a reheat – regenerative cycle with 2 – stages of regenerative heating and 1 – extraction for reheating. The turbine receives steam at 7000 KPa and 550°C, the steam expands to 2000 KPa isentropically where a fraction is extracted for feedwater heating and the remainder is reheated at constant pressure until the temperature is 540°C. The steam expands isentropically to 400 KPa where a fraction is withdrawn for LP feedwater heating on a closed type heater as shown on figure; the remainder expands isentropically in the turbine to 7.5 KPa, at which point it enters the condenser. For a mass flow rate of 10 kg/secDetermine

a) The mass fractions of extracted steam
b) The turbine work in KW
c) The pump work in KW
d) The net cycle work in KWThe heat added in KJ/hr
f) The heat rejected from the condenser in KJ/hr





$$W_t = m_s\left[(h_1 - h_2) + (1 - m_1)(h_3 - h_4) + (1 - m_1 - m_2)(h_4 - h_5\right] \text{ KW}$$

$$W_{p1} = m_s(h_{12} - h_{11})$$

$$W_{p2} = m_s(1 - m_1)(h_8 - h_6)$$

$$\Sigma W_p = W_{p1} + W_{p2}$$

$$W = W_t - \Sigma W_p$$

$$Q_A = m_s\left[(h_1 - h_{10}) + (1 - m_1)(h_3 - h_2)\right] \text{ kw}$$

$$e = \frac{W}{Q_A} \text{ x } 100\% e = \frac{W}{Q_A} \text{ x } 100\%$$

$$Q_R = m_s(1 - m_1 - m_2)(h_5 - h_6) \text{ KW}$$

SAMPLE PROBLEMS

Rerheat Cycle

Steam power plant that operates on a reheat Rakine cycle with 80 MW of net output. Steam enters the high pressure turbine at 10 MPa and 500 °C, and is reheated to and enters the low pressure turbine at 1 MPa and 500 °C. Steam leaves the condenser as a saturated liquid at 10 kPa. Isentropic efficiencies of the turbine and compressor are 80 percent and 95 percent, respectively. Find:

a) quality or temperature of steam at turbine exit
b) thermal efficiency of the cycle
c) mass flow rate of the steam
d) condenser cooling watert flow rate allowing a water temperature rise of 15 °C
e) cycle steam rate kg/KW-hr
f} cycle heat rate KJ/KW - hr

Reheat – Regenerative Cycle

Ideal reheat-regenerative Rankine cycle with one open feedwater heater. The boiler pressure is 10 MPa, condenser pressure is 15 kPa, reheater pressure is 1 MPa, and feedwater pressure is 0.6 MPa. Steam enters the high and low pressure turbines at 500 °C. Find:

a) fraction of steam extracted for regeneration

b) thermal efficiency of the cycle.

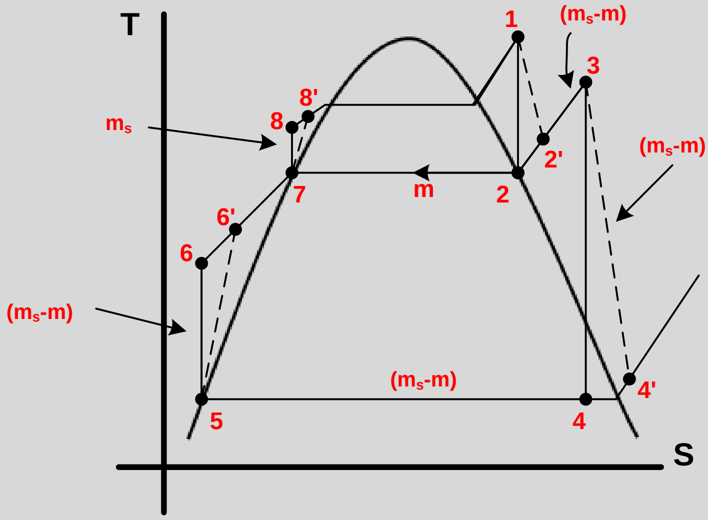
Example 3

Calculate the power plant efficiency and the net work for a steam power plant that has turbine inlet conditions of 6 MPa and 500°C, bleed steam to the heater occurring at 800 KPa and exhausted to the condenser at 15 KPa. The turbine and pump have an internal efficiencies of 90%. The flow rate is 63 kg/sec. (e = 35.6% ; W = 60,511 KW)

A steam power plant operates using the reheat Rankine cycle. Steam enters the high pressure turbine at 12.5 MPa and 550ºC at a rate of 7.7 kg/s and leaves at 2 MPa. Steam is then reheated at a constant pressure to 450ºC before it expands in the low pressure turbine. The isentropic efficiencies of the turbine and the pump are 85% and 90%, respectively. Steam leaves the condenser as a saturated liquid at 150 KPa. Determine

(a) the net power output, and

(b) the thermal efficiency.



P = 12.5 Mpa ; t = 550 C		
h1	s1	
3473.7	6.6250	

P = 2000 Kpa ; S = 6.6250		
h2	t2	tsat
2936.9	263.9	212.36

P = 2000 Kpa ; t = 450 C		
h3	s3	
3357	7.2900	

P = 96 Kpa ; S = 7.2900		
h4	tsat	
2631	96.712	

P =96 Kpa ; Sat. Liquid		
h5	s5	
412.52	1.2896	

P = 2000 Kpa ; s = 1.2896

h6	t6	
414.51	98.629	

P = 2000 Kpa ; Saturatted liq.		
h7	s7	
908.02	2.4465	

P = 12500 Kpa ; s = 2.4465		
h8	tsat	
920.35	214.14	

□t	□p
0.85	0.9
h2'	3017.42
h4'	2739.9
h6'	414.7311
h8'	921.72

$$\eta t = \frac{h1 - h2'}{h1 - h2} = \frac{h3 - h4'}{h3 - h4}$$

$$\eta p = \frac{h6 - h5}{h6' - h5} = \frac{h8 - h7}{h8' - h7}$$

A coal fired steam power plant produces 150 MW of electric power with a thermal efficiency of 35%. If the energetic efficiency of the boiler is 75%, heating value of coal is 30 MJ/kg, and the temperature rise of the cooling water in the condenser is 10ºC, determine (a) the fuel consumption rate in kg/h, and (b) mass flow rate of cooling water.

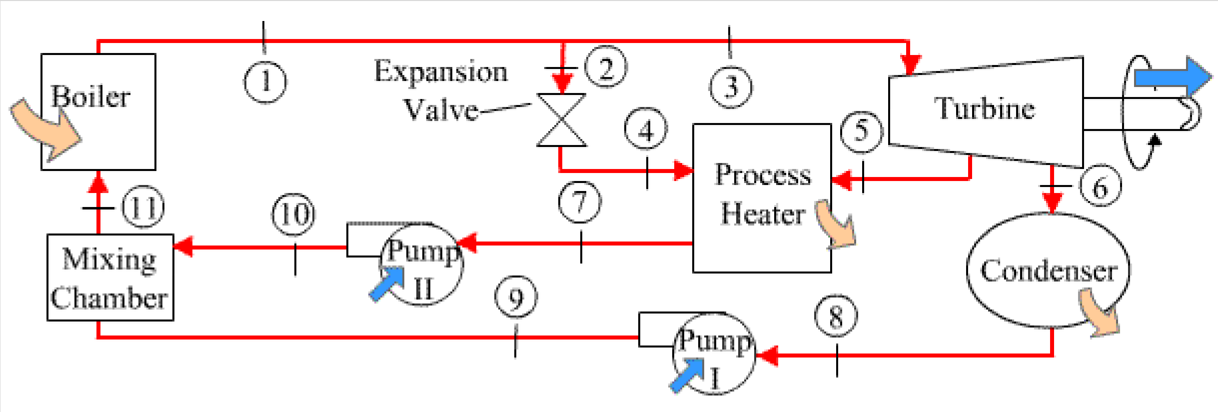
A steam power plant operates on an ideal Rankine cycle. Superheated steam flows into the turbine at 2 MPa and 500ºC with a flow rate of 100 kg/s and exits the condenser at 50ºC as saturated water. Determine (a) the net power output, (b) the thermal efficiency, and (c) the quality of steam at the turbine exit. (d) What would the efficiency be if the condenser temperature can be lowered to 30ºC.

A power plant operates on a simple Rankine cycle producing a net power of 100 MW. The turbine inlet conditions are 15 MPa and 600ºC and the condenser pressure is 10 kPa. If the turbine and pump each has an isentropic efficiency of 85% and there is a 5% pressure drop in the boiler, determine (a) the thermal efficiency, (b) the mass flow rate of steam in kg/h, and (c) the back work ratio. How sensitive is the thermal efficiency on (d) boiler pressure drop, (e) pump efficiency, and (f) turbine efficiency?

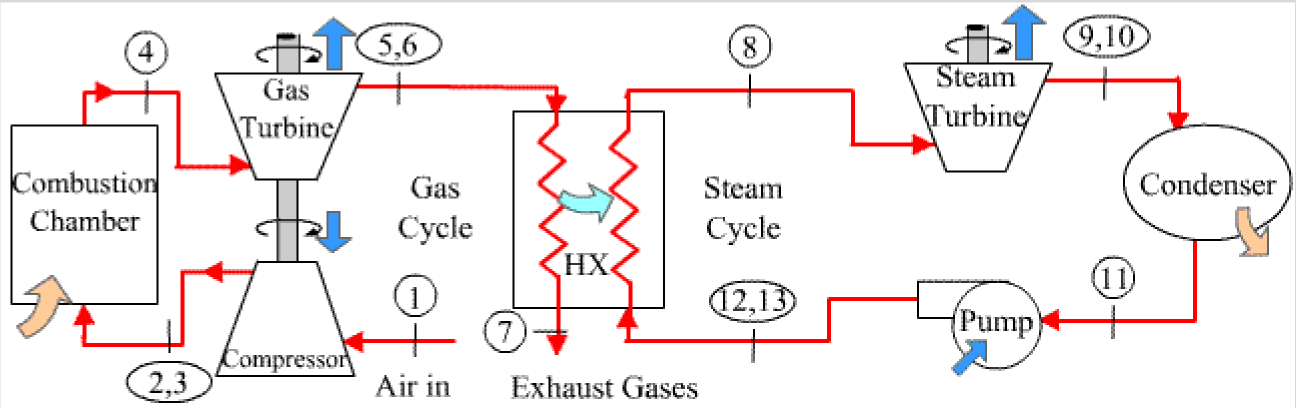
A steam power plant operates on an ideal reheat-regenerative Rankine cycle with two feedwater heaters, one open and one closed. Steam enters the first turbine at 15 MPa and 500ºC, expands to 1 MPa, and is reheated to 450ºC before it enters the second turbine where it expands to 10 kPa. Steam is extracted from the first turbine at 3 MPa and sent to the closed FWH, where the feedwater leaves at 5ºC below the temperature at which the saturated condensate leaves. The condensate is fed through the trap to the open FWH, which operates at 0.5 MPa. Steam is extracted at 0.5 MPa from the second turbine and is fed to the open FWH. The flow out of the open FWH is saturated liquid at 0.3 MPa. If the power output of the cycle is 150 MW, determine (a) the thermal efficiency, (b) the mass flow rate through the boiler, (c) the bleeding rate from the first turbine, and (d) the bleeding rate from the second turbine. What would the thermal efficiency be if (e) the closed FWH were eliminated, (f) both the FWH's were eliminated?

(a) 44.8%, (b) 116.13 kg/s, (c) ,(d) , (e) 43.8%, (f) 41.4%

The cogeneration plant has a peak capacity of 25 MW for process heating. Steam leaves the boiler at 8 MPa and 600ºC. The process heater and the condenser operate at a pressure of 400 kPa and 8 kPa respectively. Assuming ideal behavior of each component, determine (a) the mass flow rate of steam through the boiler and (b) the maximum turbine output. To meet a certain load condition, 10% of the steam is diverted to the expansion valve, and half the steam flowing through the turbine is extracted. Determine (c) the process heat load, and (d) the utilization factor.
ANSWER: (a) 8.23 kg/s, (b) 11.9 MW, (c) 10.61 MW, (d) 71.8%,



A combined gas turbine-steam power plant has a net power output of 500 MW. Air enters the compressor of the gas turbine at 100 kPa, 300 K, and has a compression ratio of 12 and an isentropic efficiency of 85%. The turbine has an isentropic efficiency of 90%, inlet conditions of 1200 kPa and 1400 K, and an exit pressure of 100 kPa. The air from the turbine exhaust passes through a heat exchanger and exits at 400 K. On the steam turbine side, steam at 8 MPa, 400°C enters the turbine, which has an isentropic efficiency of 85%, and expands to the condenser pressure of 8 kPa. Saturated water at 8 kPa is circulated back to the heat exchanger by a pump with an isentropic efficiency of 80%. Determine (a) the ratio of mass flow rates in the two cycles, (b) the mass flow rate of air, and (c) the thermal efficiency. (d) What would the thermal efficiency be if the turbine inlet temperature increased to 1600 K?
ANSWER: (a) 6.673, (b) 1107 kg/s, (c) 53.1%, (d) 54.8%

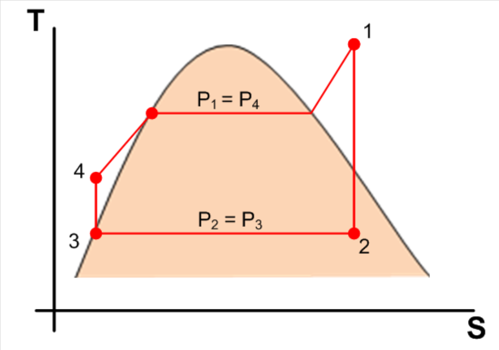


YURI G. MELLIZA

PRACTICE EXERCISES

Problem 1

Consider a steam power plant that operates on a simple ideal Rankine cycle and has a net power output of 45MW. Steam enters the turbine at 7 MPa and 500°C and is cooled in the condenser at a pressure of 10 kPa by running cooling water from a lake through the tubes of the condenser at a rate of 2000 kg/s. Show the cycle on a T-s diagram with respect to saturation lines, and determine (a) the thermal efficiency of the cycle, (b) the mass flow rate of the steam, and (c) the temperature rise of the cooling water.



State	P (KPa)	T(°C)	S	h	x (Quality)
1	7000	500	6.7940	3409.90	
2	10	45.83	6.7940	2151.30	81.924
3	10		0.6492	191.76	
4	7000	45.999	0.6492	198.81	

Problem 2

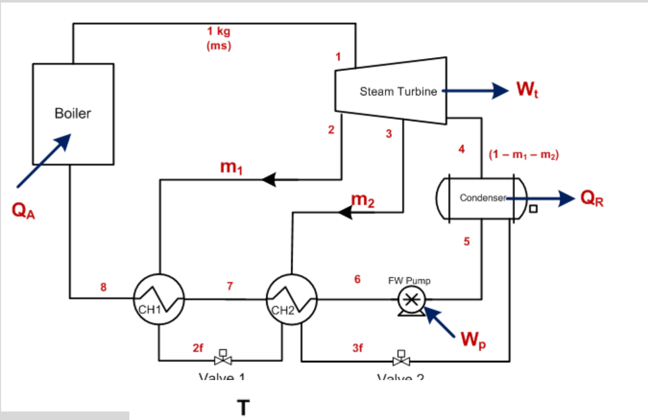
In a regenerative steam cycle as shown below employing two closed feedwater heaters, the steam is supplied to the turbine at 4000 KPa and 500°C and is exhausted to the condenser at 3.5 KPa. The intermediate bleed pressures are obtained such that the saturation pressure intervals are approximately equal, giving pressures of 1000 KPa and 110 KPa, respectively. If the plant to produces a net power output of 10 MW, Calculate the following

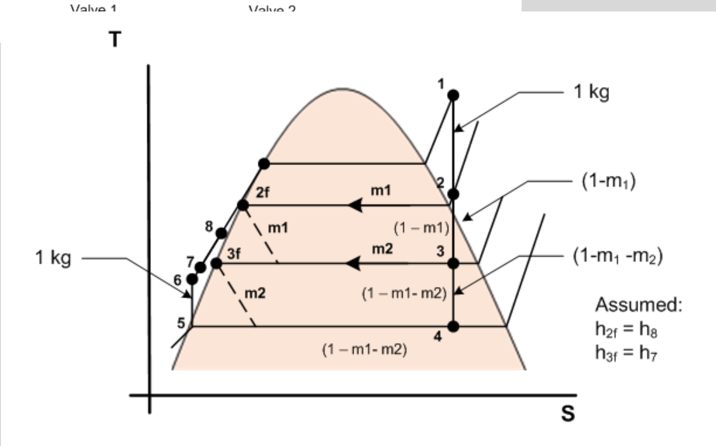
The hourly amount of bleed steam at each stage (m_{s1} and m_{s2})

The cycle thermal efficiency

Note: Draw the matching TS Diagram & assume all liquids leaving bleed steam heaters is saturated liquid at the heater pressure.

Schematic Diagram





Answer	
m_{s1}	4,653.06 Kg/hr
m_{s2}	4,041.48 Kg/hr
e	42.5 %

Problem 3
A steam power plant operates using the reheat Rankine cycle. Steam enters the high pressure turbine at 12.5 MPa and 550ºC at a rate of 7.7 kg/s and leaves at 2 MPa. Steam is then reheated at a constant pressure to 450ºC before it expands in the low pressure turbine. The isentropic efficiencies of the turbine and the pump are 85% and 90%, respectively. Steam leaves the condenser as a saturated liquid. If the moisture content of the steam at the exit of the turbine is not to exceed 5% determine (a) the condenser pressure, (b) the net power output, and (c) the thermal efficiency.

Problem 4
A steam power plant operates on an ideal Rankine cycle. Superheated steam flows into the turbine at 2 MPa and 500ºC with a flow rate of 100 kg/s and exits the condenser at 50ºC as saturated water. Determine
the net power output,
the thermal efficiency, and
the quality of steam at the turbine exit.
What would the efficiency be if the condenser temperature can be lowered to 30ºC?

Problem 5
A steam power plant operates on an ideal reheat-regenerative Rankine cycle with two feedwater heaters, one open and one closed. Steam enters the first turbine at 15 MPa and 500ºC, expands to 1 MPa, and is reheated to 450ºC before it enters the second turbine where it expands to 10 kPa. Steam is extracted from the first turbine at 3 MPa and sent to the closed FWH, where the feedwater leaves at 5ºC below the temperature at which the saturated condensate leaves. The condensate is fed through the trap to the open FWH, which operates at 0.5 MPa. Steam is extracted at 0.5 MPa from the second turbine and is fed to the open FWH. The flow out of the open FWH is saturated liquid at 0.3 MPa. If the power output of the cycle is 150 MW, determine
the thermal efficiency,
the mass flow rate through the boiler,
the bleeding rate from the first turbine, and
the bleeding rate from the second turbine.

Problem 6
Consider a regenerative vapor power cycle with one open feedwater heater. Steam enters the turbine at 8.0 MPa, 480?C and expands to 0.7 MPa, where some of the steam is extracted and diverted to the open feedwater heater operating at 0.7 MPa. The remaining steam expands through the second-stage turbine to the condenser pressure of 0.008 MPa. Saturated liquid exits the open feedwater heater at 0.7 MPa. The isentropic efficiency of each turbine stage is 85% and each pump operates isentropically. If the net power output of the cycle is 100 MW, determine
the thermal efficiency and
the mass flow rate of steam entering the first turbine stage, in kg/h.

Problem no. 7
A coal fired steam power plant produces 150 MW of electric power with a thermal efficiency of 35%. If the energetic efficiency of the boiler is 75%, heating value of coal is 30 MJ/kg, and the temperature rise of the cooling water in the condenser is 10ºC, determine (a) the fuel consumption rate in kg/h, and (b) mass flow rate of cooling water.
(a) 68.57 kg/h , (b) 23,969 kg/h

Multiple Choice Problems

In a Rankine cycle, saturated liquid water at 1 bar is compressed isentropically to 150 bar. First by reheating in a boiler and then by superheating at constant pressure of 150 bar, the water substance is brought to 750K. After adiabatic reversible expansion in a turbine to 1 bar, it is then cooled in a condenser to a saturated liquid. How much work is generated in the turbine? (Steam properties h, kJ/kg, s, kJ/kg-K: @ 150 bar&750 K, h = 3240.5, s1 = 6.2549; @ 1 bar, hf=417.46, hfg=2258, sf=1.3026, sfg=6.0568)
769.9 b. 796.9 c .967.9 d.976.9

A reheat steam has 13850 kPa throttle pressure at the turbine inlet and 2800 kPa reheat pressure. The throttle and reheat temperature of the steam is 540ºC, condenser pressure is 3.4 kPa, engine efficiency of high pressure and low pressure is 75%. Find the cycle thermal efficiency.
34.46% b. 35.56 c. 36.66 d. 37.76

In a Rankine cycle, steam enters the turbine at 2.5 MPa and condenser of 50 kPa. What is the thermal efficiency of the cycle in percent?
25.55 b. 28.87 c. 30.12 d. 31.79

A supercritical steam Rankine cycle has turbine inlet conditions of 17.5 MPa and 530ºC expands in a turbine to 7 kPa. The turbine and pump polytropic efficiencies are 0.9 and 0.7, respectively. Pressure losses between pump and turbine inlet are 1.5 MPa. What should be the pump work in kJ/kg.
27.13 b. 29.87 c. 32.47 d. 33.25

Steam enters the superheater of a boiler at a pressure of 2.5 MPa and dryness of 0.98 and leaves at the same pressure at a temperature of 370ºC. Calculate the heat energy supplied per kg of steam supplied in the superheater.
405.51 b. 504.15 c. 154.15 d. 245.25.

A back pressure steam turbine of 100 MW capacity serves as a prime mover in a cogeneration system. The boiler admits the return water at a temperature of 66ºC and produces the steam at 6.5 MPa and 455ºC. Steam then enters a back pressure turbine and expands to the pressure of the process, which is 0.52 MPa. Assuming a boiler efficiency of 80% and neglecting the effect of pumping and the pressure drops at various location, what is the incremental heat rate for electric? The following enthalpies have been found: at turbine entrance = 3306.8 kJ/kg, exit = 2700.8 kJ/kg; boiler entrance = 276.23 kJ/kg, exit = 3306.8 kJ/kg)
22,504.23 kJ/kW-hr b. 52,244.32 kJ/kW-hr
c. 12,435.72kJ/kW-hr d. 32,234.82 kJ/kW-hr

In an open feedwater for a steam power plant, saturated steam at 7 bar is mixed with sub-cooled liquid at 7 bar and 25ºC. Just enough steam is supplied to ensure that the mixed steam leaving the heater will be saturated liquid at 7 bar when heater efficiency is 90%. Calculate the mass flow rate of sub cooled liquid if steam flow rate is 0.865 kg/s. (Steam properties h, kJ/kg, @ 7 bar, h_g = 2763.5, h_f = 697.22; @ 7 bar & 25ºC, h_f= 105.5)
2.725 b. 2.286 c. 3.356 d. 3.948

A steam plant operates with an initial pressure of 1.7 MPa and 370ºC temperature and exhaust to a heating system at 0.17 MPa. The condensate from the heating system is returned to the boiler at 65.5ºC and the heating system utilizes from its intended purpose 90% of the energy transferred from the steam it receives. The turbine efficiency is 70%. If the boiler efficiency is 80%, what is the cogeneration efficiency of the system in percent? Neglect pump work. (Steam properties h, kJ/kg, s, kJ/kg-K: @ 1.7 MPa & 370ºC; h = 3787.1, s = 7.1081; @ 1.7 MPa, h_f= 483.20, h_{fg}= 2216.0, s_f= 1.4752, s_{fg}= 5.7062; @ 65oC, h_f=274.14)
69 b. 78 c. 91.24 d. 102.10

In a cogeneration plant, steam enters the turbine at 4 MPa and 400ºC. One fourth of the steam is extracted from the turbine at 600kPa pressure for process heating. The remaining steam continues to expand to 10 kPa. The extracted steam is then condensed and mixed with feedwater at constant pressure and the mixture is pumped to the boiler pressure of 4 MPa. The mass flow rate of the steam through the boiler is 30 kg/s. Disregarding any pressure drops and heat losses in the piping, and assuming the turbine and pump to be isentropic, how much process heat is required in kW? (Steam properties h, kJ/kg, s, kJ/kg-K: @ 4 MPa & 400oC, h = 3213.6 s = 6.7690; @ 600 kPa, h_f= 670.56, h_{fg}= 2086.3, s_f= 1.9312, s_{fg}= 4.8288)
1,026.90b. 2,468.2 c. 3,578.5 d. 15,646.8

A 23.5 kg/s at 5 MPa and 400ºC is produced by a steam generator. The feedwater enters economizer at 145ºC and leaves at 205ºC. The steam leaves the boiler drum with a quality of 98%. The unit consumes 2.75 kg of coal per second as received having an heating value of 25,102 kJ/kg. What would be the overall efficiency of the unit in percent? (Steam properties h, kJ/kg, s, kJ/kg-K: @ 5 MPa & 400ºC, h=3195.7; @ 0 MPa, h_f= 1154.23, h_{fg}= 1640.1; @ 205ºC , h_f= 610.63)
65 b. 78 c. 88 d. 95

A coal-fired power plant has a turbine-generator rated at 1000 MW gross. The plant required about 9% of this power for its internal operations. It uses 9800 tons of coal per day. The coal has a heating value of 6,388.9 kCal/kg, and the steam generator efficiency is 86%. What is the net station efficiency of the plant in percent?

30.12 b. 33.07 c. 36.74 d. 40.01

Steam enters the turbine of a cogeneration plant at 7 MPa and 500°C. Steam at a flow rate of 7.6 kg/s is extracted from the turbine at 600 kPa pressure for process heating. The remaining steam continues to expand to 10 kPa. The recovered condensates are pumped back to the boiler. The mass flow rate of steam that enters the turbine is 30 kg/s. Calculate the cogeneration efficiency in percent. (Steam properties h, kJ/kg, s, kJ/kg-K: @ 7 MPa & 500°C, h = 3410.3 s = 6.7975; @ 600 kPa, h_f= 670.56, h_{fg}= 2086.3, s_f= 1.9312, s_{fg}= 4.8228; @ 10 kPa, h_f= 191.83, h_{fg}= 2392.8, s_f= 0.6493, s_{fg}= 7.5009)
50 b. 55 c. 60 d. 65

A 60 MW turbine generator running at 3600 rpm receives steam at 4.0 MPa and 450ºC with back pressure of 10 kPa. Engine efficiency is 78% and the combined mechanical and electrical efficiency is 95%. What would be the exhaust enthalpy of the steam in kJ/kg.
2,400.12 kJ/kg b. 20,432.10 kJ/kg
c. 28,124.20 kJ/kg d. 30,101.15 kJ/kg

Steam enters a throttling calorimeter at a pressure of 1.03 MPa. The calorimeter downstream pressure and temperature are respectively 0.100 MPa and 125ºC. What is the percentage moisture of the supply steam? (Steam properties h, kJ/kg, s, kJ/kg-K: @1.03 MPa, hfg = 2010.7, hg = 2779.25; @ 0.1 MPa & 125oC, h=2726.6)
1.98 b. 2.62 c.3.15 d. 5.21

Steam expands adiabatically in a turbine from 2 MPa, 400ºC to 400 kPa, 250ºC. What is the effectiveness of the process in percent assuming an atmospheric temperature of 15ºC. Neglect changes in kinetic and potential energy. (Steam properties h, kJ/kg, s, kJ/kg-K: @ 2.0 MPa and 400oC; h = 3247.6 s = 7.1271; @ 400 kPa & 250oC, h= 2964.2, s= 7.3789)
79.62 b. 84.52 c. 82.45 d. 74.57

A drum containing steam with 2.5 m in diameter is 7.5 m long. Of the total volume, 1/3 contains saturated steam at 800 kPa and the other 2/3 contains saturated water. If this tank should explode, how much water would evaporate? Consider the process to be of constant enthalpy. (Steam properties h, kJ/kg, v, m3/kg @0.8 MPa, hf = 721.11, hg = 2769.1, vf= 0.0011148, vg=0.2404; @ 0.101325 MPa & 100oC, hf=419.04, hg=2676.1, vf=0.0010435, vg=2769.1)
2,123.76 kg b. 2,424.62 kg
c. 2,651.24 kg d. 2,948.11 kg

A Batangas base industrial company operates a steam power plant with reheat and regeneration. The steam enters a turbine at 300 bar and 900 K and expands to 1 bar. Steam leaves the first stage at 30 bar and part of it entering a closed heater while the rest reheated to 800K. Both section of the turbine have adiabatic efficiency of 93%. A condensate pump exists between the main condenser and the heater. Another pump lies between the heater and condensate outlet line from the heater (condensed extracted steam). Compute for the extracted fraction of the total mass flow to the heater.
0.234 b. 0.543 c. 0.765 d. 0.485

BOILER OR STEAM GENERATOR

A steam generator is a complex combination of economizer, evaporator(Boiler), superheater, reheater and air pre-heater. In addition it has various auxiliaries, such as stokers, pulverizers, burners, fans, emission control equipment, stack and ash handling equipment. A boiler is that portion in the steam generator where saturated liquid is converted to saturated steam, although it may be difficult to separate it, physically, from the economizer. The term “Boiler” is often used to mean the whole steam generator in the literature, however. Steam generators are classified in different ways. They may, for example, be classified as either (1) Utility or (2) Industrial steam generators.

Utility Steam Generators are those used by utilities for electric – power generating plants. Modern utility steam generators essentially are of two basic kinds: (1) the subcritical water-tube drum type and (2) the supercritical once-through type. The supercritical units operate at about 24 Mpa and higher, above the steam critical pressure of 22.11 Mpa. The subcritical drum group usually operate at either 13 Mpa gage or 18 Mpag. The steam capacities of modern utility steam generators are high ranging from 125 to 1250 kg/sec. They power electric power plants with an output ranging from 125 to 1300 Megawatts.

Industrial Steam Generators, on the other hand are those used by industrial and institutional concerns and are of many types. These include the water-tube pulverized-coal units similar to those used by utilities. Some are heat recovery types that use the waste heat from other industrial processes. They may be of the fire-tube variety . Industrial steam generators usually do not produced superheated steam. Rather, they usually produced saturated steam, or even only hot water (in which case they should not be called as steam generators). They operate at pressures ranging from a few Mpa to as much as 10.5 Mpag and steam or hot water capacities ranging from a few kg/sec to 125 kg/sec.
Fossil-fueled steam generators are more broadly classified [12,13] as those having the following components or characteristics.

- Fire-tube boilers
- Water-tube boilers
- Natural-circulation boilers
- Controlled-circulation boilers

- Once-through flow
- Subcritical pressure
- Supercritical pressure

FIRE-TUBE BOILER

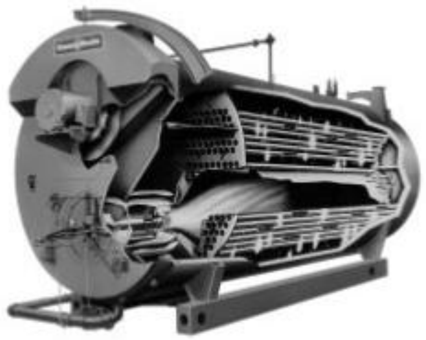
Fire tube boilers have been used in various early forms to produce steam for industrial purposes at the upper limits of 1.8 Mpag pressure and 6.3 kg/sec capacity. Although their size has increased, their general design has not change appreciably in the past 25 years.

Fire-tube boiler is a special form of the shell-type boiler. A shell-type boiler is a closed, usually cylindrical, vessel or shell that contains water. A portion of the shell such as its underside is simply exposed to heat, such as gases from an externally fired flame. The shell boiler evolve into more modern forms such as the electric boiler, in which heat is supplied by electrodes embedded in the water, or the accumulator, in which heat is supplied by steam from an outside source passing through tubes within the shell. In both cases the shell itself is no longer exposed to heat.

The shell boiler evolves into the fire-tube boiler. Hot gases, instead of steam, were now made to pass through the tubes. Because of improved heat transfer the fire-tube boiler is more efficient than the original shell boiler and can reach efficiencies of about 70 percent. The fire tubes were placed in horizontal, vertical or inclined positions. The most common was the horizontal-tube boiler, in which the furnace and grates are located underneath the front end of the shell. The gases pass horizontally along its underside to the rear, reverse direction, and pass through the horizontal tubes to the stack at the front.

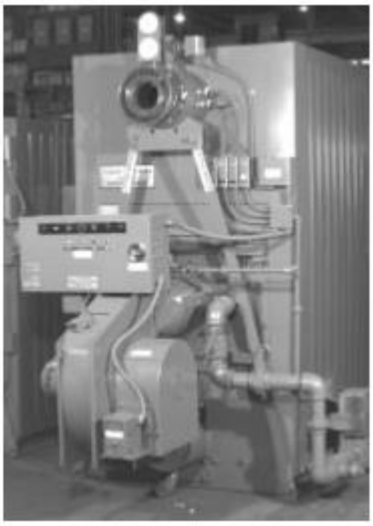
There are two types of fire-tube boilers: (1) the fir-box and (2) the scotch marine. In the fire-box boiler, the furnace or the fire box, is located within the shell, together with the fire tubes. In the scotch marine boiler, combustion takes place within one or more cylindrical chambers that are usually situated inside and near the bottom of the main shell. The gases leave these chambers at the rear, reverse direction, and return through the fire tubes to the front and out through the stack. Scotch marine boilers are usually specified with liquid or gas fuels.

Because boiling occurs in the same compartment where water is, fire-tube boilers are limited to saturated-steam production. They are presently confined to relatively small capacities and low steam pressures, such as supplying steam for space heating and in decreasing numbers, for railroad locomotive service. The largest scotch marine offered in the United States today is rated at 2000 Boiler HP, containing two combustion chambers within a 13-ft diameter, 30-ft long shell.



WATER TUBE BOILER

Demands for increased capacity and pressure over that economically obtained from the fire-tube unit led to the development of the water-tube boiler. Although available for smaller capacities, the field of the water-tube boiler begins at about 2 kg/sec of steam. The water tube boiler puts the pressure instead in tubes and relatively small diameter drums that are capable of withstanding the extreme pressures of the modern steam generators. In general appearance, the early water-tube boiler looks much like the fire tube-boiler except that the higher pressure water were inside the tubes and the combustion gases were on the outside. The water-tube boiler went through several stages of development.



Straight-Tube Boiler: The 1st water-tube boiler was the straight-tube boiler, in which straight tubes, 3 to 4 in OD, inclined at about 15° and staggered with 7 to 8 in spacing, connected two vertical headers. One header was a downcomer, or downtake, which is supplied nearly saturated water to the tubes. The water partially boiled in the tubes. The other header was a riser, or uptake, which received the water-steam mixture. The water density in the downcomer was larger than the two-phase density in the riser, which caused natural circulation in a clockwise direction. As capacity increased, more than one header each and more than one tube “deck” were used. The two-phase mixture went into the upper drum that was arranged either parallel to the tubes (the longitudinal drum) or perpendicular to them (the cross drum). These drums received the feedwater from the last feedwater heater and supplied saturated steam to the super-heater through a steam separator within the drum which separated steam from the bubbling water. The lower end of the downcomer was connected to the mud drum, which collected sediments from the circulating water.

A single longitudinal drum, usually 1.2 m in diameter, can allow only a limited number of tubes and hence a limited heating surface. Longitudinal-drum boilers were built with one more or more than one parallel drums, depending upon capacity. They were built with heating surfaces of 93 to 930 m^2 and were limited to low pressures of 1.2 Mpa to 2.3 Mpa and steam capacities from 0.63 to 10 kg/sec.

Cross-drum boilers, because of geometry, could accommodate many more tubes than longitudinal-drum boilers and were built with heating surfaces of 93 to 2300 m^2, pressures of 1.2 Mpa to 10 Mpa, and steam capacities of 0.63 to 63 kg/sec.

Baffles were installed across the tubes in both kinds to allow for up to three gas passes to ensure maximum exposure of the tubes to the hot combustion gases and minimal gas dead spots.

The Bent-Tube Boiler: There were many versions of the bent-tube boiler. In general, a bent-tube boiler used bent, rather than straight tubes between several drums or drum and headers. The tubes were bent so that they entered and left the drums radially. The number of drums usually varied from two to four. Gas baffles were installed to allow for one or more gas passages.

Water-Tube Boilers: Recent Developments

The advent of the water-cooled furnace walls, called water walls, eventually led to the integration of furnace, economizer, boiler, superheater, reheater, and air preheater into the modern steam generator. Water-cooling is also used for superheater and economizer compartment walls and various other components, such as screens, dividing walls, etc. The use of a large number of feedwater heaters (up to seven or eight) means a smaller economizer, and the high pressure means a smaller boiler surface because of the latent heat of vaporization decreases rapidly with pressure. Thus, a modern high-pressure steam generator requires more superheating and reheating surface and less boiler surface than older units. Beyond about 10 Mpa, the water tubes represent the entire boiler surface and no other tubes.

Water at 230°C to 260°C from the plant high-pressure feedwater heater enters the economizer and leaves saturated or as a two-phase mixture of low quality. It then enters the steam drum at midpoint. Water from the steam drum flows through insulated downcomers, which are situated outside the furnace, to a header. The header connects to the water tubes that line the furnace walls and act as risers. Thw water in the tubes receives heat from the combustion gases and boils further. The density differential between the water inn the downncomer and that in the water tubes helps circulation. Steam is separated from the bubbling water in the drum and goes to the superheater and the high pressure section of the turbine. The exhaust from the turbine returns to the reheater, after which it goes to the low-pressure section of the turbine.

Atmospheric air from a FD fan is preheated by the flue gases just before they exhausted to the atmosphere. From there it flows into the furnace, where it mixes with the fuel and burn to some 1650°C. The combustion gases impart portion of some of their energy to the water tubes and then the superheater, reheater, and economizer, and leave latter at about 320°C. From there they reheat the incoming atmospheric air in the air preheater, leaving it about 150°C. An ID fan draws the flue gases from the system and sends them up the stack. The temperature of about 150°C of the exiting flue gases represents an availability loss to the plant. This, however, is deem acceptable because (1) the gas temperature should be kept well above the dew point of the water vapor in the gases to prevent condensation which should form acids that will corrode metal components in its path and (2) the flue gases must have enough buoyancy to rise in a high plume above the stack for proper atmospheric dispersion.

Boiler Walls

The water tubes that cool the water walls are closely spaced for maximum heat absorption. Tube construction has varied over the years from bare tubes (a) tangent to or (2) embedded in the refractory, to (c) studded tubes, to the now-common membrane design (d). The membrane design consists of tubes spaced on centers slightly wider than their diameter, connected by bars or membranes welded to the tubes at their centerlines. The membranes acts as fins to increase the heat transfer as well as to afford a continuous rigid and pressure-tight construction for the furnace. No additional inner casing is required to contain the combustion gases. Insulation and metal lagging to protect it are provided on the outer side of the wall. One manufacturer has standardized its design on 3-in diameter tubes on 3.75 in between centers, another on 3 in on 4 1n, and yet a third on 2.75 in on 3.75 in.

Radiant Boiler

Heat is transferred from the combustion gas to the water walls by both radiation and convection. A radiant boiler, as the name implies, receives most of its heat by radiation.

The combustion gases have characteristics that depend upon the fuel used, the combustion process and the air-fuel ratio. They may be luminous, i.e., emit all wavelengths and hence strong visible radiation if there are particulates such as soot particles during the combustion process. This is the case with coal and oil. They may be nonluminous, in which case they burn cleanly without particulates, as in the case of gaseous fuels. No combustion gases are truly nonluminous because the heavier gases in the combustion products, in particular the triatomic CO_2 nd H_2O, but also SO_2, ammonia and sulfur dioxide, are selective radiators that emit (and absorbs) radiation in certain wavelengths, mostly outside the visible range. The portion of radiation within the visible range is small but gives the combustion gases a green-blue appearance. Lighter gases such as monatomic or diatomic gases are poor radiators.

The radiant energy emitted by the combustion gases depends upon the gas temperature (to the fourth power), the partial pressure of the individual constituents radiating gases, the shape and size of the gases, their proximity to the absorbing body, and the temperature of that body (to the fourth power).

The convective portion of the heat transfer follows the usual Nusselt-Reynolds turbulent forced-convection relationship. It is smaller than the radiant portion because a thick body of gas causes radiation, whereas convection is localized near the tube surface.

Heat received by the water walls is conducted through the membranes and tube walls and is then convected to the two-phase mixture inside the tubes by forced convection nucleate-boiling heat transfer. The heat transfer resistance of the latter is much smaller than the others so that it may be neglected in design calculations with little error.

Radiant boilers are designed for electric-generating stations to use coal or lignite for pulverized or cyclone furnace applications, oil or natural gas. They are built to supply a wide range of steam pressures and temperatures, but usually around 540°C and steam capacities up to 1260 kg/sec. They are limited to subcritical pressures, usually 12.5 Mpag to 17 Mpag.

Boiler Auxiliaries and Accessories

Superheater – a heat exchanger that is used to increase the temperature of the water vapor greater than the saturation temperature corresponding the boiler pressure.
Evaporator – a heat exchanger that changes saturated liquid to saturated vapor.
Economizers – is the heat exchanger that raises the temperature of the water leaving the highest pressure feedwater heater to the saturation temperature corresponding to the boiler pressure.
Air Preheater – is a heat exchanger use to preheat air that utilizes some of the energy left in the flue gases before exhausting them to the atmosphere.
Fans – a mechanical machine that assist to push the air in, pull the gas out or both.
Stoker – combustion equipment for firing solid fuels (used in water tube boilers)
Burners – combustion equipment for firing liquid and gaseous fuels.
Feedwater pump – a pump that delivers water into the boiler.
Pressure Gauge – indicates the pressure of steam in the boiler.
Safety Valve – A safety device which automatically releases the steam in case of over pressure.
Temperature Gauge – indicates the temperature of steam in the boiler.
Fusible Plug – a metal plug with a definite melting point through which the steam is released in case of excessive temperature which is usually caused by low water level.
Water Walls – water tubes installed in the furnace to protect the furnace against high temperature and also serve as extension of heat transfer area for the feed-water.
Gage Glass (Water column) – indicates the water level existing in the boiler.
Baffles – direct the flow of the hot gases to effect efficient heat transfer between the hot gases and the heated water.
Furnace – encloses the combustion equipment so that the heat generated will be utilized effectively.
Soot blower – device which uses steam or compressed air to remove the soot that has accumulated in the boiler tubes and drums.
Blowdown Valve – valve through which the impurities that settle in the mud drum are remove. Sometimes called blow 0ff valve.
Breeching – the duct that connects the boiler and the chimney.
Chimney or Smokestack – a structure usually built of steel or concrete that is used to dispose the exhaust gases at suitable height to avoid pollution in the vicinity of the plant.

BOILER PERFORMANCE

1.Heat Generated by Fuel

$$Q_S = m_f\,(HHV)\ \frac{KJ}{HR}$$

Where: m_f – fuel consumption, kg/hr

HHV – higher heating value of fuel KJ/kg

2. Rated Boiler Horsepower(R.BHp)

For Water Tube Type

$$R.BHp = \frac{HS}{0.91}$$

For Fire Tube Type

$$R.BHp = \frac{HS}{1.1}$$

Where: HS – required heating surface, m^2

3. Developed Boiler Horsepower (D.BHp)

$$D.BHp = \frac{m_s(h_s - h_f)}{15.65(2257)}$$

$$D.BHp = \frac{m_s(h_s - h_f)}{35{,}322}$$

One Boiler Horsepower is equivalent to the generation of 15.65 kg/hr of steam from water at 100°C to saturated steam at 100°C. The latent heat of vaporization of water at 100°C was taken at 2257 KJ/kg.

4. Percentage Rating

$$\%R = \frac{D.BHp}{R.BHp} \times 100\%$$

5. ASME Evaporation Units

$$ASME\ \ Evap.\ \ Units = m_s(h_s - h_f)$$

6.Factor of Evaporation (FE)

$$FE = \frac{(h_s - h_f)}{2257}$$

7. Boiler Efficiency

$$\eta_{Boiler} = \frac{m_s(h_s - h_f)}{m_F(HHV)} \times 100\%$$

8. Net Boiler Efficiency

$$\eta_{Net} = \frac{m_s(h_s - h_f) - Auxiliaries}{m_F(HHV)} \times 100\%$$

9. Actual Specific Evaporation

$$Actual\ \ Sp.\ Evap. = \frac{m_s}{m_F}\ \ \frac{kg\ of\ steam}{kg\ \ of\ \ fuel}$$

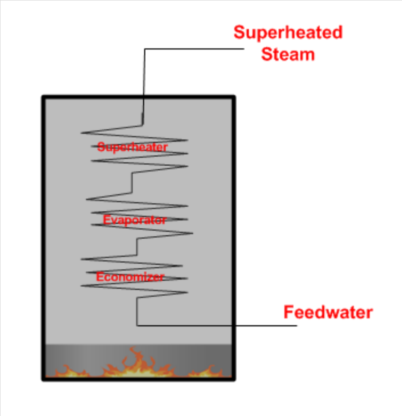
10. Equivalent Evaporation

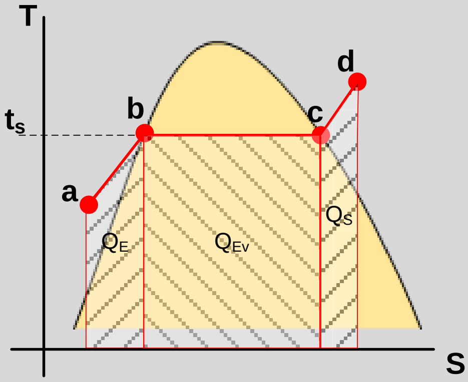
$$Equiv.\ Evap. = m_s(FE)$$

11. Equivalent Specific Evaporation

$$Equiv.\ Sp. Evap. = \frac{m_s}{m_F}(FE)$$

Economizer, Evaporator, Superheater, Reheater and Air Pre–Heater of a thermal Power Plant





a - subcooledliquid

b - saturated liquid

c - saturated vapor

d - superheated vapor

Q_E - heat required by the economizer

Q_{Ev} - heat required by the evaporator

Q_s - heat required by the superheater

For Shell and Tube type Heat Exchanger

$$Q = F(UA)(LMTD)$$

$$LMTD = \frac{\theta_2 - \theta_1}{\ln \frac{\theta_2}{\theta_1}}$$

where

Q - Total heat Transfer, KW

F - Correction Factor for complex design

U - Overall Coefficient of heat transfer, $\frac{KW}{m^2 - K}$ or $\frac{KW}{m^2 - °C}$

A - total heat transfer surface area, m^2

(LMTD) - Log Mean Temperature Difference, °C or K

θ - Terminal Temperature Difference

$A = \pi DL(N_t) \rightarrow$ For U based on Outside surface

$A = \pi dL(N_t) \rightarrow$ For U based on inside surface

N_t - number of tubes per Pass

L - length of tubes for single pass

For multiple tube passes

Tube length is $\frac{L}{\text{Number of passes}}$

Total Number of tubes is N_t(Number of passes)

Actual tube length

$L' = \text{Tube Length} + 2t$

where t = tube sheet thickness

Volume Flow rate of Tube Fluid

$$Q_V = \frac{\pi}{4} d^2 (\text{Velocity})(N_t)$$

$$Q_V = \frac{\text{Mass flow rate of tube fluid in kg/sec}}{\text{Density of tube fluid in kg/m}^3} \frac{m^3}{sec}$$

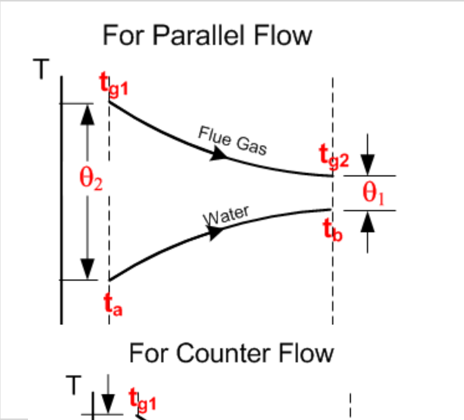
Economizer

$$Q_E = m_g(C_{pg})(t_{g1} - t_{g2}) = m_s(h_c - h_b) = F(UA)(LMTD)$$

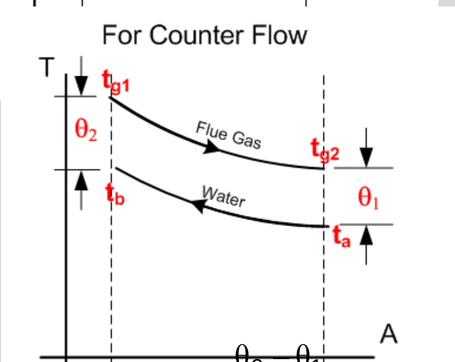
m_g – mass flow rate of flue gas, kg/sec

m_s - mass flow rate of feed water or steam produced, kg/sec

C_{pg} – specific heat of flue gas, $\frac{KJ}{kg\text{-}°C}$ or $\frac{KJ}{kg\text{-}K}$



$$\theta_2 = (t_{g1} - t_a)$$
$$\theta_1 = (t_{g2} - t_b)$$



$$\theta_2 = (t_{g1} - t_b)$$
$$\theta_1 = (t_{g2} - t_a)$$

$$LMTD = \frac{\theta_2 - \theta_1}{\ln \frac{\theta_2}{\theta_1}}$$

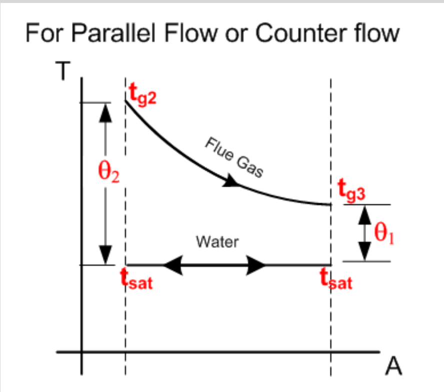
Evaporator (Boiler)

$$Q_{Ev} = m_g(C_{pg})(t_{g2} - t_{g3}) = m_s(h_b - h_a) = F(UA)(LMTD)$$

m_g – mass flow rate of flue gas, kg/sec

m_s - mass flow rate of feed water or steam produced, kg/sec

C_{pg} – specific heat of flue gas, $\frac{KJ}{kg\text{-}°C}$ or $\frac{KJ}{kg\text{-}K}$



$$\theta_2 = (t_{g2} - t_{sat})$$
$$\theta_1 = (t_{g3} - t_{sat})$$

Superheater

$$Q_S = m_g(C_{pg})(t_{g3} - t_{g4}) = m_s(h_d - h_c) = F(UA)(LMTD)$$

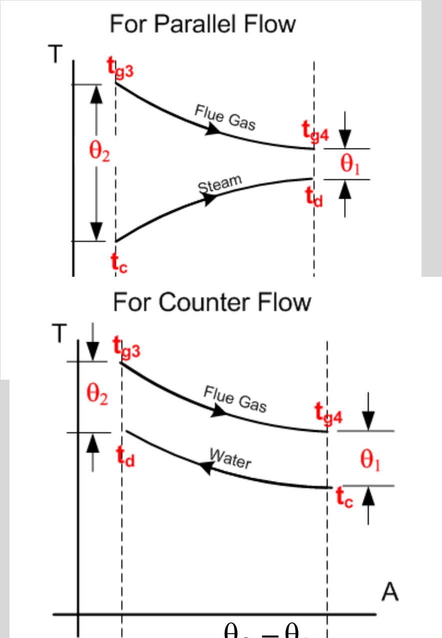
$$Q_S = m_s(C_{psteam})(t_d - t_{sat})$$

m_g – mass flow rate of flue gas, kg/sec

m_s - mass flow rate of feed water or steam produced, kg/sec

C_{pg} – specific heat of flue gas, $\frac{KJ}{kg\text{-}°C}$ or $\frac{KJ}{kg\text{-}K}$

$C_{psteam} = 1.86 \frac{KJ}{kg\text{-}°C}$ or $\frac{KJ}{kg\text{-}K}$



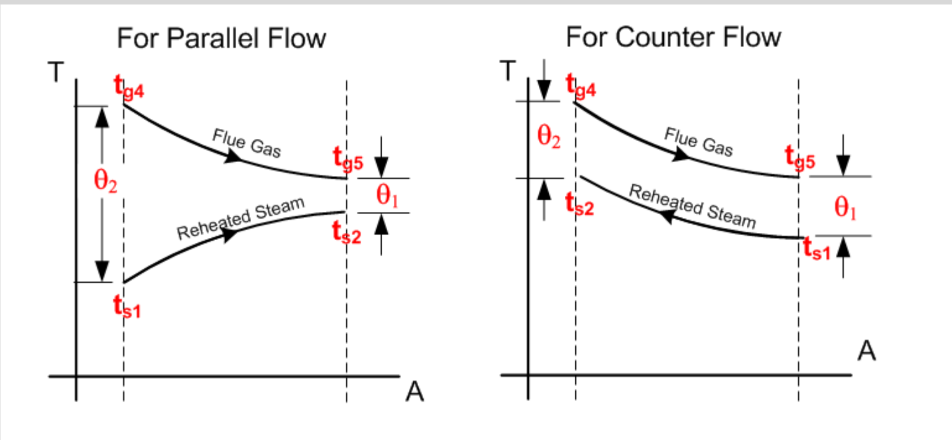
$$\theta_2 = (t_{g3} - t_c)$$
$$\theta_1 = (t_{g4} - t_d)$$

$$\theta_2 = (t_{g3} - t_d)$$
$$\theta_1 = (t_{g4} - t_c)$$

$$LMTD = \frac{\theta_2 - \theta_1}{\ln \frac{\theta_2}{\theta_1}}$$

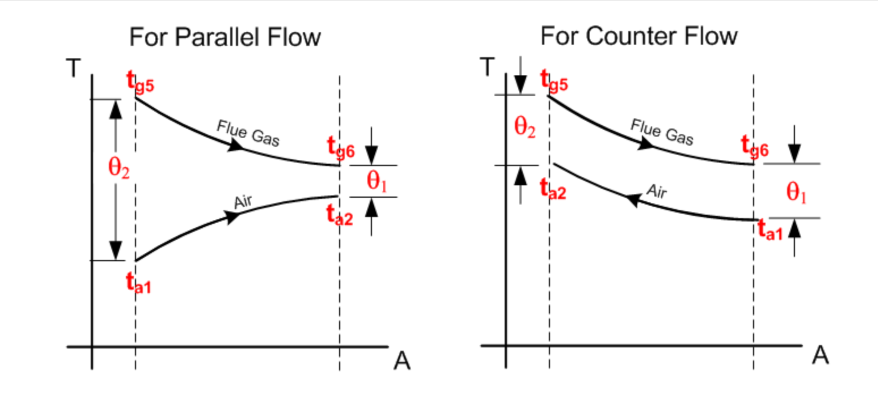
Re - Heater

Hot Fluid – Flue Gas
Cold fluid – Reheated Steam



Air Preheater

Hot Fluid – Flue Gas
Cold fluid – Combustion Air



Why Perform Combustion Analysis?

In actual combustion processes, other products are often formed. A typical example of an actual combustion process is shown in Figure 1. Fuel has reacted with air to produce the products shown on the right.

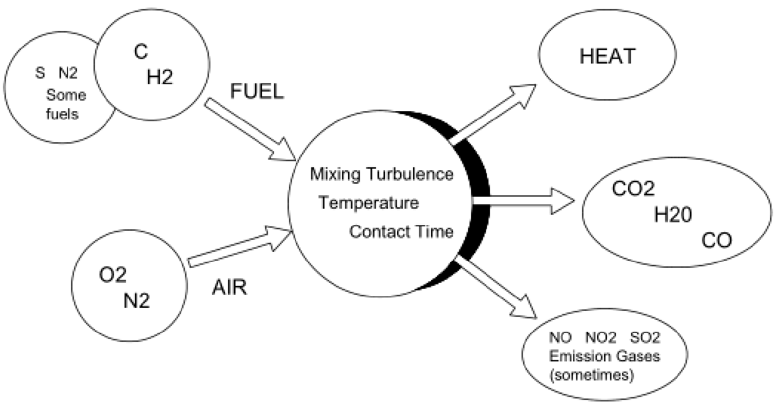


Figure 1. Combustion Diagram

Combustion Analysis

Combustion analysis is part of a process intended to improve fuel economy, reduce undesirable exhaust

Improve Fuel Efficiency

The largest sources of boiler heat losses are shown Figure 2. Heat energy leaving the system exhaust flue (or stack) is often the largest single source of lost fuel energy and is made up of the Dry Gas loss and Latent Heat Loss. Although some flue loss is unavoidable, an equipment *tune-up* using combustion analysis data can often significantly reduce this source of heat loss and save fuel costs by improving

Reduce Emissions

Carbon monoxide, sulfur dioxide, nitrogen oxides and particles are undesirable *emissions* associated with burning fossil fuels. These compounds are toxic, contribute to acid rain and smog and can ultimately cause respiratory problems. Federal and state laws govern the permissible emission rates for these pollutants under the guidance of the Clean Air Act and oversight of the federal Environmental Protection Agency (EPA). State and local environmental agencies also exert authority in regulating the emissions of these pollutants. Combustion analysis is performed to monitor toxic and acid rain forming emissions in order to meet these federal, state and local regulations.

Improve Safety

Good equipment maintenance practice, which includes combustion analysis, enables the boiler technician to fully verify and maintain the equipment operating specifications for safe and efficient operation. Many boiler manufacturers suggest that flue gas analysis be performed at least monthly. Boiler adjustments that affect combustion will tend to drift with time. Wind conditions and seasonal changes in temperature and barometric pressure can cause the excess air in a system to fluctuate several percent. A reduction in excess air can cause, in turn, a rapid increase of highly toxic carbon monoxide and explosive gases, resulting in rapid deterioration in system safety and efficiency. Low draft pressures in the flue can further result in these combustion gases building up in the combustion chamber or being vented indoors. Excessive draft pressures in the flue also can cause turbulence in the system. This can prevent complete combustion and pull explosive gases into the flue or cause flame impingement and damage in the combustion chamber and to the heat exchanger material.

What's Measured?

Combustion analysis involves the measurement of gas concentrations, temperatures and pressure for boiler tune-ups, emissions checks and safety improvements. Parameters that are commonly examined include:

- Oxygen (O_2)
- Carbon Monoxide (CO)
- Carbon Dioxide (CO_2)
- Exhaust gas temperature
- Supplied combustion air temperature
- Draft
- Nitric Oxide (NO)
- Nitrogen Dioxide (NO_2)
- Sulfur Dioxide (SO_2)

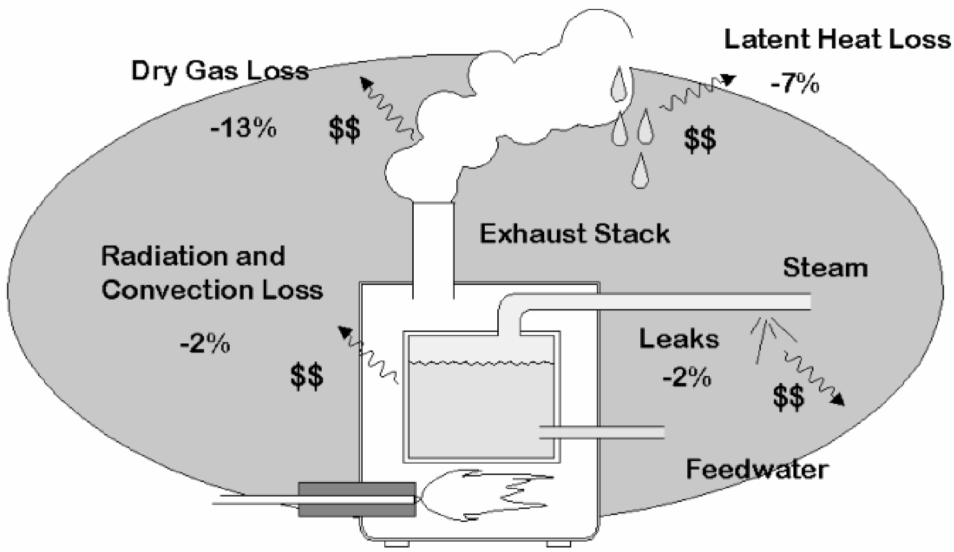


Figure 2. Boiler Heat Losses

BOILER HEAT BALANCE

Energy supplied to the boiler by 1 kg of fuel is distributed among the following items in the ASME short-form heat balance, all expressed in units of KJ/kg of fuel.

BOILER HEAT BALANCE
HEAT SUPPLIED BY FUEL = Fuel Flow Rate x Heating Value of Fuel
1. Heat absorbed by steam generating unit
2. Heat loss due to moisture in fuel
3. Heat loss due to water formed from the combustion of hydrogen in the fuel
4. Heat loss due to moisture in air supplied
5. Heat loss due to dry flue gas
6. Heat loss due to unburned gaseous combustible
7. Heat loss due to unconsumed combustible in the refuse
8. Heat loss due to unconsumed hydrogen, hydrocarbon, radiaton & unaccounted for

1. Heat absorbed by steam generating unit

$$Q_1 = \frac{m_s(h_s - h_f)}{m_F} \frac{KJ}{kg}$$

2. Heat loss due to moisture in fuel

$$Q_2 = M(h'' - hf') \frac{KJ}{kg}$$

3. Heat loss due to water formed from the combustion of hydrogen in the fuel

$$Q_3 = 9H_{2'}(h'' - hf') \frac{KJ}{kg}$$

$$H_{2'} = \left(H_2 - \frac{M}{9}\right) \frac{\text{kg og Hydrogen}}{\text{kg of fuel}}$$

4. Heat loss due to moisture in air supplied

$$Q_4 = m_{aa}W(1.926)(t_g - t_a)\frac{KJ}{kg}$$

5. Heat loss due to dry flue gas

$$Q_5 = m_{dg}C_{Pdg}(t_g - t_a)\frac{KJ}{kg}$$

6. Heat loss due to unburned gaseous combustible

$$Q_6 = \frac{CO}{CO_2 + CO}(24{,}000)C_1 \frac{KJ}{kg}$$

7. Heat loss due to unconsumed combustible in the refuse

$$Q_7 = 33{,}820(C - C_1)\frac{KJ}{kg}$$

8. Heat loss due to unconsumed hydrogen, hydrocarbon, radiaton & unaccounted for

$$Q_8 = HHV - (Q_1 + Q_2 + Q_3 + Q_4 + Q_5 + Q_6 + Q_7)$$

m_s - steam flow rate in kg/hr
m_F - fuel consumption in kg/hr
h_s - enthalpy of steam in KJ/kgh
h_f - enthalpy of feedwater to boiler in KJ/kg
m_{H_2O} - kg of free moisture per kg of fuel as fired
H_2 - kg of hydrogen in fuel, kg/kg
m_{aa} - actual air used per kg of fuel for combustion

$$m_{aa} = m_c C_1$$

$$m_c = \frac{28N_2}{12(CO_2 + CO)(0.77)} \frac{kg \text{ of air}}{kg \text{ of Carbon}}$$

$$C_1 = \frac{m_F C - m_r C_r}{m_F} \frac{kg \text{ of Carbon Burned}}{kg \text{ of fuel}}$$

$$m_{aa} = m_{dg} + 8\left(H_2 - \frac{O_2}{8}\right) - C_1 - S - N_2 \ \frac{kg \text{ of air}}{kg \text{ of fuel}}$$

C_1 - carbon actually burned
C - Carbon in fuel in kg/kg
C_r - carbon conternt in the refuse kg/kg
m_r - weight of refuse, kg

$$C_r = \frac{m_r}{m_F} - Ash$$

N_2, CO, and CO_2 - are in %age or decimals from the product analysis
m_{dg} - mass of dry flue gas, kg/kg of fuel

$$m_{dg} = \frac{44CO_2 + 32O_2 + 28(CO + N_2)}{12(CO_2 + CO)}(C_1)\frac{kg \text{ of dry gas}}{kg \text{ of fuel}}$$

S in%

h'' - enthalpy of superheated steam at flue gas temperature and P_{H2O}

h' - enthalpy of liquid at entering air and fuel temperature

tg - temperature of flue gas at boiler outlet, C

ta - temperature of air entering, C

C_{pdg} - specific heat of dry flue gas

For Solid Fuels

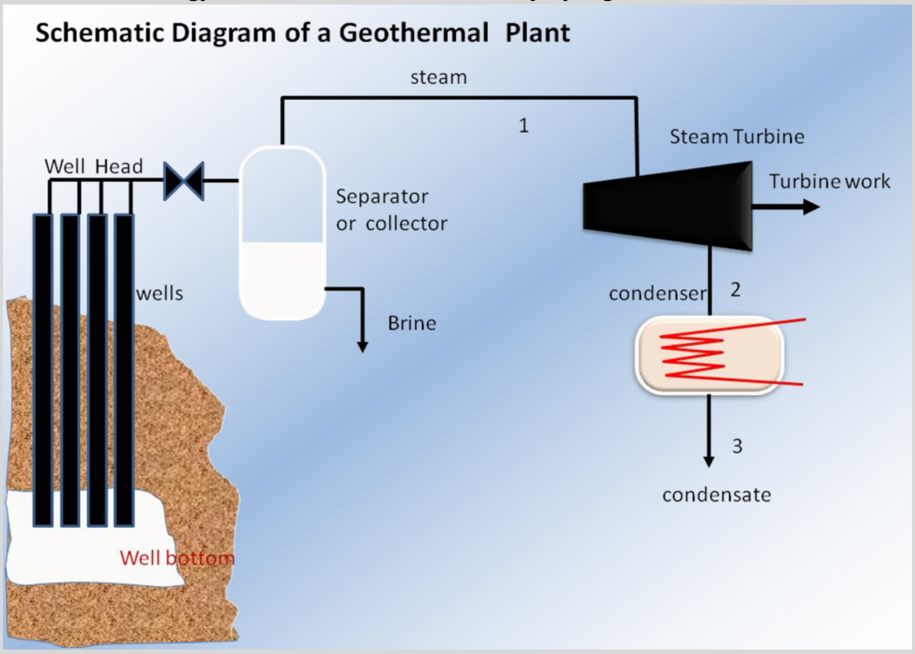
HHV = 33 820C + 144 212 (H- O/8) + 9304S KJ/kg

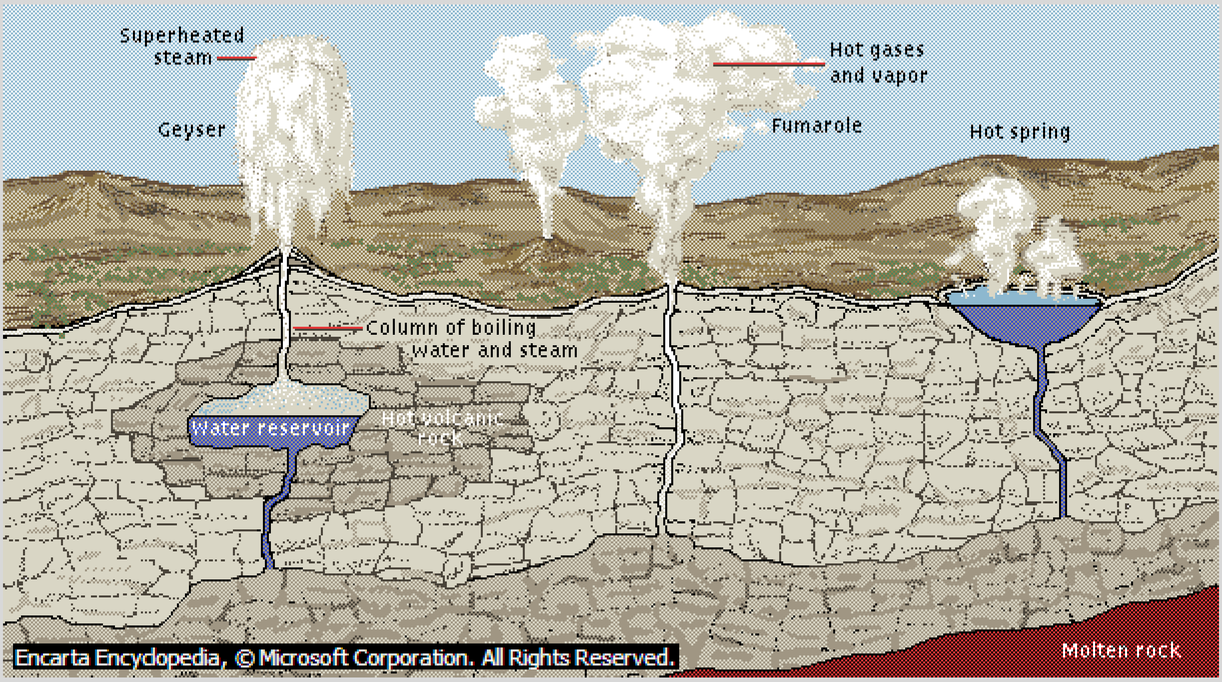
where: C, H_2, O_2, and S are in decimals from the ultimate analysis

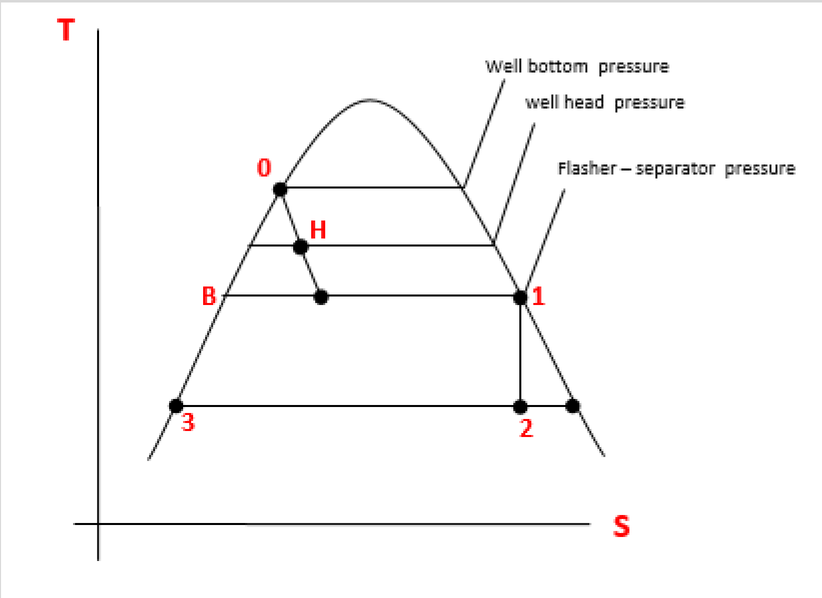
GEOTHERMAL POWER PLANT

Geothermal energy is the power obtained by using heat from the Earth's interior. Most geothermal resources are in regions of active volcanism. Hot springs, geysers, pools of boiling mud, and fumaroles (vents of volcanic gases and heated groundwater) are the most easily exploited sources of such energy

The most useful geothermal resources are hot water and steam trapped in subsurface formations or reservoirs and having temperatures ranging from 176° to 662° F (80° to 350° C). Water and steam hotter than 356° F (180° C) are the most easily exploited for electric-power generation and are utilized by most existing geothermal power plants. In these plants hot underground water is drilled from wells and passes through a separator-collector where the hot water is flashed to steam, which is then used to drive a steam turbine whose mechanical energy is then converted to electricity by a generator.







IDEAL TURBINE WORK

$$W_t = m_s(h_1 - h_2) \text{ KW}$$

ACTUAL TURBINE WORK

$$W_{t'} = \eta_T m_s(h_1 - h_2) \text{ KW}$$

GENERATOR POWER OUTPUT

$$W_0 = \eta_G \eta_T m_s(h_1 - h_2) \text{ KW}$$

where

m_s – steam flow rate in kg/sec
η_T - turbine efficiency
η_G – generator efficiency
o– underground water
H – well head condition
1 – saturated vapor condition leaving flasher – separator
B – saturated liquid condition leaving flasher – separator
2 – turbine exhaust
3 – saturated liquid leaving condenser

DRY STEAM POWER PLANT (Vapour Dominated System)

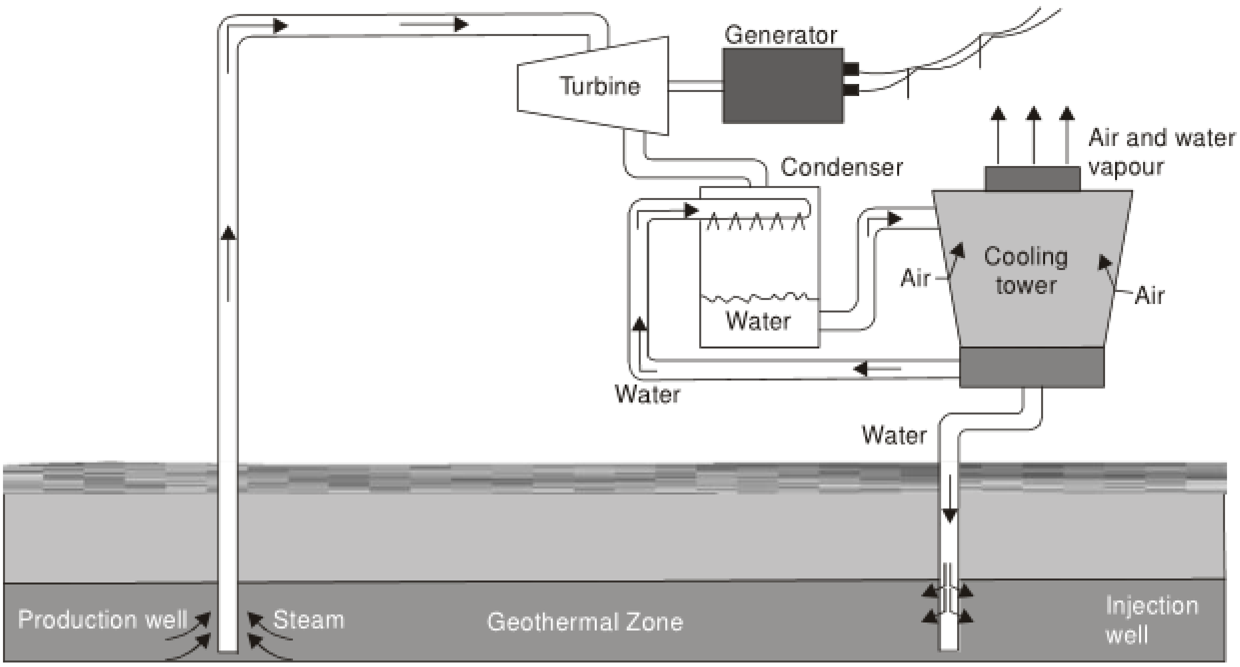


Fig. 2.34. Schematic of the Dry Steam Power Plant.

Power plants using dry steam systems were the first type of geothermal power generation plants built. They use the steam from the geothermal reservoir as it comes from wells, and route it directly through turbine/generator units to produce electricity. It is the rarest form of geothermal energy but the most suitable generation and the most developed of all geothermal resources or system.

Fig. 2.34 shows a schematic diagram of a dry steam power system also called vapour dominated system. Dry steam from the turbine at perhaps 200°C is used. It is near saturated at the bottom of the wall and may have a shut-off pressure up to 35 bar. Pressure drops through the well cause it to slightly superheated at the well head. An example of a dry steam generation operation is at the Geysers in northern California.

FLASH STEAM GEOTHERMAL POWER PLANT (Liquid Domain System)

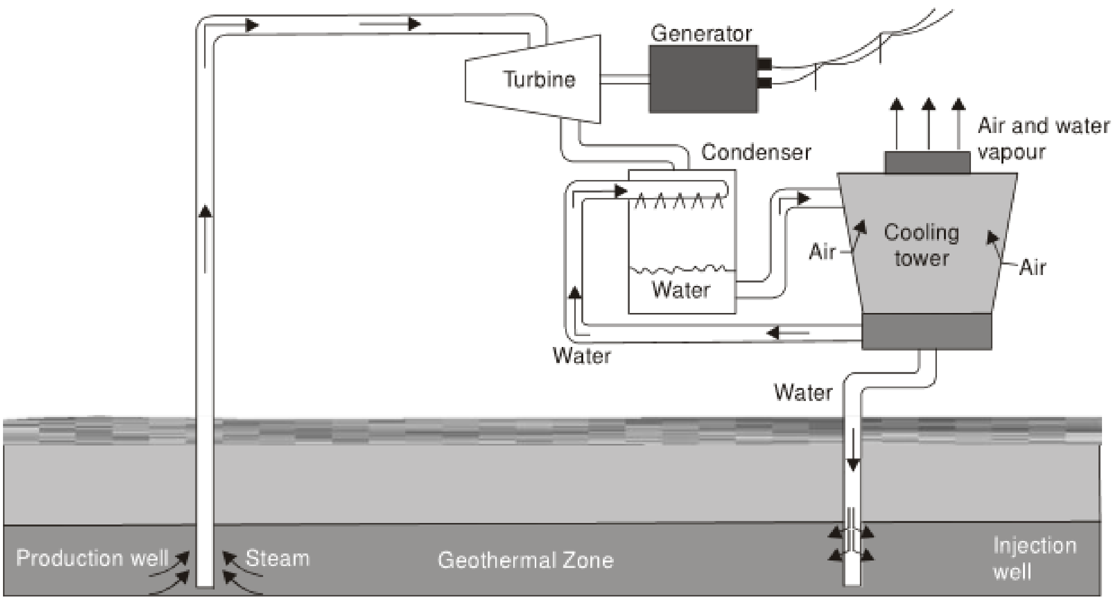


Fig. 2.34. Schematic of the Dry Steam Power Plant.

Flash Steam Power Plant (Liquid Domain System). Flash steam power plants are the most common. They use geothermal reservoirs of water with temperatures greater than 182°C. This very hot water flows up through wells in the ground under its own pressure. As it flows upward, the pressure decreases and some of the hot water boils into steam. The steam is then separated from the water and used to power a turbine/generator. Any leftover water and condensed steam are injected back into the reservoir, making this a sustainable resource.

Flash-steam power plants built in the 1980s tapped into reservoirs of water with temperatures greater than 182°C. The hot water flows up through wells in the ground under its own pressure. As it flows upward, the pressure decreases and some of the hot water "flashes" into steam. The Geysers in northern California, which uses steam piped directly from wells, produces the world's largest single

Binary Cycle Power Plant (Liquid Dominatd Systems). Binary cycle power plants operate on water at lower temperatures of about 107°–182°C. These plants use the heat from the hot water to boil a working fluid, usually an organic compound with a low boiling point. The working fluid is vaporized in a heat exchanger and used to turn a turbine. The water is then injected back into the ground to be reheated. The water and the working fluid are kept separated during the whole process, so there are little or no air emissions.

Binary cycle power plant operates on water at lower temperatures of about 107 degrees Celsius to 182 degrees Celsius. These plants use the heat from the hot water to boil a fluid, usually an organic compound with a low boiling point.

Binary cycle geothermal power generation plants differ from Dry Steam and Flash Steam systems in that the water or steam from the geothermal reservoir never comes in contact with the turbine/generator units. In the Binary system, the water from the geothermal reservoir is used to heat another "working fluid" which is vaporized and used to turn the turbine/generator units. The geothermal water, and the "working fluid" are each confined in separate circulating systems or "closed loops" and never come in contact with each other. The advantage of the binary cycle plant is that they can operate with lower temperature waters (225°F–360°F), by using working fluids that have an even lower boiling point than water. They also produce no air.

Hybrid Geothermal Power Plant-Fossil System. The concept of hybrid geothermal-fossil-fuel systems utilizes the relatively low-tem-perature heat of geothermal sources in the low-temperature end of a conventional cycle and the high-temperature heat from fossil-fuel combustion in the high-temperature end of that cycle. The concept thus combines the high-efficiency of a high-temperature cycle with a natural source of heat for part of the heat addition, thus reducing the consumption of the expensive and nonrenewable fossil fuel.

There are two possible arrangements for hybrid plants. These are

(1) Geothermal preheat, suitable for low-temperature liquid-dominated systems, and

(2) Fossil superheat, suitable for vapor-dominated and high-temperature liquid-dominated systems.

Geothermal-Preheat Hydrid Systems. In these systems the low-temperature geothermal energy is used for feed water heating of an otherwise conventional fossil-fueled steam plant. Geothermal heat replaces some, or all, of the feed water heaters, depending upon its temperature. A cycle operating on this principle is illustrated in Fig. 2.37. As shown, geothermal heat heats the feed water throughout the low-temperature end prior to an open-type deaerating heater. The DA is followed by a boiler feed pump and three closed-type feed water heaters with drains cascaded backward. These receive heat from steam bled from higher-pressure stages of the turbine. No steam is bled from the lower-pressure stages because geothermal brine fulfills this function.

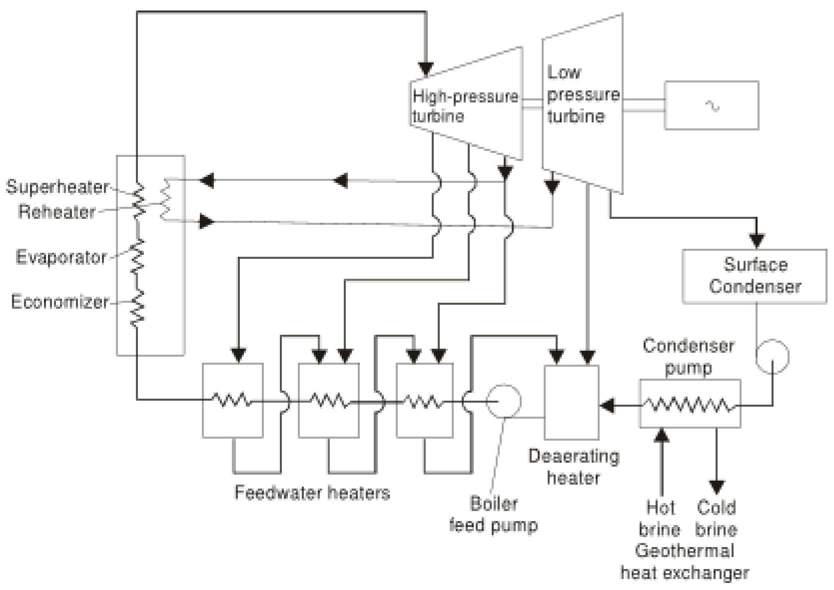


Fig 2.37. Schematic of a Geothermal-Preheat Hybrid System.

Fossil-Superheat Hybrid Systems. In these systems, the vapor-dominated steam, or the vapor obtained from a flash separator in a high-temperature liquid-dominated system, is superheated in a fossil fired super heater.

Fig. 2.38 show schematic flow. It comprises a double-flash geothermal steam system. Steam pro-duced at 4 in the first-stage flash separator is preheated from 4 to 5 in a regenerator by exhaust steam from the high-pressure turbine at 7. It is then superheated by a fossil fuel fired super heater to 6 and expands in the high pressure turbine to 7 at a pressure near that of the second stage steam separator. It than enters the regenerator, leaves it at 8, where it mixes with the lower pressure steam produces in the second stage flash separator at 15, and produces steam at 9, which expands in the lower pressure turbine to 10. The condensate at 11 is pumped and reinjected into ground at 12. The spent brine from the second stage evaporator is also reinjected in to ground at 16.

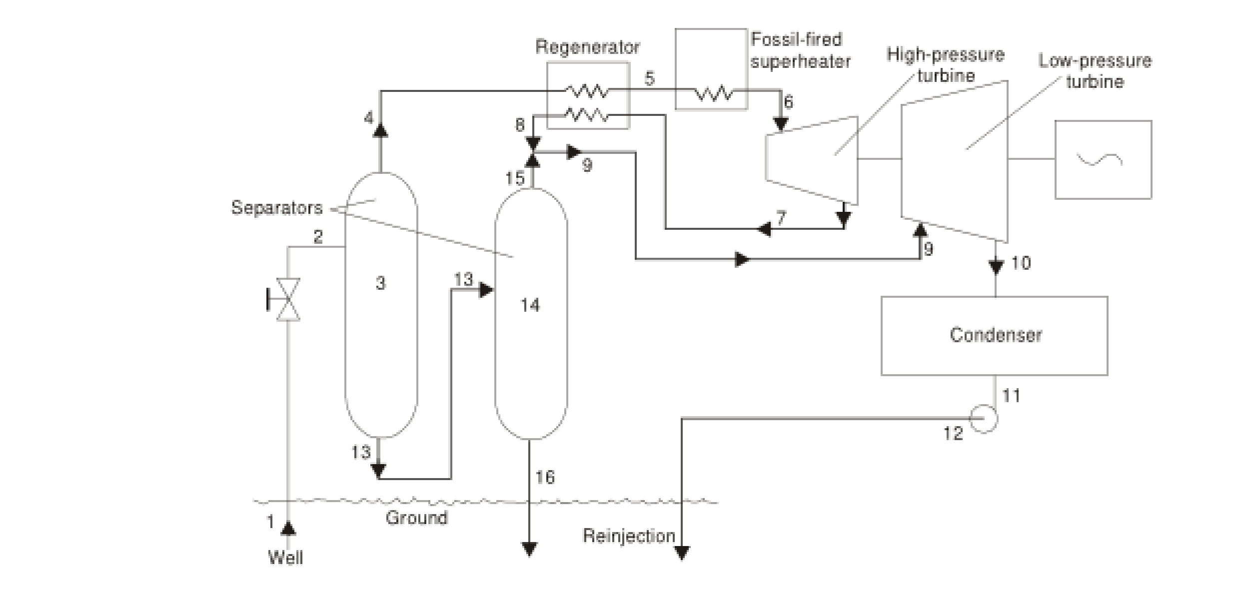


Fig. 2.38. Schematic of a fossil-superheat hybrid system with two-stage flash evaporation, regenerator, and fossil-fired super heater.

Example

A geothermal power plant has an output of 16,000 KW and mechanical - electrical efficiency of 80%. The pressurized groundwater at 17.0 MPa, 280°C leaves the well to enter the flash chamber maintained at 1.4 MPa. the flash vapor passes through the separator - collector to enter the turbine as saturated vapor at 1.4 MPa. the turbine exhaust at 0.1 MPa. The unflashed water runs to waste. If one well discharges 195,000 kg/hr of hot water, how many wells are required. (4 wells)

From Steam Table

At 17,000 KPa and 280°C

ho = 1231.7 KJ/kg

At 1400 KPa Saturated Vapor

h_1 = 2786.4 KJ/kg ; S_1 = 6.4642 KJ/kg-K

At $S_1 = S_2$ to 100 KPa

h_2 = 2341.1 KJ/kg

At 1400 KPa saturated Liquid

h_B = 829.6 KJ/kg

$$W_0 = 0.80 m_s (h_1 - h_2)$$
$$16000 = m_s (0.80)(2786.4 - 2341.1)$$
$$m_s = 45 \text{ kg/sec}$$

By mass and energy balance on the flasher - collector

$$m_o = m_s + m_B$$
$$m_B = m_o - m_s \rightarrow \text{eq. 1}$$
$$m_o h_o = m_s h_1 + (m_o - m_s) h_B$$
$$m_o = \frac{m_s (h_1 - h_B)}{h_o - h_B} = 219 \text{ kg/sec} = 788{,}400 \text{ kg/hr}$$
$$N_{wells} = \frac{788{,}400}{195{,}000} = 4 \text{ wells}$$

HYDRO ELECTRIC POWER PLANT

A. IMPULSE TYPE (Pelton type)

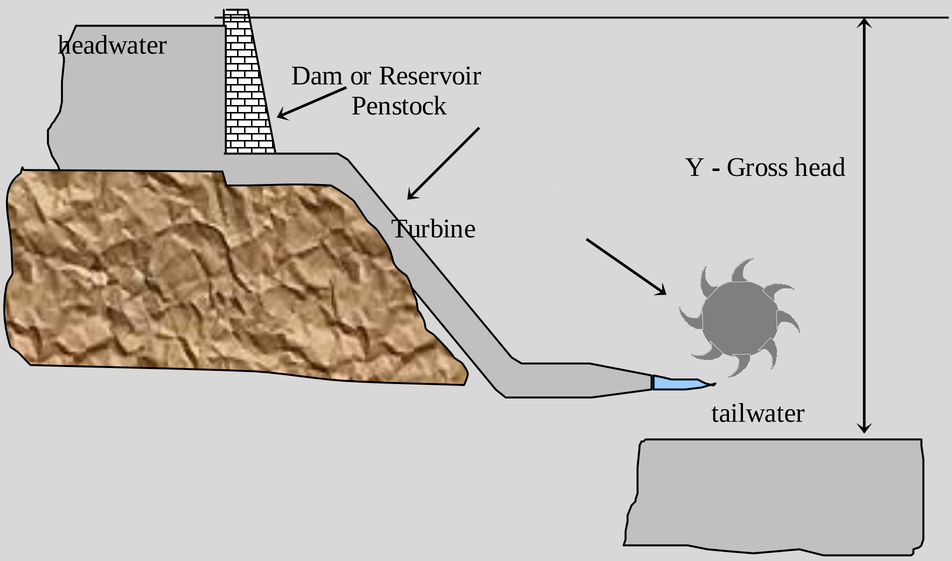


Figure 26.1 Typical PELTON WHEEL with 21 Buckets

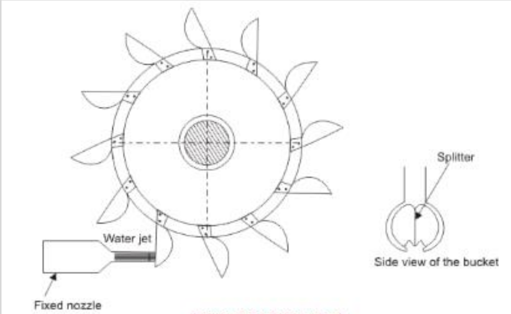
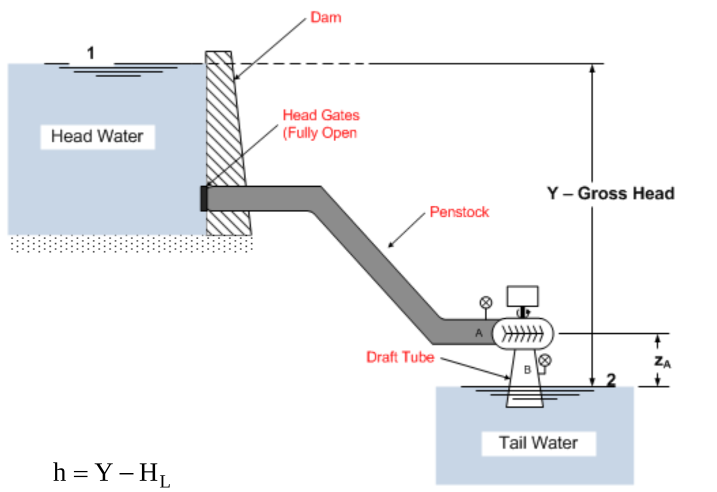


Figure 26.2 A Pelton wheel

B. REACTION TYPE (Francis Type)



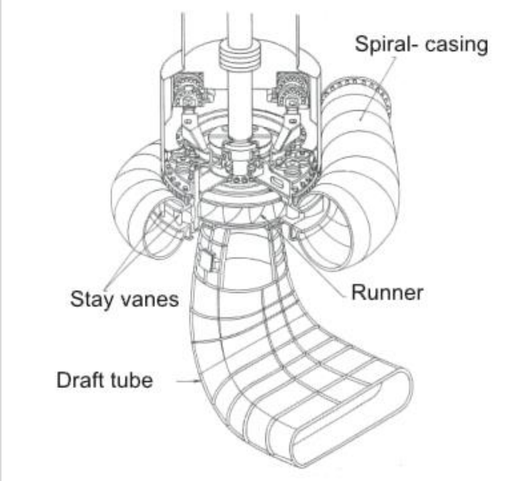


Figure 28.1 A Francis turbine

$$h = Y - H_L$$

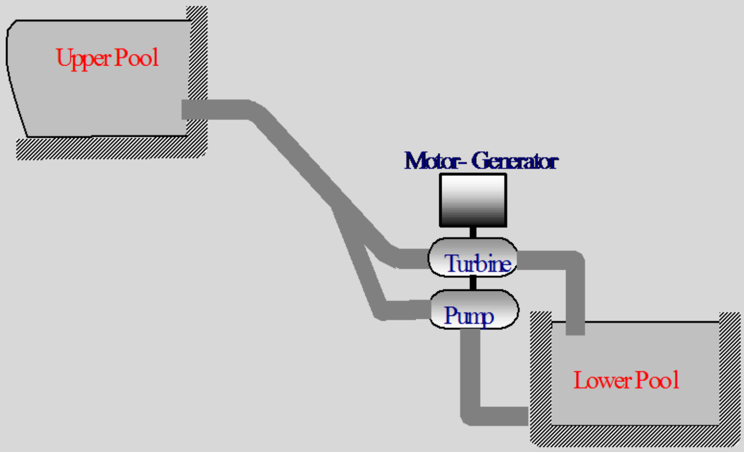
$$h = \frac{P_A}{\gamma} + \frac{V_A^2}{2g} + Z_A$$

$$V = \frac{Q}{A}$$

$$A = \frac{\pi}{4} D^2$$

D - penstock diameter, m
Y - Gross head, m
V_B - velocity at inlet, m/sec
A - area of penstock, m^2
H_L - head loss, m
Z_A - turbine setting above tail - water level, m

PUMP STORAGE HYDRO-ELECTRIC PLANT



How turbines work
The rotor is the rotating part of a turbine. In a simple turbine, it consists of a disc or wheel mounted on an axle. The axle sits either horizontally or vertically. The wheel has curved blades or buckets around the edges. Nozzles or movable gates called guide vanes aim the fluid at the blades or buckets and adjust its speed. In many turbines, a casing encloses the rotor. The casing holds the fluid against the rotor so that none of the fluid's energy is lost.
As a fluid passes through a turbine, it hits or pushes against the blades or buckets and causes the wheel to turn. When the wheel rotates, the axle turns with it. The axle is connected directly or through a series of gears to an electric generator, air compressor, or other machine. Thus, the circular motion of the spinning rotor drives a machine.
The rotors of some turbines have only one wheel. However, the rotors of others have as many as 50 or more. Multiple wheels increase the efficiency of turbines, because each wheel extracts additional energy from the moving fluid. In a turbine with more than one wheel, the wheels are mounted on a common axle, one behind the other. A stationary ring of curved blades is attached to the inside of the casing in front of each wheel. These stationary blades direct the flow of the fluid toward the wheels. A wheel and a set of stationary blades is called a stage. Multistage turbines are those that have many stages.
Kinds of turbines
Turbines are sometimes classified according to their principle of operation. All turbines operate by impulse or reaction, or by a combination of these principles. In an impulse turbine, the force of a fast-moving fluid striking the blades makes the rotor spin. In a reaction turbine, the rotor turns primarily as a result of the weight or pressure of a fluid on the blades.
Turbines are more commonly classified by the type of fluid that turns them. According to this method, there are four main kinds of turbines: (1) water turbines, (2) steam turbines, (3) gas turbines, and (4) wind turbines.
Water turbines are also called hydraulic turbines. Most water turbines are driven by water from waterfalls or by water that is stored behind dams. The turbines are used primarily to power electric generators at hydroelectric power plants. There are three main kinds of water turbines: (1) the Pelton wheel, (2) the Francis turbine, and (3) the Kaplan turbine. The type of water turbine used at a plant depends on the head available. A head is the distance the water falls before it strikes the turbine. Heads range from about 2.5 meters to more than 300 meters.
The Pelton wheel is an impulse turbine. It is used with heads of more than 300 meters. A Pelton's rotor consists of a single wheel mounted on a horizontal axle. The wheel has cup-shaped buckets around its perimeter. Water from a lake or reservoir drops toward the turbine through a long pipe called a penstock. One to six nozzles at the end of the penstock increase the water's velocity and aim the water toward the buckets. The force of these high-speed jets of water against the buckets turns the wheel.
The Francis turbine is used when the head is between about 30 meters and 300 meters. A Francis turbine's rotor is enclosed in a casing. Its wheel has as many as 24 curved blades. Its axle is vertical. The wheel of a Francis turbine operates underwater. It is encircled by a ring of guide vanes, which can be opened or closed to control the amount of water flowing past the wheel. The spaces between the vanes act as nozzles to direct the water toward the center of the wheel. The rotor is turned chiefly by the weight or pressure of the flowing water.
The Kaplan turbine is used for heads of less than 30 meters. The Kaplan rotor resembles a ship's propeller. It has from three to eight blades on a vertical axle. It works in a manner similar to that of a Francis turbine. Both the Kaplan turbine and the Francis turbine are reaction turbines.

FUNDAMENTAL EQUATIONS

1. Total dynamic head or Net effective head

a. For an Impulse type

$$h = Y - H_L$$

Y - Gross head at plant

Gross head - difference in elevation between head water level and tail water level.

b. For a Reaction type

$$h = Y - H_L$$

$$h = \frac{P_A}{\gamma} + \frac{V_A^2}{2g} + Z_A$$

$$V = \frac{Q}{A}$$

$$A = \frac{\pi}{4} D^2$$

where:

P_B - pressure at turbine inlet in KPa

V_B - velocity of water at penstock, m/sec

2. Discharge or Rate of Flow (Q)

$$Q = AV \text{ m}^3/\text{sec}$$

$$A = \frac{\pi}{4} D^2$$

where: D - diameter of penstock

3. Water Power (WP)

$$WP = Q\gamma h \quad KW$$

4. Brake Power (BP)

$$BP = \frac{2\pi TN}{60,000} \ KW$$

where:

T - brake torque N-m

N - no. of RPM

5. Head loss

$$H_L = \frac{fLV^2}{2gD} \ \text{meters}$$

f - Moody friction factor

L - length of penstock

6. Turbine Efficiency (e)

$$e = e_h e_m e_v$$

$$e = \frac{BP}{WP} \times 100\%$$

where: e_h - hydraulic efficiency

e_m - mechanical efficiency

e_v - volumetric efficiency

7. Generator Efficiency (η_g)

$$\eta_g = \frac{GP}{BP} \times 100\%$$

where: GP - electrical output of the generator, KW

8. Rotative Speed (N)

$$N = \frac{120f}{n} \ RPM$$

where:

n - number of generator poles (usually divisible by 4)

9. Turbine Specific Speed

$$Ns = \frac{N\sqrt{BP}}{3.813(h)^{5/4}} \ RPM$$

10. Wheel Diameter

$$D = \frac{60\varphi\sqrt{2gh}}{\pi N} \ \text{meters}$$

where

φ – peripheral velocity factor

TERMS AND DEFINITION

Reservoir - stores the water coming from the upper river or waterfalls.

Headwater - the water in the reservoir or upper pool.

Spillway - a weir in the reservoir which discharges excess water so that the head of the plant will be maintained.

Dam - the concrete structure that encloses the reservoir to impound water.

Silt Sluice - a chamber which collects the mud and through which the mud is discharged.

Trash Rack - a screen which prevents the leaves, branches and other water contaminants to enter into the penstock.

Valve - opens or closes the entrance of the water into the penstock.

Surge Chamber - a standpipe connected to the atmosphere and attached to the penstock so that the water will be at atmospheric pressure.

Penstock - a channel or a large pipe that conducts the water from the reservoir to the turbine.

Turbine - a device or a machine that converts the energy of the water to mechanical energy.

Generator - a device or a machine that converts mechanical energy of the turbine into electrical energy.
Draft Tube - a pipe that conducts the water from the turbine to the tailrace so that the turbine can be set above the tail water level.
Tailrace - is the canal that is used to carry the water away from the plant.
Undershot wheel - water enters at the bottom of the wheel tangential to its periphery and impinges on the buckets or vanes.
Breast shot wheel - a wheel used for heads up to 16 ft, where water enters between the bottom and top of the wheel at an angle and is prevented from leaving the wheel by a breast wall on the side of the wheel.
Over shot wheel - a wheel used for high heads, where water enters the wheel at the top by being discharged from a flume.
Gross head - is the difference between the headwater and tail water elevation.
Spiral case - it conducts the water around a reaction type turbine.

SAMPLE PROBLEMS

1. A pelton type turbine was installed 30 m below the head gate of the penstock. The head loss due to friction is 15% of the given elevation. The length of the penstock is 80 m and the coefficient of friction is 0.00093. Determine
   a) the diameter of the penstock in mm. (421.6 mm)
   b) the power output in KW (781.234 KW)
2. What power in KW can be developed by the impulse turbine shown if the turbine efficiency is 85%. Assume that the resistance coefficient f of the penstock is 0.015 and the head loss in the nozzle itself is negligible. What will be the speed of the wheel , assuming ideal conditions where $V_{JET} = 2V_{BUCKET}$ and what torque will be exerted on the turbine shaft.

3. A hydroelectric plant has a 20 MW generator with an efficiency of 96%. The generator is directly coupled to a vertical Francis type hydraulic turbine having an efficiency of 80%.The total gross head of the turbine is 150 m while the loss of head due to friction at the penstock up to the turbine inlet flange is 4% of gross head. the runaway speed is not to exceed 750 RPM, determine:
   a) Brake horsepower rating of the turbine (27 927 hp)
   b) the flow of water through the turbine in cfs (650.53 cfs)
   c) check if the specific speed falls under that of Francis type turbine. (Ns = 29)
   d) the rated speed of the turbine (N = 400 RPM)
4. A Francis turbine is installed with a vertical draft tube. The pressure gauge located at the penstock leading to the turbine casing reads 372.6 KPa and velocity of water at inlet is 6 m/sec. The discharge is 2.5 m^3/sec. The hydraulic efficiency is 85%, and the overall efficiency is 82%. The top of the draft tube is 1.5 m below the centerline of the spiral casing, while the tailrace level is 2.5 m from the top of the draft tube. There is no velocity of whirl at the top or bottom of the draft tube and leakage losses are negligible. Calculate,
   a) the net effective head in meters (43.817 m)
   b) the brake power in kw. (881.2 kw)
   c) the plant output for a generator efficiency of 92%. (810.7 kw)
   d) the mechanical efficiency (96.550)
5. A hydroelectric power plant using a Francis type turbine has the following data:
   Headwater elevation - 190 m
   Tailwater elevation - 50 m
   Head loss due to friction - 3.5% of gross head
   Turbine discharge at full gate opening - 6 m^3/sec
   Turbine – Generator Speed – 600 RPM
   Turbine efficiency at rated capacity - 90%
   Turbine is to be direct connected to a 60 hertz a-c generator

   Determine:
   a) the brake power in kw (7156.8 kw)
   b) the number of generator poles (12 poles)
   c) the electrical power output of the generator if the efficiency is 94% (6727.4 kw)
   d) the torque developed in N-m (102 925.31 N-m)
   e) the approximate length of the penstock, if the diameter is 1.5 m and friction factor f is 0.018. (693 m)

6. The flow of a river is 21.25 m^3/sec and the head on the site is 30.5 m. It is proposed to developed the maximum capacity at the site with the installation of two turbines, one of which is twice the capacity of the other. The efficiency of both units is assumed to be 85%. Determine:
   a) Rotative speed of each unit in rpm if the specific of both is 70 rpm.
   b) Brake power of each unit in kw.
   c) Number of poles of the generator for 60 cycle current

7. The flow of a river is 21 m^3/sec and the head at the power site is 24.4 m (≅80 ft). It is proposed to developed this site with an installation of 3-turbines, 2 similar units and another of half their size, all having the same efficiency of 85%. Find the rotative speed of all these units. (348 rpm; 492 rpm)

   Ns at 80 ft head = 70 rpm
8. A hydroelectric power plant discharging water at the rate of 0.75 m^3/sec and entering the turbine at 0.35 m/sec with a pressure of 275 KPag has a runner of 55 cm internal diameter. Speed is 514 rpm at 260 BHP. The casing is 2 m above the tailwater level. Calculate:
   a) the net effective head in m (30.039 m)
   b) the peripheral coefficient (0.61)
   c) the efficiency (87.8%)
9. A Mindanao province where a mini-hydroelectric plant is to be constructed has an average annual rainfall of 139 cm. The catchment's area is 206 km^2 with an available head of 23 m.Only 82% of the rainfall can be collected and 75% of the impounded water is available for power. Hydraulic friction loss is 6%, turbine efficiency is 78% and generator efficiency is 93%. Determine the average kw power that could be generated for continuous operation.

$$\text{POWER} = \frac{1.39(206)(1000)^2(0.82)(0.75)(9.81)(23)(1-.06)(0.78)(0.93)}{8760(3600)}$$

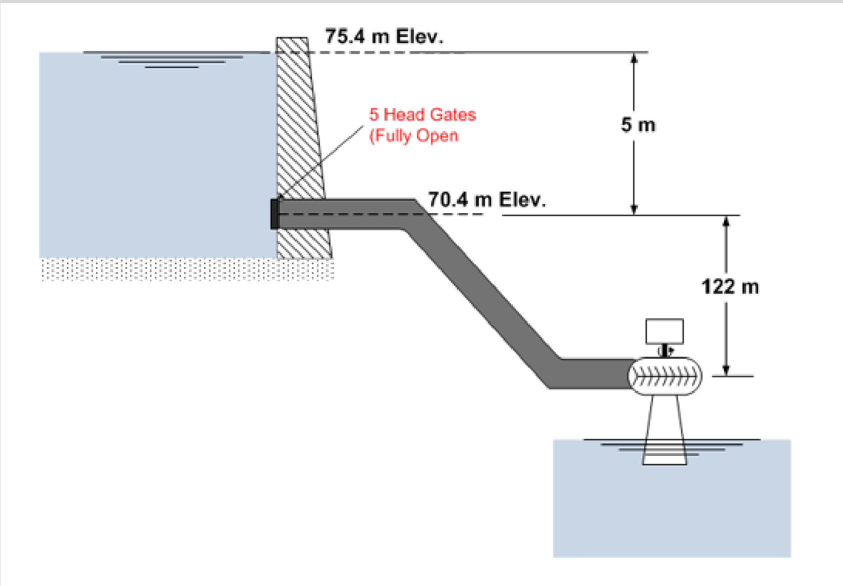
$$= 859 \text{ kw}$$

10. A Mindanao province where a hydroelectric power station is to be constructed has an average annual rainfall of 1.93 m. The catchment's area 260 km^2 with an available head of 32 m. Only 82% of the rainfall can be collected and 5/8 of the impounded water is available for power.The load factor of the station is 80%, the penstock efficiency is 94%, turbine efficiency 78% and the generator efficiency is 88%.

    a) Calculate the average power that could be generated
    b) Calculate the installed capacity of the station
    c) Assuming no standby unit, give the possible number and size of the units

    Average power = load factor(peak power)
11. The difference in elevation between the surfaces of water in the storage reservoir and the intake to a turbine was 40.4 m. During a test the pressure head at the later point was 38.6 m and the discharge was1.25 m/sec. The inside diameter of the penstock is762 mm.
    a) What is the efficiency of the pipeline
    b) What was the power delivered to the turbine in KW
12. The difference in elevation between the source of the water supply and the centerline at the base of the nozzle of a pelton wheel is 373 m. During a test the pressure at the end of the nozzle was 3,520 KPa when the flow was 1.3 m^3/sec. Inside diameter of the penstock is 762 mm. Compute:
    a) What power at the base of the nozzle.
    b) Shaft power developed by the pelton wheel if efficiency is 80%.

13. In a hydroelectric power plant, the head water surface on the dam is at elevation 75.4 m while the water surface just at the outlet of the head gate is at elevation 70.4 m. The head gate has 5 gates of 1 m x 1 m leading to the penstock and are fully opened. Turbine is 122 m below the entrance of the penstock with an efficiency of 90%. Assuming 60% as coefficient of discharge, and Generator efficiency of 87% ,determine the Megawatt output of electrical power produced by the plant.



14. A hydro - storage plant has a 20,000 KW rated capacity, with a utilization factor of 76%. For a 1 1/2 hour peak, determine the hydraulic impoundment in cubic meters of water required with friction factor loss of 6%. Dam elevation at 40 m and hydraulic turbine elevation of 14 m and tailrace at elevation of 7 m. Generator efficiency is 92% and turbine efficiency of 85%, evaporation factor of 20%. If the water is pumped from the lower reservoir with friction factor of 8%, pump efficiency of 76% and motor efficiency of 85%.
    a) How many KW-hr of power is required to pump the water required to carry the peak
    b) What is the overall thermal efficiency of the hydro plant.
15. A remote community in the mountain province plans to put up a small hydroelectric plant to service six closely located barangays estimated to consume 52,650,000 KW-hrs per annum. Expected flow of water is 28 m^3/sec. The most favorable location for the plant fixes the tail water level at 480 m. The manufacturer of the turbine generator set have indicated the following performance data:
    Turbine efficiency - 87&
    Generator Efficiency - 92%
    Loss in headwork will not exceed 3.8% of the available head

    In order to pinpoint the most suitable area for the dam, determine
    a. Headwater elevation
    b. Type of turbine to be used
    c. Synchronous speed of generator if number of poles is 6, and frequency is 60 hertz.
16. A proposed hydro electric power plant has the following data:
    Elevation of normal headwater surface - 194 m
    Elevation of normal tailwater surface - 60 m
    Loss of head due to friction - 6.5 m
    Turbine discharge at full gate opening - 5 m^3/sec
    Turbine efficiency at rated capacity - 90%
    Turbine is direct connected to a 60 cycle AC generator

    Required:
    a. What type of hydraulic turbine would you specify? (Francis)
    b. Find the Brake Power of the turbine in KW (5628.5 KW)
    c. Find the number of poles of the generator if Ns = 33 RPM (10 poles)
    d. Find the KW output of the direct-connected generator if the efficiency is 94% (5,291 KW)
17. The flow of a river of 22 m^3/sec produces a total brake power of 5400 KW. It is proposed to install two turbines one of which is twice the capacity of the other. The efficiency and specific speed of both units are assumed to be 85% and 70 RPM, respectively. Determine
    a. Head in meters (30.5 m)
    b. Rotative speed of each unit (450 RPM; 320 RPM)
    c. Number of poles of generator if f = 60 Hz. (16 Poles ; 24 Poles)
18. At a proposed hydro-electric power plant site, the average elevation of the headwater is 600 m, the tailwater elevation is 480 m. Average annual water flow is determined to be equal to that volume flowing through a rectangular channel 4 m wide and0.5 m deep and average velocity of 5.5 m/sec. Assuming that the plant will operate 350 days per year, find the annual energy in KW-HR that the power site can develop if the hydraulic turbine that will be used has an efficiency 80% and generator efficiency of 92%. Consider a headwork loss of 4% of the available head. (76,854,851 KW-hr)

I

A Francis turbine is installed with a vertical draft tube. The pressure gauge located at the penstock leading to the turbine casing reads 372.6 KPa and velocity of water at inlet is 6 m/sec. The discharge is 2.5 m^3/sec. The hydraulic efficiency is 85%, and the overall efficiency is 82%. The top of the draft tube is 1.5 m below the centerline of the spiral casing, while the tailrace level is 2.5 m from the top of the draft tube. There is no velocity of whirl at the top or bottom of the draft tube and leakage losses are negligible. Calculate,

a) the net effective head in meters
b) the brake power in KW
c) the plant output for a generator efficiency of 92%.
d) the mechanical efficiency

II

The discharge gauge of a centrifugal pump handling potable water for a class A subdivision reads 175 KPa, while the suction gauge indicates a vacuum of 305 mm Hg. The discharge pressure gauge is 10 m above the pump centerline and the point of attachment of the suction gauge is 3 m below the centerline. The diameters of the suction and discharge pipes are 76 mm and 63.5 mm, respectively. Assuming a head loss of 6 m, and a pump-motor efficiency of 75, Calculate the power required if the flow is 12.5 L/sec of water. (γ = 9.81 KN/m^3)

DIESEL POWER PLANT

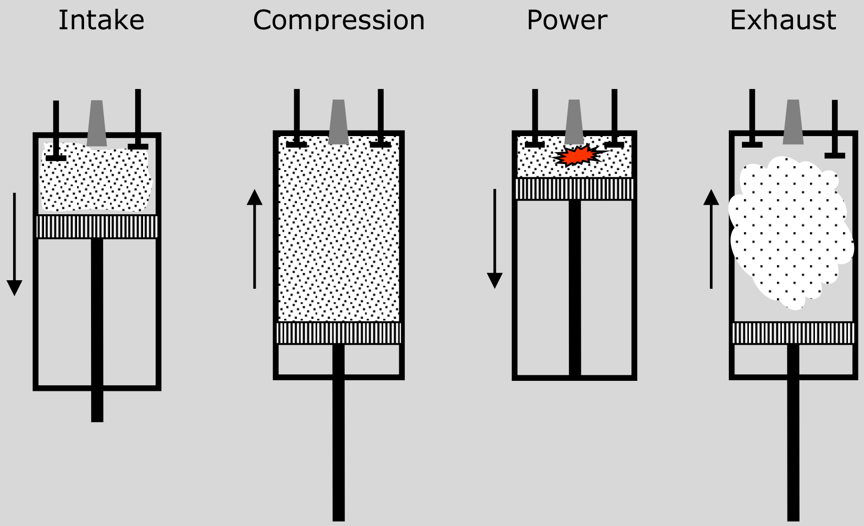
Diesel engine is a type of internal combustion engine that uses low grade fuel oil and which burns this fuel inside the cylinder by heat of compression. It is used chiefly for heavy-duty work. Diesel engines drive huge freight trucks, large buses, tractors, and heavy road-building equipment. They are also used to power submarines and ships, and the generators of electric-power stations in small cities. Some motor cars are powered by diesel engines.

Gasoline engine - is a type of internal combustion engine, which uses high grade of oil. It uses electricity and spark plugs to ignite the fuel in the engine's cylinders.

Kinds of diesel engines. There are two main types of diesel engines. They differ according to the number of piston strokes required to complete a cycle of air compression, exhaust, and intake of fresh air. A stroke is an up or down movement of a piston. These engines are (1) the four-stroke cycle engine and (2) the two-stroke cycle engine.

Four Stroke Cycle Engine

1. Intake
2. Compression
3. Power
4. Exhaust

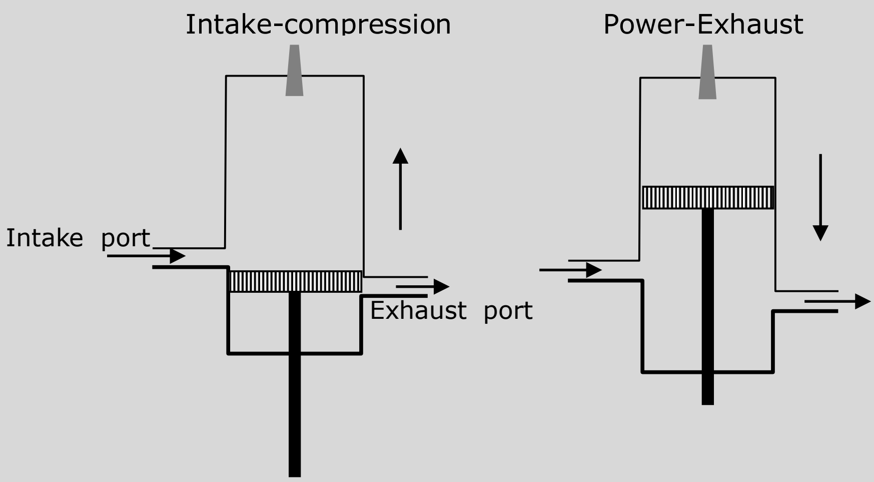


In a four-stroke engine, each piston moves down, up, down, and up to complete a cycle. The first down stroke draws air into the cylinder. The first upstroke compresses the air. The second down stroke is the power stroke. The second upstroke exhausts the gases produced by combustion. A four-stroke engine requires exhaust and air-intake valves.

It completes one cycle in two revolutions of the crankshaft.

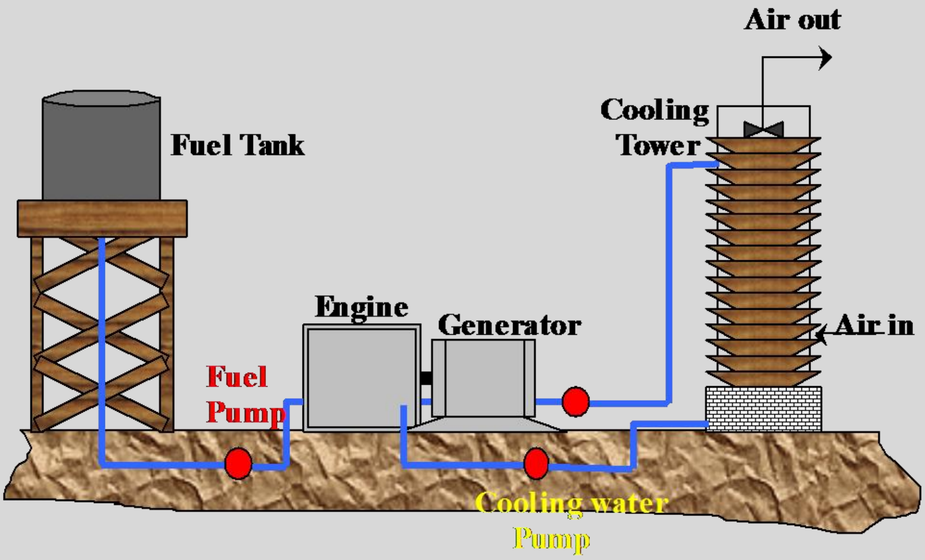
Two Stroke Cycle Engine

1. Intake-Compression stroke
2. Power-exhaust stroke



In a two-stroke engine, the exhaust and intake of fresh air occur through openings in the cylinder near the end of the down stroke, or power stroke. The one upstroke is the compression stroke. A two-stroke engine does not need valves. These engines have twice as many power strokes per cycle as four-stroke engines, and are used where high power is needed in a small engine. It completes one cycle in one revolution of the crankshaft.

INTERNAL COMBUSTION ENGINE POWER PLANT



Two stroke cycle engine: An engine that completes one cycle in one revolution of the crankshaft.
Four stroke cycle engine: An engine that completes one cycle in two revolution of the crankshaft.

ENGINE PERFORMANCE

HEAT SUPPLIED BY FUEL

$$Qs = m_F(HV)\,\frac{KJ}{hr}$$

where:
m_f - fuel consumption in kg/hr
HV - heating value of fuel in KJ/kg

INDICATED POWER

$$IP = \frac{P_{mi}\pi LD^2 Nn'}{4(60)}\ KW$$

where:
P_{mi} - indicated mean effective pressure in KPa
L - length of stroke in m
D - diameter of bore in m
n' - no. of cylinders
IP - indicated power in KW

$$N = (RPM) \rightarrow (\text{For 2 Stroke - Single acting})$$
$$N = 2(RPM) \rightarrow (\text{For 2 Stroke - Double acting})$$
$$N = \frac{(RPM)}{2} \rightarrow (\text{For 4 Stroke - Single acting})$$
$$N = (RPM) \rightarrow (\text{For 4 Stroke - Double acting})$$

BRAKE POWER

$$IP = \frac{P_{mb}\pi LD^2 Nn'}{4(60)}\ KW$$

where:
P_{mb} - brake mean effective pressure in KPa

$$N = (RPM) \rightarrow (\text{For 2 Stroke - Single acting})$$
$$N = 2(RPM) \rightarrow (\text{For 2 Stroke - Double acting})$$
$$N = \frac{(RPM)}{2} \rightarrow (\text{For 4 Stroke - Single acting})$$
$$N = (RPM) \rightarrow (\text{For 4 Stroke - Double acting})$$

$$BP = \frac{2\pi TN}{60,000}\ KW$$

where:
T - brake torque in N-m
N = RPM

4. FRICTION POWER

$$FP = IP - BP$$

5. INDICATED MEAN EFFECTIVE PRESSURE

$$Pmi = \frac{AS'}{L'}\ KPa$$

where:
A - area of indicator card, m^2
S - spring scale, KPa/m
L' - length of indicator card, m

6. BRAKE TORQUE

$$T = (P - Tare)R\ \ N\text{-}m$$

where:
P - Gross load on scales, N
Tare - tare weight, N
R - length of brake arm, m

7. PISTON SPEED

$$PS = 2LN\ \frac{m}{min}$$

8. DISPLACEMENT VOLUME

$$V_D = \frac{\pi L D^2 N n'}{4(60)}\ KPa$$

$$V_D = \frac{IP}{P_{mi}}\ KPa$$

$$V_D = \frac{BP}{P_{mb}}\ KPa$$

Where

$$N = (RPM) \rightarrow (\text{For 2 Stroke - Single acting})$$

$$N = 2(RPM) \rightarrow (\text{For 2 Stroke - Double acting})$$

$$N = \frac{(RPM)}{2} \rightarrow (\text{For 4 Stroke - Single acting})$$

$$N = (RPM) \rightarrow (\text{For 4 Stroke - Double acting})$$

9. SPECIFIC FUEL CONSUMPTION

a. Indicated specific fuel consumption

$$m_{fi} = \frac{m_f}{IP}\ \frac{kg}{KW\text{-}hr}$$

b. Brake specific fuel consumption

$$m_{fb} = \frac{m_f}{BP}\ \frac{kg}{KW\text{-}hr}$$

c. Combined specific fuel consumption

$$m_{fc} = \frac{m_f}{GP}\ \frac{kg}{KW\text{-}hr}$$

where: GP - Generator output

10. HEAT RATE (HR): Heat supplied divided by the KW produced.

a. Indicated heat rate

$$HRi = \frac{Q_s}{IP} \quad \frac{KJ}{KW\text{-}hr}$$

b. Brake heat rate

$$HRb = \frac{Q_s}{BP} \quad \frac{KJ}{KW\text{-}hr}$$

c. Combined heat rate

$$HRc = \frac{Q_s}{GP} \quad \frac{KJ}{KW\text{-}hr}$$

11. GENERATOR SPEED

$$N = \frac{120f}{n} \quad \text{RPM}$$

WHERE: n - number of generator poles (usually divisible by 4)

12. MECHANICAL EFFICIENCY

$$\eta_m = \frac{BP}{IP} \text{ x } 100\%$$

13. GENERATOR EFFICIENCY

$$\eta_g = \frac{GP}{BP} \text{ x } 100\%$$

14. Indicated Thermal Efficiency

$$e_i = \frac{3600(IP)}{Q_s} \text{ x } 100\%$$

15. Brake Thermal Efficiency

$$e_b = \frac{3600(BP)}{Q_s} \text{ x } 100\%$$

16. Combined Thermal efficiency

$$e_c = \frac{3600(GP)}{Q_s} \text{ x } 100\%$$

17. Indicated Engine Efficiency

$$\eta_i = \frac{e_i}{e} \text{ x } 100\%$$

18. Brake Engine Efficiency

$$\eta_b = \frac{e_b}{e} \text{ x } 100\%$$

where:

e - cycle thermal efficiency

19. Volumetric Efficiency

$$\eta_v = \frac{\text{Actual volume of air entering}, \frac{m^3}{sec}}{\text{Displacement Volume}, \frac{m^3}{sec}} \text{ x } 100\%$$

20. Correction Factor for Nonstandard condition

a. Considering temperature and pressure effect

$$P_h = P_s\left(\frac{B_S}{B_h}\right)\left(\sqrt{\frac{T_h}{T_s}}\right)$$

b. Considering temperature effect alone

$$P_h = P_s\left(\sqrt{\frac{T_h}{T_s}}\right)$$

c. Considering pressure effect alone

$$P_h = P_s\left(\frac{B_S}{B_h}\right)$$

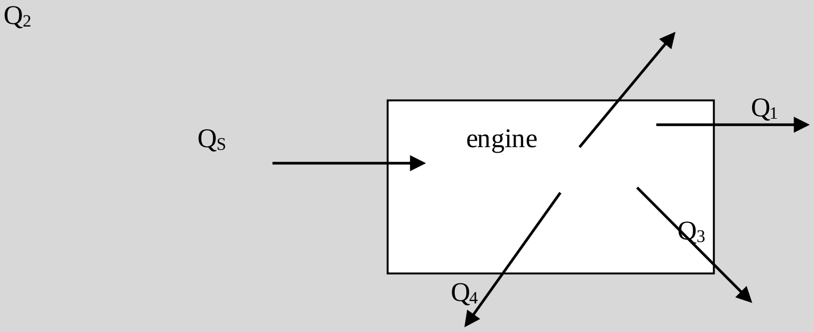
Note: From US Standard atmosphere:

$$B_h = \left(B_s - \frac{83.312h}{1000}\right) \text{ mm Hg}$$

$$T_h = \left(T_s - \frac{6.5h}{1000}\right) °K$$

where: B - barometric pressure, mm Hg
T - absolute temperature, K
h - at elevation h condition
s - at sea level condition

21. ENGINE HEAT BALANCE: The total heat supplied to th engine was broken down into four heat items.



$Q_S = Q_1 + Q_2 + Q_3 + Q_4$

Q_1 - heat converted to useful work
Q_2 - heat lost to cooling water
Q_3 - heat lost to exhaust gases
Q_4 - heat lost due to friction, radiation and unaccounted for

$Q_1 = 3600(BP)$ KJ/hr
$Q_2 = m_wCp_w(t_{wo} - t_{wi})$ KJ/hr
$Q_3 = Q_a + Q_b$ KJ/hr
$Q_a = m_gC_{pg}(t_g - t_a)$ KJ/hr
$Q_b = m_f(9H_2)(2442.7)$ KJ/hr
$Q_4 = Q_S - (Q_1 + Q_2 + Q_3)$ KJ/hr
$H_2 = 0.26 - 0.15S$ kgH/kgfuel

$$S = \frac{141.5}{131.5 + °API}$$

$$S = \frac{140}{130 + °Bel}$$

where:
Q_a - sensible heat of products of combustion
Q_b - heat required to evaporate and superheat moisture formed from the combustion of hydrogen in the fuel
t_g - temperature of flue gas, °C
t_a - temperature of air, °C
H_2 - amount of hydrogen in the fuel kg H/kg fuel

TERMS AND DEFINITIONS

Diesel engine is a type of internal combustion engine that uses low grade fuel oil and which burns this fuel inside the cylinder by heat of compression. It is used chiefly for heavy-duty work. Diesel engines drive huge freight trucks, large buses, tractors, and heavy road-building equipment. They are also used to power submarines and ships, and the generators of electric-power stations in small cities. Some motor cars are powered by diesel engines.

Gasoline engine - is a type of internal combustion engine, which uses high grade of oil. It uses electricity and spark plugs to ignite the fuel in the engine's cylinders.
Kinds of diesel engines. There are two main types of diesel engines. They differ according to the number of piston strokes required to complete a cycle of air compression, exhaust, and intake of fresh air. A stroke is an up or down movement of a piston.These engines are (1) the four-stroke cycle engine and (2) the two-stroke cycle engine.

Four Stroke Cycle Engine

1. Intake
2. Compression
3. Power
4. Exhaust

In a four-stroke engine, each piston moves down, up, down, and up to complete a cycle. The first down stroke draws air into the cylinder. The first upstroke compresses the air. The second down stroke is the power stroke. The second upstroke exhausts the gases produced by combustion. A four-stroke engine requires exhaust and air-intake valves.
It completes one cycle in two revolutions of the crankshaft.
Two Stroke Cycle Engine

1. Intake-Compression stroke
2. Power-exhaust stroke

In a two-stroke engine, the exhaust and intake of fresh air occur through openings in the cylinder near the end of the down stroke, or power stroke. The one upstroke is the compression stroke. A two-stroke engine does not need valves. These engines have twice as many power strokes per cycle as four-stroke engines, and are used where high power is needed in a small engine. It completes one cycle in one revolution of the crankshaft.

Governor - is a device used to govern or control the speed of an engine under varying load conditions.
Purifier - a device used to purify fuel oil and lube oil.
Generator - a device used to convert mechanical energy.
Crank scavenging - is one that the crankcase is used as compressor.
Thermocouple - is made of rods of different metal that are welded together at one end.
Centrifuge - is the purification of oil for separation of water.

Unloader - is a device for automatically keeping pressure constant by controlling the suction valve.
Planimeter - is a measuring device that traces the area of actual P-V diagram.
Tachometer - measures the speed of the engine.
Engine indicator - traces the actual P-V diagram.
Dynamometer - measures the torque of the engine.
Supercharging - admittance into the cylinder of an air charge with density higher than that of the surrounding air.
Bridge Gauge - is an instrument used to find the radial position of crankshaft motor shaft.
Piston - is made of cast iron or aluminum alloy having a cylinder form.
Atomizer - is used to atomize the fuel into tiny spray which completely fill the furnace in the form of hollow cone.
Scavenging - is the process of cleaning the engine cylinder of exhaust gases by forcing through it a pressure of m fresh air.
Flare back - is due the explosion of a maximum fuel oil vapor and air in the furnace.
Single acting engine - is one in which work is done on one side of the piston.
Double acting engine - is an engine in which work is done on both sides of the piston.
Triple-expansion engine - is a three-cylinder engine in which there are three stages of expansion.
The working pressure in power cylinder is from 50 psi to 500 psi.
The working temperature in the cylinder is from 800°F to 1000°F.
Air pressure used in air injection fuel system is from 600 psi to 1000 psi.
Effect of over lubricating a diesel engine is:
Carbonization of oil on valve seats and possible explosive mixture is produced.
The average compression ratio of diesel engine is from 14:1 to 16:1.
Three types of piston:

1. barrel type
2. trunk type
3. closed head type

Three types of cam follower:

1. flat type
2. pivot type
3. roller type

Methods of mechanically operated starting valve:
1. the poppet
2. the disc type

Three classes of fuel pump:
1. continuous pressure
2. constant stroke
c. variable stroke

Type of pump used in transferring oil from the storage to the service tanks:
1. rotary pump
2. plunger pump
3. piston pump
4. centrifugal pump

Valve that is found in the cylinder head of a 4-stroke cycle engine:
1. fuel valve
2. air starting valve
3. relief valve
4. test valve
5. intake valve
6. exhaust valve

Four common type of governors used on a diesel engine:
1. constant speed governor
2. variable speed governor
3. speed limiting governor
4. load limiting governor

Kinds of piston rings used in an internal combustion engines:
1. compression ring
2. oil ring
3. firing ring
4. oil scraper ring

Reasons of smoky engine:
1. overload
2. injection not working
3. choked exhaust pipe
4. fuel or water and leaky things

Methods of reversing diesel engines:
1. sliding camshaft
2. shifting roller
c. rotating camshaft

Arrangements of cylinders:
1. in-line
2. radial
3. opposed cylinder
4. V
5. opposed piston

Position of cylinders:
1. vertical
2. horizontal
3. inclined

Methods of starting:
1. manual, crank, rope, and kick
2. electric (battery)
3. compressed air
4. using another engine

Applications:
1. automotive
2. marine
3. industrial
4. stationary power
5. locomotive
6. aircraft

Types of internal combustion engine:
1. Gasoline engine
2. Diesel engine
3. Kerosene engine
4. Gas engine
5. Oil-diesel engine

Methods of ignition:
1. Spark
2. Heat of compression

Reasons for supercharging:
1. to reduce the weight to power ratio
2. to compensate the power loss due to high altitude

Types of superchargers:
1. engine-driven compressor
2. exhaust-driven compressor
3. separately-driven compressor

Auxiliary systems of a diesel engine:
1. Fuel system
   a. fuel storage tank
   b. fuel filter
   c. transfer pump
   d. day tank
   e. fuel pump
2. Cooling system
   a. cooling water pump
   b. heat exchanger
   c. surge tank
   d. cooling tower
   e. raw water pump
3. Lubricating system:
   a. lub oil tank
   b. lub oil pump
   c. oil filter
   d. oil cooler
   e. lubricators

4. Intake and exhaust system
  a. air filter
  b. intake pipe
  c. exhaust pipe
  d. silencer
5. Starting system
  a. air compressor
  b. air storage tank

Advantages of diesel engine over other internal combustion engines:
1. low fuel cost
2. high efficiency
3. needs no large water supply
4. no long warm-up period
5. simple plant layout

Types of scavenging:
1. direct scavenging
2. loop scavenging
3. uniflow scavenging

Color of the smoke:
1. efficient combustion - light brown baze
2. insufficient air - black smoke
3. excess air - white smoke

Causes of black smoke:
1. fuel valve open too long
2. too low compression pressure
3. carbon in exhaust pipe
4. overload on engine

Causes of white smoke:
1. one or more cylinders not getting enough fuel
2. too low compression pressure
3. water inside the cylinder

ENGINE FOUNDATION

Functions of a Foundation:

Support the weight of the engine.
Maintain proper alignment with the machinery and
Absorb the vibration produced by unbalanced forces created by reciprocating revolving masses.

Materials:

Mixture: 1 : 2 : 4
1 part Cement
2 parts sand
4 parts broken stone or Gravel

For good firm soil, reinforced concrete foundations for large engines may use a leaner mixture down to 1 : 3 : 6

Soil Bearing Pressure:
The safe loads vary from about 4,890 kg/m^2 for alluvial soil or wet clay, to 19,560 kg/m^2 for gravel, coarse sand and dry clay. (12,225 kg/m^2 is assumed to be safe load average).
In computation 2,406 kg/m^2 may be used as weight of concrete.

Depth:
The foundation depth maybe taken as a good practical rule, to be 3.2 to 4.2 times the engine stroke; the lower factor for well-balanced multi-cylinder engines and increased factor for engines with fewer cylinders or on less firm soil.

Weight:
The minimum weight required to absorb vibration could be expressed as a function of the reciprocating masses and the speed of the engine. However, for practical purposes it is simpler to use the empirical formula.

Volume:
If the weight and speed of the engine is not known, the volume of concrete for the foundation may be estimated from the data in the table.
Anchor Bolts:
To prevent pulling out of the bolts when the nuts are tightened, the length embedded in concrete should be equal to at least thirty (30) times the bolt diameter. The upper ends are surrounded by a 50 mm or 75 mm sheet metal pipe, 460 mm to 610 mm long to permit them to be bent slightly to fit the holes of the bedplate.

FORMULA:

$$W_F = e\,(W_E)\,\sqrt{N}$$

Where:

W_F – weight of foundation in kgs
W_E – weight of engine in kgs
e – empirical coefficient
N – engine speed in RPM

Values of e

Type of Engine	Cylinder Arrangement	No. of Cylinders	e
Single Acting	Vertical	1	0.15
Single Acting	Vertical	2	0.14
Single Acting	Vertical	3	0.12
Single Acting	Vertical	4, 6, 8	0.11
Single Acting	Horizontal	1	0.25
Single Acting	Horizontal Duplex	2	0.24
Single Acting	Horizontal Twin Duplex	4	0.23
Double acting	Horizontal	1, 2	0.32
Double acting	Horizontal Twin Tandem	4	0.20

Volume of Concrete Foundation (Cubic Feet per HP)

No. of Cylinders	1	2	3	4	5 - 8
High speed engine	4.0	2.5	2.0	1.7	1.5
Medium speed engine	5.0	3.1	2.5	2.1	1.9
Low speed engine	6.0	4.0	3.0	2.6	2.3

Note: 1 cubic meter (m^3) = 35.315 ft^3
1 Horsepower = 0.746 KW

General Requirements

1. All heavy machinery shall be supported on solid foundation of sufficient mass and base area to prevent or minimize the transmission of objectionable vibration to the building and occupied space and to maintain the supported machine at its proper elevation and alignment.
2. Foundation mass should be from 3 to 5 times the weight of the machinery it is supposed to support. If the unbalanced inertial forces produced by the machine can be calculated, a mass of weight equal to 10 to 20 times the forces should be used to dampen vibration.
   For stability, the total combined engine, driven equipment and foundation center of gravity must be kept below the foundation's top.
3. the weight of the machine plus the weight of the foundation should be distributed over a sufficient soil area which is large enough to cause a bearing stress within the safe bearing capacity of the soil with a factor of safety of 5.
4. Foundation should be isolated from floor slabs and building footings by at least 25 mm around its perimeter to eliminate transmission of vibration. Fill opening with water tight-mastic.
   When installing machinery above grade level of a building, additional stiffness must be provided on its structural members of the building to dampen machine vibration.
5. Foundations are preferably built of concrete in the proportions of one (1) measure of Portland Cement to two (2) measures of sand and four (4) measures of screened crushed stones. The machine should not be placed on thefoundation until ten (10) days have elapsed or operated until another ten (10) days have passed.

6. Concrete foundation should have steel bar reinforcements placed both vertically and horizontally, to avoid thermal cracking. Weight of reinforcing steel should be from ½ % to 1 % of the weight of the foundation.

7. Foundation bolts of specified size should be used and surrounded by a pipe sleeve with an inside diameter of at least three (3) times the diameter of the anchor bolt and a length of at least eighteen (18) times the diameter of the bolt. No foundation bolts shall be less than 12 mm diameter.
8. Machine should be leveled by driving wedges between the machine's base and concrete foundation and with The aid of a spirit level. Grout all spaces under the machine bed with a thin mixture of one (1) part cement and one part sand. The level wedges should be removed after grout has thoroughly set and fill wedges holes with grout.

SAMPLE PROBLEMS

No. 1

A 2 - stroke, 4 - cylinders, 38 cm x 53 cm diesel engine is guaranteed to deliver 522 KW at 300 rpm.The fuel rate is 0.26 kg/KW-hr. If the heating value of the fuel is 44,320 KJ/kg Calculate:

a) the brake thermal efficiency (32%)
b) the brake mean effective pressure (422 KPa)
c) suction displacement in m^3/min-KW of shaft power (0.15)
d) heat supplied to cylinder per Liter of displacement

Given: 2-stroke, n' = 4; D = 0.38 m; L = 0.53 m; BP = 522 KW; N = 300 RPM
m_{FB} = 0.26 kg/KW-hr; HV = 44,320 KJ/kg

$$\text{a. } e_b = \frac{3600BP}{Qs} = \frac{3600BP}{m_F(HV)} = \frac{3600}{m_{FB}(HV)} = 31.2\%$$

$$\text{b. } BP = \frac{P_{mB}\pi LD^2Nn'}{4(60)}$$

$$N = RPM \ \text{(for 2 - stroke, single acting)}$$

$$P_{mB} = \frac{240(BP)}{\pi LD^2Nn'} = 434 \ \text{KPa}$$

.

$$\text{c. } BP = P_{mB}(V_D)$$

$$\frac{V_D}{BP} = \frac{1(60)}{P_{mB}} \ \frac{m^3}{\text{min - KW}} = 0.14$$

$$\text{d. } \frac{Q_S}{V_D} = \frac{m_F(HV)}{\frac{BP(3600)}{P_{mB}}} \ \frac{KJ}{m^3} = \frac{m_{FB}(BP)(HV)P_{mB}}{BP(3600)} = \frac{m_{FB}(HV)P_{mB}}{(3600)(1000)} \ \text{KJ/L} = 1.4 \ \text{KJ/L}$$

No. 2

Find the volume in Liters needed for a two weeks supply of 26°API fuel oil to operate a 750 KW engine 70 % of the time at full load, 10 % at 3/4 load and idle 20% of the time. Fuel rate is 0.25 kg/KW-hr at full load and 0.24 kg/KW-hr at 3/4 load. Temperature of oil is 21°C.

$$T = 2\text{weeks}(7 \text{ days})(24 \text{ hours}) = 336 \text{ hrs}$$

$$m_F = 336(0.70)0.25(750) + 0.10(336)(0.75)(750)(0.24) = 48{,}636 \ \text{kg}$$

$$S = \frac{141.5}{131.5 + 26} = 0.898$$

$$S@t = 0.898 - 0.0007(21 - 15.56) = 0.895$$

$$\rho = 895\frac{kg}{m^3} = 0.895\frac{kg}{L}$$

$$V_F = \frac{48{,}636}{0.895} = 54{,}342 \ \text{Liters}$$

No. 3

A diesel power plant is to have 3- supercharged, 7 cylinders, 590 KW, 720 RPM, 4-stroke cycle diesel engine. The full load brake specific fuel consumption is 0.24 kg/KW-hr. Determine the most economical size and dimensions of a day tank you would install to contain enough fuel for the three units to operate for 24 hours.

Assume D = 0.75H and fuel oil is at 26°API.
MF = 3(0.24)(590)(24) = 10,195.2 kg (total mass of fuel consumed)
S of fuel at 26°API = 0.898; ρ = 898 kg/m^3

$$VF = \frac{10{,}195.2}{898} = 11.4 \ m^3 \quad \text{(volume of fuel)}$$

$$V = \frac{\pi}{4}D^2H$$

$$D = 0.75H$$

$$H = 3\,m\,;\, D = 2.25\,M$$

No.4

It is claimed that 1 KW is developed for each 9200 KJ supplied per hour based on a lower heating value, bya supercharged Dorman diesel engine when operated at a brake mean effective pressure of 900 KPa. The lower heating value of the fuel is 41,900 KJ/kg and the engine is a 4 - stroke cycle with 16 cylinders, 39 cm x 56 cm. If the engine requires 20 kg of combustion air per kg of fuel, Determine:

a) the brake specific fuel consumption in kg/KW-hr
b) the brake power in KW at N = 360 RPM
c) the volumetric efficiency assuming combustion air is at P =101 KPa and T = 298°K.

a.

$$BP = \frac{Q_S}{9200} = \frac{m_F(HV)}{9200}$$

$$m_{FB} = \frac{9200}{41{,}900} = 0.22 \text{ kg/KW - hr}$$

b.

$$BP = \frac{P_{mB}\pi LD^2Nn'}{4(60)}$$

$$N = \frac{RPM}{2} \text{ for 4 - stroke single acting}$$

$$BP = 2{,}890 \text{ KW}$$

c. m_F = 0.22(2,890) = 635.8 kg/hr = 0.177 kg/sec

m_a = 0.177(20) = 3.54 kg/sec

$$V = \frac{mRT}{P}$$

Va = 3 m^3/sec

$$V_D = \frac{BP}{P_{mB}}$$

V_D = 3.211 m^3/sec

$$e_v = \frac{Va}{V_D} x100\%$$

e_v = 93.5 %

No. 5

A single acting, 4-cycle diesel engine uses 11 kg/hr of 24°API fuel when running at 420 RPM. Engine specifications: 23 cm x 36 cm. The prony brake used to determine the brake power has 1 m arm and registers on the scale 130 kg gross. If the tare mass is 12 kg, calculate the brake thermal efficiency based on the lower heating value of fuel.

T = (P – tare)R = (130 – 12)1 = 118 kg-m (9.81 N/kg) = 1,157.58 N-m

$$BP = \frac{2\pi TN}{60{,}000}$$

BP = 51 KW

LHV = 38,105 + 139.6(°API) = 41,455.4 KJ/kg

$$e_B = \frac{3600BP}{m_F(LHV)} \text{ x100\%}$$

e_B = 40.3 %

SAMPLE PROBLEMS

1. A 2 - stroke, 4 - cylinders, 38 cm x 53 cm diesel engine is guaranteed to deliver 522 KW at 300 rpm. The fuel rate is 0.26 kg/KW-hr. If the heating value of the fuel is 44,320 KJ/kg Calculate:
   a) the brake thermal efficiency (32%)
   b) the brake mean effective pressure (422 KPa)
   c) suction displacement in m^3/min-KW of shaft power (0.15)
   d) heat supplied to cylinder per Liter of displacement
2. Find the volume in Liters needed for a two weeks supply of 26°API fuel oil to operate a 750 KW engine 70 % of the time at full load, 10 % at 3/4 load and idle 20% of the time. Fuel rate is 0.25 kg/KW-hr at full load and 0.24 kg/KW-hr at 3/4 load. Temperature of oil is 21°C.
3. A diesel power plant is to have 3- supercharged, 7 cylinders, 590 KW, 720 RPM, 4-stroke cycle diesel engine. The full load brake specific fuel consumption is 0.24 kg/KW-hr. Determine the most economical size and dimensions of a day tank you would install to contain enough fuel for the three units to operate for 24 hours. Assume D = 0.75H.
4. It is claimed that 1 KW is developed for each 9200 KJ supplied per hour based on a lower heating value, by a supercharged Dorman diesel engine when operated at a brake mean effective pressure of 900 KPa. The lower heating value of the fuel is 41,900 KJ/kg and the engine is a 4 - stroke cycle with 16 cylinders, 39 cm x 56 cm. If the engine requires 20 kg of combustion air per kg of fuel, Determine:
   a) the brake specific fuel consumption in kg/KW-hr
   b) the brake power in KW
   c) the volumetric efficiency
5. A single acting, 4-cycle diesel engine uses 11 kg/hr of 24°API fuel when running at 420 RPM. Engine specifications: 23 cm x 36 cm. The prony brake used to determine the brake power has 1 m arm and registers on the scale 130 kg gross. If the tare mass is 12 kg, calculate the brake thermal efficiency based on the lower

heating value of fuel.

6. A 4 cylinders, 4 stroke cycle diesel engine, 30.5 cm x 46 cm, 260 RPM, single acting diesel engine is rated at 200 KW. The fuel rate at rated load is 0.26 kg/KW-hr, and the fuel used has a heating value of 44,000 KJ/kg. Determine
   a) the brake mean effective pressure
   b) displacement in m^3/min-KW of shaft power
   c) the brake thermal efficiency
7. A torque of 28 kg-m is developed by a diesel engine when running at 1200 RPM and using 10 kg of fuel per hour. the heating value of the fuel is 44,200 KJ/kg. The engine is 4-stroke cycle and has 4 cylinders and the bore is equal the stroke. The engine takes in 25 kg of air per kg of fuel and the volumetric efficiency is 80%. Calculate the brake thermal efficiency and the bore in cm.
8. Performance values of a 3000 KW Diesel power plant unit are as follows:
   Fuel rate: 1.5 barrels for 900 KW-hr of 25°API fuel
   Generator Efficiency: 92%
   Mechanical efficiency: 82%
   Determine:
   a) Engine fuel rate in kg/KW-hr
   b) Engine-Generator fuel rate in kg/KW-hr
   c) Indicated thermal efficiency
   d) Brake thermal efficiency
   e) Overall thermal efficiency

9. A twin tandem, 4-stroke cycle, double acting Blast furnace gas engine is to developed 2700 KW of brake power at 90 RPM. Expected operating data are:

   P_{mB} = 450 KPa
   η_m = 83%
   η_b = 65%
   HHV = 31,700 KJ/m^3
   L/D = 1.35
   r = 7.1
   k = 1.32

Determine:

a) the cylinder dimensions in cm

b) fuel consumption in L/hr if the specific gravity of fuel is 0.88.

10. An 8 cylinders,2-stroke cycle, single acting diesel engine rated at 940 KW at standard condition (P = 760 mm Hg; t = 15.56°C) is directly coupled to a 24 poles alternator; 3-phase; 60 Hertz. Assuming brake mean effective pressure of 520 KPa; η_m = 85% & η_g = 94%

    a) Find the diameter of the cylinder and the length of stroke of the piston if the average piston speed is 310 m/min.

    b) What will be the actual thermal efficiency for a combined fuel rate of 0.20 kg of diesel fuel per KW hour with a HV = 46,530 KJ/kg

    c) What will be the KW output if the above diesel engine is operated in Baguio City whose elevation is 1525 m above sea level and the actual sea level temperature is 29°C.

11. A diesel engine under test gave the following performance data;

    Brake Power - 3360 KW
    Jacket loss - 23%
    Mechanical Efficiency - 83%
    Indicated thermal efficiency - 34%

    A waste heat recovery boiler recovers 35% of the exhaust loss. If cooling water leaving the engine is at 66°C and is used as feedwater to boiler, determine the quantity of 0.14 MPa steam that can be produced in kg/hr.

12. When the pressure is 101.3 KPa and the temperature is 27°C, a diesel engine has the full throttle characteristics listed:

    Brake power - 275 KW
    Brake specific fuel consumption - 0.25 kg/KW-hr
    Air - Fuel ratio - 22
    Mechanical efficiency - 88%

    If the engine is operated at a pressure of 84.5 KPa and temperature of 16°C, Find:

    a) Brake power in KW

    b) Mechanical efficiency

    c) Brake specific fuel consumption

    d) air-Fuel ratio

13. A 40 KW blast furnace engine shows by test a gas consumption of 4.2 m^3/KW-hr. Heating value of gas is 33,500 KJ/m^3. Mechanical efficiency of the gas engine is 86%, Calculate:

    a) The brake thermal efficiency

    b) Indicated thermal efficiency

    c) Heat rate in KJ/KW-hr

    d) If the heat rejected to cooling water is 28% of the heat generated in engine cylinder, determine the quantity of cooling water in L/hr to be circulated if the allowable rise in temperature of the cooling water is 10°C

14. On the test bed, a 16 cylinders, V type, 4-stroke cycle, turbo charge diesel engine developed 7350 KW at 514 RPM. Heat rejection to lubricating oil was 460 KJ/KW-hr. Specific gravity of lubricating oil was 0.90 and its specific heat was 1.9 KJ/kg-°C. Direct connected generator is rated at 9,150 KVA at 80% power factor and efficiency of 95%, Calculate

    a) Liters of lubricating oil per hour to be circulated through engine if lubricating oil temperature entering engine is 54°C and the allowable rise in temperature of oil is 6°C.

    b) Liters of make up new lubricating oil per 24 hours operated at rated load to maintain the level of oil in the sump tank constant if manufacturers guarantees a lubricating oil consumption of 1.36 gram/KW-hr.

15. A diesel generating set has the following specifications:

    Mitsubishi, 16 cylinders, Model V-280, 4 stroke cycle, supercharge
    Bore and stroke - 280 mm x 290 mm
    Brake power and RPM - 225 kw per cylinder at 900 RPM
    Fuel type and heating value - Bunker C, 10 000 KCal/kg
    Brake specific fuel consumption - 0.2064 kg/kw-hr

    If the engine is operating at full load with an ambient room temperature of 32°C db and 24 C wb and 760 mm Hg barometric pressure, determine

a) the fuel consumption in L/hr (743.04 kg/hr)
b) the brake thermal efficiency (42%;39%)
c) the amount of cooling water in L/min, if the cooling loss is 30% and the temperature rise of the cooling water is 22 C. (1689 L/min)
d) the cubic meter per minute of air entering the engine at ambient conditions if the air-fuel ratio is 22:1 (241 m^3/min)

at 32 C db and 24 C wb

$v = 0.885\ m^3/kg$

16. The following data were obtained during a full load test on a 2 cylinder, single acting, 4 stroke cycle diesel engine, 280 mm bore by 381 mm stroke; average RPM, 385; length of prony brake arm, 92 cm; net weight on scales (due to torque), 227 kg; total; fuel used in 15 min. run, 5.2 kg; heating value of fuel oil, 45 400 KJ/kg; Mean effective pressure from indicator cards (average of each cylinder), 620 KPa; Find
    a) Brake power (82.6 KW
    b) Brake specific fuel consumption (0.252 kg/KW-hr)
    c) Brake thermal efficiency (32%)
    d) Indicated power (93.33 KW)
    e) Friction Power (10.7 KW)
    f)Mechanical efficiency (89%)
17. A diesel engine under test gave the following performance data:
    Brake power - 3357 KW
    Jacket loss - 23%
    Mechanical efficiency - 83%
    Indicated thermal efficiency - 34%
    A waste heat recovery boiler recovers 35% of the exhaust loss. If cooling water leaving the engine is 66C and used as feed water to boiler, determine the quantity of 34.5 KPa steam that can be produced in kg/hr.

$$\eta_m = \frac{BP}{IP} \times 100\%$$

IP = 4044.6 KW

$$e_i = \frac{3600(IP)}{m_f(HV)} \times 100\%$$

$Q_s = m_f(HV) = 42\ 825\ 176$ KJ/hr

BP = 3600(3357) = 12 085 200 KJ/hr

FP = 3600(IP - BP) = 2 475 360 KJ/hr

Q_c = cooling loss = $0.23Q_s$ = 9 849 791 KJ/hr

Q_G = exhaust gas loss = $Q_s - (BP + Q_c + FP)$ = 18 414 825 KJ/hr

Q_B = heat absorbed by waste heat recovery boiler

$Q_B = 0.35Q_G$ = 6 445 189 KJ/hr

$Q_B = m_s$ (hs - hf)

hs = 2689 KJ/kg

hf = 276.23 KJ/kg

m_s = 2672 kg/hr

18. A spark ignition engine produces 224 KW while using 0.0169 kg/sec of fuel. The fuel has a higher heating value of 44 186 KJ/kg and the engine has a compression ratio of 8:1. The friction power is found to be 22.4 KW. Determine the indicated engine efficiency. (58%)
19. A six cylinder automotive engine with a 9 cm x 9 cm has a fuel consumption of 0.306 kg/KW-hr at 300 RPM. The brake power developed is 86 KW and the indicated power is 105 KW. The thermal efficiency of the ideal cycle is 47%, and the fuel has a heating value of 44 186 KJ/kg. Compute the brake and indicated engine efficiency. (56.6%;69.2%)
20. A 6 cylinder, 4 stroke cycle, single acting, spark ignition engine with a compression ratio of 9.5 is required to develop 67.14 KW with a torque of 194 N-m. Under the conditions the η_m = 78% and the P_{mb} = 552 KPa. For the ideal cycle, P_1 = 101.35 KPa, t_1 = 35°C and the hot air standard k = 1.32. If D/L =1.1 and the m_{fi} = 0.353 kg/KW-hr and HV = 43 967 KJ/kg, Determine
    a) the bore and stroke in cm (10.1 cm; 9.2 cm)

b) the indicated thermal efficiency (23.3%)
c) the brake engine efficiency (34.4%)
d) the percentage clearance (11.8%)

21. A 6 cylinder, 700 mm x 900 mm, single acting, 2 stroke cycle diesel engine develops 22.35 KW at 128 RPM. compression ratio is 13, cut off ratio is 2.45 and overall k = 1.33. Suction pressure is 97 KPa, suction temperature 54°C. the average Pm_i = 620 KPa. The fuel consumed during a 30 minutes test was 286 kg with a LHV = 42 563 KJ/kg. Determine
    a) the brake thermal efficiency
    b) the mechanical efficiency
    c) the indicated power
    d) the brake engine efficiency
22. A single cylinder,4 stroke, compression ignition oil engine gives 15 KW at 60 RPM and uses fuel having the composition by mass of 84% carbon and 16% hydrogen. The air supply is 100% in excess of that required for perfect combustion. The fuel has a calorific value of 45 000KJ/kg and the brake thermal efficiency is 30%. Find:
    a) the mass of fuel used per cycle (0.0022)
    b) the actual mass of air taken in per cycle
    c) the volume of air taken in per cycle at 100 KPa and 15°C.Take R = 0.29 KJ/kg-°C

ME 413
Semi Final Exam 2004

1. A 6 cylinder, 4 stroke cycle, single acting, spark ignition engine with a compression ratio of 9.5 is required to develop 67.14 KW with a torque of 194 N-m. Under the conditions the η_m = 78% and the P_{mb} = 552 KPa. For the ideal cycle, P_1 = 101.35 KPa, t_1 = 35°C and the hot air standard k = 1.32. If D/L =1.1 and the m_{fi} = 0.353 kg/KW-hr and HV = 43 967 KJ/kg, Determine
    a) the bore and stroke in cm
    b) the indicated thermal efficiency
    c) the brake engine efficiency
    d) the percentage clearance
2. A spark ignition engine produces 224 KW while using 0.0169 kg/sec of fuel. The fuel has a higher heating value of 44 186 KJ/kg and the engine has a compression ratio of 8:1. The friction power is found to be 22.4 KW. Determine the indicated engine efficiency. (58%)
3. Performance values of a 3000 KW Diesel power plant unit are as follows:
   Fuel rate: 1.5 barrels for 900 KW-hr of 25°API fuel
   Generator Efficiency: 92%
   Mechanical efficiency: 82%
   Determine:
    a) Engine fuel rate in kg/KW-hr
    b) Engine-Generator fuel rate in kg/KW-hr
    c) Indicated thermal efficiency
    d) Brake thermal efficiency
    e) Overall thermal efficiency

ME 413
Final Exam 2004

1. A 40 KW blast furnace engine shows by test a gas consumption of 4.2 m^3/KW-hr. Heating value of gas is 33,500 KJ/m^3. Mechanical efficiency of the gas engine is 86%, Calculate:
    a) The brake thermal efficiency
    b) Indicated thermal efficiency
    c) Heat rate in KJ/KW-hr
2. An 8 cylinders,2-stroke cycle, single acting diesel engine rated at 940 KW at standard condition (P = 760 mm Hg; t = 15.56°C) is directly coupled to a 24 poles alternator; 3-phase; 60 Hertz. Assuming brake mean effective pressure of 520 KPa; η_m = 85% & η_g = 94%

a) Find the diameter of the cylinder and the length of stroke of the piston if the average piston speed is 310 m/min.

b) What will be the actual thermal efficiency for a combined fuel rate of 0.20 kg of diesel fuel per KW hour with a HV = 46,530 KJ/kg

3. A single cylinder,4 stroke, compression ignition oil engine gives 15 KW at 60 RPM and uses fuel having the composition by mass of 84% carbon and 16% hydrogen. The air supply is 100% in excess of that required for perfect combustion. The fuel has a calorific value of 45 000KJ/kg and the brake thermal efficiency is 30%. Find

a) the mass of fuel used per cycle

b) the actual mass of air taken in per cycle

c) the volume of air taken in per cycle at 100 KPa and 15°C.Take R = 0.29 KJ/kg-°C

ME 413

Final Exam 2004

1. A 40 KW blast furnace engine shows by test a gas consumption of 4.2 m^3/KW-hr. Heating value of gas is 33,500 KJ/m^3. Mechanical efficiency of the gas engine is 86%, Calculate:

a) The brake thermal efficiency

b) Indicated thermal efficiency

c) Heat rate in KJ/KW-hr

2. An 8 cylinders,2-stroke cycle, single acting diesel engine rated at 940 KW at standard condition (P = 760 mm Hg; t = 15.56°C) is directly coupled to a 24 poles alternator; 3-phase; 60 Hertz. Assuming brake mean effective pressure of 520 KPa; η_m = 85% & η_g = 94%

a) Find the diameter of the cylinder and the length of stroke of the piston if the average piston speed is 310 m/min.

b) What will be the actual thermal efficiency for a combined fuel rate of 0.20 kg of diesel fuel per KW hour with a HV = 46,530 KJ/kg

3. A single cylinder,4 stroke, compression ignition oil engine gives 15 KW at 60 RPM and uses fuel having the composition by mass of 84% carbon and 16% hydrogen. The air supply is 100% in excess of that required for perfect combustion. The fuel has a calorific value of 45 000 KJ/kg and the brake thermal efficiency is 30%. Find

a) the mass of fuel used per cycle

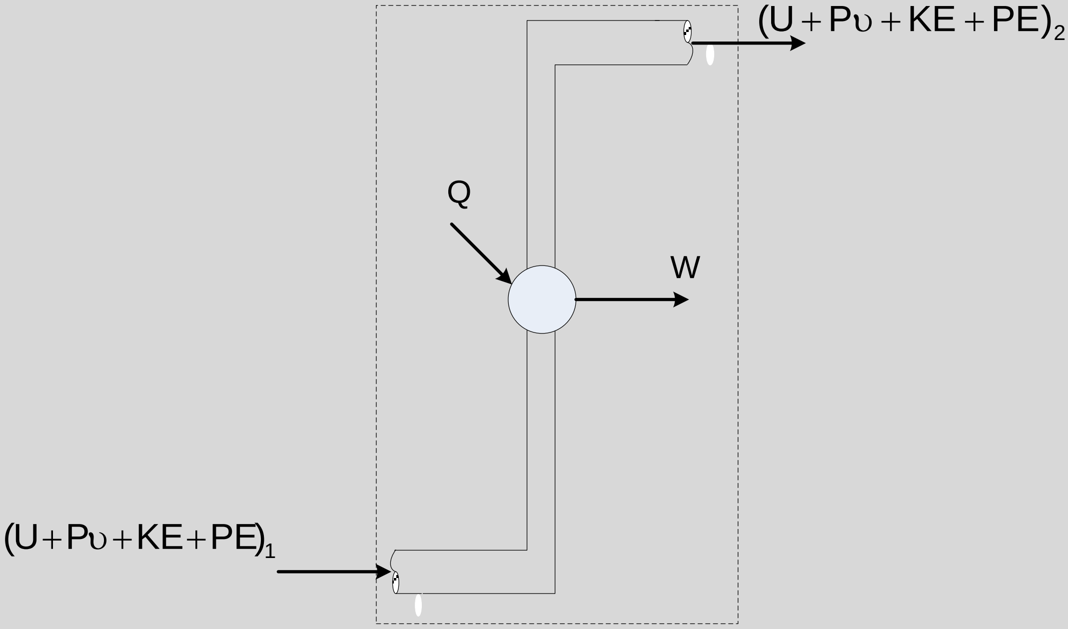
b) the actual mass of air taken in per cycle

c) the volume of air taken in per cycle at 100 KPa and 15°C. Take R = 0.29 KJ/kg-°C

PUMPS AND PIPING

BERNOULLI'S ENERGY EQUATION

From 1st Law (OPEN SYSTEM)



GENERAL ENERGY EQUATION

$$Q = \Delta U + \Delta(P\upsilon) + \Delta KE + \Delta PE + W$$

Considering

$$Q = 0;\ \Delta U = 0 \text{ and } W = 0$$

$$0 = \Delta(P\upsilon) + \Delta KE + \Delta PE$$

$$0 = P_2\upsilon_2 - P_1\upsilon_1 + \frac{v_2^{\,2} - v_1^{\,2}}{2(1000)} + \frac{g(z_2 - z_1)}{1000}$$

For one dimensional flow

$$\rho_1 = \rho_2 = \rho$$

$$\upsilon = \frac{1}{\rho} \quad \therefore \upsilon_2 = \upsilon_1 = \upsilon$$

$$0 = \frac{(P_2 - P_1)}{\rho} + \frac{v_2^{\,2} - v_1^{\,2}}{2(1000)} + \frac{g(z_2 - z_1)}{1000}$$

Multiplying both sides of the equation by $\dfrac{1000}{g}$

$$0 = \frac{1000(P_2 - P_1)}{\rho g} + \frac{v_2^{\,2} - v_1^{\,2}}{2g} + (z_2 - z_1)$$

$$0 = \frac{(P_2 - P_1)}{\rho g/1000} + \frac{v_2^{\,2} - v_1^{\,2}}{2g} + (z_2 - z_1)$$

but

$$\gamma = \rho g/1000$$

$$0 = \frac{(P_2 - P_1)}{\gamma} + \frac{v_2^{\,2} - v_1^{\,2}}{2g} + (z_2 - z_1)$$

$$\frac{P_1}{\gamma} + \frac{v_1^{\,2}}{2g} + z_1 = \frac{P_2}{\gamma} + \frac{v_2^{\,2}}{2g} + z_2$$

Bernoulli's Energy Equation

$$\frac{P_1}{\gamma}+\frac{v_1^2}{2g}+z_1=\frac{P_2}{\gamma}+\frac{v_2^2}{2g}+z_2$$

$\frac{P}{\gamma}$ – Pressure head, meters

$\frac{v^2}{2g}$ – Velocity head, meters

z - Elevation head, meters

P - Pressure, KPa

v - velocity, m/sec

γ - specific weight, KN/m^3

z – elevation, m

g - gravitational acceleration, $\frac{m}{sec^2}$

Note

z is positive if meassured above datum

z is negative if measured below datum

Considering Head Loss and W = 0

$\Delta U - Q = H_L$ in meters

$$\frac{P_1}{\gamma}+\frac{v_1^2}{2g}+z_1=\frac{P_2}{\gamma}+\frac{v_2^2}{2g}+z_2+H_L$$

With Energy Head added to the fluid (Work done on the system; let –W = h_t in meters)

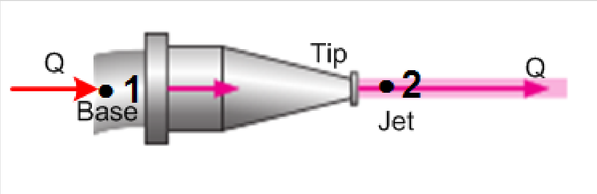
$$\frac{P_1}{\gamma}+\frac{v_1^2}{2g}+z_1+h_t=\frac{P_2}{\gamma}+\frac{v_2^2}{2g}+z_2+H_L$$

With Energy Head given up by the fluid (Work done on the system; let W = h in meters)

$$\frac{P_1}{\gamma}+\frac{v_1^2}{2g}+z_1=\frac{P_2}{\gamma}+\frac{v_2^2}{2g}+z_2+H_L+h$$

APPLICATION OF BERNOULLI'S ENERGY THEOREM

Nozzle



$$\frac{P_1}{\gamma}+\frac{v_1^2}{2g}+Z_1=\frac{P_2}{\gamma}+\frac{v_2^2}{2g}+Z_2+H_L$$

From continuity equation: Q = Av ; for $\rho_1 = \rho_2 = \rho$

$$Q = A_1v_1 = A_2v_2$$

For a nozzle the head loss H_L is equal to:

$$\left(\frac{1}{C_v^2}-1\right)\frac{v_2^2}{2g}$$

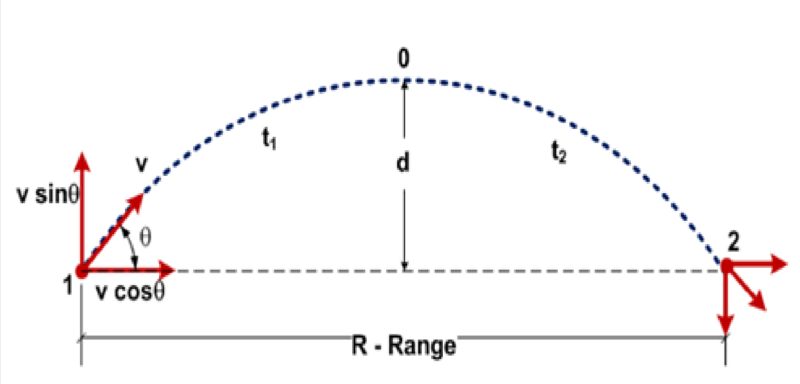
where: C_v - velocity coefficient

JET POWER

$$P_{Jet} = Q\gamma\frac{v_2^2}{2g}$$

If a nozzle makes an angle θ with the horizontal,

JET TRAJECTORY:



v – velocity of water at tip of nozzle

at 1 to 0

$$v_o^2 = v_1^2 - 2gd$$

$$0 = v_1^2 - 2gd$$

$$d = \frac{(v\sin\theta)^2}{2g} \rightarrow 1$$

$$v_0 = v_1 - gt$$

$$t = \frac{v\sin\theta}{g} \rightarrow 2$$

at 0 to 2

$$d = v_1 t + \tfrac{1}{2}gt^2$$

$$d = 0 + \tfrac{1}{2}gt^2$$

$$d = \tfrac{1}{2}gt^2 \rightarrow 3$$

$$\frac{(v\sin\theta)^2}{2g} = \tfrac{1}{2}gt^2$$

$$t^2 = \frac{(v\sin\theta)^2}{g^2}$$

$$t = \frac{(v\sin\theta)}{g} \rightarrow 4$$

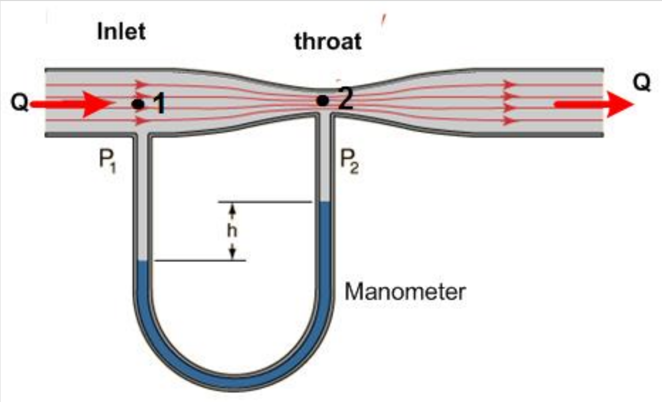
$$R = v\cos\theta(2t)$$

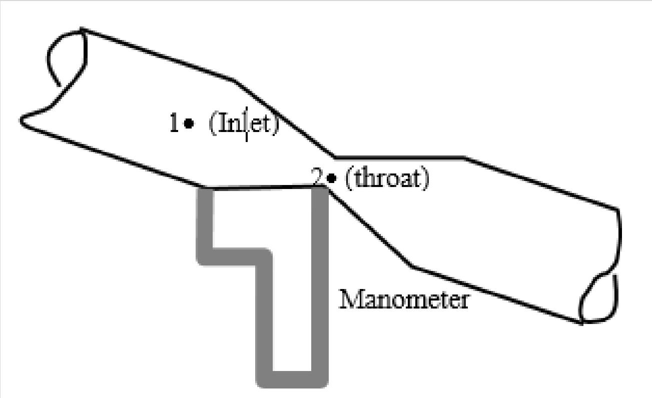
$$R = \frac{2(v\sin\theta)(v\cos\theta)}{g} = \frac{v^2(2\sin\theta\cos\theta)}{g}$$

$$(2\sin\theta\cos\theta) = \sin(2\theta)$$

$$R = \frac{v^2(\sin 2\theta)}{g} \rightarrow 5$$

Venturi Meter





a. Without considering H_L:

$$\frac{P_1}{\gamma}+\frac{v_1^2}{2g}+Z_1=\frac{P_2}{\gamma}+\frac{v_2^2}{2g}+Z_2$$

$$Q = A_1v_1 = A_2v_2$$

$$Q - \text{Theoretical Flow}$$

b. Considering H_L:

$$\frac{P_1}{\gamma}+\frac{v_1^2}{2g}+Z_1=\frac{P_2}{\gamma}+\frac{v_2^2}{2g}+Z_2+H_L$$

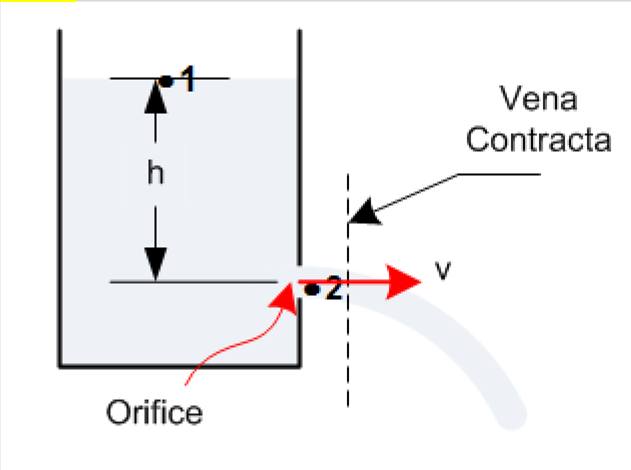
$$Q' = A_1v_1 = A_2v_2$$

$$Q' - \text{Actual Flow}$$

METER COEFFICIENT

$$C = \frac{Q'}{Q}$$

Orifice



By applying Bernoulli's Energy theorem:

$$\frac{P_1}{\gamma}+\frac{v_1^2}{2g}+Z_1=\frac{P_2}{\gamma}+\frac{v_2^2}{2g}+Z_2$$

But $P_1 = P_2 = P_a$ and v_1 is negligible, then

$$\frac{v_2^2}{2g} = Z_1 - Z_2$$

and from figure: $Z_1 - Z_2 = h$, therefore

$$\frac{v_2^2}{2g} = h$$

$$v_2 = \sqrt{2gh}$$

let $v_2 = v_t$

$$v_t = \sqrt{2gh}$$

where:

v_t - theoretical velocity, m/sec
h - head producing the flow, meters
g - gravitational acceleration, m/sec^2

COEFFICIENT OF VELOCITY (Cv)

$$Cv = \frac{\text{actual velocity}}{\text{theoretical velocity}}$$

$$Cv = \frac{v'}{v_t}$$

COEFFICIENT OF CONTRACTION

$$Cc = \frac{\text{area of jet @ vena contracta}}{\text{area of the orifice}}$$

$$Cc = \frac{a}{A}$$

where: a - area of jet at vena contracta, m^2
A - area of the orifice, m^2

COEFFICIENT OF DISCHARGE

$$Cd = \frac{\text{actual flow}}{\text{theoretical flow}}$$

$$Cd = \frac{Q'}{Q}$$

$$C_d = C_v C_c$$

where:

v' - actual velocity
v_t - theoretical velocity
a - area of jet at vena contracta
A - area of orifice
Q' - actual flow
Q - theoretical flow
C_v - coefficient of velocity
C_c - coefficient of contraction
C_d - coefficient of discharge

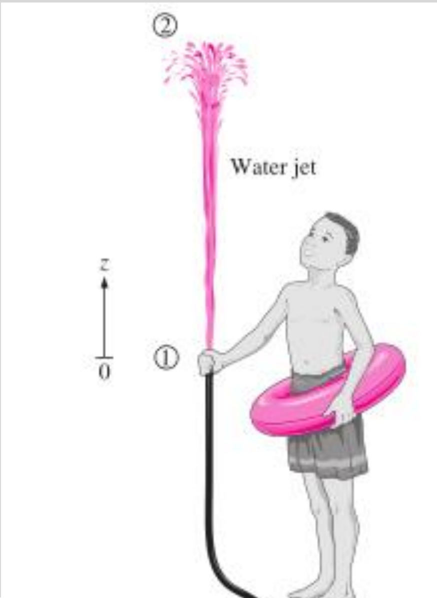
Sample Problem no. 1

Water is flowing from a hose attached to a water main at 400 kPa gage (Fig. 5–38). A child places his thumb to cover most of the hose outlet, causing a thin jet of high-speed water to emerge. If the hose is held upward, what is the maximum height that the jet could achieve?

$$\frac{P_1}{\gamma} + \frac{v_1^2}{2g} + z_1 = \frac{P_2}{\gamma} + \frac{v_2^2}{2g} + z_2$$

$$\frac{400}{\gamma} + 0 + 0 = 0 + 0 + z_2$$

$$z_2 = \frac{400}{9.81} = 40.8\ \text{m}$$



Sample Problem no. 2

A large tank open to the atmosphere is filled with water to a height of 5 m from the outlet tap (Fig. 5–39). A tap near the bottom of the tank is now opened, and water flows out from the smooth and rounded outlet. Determine the water velocity at the outlet.

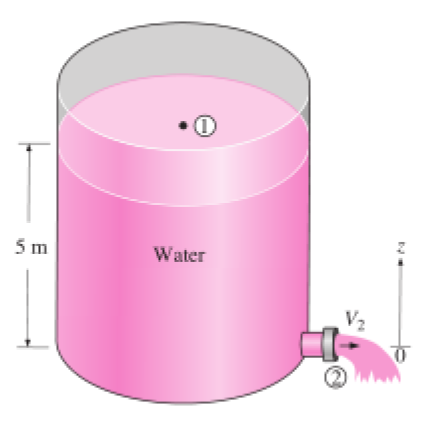
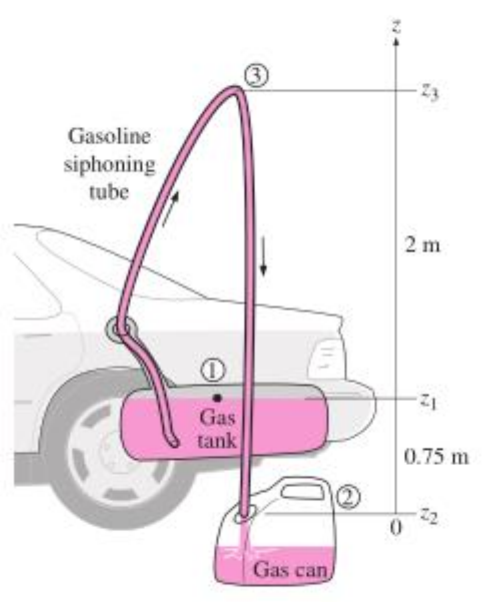


FIGURE 5–39

Sample Problem No. 3

During a trip to the beach (P_{atm} = 1 atm = 101.3 kPa), a car runs out of gasoline, and it becomes necessary to siphon gas out of the car of a Good Samaritan (Fig. 5–40). The siphon is a small-diameter hose, and to start the siphon it is necessary to insert one siphon end in the full gas tank, fill the hose with gasoline via suction, and then place the other end in a gas can below the level of the gas tank. The difference in pressure between point 1 (at the free surface of the gasoline in the tank) and point 2 (at the outlet of the tube) causes the liquid to flow from the higher to the lower elevation. Point 2 is located 0.75 m below point 1 in this case, and point 3 is located 2 m above point 1. The siphon diameter is 4 mm, and frictional losses in the siphon are to be disregarded. Determine (*a*) the minimum time to withdraw 4 L of gasoline from the tank to the can and (*b*) the pressure at point 3. The density of gasoline is 750 kg/m^3.



PUMPS
By Engr. Yuri G. Melliza

Pump is a device that moves or compresses liquids and gases. Pumps are used in a variety of machines and other devices, including home heating systems, refrigerators, oil wells and water wells, and turbojet and car engines. The fluids (gases or liquids) moved by pumps range from air for inflating bicycle tires to liquid sodium and liquid potassium for cooling nuclear reactors. Most pumps are made of steel, but some are made of glass or plastic. Gas pumps are also called compressors, fans, or blowers.

Types of Pumps

Dynamic Pump: Dynamic pumps maintain a steady flow of fluid.

Positive Displacement Pump: Positive displacement pumps, on the other hand, trap individual portions of fluid that are in an enclosed area before moving them along.

Dynamic pumps

Centrifugal pumps consist of a motor-driven propeller like device, called an impeller, which is contained within a circular housing. The impeller is a wheel of curved blades that rotates on an axis. Before most centrifugal pumps can start pumping liquid, they must be primed (filled with liquid). As the impeller rotates, it creates suction that draws a continuous flow of fluid through an inlet pipe. Fluid enters the pump at the center of the impeller and travels out along the blades due to centrifugal (outward) force. The curved ends of the blades sweep the fluid to an outlet port. Centrifugal pumps are inexpensive and can handle large amounts of fluid. They are widely used in chemical processing plants and oil refineries.

Axial-flow pumps have a motor-driven rotor that directs fluid along a path parallel to its axis. The fluid thus travels in a relatively straight path from the inlet pipe through the pump to the outlet pipe. Axial-flow pumps are most often used as compressors in turbojet engines. Centrifugal pumps are also used for this purpose, but axial-flow pumps are more efficient. Axial-flow compressors consist of alternating rows of rotors and stationary blades. The blades and rotors produce a pressure rise in the air as it moves through the axial-flow compressor. Air then leaves the compressor under high pressure.

Jet pumps get their name from the way they move fluid. They operate on the principle that a high-velocity fluid will carry along any other fluid it passes through. Most jet pumps send a jet of steam or water through the fluid that needs to be moved. The jet carries the fluid with it directly into the outlet pipe and, at the same time, creates a vacuum that draws more fluid into the pump. The amount of fluid carried out of most jet pumps is several times the amount in the jet itself. Jet pumps can be used to raise water from wells deeper than 60 meters. In such cases, a centrifugal pump at ground level supplies water for a jet at the bottom of the well. The jet carries well water with it back up to ground level. Jet pumps are also used in high vacuum diffusion pumps to create a vacuum in an enclosed area. In high vacuum diffusion pumps, a high-velocity jet of mercury or oil vapor is sent into the enclosed area. The vapor molecules collide with the molecules of air and force them out the outlet port.

Electromagnetic pumps are used chiefly to move liquid sodium and liquid potassium, which serve as coolants in nuclear reactors. These pumps consist of electrical conductors and magnetized pipes. The conductors send current through the fluid, which thereby becomes an electromagnet. The fluid is then moved by the magnetic attraction and repulsion (pushing away) between the fluid's magnetic field and that of the pipes. The fluid is therefore moved in an electromagnetic pump in much the same way as an armature is moved in an electric motor.

Positive displacement pumps

Rotary pumps are the most widely used positive displacement pumps. They are often used to pump such viscous (sticky) liquids as motor oil, syrup, and paint. There are three main types of rotary pumps. These types are: (1) *gear pumps*, (2) *lobe pumps*, and (3) *sliding vane pumps.*

Gear pumps consist of two gears that rotate against the walls of a circular housing. The inlet and outlet ports are at opposite sides of the housing, on line with the point where the teeth of the gears are fitted together. Fluid that enters the pump is trapped by the rotating gear teeth, which sweep the fluid along the pump wall to the outlet port.

Lobe pumps operate in a manner similar to gear pumps. However, instead of gears, lobe pumps are equipped with impellers that have lobes (rounded projections) fitted together. Lobe pumps can discharge large amounts of fluid at low pressure.

Sliding vane pumps consist of a slotted impeller mounted off-center in a circular housing. Sliding vanes (blades) move in and out of the slots. As the vanes rotate by the inlet port, they sweep up fluid and trap it against the pump wall. The distance between the impeller and the pump wall narrows near the outlet port. As the fluid is carried around to this port, the vanes are pushed in and the fluid is compressed. The pressurized fluid then rushes out the outlet port.

Reciprocating pumps consist of a piston that moves back and forth within a cylinder. One end of the cylinder has an opening through which the connecting rod of the piston passes. The other end of the cylinder, called the closed end, has an inlet valve or an outlet valve, or both, depending on the type of pump. In some reciprocating pumps, the inlet valve or the outlet valve is on the piston. Common reciprocating pumps include lift pumps, force pumps, and bicycle tire pumps.

Lift pumps draw water from wells. In a lift pump, the inlet valve is at the closed end of the cylinder and the outlet valve is on the piston. As the piston is raised, water is drawn up through the inlet valve. As the piston moves down, the inlet valve closes, forcing water through the outlet valve and up above the piston. As the piston is raised again, the outlet valve closes and the water is lifted to an opening, where it leaves the pump. At the same time, more water is drawn through the inlet valve. It is theoretically possible for a lift pump to raise water in a well almost 10 meters. However, because of leakage and resistance, a lift pump cannot raise water that is deeper than about 7.5 meters.

Force pumps are similar to lift pumps. However, in force pumps, both the inlet valve and the outlet valve are at the closed end of the cylinder. As the piston moves away from the closed end, fluid enters the cylinder. When the piston moves toward the closed end, the fluid is forced out the outlet valve.

Bicycle tire pumps differ in the number and location of the valves they have and in the way air enters the cylinder. Some simple bicycle tire pumps have the inlet valve on the piston and the outlet valve at the closed end of the cylinder. Air enters the pump near the point where the connecting rod passes through the cylinder. As the rod is pulled out, air passes through the piston and fills the areas between the piston and the outlet valve. As the rod is pushed in, the inlet valve closes and the piston forces air through the outlet valve.

History

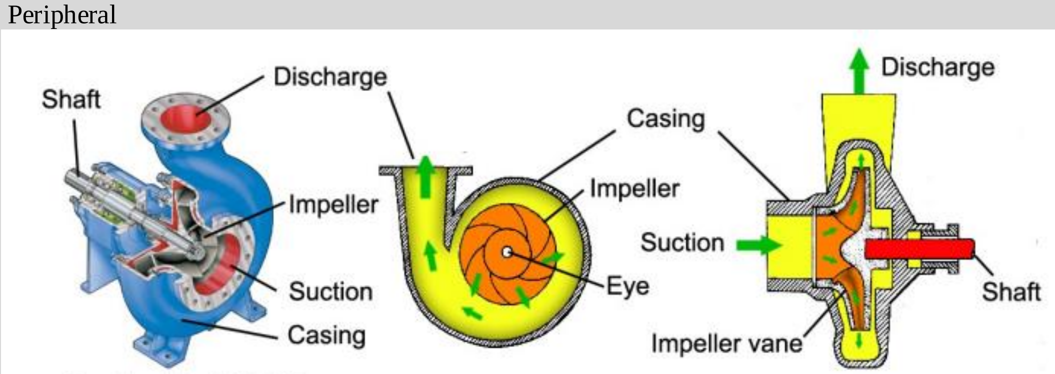
Pumping devices have been an important means of moving fluids for thousands of years. The ancient Egyptians used water wheels with buckets mounted on them to move water for irrigation. The buckets scooped water from wells and streams and deposited it in ditches that carried it to fields. In the 200's B.C., Ctesibius, a Greek inventor, made a reciprocating pump for pumping water. About the same time, Archimedes, a Greek mathematician, invented a screw pump that was made up of a screw rotating in a cylinder. This type of pump was used to drain and to irrigate the Nile Valley.

True centrifugal pumps were not developed until the late 1600's, when Denis Papin, a French-born inventor, made one with straight vanes. The British inventor John Appold introduced a curved-vane centrifugal pump in 1851. Axial-flow compressors were first used on turbojet engines in the 1940's.

CLASSIFICATION OF PUMPS

Centrifugal: It consist essentially of an impeller arranged to rotate within a casing so that the liquid will enter at the center or eye of the impeller and be thrown outward by centrifugal force to the outer periphery of the impeller and discharge into the outer case. It operates at high discharge pressure, low head, high speed and they are not self priming.

a. Centrifugal
b. Mixed Flow
1. single stage
2. multi stage
3. Propeller or axial flow
4. Peripheral



Rotary: It is a positive displacement pump consisting of a fixed casing containing gears, cams, screws, vanes, plungers or similar element actuated by the rotation of the drive shaft. A rotary pump traps a quantity of liquid and moves it along toward the discharge point. For a gear type rotary pump the unmeshed gears at the pump provides a space for the liquid to fill as the gears rotate. The liquid trapped between the teeth and the pump casing is eventually released at the discharge line. It operates at low heads, low discharge and is used for pumping viscous liquids like oil.

a. cam   c. gear
b. screw   d. vane

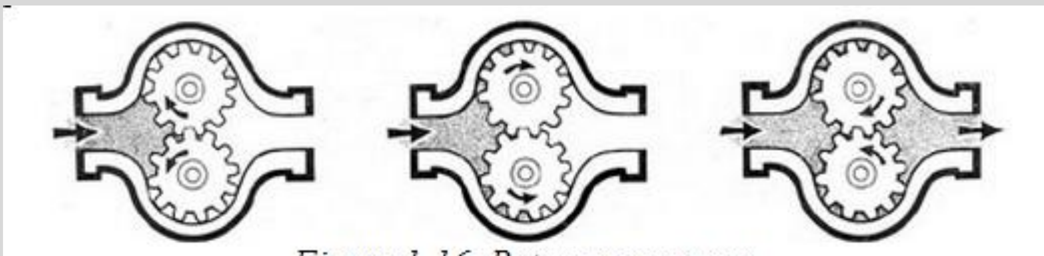
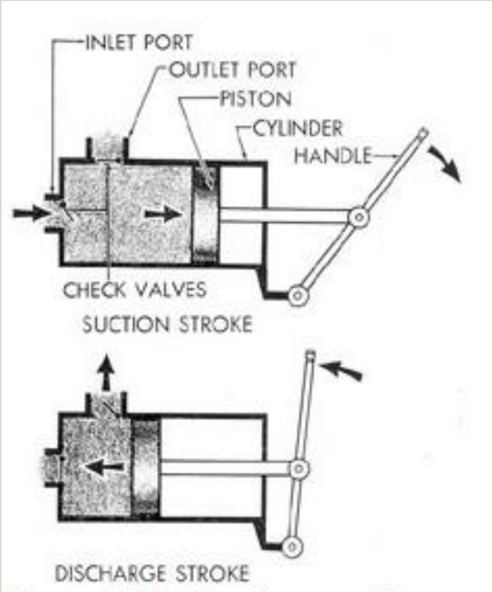


Figure 1-16. Rotary gear pump.

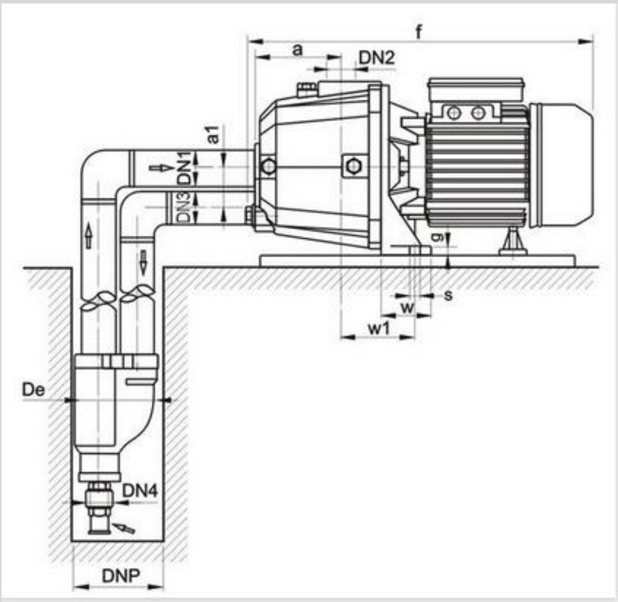
Reciprocating: It is a positive displacement unit wherein the pumping action is accomplished by the forward and backward movement of a piston or a plunger inside a cylinder usually provided with valves.

a. Piston
b. Direct Acting
    1. single
    2. duplex
c. Crank and Flywheel
d. Plunger
e. Power Driven
    1. simplex
    2. duplex
    3. triplex

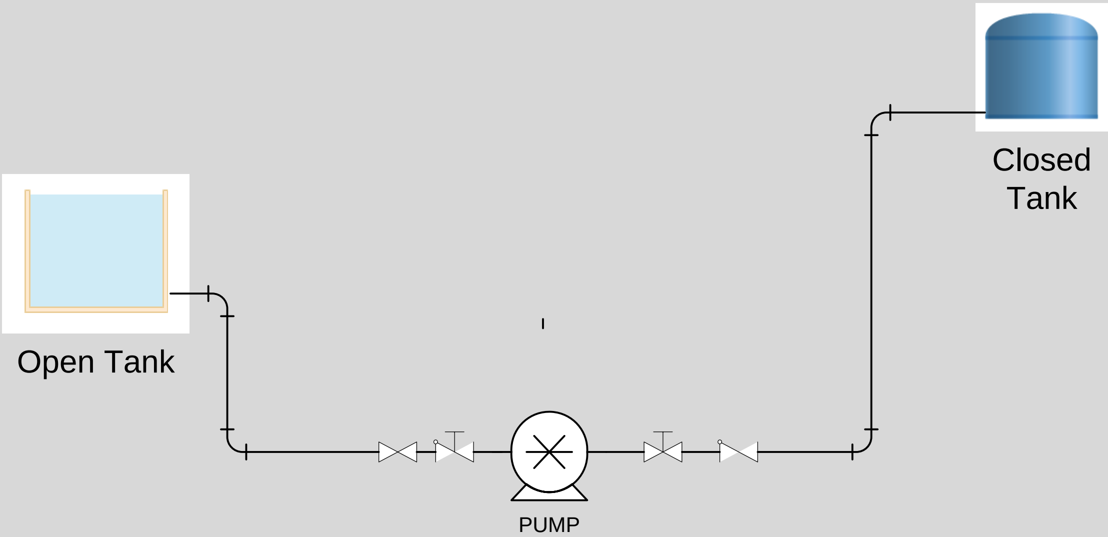


Deepwell pumps: It is used when pumping water from deep wells. The pump is lowered into the well and operated close to water level. They are usually motor driven with the motor being at the ground level and connected to the pump by a long vertical line shaft.

a. Turbine
b. Ejector or centrifugal
c. reciprocating
d. Airlift



For a final choice of a pump for a particular operation the following data is needed.
Number of units required
Nature of liquid
Capacity
Suction conditions
Discharge conditions
Intermittent or continuous service
Total dynamic head
Position of pump, vertical or horizontal
Location, geographical, indoor, outdoor, elevation
Type of power drive



FUNDAMENTAL EQUATIONS

TOTAL DYNAMIC HEAD

$$H_t = \frac{P_2 - P_1}{\gamma} + \frac{v_2^2 - v_1^2}{2g} + Z_2 - Z_1 + H_L \quad \text{meters}$$

FLUID POWER or WATER POWER

$WP = Q\gamma H_t$ KW

DISCHARGE or CAPACITY

$Q = A_s v_s = A_d v_d$

s – refers to suction
d – refers to discharge

BRAKE or SHAFT POWER

$$BP = \frac{2\pi TN}{60{,}000} \quad \text{KW}$$

PUMP EFFICIENCY

$$\eta_P = \frac{WP}{BP} \times 100\%$$

MOTOR EFFICIENCY

$$\eta_m = \frac{BP}{MP} \times 100\%$$

COMBINED PUMP-MOTOR EFFICIENCY

$$\eta_C = \frac{WP}{MP} \times 100\%$$

$$\eta_C = \eta_P \eta_m$$

MOTOR POWER

For Single Phase Motor

$$MP = \frac{EI(\cos\theta)}{1000}\ KW$$

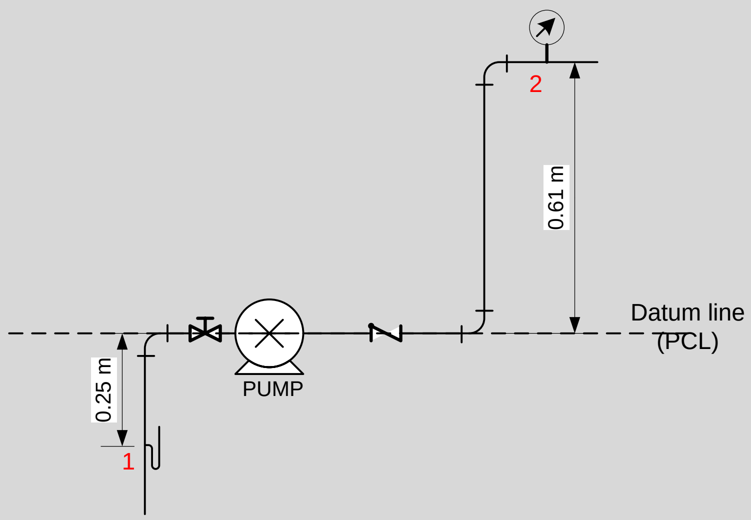
For 3 Phase Motor

$$MP = \frac{\sqrt{3}\ EI(\cos\theta)}{1000}\ KW$$

where: P - pressure in KPa
- v - velocity, m/sec
- γ - specific weight of liquid, KN/m^3
- Z - elevation, meters
- g - gravitational acceleration, m/sec^2
- H_L - total head loss, meters
- E - energy, Volts
- I - current, amperes
- $(\cos\theta)$ - power factor
- T - brake torque, N-m
- N - no. of RPM
- FP - fluid power, KW
- WP or BP - brake power, KW
- MP - power input to motor
- s – suction
- d - discharge
- 1 & 2 – Reference Points (@ suction and discharge)

Sample Problem no. 1

A pump handles brine having a temperature of 0°C and S = 1.2 when the discharge pressure is 173 KPag and the suction gauge indicates a vacuum of 305 mm Hg. The discharge gauge is 61 cm above the pump centerline and the point of attachment of the suction gauge is 25 cm below the pump centerline. Suction and discharge pipes inside diameters are 100 mm and 80 mm respectively. Find the total head of the pump in meters of the fluid handled. If the efficiency of the motor-driven pump is 88%, find the brake power load of the motor when the pump handles 32 L/sec of brine. (8.5 KW)



$\gamma = 1.2(9.81) = 11.772 \frac{KN}{m^3}$

$z_1 = -0.25\,m$

$z_2 = 0.61\,m$

Reference Datum : Pump Centerline

$P_1 = -305\left(\frac{101.325}{760}\right) = -40.7\,KPa$

$P_2 = 173\,KPa$

$v = \frac{Q}{A}; Q = 0.032 \frac{m^3}{sec}$

$As = \frac{\pi}{4}(0.100)^2 = 0.00785\,m^2$

$Ad = \frac{\pi}{4}(0.08)^2 = 0.005\,m^2$

$v_1 = 4.08 \frac{m}{sec}$

$v_2 = 6.4 \frac{m}{sec}$

$h = \left(\frac{P_2 - P_1}{\gamma}\right) + \left(\frac{v_2{}^2 - v_1{}^2}{2g}\right) + (z_2 - z_1) + H_{LL}$

$h = \left(\frac{172 + 40.7}{11.772}\right) + \left(\frac{\overline{6.4}^2 - \overline{4.08}^2}{2(9.81)}\right) + (0.61 + 0.25) + 0$

$h = 18.07 + 0.85 + 0.86 = 19.78\,m$

$\text{Power} = \frac{0.032(11.772)(19.78)}{0.88} = 8.5\,KW$

HEAD LOSSES

H_L = Major loss + Minor losses

Major Loss: Head loss due to friction and turbulence in pipes
Minor Losses: Minor losses include, losses due to valves and fittings, enlargement, contraction, pipe entrance and pipe exit. Minor losses are most easily obtained in terms of equivalent length of pipe "Le". The advantage of this approach is that both pipe and fittings are expressed in terms of "Equivalent Length" of pipe of the same relative roughness.
Considering Major loss only

Using the Darcy-Weisbach Equation

(Henry Darcy, a French Engineer and Julius Weisbach, a German engineer)

$$h_f = \frac{fLv^2}{2gD} \quad \text{meters}$$

$$h_f = K\frac{v^2}{2g}$$

$$K = \frac{fL}{D}$$

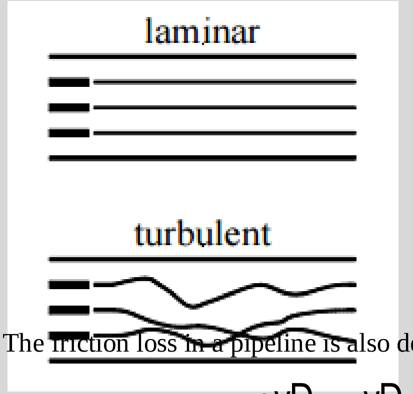
Considering Major and Minor losses

$$h_f = \frac{f(L + \Sigma L_e)v^2}{2gD} \quad \text{meters}$$

Where;
f - friction factor from Moody's Chart
L - length of pipe, m
Le - equivalent length in straight pipe of valves and fittings, m
v - velocity, m/sec
D - pipe inside diameter, m

g - gravitational acceleration, m/sec^2

REYNOLD'S NUMBER: Reynold's Number is a non dimensional one which combines the physical quantities which describes the flow either *Laminar* or *Turbulent* flow.



The friction loss in a pipeline is also dependent upon this dimensionless factor.

$$N_R = \frac{\rho v D}{\mu} = \frac{vD}{\nu}$$

$$\nu = \frac{\mu}{\rho} \frac{m^2}{sec}$$

where;

μ - absolute or dynamic viscosity, Pa-sec

ν - kinematic viscosity, m^2/sec

For a Reynold's Number of less 2100 flow is said to *Laminar*

For a Reynold's Number of greater than 3000 the flow is *Turbulent*

FRICTION FACTOR (Resistance Coefficient)

For Laminar Flow

$$f = \frac{64}{N_R}$$

For Turbulent Flow

Prandtl Equation for smooth pipes for NR > 3000

$$\frac{1}{\sqrt{f}} = 2\log\left(N_R\sqrt{f}\right) - 0.8$$

Cyril F. Colebrook Equation (1910 – 1997)

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{\in/D}{3.7} + \frac{2.51}{N_R\sqrt{f}}\right)$$

$$f = \frac{0.25}{\left[\log\left(\frac{\in/D}{3.7} + \frac{5.74}{N_R^{0.9}}\right)\right]^2}$$

$$Q = -2.22 D^{5/2}\sqrt{g h_f / L} \bullet \log\left(\frac{\in/D}{3.7} + \frac{1.78\nu}{D^{3/2}\sqrt{g h_f / L}}\right) \frac{m^3}{sec}$$

Zigrang-Sylvester Equation (From Fluid Mechanics by F.M. White)

$$\frac{1}{\sqrt{f}} = -4.0\log\left[\frac{\in/D}{3.7} - \frac{5.02}{N_R}\log\left(\frac{\in/D}{3.7} + \frac{13}{N_R}\right)\right]$$

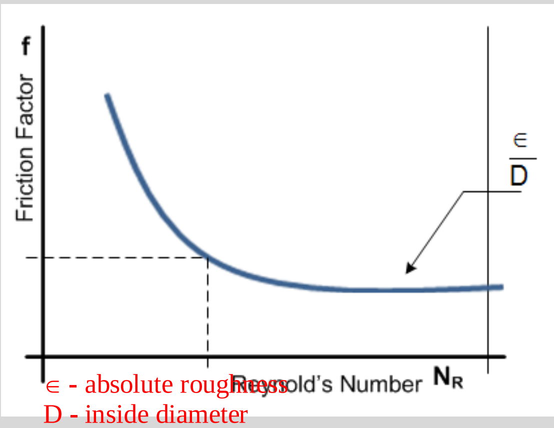
$$\sqrt{f} = \frac{1}{-4.0\log\left[\frac{\in/D}{3.7} - \frac{5.02}{N_R}\log\left(\frac{\in/D}{3.7} + \frac{13}{N_R}\right)\right]}$$

S. E. Haaland Equation (1983)

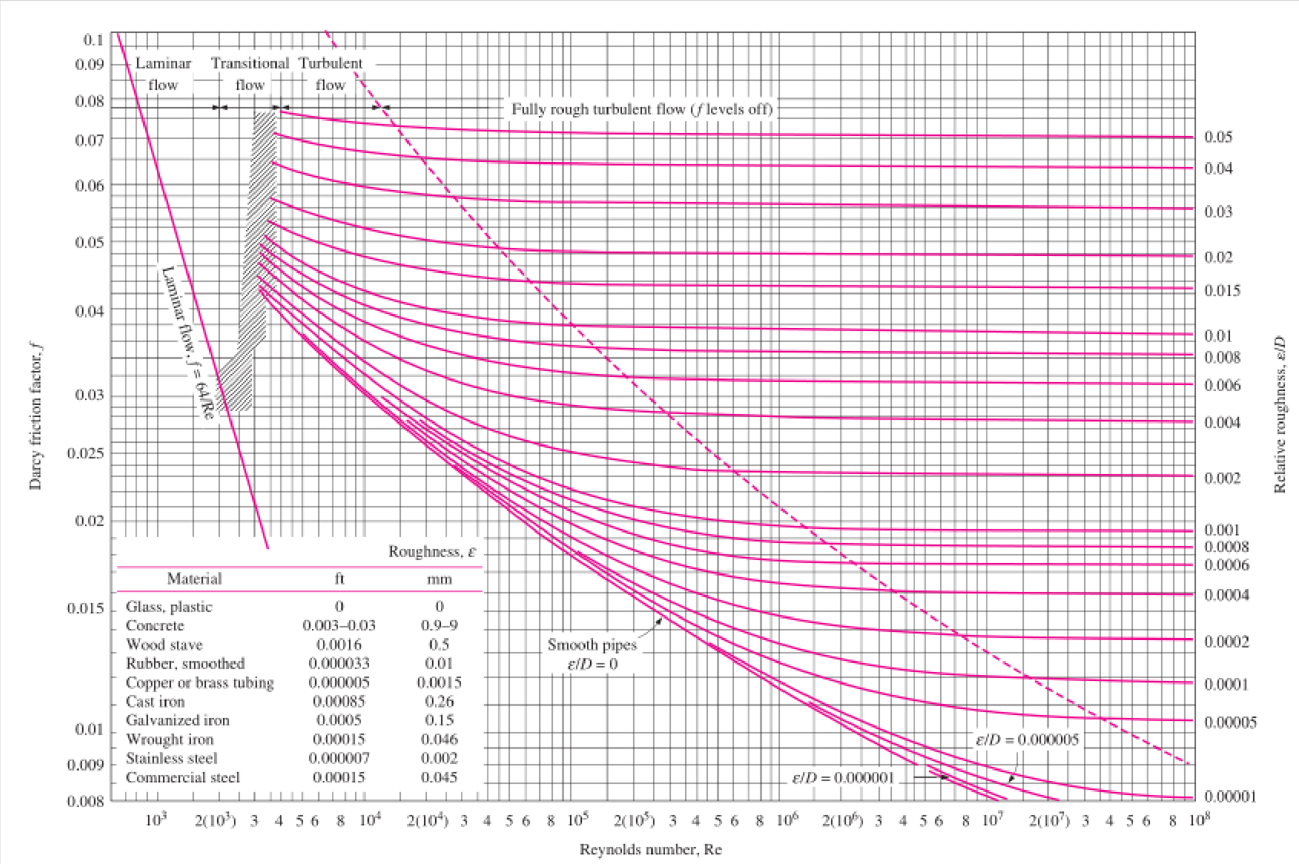
$$\frac{1}{\sqrt{f}} \cong -1.8\log\left[\frac{6.9}{N_R} + \left(\frac{\in/D}{3.7}\right)^{1.11}\right]$$

In 1944, Lewis Ferry Moody plotted the Darcy–Weisbach friction factor against Reynolds number N_R for various values of relative roughness $\frac{\in}{D}$ This chart became commonly known as the *Moody Chart or Moody Diagram*.

Moody's Chart



∈ - absolute roughness
D - inside diameter
∈/D - relative roughness



Absolute Pipe Roughness (∈) is a measure of pipe wall irregularities of commercial pipes. Other than pipes, absolute roughness is also used for representing the irregularities of other equipment walls, for example, walls of heat exchanger shell. The absolute roughness has dimensions of length and is usually expressed in millimeter (mm).

Relative Roughness' or 'Roughness factor of a pipe wall can be defined as the ratio of absolute roughness to the pipe nominal diameter. Relative roughness factor is often used for pressure drop calculations for pipes and other equipment. The relative roughness factor is an important parameter for determining friction factor based on Reynold's number for flow in a pipe.

Relative roughness = ∈/D

Absolute Roughness is usually defined for a material and can be measured experimentally.

Following table gives typical roughness values in millimeters for commonly used piping materials.

ABSOLUTE ROUGHNESS (∈) OF PIPE MATERIAL	
MATERIAL	∈ (mm)
Aluminum	0.0015
Aluminum, Lead	0.001 - 0.002
Drawn Brass, Drawn Copper	0.0015
Drawn Tubing, Glass, Plastic	0.0015-0.01
Drawn Brass, Copper, Stainless Steel (New)	0.0015-0.01
Flexible Rubber Tubing - Smooth	0.006-0.07
Flexible Rubber Tubing - Wire Reinforced	0.3-4
Wrought Iron (New)	0.045
Carbon Steel (New)	0.02-0.05
Carbon Steel (Slightly Corroded)	0.05-0.15
Carbon Steel (Moderately Corroded)	0.15-1
Carbon Steel (Badly Corroded)	1-3
Carbon Steel (Cement-lined)	1.5

Asphalted Cast Iron	0.1-1
Cast Iron (new)	0.25
New cast iron	0.25 - 0.8
Cast Iron (old, sandblasted)	1
Fiberglass	0.005
Worn cast iron	0.8 - 1.5
Corroding cast iron	1.5 - 2.5
Asphalted cast iron	0.012
Sheet Metal Ducts (with smooth joints)	0.02-0.1
Galvanized Iron	0.025-0.15
Galvanized steel	0.15
Wood Stave	0.18-0.91
Wood Stave, used	0.25-1
Smooth Cement	0.5
Concrete – Very Smooth	0.025-0.2
Concrete – Fine (Floated, Brushed)	0.2-0.8
Concrete – Rough, Form Marks	0.8-3
PVC, Plastic Pipes	0.0015
Galvanized iron	0.015
Riveted Steel	0.91-9.1
Rusted steel	0.15 - 4
Stainless steel	0.015 – 0.03
Steel commercial pipe	0.045 - 0.09
Stretched steel	0.015
Water Mains with Tuberculations	1.2
Weld steel	0.045
Brickwork, Mature Foul Sewers	3
Smoothed cement	0.3
Ordinary concrete	0.3 - 3
Well planed wood	0.18 - 0.9
Ordinary wood	5
PVC	0.0015

PIPE, TUBING AND FITTINGS

Nominal Pipe Diameter: Pipe sizes are based on the approximate diameter and are reported as nominal pipe sizes. Regardless of wall thickness, pipes of the same nominal diameter have the same outside diameter. This permits interchange of fittings. Pipe may be manufactured with different and various wall thickness, so some standardization is necessary. A method of identifying pipe sizes has been established by ANSI (American National Standard Institute). By convention, pipe size and fittings are characterized in terms of Nominal Diameter and wall thickness.

For steel pipes, nominal diameter is approximately the same as the inside diameter for 12" and smaller. For sizes of 14" and larger, the nominal diameter is exactly the outside diameter.

SCHEDULE NUMBER: The wall thickness of pipe is indicated by a *schedule number*, which is a function of *internal pressure* and *allowable stress*.

Schedule Number ≅ 1000P/S

where P - internal working pressure, KPa

S - allowable stress, KPa

Schedule number in use: 10,20,30, 40,60, 80, 100, 120, 140, and 160.

Schedule 40 "Standard Pipe"
Schedule 80 " Extra Strong Pipe"

TUBINGS: Tubing specifications are based on the actual outside diameter with a designated wall thickness. Conventional systems such as the *Birmingham Wire Gauge* "BWG" are used to indicate the wall thickness.

FITTING: The term fitting refers to a piece of pipe that can:

1. Join two pieces of pipe
   ex. couplings and unions
2. Change pipeline directions
   ex. elbows and tees
3. Change pipeline diameters
   ex. reducers
4. Terminate a pipeline
   ex. plugs and valves
5. Join two streams to form a third
   ex. tees, wyes, and crosses
6. Control the flow
   ex. valves

VALVES: A valve is also a fitting, but it has more important uses than simply to connect pipe. Valves are used either to control the flow rate or to shut off the flow of fluid.

DESIGN OF A PIPING SYSTEM

The engineer should consider the following items when he is developing the design of a piping system.

1. Choice of material and sizes
2. Effects of temperature level and temperature changes.
   a. insulation
   b. thermal expansion
   c. freezing
3. Flexibility of the system for physical and thermal shocks.
4. Adequate support and anchorage
5. Alteration in the system and the service.
6. Maintenance and inspection.
7. Ease of installation
8. Auxiliary and standby pumps and lines
9. Safety
   a. Design factors
   b. Relief valves and flare systems

TABLE FOR DN and NPS

Diameter Nominal DN *(mm)*	**Nominal Pipe Size NPS** *(inches)*
6	1/8
8	1/4
10	3/8
15	1/2
20	3/4
25	1
32	1 1/4
40	1 1/2
50	2
65	2 1/2
80	3
100	4
150	6
200	8
250	10

300	12
350	14
400	16
450	18
500	20
550	22
600	24
650	26
700	28
750	30
800	32
900	36
1000	40
1050	42
1100	44
1200	48
1300	52
1400	56
1500	60
1600	64
1700	68
1800	72
1900	76
2000	80
2200	88

O.D., I.D., & Wall Thickness Dimensions For Given Pipe Sizes									
Pipe Size	Schedule 40 Thickness			Schedule 10 Thickness			Schedule 5 Thickness		
	O.D.	I.D.	Wall Thickness	O.D.	I.D.	Wall Thickness	O.D.	I.D.	Wall Thickness
3/4	1.050	0.824	0.113	1.050	0.884	0.083	1.050	0.920	0.065
1	1.315	1.049	0.133	1.315	1.097	0.109	1.315	1.185	0.065
1-1/4	1.660	1.380	0.140	1.660	1.442	0.109	1.660	1.530	0.065
1-1/2	1.900	1.610	0.145	1.900	1.682	0.109	1.900	1.770	0.065
2	2.375	2.067	0.154	2.375	2.157	0.109	2.375	2.245	0.065
2-1/2	2.875	2.469	0.203	2.875	2.635	0.120	2.875	2.709	0.083
3	3.500	3.068	0.216	3.500	3.260	0.120	3.500	3.334	0.083
3-1/2	4.000	3.548	0.226	4.000	3.760	0.120	4.000	3.834	0.083
4	4.500	4.026	0.237	4.500	4.260	0.120	4.500	4.334	0.083
6	6.625	6.065	0.280	6.625	6.357	0.134	6.625	6.407	0.109
8	8.625	7.981	0.322	8.625	8.329	0.148	8.625	8.407	0.109

*Table values are in inches
*O.D. = Outside Diameter
*I.D. = Inside Diameter

Nominal Pipe Size NPS, Nominal Bore NB, Outside Diameter OD
What is a relation between NPS, NB & OD of a pipe?
What is NPT & what are its applications?
What is a relation between NPS, NB & OD of a pipe?
Nominal Pipe Size (NPS) is a North American set of standard sizes for pipes used for high or low pressures and temperatures. Pipe size is specified with two non-dimensional numbers: a nominal pipe size (NPS) for diameter based on inches, and a schedule (Sched. or Sch.) for wall thickness. NPS is often incorrectly called National Pipe Size, due to confusion with national pipe thread (NPT)
NB (nominal bore) is the European designation equivalent to NPS is *DN* (diamètre nominal/nominal diameter/Durchmesser nach Norm), in which sizes are measured in millimeters. NB (nominal bore) is also frequently used interchangeably with NPS.
OD is the outside diameter of the pipe and is fixed for a given size.
The NPS is very loosely related to the inside diameter in inches, but only for NPS 1/8 to NPS 12. For NPS 14 and larger, the NPS is equal to the outside diameter (OD) in inches. For a given NPS, the OD stays constant and the wall thickness increases with larger SCH. For a given SCH, the OD increases with increasing NPS while the wall thickness increases or stays constant. Pipe sizes are documented by a number of standards, including API 5L, ANSI/ASME B36.10M in the US, BS 1600 and BS EN 10255 in the United Kingdom and Europe, and ISO 65 internationally.
For NPS of 5 and larger, the DN is equal to the NPS multiplied by 25 (not 25.4).
Table below shows the relation between Nominal pipe size, Nominal diameter & outside diameter for pipes:

Nominal Pipe size NPS [inches]	Nominal Diameter DN [mm]	Outside Diameter OD [mm]
1/2	15	21.3
3/4	20	26.7
1	25	33.4
1 1/4	32	42.16
1 1/2	40	48.26
2	50	60.3
2 1/2	65	73.03
3	80	88.9
4	100	114.3
5	125	141.3
6	150	168.28
8	200	219.08
10	250	273.05
12	300	323.85
14	350	355.6
16	400	406.4
18	450	457.2
20	500	508
24	600	609.6
28	700	711.2
32	800	812.8
36	900	914.4
40	1000	1016
42	1050	1066.8
44	1100	1117.6
48	1200	1219.2

52	1300	1320.8
56	1400	1422.4
60	1500	1524
64	1600	1625.6
68	1700	1727.2
72	1800	1828.8
76	1900	1930.4
80	2000	2032
88	2200	2235.2
96	2400	2438.4
104	2600	2641.6
112	2800	2844.8
120	3000	3048
128	3200	3251.2

Definition and Details of Nominal Pipe Size

NOMINAL PIPE SIZE

Nominal Pipe Size (NPS) is a North American set of standard sizes for pipes used for high or low pressures and temperatures. The name NPS is based on the earlier "Iron Pipe Size" (IPS) system.

That IPS system was established to designate the pipe size. The size represented the approximate inside diameter of the pipe in inches. An IPS 6" pipe is one whose inside diameter is approximately 6 inches. Users started to call the pipe as 2inch, 4inch, 6inch pipe and so on. To begin, each pipe size was produced to have one thickness, which later was termed as standard (STD) or standard weight (STD.WT.). The outside diameter of the pipe was standardized.

As the industrial requirements handling higher pressure fluids, pipes were manufactured with thicker walls, which has become known as an extra strong (XS) or extra heavy (XH). The higher pressure requirements increased further, with thicker wall pipes. Accordingly, pipes were made with double extra strong (XXS) or double extra heavy (XXH) walls, while the standardized outside diameters are unchanged. Note that on this website only terms XS and XXS are used.

PIPE SCHEDULE

So, at the IPS time only three wall thickness were in use. In March 1927, the American Standards Association surveyed industry and created a system that designated wall thicknesses based on smaller steps between sizes. The designation known as nominal pipe size replaced iron pipe size, and the term schedule (SCH) was invented to specify the nominal wall thickness of pipe. By adding schedule numbers to the IPS standards, today we know a range of wall thicknesses, namely:

SCH 5, 5S, 10, 10S, 20, 30, 40, 40S, 60, 80, 80S, 100, 120, 140, 160, STD, XS AND XXS.

Nominal pipe size (NPS) is a dimensionless designator of pipe size. It indicates standard pipe size when followed by the specific size designation number without an inch symbol. For example, NPS 6 indicates a pipe whose outside diameter is 168.3 mm.

The NPS is very loosely related to the inside diameter in inches, and NPS 12 and smaller pipe has outside diameter greater than the size designator. For NPS 14 and larger, the NPS is equal to 14inch.

For a given NPS, the outside diameter stays constant and the wall thickness increases with larger schedule number. The inside diameter will depend upon the pipe wall thickness specified by the schedule number.
SUMMARY:
Pipe size is specified with two non-dimensional numbers,
nominal pipe size (NPS)
schedule number (SCH)
and the relationship between these numbers determine the inside diameter of a pipe.
Stainless Steel Pipe dimensions determined by ASME B36.19 covering the outside diameter and the Schedule wall thickness. Note that stainless wall thicknesses to ASME B36.19 all have an "S" suffix. Sizes without an "S" suffix are to ASME B36.10 which is intended for carbon steel pipes.
The International Standards Organization (ISO) also employs a system with a dimensionless designator. Diameter nominal (DN) is used in the metric unit system. It indicates standard pipe size when followed by the specific size designation number without a millimeter symbol. For example, DN 80 is the equivalent designation of NPS 3. Below a table with equivalents for NPS and DN pipe sizes.

NPS	1/2	3/4	1	1¼	1½	2	2½	3	3½	4
DN	15	20	25	32	40	50	65	80	90	100

Note: For NPS ≥ 4, the related DN = 25 multiplied by the NPS number.
Do you now what is "ein zweihunderter Rohr"?. Germans means with that a pipe NPS 8 or DN 200. In this case, the Dutch talking about a "8 duimer". I'm really curious how people in other countries indicates a pipe.
EXAMPLES OF ACTUAL O.D. AND I.D.
ACTUAL OUTSIDE DIAMETERS
NPS 1 actual O.D. = 1.5/16" (33.4 mm)
NPS 2 actual O.D. = 2.3/8" (60.3 mm)
NPS 3 actual O.D. = 3½" (88.9 mm)
NPS 4 actual O.D. = 4½" (114.3 mm)
NPS 12 actual O.D. = 12.3/4" (323.9 mm)
NPS 14 ACTUAL O.D. = 14"(355.6 MM)
ACTUAL INSIDE DIAMETERS OF A 1 INCH PIPE.
NPS 1-SCH 40 = O.D.33,4 mm - WT. 3,38 mm - I.D. 26,64 mm
NPS 1-SCH 80 = O.D.33,4 mm - WT. 4,55 mm - I.D. 24,30 mm
NPS 1-SCH 160 = O.D.33,4 mm - WT. 6,35 mm - I.D. 20,70 mm
Such as above defined, no inside diameter corresponds to the truth 1" (25,4 mm).
The inside diameter is determined by the wall thickness (WT).

ANSI Standard Steel Pipe Chart ASME B36.10 ASME B36.19

Gauge Sizes | Sheet Metal Gauge Sizes | Pipe Sizes | Pipe Specification | Pipe Schdule | ANSI Pipe Chart | NPS
According to ASME B36.10 and ASME B 36.19.

NPS	OD	Schedule Designations	Wall Thickness	Inside Diameter	Weight
	(Inches)	(ANSI/ASME)	(Inches)	(Inches)	(lbs./ft.)
1/8	0.405	10/10S	0.049	0.307	0.1863
		Std./40/40S	0.068	0.269	0.2447
		XS/80/80S	0.095	0.215	0.3145
1/4	0.54	10/10S	0.065	0.41	0.3297
		Std./40/40S	0.088	0.364	0.4248
		XS/80/80S	0.119	0.302	0.5351
3/8	0.675	10/10S	0.065	0.545	0.4235
		Std./40/40S	0.091	0.493	0.5676
		XS/80/80S	0.126	0.423	0.7388

1/2	0.84	5/5S	0.065	0.71	0.5383	
		10/10S	0.083	0.674	0.671	
		Std./40/40S	0.119	0.622	0.851	
		XS/80/80S	0.147	0.546	1.088	
		160	0.188	0.466	1.309	
		XX	0.294	0.252	1.714	
3/4	1.05	5/5S	0.065	0.92	0.6838	
		10/10S	0.083	0.884	0.8572	
		Std./40/40S	0.113	0.824	1.131	
		XS/80/80S	0.154	0.742	1.474	
		160	0.219	0.618	1.944	
		XX	0.308	0.434	2.441	
1	1.315	5/5S	0.065	1.185	0.8678	
		10/10S	0.109	1.097	1.404	
		Std./40/40S	0.133	1.049	1.679	
		XS/80/80S	0.179	0.957	2.172	
		160	0.25	0.815	2.844	
		XX	0.358	0.599	3.659	
1 1/4	1.66	5/5S	0.065	1.53	1.107	
		10/10S	0.109	1.442	1.806	
		Std./40/40S	0.14	1.38	2.273	
		XS/80/80S	0.191	1.278	2.997	
		160	0.25	1.16	3.765	
		XX	0.382	0.896	5.214	
1 1/2	1.9	5/5S	0.065	1.77	1.274	
		10/10S	0.109	1.682	2.085	
		Std./40/40S	0.145	1.61	2.718	
		XS/80/80S	0.2	1.5	3.631	
		160	0.281	1.338	4.859	
		XX	0.4	1.1	6.408	
2	2.375	5/5S	0.065	2.245	1.604	
		10/10S	0.109	2.157	2.638	
		Std./40/40S	0.154	2.067	3.653	
		XS/80/80S	0.218	1.939	5.022	
		160	0.344	1.689	7.462	
		XX	0.436	1.503	9.029	
2 1/2	2.875	5/5S	0.083	2.709	2.475	
		10/10S	0.12	2.635	3.531	
		Std./40/40S	0.203	2.469	5.793	
		XS/80/80S	0.276	2.323	7.661	

		160	0.375	2.125	10.01	
		XX	0.552	1.771	13.69	
3	3.5	5/5S	0.083	3.334	3.029	
		10/10S	0.12	3.26	4.332	
		Std./40/40S	0.216	3.068	7.576	
		XS/80/80S	0.3	2.9	10.25	
		160	0.438	2.624	14.32	
		XX	0.6	2.3	18.58	
3 1/2	4	5/5S	0.083	3.834	3.472	
		10/10S	0.12	3.76	4.973	
		Std./40/40S	0.226	3.548	9.109	
		XS/80/80S	0.318	3.364	12.5	
		XX	0.636	2.728	22.85	
4	4.5	5/5S	0.083	4.334	3.915	
		10/10S	0.12	4.26	5.613	
		Std./40/40S	0.237	4.026	10.79	
		XS/80/80S	0.337	3.826	14.98	
		120	0.438	3.624	19	
		160	0.531	3.438	22.51	
		XX	0.674	3.152	27.54	
4 1/2	5	Std./40/40S	0.247	4.506	12.53	
		XS/80/80S	0.355	4.29	17.61	
		XX	0.71	3.58	32.43	
5	5.563	5/5S	0.109	5.345	6.349	
		10/10S	0.134	5.295	7.77	
		Std./40/40S	0.258	5.047	14.62	
		XS/80/80S	0.375	4.813	20.78	
		120	0.5	4.563	27.04	
		160	0.625	4.313	32.96	
		XX	0.75	4.063	38.55	
6	6.625	5/5S	0.109	6.407	7.585	
		10/10S	0.134	6.357	9.289	
		Std./40/40S	0.28	6.065	18.97	
		XS/80/80S	0.432	5.761	28.57	
		120	0.562	5.491	36.39	
		160	0.719	5.189	45.35	
		XX	0.864	4.897	53.16	
7	7.625	Std./40/40S	0.301	7.023	23.57	
		XS/80/80S	0.5	6.625	38.05	
		XX	0.875	5.875	63.08	

8	8.625	5S	0.109	8.407	9.914	
		10/10S	0.148	8.329	13.4	
		20	0.25	8.125	22.36	
		30	0.277	8.071	24.7	
		Std./40/40S	0.322	7.981	28.55	
		60	0.406	7.813	35.64	
		XS/80/80S	0.5	7.625	43.39	
		100	0.594	7.439	50.95	
		120	0.719	7.189	60.71	
		140	0.812	7.001	67.76	
		XX	0.875	6.875	72.42	
		160	0.906	6.813	74.69	
9	9.625	Std./40/40S	0.342	8.941	33.9	
		XS/80/80S	0.5	8.625	48.72	
		XX	0.875	7.875	81.77	
10	10.75	5S	0.134	10.482	15.19	
		10S	0.165	10.42	18.7	
		20	0.25	10.25	28.04	
		30	0.307	10.136	34.24	
		Std./40/40S	0.365	10.02	40.48	
		XS/60/80S	0.5	9.75	54.74	
		80	0.594	9.564	64.43	
		100	0.719	9.314	77.03	
		120	0.844	9.064	89.29	
		140	1	8.75	104.13	
		160	1.125	8.5	115.64	
11	11.75	Std./40/40S	0.375	11	45.55	
		XS/80/80S	0.5	10.75	60.07	
		XX	0.875	10	101.63	
12	12.75	5S	0.165	12.42	22.18	
		10S	0.18	12.39	24.2	
		20	0.25	12.25	33.38	
		30	0.33	12.09	43.77	
		Std./40S	0.375	12	49.56	
		40	0.406	11.938	53.53	
		XS/80S	0.5	11.75	65.42	
12	12.75	60	0.562	11.626	73.15	
		80	0.688	11.376	88.63	
		100	0.844	11.064	107.32	
		120	1	10.75	125.49	

		140	1.125	10.5	139.67	
		160	1.312	10.126	160.27	
14	14	10S	0.188	13.624	27.73	
		10	0.25	13.5	36.71	
		20	0.312	13.376	45.61	
		Std./30/40S	0.375	13.25	54.57	
		40	0.438	13.124	63.44	
		XS/80S	0.5	13	72.09	
		60	0.594	12.814	85.05	
		80	0.75	12.15	106.13	
		100	0.938	12.126	130.85	
		120	1.094	11.814	150.9	
		140	1.25	11.5	170.21	
		160	1.406	11.188	189.1	
16	16	10S	0.188	15.624	31.75	
		10	0.25	15.5	42.05	
		20	0.312	15.376	52.27	
		Std./30/40S	0.375	15.25	62.58	
		XS/40/80S	0.5	15	82.77	
		60	0.656	14.688	107.5	
		80	0.844	14.314	136.61	
		100	1.031	13.938	164.82	
		120	1.219	13.564	192.43	
		140	1.438	13.124	223.64	
		160	1.594	12.814	245.25	
18	18	10S	0.188	17.624	35.76	
		10	0.25	17.5	47.39	
		20	0.312	17.376	58.94	
		Std./40S	0.375	17.25	70.59	
		30	0.438	17.124	82.15	
		XS/80S	0.5	17	93.45	
		40	0.562	16.876	104.67	
		60	0.75	16.5	138.17	
		80	0.938	16.126	170.92	
		100	1.156	15.688	207.96	
		120	1.375	15.25	244.14	
		140	1.562	14.876	274.22	
		160	1.781	14.438	308.5	
20	20	10	0.25	19.5	52.73	
		20	0.375	19.25	78.6	

		30	0.5	19	104.13
		40	5.94	18.814	123.11
		60	8.12	18.376	166.4
		80	1.031	17.938	208.87
		100	1.281	17.438	256.1
		120	1.5	17	296.37
		140	1.75	16.5	341.09
		160	1.969	16.064	379.17
NPS	OD	Schedule Designations	Wall Thickness	Inside Diameter	Weight
	(Inches)	(ANSI/ASME)	(Inches)	(Inches)	(lbs./ft.)
22	22	10/10S	0.25	21.5	58.07
		Std./20/40S	0.375	21.25	86.61
		XS/30/80S	0.5	21	114.81
		60	0.875	20.25	197.41
		80	1.125	19.75	250.81
		100	1.375	19.25	302.88
		120	1.625	18.75	353.61
		140	1.875	18.25	403
		160	2.125	17.75	451.06
24	24	10/10S	0.25	23.5	63.41
		Std./20/40S	0.375	23.25	94.62
		XS/80S	0.5	23	125.49
		30	0.562	22.876	140.68
		40	0.688	22.626	171.29
		60	0.969	22.064	238.35
		80	1.219	21.564	296.58
		100	1.531	20.938	367.39
		120	1.812	20.376	429.39
		140	2.062	19.876	483.1
		160	2.344	19.314	542.13
26	26	10	0.312	25.376	85.6
		Std./40S	0.375	25.25	102.63
		XS/80S	0.5	25	136.17
28	28	10	0.312	27.376	92.26
		Std./40S	0.375	27.25	110.64
		20/80S	0.5	27	146.25
		30	0.625	26.75	182.73
30	30	10	0.312	29.376	98.93
		Std./40S	0.375	29.25	118.65
		XS/20/80S	0.5	29	157.53

		30	0.625	28.75	196.08	
32	32	10	0.312	31.376	105.59	
		Std.	0.375	31.25	126.66	
		20	0.5	31	168.21	
		30	0.625	30.75	109.43	
		40	0.688	30.624	230.08	
34	34	10	0.312	33.376	112.25	
		Std.	0.375	33.25	134.67	
		20	0.5	33	178.89	
		30	0.625	32.75	222.78	
		40	0.688	32.624	244.77	
36	36	10	0.312	35.375	118.92	
		Std./40S	0.375	35.25	142.68	
		XS/80S	0.5	35	189.57	
42	42	Std./40S	0.375	41.25	166.71	
		XS/80S	0.5	41	221.61	
		30	0.625	40.75	276.18	
		40	0.75	40.5	330.41	
48	48	Std./40S	0.375	47.25	190.74	
		XS/80S	0.5	47		

Related References:

PVC Pipe Reference
Select Mechanical or Inflatable Pipe Plugs based on the below pipe ID and along with the pressure to be blocked

PVC and CPVC Pipes - Schedule 40					
Nominal Pipe Size (inches)	Outside Diameter (inches)	Minimum Wall Thickness (inches)	Inside Diameter*) (inches)	Weight (lb/ft)	
				PVC	CPVC
1/2	0.840	0.109	0.622	0.16	0.17
3/4	1.050	0.113	0.824	0.21	0.23
1	1.315	0.133	1.049	0.32	0.34
1 1/4	1.660	0.140	1.380	0.43	0.46
1 1/2	1.900	0.145	1.610	0.51	0.55
2	2.375	0.154	2.067	0.68	0.74
2 1/2	2.875	0.203	2.469	1.07	1.18
3	3.500	0.216	3.068	1.41	1.54
4	4.500	0.237	4.026	2.01	2.20
5	5.563	0.258	5.047	2.73	

PVC and CPVC Pipes - Schedule 40					
Nominal Pipe Size (inches)	Outside Diameter (inches)	Minimum Wall Thickness (inches)	Inside Diameter*) (inches)	Weight (lb/ft)	
				PVC	CPVC
6	6.625	0.280	6.065	3.53	3.86
8	8.625	0.322	7.981	5.39	5.81
10	10.750	0.365	10.020	7.55	8.24
12	12.750	0.406	11.938	10.01	10.89
14	14.000	0.438	13.124	11.80	
16	16.000	0.500	15.000	15.43	
PVC and CPVC Pipes - Schedule 80					
1/2	0.840	0.147	0.546	0.20	0.22
3/4	1.050	0.154	0.742	0.27	0.30
1	1.315	0.179	0.957	0.41	0.44
1 1/4	1.660	0.191	1.278	0.52	0.61
1 1/2	1.900	0.200	1.500	0.67	0.74
2	2.375	0.218	1.939	0.95	1.02
2 1/2	2.875	0.276	2.323	1.45	1.56
3	3.500	0.300	2.900	1.94	2.09
4	4.500	0.337	3.826	2.75	3.05
5	5.563	0.375	4.813	3.87	
6	6.625	0.432	5.761	5.42	5.82
8	8.625	0.500	7.625	8.05	8.83
10	10.750	0.593	9.564	12.00	13.09
12	12.750	0.687	11.376	16.50	18.0
14	14.000	0.750	12.500	19.30	
16	16.000	0.843	14.314	25.44	

Inside Diameter = Outside Diameter - 2 x Minimum Wall Thickness
PVC - Polyvinyl Chloride
strong and rigid
resistant to a variety of acids and bases
may be damaged by some solvents and chlorinated hydrocarbons
maximum usable temperature *140°F (60°C)*
usable for water, gas and drainage systems
not useable in hot water systems

Operating Temp (°F)	Operating Temp (°C)	De-Rating Factor
73	22.8	1.00
80	26.7	0.88

Operating Temp (°F)	Operating Temp (°C)	De-Rating Factor
90	32.2	0.75
100	37.8	0.62
110	43.3	0.51
120	48.9	0.40
130	54.4	0.31
140	60	0.22

CPVC - Chlorinated Polyvinyl Chloride
similar to PVC - but designed for water up to approx. 180ºF (82ºC)

Operating Temp (°F)	Operating Temp (°C)	De-Rating Factor
73 - 80	22.8 - 26.7	1.00
90	32.2	0.91
100	37.8	0.82
110	43.3	0.72
120	48.9	0.65
130	54.4	0.57
140	60	0.50
150	65.6	0.42
160	71.1	0.40
170	76.7	0.29
180	82.2	0.25
200	93.3	0.20

SAMPLE PROBLEMS

Water at 15°C is flowing in a 5 cm diameter horizontal pipe made of stainless steel at a rate of 0.0057 m^3/sec. Determine the pressure drop, the head loss over a 60 m long section of the pipe.

For water at 15°C

$$\rho = 999.14 \frac{kg}{m^3}$$

$$\mu = 0.001140 \text{ Pa - sec}$$

$$\nu = 1.141 \times 10^{-6} \frac{m^2}{sec}$$

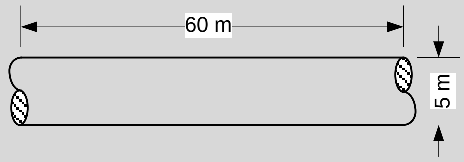
for stainless steel

$$\in = 0.015 \text{ mm}$$

$$\frac{\in}{D} = \frac{0.015}{50} = 0.0003$$

using Colebrook Eq.

$$f \cong 0.015$$



$$h_L = \frac{fLv^2}{2gD} =$$

Example:
Water at 20°C the rate of 30 L/sec is flowing in a 70 m long pipeline, (ND = 100 mm; OD = 114.23 mm; ID = 102.29 mm) containing 1-Open Gate valve; 1 – Check valve and 15 90°-Elbows. Determine the pipe material with minimal head losses if pipe material available are Stainless Steel, Cast Iron and Galvanized Iron.

Pipe Material	Absolute Roughness (∈)
Stainless Steel	0.015
Cast Iron	0.25
Galvanized Iron	0.025

At 20°C Water	Properties	Unit
Density	998.29	Kg/m^3
Absolute Viscosity	0.001003	Pa-sec
Kinematic Viscosity	1.005×10^{-6}	m^2/sec

Fitting	Qty.	L/D	Le	Unit
Gate Valve	1	8	0.8	m
Check Valve	1	135	13.5	m
90°Elbow	15	30	45	m
Total			59.3	m

Formulas

$$v = \frac{Q}{A}$$

$$A = \frac{\pi}{4} D^2$$

$$hf = \frac{f(L + \Sigma Le)v^2}{2gD}$$

$$N_R = \frac{vD}{\nu}$$

$$f = \frac{0.25}{\left[\log\left(\frac{\in/D}{3.7} + \frac{5.74}{N_R^{0.9}}\right)\right]^2}$$

For Stainless Steel

HEAD LOSS CALCULATION		
Q (Capacity)	30	L/sec
Nominal Diameter	100	mm
Outside Diameter (OD)	114.33	mm
Inside Diameter (ID)	102.29	mm
Wall Thickness (t)	6.02	mm
Area (A)	0.0082	m^2
Velocity (v)	3.65	m/sec
Velocity Head (V^2/2g)	0.679	m
(Density) □	998.29	kg/m^3
Absolute Viscocity (m)	0.001003	Pa-sec
Kinematic Viscosity (n)	1.005E-06	m^2/sec
Reynold's Number(NR)	69,161.354	x
L (Length of Pipeline)	70	m
□Le (Equivalent Length)	59.3	m
(L + □Le)		m
Pipe Roughness ∊	0.015	mm
Relative Roughness	0.00015	x
Friction Factor (f)	0.0200	
Head Loss (hf)		m

For Cast Iron Steel

HEAD LOSS CALCULATION		
Q (Capacity)	30	L/sec
Nominal Diameter	100	mm
Outside Diameter (OD)	114.33	mm
Inside Diameter (ID)	102.29	mm
Wall Thickness (t)	6.02	mm
Area (A)	0.0082	m^2
Velocity (v)	3.65	m/sec
Velocity Head (V^2/2g)	0.679	m
(Density) r	998.29	kg/m^3
Absolute Viscocity (m)	0.001003	Pa-sec
Kinematic Viscosity (n)	1.005E-06	m^2/sec

Reynold's Number(NR)	69,161.354	x
L (Length of Pipeline)	70	m
□Le (Equivalent Length)	59.3	m
(L + □Le)		m
Pipe Roughness (Î)	0.25	mm
Relative Roughness	0.00244	x
Friction Factor (f)	0.0271	
Head Loss (hf)		m

For Galvanized Iron

HEAD LOSS CALCULATION		
Q (Capacity)	30	L/sec
Nominal Diameter	100	mm
Outside Diameter (OD)	114.33	mm
Inside Diameter (ID)	102.29	mm
Wall Thickness (t)	6.02	mm
Area (A)	0.0082	m^2
Velocity (v)	3.65	m/sec
Velocity Head (V^2/2g)	0.679	m
(Density) r	998.29	kg/m^3
Absolute Viscocity (m)	0.001003	Pa-sec
Kinematic Viscosity (n)	1.005E-06	m^2/sec
Reynold's Number(NR)	69,161.354	x
L (Length of Pipeline)	70	m
□Le (Equivalent Length)	59.3	m
(L + □Le)		m
Pipe Roughness □□□	0.025	mm
Relative Roughness	0.00024	x
Friction Factor (f)	0.0205	
Head Loss (hf)		m

FUNDAMENTAL EQUATIONS OF CENTRIFUGAL PUMPS

1. Total Head

$H_T = nH$

n - number of stages

H - head per stage

2. Specific Speed: Is the speed in RPM at which a theoretical pump geometrically similar to the actual pump would run at its best efficiency if proportion to deliver 1 m^3/sec against a total head of 1 m. It serves as a convenient index of the actual pump type.

$$N_S = \frac{N\sqrt{Q}}{0.0194(H)^{\frac{3}{4}}}$$

where: Q - flow for a single suction pump in m^3/sec and H is the head per stage

3. Suction Specific Speed (S)01

$$S = \frac{N\sqrt{Q}}{0.0194(NPSH)^{\frac{3}{4}}}$$

$$\frac{N_S}{S} = \left(\frac{NPSH}{H}\right)^{\frac{3}{4}}$$

NPSH - net positive suction head, meters

4. Net Positive Suction Head: The amount of pressure in excess of the vapor pressure of the liquid to prevent cavitation.

$$NPSH = Hp \pm Hz - Hvp - h_{fs}\text{, meters}$$

where: Hp - absolute pressure head at liquid surface at suction, m

Hz - elevation of liquid surface at suction, above or below the pump centerline, m

(+) if above PCL

(-) if below PCL

Hvp - vapor pressure head corresponding the temperature of the liquid, m

h_{fs} - friction head loss from liquid surface at suction to PCL.

5. Cavitation: The formation of cavities of water vapor in the suction side of the pump due to low suction pressure.

CAUSES OF CAVITATION

- Sharp bends.
- High temperature
- High velocity
- Rough surface
- Low atmospheric pressure

EFFECTS OF CAVITATION

- Noise
- Vibration
- Corrosion
- Decreased capacity
- 6. Cavitation Parameter:

$$\delta = \frac{NPSH}{H} = \left(\frac{N_S}{S}\right)^{\frac{4}{3}}$$

$$D = \frac{60\phi\sqrt{2gH}}{\pi N} \quad \text{meter}$$

7. Impeller Diameter;

where: ϕ - peripheral velocity factor whose value ranges from 0.95 to1.09

8. Affinity Laws or Similarity Laws for Centrifugal Machines

a. For Geometrically similar pumps

$Q \propto ND^3$ Power $\propto \gamma N^3D^5$

$H \propto N^2D^2$ $T \propto \gamma N^2D^5$

b. For pumps with Variable Speed and Constant impeller diameter

$Q \propto N$ Power $\propto N^3$

$H \propto N^2$

c. For pumps at Constant Speed with Variable impeller diameter

$Q \propto D$ Power $\propto D^3$

$H \propto D^2$

FUNDAMENTAL EQUATIONS FOR RECIPROCATING PUMPS

Specification: Ds x Dw x L

where: Ds - diameter of steam cylinder

Dw - diameter of water cylinder

L - length of stroke

1. Volumetric Efficiency

$$\eta_V = \frac{Q}{V_D} \text{ x } 100\%$$

where: Q - capacity, m^3/sec

V_D - displacement volume, m^3/sec

$$F_w = e_m F_s$$

$$F_w = \frac{\pi (Dw)^2 (P_d - P_{su})}{4} \text{ KPa}$$

$$\frac{D_s}{D_w} = \sqrt{\frac{(P_d - P_{su})}{e_m (P_s - P_e)}}$$

2. Displacement Volume

a. For Single acting

$$V_D = \frac{\pi L (D_W)^2 Nn'}{4(60)} \frac{m^3}{sec}$$

b. For Double acting without considering piston rod

$$V_D = \frac{2\pi L (D_W)^2 Nn'}{4(60)} \frac{m^3}{sec}$$

c. For Double acting considering piston rod

$$V_D = \frac{\pi LNn'}{4(60)} \left(2D_W{}^2 - d^2\right) \frac{m^3}{sec}$$

where: N - no. of strokes per minute

L - length of stroke, m

D - diameter of bore, .

d - diameter of piston rod, m

n' - no. of cylinders

n' = 1 (For Simplex)

n' = 2 (For Duplex)

n' = 3 (For Triplex)

3. Percent Slip

$$\%Slip = 100 - \eta_v$$

4. Slip

$$Slip = V_D - Q$$

5. Thermal Efficiency

$$e = \frac{3600(WP)}{m_s(h_s - h_e)} \text{ x } 100\%$$

where: hs - enthalpy of supply steam, KJ/kg
he - enthalpy of exhaust steam, KJ/kg
m_s - steam flow rate, kg/hr
FP - fluid power, KW

$$F_s = \frac{\pi D_S{}^2 (P_s - P_e)}{4} \quad \text{KPa}$$

6. Force produced and acting on the piston rod
where: Ps - supply steam pressure, KPa
Pe - exhaust steam pressure, KPa
Ds - diameter of steam cylinder, m
(Ps - Pe) - mean effective pressure

7. Force transmitted to the liquid piston
where: e_m - mechanical efficiency

$$F_w = e_m F_s$$

$$F_W = \frac{\pi (Dw)^2 (P_d - P_{su})}{4} \text{ KPa}$$

$$\frac{D_s}{D_w} = \sqrt{\frac{(P_d - P_{su})}{e_m (P_s - P_e)}}$$

Psu - suction pressure of water cylinder, KPa
Pd - discharge pressure of water cylinder, KPa

8. Pump Duty: Work done on the water cylinder expressed in Newton-meter per Million Joules

$$\text{Pump Duty} = \frac{9.81 m_w (H_d - H_{su}) \text{ x } 10^6}{1000 m_s (h_s - h_e)} \quad \frac{\text{N - m}}{\text{Million Joules}}$$

where: m_w - water flow rate, kg/hr
Hd - discharge head of pump, m
Hsu - suction head of pump, m

9. Pump Speed

$$V = 43.64(L)^{1/2}(f_t), \text{ m/min}$$

$$V_D = \frac{\pi (D_W)^2 Vn'}{4(60)} \quad \frac{m^3}{sec}$$

where: f_t - temperature correction factor
L - length of stroke, m

10. Temperature Correction Factor

f_t	= 1 For cold water
	= 0.85 for 32.2°C water
	= 0.71 for 65.5°C water
	= 0.55 for 204.4°C water

11. For Indirect Acting pumps

$$N = \frac{907 f_t}{\sqrt{L}}$$

EXAMPLE

A mechanical engineer of an industrial plant wishes to install a pump to lift 15 L/sec of water at 20°C from a sump to a tank on a tower. The water is to be delivered into a tank 200 KPag. The water level in the tank is 20 m above the pump centerline and the pump is 4 m above the water level in the sump. The suction pipe is 100 mm in diameter, 7.0 m long and will contain 2 - standard elbows and 1 - Foot valve. The discharge pipe to the tank is 75 mm in diameter and is 120 m long and contains 5 - 90°elbows, 1 - check valve, and 1 - gate valve. Pipe material is Cast iron. Assuming a motor-pump efficiency of 75%, calculate the KW power required and the current drawn by the 3 – Phase motor drive if E = 220 Volts and 0.90 Power Factor.

From table

At 20°C

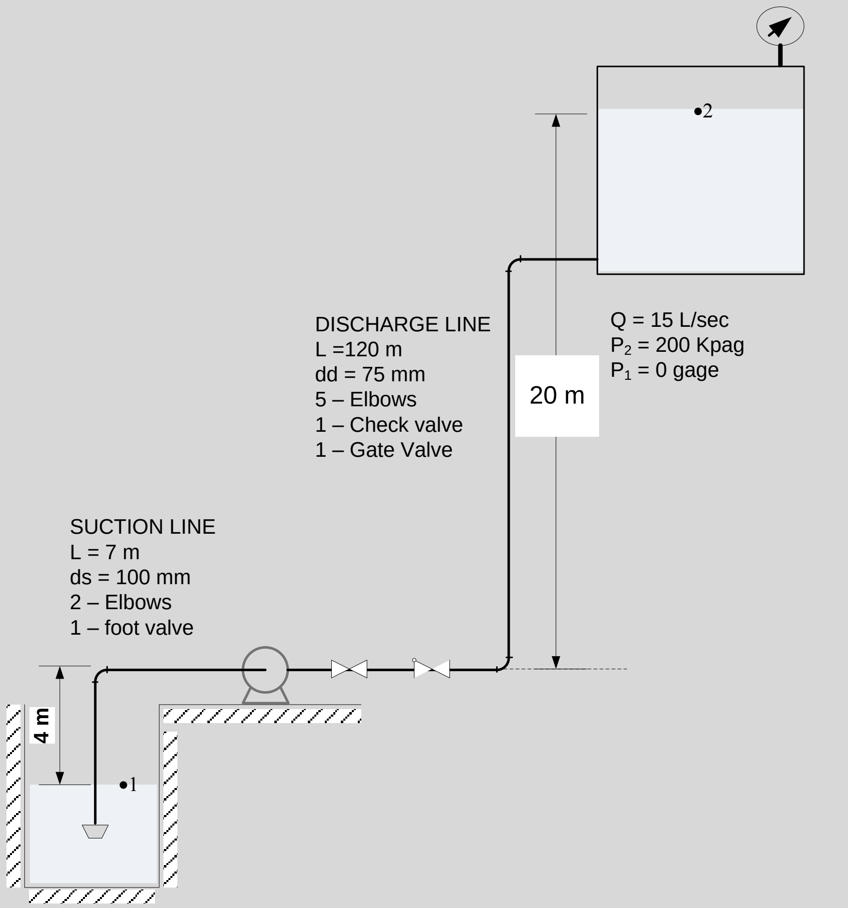
$\rho = 998\ \text{kg/m}^3$

$\gamma = 9.79\ \text{KN/m}^3$

$\in = 0.25$ mm

$\mu = 1.002 \times 10^{-3}$ Pa-sec

$\nu = 1.004 \times 10^{-6}\ \text{m}^2/\text{sec}$



$$h_f = \frac{fLv^2}{2gD}$$

$$h_f = \mathbf{K}\frac{v^2}{2g}$$

$$\mathbf{K} = \frac{fL}{D}$$

$$N_R = \frac{\rho v D}{\mu} = \frac{vD}{\nu}$$

$\in$ - absolute roughness
D - inside diameter
$\in$/D - relative roughness

Considering Major and Minor losses

$$h_f = \frac{f(L + \Sigma L_e)v^2}{2gD} \text{ meters}$$

Suction Line

L	7 m
ID	0.100 m
∈	0.25 mm
∈/D	0.0025
NR	190,224
v_d	1.91 m/sec
fd	0.026

$$\frac{\in}{D} = \frac{0.25}{100} = 0.0025$$

$$A = \frac{\pi}{4}(0.100)^2 = 0.00785\,m^2$$

$$v_s = \frac{0.015}{0.00785} = 1.91\frac{m}{sec}$$

$$N_R = \frac{1.91(0.100)}{1.004x10^{-6}} = 190{,}224$$

From Moody diagram

$$f = 0.026$$

$$h_{L\,Suction} = \frac{0.026(7+13.5)(1.91)^2}{2(9.81)(0.100)} = 0.99\,m$$

Fitting	L/D	Quantity	D	Le
90°Standard Elbow	30	2	0.100	6
Foot Valve	75	1	0.100	7.5
Total (Le)				13.5

Discharge Line

L	120 m
ID	0.075 m
∈	0.25 mm
∈/D	0.0033
NR	253,631
v_d	3.40 m/sec
fd	0.028

$$\frac{\in}{D} = \frac{0.25}{75} = 0.0033$$

$$A_d = \frac{\pi}{4}(0.075)^2 = 0.00442\,m^2$$

$$v_d = \frac{0.015}{0.00442} = 3.4\frac{m}{sec}$$

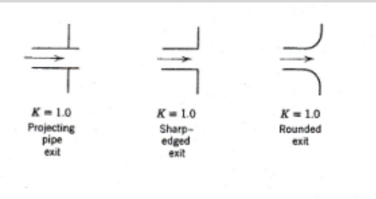
$$N_R = \frac{3.4(0.100)}{1.004x10^{-6}} = 253{,}631$$

From Moody diagram

$$f_d = 0.028$$

$$h_{L\,Discharge} = \frac{0.028(120+25.03)(3.40)^2}{2(9.81)(0.075)} = 32\,m$$

For Pipe Exit



$$K = 1$$

$$K = \frac{fL}{D}$$

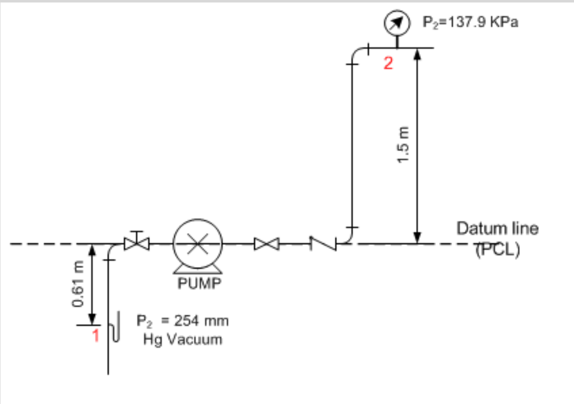
$$\frac{L}{D} = \frac{K}{f} = \frac{1}{0.028} = 35.7$$

Fitting	L/D	Quantity	D	Le
90°Standard Elbow	30	5	0.075	11.25
Check Valve	135	1	0.075	10.125
Gate Valve	13	1	0.075	0.975
Pipe Exit	35.7	1	0.075	2.68
Total (Le)				25.03

$$H_L = h_{Ls} + h_{Ld}$$
$$H_L = 32 + 0.99 = 32.99\, m$$
$$ht = \frac{P_2 - P_1}{\gamma} + \frac{v_2^{\,2} - v_1^{\,2}}{2g} + (z_2 - z_1) + H_L$$
$$v_1 = 0$$
$$v_2 = 0$$
$$P_1 = 0 \text{ gage}$$
$$z_1 = 20\, m$$
$$z_2 = \text{-}4\, m$$
$$ht = \frac{200}{9.79} + 0 + (20 + 4) + 32.99 = 77.42\, m$$
$$\text{Water Power} = Q\gamma h_t$$
$$\text{Efficiency} = \frac{\text{Water Power}}{\text{Motor Power}} \text{x}100\%$$
$$MP = \frac{0.015(9.79)(77.42)}{0.75} = 15.2\ \ KW$$
$$MP = \frac{\sqrt{3}\, EI(\text{Power Factor})}{1000}$$
$$I = \frac{1000(15.2)}{220(0.90)\sqrt{3}} = 44.3 \text{ Amperes}$$

EXAMPLE NO. 1

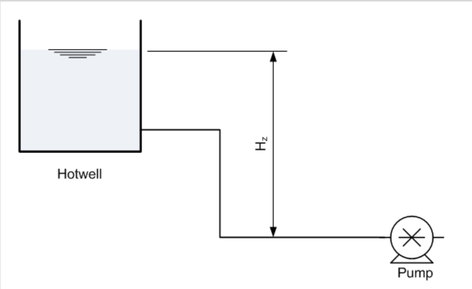
The diameters of the suction and discharge pipes of a pump are 152 mm and 102 mm, respectively. The discharge pressure is read by a gauge at a point 1.5 m above the pump centerline and the suction pressure is read by a gauge 61 cm below the pump centerline. If the discharge pressure gauge reads 137.9 KPa and the suction gauge reads a vacuum of 254 mm Hg, when gasoline with s = 0.75 is pumped at the rate of 34 L/sec, find the power delivered to the fluid. If the pump is 75% efficient and is driven by a 3-phase, 440 V, 0.9 power factor and 92% efficiency induction motor, determine the cost of power for 24 hours operation, if power costs P 0.30/KW-hr, and the line current drawn by the motor. (FP = 6.54 KW; C = P51/day; I = 10.37 amp; ht = 26.16 m)



INPUT DATA		CALCULATED DATA	
Q	0.034	A1	0.018
d1	152	A2	0.008
d2	102	v1	1.87
P1	-33.9	v2	4.16
P2	137.9	(v2^2-v1^2)/2g	0.70
z1	-0.61	(P2-P1/γ)	23.35
z2	1.5	(z2-z1)	2.11
g	9.81	ht (m)	26.16
HL	0	FP	6.54
γ	7.36	MP	7.11
Eff.(Overall)	0.92	I (Amperes)	10.37
E (Volts)	440	Pumping Cost	51.21
3 - Phase	1.73		
Power Factor	0.9		
Power Cost	0.30		
Pumping Time	24		

EXAMPLE NO. 2

A boiler feed pump delivers water at 100C (ρ = 958.31 kg/m^3;γ = 9.40 KN/m3) which it draws from an open hot-well with a friction loss of 610 mm in the intake pipe between it and the hot-well. The barometric pressure is 737 m Hg and the value of the cavitation parameter for the pump is 0.10. What must be the elevation of the water surface in the hot-well relative to that of the pump intake? The total pumping head is 73 m. (Hz = 8.24 m)



INPUT DATA		CALCULATED DATA	
t	100	NPSH	7.3
ρ	958.31	Hp	10.45
γ	9.4	Hvp	10.78
Pa	98.25858553	Hz	8.24
Psat	101.33		
hfs	0.61		
δ	0.1		
H	73		

NPSH = Hp ± Hz - Hvp - h_{fs} , meters

$$\delta = \frac{NPSH}{H}$$

EXAMPLE 3

A centrifugal pump design for a 1800 RPM operation and a head of 61 m has a capacity of 190 L/sec with a power input of 132 KW. What effect will a speed reduction to 1200 RPM have on the head, capacity and power input of the pump? What will be the change in H, Q and BP if the impeller diameter is reduced from 305 mm to 254 mm while the speed is held constant at 180 RPM. Neglect effects of fluid viscosity.

INPUT DATA		CALCULATED DATA	
Constant Impeller Diameter (Variable Speed)			
N1	1800	N2/N1	0.67
H1	61	Q2	126.67
Q1 (L/sec)	190	H2	27.11
BP1	132	BP2	39.11
N2	1200		
Constant Speed (Variable Impeller Diameter)			
D1 (mm)	305	D2/D1	0.8
D2 (mm)	254	Q2	158.2
		H2	42.3
		BP2	76.2

For pumps with Variable Speed and Constant impeller diameter

$Q \propto N$ Power $\propto N^3$

$H \propto N^2$

For pumps at Constant Speed with Variable impeller diameter

$Q \propto D$ Power $\propto D^3$

$H \propto D^2$

EXAMPLE 4

A plant has installed a single suction centrifugal pump with a discharge of 68 m^3/hr under a 60 m head and running at 1200 RPM. It is proposed to install another pump with double suction but of the same type to operate at 30 m head and deliver 90 m^3/hr.

a) Determine the speed of the proposed pump (877 RPM)

b) What must be the impeller diameter of the proposed pump if the diameter of the existing pump is 150 mm. (145 mm)

INPUT DATA		CALCULATED DATA	
Q1 (m3/sec)	0.0189	Ns1 = Ns2	394.34
H1	60	N2	877.12
N1	1200	Q2/Q1	0.66
Q2 (Double Suction)	0.025	N2/N1	0.73
Q2(single suction)	0.0125	D2 (mm)	145
H2	30		
ϕ	0.95		
D1 (mm)	150		

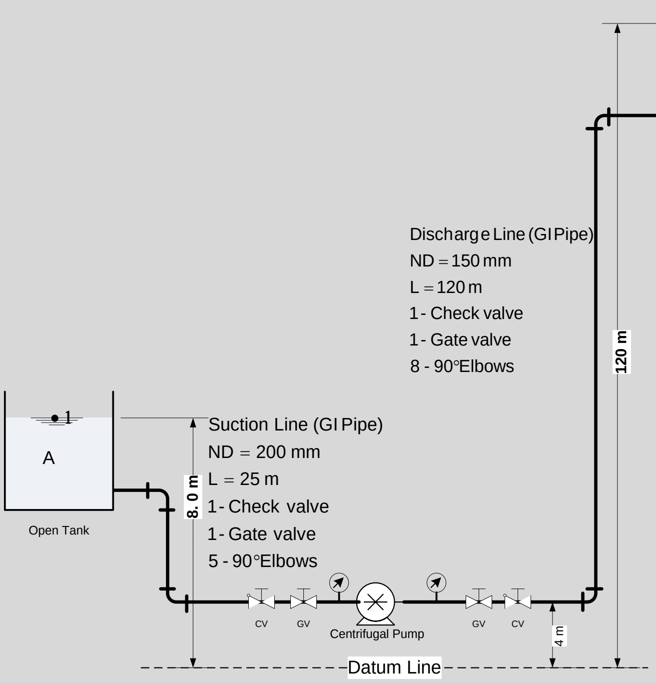
$$N_s = \frac{N\sqrt{Q}}{0.0194\,(H)^{\frac{3}{4}}}$$

$Q \propto ND^3$

$H \propto N^2D^2$

Design Problem No. 1

Water (@ t = 20°C) from an open reservoir A at 8 m elevation is drawn by a motor driven centrifugal pump to an open reservoir B at 70 m elevation. The suction line is GI pipe; ND = 200 mm (NPS = 8"); L = 25 m containing 1 –check valve; 1 – Gate valve; 5 standard 90° elbows. Discharge line is GI pipe; ND = 150 mm (NPS = 6"); L = 120 m containing 1-Gate valve, 1-Check valve, and 8 standard 90° elbows. Assume shard edge for pipe entrance and pipe exit. Pump elevation is at 4 m. Overall efficiency of the system is 78%. For a discharge rate of 10 L/sec and power cost of P 7.00 per KW-hr. Determine
The total dynamic head
The water power
The power required by the motor
The power cost for 10 hrs pumping time
The line current drawn by the motor for E = 220 Volts and power factor PF = 0.92
The pressure gauge reading installed just at the inlet and outlet of the pump
Note: Draw the sketch of the problem



HYDRAULIC MACHINERY: Activity 2 (Pump Design)
Determine the Motor power required of the pumping system shown below.

Avialable Material
1. Stainless steel
2. Galvanized Iron (new)
3. Cast Iron (new)

$t = 20°C$

$\gamma = 9.789 \frac{KN}{m^3}$

$\rho = 998.2 \frac{kg}{m^3}$

$\mu = 1.002 \times 10^{-3}$ Pa-sec

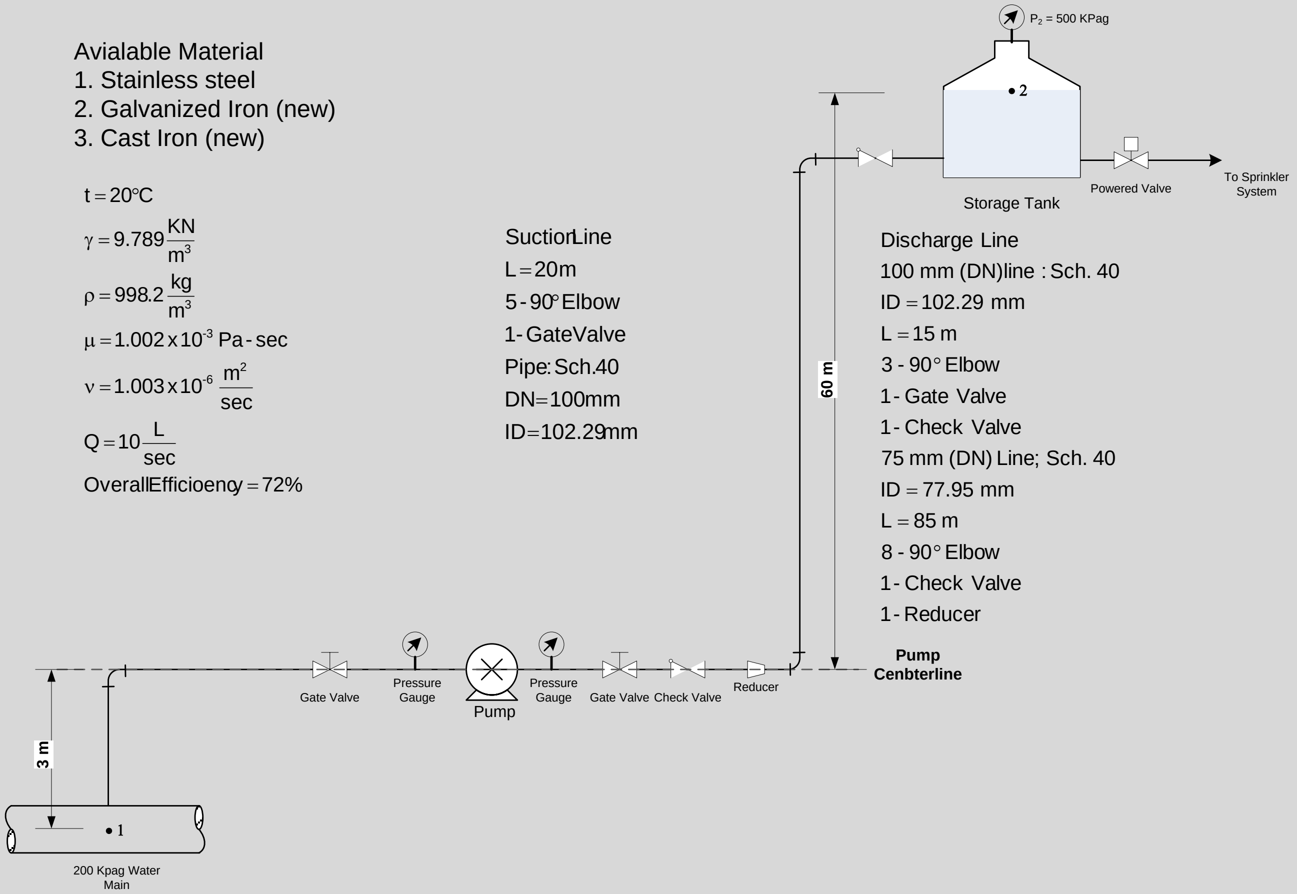
$\nu = 1.003 \times 10^{-6} \frac{m^2}{sec}$

$Q = 10 \frac{L}{sec}$

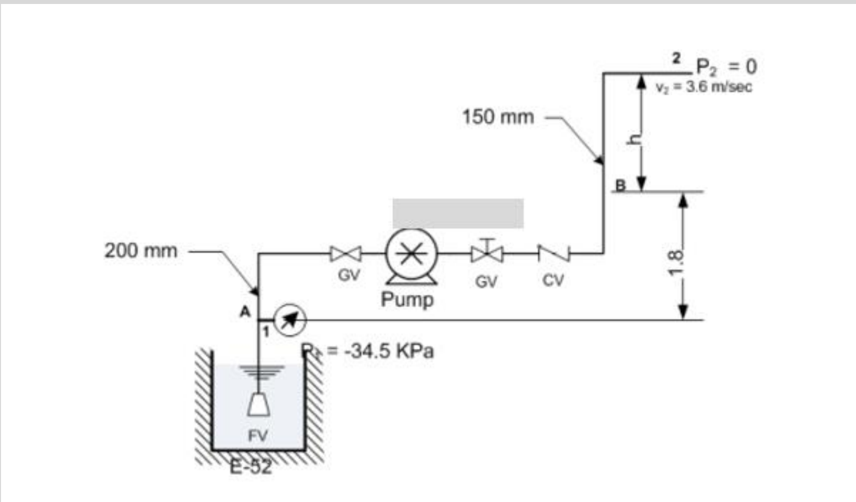
Overall Efficioency = 72%

Suction Line
L = 20m
5 - 90° Elbow
1 - Gate Valve
Pipe: Sch.40
DN = 100mm
ID = 102.29mm

Discharge Line
100 mm (DN) line : Sch. 40
ID = 102.29 mm
L = 15 m
3 - 90° Elbow
1 - Gate Valve
1 - Check Valve
75 mm (DN) Line; Sch. 40
ID = 77.95 mm
L = 85 m
8 - 90° Elbow
1 - Check Valve
1 - Reducer



A 15 KW suction pump draws water from a suction line whose diameter is 200 mm and discharges through a line whose diameter is 150 mm. The velocity in 150 mm line is 3.6 m/sec. If the pressure at point A in the suction line is 34.5 KPa below the atmosphere where A is 1.8 m below that of B on the 150 mm line, Determine the maximum elevation above B to which water can be raised assuming a head loss of 3 m due to friction.

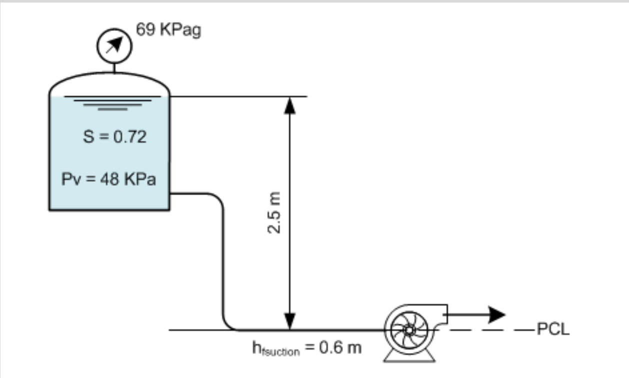


HYDRAULIC MACHINERY(QUIZ NO. 2)

Gasoline at 38°C being drawn from a closed tank having a pressure of 69 KPag in a plant located 900 m above sea level. The level of gasoline in the tank is 2.5 m above the pump centerline. The suction line friction and turbulence head losses amounts to 0.6 m. The vapor pressure of the gasoline is 48 KPa absolute and the relative density is 0.72.

a) What is the NPSH of the system

b) If the cavitation parameter is 0.10 and the discharge is 400 L/sec, what is the size of the drive motor required? Combined pump-motor efficiency is 55%.



FROM NASA

1. Atmospheric pressure decreases at the rate of 83.312 mm Hg per 1000 meters rise in elevation.
2. Atmospheric temperature decreases at the rate of 6.5°K per 1000 meters rise in elevation.

$$NPSH = Hp \pm Hz - Hvp - h_{fs} \text{, meters}$$

$$\delta = \frac{NPSH}{H} \qquad \eta_c = \frac{WP}{MP} \times 100\%$$

INPUT DATA		CALCULATED DATA	
h (Elevation)	900	Pa	91.33
Pgauge (Kpag)	69	Hp	12.93
hz	2.5	Hvp	6.80
hfs	0.6	NPSH	8.03
Pv (Kpaa)	48	H	80.34
S	0.72	Motor Power	412.72
g (m/sec^2)	9.81		
Q (m^3/sec)	0.400		
Efficiency	0.55		
δ	0.1		
γ	7.0632		

At its optimum point of operation a given centrifugal pump with an impeller diameter of 50 cm delivers 3.2 m^3/sec of water against a head of 25 m when operating at 1450 RPM.

a. If the efficiency is 82%, what is the brake power of the driving shaft.

b. If a homologous pump with an impeller diameter of 80 cm is rotating at 1200 RPM, what would be the Q, H, and BP.

c. Compute the specific speed of both pumps.

$$N_s = \frac{N\sqrt{Q}}{0.0194\,(H)^{\frac{3}{4}}}$$

For pumps with Variable Speed and Constant impeller diameter

$Q \propto N$ $Power \propto N^3$

$H \propto N^2$

For pumps at Constant Speed with Variable impeller diameter

$Q \propto D$ $Power \propto D^3$

$H \propto D^2$

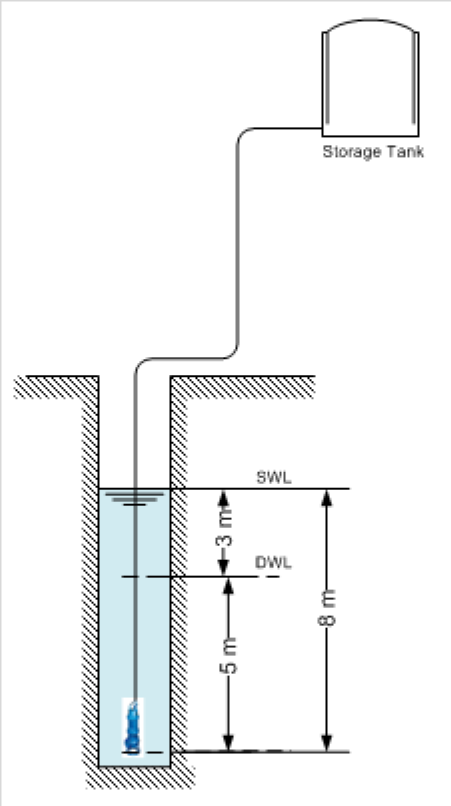
INPUT DATA		CALCULATED DATA	
D1	50	BP1	1426.9
Q1	3.2	(N2/N1)	0.83
H1	25	Q2	2.65
N1	1450	H2	17.12
Efficiency	0.82	BP2	808.79
D2	80	Ns	11,959
N2	1200		
Efficiency	0.55		
δ	0.1		
γ	784.8		

A submersible, multistage centrifugal deep-well pump 984 L/min capacity is installed in a well 8 m below the static water level. Draw-down when pumping at rated capacity is 3 m. The pump delivers the water into a 95 Liters capacity overhead storage tank. Total discharge head developed including friction in piping is 74 m. Calculate:

a) Brake power required to drive the pump if pump efficiency is 70%

b) Determine the number of stages in the pump if each impeller develops a head of 12 m at 3450 RPM

c) Determine the diameter of the impeller of this pump in mm.



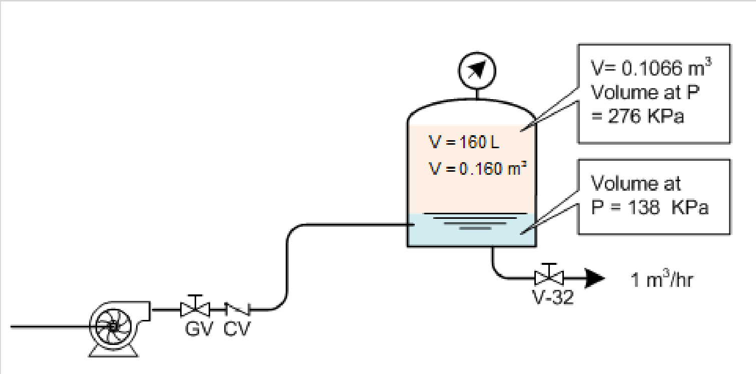
INPUT DATA		CALCULATED DATA	
Q	0.0164	Ht	69.0
SWL	8	BP	16
DWL	3	n (stages)	6
H Discharge	74	ϕ	0.95
Efficiency	0.7	D	0.19
H	12	D (mm)	193
N	3450		

$$D = \frac{60\phi\sqrt{2gH}}{\pi N} \text{ meter}$$

$$Ht = (74 - (8 - 3)) = 69 \text{ m}$$

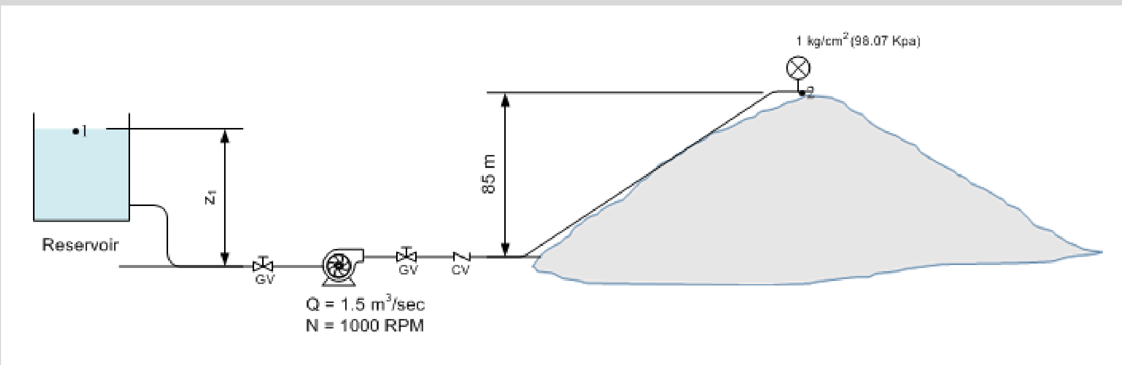
$$n = \frac{69}{12} = 6 \text{ stages}$$

HYDRAULIC MACHINERY
ACTIVITY NO. 2 (May 06, 2017)
NAME ______________________________________
No. 1
A 4 m^3/hr pump delivers water to a pressure tank. At the start, the gauge reads 138 KPa until it reads 276 KPa and then the pump was shut off. The volume of the tank is 160 Liters. At 276 KPa, the water occupied 2/3 of the tank volume.
a) Determine the volume of water that can be taken out until the gauge reads 138 KPa
b) If 1 m^3/hr of water is constantly used, in how many minutes from 138 KPa will the pump run until the gauge reads 276 KPa



$$\text{At } P = 276\,\text{KPa}$$
$$\text{Pair} = 276 + 101.325 = 377.325\,\text{KPa}$$
$$\text{Vair} = \tfrac{1}{3}(0.160) = 0.0533\,m^3$$
$$\text{at } P = 138\,\text{KPa}$$
$$\text{Pair} = 138 + 101.325 = 239.325\,\text{KPa}$$
assumng isothermal conditions for air in the tank
$$V_{\text{air at 138 KPa}} = \frac{377.325(0.0533)}{239.325} = 0.084\,m^3, \text{and the volume of water remaining in the tank is}$$
$$V_{\text{water at 138 KPa}} = 0.160 - 0.084 = 0.076\,m^3$$
$$\text{Volume of water taken out} = \tfrac{2}{3}(0.160) - 0.076 = 0.0307\,m^3$$
$$t - \text{pumping time if } 1m^3/\text{hr is constantly used}$$
$$4t - 1t = 0.0307$$
$$t = \frac{0.0307}{3}\,\text{hrs} = 0.01022\,\text{hrs} = 0.6133\,\text{min} = 36.8\,\text{sec}$$

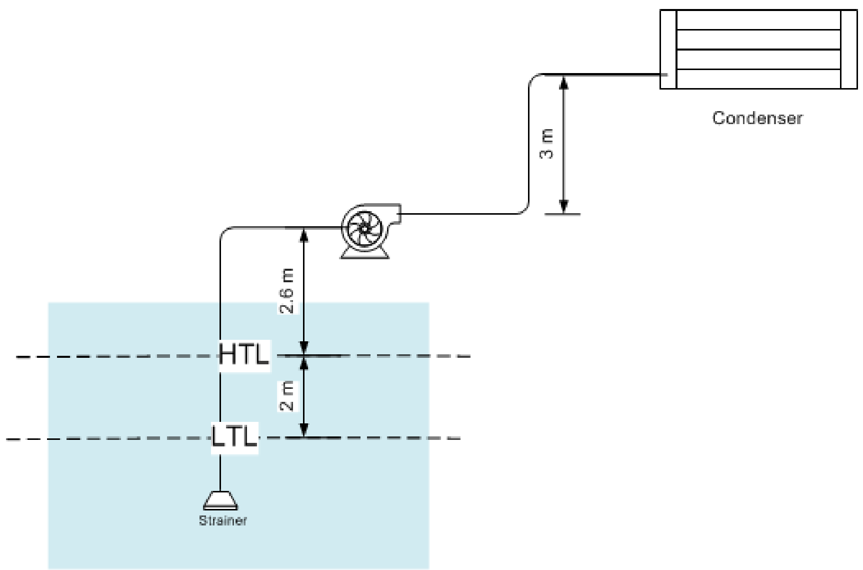
No. 2
Water from a reservoir is pumped over a hill through a pipe 900 mm diameter and a pressure 1 kg/cm^2 is maintained at the pipe discharge where the pipe is 85 m from the pump centerline. the pump has a positive suction head of 5 m. Pumping rate of the pump at 1000 RPM is 1.5 m^3/sec. Friction loss is equivalent to 3 m of head loss.
a) What amount of energy must be furnished by the pump in KW
b) If the speed of the pump is 1200 RPM, what are the new values of Q, h, and power



GIVEN DATA		CALCULATED DATA	
ID	0.9	Sp. Wgt	9.81
A2	0.636	v2	2.36
A1	0.000	v1	0
P1	0	(P2-P1)/γ	10.00
P2	98.09	(z2-z1)	80
z1	5	(v2^2-v1^2)/2g	0.28
z2	85	ht	93.28
N	1000	Pump Power	1372.6
Q	1.5	(N2/N1)	1.2
HL	3	Q2	1.8
N1	1000	ht2	134.33
N2	1200	Pump Power 2	2371.9

2. A double suction single stage centrifugal pump is delivering 900 m^3/hr of sea water (S = 1.03) from a source where the water level varies 2 m from a high tide to low tide level. The pump centerline is located 2.6 m above the surface of the water at high tide level. The pump discharges into a surface condenser 3 m above the pump centerline. Loss of head due to friction in suction pipe is 0.8 m and that in the discharge side is 3 m. Pump is directly connected to a 1750 RPM,460 Volts, 3-phase, 60 Hertz motor.

a) Make a sketch of the installation in accordance with good practice.
b) Calculate the total suction head in m
c) Calculate the total discharge head in m
d) Calculate the specific speed of the pump in RPM
e) Calculate the BP of the pump if efficiency is 82%
f) Calculate the impeller diameter in mm.



$$N_s = \frac{N\sqrt{Q}}{0.0194\,(H)^{\frac{3}{4}}} \qquad D = \frac{60\phi\sqrt{2gH}}{\pi N} \text{ meter}$$

Given		Calculated Data	
Q	0.25	Hsuction	4.6
S	1.03	Hdischarge	3
γ	10.1043	ht	11.4
z2	7.6	Pump Power	28.8
z1	0	Ns	7,270
HL	3.8	Brake Power	35.12
N	1750	Efficiency	0.82
E	460	ϕ	0.95
f	60	Diameter (Impeller)	155
Motor (3 Phase)	1.732		

SAMPLE PROBLEMS (INDUSTRIAL PLANT)

1. A centrifugal pump with a 30.5 cm. diameter impeller requires a power input of 45 KW when the flowrate is 200 L/sec against a 18 m head. The impeller is changed to one with a 25.5 cm. diameter. Determine the expected flowrate, head, and input power if the pump speed remains the same.

   $Q \alpha D$

   $H \alpha D^2$

   $\text{Power} \alpha D^3$

Q1	H1	P1	D1
200	18	45	30.5
Q2	H2	P1	D2
167.21	12.6	26.30	25.5

2. A small centrifugal pump, when tested at 2875 rpm with water, delivered a flow rate of 16 L/sec and a head of 42 m at its best efficiency point (efficiency is 76%). Determine the specific speed of the pump at this test condition. Compute the required power input to the pump.

$$N_S = \frac{N\sqrt{Q}}{0.0194(H)^{\frac{3}{4}}}$$

$$Ns = \frac{2875\sqrt{0.016}}{0.0194(42)^{\frac{3}{4}}}$$

Ns	N	Q	H
1136.21	2875	0.016	42

3. A plant has installed a single suction centrifugal pump with a discharge of 68 m^3/hr under a 60 m head and running at 1200 RPM. It is proposed to install another pump with double suction but of the same type to operate at 30 m head and deliver 90 m^3/hr.

   a) Determine the speed of the proposed pump (877 RPM)

   b) What must be the impeller diameter of the proposed pump if the diameter of the existing pump is 150 mm. (145 mm)

Given:

Q1 = 68 m^3/hr = 0.0189 m^3/sec
H1 = 60 m
N1 = 1200 RPM
For the proposed pump
H2 = 30 m
Q2 = 90 m^3/hr
Q2 = 45 m^3/hr = 0.0125 m^3/sec→ For single suction

$$N_S = \frac{N\sqrt{Q}}{0.0194(H)^{\frac{3}{4}}}$$

N_{S1}	N	Q	H
394.45	1200	0.0189	60
N	Ns	Q	H
877	394.5	0.0125	30

$$\frac{Q_2}{Q_1} = \frac{N_2}{N_1}\left(\frac{D_2}{D_1}\right)^3$$

$$\frac{H_2}{H_1} = \left(\frac{N_2}{N_1}\right)^2\left(\frac{D_2}{D_1}\right)^2$$

For Geometrically similar pumps

$Q \propto ND^3$ $\qquad$ $\text{Power} \propto \gamma N^3D^5$

$H \propto N^2D^2$ $\qquad$ $T \propto \gamma N^2D^5$

Q1	H1	D1	N1
0.0189	60	0.15	1200
D2	Q2	H2	N2
0.145	0.0125	30	877

To be continued!!!!!!!