Engineering Fundamentals-Thermodynamics

By Professor Paul A. Erickson

Basic Thermodynamics

- Conservation of Mass
- Conservation of Energy
- Principle of State and Phase
- Principle of Entropy
- Thermodynamic Cycles
  - Carnot
  - Rankine
  - Air Standard Cycles
    - Otto
    - Diesel
    - Brayton
  - Refrigeration Cycle (also Heat Pump)
- Air/Water Mixtures

Basic Thermo Processes and Terms

- Adiabatic
- Isothermal
- Isobaric
- Isometric or Isochoric
- Isentropic
- Isenthalpic
- Polytropic

- Ideal Gas
- Incompressible substances

- Intensive and Extensive Properties
- Compressibility
- Reduced Temperature and Pressure
- Constant and Variable Specific Heat
- Relative Pressure and Volume
- Flow Work
- Boundary Work
- Enthalpy
- Entropy
- Thermal Efficiency and COP
- Relative and Absolute Humidity

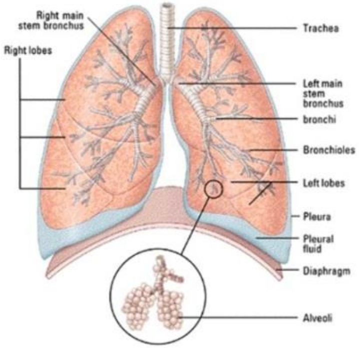
Conservation of Mass

Epicurus (341–270 BC). Describing the nature of the universe, "the sum total of things was always such as it is now, and such it will ever remain."

Input

- Output

= Storage

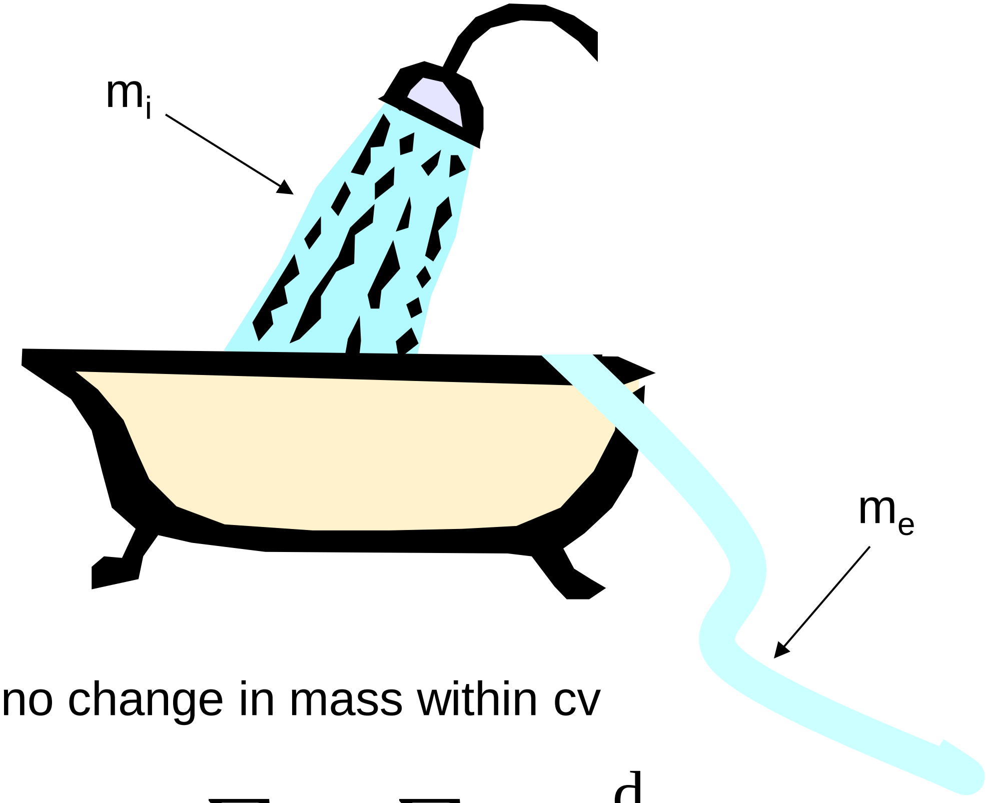


Must be linked with Conservation of Energy $E = mc^2$

Continuity Equation

Conservation of Mass

(Total Mass Entering a System)-(Total Mass Leaving the System) = Net change in mass within system



Steady flow implies no change in mass within cv

$$\sum \dot{m}_e - \sum \dot{m}_i + \frac{d}{dt} m_{cv} = 0$$

Conservation of Energy

Lights on or off Energy is always conserved… But that isn't what they mean!

- Energy is neither created nor destroyed

1st Law of Thermodynamics

Conservation of Energy
Energy can neither be created nor destroyed

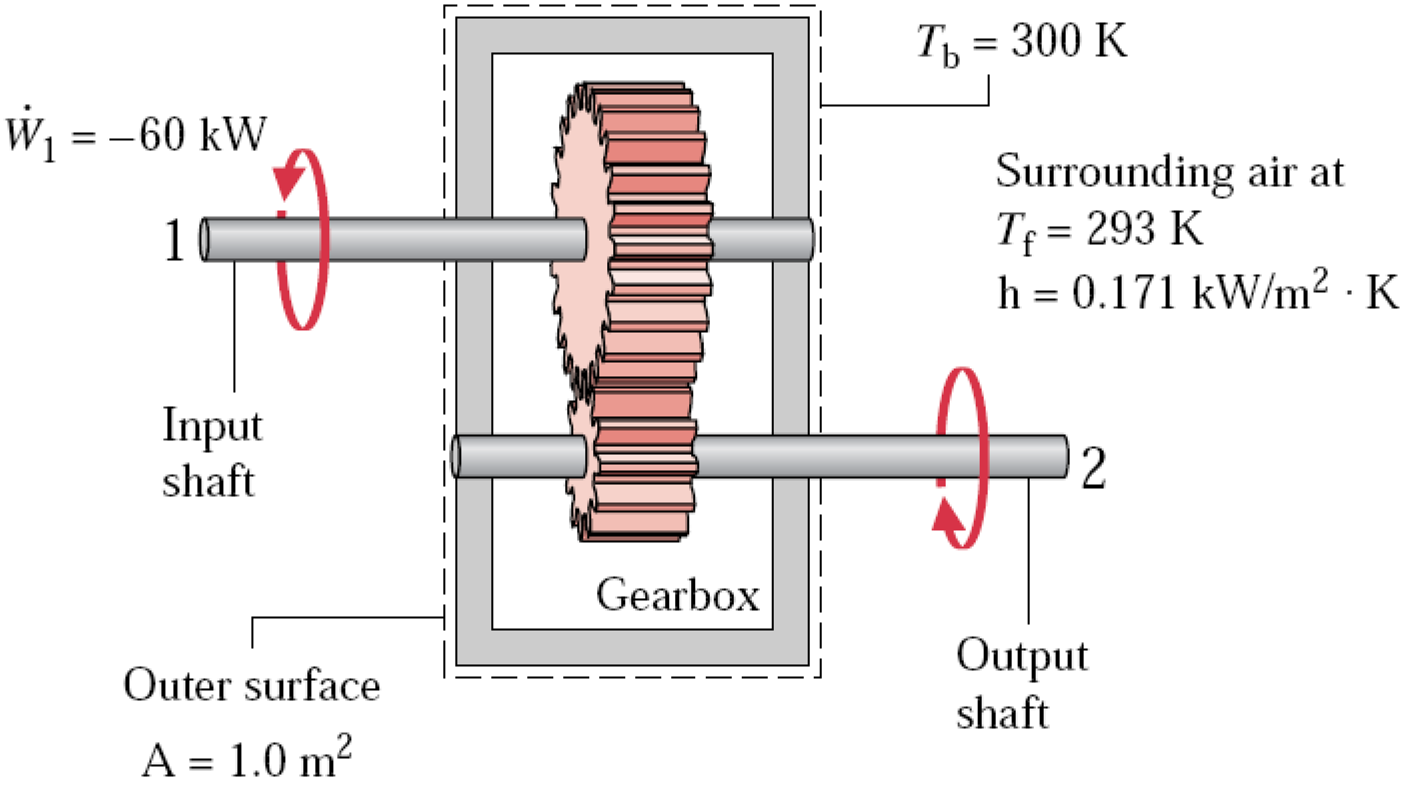
$$Input - Output = Storage$$

$$\dot{W} - \dot{Q} + \sum \dot{m}e_{out} - \sum \dot{m}e_{in} + \frac{d}{dt}\left[me_{cv}\right] = 0$$

Where W is the work done by the system Q is the heat transferred to the system and

$$e = h + \frac{V^2}{2} + gz$$

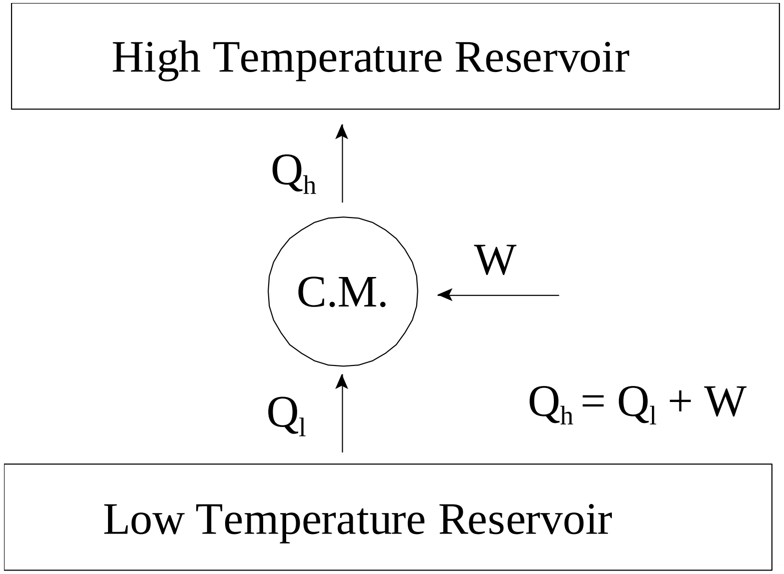
Using Conservation of Energy



What is the output shaft power?

Using Conservation of Energy

A refrigeration cycle has heat transfer Q_{out} 3200 Btu and net work of W_{cycle} 1200 Btu. Determine the coefficient of performance for the cycle.



$Q_h = Q_l + W$

$$\beta = \frac{Q_{in}}{W_{cycle}}$$

$$W_{cycle} = Q_{out} - Q_{in}$$

$$Q_{in} = Q_{out} - W_{cycle}$$

$$= 3200 - 1200$$

$$= 2000 \text{ Btu}$$

$$\beta = \frac{2000}{1200} = 1.667 \longleftarrow$$

Using Conservation of Energy

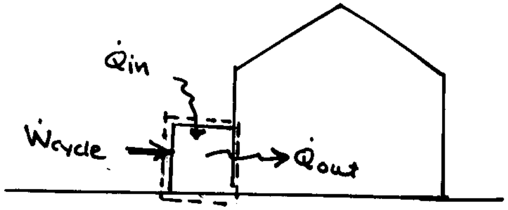
A heat pump cycle whose coefficient of performance is 2.5 delivers energy by heat transfer to a dwelling at a rate of 20 kW.

(a) Determine the net power required to operate the heat pump, in kW.

(b) Evaluating electricity at \$0.08 per determine the cost of electricity in a month when the heat pump operates for 200 hours.

KNOWN: Operating data are provided for a heat pump.

FIND: Determine the net power required to operate the heat pump and its monthly cost.



$\gamma = 2.5$

$\dot{Q}_{out} = 20\,kW$

ASSUMPTIONS: 1. The system undergoes a heat pump cycle. 2. The cycle operates steadily for 200 h monthly. 3. Electricity is valued at \$0.08/kW·h.

ANALYSIS: (a) The coefficient of performance for the heat pump is

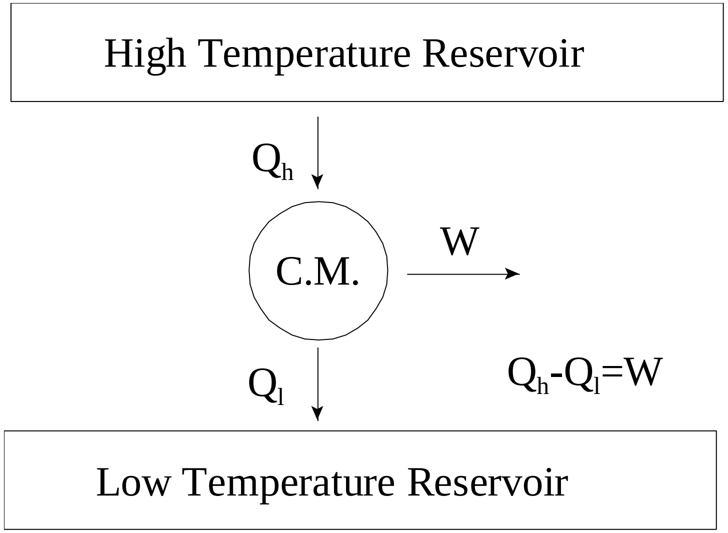
$$\gamma = \frac{\dot{Q}_{out}}{\dot{W}_{cycle}} \Rightarrow \dot{W}_{cycle} = \frac{\dot{Q}_{out}}{\gamma} = \frac{20\,kW}{2.5} = 8\,kW \longleftarrow \gamma$$

(b) With assumptions 2,3

$$\dot{\$} = (8\,kW)\left(\frac{200\,h}{month}\right)\left(\frac{\$0.08}{kW\cdot h}\right) = \$128/month \longleftarrow \dot{\$}$$

Using Conservation of Energy

For a power cycle, the heat transfers are *Q*in 25,000 kJ and *Q*out 15,000 kJ. Determine the net work, in kJ, and the thermal efficiency.

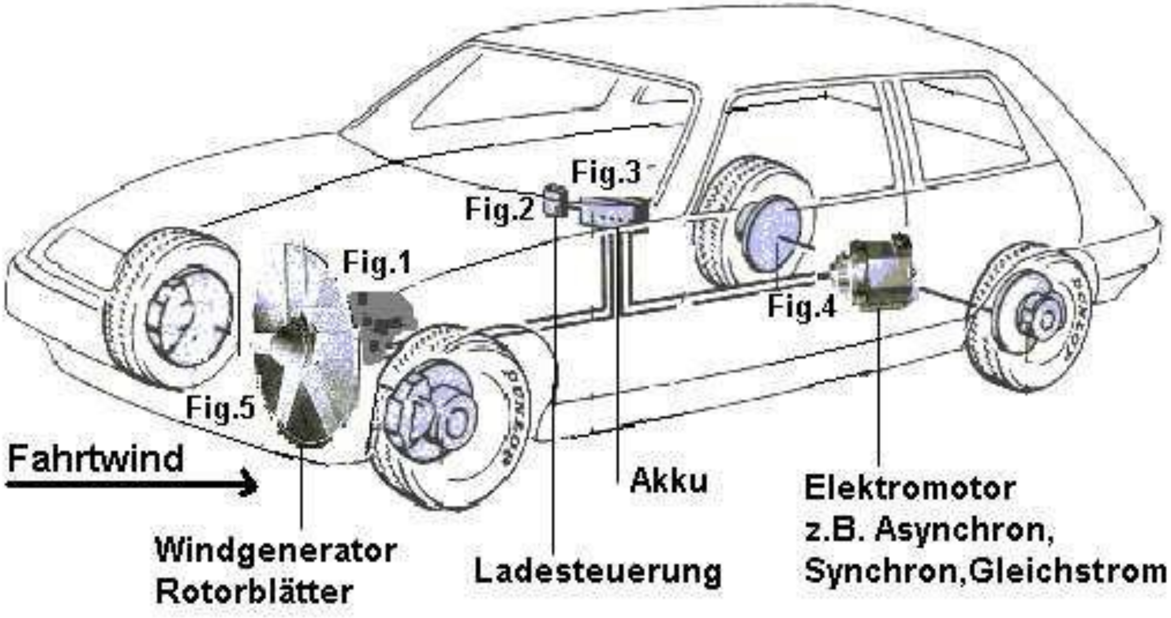


$$W_{cycle} = Q_{in} - Q_{out}$$

$$= 25,000 - 15,000 = 10,000 \text{ kJ} \leftarrow W_{cycle}$$

$$\eta = \frac{W_{cycle}}{Q_{in}} = \frac{10,000}{25,000} = 0.4 \ (40\%) \leftarrow \eta$$

Perpetual Motion



$$\dot{W} - \dot{Q} + \sum \dot{m}e_{out} - \sum \dot{m}e_{in} + \frac{d}{dt}\left[me_{cv}\right] = 0$$

Perpetual Motion is a Technical Foul

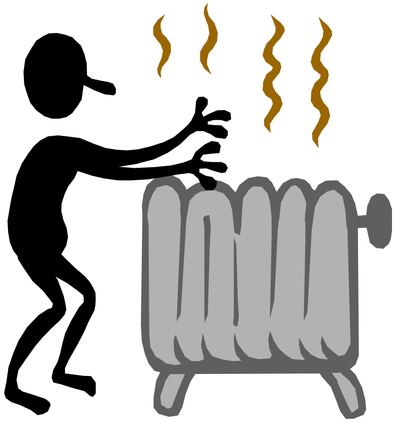
Principle of State and Phase-Terms

- Intensive and extensive properties (State Postulate)
- Pure substance
- Equilibrium
- Specific properties
- Compressed (Sub Cooled) Liquid
- Quality
- Two-phase region
- Vapor dome
- Superheated region
- Critical properties
- Ideal gas
- Gas constant
- Compressibility factor
- Reduced temperature and pressure
- Compressed liquid assumption

Can You find Properties???

Thermodynamic Properties

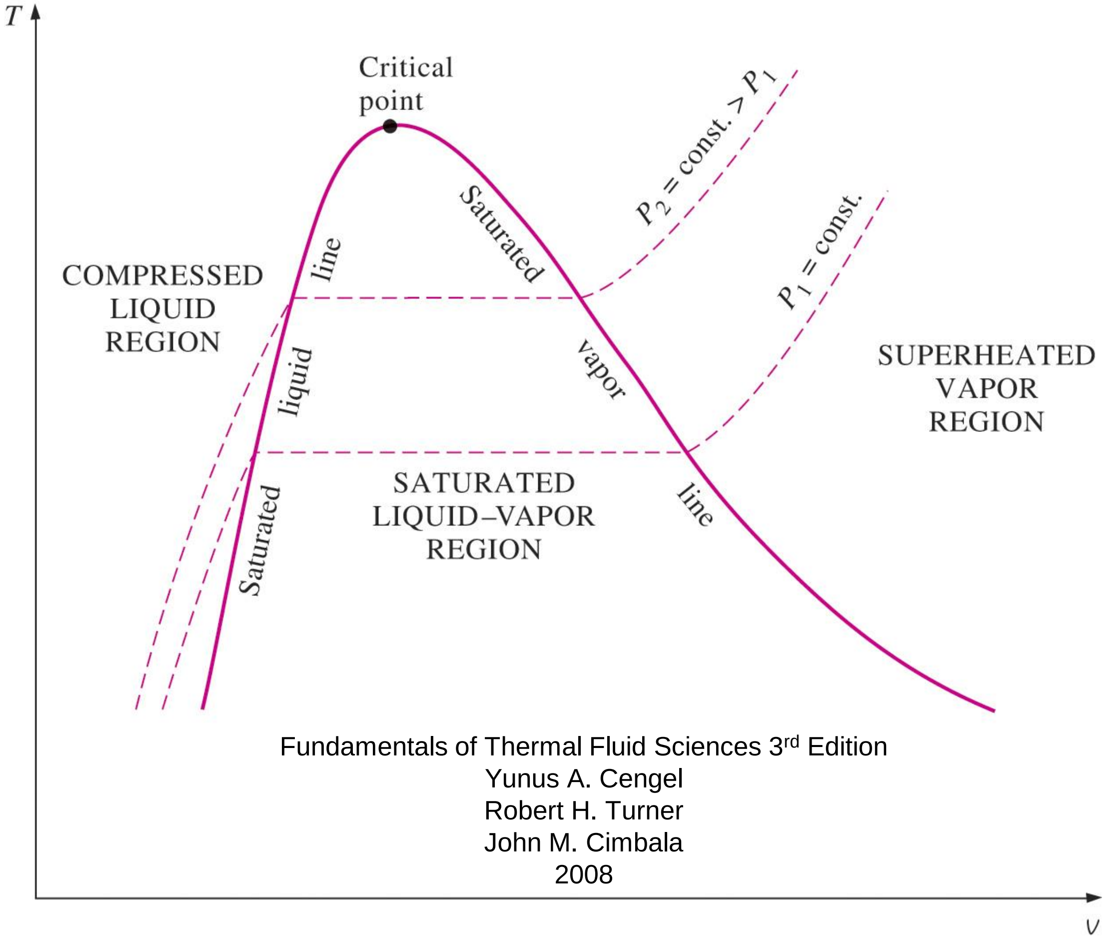
- Volume
  - Specific Volume
  - Density
- Pressure
  - Gauge
  - Absolute
  - Units
- Temperature
  - Units and Scales
- Energy
  - Internal Energy
  - Enthalpy (internal energy + boundary/flow work)
  - Kinetic and Potential
- Quality
- Entropy



The state postulate

The minimum number of independent intensive properties to find the state for a simple compressible system is two.

- Two-phase region
- Superheated vapor
- Triple line
- Saturation state
- Compressed liquid
- Vapor dome
- Critical point



Fundamentals of Thermal Fluid Sciences 3rd Edition
Yunus A. Cengel
Robert H. Turner
John M. Cimbala
2008

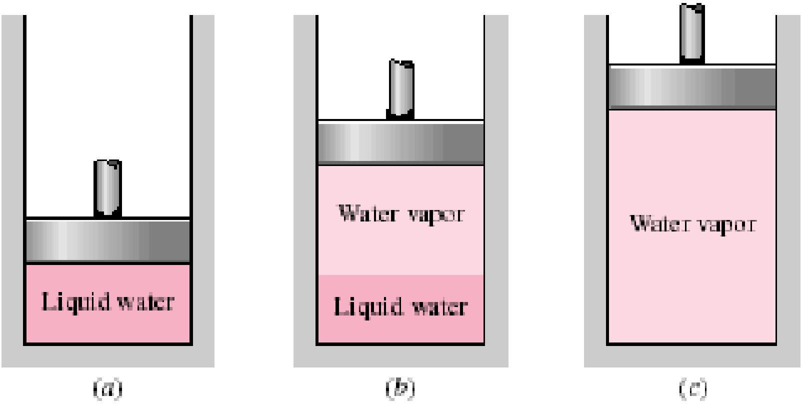
Look up Properties for Each Case

Pressure is 100kPa Review Quality Equations subscripts f,fg,g

$$x = m_g / m_t$$

$$v_{\text{avg}} = v_f + x v_{fg} \qquad (\text{m}^3/\text{kg})$$

- (a) is saturated liquid
- (a) is liquid at 20 C
- (b) is two phase quality (x) of .7
- (c) is saturated vapor
- (c) is superheated vapor at 120 C



Introduction to Thermal
Systems Engineering:
Thermodynamics, Fluid Mechanics,
and Heat Transfer
Michael J. Moran
Howard N. Shapiro
Bruce R. Munson
David P. DeWitt
John Wiley & Sons, Inc.
2003

Specific volume, specific internal energy, specific enthalpy, and specific entropy as properties

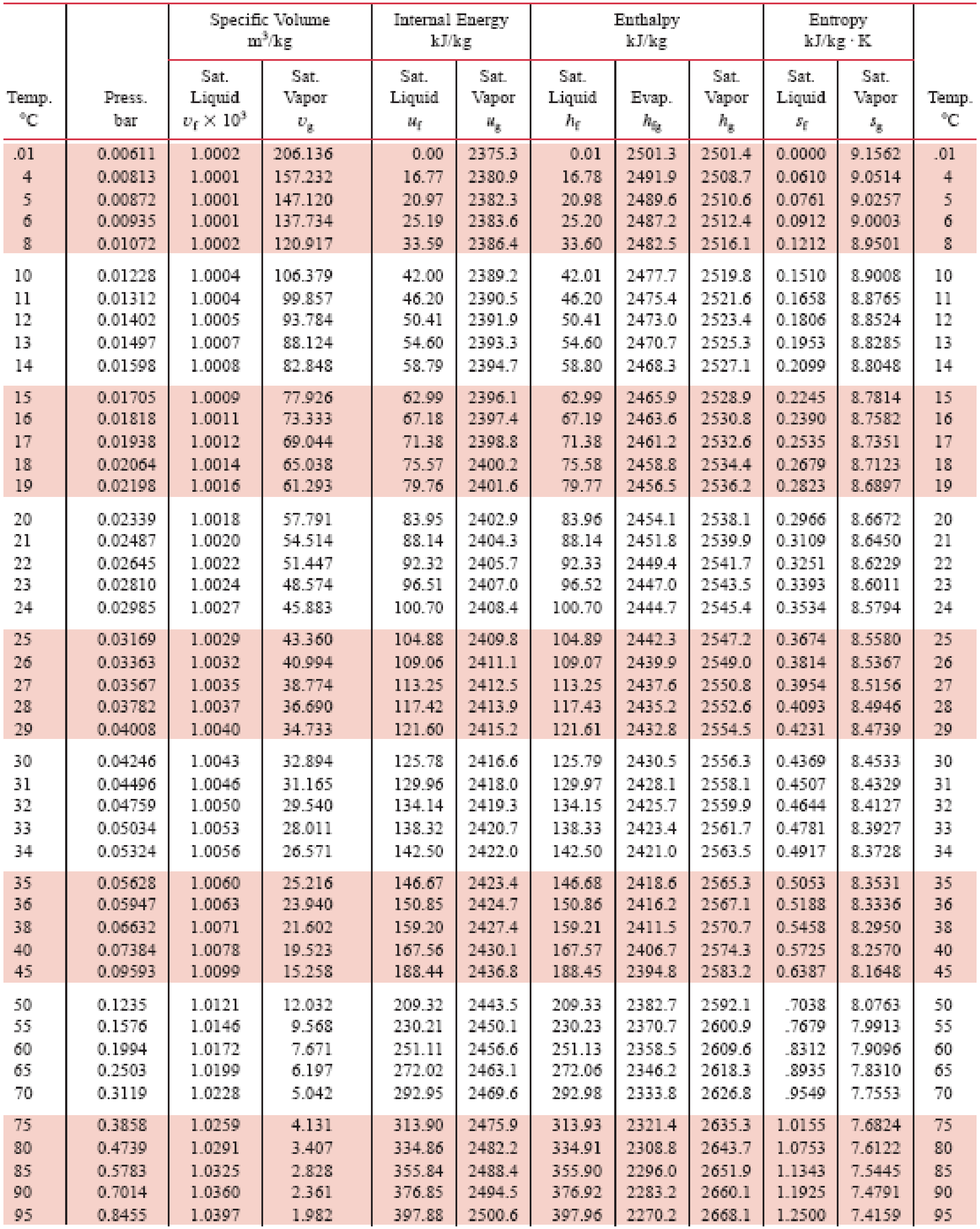
Saturated Water Pressure Table

Table T-3 *(Continued)*

Press. bar	Temp. °C	Specific Volume m³/kg Sat. Liquid $v_f \times 10^3$	Specific Volume m³/kg Sat. Vapor v_g	Internal Energy kJ/kg Sat. Liquid u_f	Internal Energy kJ/kg Sat. Vapor u_g	Enthalpy kJ/kg Sat. Liquid h_f	Enthalpy kJ/kg Evap. h_{fg}	Enthalpy kJ/kg Sat. Vapor h_g	Entropy kJ/kg · K Sat. Liquid s_f	Entropy kJ/kg · K Sat. Vapor s_g	Press. bar
0.80	93.50	1.0380	2.087	391.58	2498.8	391.66	2274.1	2665.8	1.2329	7.4346	0.80
0.90	96.71	1.0410	1.869	405.06	2502.6	405.15	2265.7	2670.9	1.2695	7.3949	0.90
1.00	99.63	1.0432	1.694	417.36	2506.1	417.46	2258.0	2675.5	1.3026	7.3594	1.00
1.50	111.4	1.0528	1.159	466.94	2519.7	467.11	2226.5	2693.6	1.4336	7.2233	1.50
2.00	120.2	1.0605	0.8857	504.49	2529.5	504.70	2201.9	2706.7	1.5301	7.1271	2.00
2.50	127.4	1.0672	0.7187	535.10	2537.2	535.37	2181.5	2716.9	1.6072	7.0527	2.50
3.00	133.6	1.0732	0.6058	561.15	2543.6	561.47	2163.8	2725.3	1.6718	6.9919	3.00
3.50	138.9	1.0786	0.5243	583.95	2546.9	584.33	2148.1	2732.4	1.7275	6.9405	3.50
4.00	143.6	1.0836	0.4625	604.31	2553.6	604.74	2133.8	2738.6	1.7766	6.8959	4.00
4.50	147.9	1.0882	0.4140	622.25	2557.6	623.25	2120.7	2743.9	1.8207	6.8565	4.50
5.00	151.9	1.0926	0.3749	639.68	2561.2	640.23	2108.5	2748.7	1.8607	6.8212	5.00
6.00	158.9	1.1006	0.3157	669.90	2567.4	670.56	2086.3	2756.8	1.9312	6.7600	6.00
7.00	165.0	1.1080	0.2729	696.44	2572.5	697.22	2066.3	2763.5	1.9922	6.7080	7.00
8.00	170.4	1.1148	0.2404	720.22	2576.8	721.11	2048.0	2769.1	2.0462	6.6628	8.00
9.00	175.4	1.1212	0.2150	741.83	2580.5	742.83	2031.1	2773.9	2.0946	6.6226	9.00
10.0	179.9	1.1273	0.1944	761.68	2583.6	762.81	2015.3	2778.1	2.1387	6.5863	10.0
15.0	198.3	1.1539	0.1318	843.16	2594.5	844.84	1947.3	2792.2	2.3150	6.4448	15.0
20.0	212.4	1.1767	0.09963	906.44	2600.3	908.79	1890.7	2799.5	2.4474	6.3409	20.0
25.0	224.0	1.1973	0.07998	959.11	2603.1	962.11	1841.0	2803.1	2.5547	6.2575	25.0
30.0	233.9	1.2165	0.06668	1004.8	2604.1	1008.4	1795.7	2804.2	2.6457	6.1869	30.0
35.0	242.6	1.2347	0.05707	1045.4	2603.7	1049.8	1753.7	2803.4	2.7253	6.1253	35.0
40.0	250.4	1.2522	0.04978	1082.3	2602.3	1087.3	1714.1	2801.4	2.7964	6.0701	40.0
45.0	257.5	1.2692	0.04406	1116.2	2600.1	1121.9	1676.4	2798.3	2.8610	6.0199	45.0
50.0	264.0	1.2859	0.03944	1147.8	2597.1	1154.2	1640.1	2794.3	2.9202	5.9734	50.0
60.0	275.6	1.3187	0.03244	1205.4	2589.7	1213.4	1571.0	2784.3	3.0267	5.8892	60.0
70.0	285.9	1.3513	0.02737	1257.6	2580.5	1267.0	1505.1	2772.1	3.1211	5.8133	70.0
80.0	295.1	1.3842	0.02352	1305.6	2569.8	1316.6	1441.3	2758.0	3.2068	5.7432	80.0
90.0	303.4	1.4178	0.02048	1350.5	2557.8	1363.3	1378.9	2742.1	3.2858	5.6772	90.0
100.	311.1	1.4524	0.01803	1393.0	2544.4	1407.6	1317.1	2724.7	3.3596	5.6141	100.
110.	318.2	1.4886	0.01599	1433.7	2529.8	1450.1	1255.5	2705.6	3.4295	5.5527	110.
120.	324.8	1.5267	0.01426	1473.0	2513.7	1491.3	1193.6	2684.9	3.4962	5.4924	120.
130.	330.9	1.5671	0.01278	1511.1	2496.1	1531.5	1130.7	2662.2	3.5606	5.4323	130.
140.	336.8	1.6107	0.01149	1548.6	2476.8	1571.1	1066.5	2637.6	3.6232	5.3717	140.
150.	342.2	1.6581	0.01034	1585.6	2455.5	1610.5	1000.0	2610.5	3.6848	5.3098	150.
160.	347.4	1.7107	0.009306	1622.7	2431.7	1650.1	930.6	2580.6	3.7461	5.2455	160.
170.	352.4	1.7702	0.008364	1660.2	2405.0	1690.3	856.9	2547.2	3.8079	5.1777	170.
180.	357.1	1.8397	0.007489	1698.9	2374.3	1732.0	777.1	2509.1	3.8715	5.1044	180.
190.	361.5	1.9243	0.006657	1739.9	2338.1	1776.5	688.0	2464.5	3.9388	5.0228	190.
200.	365.8	2.036	0.005834	1785.6	2293.0	1826.3	583.4	2409.7	4.0139	4.9269	200.

Introduction to Thermal
Systems Engineering:
Thermodynamics, Fluid Mechanics,
and Heat Transfer
Michael J. Moran
Howard N. Shapiro
Bruce R. Munson
David P. DeWitt
John Wiley & Sons, Inc.
2003

Saturated Water Temperature Table



Temp. °C	Press. bar	Specific Volume m³/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy kJ/kg · K		Temp. °C
		Sat. Liquid $v_f \times 10^3$	Sat. Vapor v_g	Sat. Liquid u_f	Sat. Vapor u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Sat. Vapor s_g	
.01	0.00611	1.0002	206.136	0.00	2375.3	0.01	2501.3	2501.4	0.0000	9.1562	.01
4	0.00813	1.0001	157.232	16.77	2380.9	16.78	2491.9	2508.7	0.0610	9.0514	4
5	0.00872	1.0001	147.120	20.97	2382.3	20.98	2489.6	2510.6	0.0761	9.0257	5
6	0.00935	1.0001	137.734	25.19	2383.6	25.20	2487.2	2512.4	0.0912	9.0003	6
8	0.01072	1.0002	120.917	33.59	2386.4	33.60	2482.5	2516.1	0.1212	8.9501	8
10	0.01228	1.0004	106.379	42.00	2389.2	42.01	2477.7	2519.8	0.1510	8.9008	10
11	0.01312	1.0004	99.857	46.20	2390.5	46.20	2475.4	2521.6	0.1658	8.8765	11
12	0.01402	1.0005	93.784	50.41	2391.9	50.41	2473.0	2523.4	0.1806	8.8524	12
13	0.01497	1.0007	88.124	54.60	2393.3	54.60	2470.7	2525.3	0.1953	8.8285	13
14	0.01598	1.0008	82.848	58.79	2394.7	58.80	2468.3	2527.1	0.2099	8.8048	14
15	0.01705	1.0009	77.926	62.99	2396.1	62.99	2465.9	2528.9	0.2245	8.7814	15
16	0.01818	1.0011	73.333	67.18	2397.4	67.19	2463.6	2530.8	0.2390	8.7582	16
17	0.01938	1.0012	69.044	71.38	2398.8	71.38	2461.2	2532.6	0.2535	8.7351	17
18	0.02064	1.0014	65.038	75.57	2400.2	75.58	2458.8	2534.4	0.2679	8.7123	18
19	0.02198	1.0016	61.293	79.76	2401.6	79.77	2456.5	2536.2	0.2823	8.6897	19
20	0.02339	1.0018	57.791	83.95	2402.9	83.96	2454.1	2538.1	0.2966	8.6672	20
21	0.02487	1.0020	54.514	88.14	2404.3	88.14	2451.8	2539.9	0.3109	8.6450	21
22	0.02645	1.0022	51.447	92.32	2405.7	92.33	2449.4	2541.7	0.3251	8.6229	22
23	0.02810	1.0024	48.574	96.51	2407.0	96.52	2447.0	2543.5	0.3393	8.6011	23
24	0.02985	1.0027	45.883	100.70	2408.4	100.70	2444.7	2545.4	0.3534	8.5794	24
25	0.03169	1.0029	43.360	104.88	2409.8	104.89	2442.3	2547.2	0.3674	8.5580	25
26	0.03363	1.0032	40.994	109.06	2411.1	109.07	2439.9	2549.0	0.3814	8.5367	26
27	0.03567	1.0035	38.774	113.25	2412.5	113.25	2437.6	2550.8	0.3954	8.5156	27
28	0.03782	1.0037	36.690	117.42	2413.9	117.43	2435.2	2552.6	0.4093	8.4946	28
29	0.04008	1.0040	34.733	121.60	2415.2	121.61	2432.8	2554.5	0.4231	8.4739	29
30	0.04246	1.0043	32.894	125.78	2416.6	125.79	2430.5	2556.3	0.4369	8.4533	30
31	0.04496	1.0046	31.165	129.96	2418.0	129.97	2428.1	2558.1	0.4507	8.4329	31
32	0.04759	1.0050	29.540	134.14	2419.3	134.15	2425.7	2559.9	0.4644	8.4127	32
33	0.05034	1.0053	28.011	138.32	2420.7	138.33	2423.4	2561.7	0.4781	8.3927	33
34	0.05324	1.0056	26.571	142.50	2422.0	142.50	2421.0	2563.5	0.4917	8.3728	34
35	0.05628	1.0060	25.216	146.67	2423.4	146.68	2418.6	2565.3	0.5053	8.3531	35
36	0.05947	1.0063	23.940	150.85	2424.7	150.86	2416.2	2567.1	0.5188	8.3336	36
38	0.06632	1.0071	21.602	159.20	2427.4	159.21	2411.5	2570.7	0.5458	8.2950	38
40	0.07384	1.0078	19.523	167.56	2430.1	167.57	2406.7	2574.3	0.5725	8.2570	40
45	0.09593	1.0099	15.258	188.44	2436.8	188.45	2394.8	2583.2	0.6387	8.1648	45
50	0.1235	1.0121	12.032	209.32	2443.5	209.33	2382.7	2592.1	.7038	8.0763	50
55	0.1576	1.0146	9.568	230.21	2450.1	230.23	2370.7	2600.9	.7679	7.9913	55
60	0.1994	1.0172	7.671	251.11	2456.6	251.13	2358.5	2609.6	.8312	7.9096	60
65	0.2503	1.0199	6.197	272.02	2463.1	272.06	2346.2	2618.3	.8935	7.8310	65
70	0.3119	1.0228	5.042	292.95	2469.6	292.98	2333.8	2626.8	.9549	7.7553	70
75	0.3858	1.0259	4.131	313.90	2475.9	313.93	2321.4	2635.3	1.0155	7.6824	75
80	0.4739	1.0291	3.407	334.86	2482.2	334.91	2308.8	2643.7	1.0753	7.6122	80
85	0.5783	1.0325	2.828	355.84	2488.4	355.90	2296.0	2651.9	1.1343	7.5445	85
90	0.7014	1.0360	2.361	376.85	2494.5	376.92	2283.2	2660.1	1.1925	7.4791	90
95	0.8455	1.0397	1.982	397.88	2500.6	397.96	2270.2	2668.1	1.2500	7.4159	95

Introduction to Thermal
Systems Engineering:
Thermodynamics, Fluid Mechanics,
and Heat Transfer
Michael J. Moran
Howard N. Shapiro
Bruce R. Munson
David P. DeWitt
John Wiley & Sons, Inc.
2003

Table T-4 Properties of Superheated Water Vapor

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg · K	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg · K	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg · K
	p = 0.06 bar = 0.006 MPa (T_{sat} = 36.16°C)				p = 0.35 bar = 0.035 MPa (T_{sat} = 72.69°C)				p = 0.70 bar = 0.07 MPa (T_{sat} = 89.95°C)			
Sat.	23.739	2425.0	2567.4	8.3304	4.526	2473.0	2631.4	7.7158	2.365	2494.5	2660.0	7.4797
80	27.132	2487.3	2650.1	8.5804	4.625	2483.7	2645.6	7.7564	2.434	2509.7	2680.0	7.5341
120	30.219	2544.7	2726.0	8.7840	5.163	2542.4	2723.1	7.9644	2.571	2539.7	2719.6	7.6375
160	33.302	2602.7	2802.5	8.9693	5.696	2601.2	2800.6	8.1519	2.841	2599.4	2798.2	7.8279
200	36.383	2661.4	2879.7	9.1398	6.228	2660.4	2878.4	8.3237	3.108	2659.1	2876.7	8.0012
240	39.462	2721.0	2957.8	9.2982	6.758	2720.3	2956.8	8.4828	3.374	2719.3	2955.5	8.1611
280	42.540	2781.5	3036.8	9.4464	7.287	2780.9	3036.0	8.6314	3.640	2780.2	3035.0	8.3162
320	45.618	2843.0	3116.7	9.5859	7.815	2842.5	3116.1	8.7712	3.905	2842.0	3115.3	8.4504
360	48.696	2905.5	3197.7	9.7180	8.344	2905.1	3197.1	8.9034	4.170	2904.6	3196.5	8.5828
400	51.774	2969.0	3279.6	9.8435	8.872	2968.6	3279.2	9.0291	4.434	2968.2	3278.6	8.7086
440	54.851	3033.5	3362.6	9.9633	9.400	3033.2	3362.2	9.1490	4.698	3032.9	3361.8	8.8286
500	59.467	3132.3	3489.1	10.1336	10.192	3132.1	3488.8	9.3194	5.095	3131.8	3488.5	8.9991
	p = 1.0 bar = 0.10 MPa (T_{sat} = 99.63°C)				p = 1.5 bar = 0.15 MPa (T_{sat} = 111.37°C)				p = 3.0 bar = 0.30 MPa (T_{sat} = 133.55°C)			
Sat.	1.694	2506.1	2675.5	7.3594	1.159	2519.7	2693.6	7.2233	0.606	2543.6	2725.3	6.9919
100	1.696	2506.7	2676.2	7.3614								
120	1.793	2537.3	2716.6	7.4668	1.188	2533.3	2711.4	7.2693				
160	1.984	2597.8	2796.2	7.6597	1.317	2595.2	2792.8	7.4665	0.651	2587.1	2782.3	7.1276
200	2.172	2658.1	2875.3	7.8343	1.444	2656.2	2872.9	7.6433	0.716	2650.7	2865.5	7.3115
240	2.359	2718.5	2954.5	7.9949	1.570	2717.2	2952.7	7.8052	0.781	2713.1	2947.3	7.4774
280	2.546	2779.6	3034.2	8.1445	1.695	2778.6	3032.8	7.9555	0.844	2775.4	3028.6	7.6299
320	2.732	2841.5	3114.6	8.2849	1.819	2840.6	3113.5	8.0964	0.907	2838.1	3110.1	7.7722
360	2.917	2904.2	3195.9	8.4175	1.943	2903.5	3195.0	8.2293	0.969	2901.4	3192.2	7.9061
400	3.103	2967.9	3278.2	8.5435	2.067	2967.3	3277.4	8.3555	1.032	2965.6	3275.0	8.0330
440	3.288	3032.6	3361.4	8.6636	2.191	3032.1	3360.7	8.4757	1.094	3030.6	3358.7	8.1538
500	3.565	3131.6	3488.1	8.8342	2.376	3131.2	3487.6	8.6466	1.187	3130.0	3486.0	8.3251
	p = 5.0 bar = 0.50 MPa (T_{sat} = 151.86°C)				p = 7.0 bar = 0.70 MPa (T_{sat} = 164.97°C)				p = 10.0 bar = 1.0 MPa (T_{sat} = 179.91°C)			
Sat.	0.3749	2561.2	2748.7	6.8213	0.2729	2572.5	2763.5	6.7080	0.1944	2583.6	2778.1	6.5865
180	0.4045	2609.7	2812.0	6.9656	0.2847	2599.8	2799.1	6.7880				
200	0.4249	2642.9	2855.4	7.0592	0.2999	2634.8	2844.8	6.8865	0.2060	2621.9	2827.9	6.6940
240	0.4646	2707.6	2939.9	7.2307	0.3292	2701.8	2932.2	7.0641	0.2275	2692.9	2920.4	6.8817
280	0.5034	2771.2	3022.9	7.3865	0.3574	2766.9	3017.1	7.2233	0.2480	2760.2	3008.2	7.0465
320	0.5416	2834.7	3105.6	7.5308	0.3852	2831.3	3100.9	7.3697	0.2678	2826.1	3093.9	7.1962
360	0.5796	2898.7	3188.4	7.6660	0.4126	2895.8	3184.7	7.5063	0.2873	2891.6	3178.9	7.3349
400	0.6173	2963.2	3271.9	7.7938	0.4397	2960.9	3268.7	7.6350	0.3066	2957.3	3263.9	7.4651
440	0.6548	3028.6	3356.0	7.9152	0.4667	3026.6	3353.3	7.7571	0.3257	3023.6	3349.3	7.5883
500	0.7109	3128.4	3483.9	8.0873	0.5070	3126.8	3481.7	7.9299	0.3541	3124.4	3478.5	7.7622
600	0.8041	3299.6	3701.7	8.3522	0.5738	3298.5	3700.2	8.1956	0.4011	3296.8	3697.9	8.0290

Introduction to Thermal
Systems Engineering:
Thermodynamics, Fluid Mechanics,
and Heat Transfer
Michael J. Moran
Howard N. Shapiro
Bruce R. Munson
David P. DeWitt
John Wiley & Sons, Inc.
2003

THE IDEAL-GAS EQUATION OF STATE

- **Equation of state**: Any equation that relates the pressure, temperature, and specific volume of a substance.
- The simplest and best-known equation of state for substances in the gas phase is the ideal-gas equation of state. This equation predicts the *P*-*v*-*T* behavior of a gas quite accurately within some properly selected region.

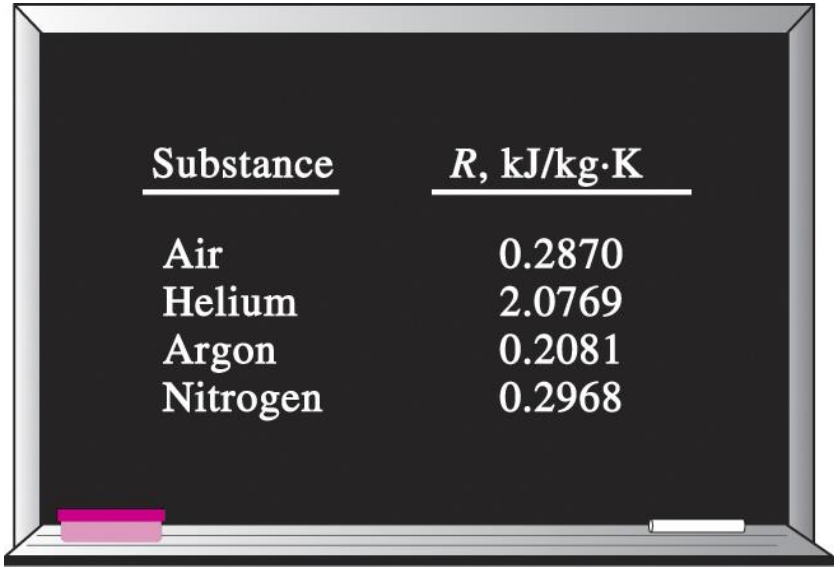
$$P = R\left(\frac{T}{v}\right) \qquad Pv = RT$$

Ideal gas equation of state

$$R = \frac{R_u}{M} \qquad (\mathrm{kJ/kg \cdot K \text{ or } kPa \cdot m^3/kg \cdot K})$$

R: gas constant
M: molar mass (kg/kmol)
R_u: universal gas constant

$$R_u = \begin{cases} 8.31447 \text{ kJ/kmol} \cdot \text{K} \\ 8.31447 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K} \\ 0.0831447 \text{ bar} \cdot \text{m}^3/\text{kmol} \cdot \text{K} \\ 1.98588 \text{ Btu/lbmol} \cdot \text{R} \\ 10.7316 \text{ psia} \cdot \text{ft}^3/\text{lbmol} \cdot \text{R} \\ 1545.37 \text{ ft} \cdot \text{lbf/lbmol} \cdot \text{R} \end{cases}$$



Substance	R, kJ/kg·K
Air	0.2870
Helium	2.0769
Argon	0.2081
Nitrogen	0.2968

Different substances have different gas constants.

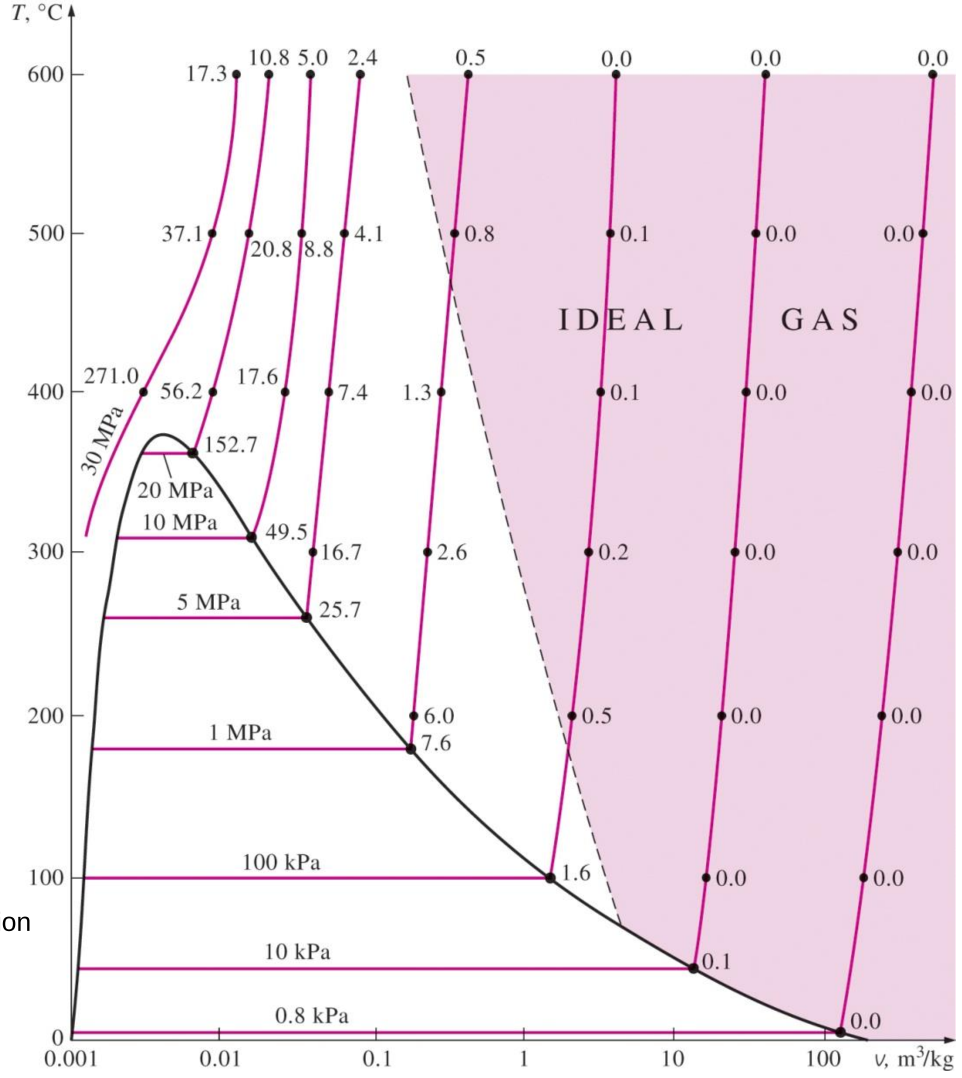
$pv - RT$ requires the use of *absolute* temperature T and *absolute* pressure p.

Ideal Gas

$pV = mRT$

$pV = nR_uT$

Ideal gas model only valid away from two phase region and critical point



Fundamentals of Thermal Fluid Sciences 3rd Edition
Yunus A. Cengel
Robert H. Turner
John M. Cimbala
2008

An automobile tire is inflated with air. The pressure rise of air in the tire when the tire is heated to 50 C and the amount of air that must be bled off to reduce the pressure to the original value are to be determined.

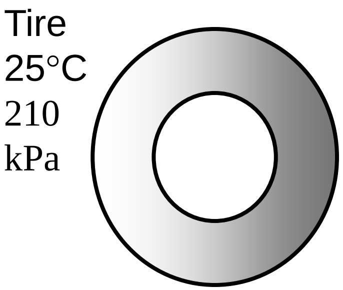
Assumptions **1** At specified conditions, air behaves as an ideal gas. **2** The volume of the tire remains constant.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K

Analysis Initially, the absolute pressure in the tire is

$$P_1 = P_g + P_{\text{atm}} = 210 + 100 = 310\text{kPa}$$

Initial Condition



Treating air as an ideal gas and assuming the volume of the tire to remain constant, the final pressure in the tire can be determined from

$$\frac{P_1\boldsymbol{V}_1}{T_1} = \frac{P_2\boldsymbol{V}_2}{T_2} \longrightarrow P_2 = \frac{T_2}{T_1}P_1 = \frac{323\text{ K}}{298\text{ K}}(310\text{ kPa}) = 336\text{ kPa}$$

Thus the pressure rise is

$$\Delta P = P_2 - P_1 = 336 - 310 = \mathbf{26\ kPa}$$

The amount of air that needs to be bled off to restore pressure to its original value is

$$m_1 = \frac{P_1\boldsymbol{V}}{RT_1} = \frac{(310\text{ kPa})(0.025\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 0.0906\text{ kg}$$

$$m_2 = \frac{P_1\boldsymbol{V}}{RT_2} = \frac{(310\text{ kPa})(0.025\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323\text{ K})} = 0.0836\text{ kg}$$

$$\Delta m = m_1 - m_2 = 0.0906 - 0.0836 = \mathbf{0.0070 kg}$$

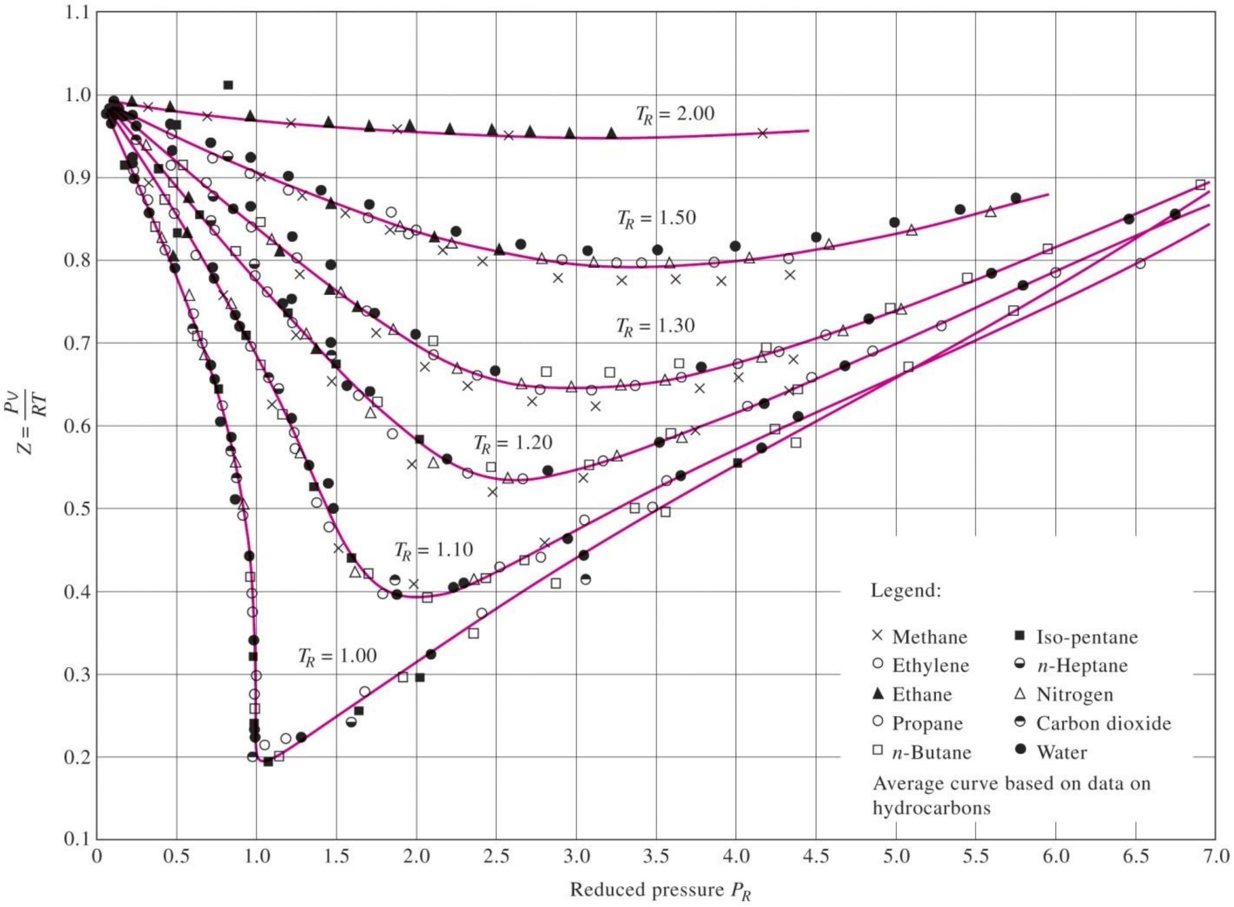
Compressibility Factor

Reduced pressure $P_R = \frac{P}{P_{cr}}$

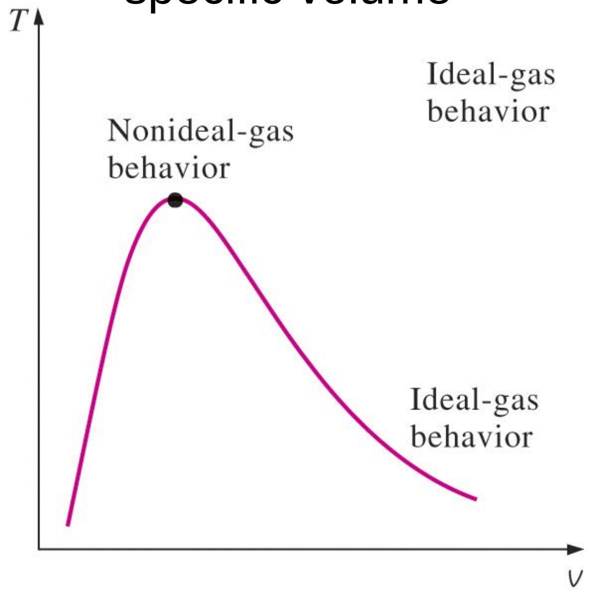
Reduced temperature $T_R = \frac{T}{T_{cr}}$

$v_R = \frac{v_{\text{actual}}}{RT_{cr}/P_{cr}}$

Pseudo-reduced specific volume



Comparison of *Z* factors for various gases.



Gases deviate from the ideal-gas behavior the most in the neighborhood of the critical point.

Charles’ and Boyle’s Law

Uses ideal gas law and conservation of mass in steady flow to determine volume at another pressure and temperature.

$$V_1 = \frac{V_2 \times P_2 \times T_1}{T_2 \times P_1}$$

Derive

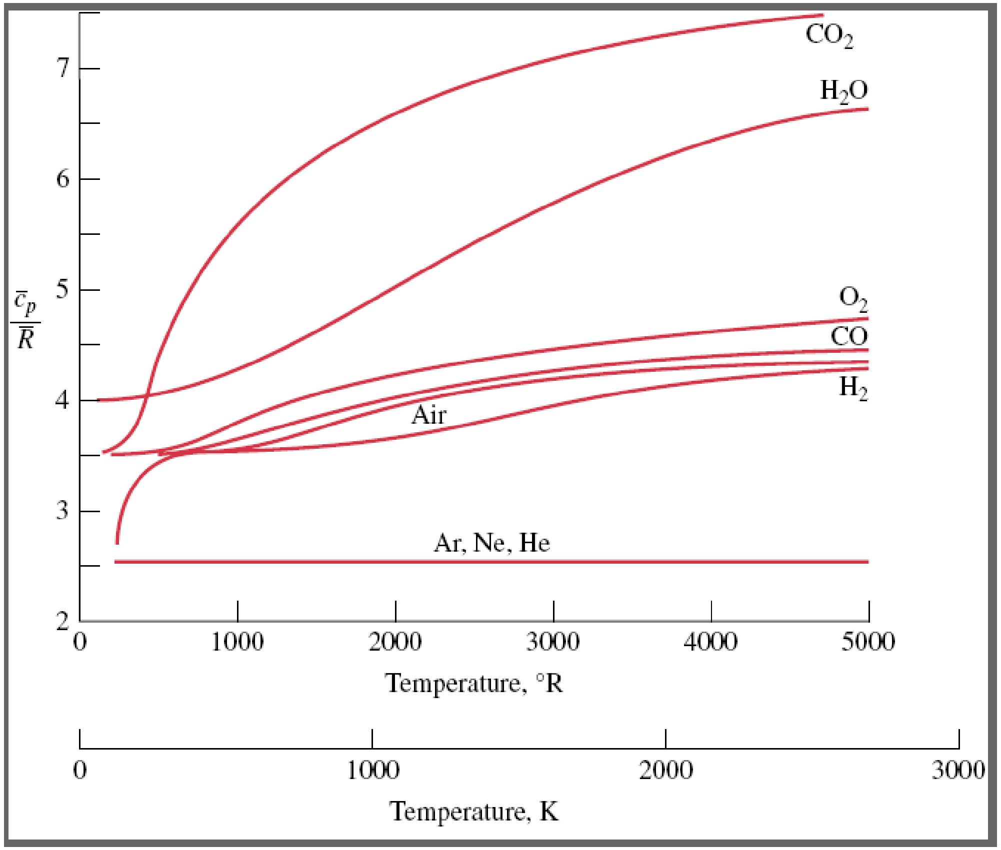
Exhaust emissions at collected temperature and pressure to standard temperature and pressure

Ambient Standard 1 bar 25°C

Standard 1 atm 0°C

Specific Heat

- Constant volume (Internal Energy)
- Constant pressure (Enthalpy)
- Ratio of specific heats
- Specific heat
  - – Solids
  - – Liquids



Introduction to Thermal
Systems Engineering:
Thermodynamics, Fluid Mechanics,
and Heat Transfer
Michael J. Moran
Howard N. Shapiro
Bruce R. Munson
David P. DeWitt
John Wiley & Sons, Inc.
2003

Compressed Liquid Assumption

The compressed liquid properties depend on temperature much more strongly than they do on pressure.

$y \cong y_{f\,@\,T}$ **y** → v, u, or h

A more accurate relation for h

$$h \cong h_{f\,@\,T} + v_{f\,@\,T}(P - P_{\text{sat}\,@\,T})$$

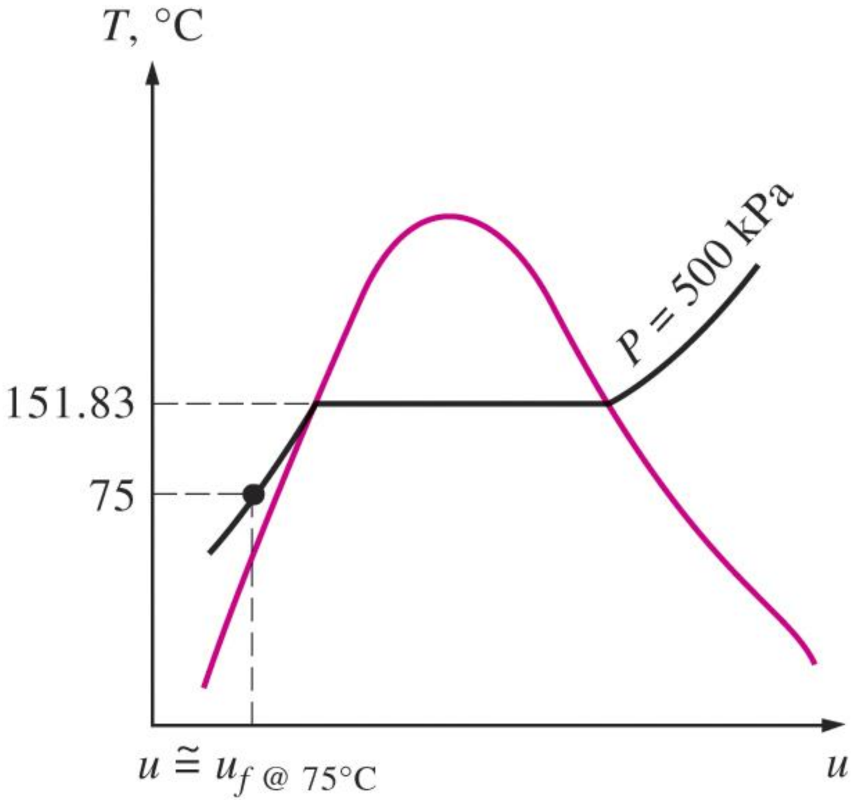
Given: P and T

$$v \cong v_{f\,@T}$$
$$u \cong u_{f\,@T}$$
$$h \cong h_{f\,@T}$$

A compressed liquid may be approximated as a saturated liquid at the given temperature.

At a given P and T, a pure substance will exist as a compressed liquid if

$T < T_{\text{sat}\,@\,P}$

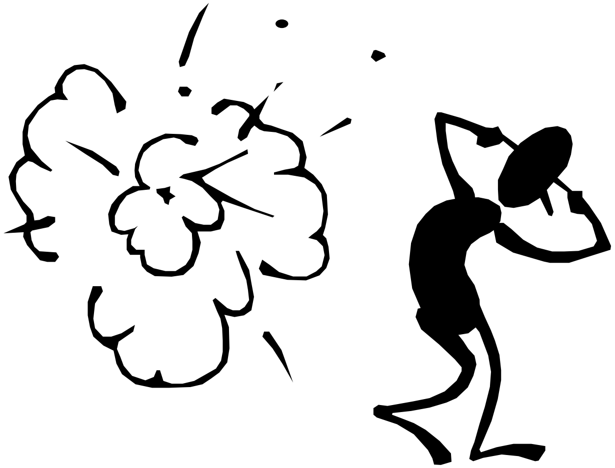


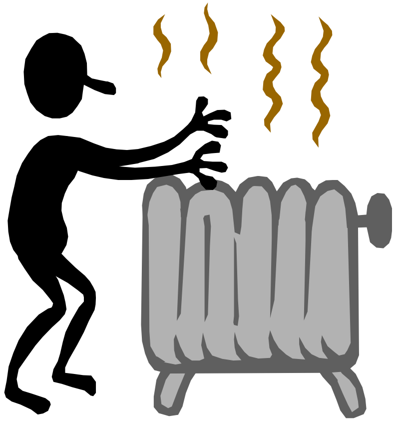
Principle of Entropy

- Total Entropy is ALWAYS increasing
- 2nd Law of Thermodynamics
- Irreversibilities
  - Friction
  - Unrestrained expansion
  - Heat Transfer
  - Mixing
  - Resistance etc
- Carnot Efficiency and Coefficient of Performance
- Isentropic Efficiency

What creates Entropy

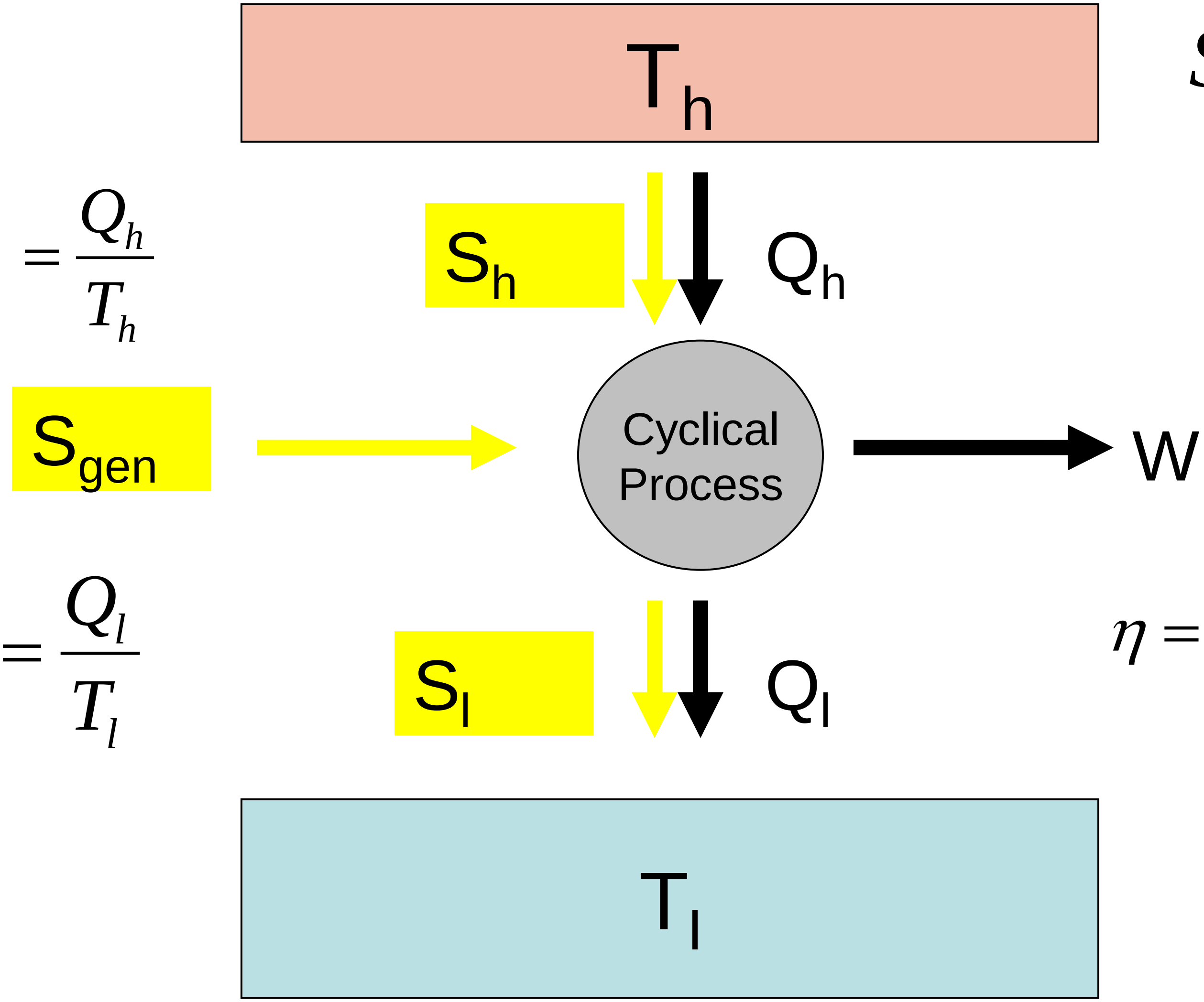
- Major Irreversibilities
  - Friction
  - Unrestrained Expansion
  - Heat Transfer
- Minor
  - Mixing, spontaneous reaction
  - Resistance





Total entropy always increases

Carnot Heat Engine



$$S_h = \frac{Q_h}{T_h}$$

$$S_l = \frac{Q_l}{T_l}$$

$$S_h + \cancelto{0}{S_{gen}} = S_l$$

$$\frac{Q_l}{T_l} = \frac{Q_h}{T_h}$$

$$\eta = 1 - \frac{Q_l}{Q_h} = 1 - \frac{T_l}{T_h}$$

$$\eta_{\max} = 1 - \frac{T_l}{T_h}$$

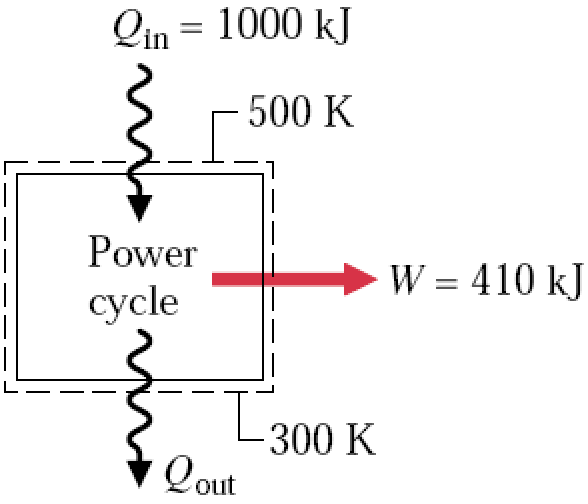
Any thermal efficiency above Carnot thermal efficiency is a violation of the Second Law

- Perpetual Motion Machine of the 2nd Kind PMM-2
- This concept applies to heat engines, refrigerators, heat pumps and any other system
- The best you can do is to have no change in entropy on the WHOLE system. (individual processes may see reductions in entropy)

Perpetual Motion is a Technical Foul

Examples

An “inventor” claims to have built a device which receives 1000 kJ at 500K and converts it to 410 kJ exhausting heat to the environment at 300K. Is this Possible?



- Max Efficiency = 1-Tl/Th =1-(300K/500K)=40%

Isentropic Efficiency

- Processes are not Isentropic
- Isentropic cases are a best case scenario
- Efficiency can be related to the Isentropic case
  - Turbines
  - Pumps and Compressors
  - Nozzles

Isentropic Efficiency

$$\eta_T = \frac{Actual\ Turbine\ Work}{Isentropic\ Turbine\ Work} = \frac{W_a}{W_s} = \frac{h_1 - h_{2a}}{h_1 - h_{2s}}$$

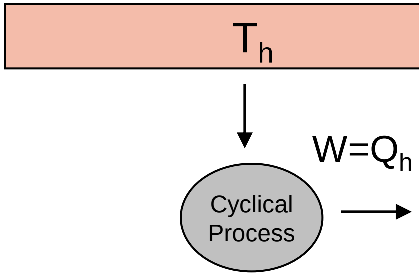
$$\eta_C = \frac{Isentropic\ Compressor\ Work}{Actual\ Compressor\ Work} = \frac{W_s}{W_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

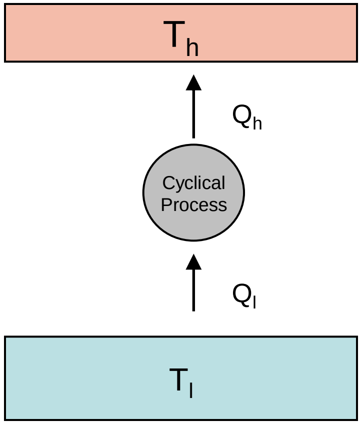
$$\eta_N = \frac{Actual\ KE}{Isentropic\ KE} = \frac{V_a^{\,2}}{V_s^{\,2}}$$

2nd Law of Thermodynamics

Kelvin-Planck: It is impossible to construct a device which will operate **in a cycle** and produce no effect other than (work) and the exchange of heat with a single reservoir.

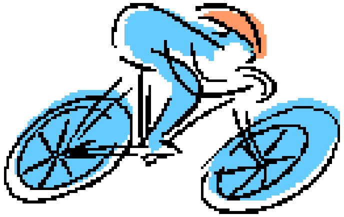
Clausius: It is impossible to construct a device which operates **in a cycle** and produces no effect other than the transfer of heat from a cooler body to a hotter body





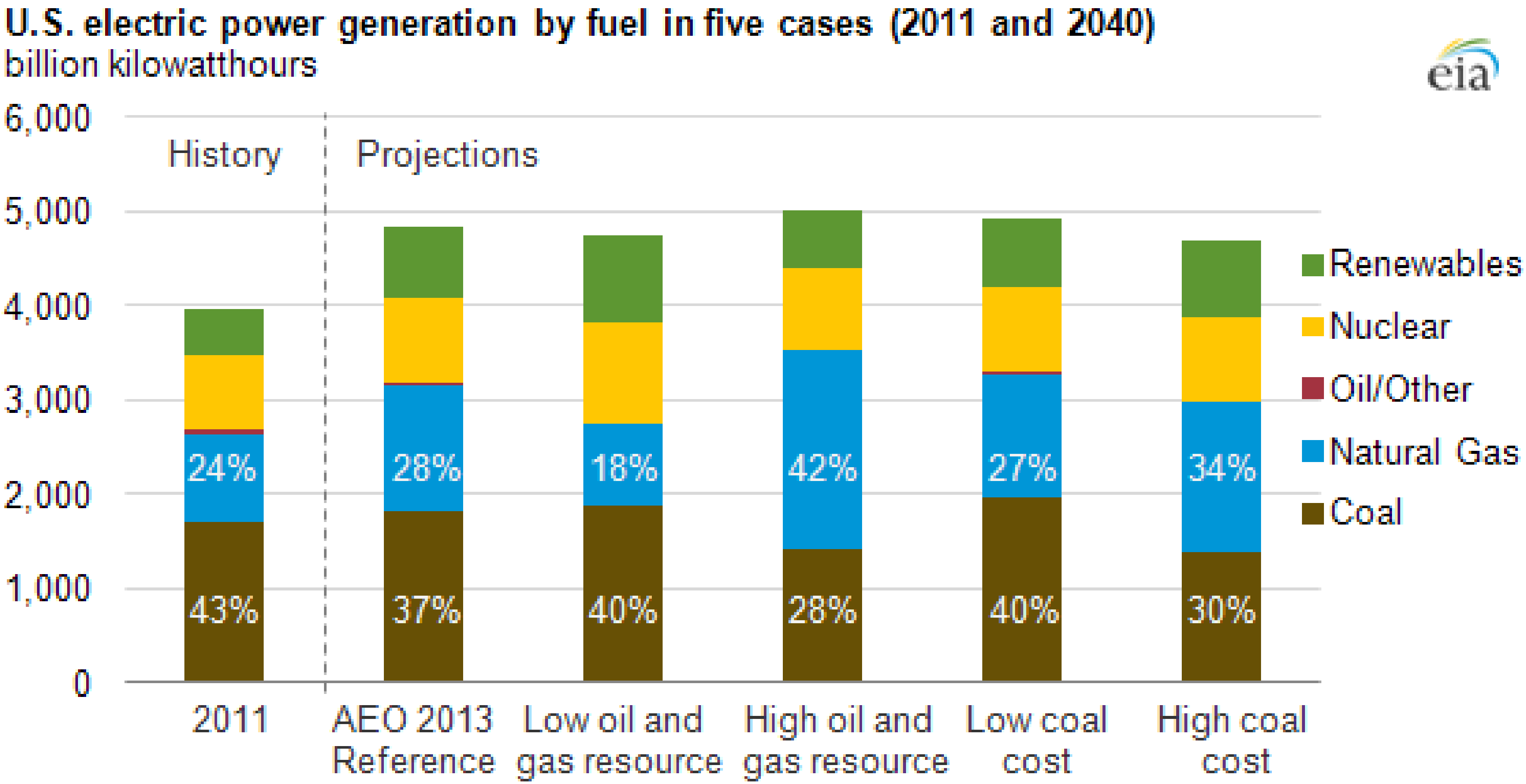
Thermodynamic Cycles

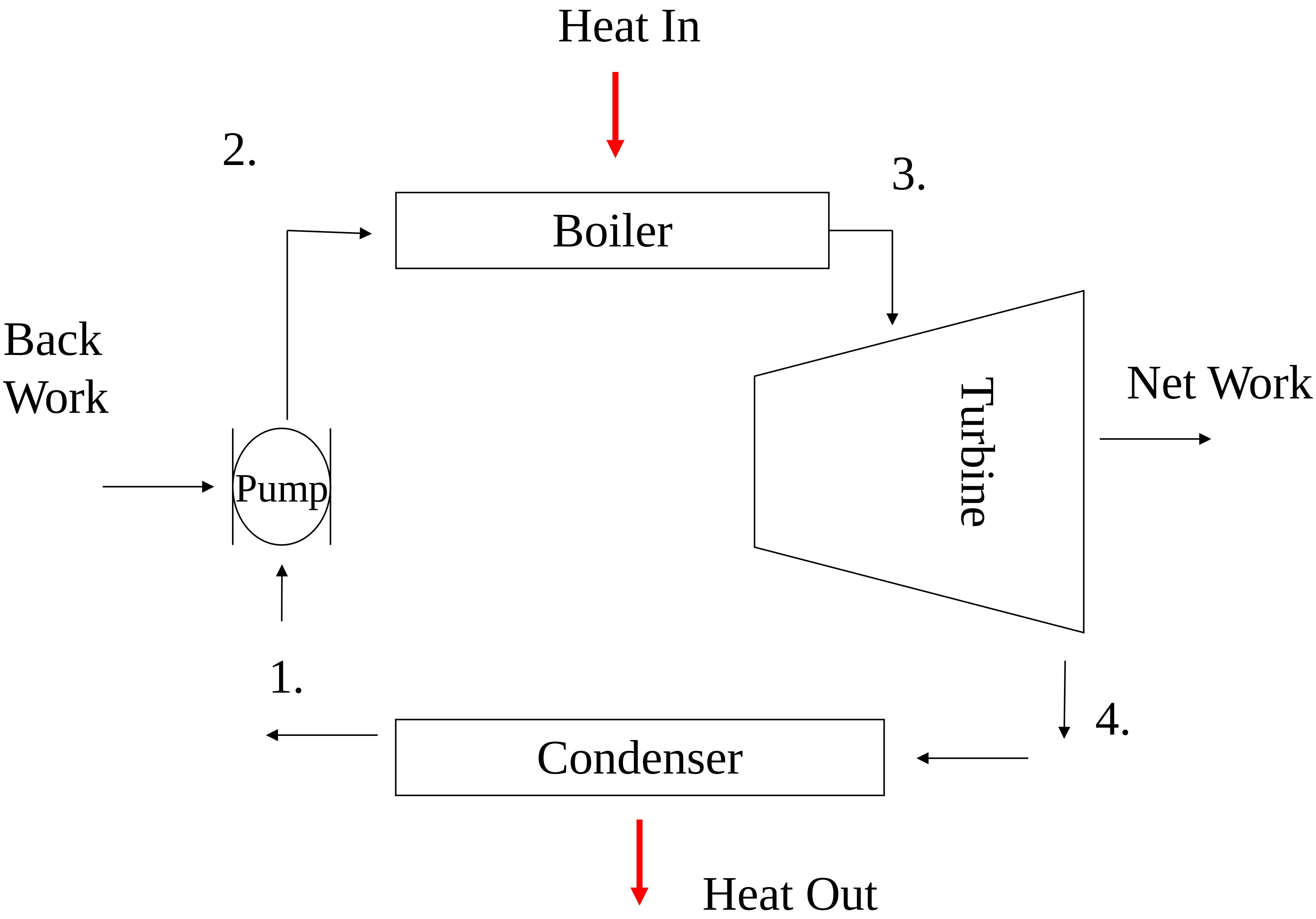
- Carnot Heat Engine
- Rankine
  - Reheat
- Air Standards Cycles
  - Otto
  - Diesel
  - Brayton
    - Regeneration
- Carnot Refrigerator and Heat Pump
- Ideal and Real Refrigeration Cycles



The Rankine Cycle

- The majority (75%+) of power plants are based on the Rankine Cycle





Heat In

2.

3.

Boiler

Back
Work

Pump

Turbine

Net Work

1.

4.

Condenser

Heat Out

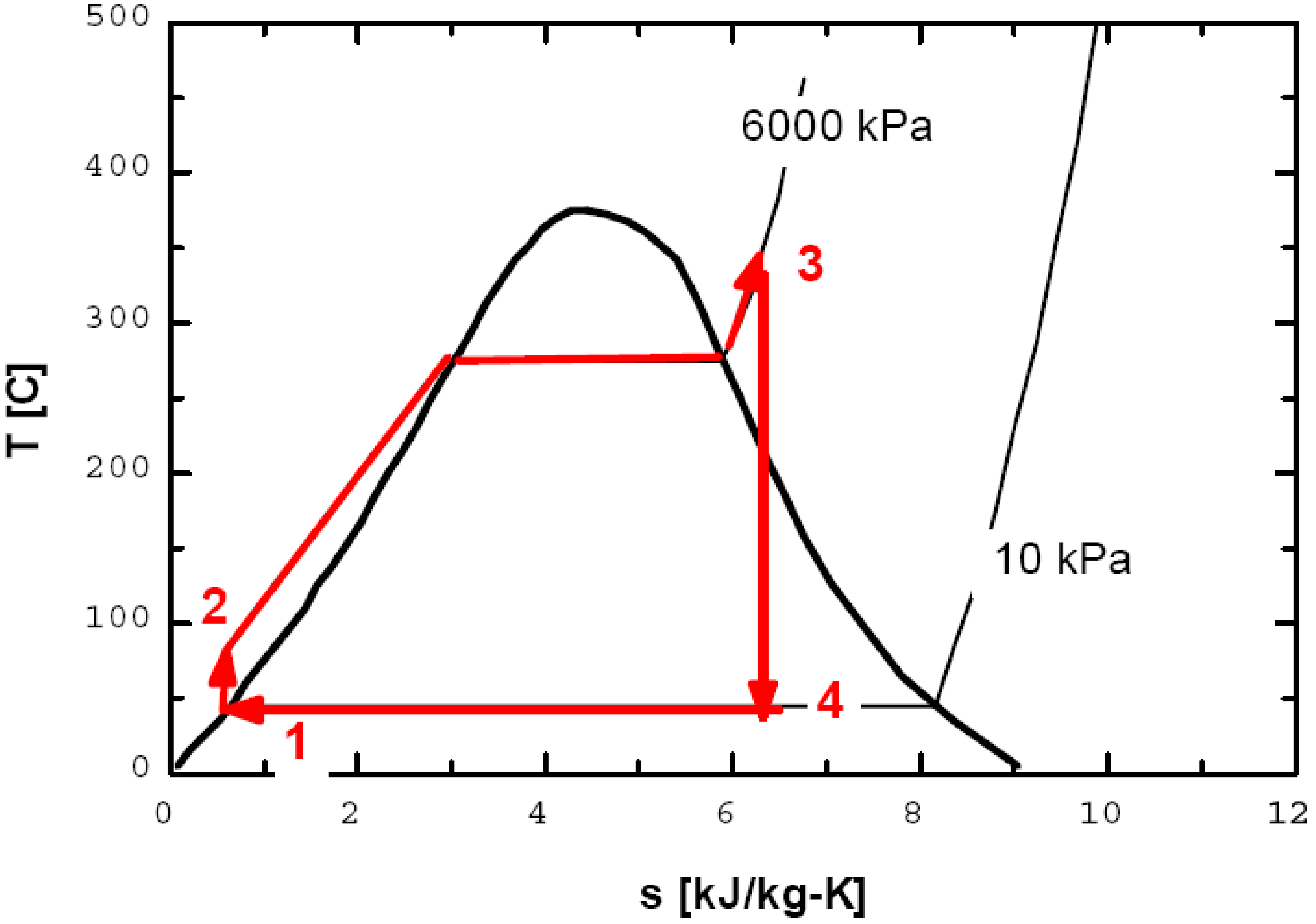
Simple Ideal Rankine

- Isentropic Compression of Liquid (Pump)
- Isobaric Heat Addition (Boiler)
- Isentropic Expansion (Turbine or Piston)
- Isobaric Heat Rejection (Condenser)

Open and Closed Cycles

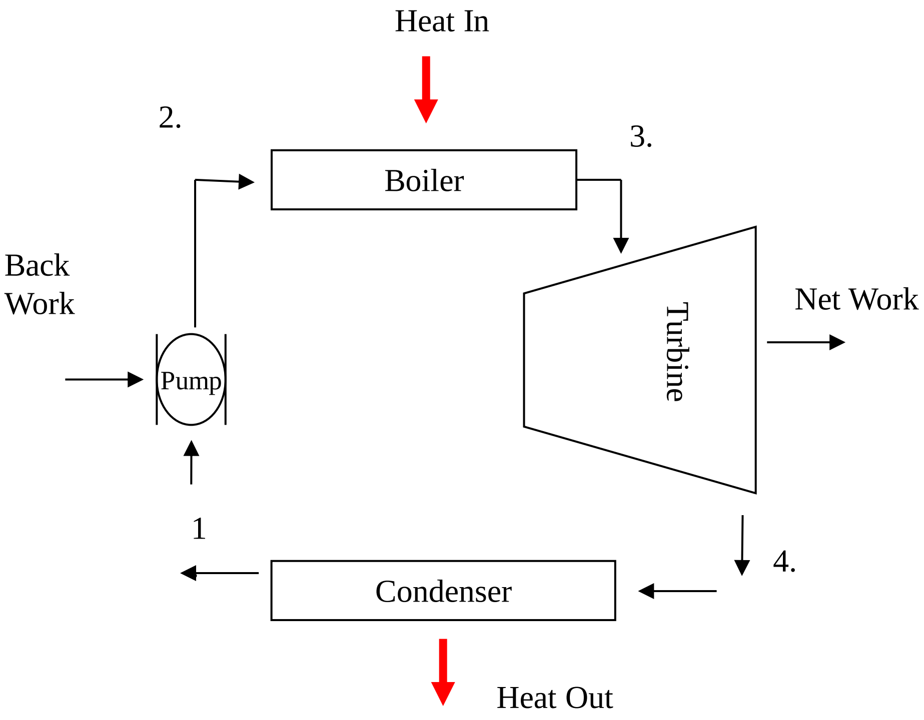
Working Fluid is typically water

Rankine Vapor Power Cycle



Simple Analysis

- State 1 must be liquid
- State 2 must be liquid
- State 3 must be superheated vapor
- State 4 must have Quality better than 0.85



Pump

- Pumps Liquid not Vapor (Cavitation)

From 1st Law $\dot{W}_{pump} = \dot{m}(h_2 - h_1)$

$dh = T\,ds + v\,dP$ For isentropic case $dh = v\,dP$

$$\Delta h = h_2 - h_1 = \int_1^2 v\,dP$$

For incompressible substances like liquids

$$v \cong v_1 = const.$$

$$h_2 - h_1 \cong v_1(P_2 - P_1)$$

$$\dot{W}_{pump} = \dot{m}(h_2 - h_1) \cong \dot{m}v_1(P_2 - P_1)$$

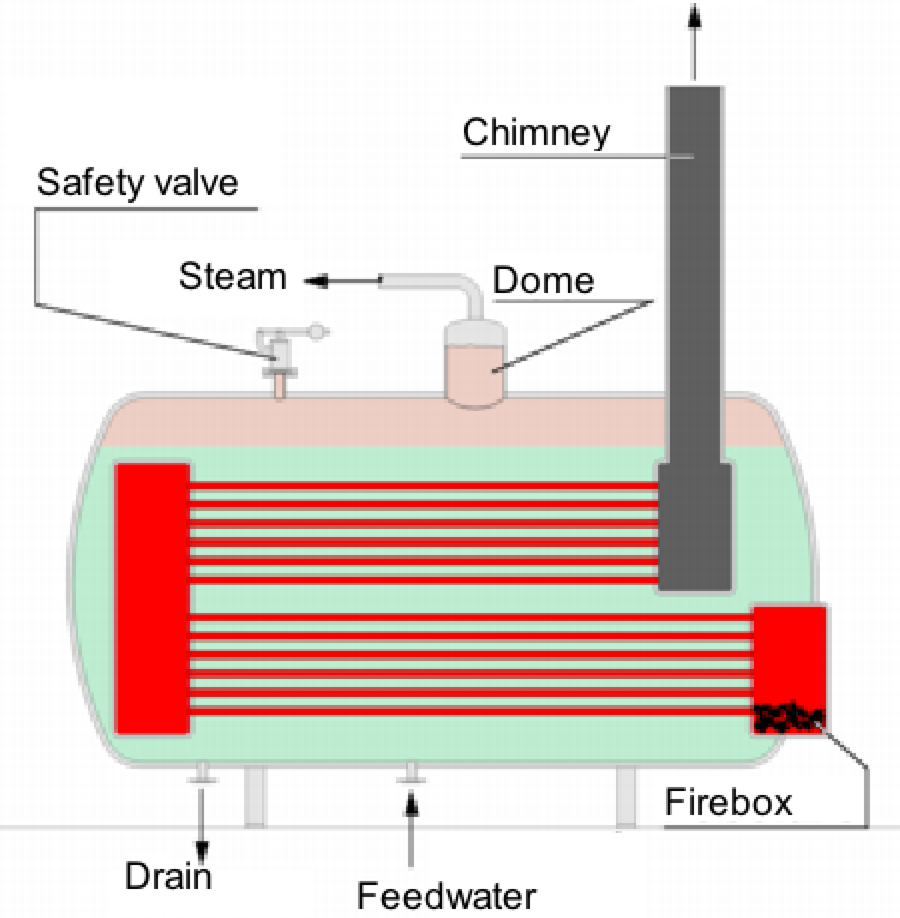
$$w_{pump} = \frac{\dot{W}_{pump}}{\dot{m}} = v_1(P_2 - P_1)$$

Boiler

- Change in Enthalpy of Steam

From 1st Law

$$\dot{Q}_{in} = \dot{m}(h_3 - h_2)$$



Steam Turbine

From 1st Law

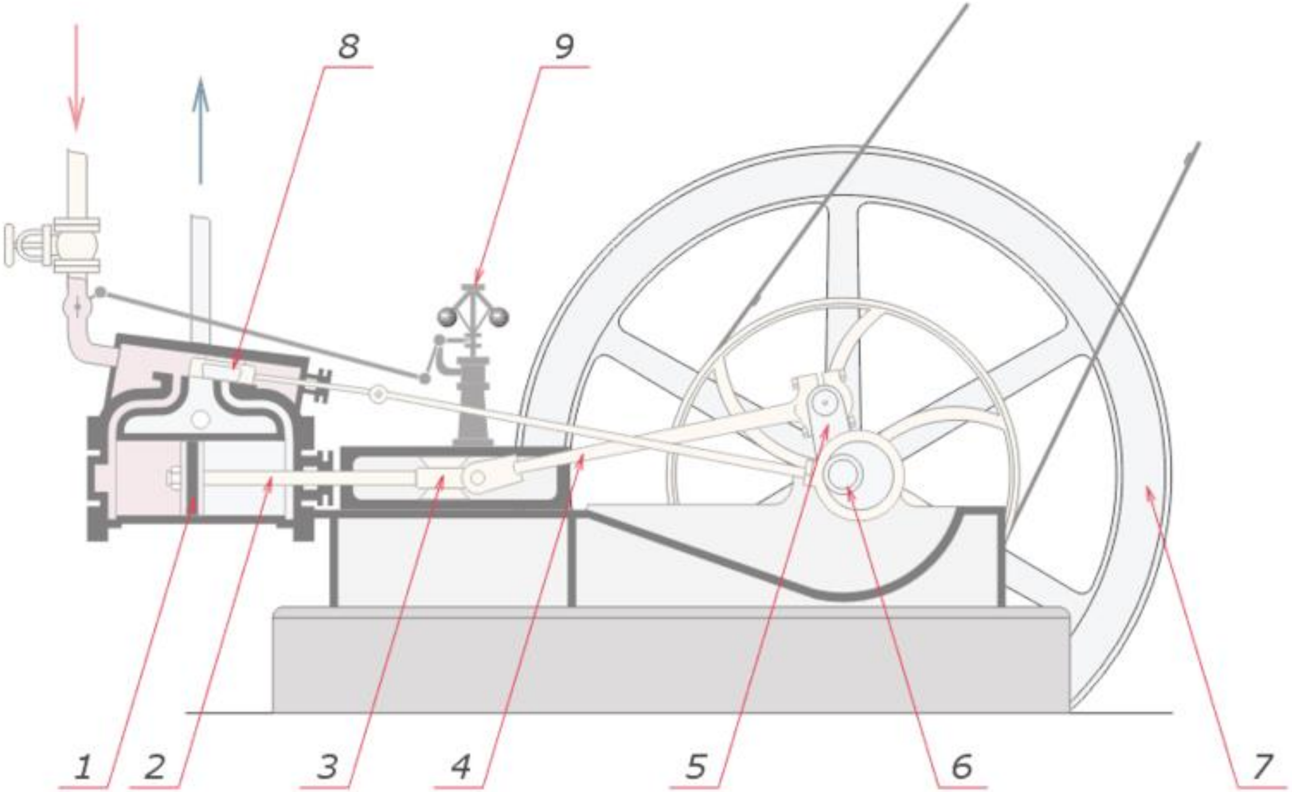
$$\dot{W}_{turb} = \dot{m}(h_3 - h_4)$$

Quality of Turbine exit needs to stay above 0.85

Steam Piston

A typical single cylinder, simple expansion, double-acting high pressure steam engine. Power takeoff from the engine is by way of a belt.

- **1** - Piston
- **2** - Piston rod
- **3** - Crosshead bearing
- **4** - Connecting rod
- **5** - Crank
- **6** - Eccentric valve motion
- **7** - Flywheel
- **8** - Sliding valve
- **9** - Centrifugal governor.



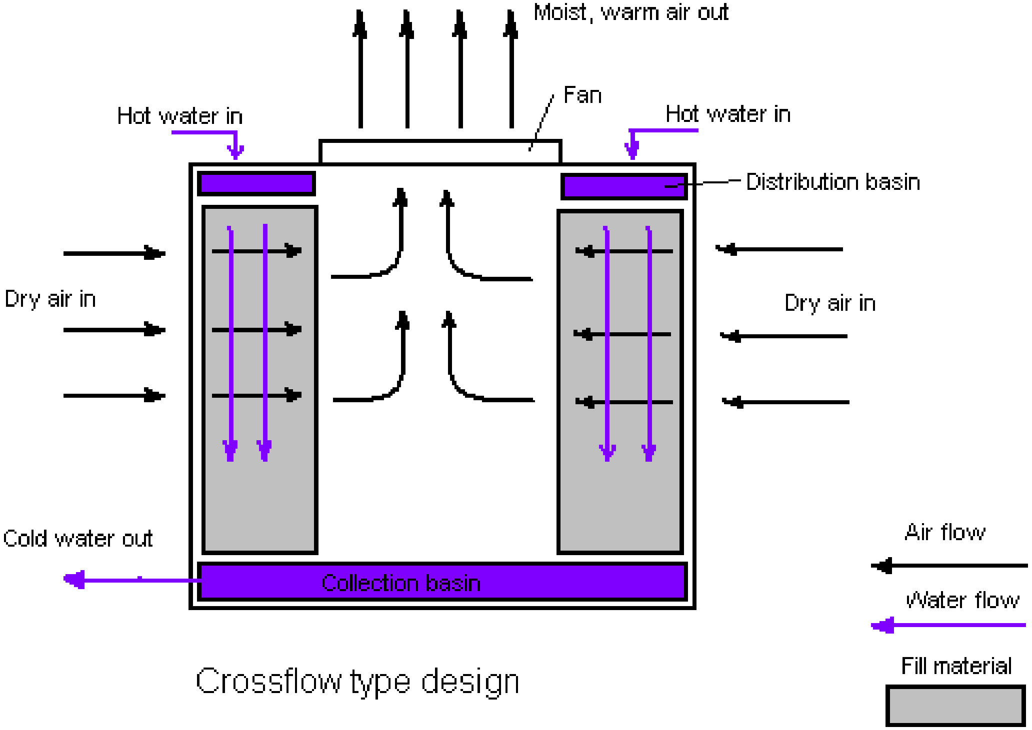
Shaft Power = torque x ω

Condenser

- Atmosphere
- Rivers or Oceans

From 1st Law

$$Q_{out} = \dot{m}(h_4 - h_1)$$



Crossflow type design

A Run Around the Simple Ideal Rankine Cycle

Compute the thermal efficiency of an ideal Rankine cycle for which steam leaves the boiler as superheated vapor at 6 MPa, 350°C, and is condensed at 10 kPa.

Start at Pump Inlet

$$\left.\begin{array}{l} P_1 = 10\ kPa \\ Sat.\ liquid \end{array}\right\} \begin{cases} h_1 = h_f = 191.83\dfrac{kJ}{kg} \\ v_1 = v_f = 0.00101\dfrac{m^3}{kg} \end{cases}$$

$$\begin{aligned} w_{pump} &= v_1(P_2 - P_1) \\ &= 0.00101\frac{m^3}{kg}(6000-10)\ kPa\,\frac{kJ}{m^3 kPa} \\ &= 6.05\frac{kJ}{kg} \end{aligned}$$

$$\begin{aligned} h_2 &= w_{pump} + h_1 \\ &= 6.05\frac{kJ}{kg} + 191.83\frac{kJ}{kg} \\ &= 197.88\frac{kJ}{kg} \end{aligned}$$

Boiler

$$\left.\begin{array}{l} P_3 = 6000\ kPa \\ T_3 = 350^o C \end{array}\right\} \left\{\begin{array}{l} h_3 = 3043.0 \dfrac{kJ}{kg} \\ s_3 = 6.335 \dfrac{kJ}{kg \cdot K} \end{array}\right.$$

$$q_{in} = \frac{\dot{Q}_{in}}{\dot{m}} = h_3 - h_2$$

$$= (3040.3 - 197.88)\frac{kJ}{kg}$$

$$= 2845.2\frac{kJ}{kg}$$

Turbine is Isentropic

$$s_4 = s_3$$

$$at\ \ P_4 = 10kPa\!:\ s_f = 0.6483\frac{kJ}{kg \cdot K};\ s_g = 8.1502\frac{kJ}{kg \cdot K}$$

$$s_4 = s_f + x_4 s_{fg}$$

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.335 - 0.6493}{7.5009} = 0.758$$

$$h_4 = h_f + x_4 h_{fg}$$

$$= 191.83\frac{kJ}{kg} + 0.758(2584.7 - 191.83)\frac{kJ}{kg}$$

$$= 2005.6\frac{kJ}{kg}$$

$$w_{turb} = h_3 - h_4$$

$$= (3043.0 - 2005.63)\frac{kJ}{kg}$$

$$= 1037.4\frac{kJ}{kg}$$

$$w_{net} = w_{turb} - w_{pump}$$

$$= (1037.4 - 6.05)\,\frac{kJ}{kg}$$

$$= 1031.4\,\frac{kJ}{kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$= \frac{1031.4\,\frac{kJ}{kg}}{2845.2\,\frac{kJ}{kg}}$$

$$= 0.363 \; or \;\; 36.3\%$$

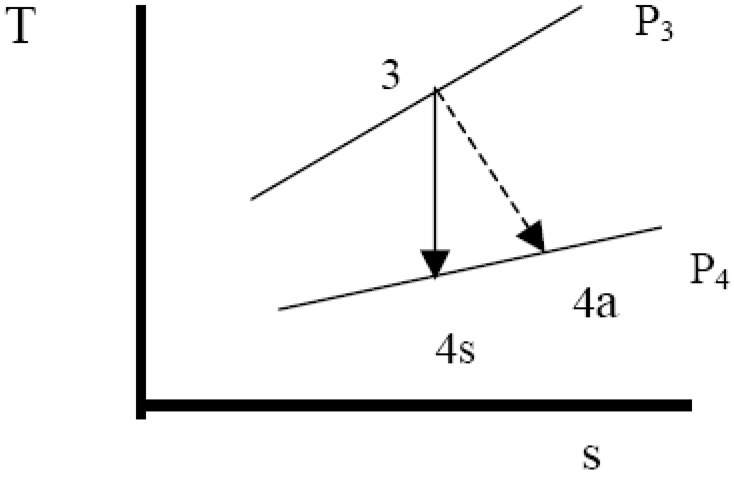
Summary

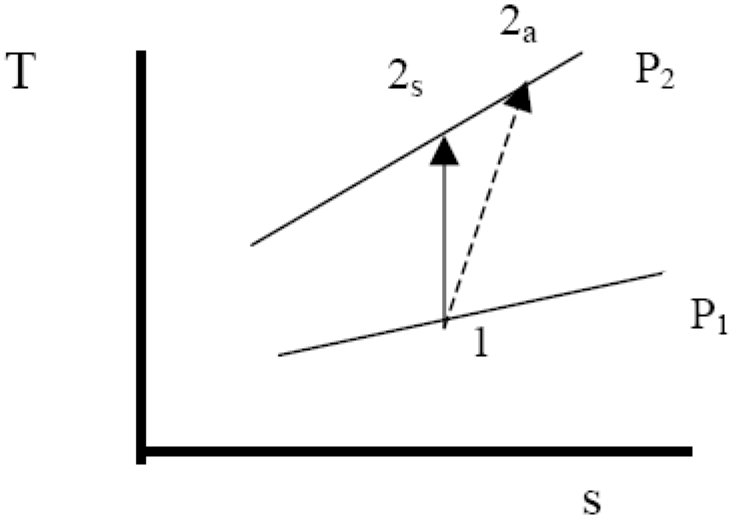
1) Find enthalpy at all States

2) Turn the mathematical crank

Simple Real Rankine

- Piping losses--frictional effects reduce the available energy content of the steam.
- Turbine losses--turbine isentropic (or adiabatic) efficiency.
- Pump losses--pump isentropic (or adiabatic) efficiency.
- Condenser losses--relatively small losses that result from cooling the condensate below the saturation temperature in the condenser.





Real Systems

$$\eta_{turb} = \frac{w_{actual}}{w_{isentropic}} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

$$\eta_{pump} = \frac{w_{isentropic}}{w_{actual}} = \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

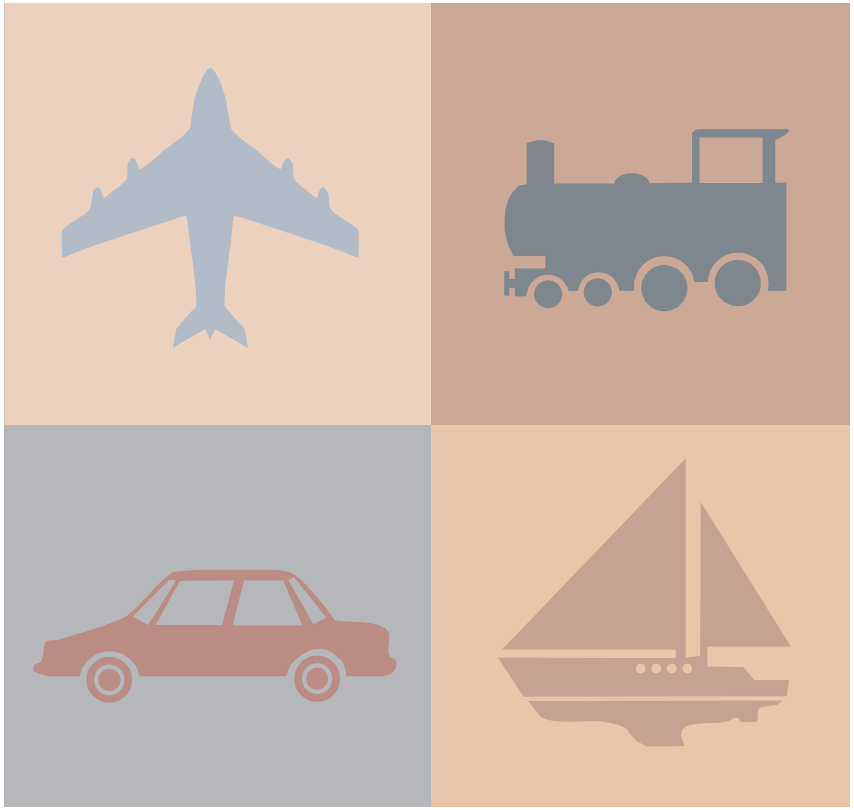
How to Increase Rankine Cycle Efficiency

- Increase Boiler Temperature and Pressure (materials constraints)
- Decrease Condenser Pressure
  - increases moisture in turbine outlet
  - Large turbines required as density of steam drops
- Reheat
- Preheat using Open and Closed Feedwater heaters

Air Standard Cycles

- Otto
- Diesel
- Brayton

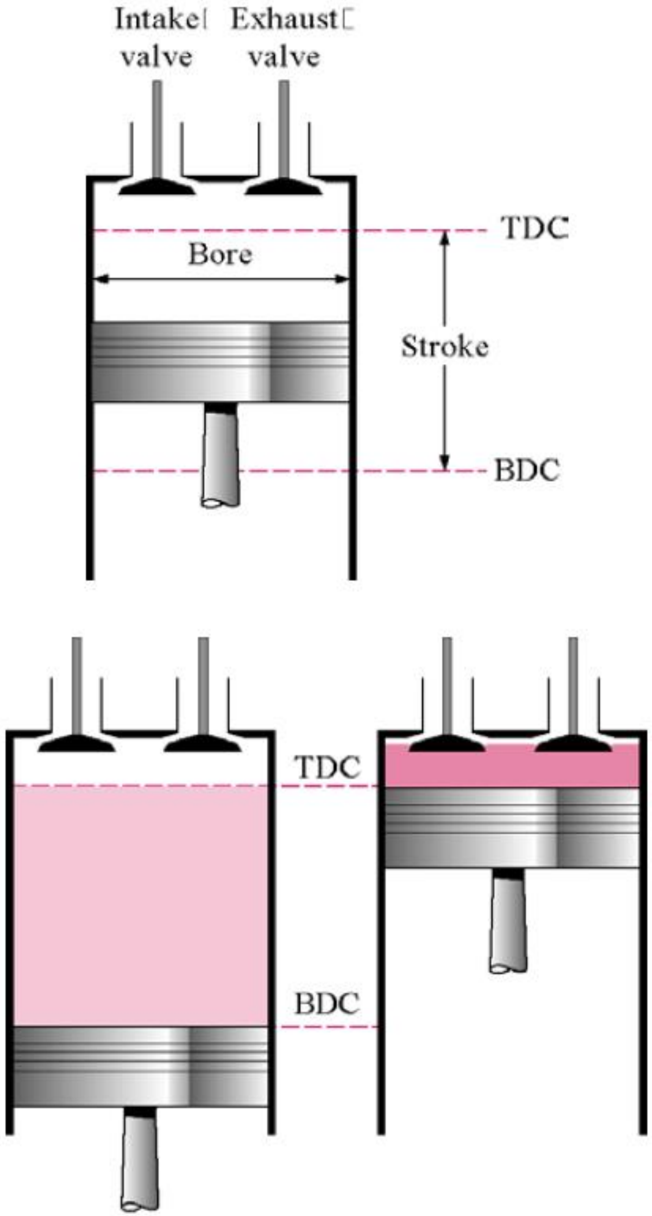
Planes, Trains, and Automobiles



Air Standard Assumptions

- Working fluid is air and acts as an ideal gas
- All Processes are internally reversible
- Combustion process is replaced by heat addition
- Exhaust process is replaced by heat rejection
- Cold ASA are that properties of air are constant at 25C

Reciprocating Internal Combustion Engines



(*a*) Displacement volume

(*b*) Clearance volume

Otto Engine Spark Ignition (SI)

- High Power Density (aircraft, cars, etc)
- Most common power plant in US automotive industry

0.25 mile race at speeds over 300 mph
(15 gallons of methanol used)

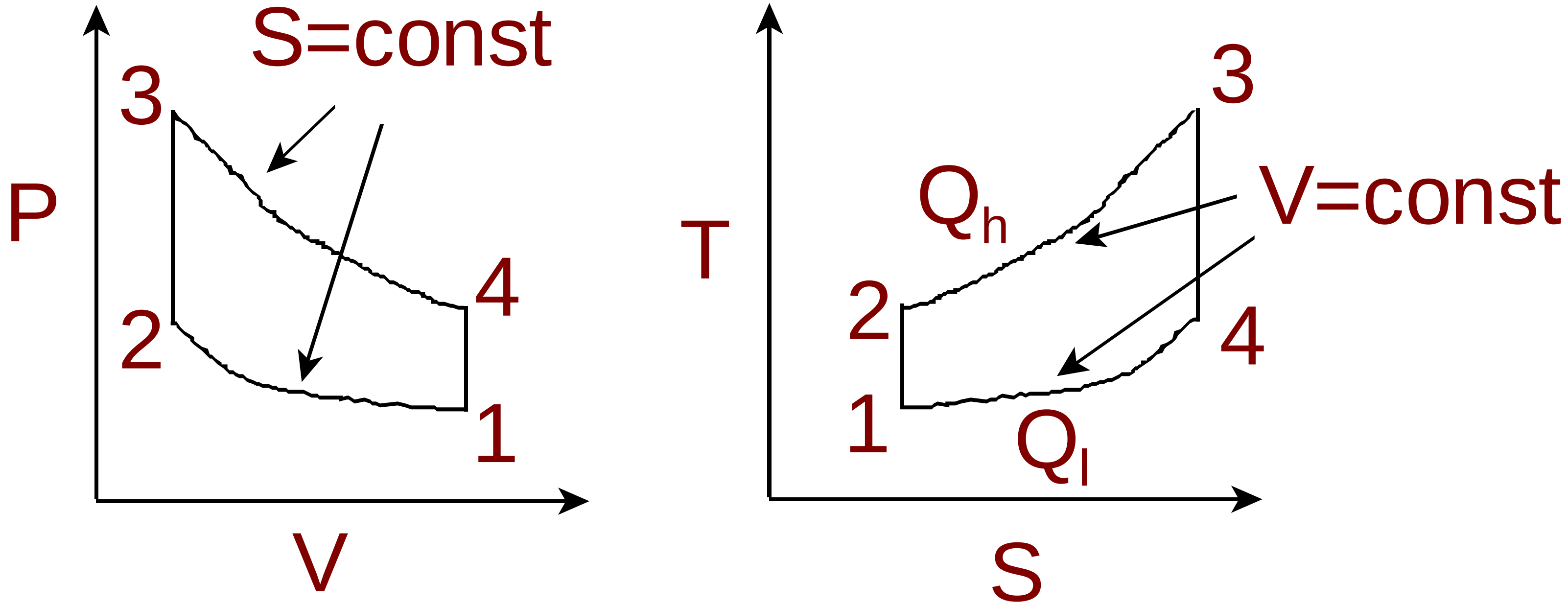
Diagrams for Ideal Otto Cycle

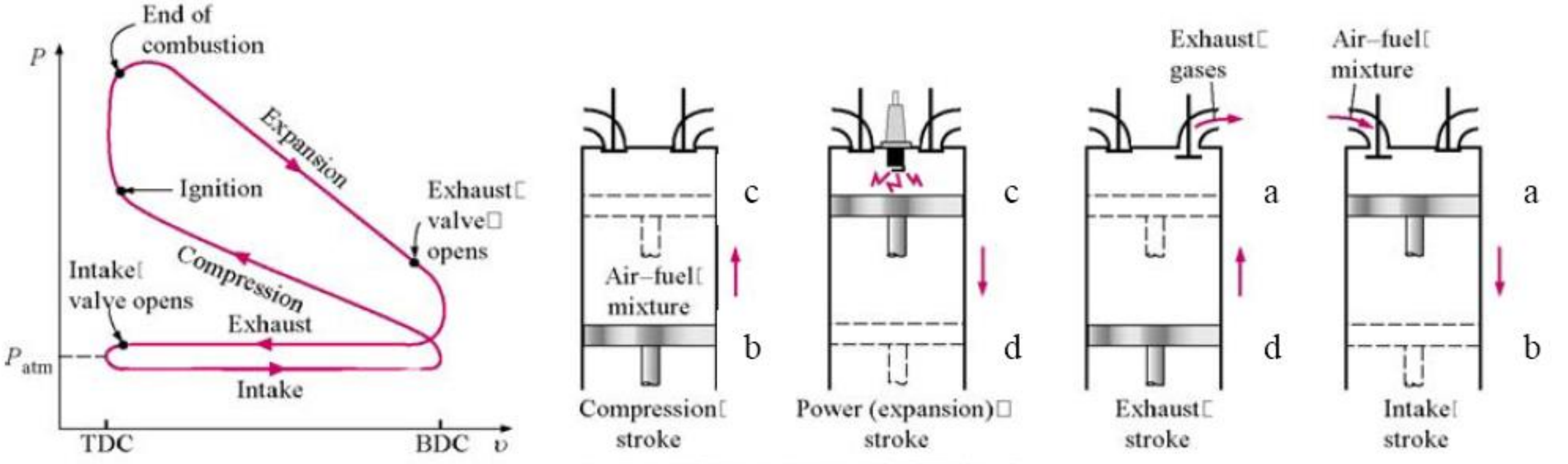
Isentropic compression

Isometric heat addition

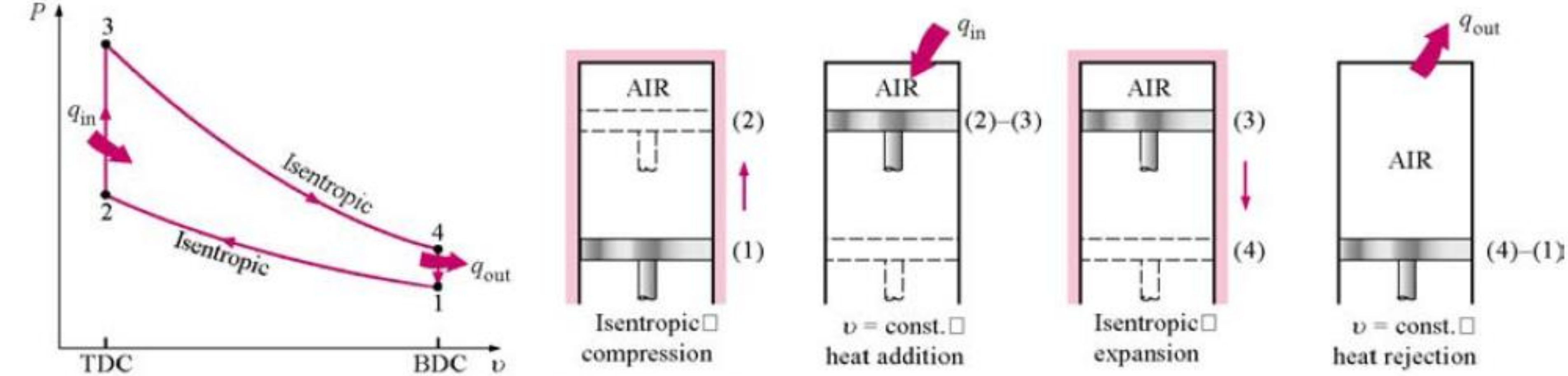
Isentropic expansion

Isometric heat rejection





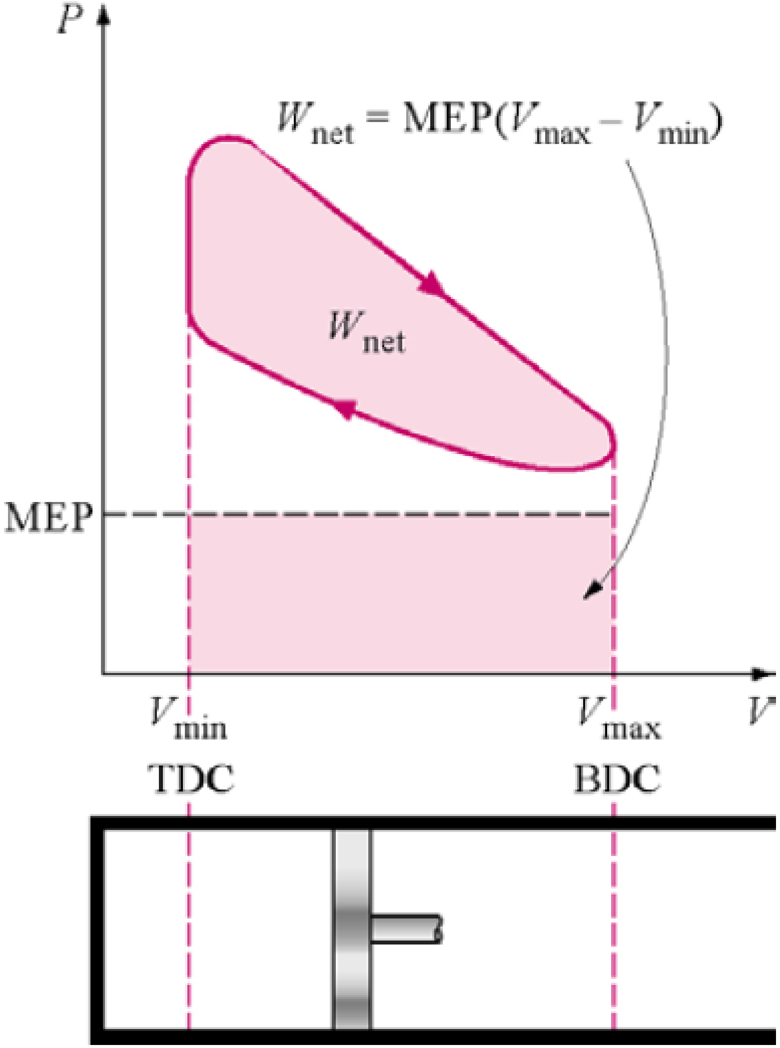
(*a*) Actual four-stroke spark-ignition engine



(*b*) Ideal Otto cycle

REAL (a)   IDEAL(b)

Mean Effective Pressure MEP



$$MEP = \frac{W_{net}}{V_{max} - V_{min}} = \frac{w_{net}}{v_{max} - v_{min}}$$

Fictitious pressure for cycle in terms of average pressure for a given displacement

Much lower than real pressures seen through cycle

DO NOT use MEP to calculate required strength of materials

Otto Cycle Efficiency

$$\eta_{th,Otto} = \frac{Q_h - Q_l}{Q_h} = 1 - \frac{Q_l}{Q_h} = 1 - \frac{mc_p\left(T_4 - T_1\right)}{mc_p\left(T_3 - T_2\right)} = 1 - \frac{T_1}{T_2}\left(\frac{\frac{T_4}{T_1} - 1}{\frac{T_3}{T_2} - 1}\right)$$

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} \qquad \frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{k-1}$$

$$V_4 = V_1$$

$$V_3 = V_2$$

$$\left(\frac{V_1}{V_2}\right)^{k-1} = \left(\frac{V_4}{V_3}\right)^{k-1}$$

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

Derivation continued

$$\eta_{th,Otto} = \frac{Q_h - Q_l}{Q_h} = 1 - \frac{Q_l}{Q_h} = 1 - \frac{mc_p\left(T_4 - T_1\right)}{mc_p\left(T_3 - T_2\right)} = 1 - \frac{T_1}{T_2}\left(\frac{\frac{T_4}{T_1} - 1}{\frac{T_3}{T_2} - 1}\right)$$

$$\eta_{th,Otto} = 1 - \left(\frac{T_1}{T_2}\right)$$

But remember

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1}$$

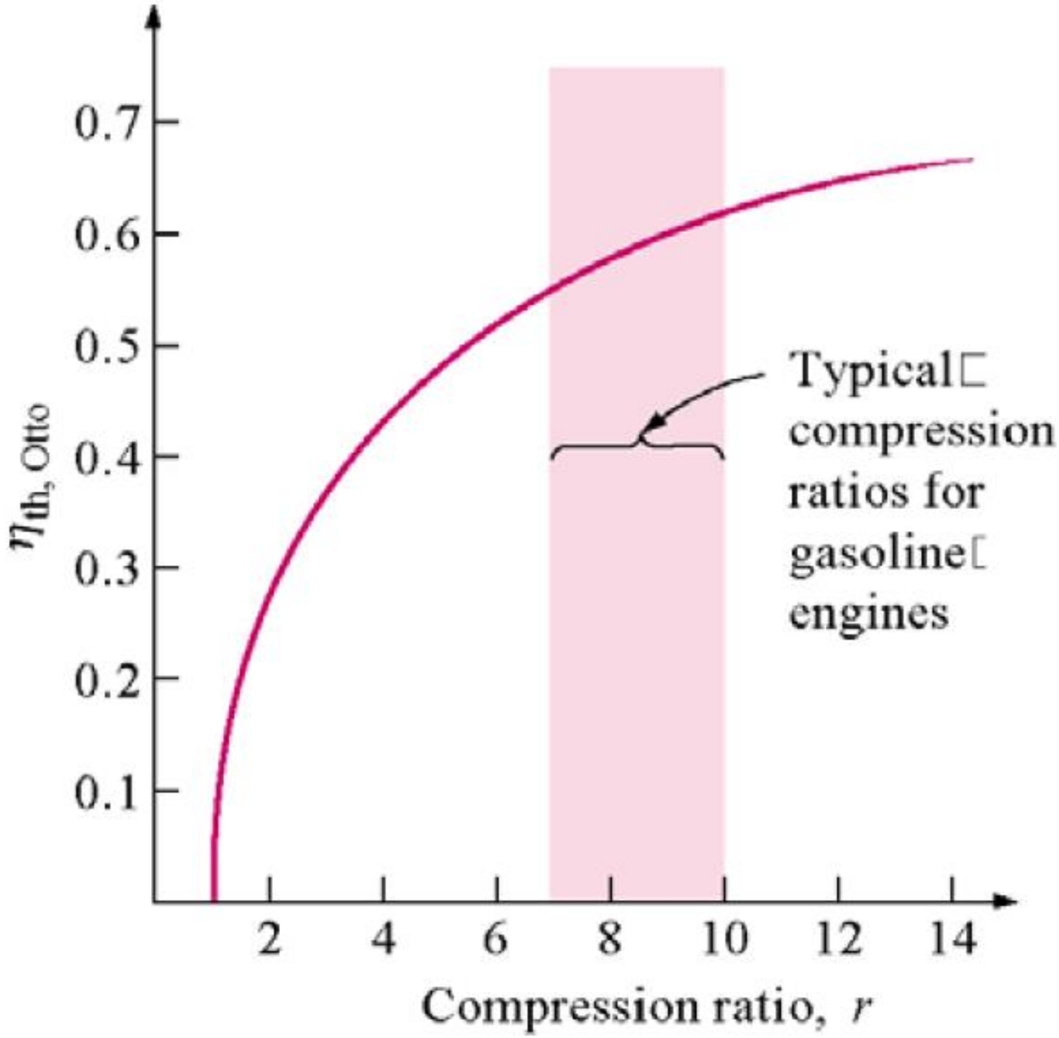
$$r = \frac{V_{1,4}}{V_{3,2}}$$

$$\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}}$$

Modeled Otto Cycle efficiency

$$\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}}$$

- Increase temperature by increasing compression ratio but autoignition occurs
- Use knock inhibitors to raise Octane number (lead was used in US until 1973-1996)


0.7
0.6
0.5
0.4
0.3
0.2
0.1
$\eta_{th,\,Otto}$
2
4
6
8
10
12
14
Compression ratio, r
Typical compression ratios for gasoline engines

A Run Around the Otto Cycle

An Otto cycle having a compression ratio of 9:1 uses air as the working fluid. Initially P_1 = 95 kPa, T_1 = 17°C, and V_1 = 3.8 liters. During the heat addition process, 7.5 kJ of heat are added. Determine all T's, P's, η_{th}, the back work ratio, and the mean effective pressure.

Assume constant specific heats with $C_v = 0.718$ kJ/kg ·K, $k = 1.4$

1 – 2 is Isentropic Compression

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = T_1(r)^{k-1}$$

$$= (17+273)K(9)^{1.4-1}$$

$$= 698.4K$$

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^{k} = P_1(r)^{k}$$

$$= 95kPa(9)^{1.4}$$

$$= 2059\,kPa$$

2 – 3 Isometric or Isochoric Heat Addition

$$Q_{in} = mC_v(T_3 - T_2)$$

$$q_{\text{in}} = Q_{\text{in}} / m \text{ and } m = V_1 / v_1$$

$$v_1 = \frac{RT_1}{P_1}$$

$$= \frac{0.287\frac{kJ}{kg \cdot K}(290\,K)}{95\,kPa}\frac{m^3 kPa}{kJ}$$

$$= 0.875\frac{m^3}{kg}$$

$$q_{in} = \frac{Q_{in}}{m} = Q_{in}\frac{v_1}{V_1}$$

$$= 7.5kJ\frac{0.875\frac{m^3}{kg}}{3.8 \cdot 10^{-3} m^3}$$

$$= 1727\frac{kJ}{kg}$$

$$T_3 = T_2 + \frac{q_{in}}{C_v}$$

$$= 698.4K + \frac{1727\frac{kJ}{kg}}{0.718\frac{kJ}{kg \cdot K}}$$

$$= 3103.7\,K$$

$$P_3 = P_2\frac{T_3}{T_2} = 9.15\,MPa$$

3 - 4 Expansion is Isentropic

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = T_3\left(\frac{1}{r}\right)^{k-1}$$

$$= (3103.7)K\left(\frac{1}{9}\right)^{1.4-1}$$

$$= 1288.8\,K$$

$$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{k} = P_3\left(\frac{1}{r}\right)^{k}$$

$$= 9.15\,MPa\left(\frac{1}{9}\right)^{1.4}$$

$$= 422\,kPa$$

4 – 1 Heat Rejection is Isometric

$$Q_{out} = mC_v(T_4 - T_1)$$

$$q_{out} = \frac{Q_{out}}{m} = C_v(T_4 - T_1)$$

$$= 0.718\frac{kJ}{kg\cdot K}(1288.8 - 290)K$$

$$= 717.1\frac{kJ}{kg}$$

$$w_{net} = q_{net} = q_{in} - q_{out}$$
$$= (1727 - 717.4)\frac{kJ}{kg}$$
$$= 1009.6\frac{kJ}{kg}$$

$$\eta_{th,\,Otto} = \frac{w_{net}}{q_{in}} = \frac{1009.6\frac{kJ}{kg}}{1727\frac{kJ}{kg}}$$
$$= 0.585 \quad or \quad 58.5\%$$

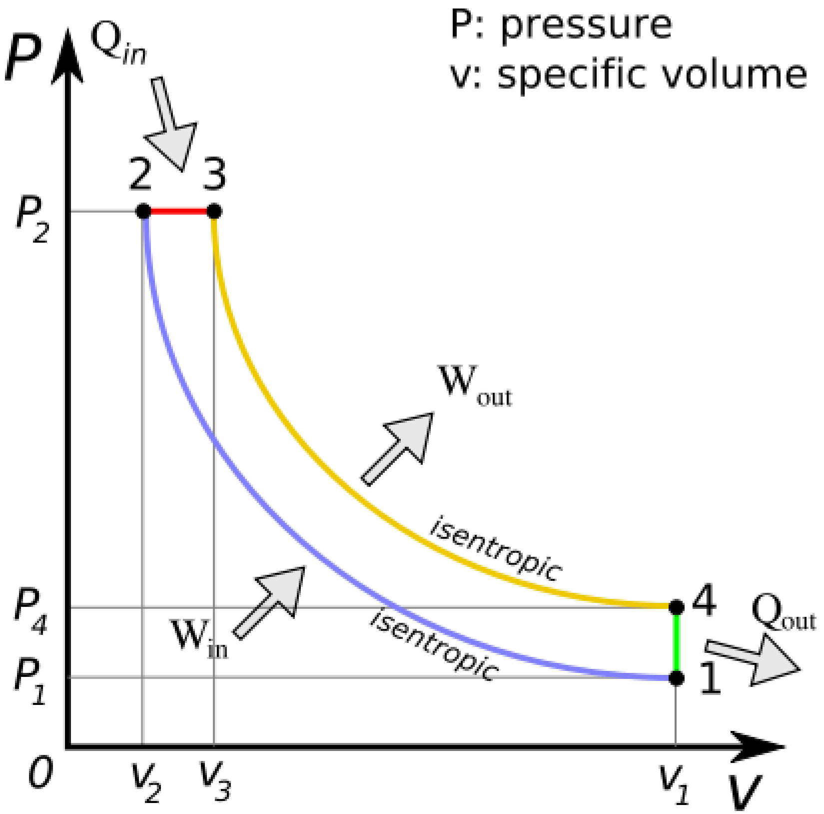
$$MEP = \frac{W_{net}}{V_{\max} - V_{\min}} = \frac{w_{net}}{v_{\max} - v_{\min}}$$
$$= \frac{w_{net}}{v_1 - v_2} = \frac{w_{net}}{v_1(1 - v_2/v_1)} = \frac{w_{net}}{v_1(1 - 1/r)}$$
$$= \frac{1009.6\frac{kJ}{kg}}{0.875\frac{m^3}{kg}(1 - \frac{1}{9})}\frac{m^3 kPa}{kJ} = 1298\ kPa$$

$$BWR = \frac{w_{comp}}{w_{\exp}} = \frac{\Delta u_{12}}{-\Delta u_{34}}$$
$$= \frac{C_v(T_2 - T_1)}{C_v(T_3 - T_4)} = \frac{(T_2 - T_1)}{(T_3 - T_4)}$$
$$= 0.225 \quad or \quad 22.5\%$$

Diesel Engines CI

- High Efficiency (trucks, buses, construction equipment, ships, etc) (cars esp europe)
- Low power density (runs fuel lean)

P-v Diagram for Diesel



Isentropic compression

Isobaric heat addition

Isentropic expansion

Isometric heat rejection

Diesel Efficiency

$$\eta_{th,\,Diesel} = 1 - \frac{Q_{out}}{Q_{in}}$$

$$= 1 - \frac{mC_v(T_4 - T_1)}{mC_p(T_3 - T_2)}$$

$$\frac{P_3V_3}{T_3} = \frac{P_2V_2}{T_2} \quad \text{where } P_3 = P_2$$

$$\frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c$$

$$\eta_{th,\,Diesel} = 1 - \frac{C_v(T_4 - T_1)}{C_p(T_3 - T_2)}$$

$$= 1 - \frac{1}{k}\frac{T_1(T_4 / T_1 - 1)}{T_2(T_3 / T_2 - 1)}$$

r_c or Alpha α is the cutoff ratio corresponding to the volume change during heat addition

$$\frac{P_4 V_4}{T_4} = \frac{P_1 V_1}{T_1} \quad \text{where } V_4 = V_1$$

$$P_1 V_1^k = P_2 V_2^k \quad \text{and} \quad P_4 V_4^k = P_3 V_3^k$$

$$\frac{T_4}{T_1} = \frac{P_4}{P_1}$$

$V_4 = V_1$ and $P_3 = P_2$,

$$\frac{P_4}{P_1} = \left(\frac{V_3}{V_2}\right)^k = r_c^k$$

$$\eta_{th,\,Diesel} = 1 - \frac{1}{k}\frac{T_1(T_4 / T_1 - 1)}{T_2(T_3 / T_2 - 1)}$$

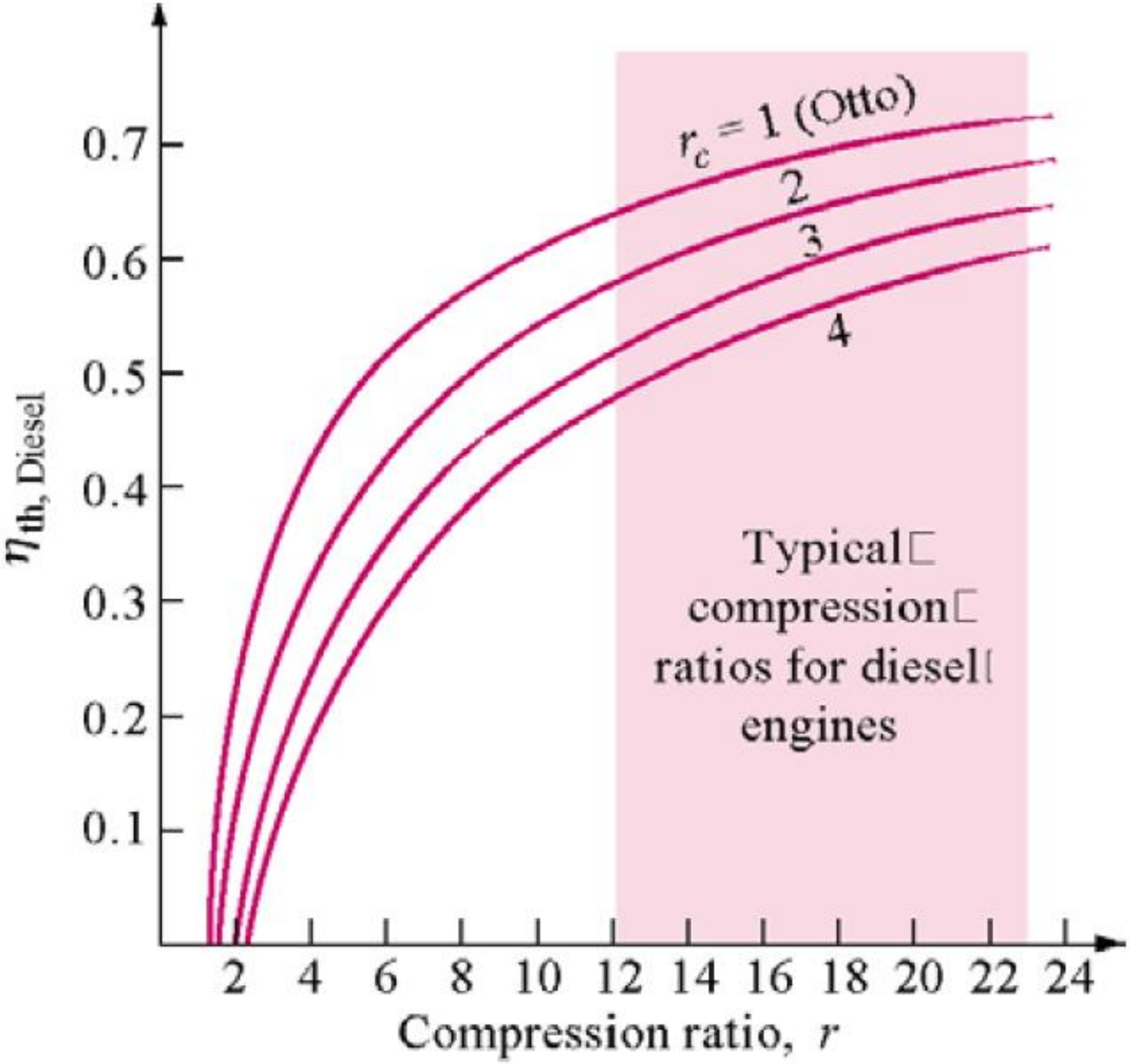
$$= 1 - \frac{1}{k}\frac{T_1}{T_2}\frac{r_c^k - 1}{(r_c - 1)}$$

$$= 1 - \frac{1}{r^{k-1}}\frac{r_c^k - 1}{k(r_c - 1)}$$

Diesel efficiency

$$\eta_{th,Diesel} = 1 - \frac{1}{r^{k-1}}\left(\frac{\alpha^{k} - 1}{k(\alpha - 1)}\right)$$

Where α is the cutoff ratio v3/v2 corresponding to the burn duration


r_c = 1 (Otto)
2
3
4
0.7
0.6
0.5
0.4
0.3
0.2
0.1
$\eta_{\text{th, Diesel}}$
2
4
6
8
10
12
14
16
18
20
22
24
Compression ratio, r
Typical compression ratios for diesel engines

A Run Around the Diesel Cycle

At the beginning of the compression process of an air-standard Diesel cycle operating with a compression ratio of 18, the temperature is 300 K and the pressure is 0.1 MPa. The cutoff ratio for the cycle is 2. Determine **(a)** the temperature and pressure at the end of each process of the cycle, **(b)** the thermal efficiency, **(c)** the mean effective pressure, in MPa.

Values from Tables (variable specific heat)

isentropic compression process 1–2

$p2 = 5.39$ MPa $T2 = 898.3$ K and $h2 = 930.98$ kJ/kg.

isobaric heat addition process 2–3

$$\frac{P_3 V_3}{T_3} = \frac{P_2 V_2}{T_2} \quad \text{where } P_3 = P_2$$

$T3 = r_c T2 = 2*(898.3) = 1796.6$ K

$$\frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c$$

$h3 = 1999.1$ kJ/kg

isentropic expansion process 3–4

$T4$ = 887.7 K.

$p4$ = 0.3 MPa

$u4$ = 664.3 kJ/kg

$$\eta = 1 - \frac{Q_{41}/m}{Q_{23}/m} = 1 - \frac{u_4 - u_1}{h_3 - h_2}$$

$$= 1 - \frac{664.3 - 214.07}{1999.1 - 930.98} = 0.578(57.8\%)$$

$$\frac{W_{\text{cycle}}}{m} = \frac{Q_{23}}{m} - \frac{Q_{41}}{m} = (h_3 - h_2) - (u_4 - u_1)$$

$$= (1999.1 - 930.98) - (664.3 - 214.07)$$

$$= 617.9 \text{ kJ/kg}$$

$$\text{mep} = \frac{W_{\text{cycle}}/m}{v_1 - v_2} = \frac{W_{\text{cycle}}/m}{v_1(1 - 1/r)}$$

$$v_1 = \frac{(\overline{R}/M)T_1}{p_1} = \frac{\left(\dfrac{8314}{28.97}\,\dfrac{\text{N}\cdot\text{m}}{\text{kg}\cdot\text{K}}\right)(300\text{ K})}{10^5\text{ N/m}^2} = 0.861\text{ m}^3/\text{kg}$$

$$\text{mep} = \frac{617.9\text{ kJ/kg}}{0.861(1 - 1/18)\text{m}^3/\text{kg}}\left|\frac{10^3\text{ N}\cdot\text{m}}{1\text{ kJ}}\right|\left|\frac{1\text{ MPa}}{10^6\text{ N/m}^2}\right|$$

$$= 0.76\text{ MPa} \triangleleft$$

Large Diesel Engines for Ships

Wartsila-Sulzer RTA96-C turbocharged two-stroke diesel engine

Total engine weight: 2300 tons (The crankshaft alone weighs 300 tons.)

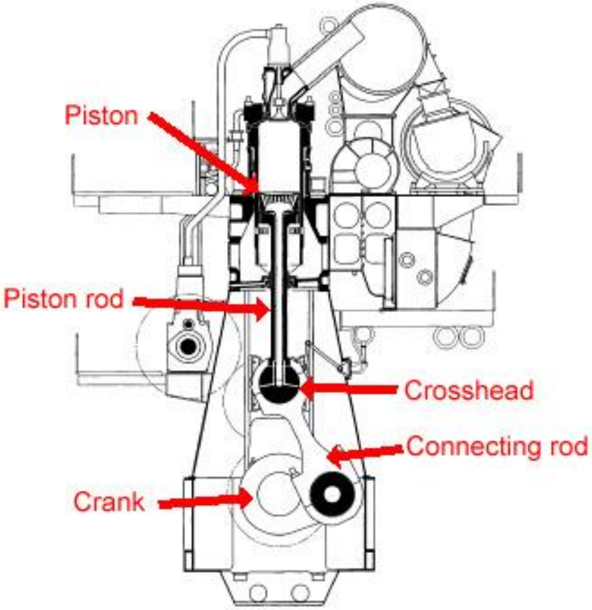
Length: 89 feet

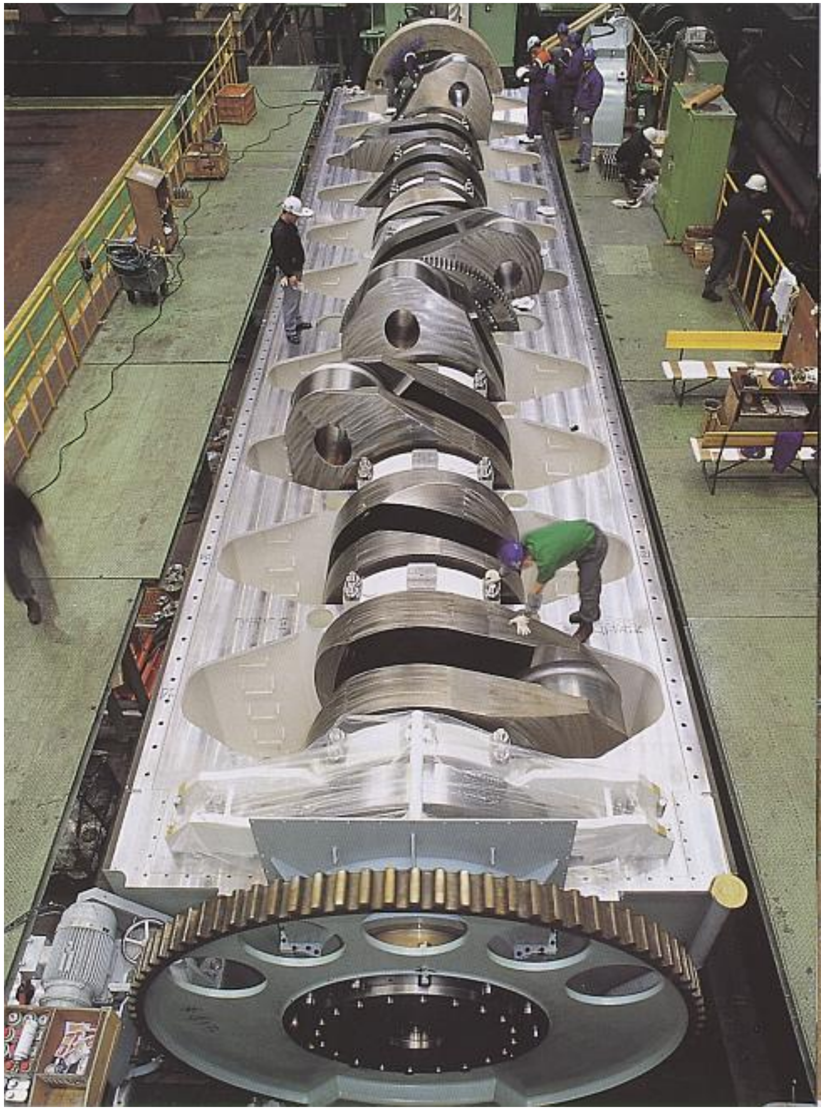
Height: 44 feet

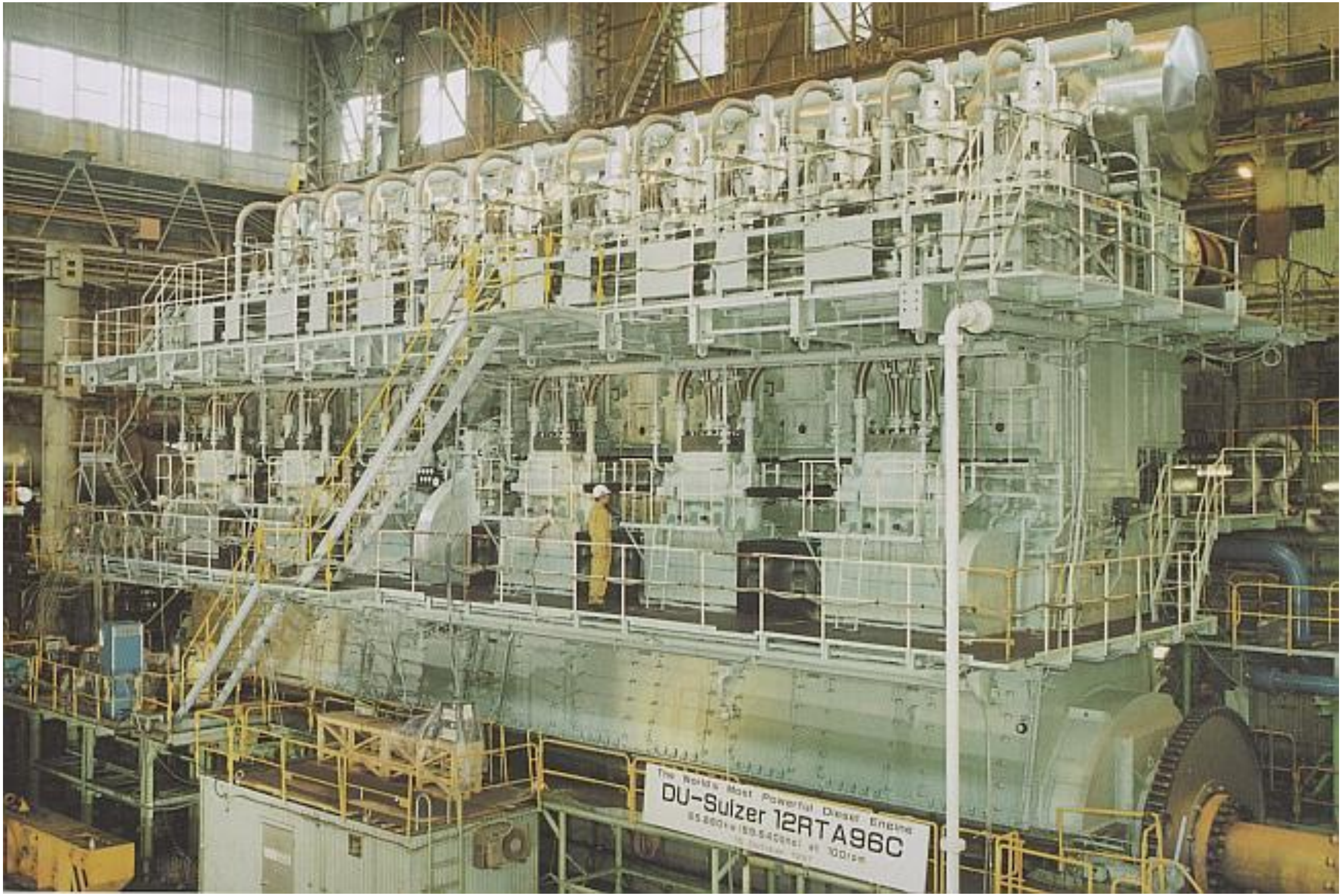
Maximum power: 108,920 hp at 102 rpm

Maximum torque: 5,608,312 lb/ft at 102rpm

50% efficiency



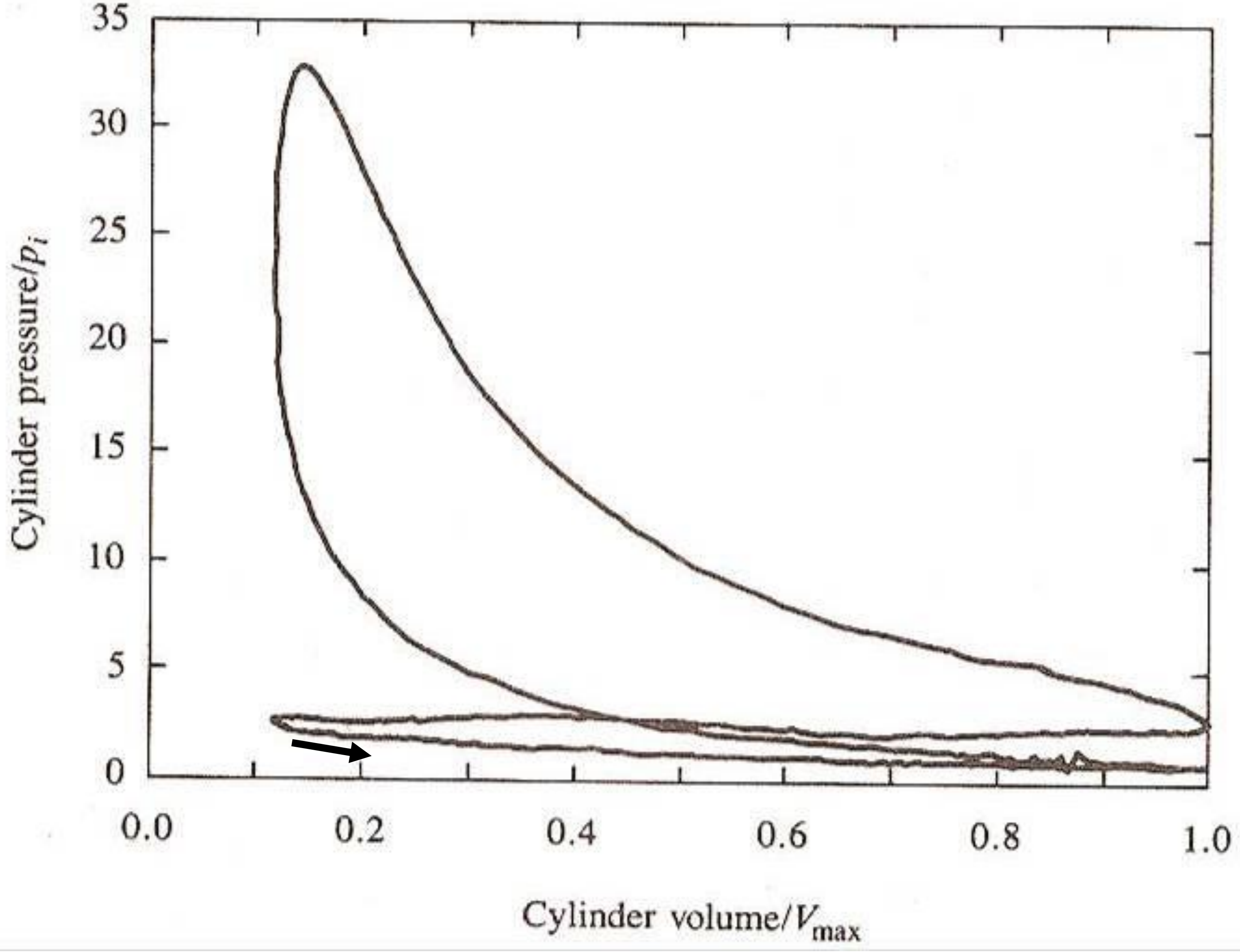



The World's Most Powerful Diesel Engine
DU-Sulzer 12RTA96C
65,880kW(89,640ps) at 100rpm

Reality vs. Models

- Irreversible processes
  - Friction
  - Unrestrained Expansion
  - Heat Transfer
- Exhaust stroke
- Intake stroke
- Heat addition takes time in otto cycle and changes pressure in diesel cycle
- Throttling losses (esp at low power/idle)
- Heat losses to cylinder wall
- k ≠ 1.4

This accounts for the large differences (1/2) in modeled versus real efficiencies in the ICE

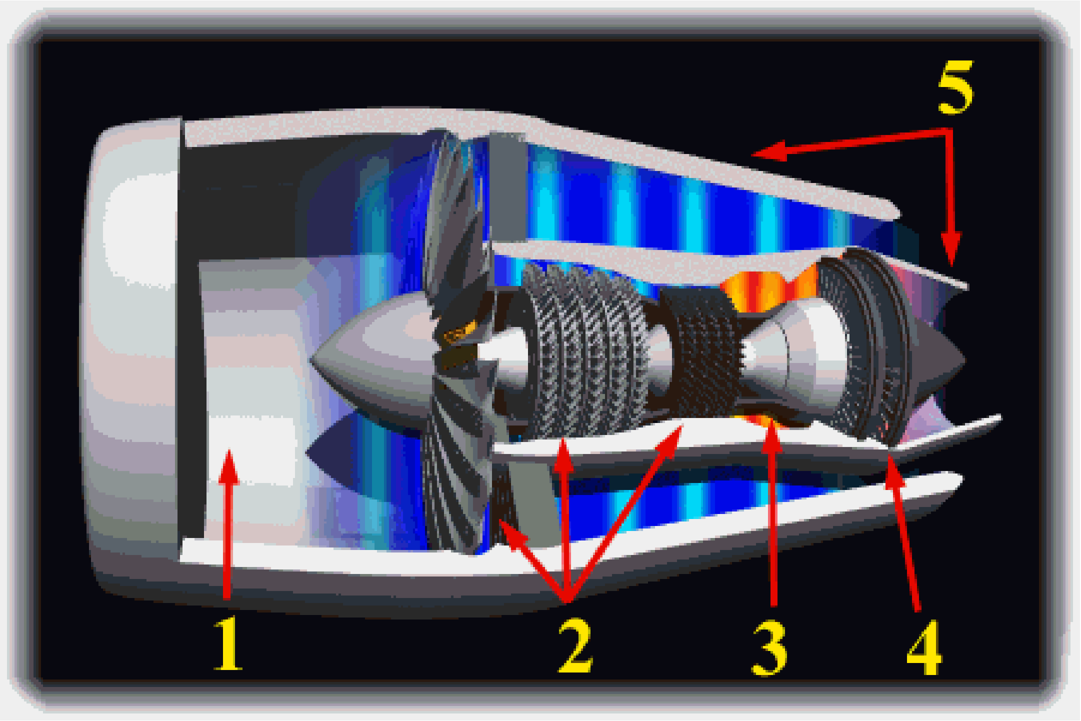

35
30
25
20
15
10
5
0
0.0
0.2
0.4
0.6
0.8
1.0
Cylinder pressure/p_i
Cylinder volume/V_{max}

Improving Efficiency

- Decrease entropy generation
  - Friction
  - Unrestrained expansion of gases
  - Heat transfer
- Increase temperature
- Shorten burn time for diesel

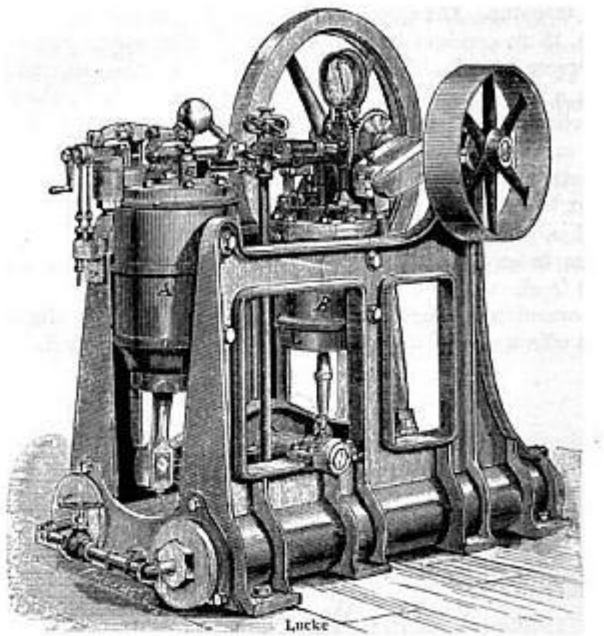
$$\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}} \qquad \eta_{th,Diesel} = 1 - \frac{1}{r^{k-1}}\left(\frac{\alpha^{k} - 1}{k(\alpha - 1)}\right)$$

Brayton Cycle

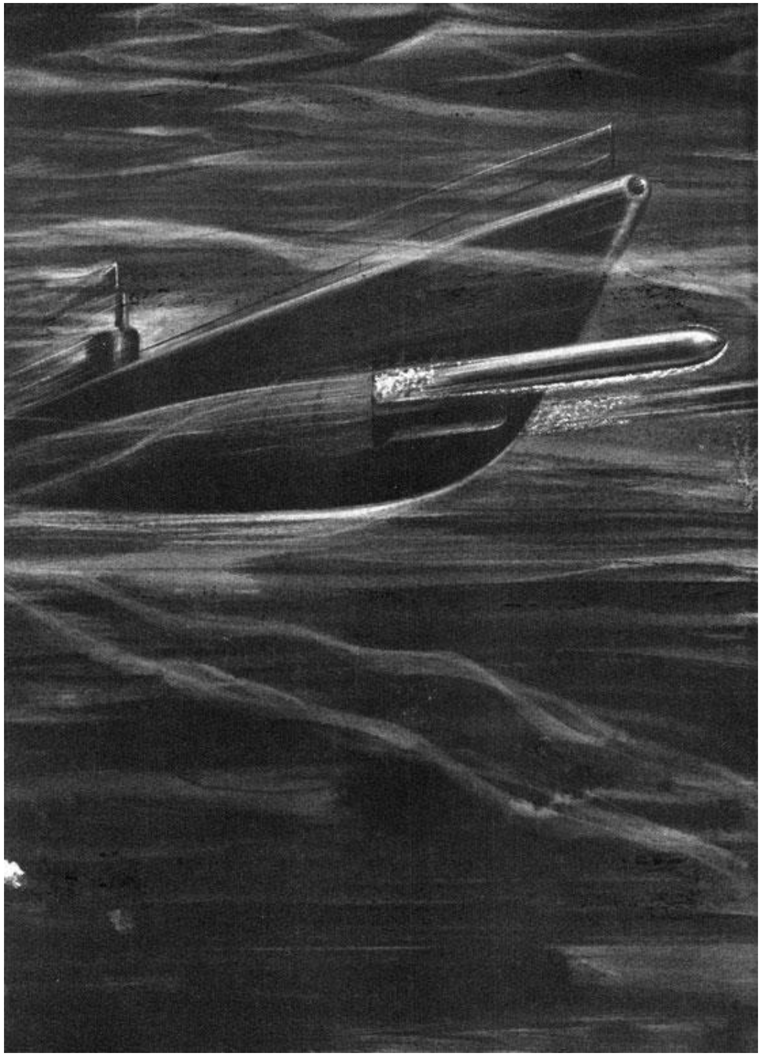


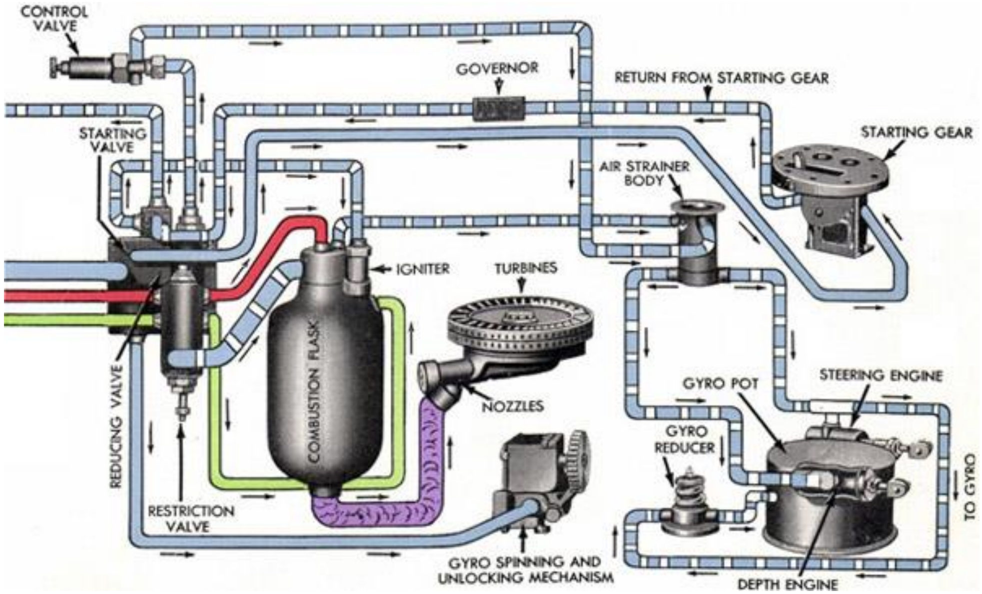
George Brayton (1830-1892)

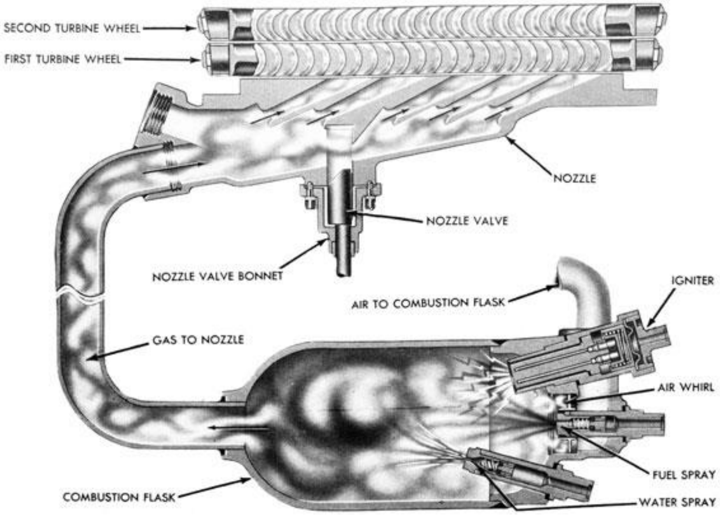
- Envisioned a continuous heat addition at constant pressure for a reciprocating engine.
- Eventually this idea morphed into a compressor, followed by burner followed by an expansion device.
- Gas turbine!



Torpedo Development (wet heater)







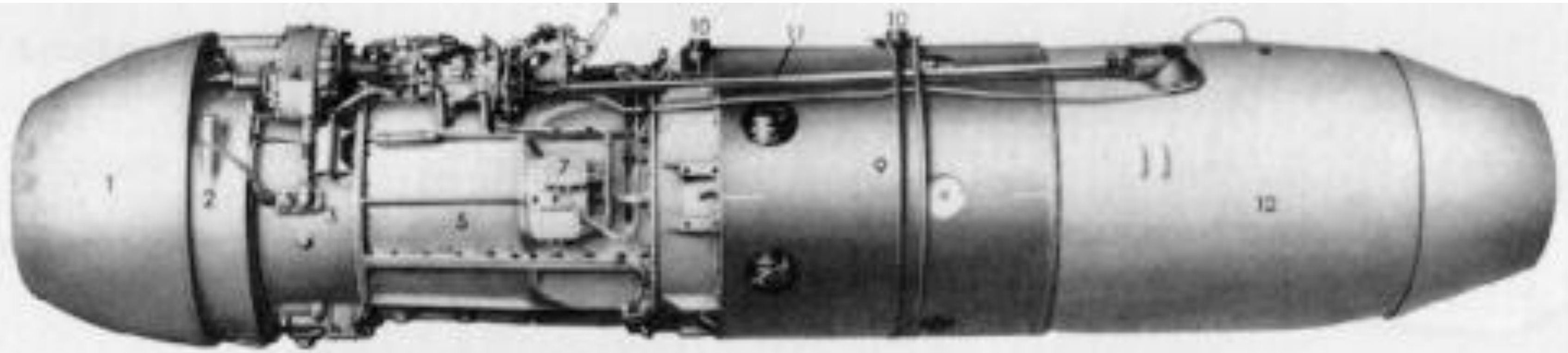
Gas Turbines

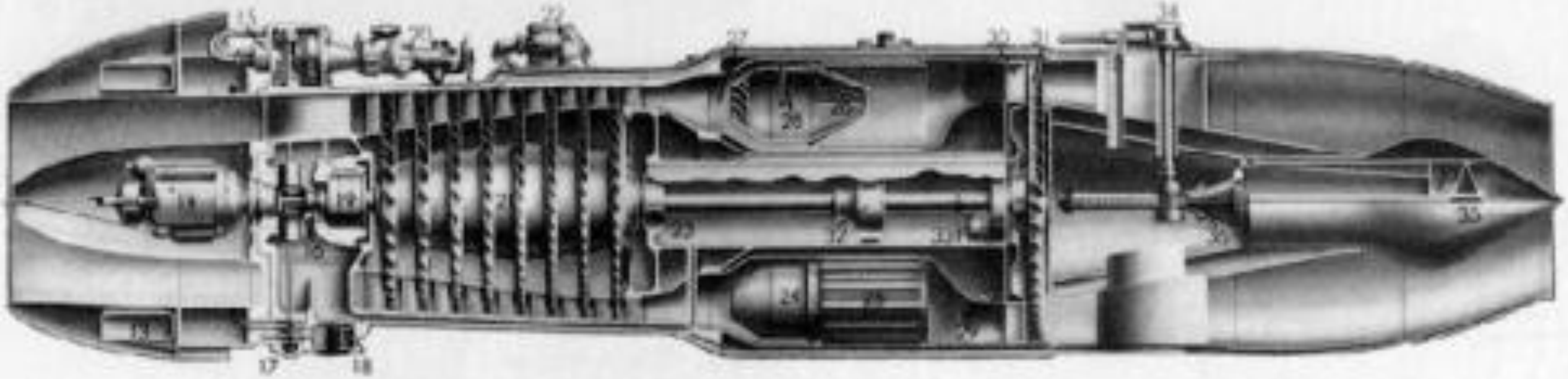
- Developed in WW2 independently by British and German researchers (concepts known in 1920s)
- Efficiency is lower than Otto and Diesel High backwork required
- Significantly higher power density
- ME 262 could outclimb the P-51 Mustang and fly 100 mph faster. Less efficient but roughly double the power!
- Open Brayton Cycle

http://www.rolls-royce.com/interactive_games/journey02/flash.html

In 1942 Adolf Galland—director general of fighters for the Luftwaffe, veteran of the Battle of Britain, and one of Germany's top aces—flew a prototype Messerschmitt ME 262. "For the first time, I was flying by jet propulsion and there was no torque, no thrashing sound of the propeller, and my jet shot through the air," he commented. "It was as though angels were pushing."

Hitler delays production of ME 262 in favor of “Offensive” Aircraft and Weapons





1 Vorhaube
2 Schmierstoffbehälter
3 Stirngehäuse
4 Geräteträger
5 Verdichtergehäuse
6 Schubregler
7 Zündgeräte
8 Bedienungsgestängehebel
9 Tragzapfen
10 Aufhängung

11 Düsennadelverstellwelle
12 Schubdüse
13 Kraftstoffringbehälter
14 Riedel-Anlasser
15 Einspritzpumpe
16 Abzweiggetriebe
17 Schmierstoffpumpen
18 Schmierstofffilter
19 Vorderes Verdichterlager

20 Drehzahlregler
21 Verdichterläufer
22 Kraftstoffilter
23 Hinteres Verdichterlager
24 Muffel
25 Heizkammereinsatz
26 [illegible]
27 Heizkammer
28 Einspritzdüse

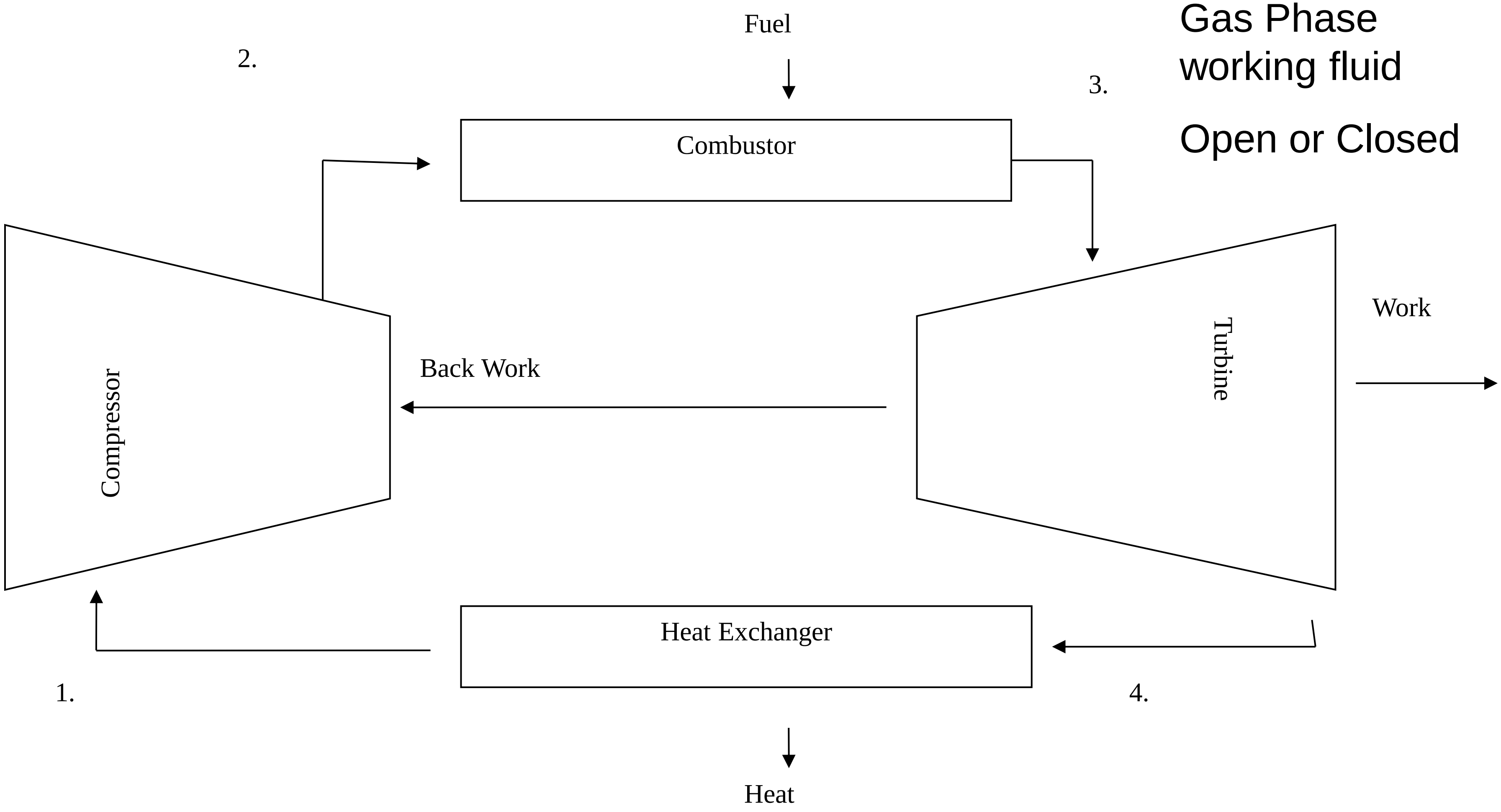
29 Sammler
30 Leitkranz
31 Turbinenläufer
32 Vorderes Turbinenlager
33 Hinteres Turbinenlager mit Schmierstoffrückförderpumpen
34 Düsennadelverstellgetriebe
35 Düsennadel
36 Düsennadellagerung

Jumo 004 - B
Ansicht und Schnitt
Fl. Üb. Nr. 9-062/1
Stand vom September 1944
Inzwischen eingetretene Änderungen beachten und darauf hinweisen

Types of turbine engines

- Turbojet
- Turbofan
- Turboprop
- Turboshaft

Simple Brayton Cycle



Gas Phase working fluid

Open or Closed

Open Brayton Cycle

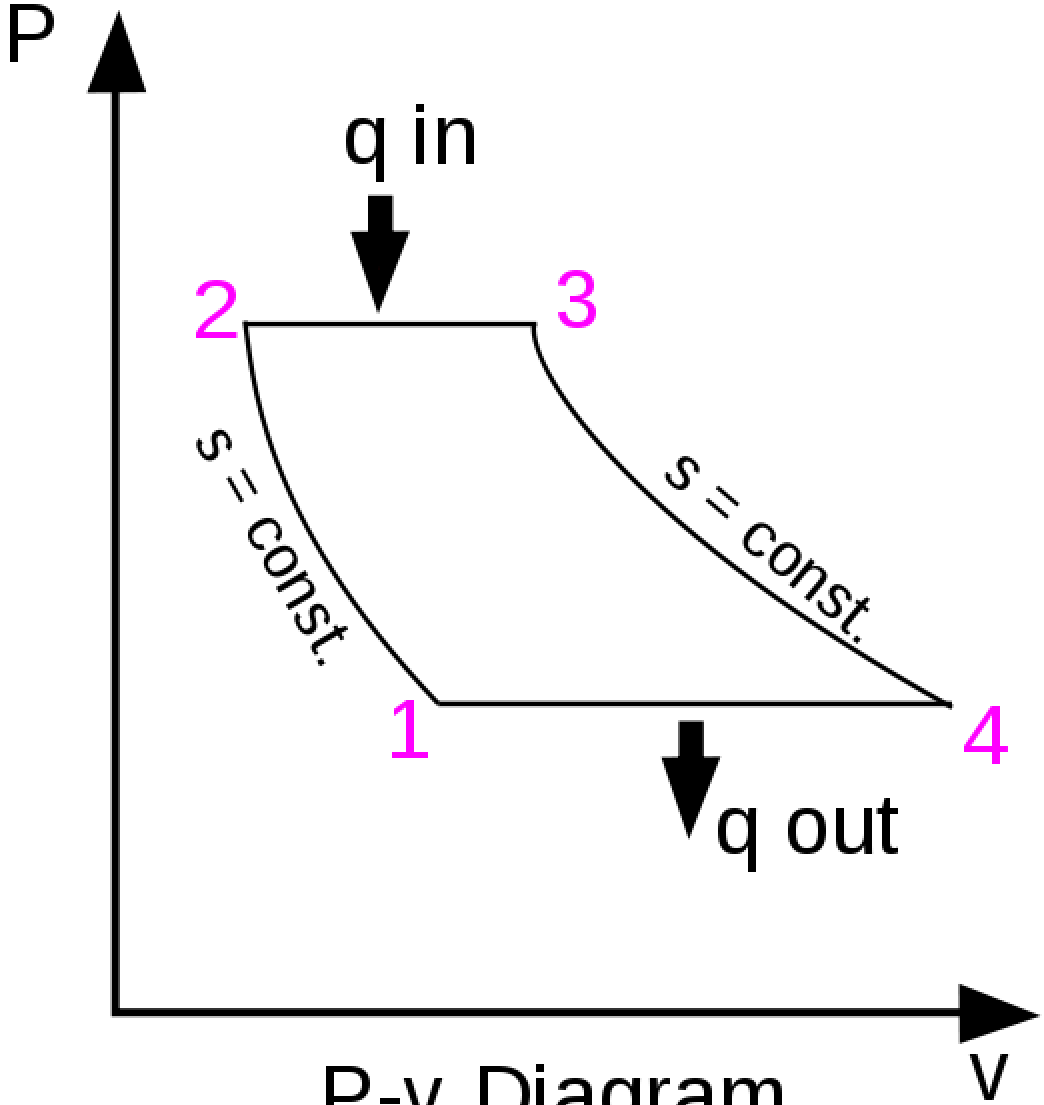
- Uses Atmosphere as the Heat Exchanger

Air Standard Assumptions

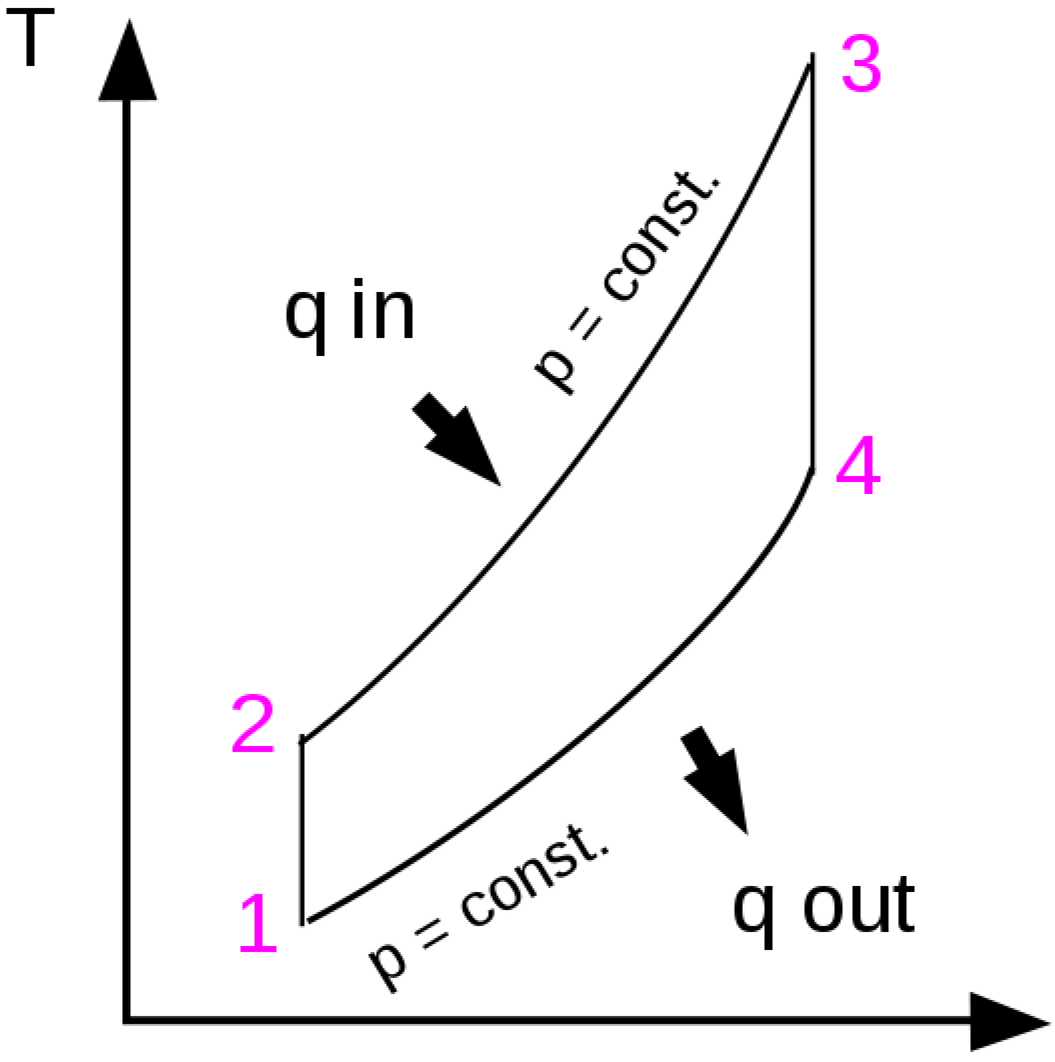
- Working fluid is air and acts as an ideal gas
- All Processes are internally reversible
- Combustion process is replaced by heat addition
- Exhaust process is replaced by heat rejection
- Cold ASA are that properties of air are constant at 25C

Ideal Brayton Cycle

- Isentropic Compression
- Isobaric Heat Addition
- Isentropic Expansion
- Isobaric Heat Rejection

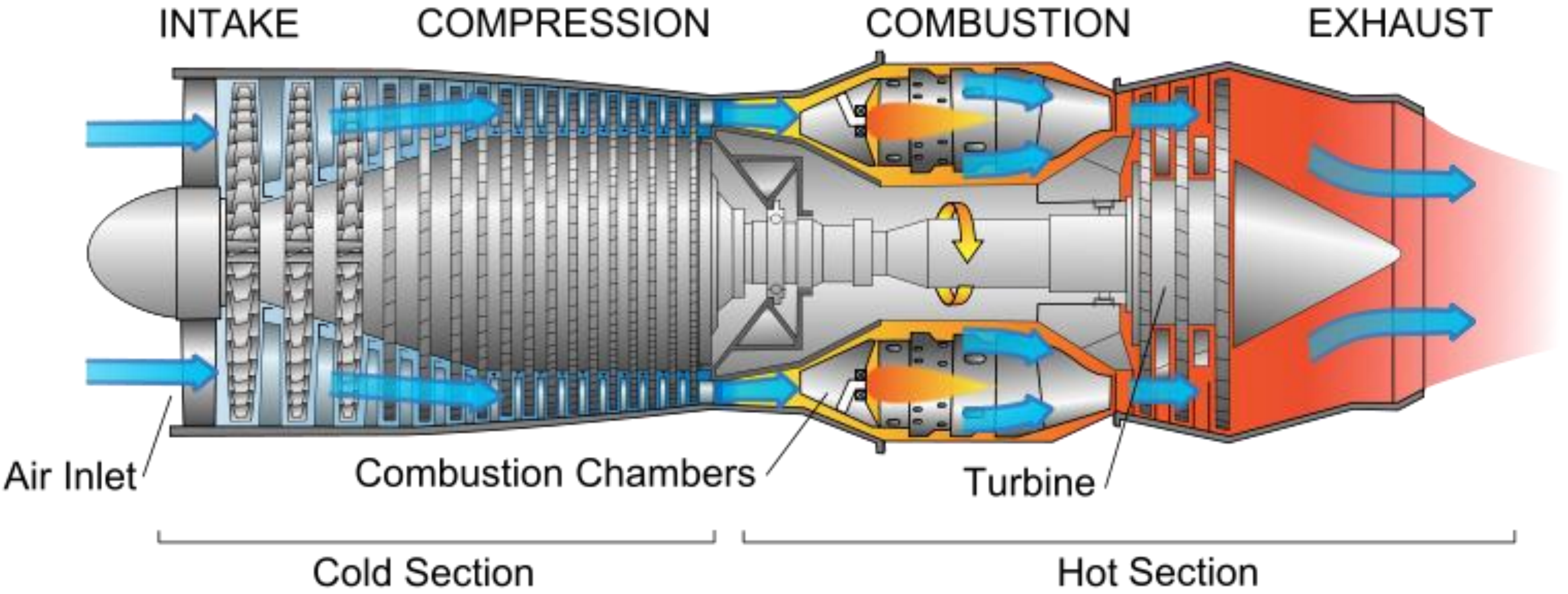


P-v Diagram



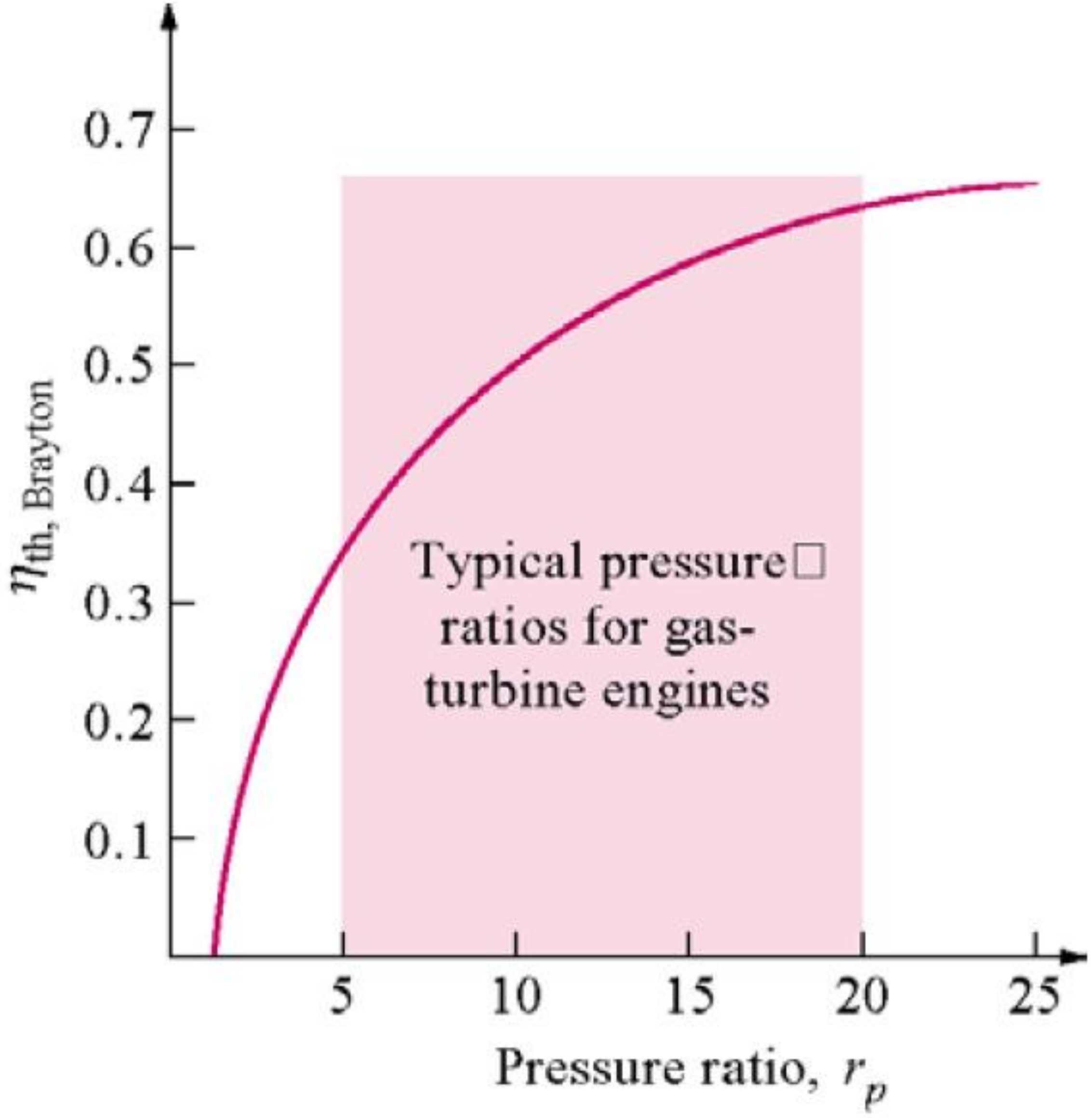
T-s Diagram

Gas Turbines



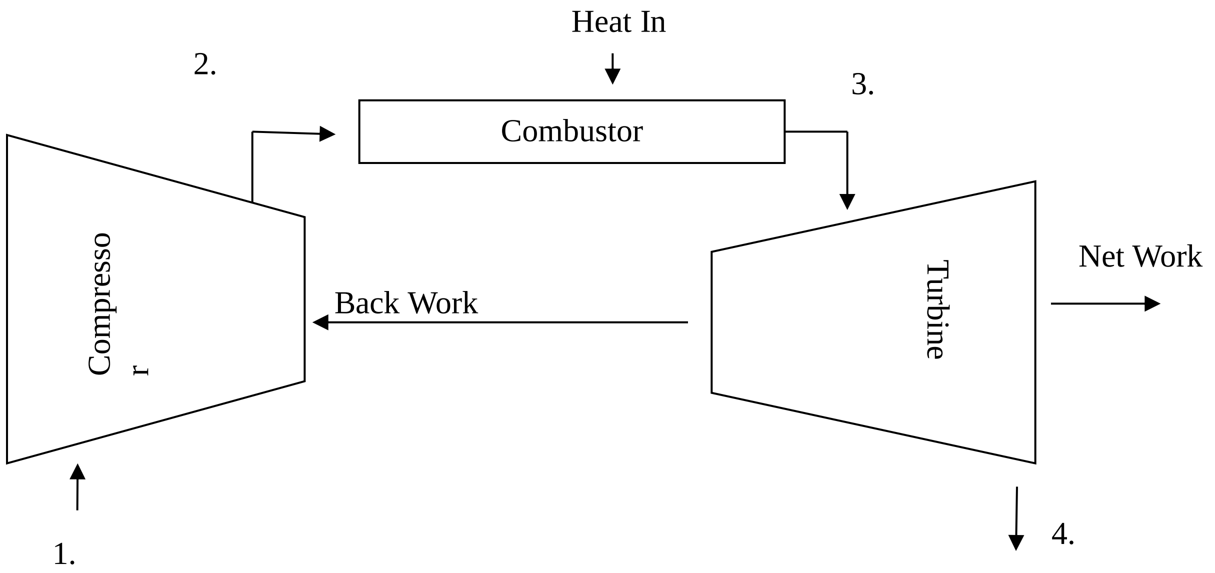
Can use direct thrust from momentum (mass flow rate x Velocity = Force) or can drive a shaft (helicopters, boats, turbo propeller aircraft).

$$\eta_{th,brayton} = 1 - \frac{1}{r_p^{\left(\frac{k-1}{k}\right)}}$$


0.7
0.6
0.5
0.4
0.3
0.2
0.1
$\eta_{\text{th, Brayton}}$
Typical pressure□ ratios for gas-turbine engines
5
10
15
20
25
Pressure ratio, r_p

A Run around the Brayton Cycle

The ideal air-standard Brayton cycle operates with air entering the compressor at 95 kPa, 22°C. The pressure ratio r_p is 6:1 and the air leaves the heat addition process at 1100 K. Determine the compressor work and the turbine work per unit mass flow, the cycle efficiency, the back work ratio, and compare the compressor exit temperature to the turbine exit temperature. Assume constant properties.



Example of Calculations (continued)

Isentropic Compressor Work

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

$$\begin{aligned} T_2 &= T_1 r_p^{(k-1)/k} \\ &= (22+273)K(6)^{(1.4-1)/1.4} \\ &= 492.5\,K \end{aligned}$$

$$\begin{aligned} w_{comp} &= C_p(T_2 - T_1) \\ &= 1.005\frac{kJ}{kg\cdot K}(492.5-295)K \\ &= 198.15\frac{kJ}{kg} \end{aligned}$$

Isentropic Turbine Work

$$\frac{T_4}{T_3} = \left(\frac{1}{r_p}\right)^{(k-1)/k}$$

$$\begin{aligned} T_4 &= T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} \\ &= 1100K\left(\frac{1}{6}\right)^{(1.4-1)/1.4} \\ &= 659.1\,K \end{aligned}$$

$$\begin{aligned} w_{turb} &= C_p(T_3 - T_4) = 1.005\frac{kJ}{kg\cdot K}(1100-659.1)K \\ &= 442.5\frac{kJ}{kg} \end{aligned}$$

Example of Calculations (continued)

Heat In

$$q_{in} = \frac{\dot{Q}_{in}}{\dot{m}} = h_3 - h_2$$

$$= C_p(T_3 - T_2) = 1.005\frac{kJ}{kg \cdot K}(1100 - 492.5)K$$

$$= 609.6\frac{kJ}{kg}$$

Work Out

$$w_{net} = w_{turb} - w_{comp}$$

$$= (442.5 - 198.15)\frac{kJ}{kg}$$

$$= 244.3\frac{kJ}{kg}$$

Example of Calculations (continued)

$$\eta_{th,\ Brayton} = \frac{w_{net}}{q_{in}}$$

$$= \frac{244.3\,\frac{kJ}{kg}}{609.6\,\frac{kJ}{kg}} = 0.40 \quad or \quad \boxed{40\%}$$

$$BWR = \frac{w_{in}}{w_{out}} = \frac{w_{comp}}{w_{turb}}$$

$$= \frac{198.15\,\frac{kJ}{kg}}{442.5\,\frac{kJ}{kg}} = \boxed{0.448}$$

Turbine exit temperature (659K) is significantly hotter than compressor exit (492K) !

Entropy

- Major Irreversibilities
  - Friction
  - Unrestrained Expansion
  - Heat Transfer

Total entropy always increases

Isentropic Efficiency

- Referenced to Isentropic Cases
- **Always** less than Unity.

$$\eta_{turbine} = \frac{W_{out_{actual}}}{W_{out_{isentropic}}} = \frac{W_a}{W_s}$$

$$\eta_{compressor} = \frac{W_{in_{isentropic}}}{W_{in_{actual}}} = \frac{W_s}{W_a}$$

Hot Exhaust (even without afterburners)

Hot Exhaust brings the advent of Tricycle Landing Gear

Note how field catches on fire

Early Model Taildragger ME 262

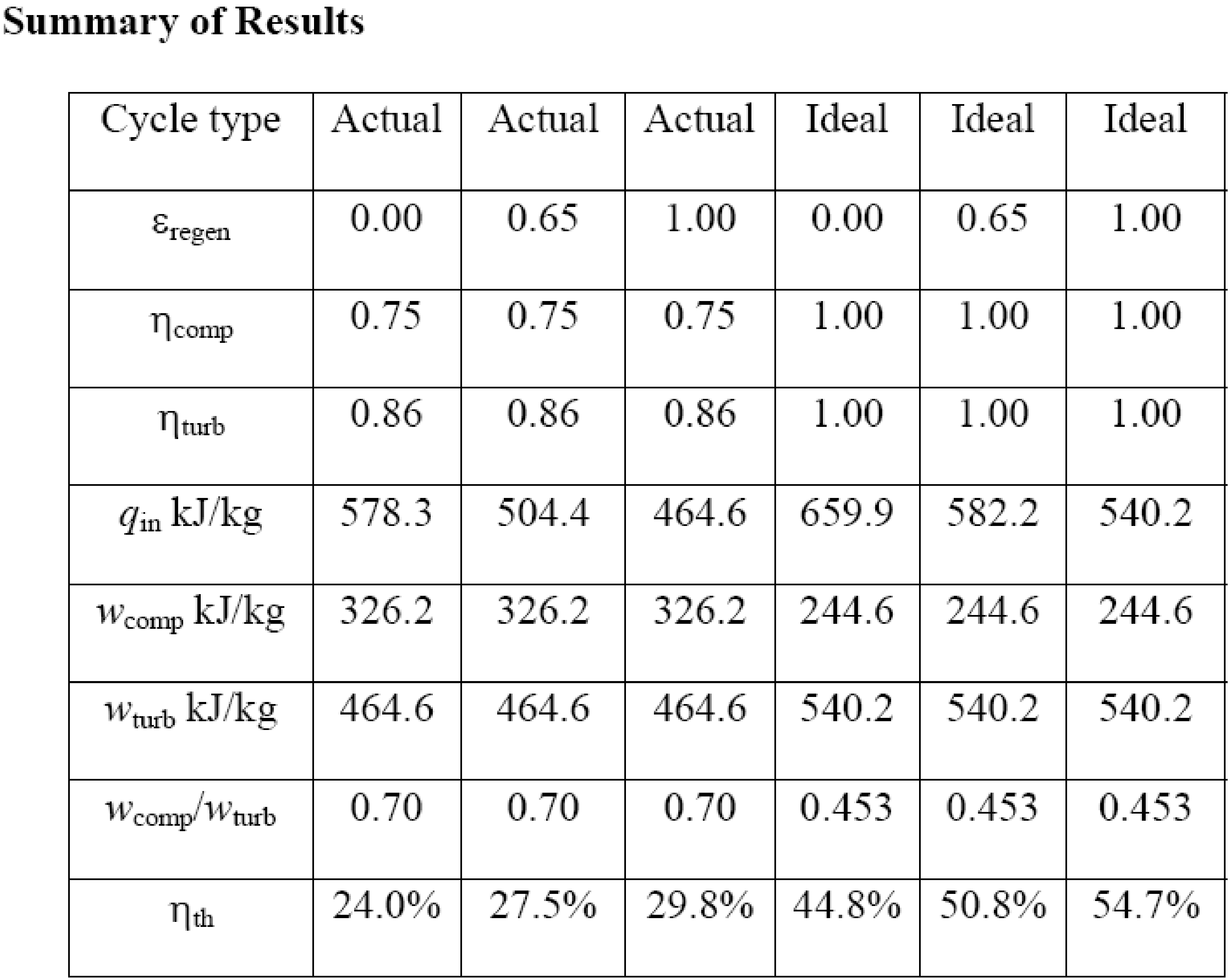
Later Model ME 262 w/ tricycle gear

Recuperation

- If temperature at exit of turbine is higher than the temperature at exit of compressor one can use a recuperator to preheat air before the combustion process.
- Adds weight but increases efficiency.
- Closed heat exchanger required (the two streams are at different pressures).

$$\varepsilon_{regen} = \frac{q_{regen,\,act}}{q_{regen,\,\max}}$$

For 8:1 pressure ratio, Compressor entrance at 300K 1 bar, turbine entrance at 1200K (ideal and non ideal cases with and without regeneration)



Summary of Results

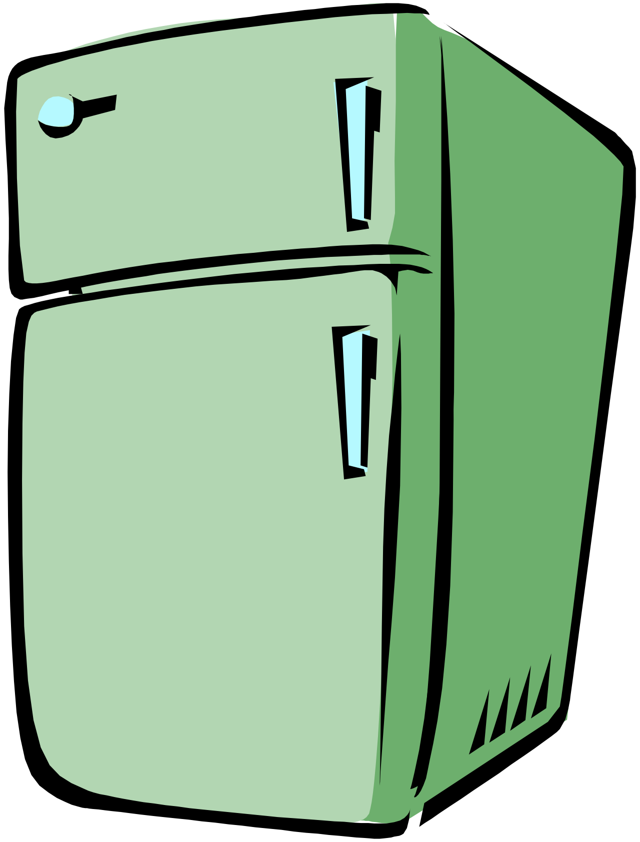
Cycle type	Actual	Actual	Actual	Ideal	Ideal	Ideal
ε_{regen}	0.00	0.65	1.00	0.00	0.65	1.00
η_{comp}	0.75	0.75	0.75	1.00	1.00	1.00
η_{turb}	0.86	0.86	0.86	1.00	1.00	1.00
q_{in} kJ/kg	578.3	504.4	464.6	659.9	582.2	540.2
w_{comp} kJ/kg	326.2	326.2	326.2	244.6	244.6	244.6
w_{turb} kJ/kg	464.6	464.6	464.6	540.2	540.2	540.2
w_{comp}/w_{turb}	0.70	0.70	0.70	0.453	0.453	0.453
η_{th}	24.0%	27.5%	29.8%	44.8%	50.8%	54.7%

Gas Turbine Engines
Brayton Cycle

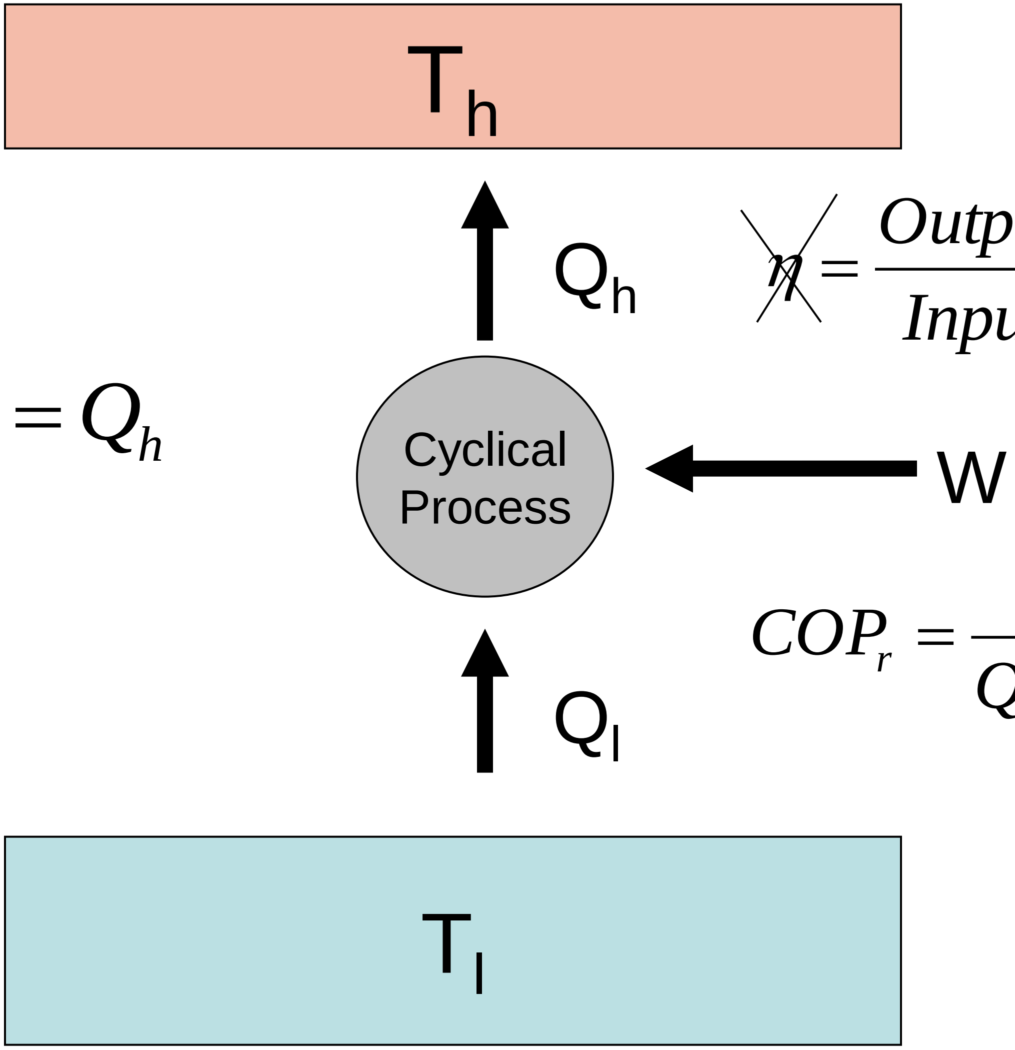
- High back work
- Need thermal recuperation for high efficiency
- High Power Density
- Smooth
- Slow Dynamic Response (turbo lag)
- Relatively Low Torque
- Scalability??

Vapor compression Refrigeration Cycle

- Can you imagine life without refrigeration?



Refrigeration Cycle

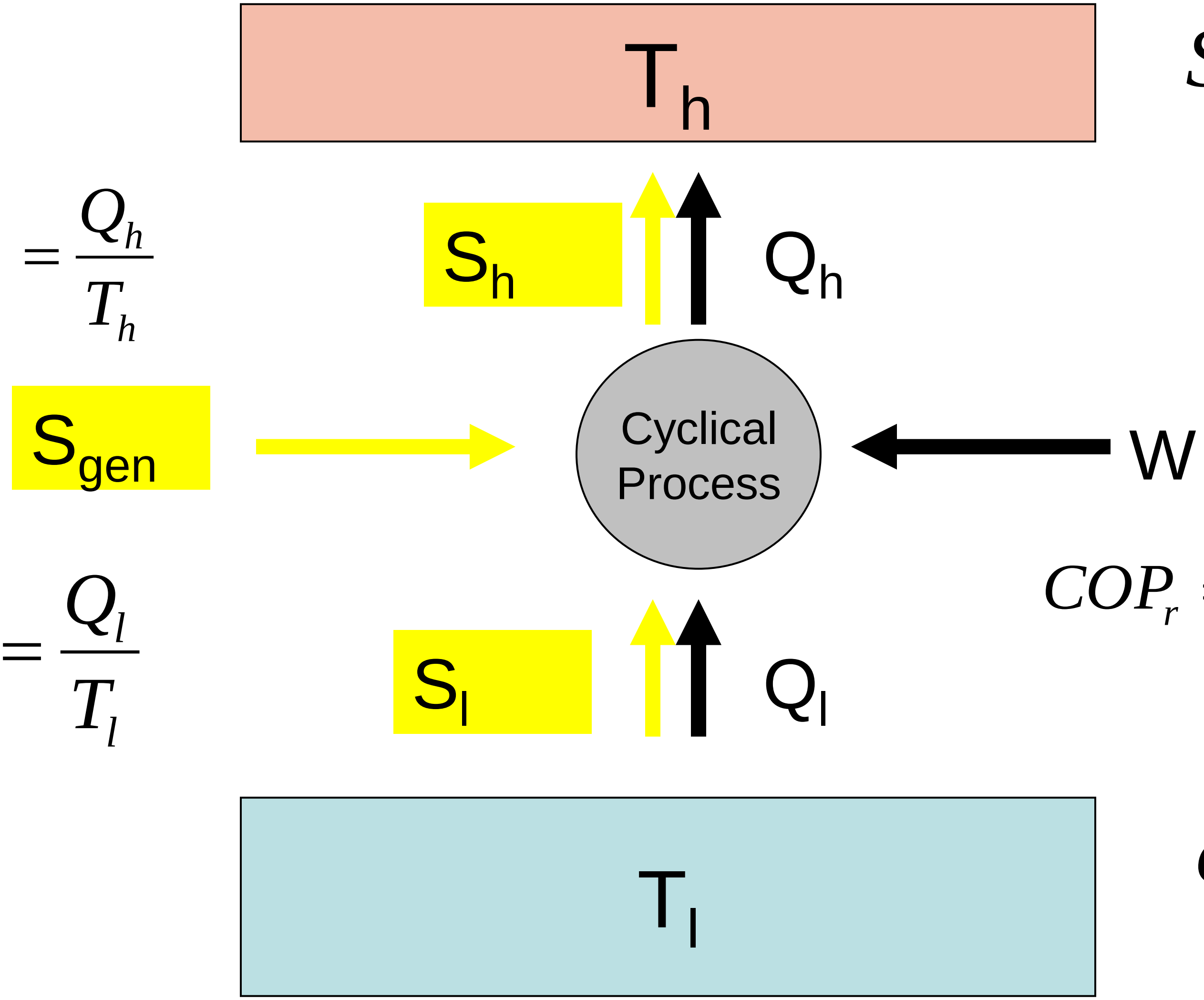


$$\cancel{\eta} = \frac{Output}{Input} = \frac{Q_l}{W} = COP_r$$

$$Q_l + W = Q_h$$

$$COP_r = \frac{Q_l}{Q_h - Q_l} = \frac{1}{\frac{Q_h}{Q_l} - 1}$$

Carnot Refrigeration Cycle



$$S_l + \cancelto{0}{S_{gen}} = S_h$$

$$S_h = \frac{Q_h}{T_h}$$

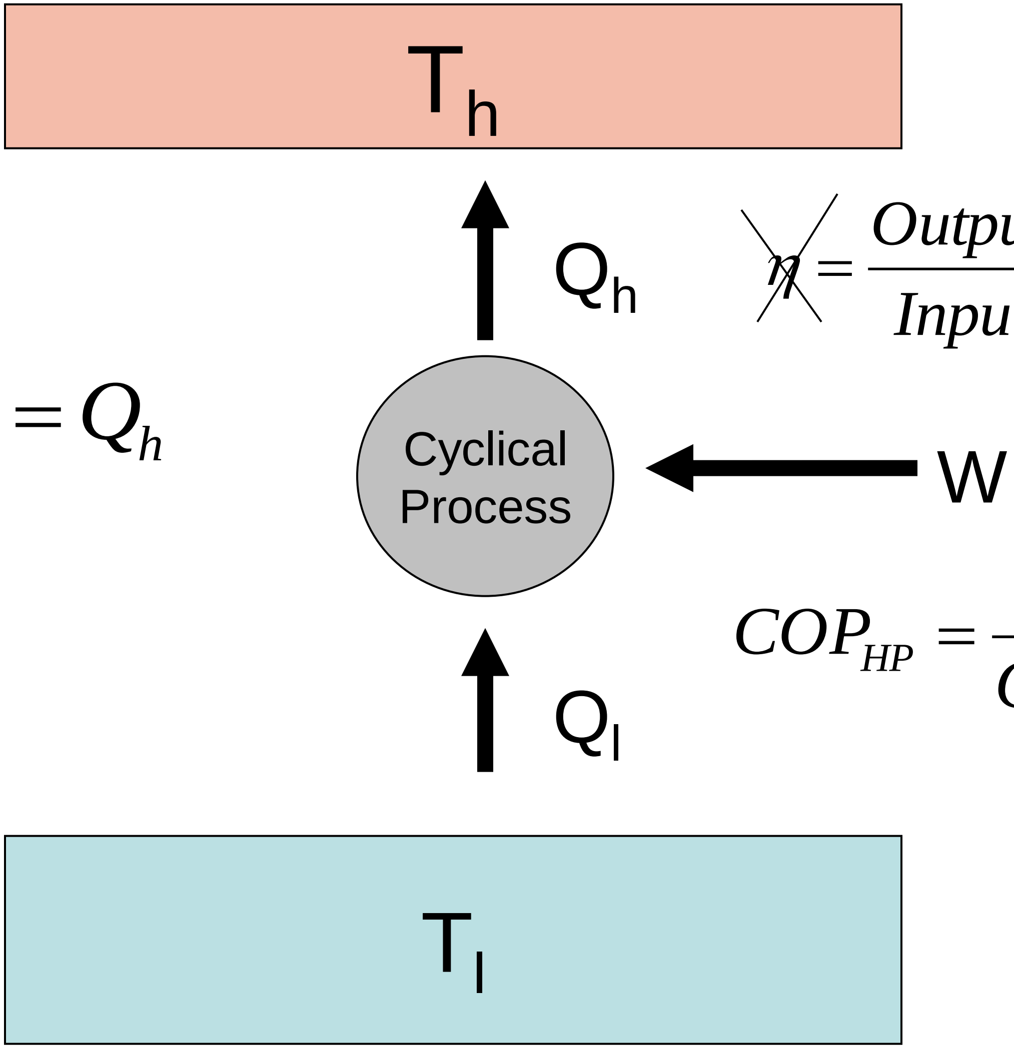
$$\frac{Q_l}{T_l} = \frac{Q_h}{T_h}$$

$$COP_r = \frac{1}{\frac{Q_h}{Q_l} - 1} = \frac{1}{\frac{T_h}{T_l} - 1}$$

$$S_l = \frac{Q_l}{T_l}$$

$$COP_{\max} = \frac{1}{\frac{T_h}{T_l} - 1}$$

Heat Pump Cycle

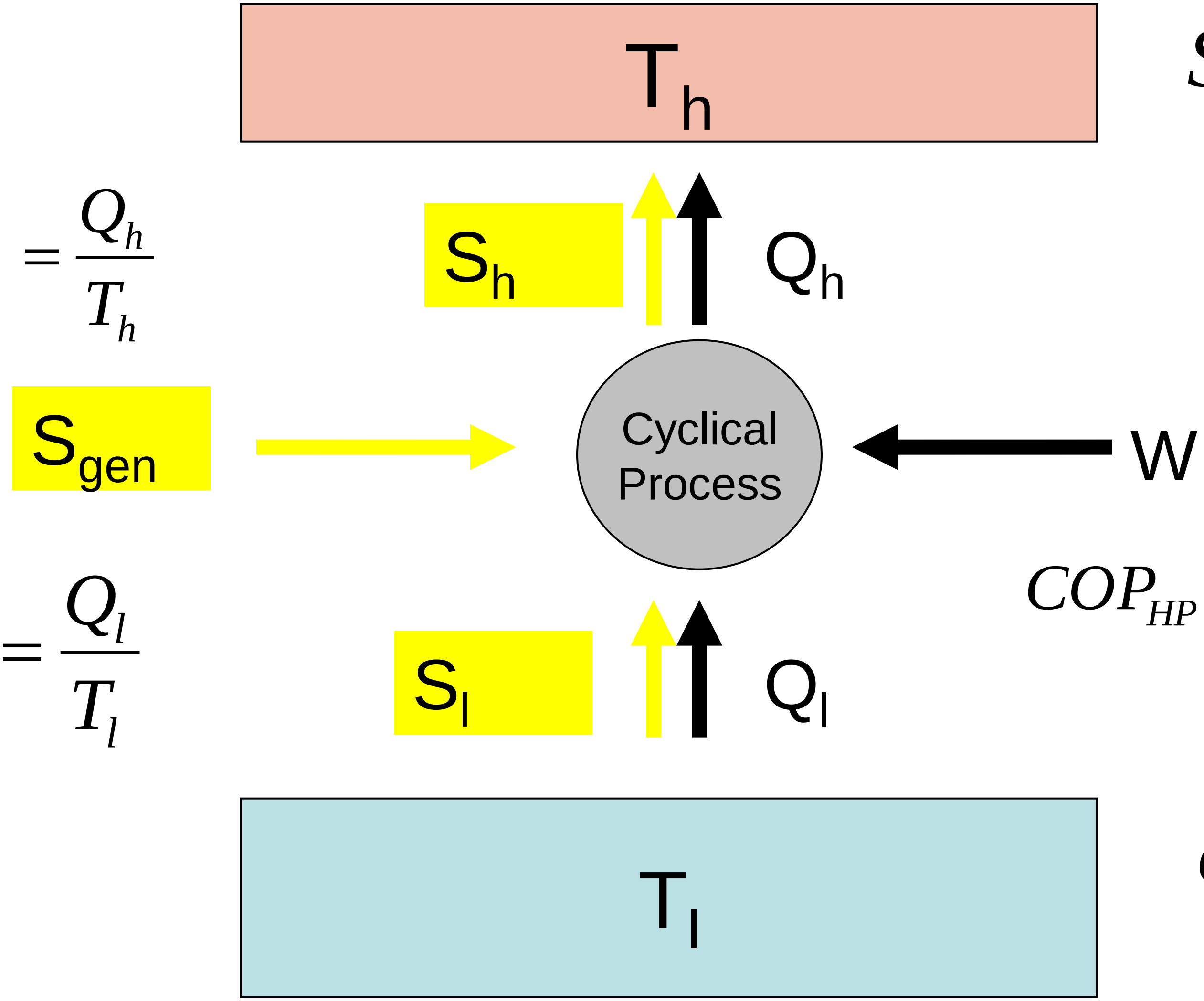


$$\cancel{\eta} = \frac{Output}{Input} = \frac{Q_h}{W} = COP_{HP}$$

$$Q_l + W = Q_h$$

$$COP_{HP} = \frac{Q_h}{Q_h - Q_l} = \frac{1}{1 - \frac{Q_l}{Q_h}}$$

Carnot Heat Pump Cycle



$$S_l + \cancelto{0}{S_{gen}} = S_h$$

$$S_h = \frac{Q_h}{T_h}$$

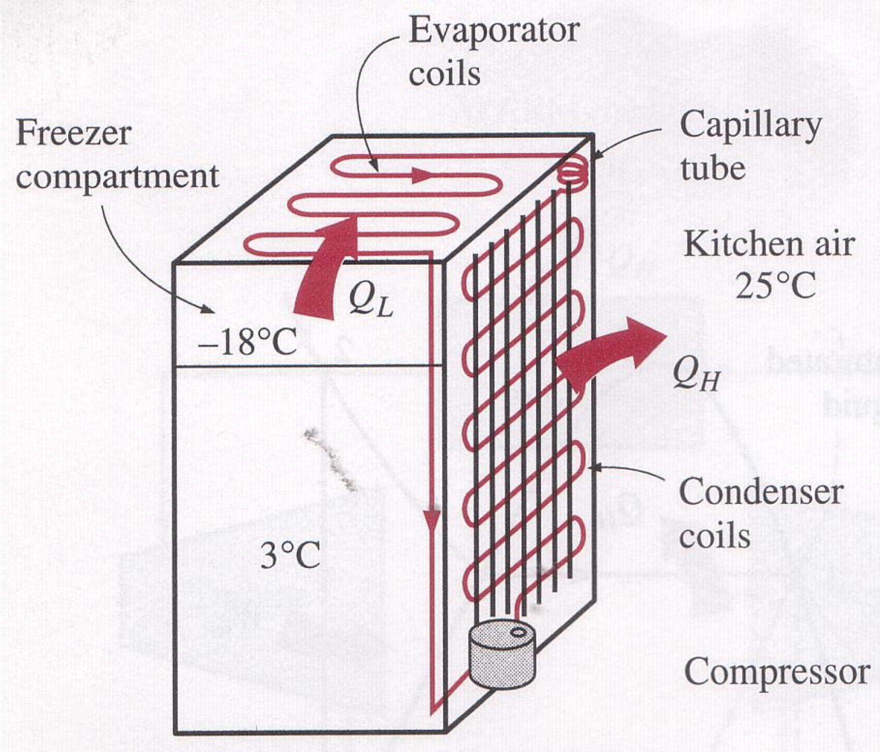
$$\frac{Q_l}{T_l} = \frac{Q_h}{T_h}$$

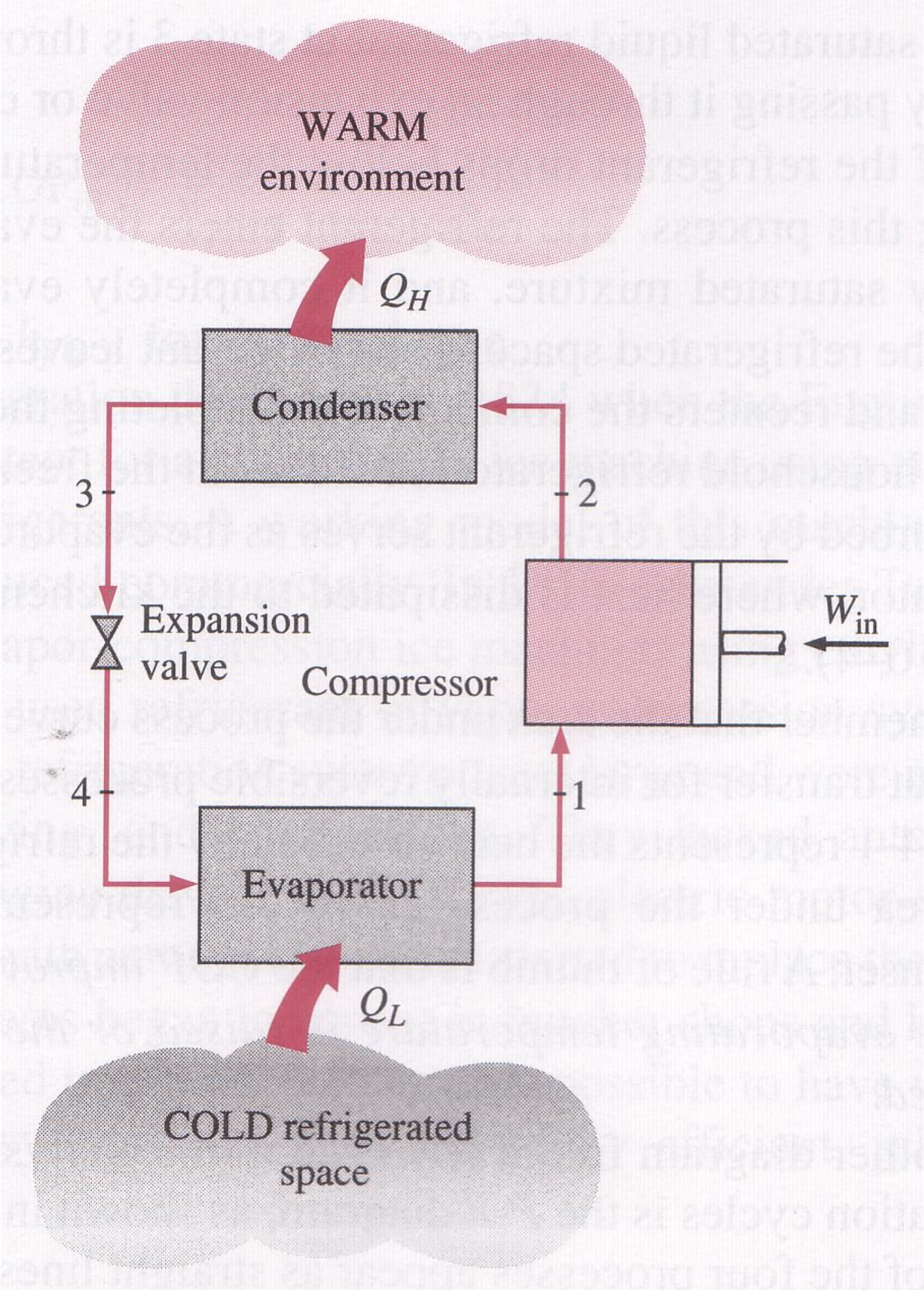
$$S_l = \frac{Q_l}{T_l}$$

$$COP_{HP} = \frac{1}{1 - \frac{Q_l}{Q_h}} = \frac{1}{1 - \frac{T_l}{T_h}}$$

$$COP_{\max} = \frac{1}{1 - \frac{T_l}{T_h}}$$

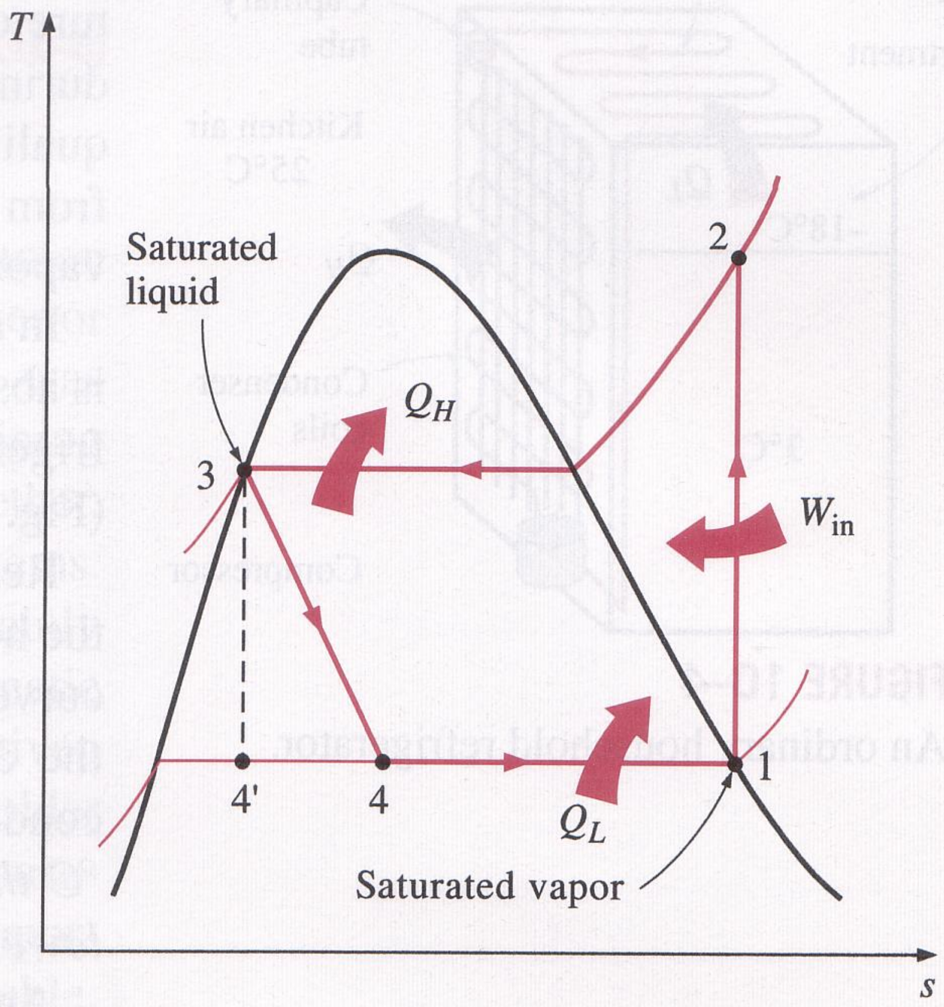
The System


Evaporator coils
Freezer compartment
Capillary tube
Kitchen air 25°C
Q_L
–18°C
Q_H
3°C
Condenser coils
Compressor


WARM environment
Q_H
Condenser
3
2
Expansion valve
Compressor
W_{in}
4
1
Evaporator
Q_L
COLD refrigerated space

Thermodynamics – T-S

- 1-2 Isentropic Compression
  - Compressor
- 2-3 Isobaric heat transfer
  - Condenser
- 3-4 Throttling process
  - Expansion valve
- 4-1 Isobaric heat transfer
  - Evaporator

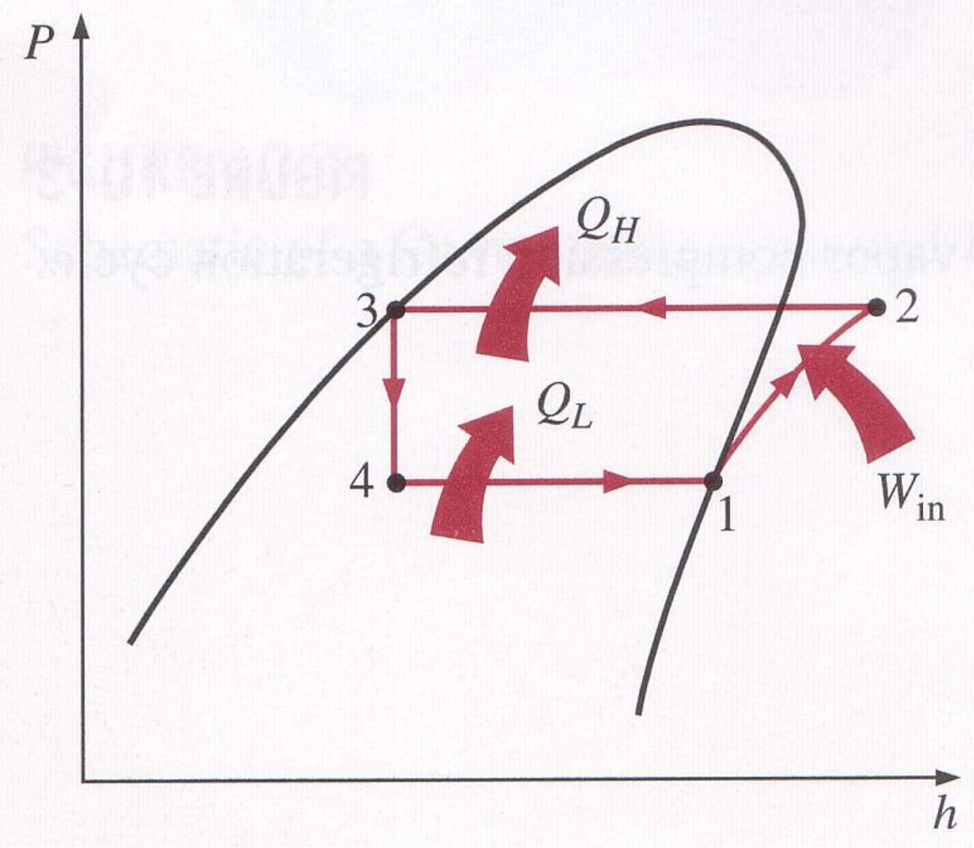


Thermodynamics – P-*h*

- Pressure - Enthalpy

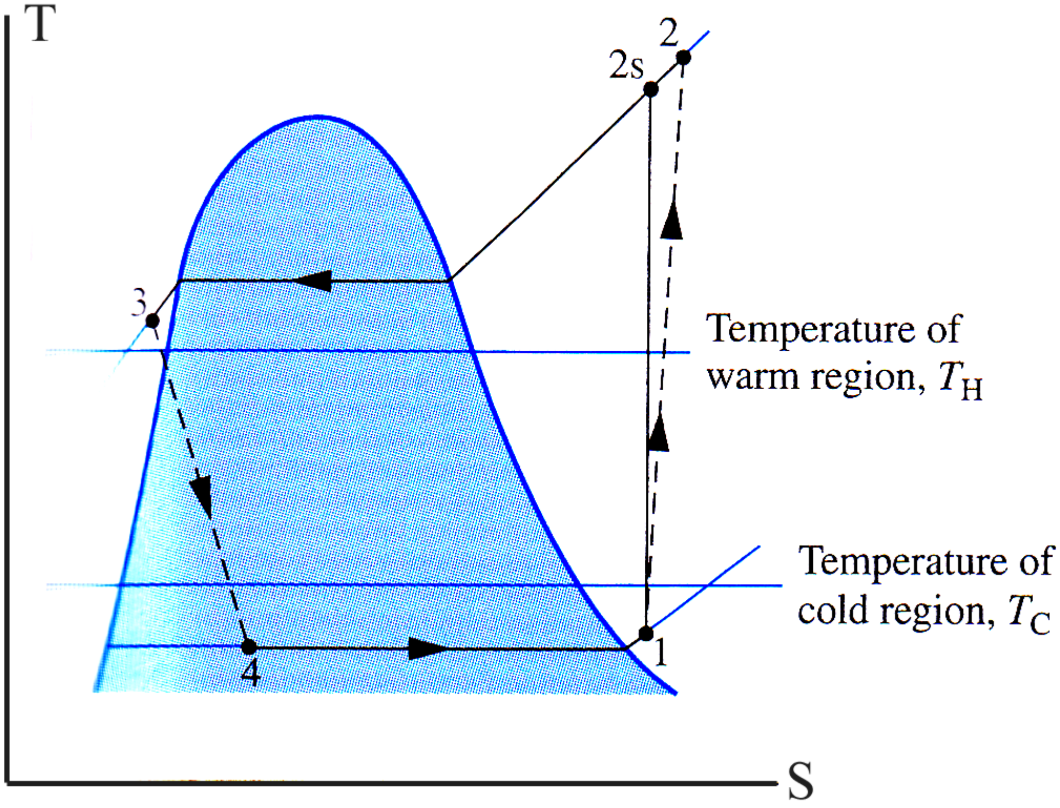
- 1 – Saturated vapor
- 2 – Superheated vapor
- 3 – Saturated liquid
- 4 – Vapor + liquid

- Will vary for each refrigerant gas



Actual Process – T-S

- Non-Isentropic compression
- Superheated gas at 1
- Subcooled liquid at 3
- Ambient temperatures
  - $T_H < T_{CONDENSER}$
  - $T_C > T_{EVAPORATOR}$
  - Necessary for heat transfer



You roughly need a Delta T of 10 C for efficient Heat Transfer

Performance

$$COP_R = \frac{q_L}{w_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$q = \frac{\dot{Q}}{\dot{m}}, \quad w = \frac{\dot{W}}{\dot{m}}$$

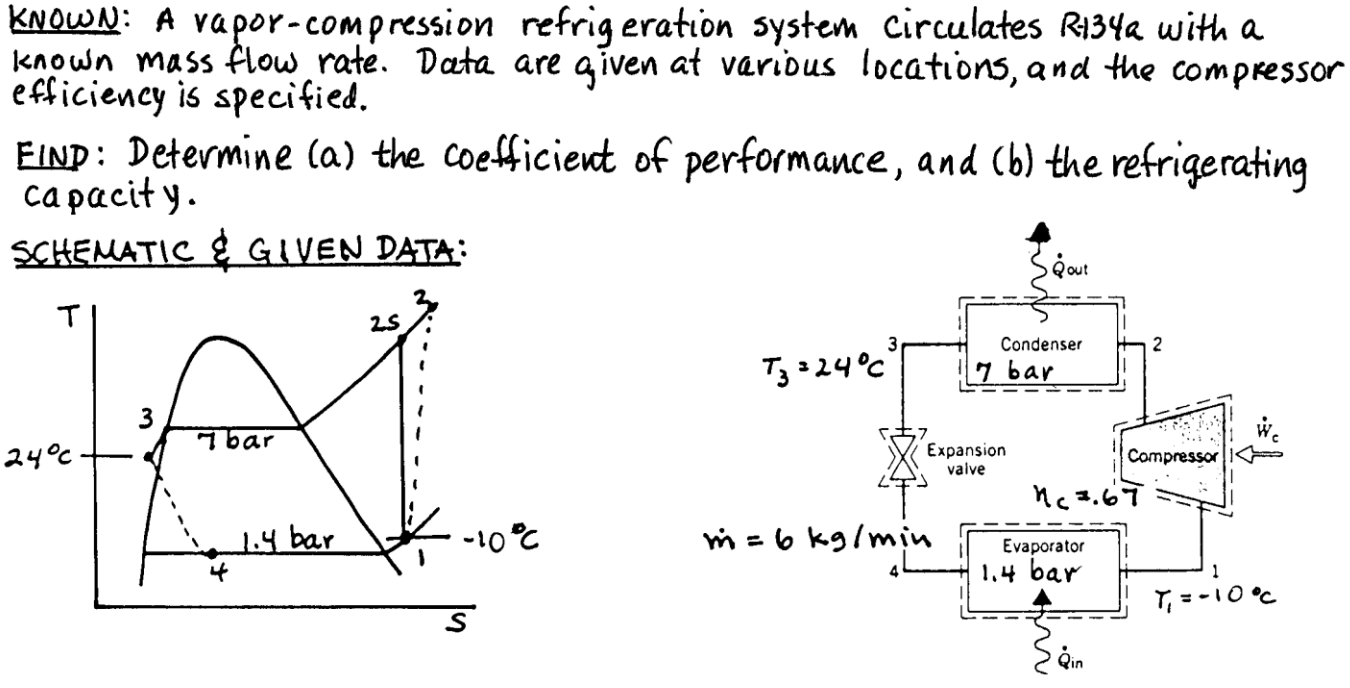
- Heat WILL flow from cold to hot, if work is done
- More heat flows than work is done
- Typical $COP_R \approx 4$

A Run around the Refrigeration Cycle

A vapor-compression refrigeration system circulates Refrigerant 134a at a rate of 6 kg/min. The refrigerant enters the compressor at 10C, 1.4 bar, and exits at 7 bar. The isentropic compressor efficiency is 67%. There are no appreciable pressure drops as the refrigerant flows through the condenser and evaporator. The refrigerant leaves the condenser at 7 bar, 24C. Ignoring heat transfer between the compressor and its surroundings, determine

(a) the coefficient of performance.

(b) the refrigerating capacity, in tons.



KNOWN: A vapor-compression refrigeration system circulates R134a with a known mass flow rate. Data are given at various locations, and the compressor efficiency is specified.

FIND: Determine (a) the coefficient of performance, and (b) the refrigerating capacity.

SCHEMATIC & GIVEN DATA:

A Run around the Refrigeration Cycle

<u>ANALYSIS</u>: First, fix each of the principal states.

<u>State 1</u> $T_1 = -10°C$, $p_1 = 1.4$ bar $\Rightarrow h_1 = 243.40$ kJ/kg, $s_1 = .9606$ kJ/kg·K

<u>State 2</u> For isentropic compression, $p_2 = 7$ bar, $s_{2s} = s_1 \Rightarrow h_{2s} = 278.06 \ \frac{\text{kJ}}{\text{kg}}$

Using the compressor efficiency

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} \Rightarrow h_2 = h_1 + \left(\frac{h_{2s} - h_1}{\eta_c}\right) = 295.13 \ \text{kJ|kg}$$

$$s_2 = 1.0135 \ \text{kJ/kg·K}$$

<u>State 3</u> $p_3 = 7$ bar, $T = 24°C \Rightarrow h_3 \approx h_{f@24} = 82.90$ kJ/kg, $s_3 \approx .3113$ kJ/kg·K

<u>State 4</u> Throttling process $\Rightarrow h_4 = h_3 = 82.90$ kJ/kg, $s_4 = .33011$ kJ/kg·K

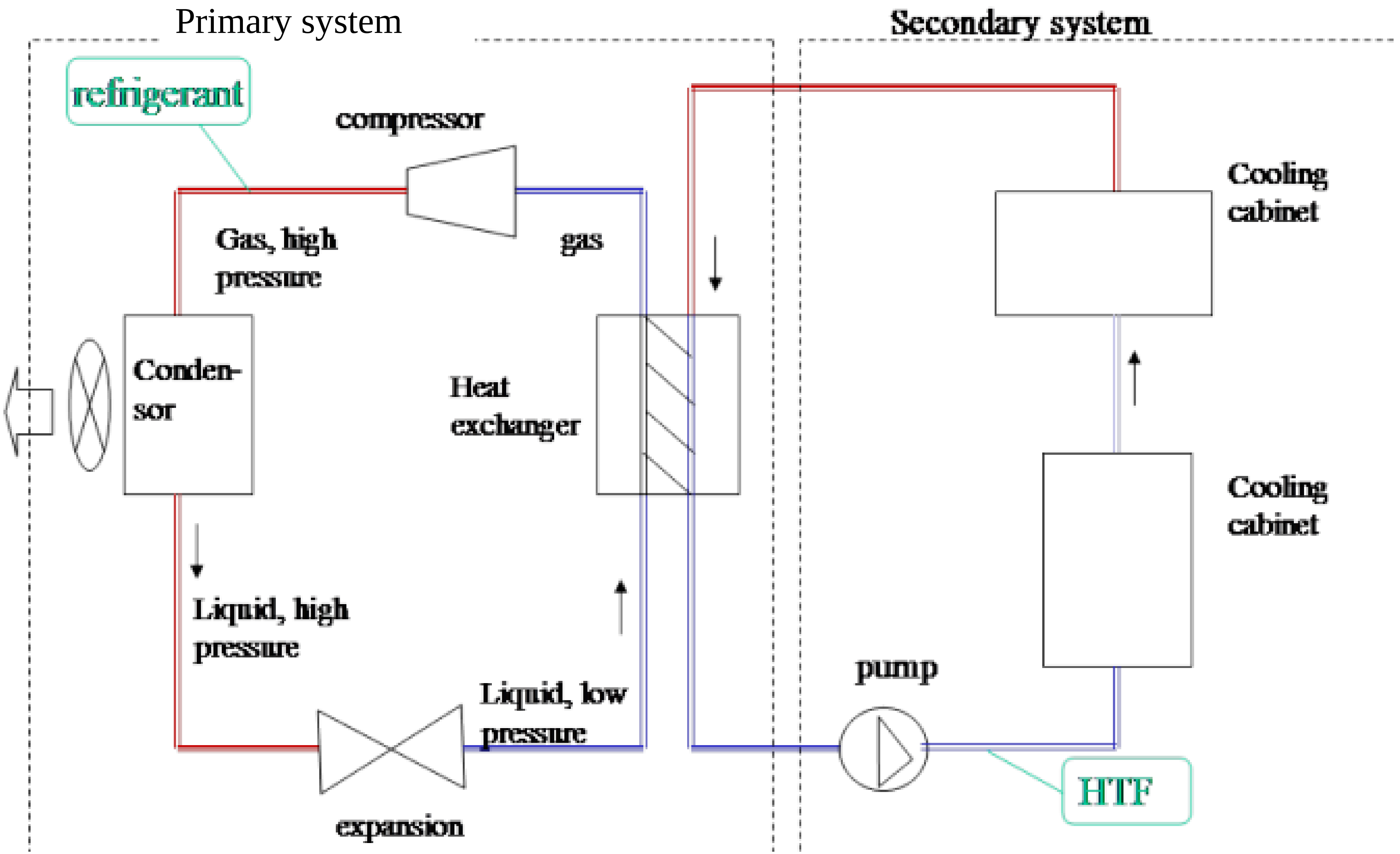
(a) The coefficient of performance is

$$\beta = \frac{h_1 - h_4}{h_2 - h_1} = 3.10 \longleftarrow \beta$$

(b) The refrigerating capacity is

$$\dot{Q}_{in} = \dot{m}(h_1 - h_4)$$

$$= \left(6 \ \frac{\text{kg}}{\text{min}}\right)(243.40 - 82.90) \ \frac{\text{kJ}}{\text{kg}} \left|\frac{1 \ \text{ton}}{211 \ \text{kJ/min}}\right| = 4.564 \ \text{tons} \longleftarrow \dot{Q}_{in}$$


Primary system
Secondary system
refrigerant
compressor
Gas, high pressure
gas
Conden-sor
Heat exchanger
Liquid, high pressure
Liquid, low pressure
expansion
Cooling cabinet
Cooling cabinet
pump
HTF

Desirable Characteristics of Refrigerants

- High H_{fg}
- Low Cost
- Non-toxic
- Non-Corrosive
- Liquid at STP
- Reasonable pressures and temperatures for application
- Environmentally “friendly” GHG, Ozone, etc

R-11

R-22

R-134a

R-410

Ammonia

Typical Secondary Refrigeration Fluids (or heat transfer fluids)

- Ethylene glycol
- Propylene glycol (In your sports drink)
- Water

- High heat capacity
- Low viscosity (pumping losses)
- Non-toxic
- Cheap

What can be done to improve system

- Decrease Entropy Generation (Friction, Unrestrained Expansion,
- Decrease Heat Gain

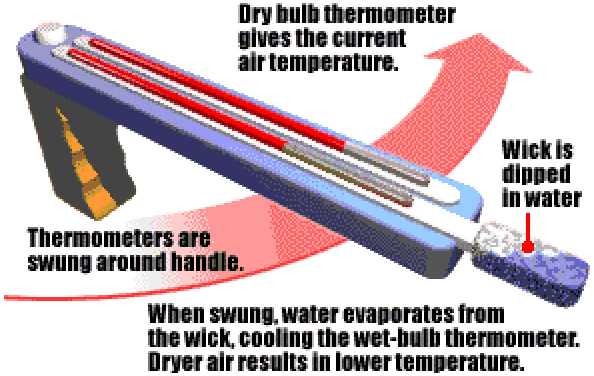
- Remove/Reduce Transient Loads

Air / Water Mixtures

- Several tools
  - Wet / dry bulb thermometer
  - Psychrometric chart
  - Hygrometer
- Allow calculation of:
  - Humidity
  - Enthalpy
  - Saturation temperature (dew point)
  - Density

Wet / dry bulb thermometer

- Two thermometers
  - Wet bulb – bulb is surrounded by moistened wick
  - Dry bulb – exposed “dry” bulb”
  - Moisture content of air determines evaporation rate - cooling effect
  - 100% relative humidity – wet=dry
  - Low humidity = wet <<dry



Humidity Measures

- Absolute Humidity:

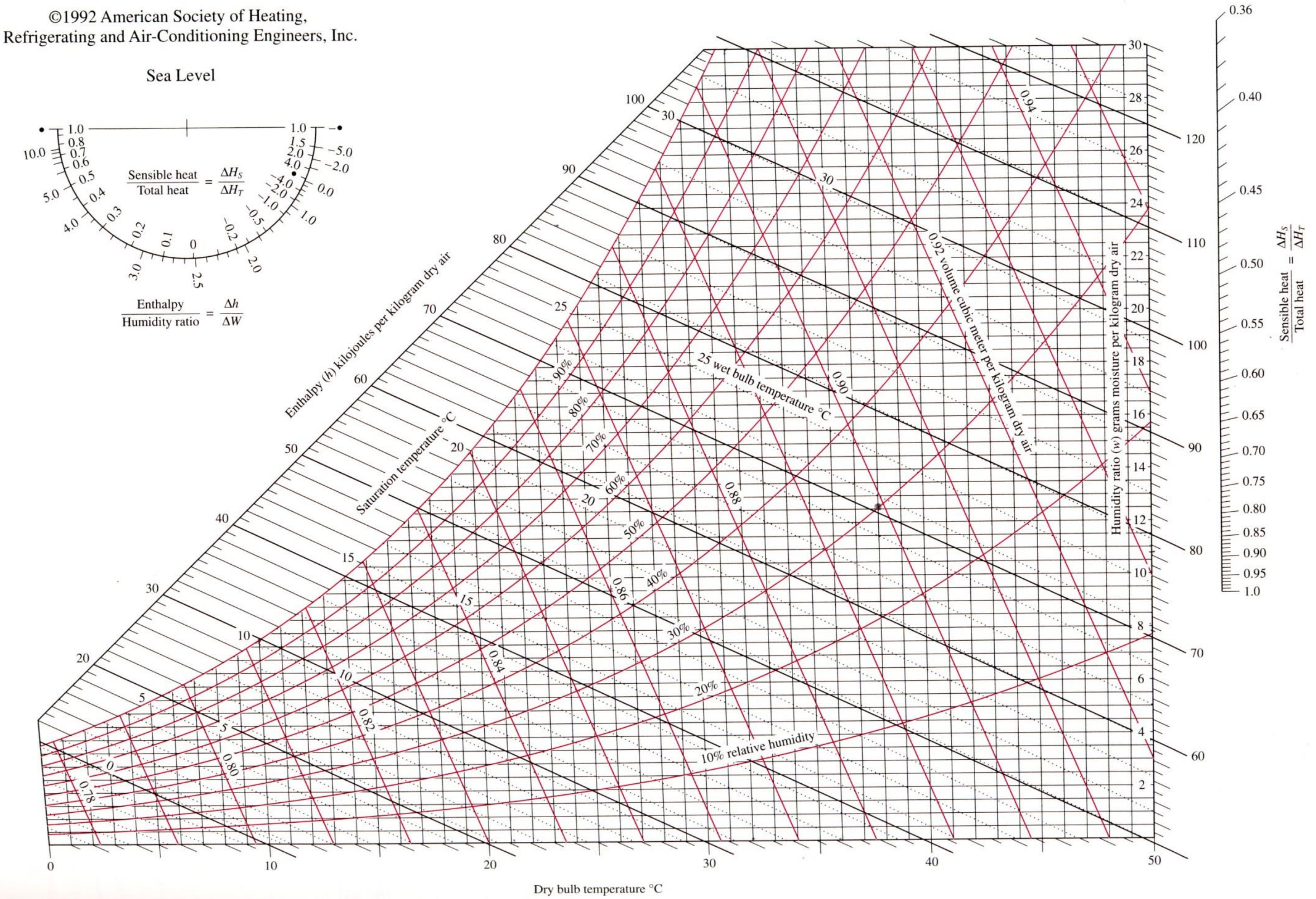
$$\gamma = \frac{m_{H_2O\,vapour}}{m_{air}} = \frac{mass\,of\ water\,vapor}{mass\,of\ dry\,air}$$

- Relative Humidity:

$$\phi = \frac{p_{H_2O\,vapour}}{p_{saturation}} = \frac{partial\ pressure\,of\ H_2O\,vapor}{saturation\ pressure\,at\,mixture\,temperature}$$

ASHRAE Psychrometric Chart No. 1
Normal Temperature
Barometric Pressure: 101.325 kPa

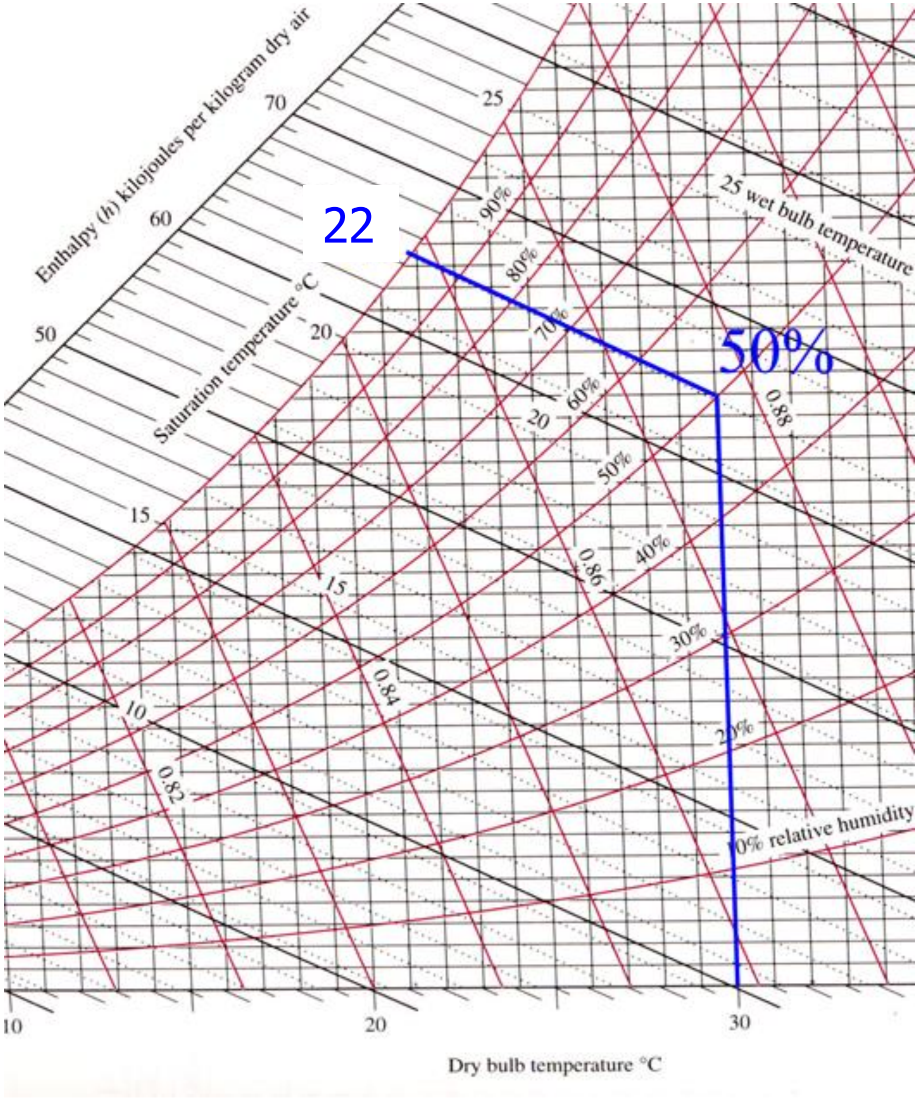
Sea Level



Calculating Humidity

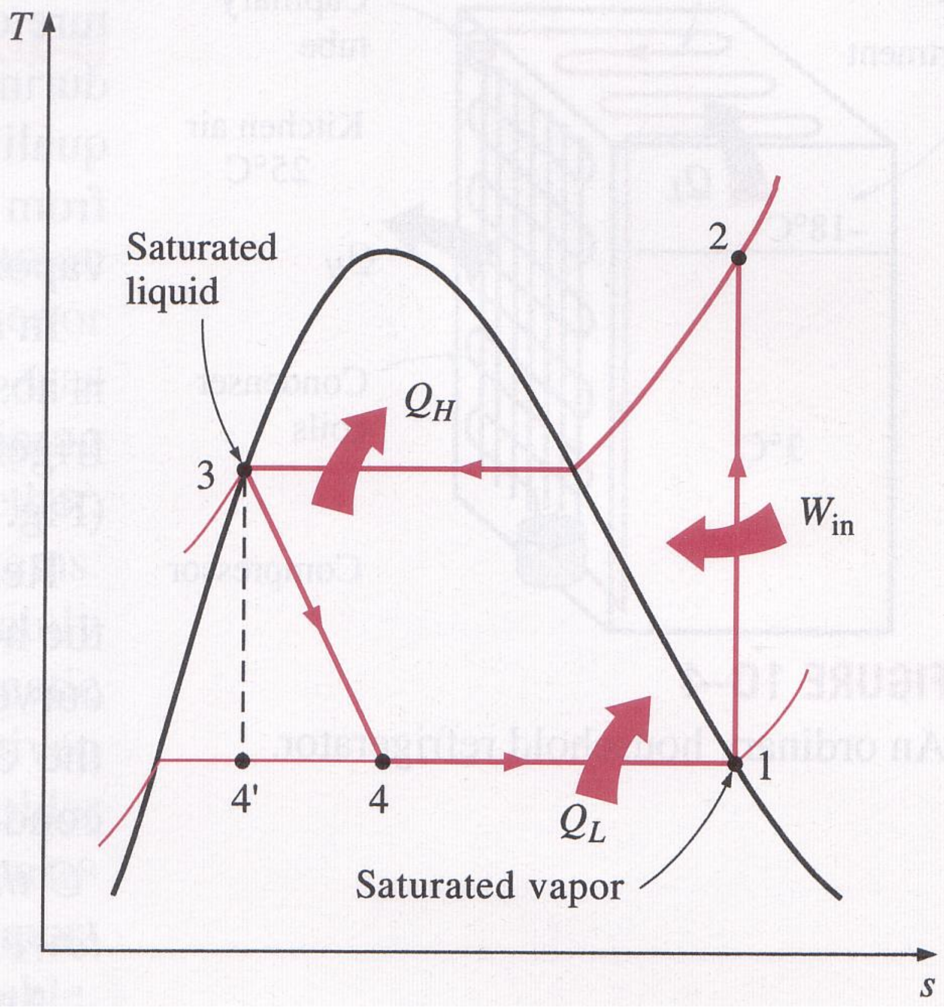
- Dry bulb = 30° C
- Wet bulb = 22° C

- Humidity = 50%
- Dew Point = 18.5° C

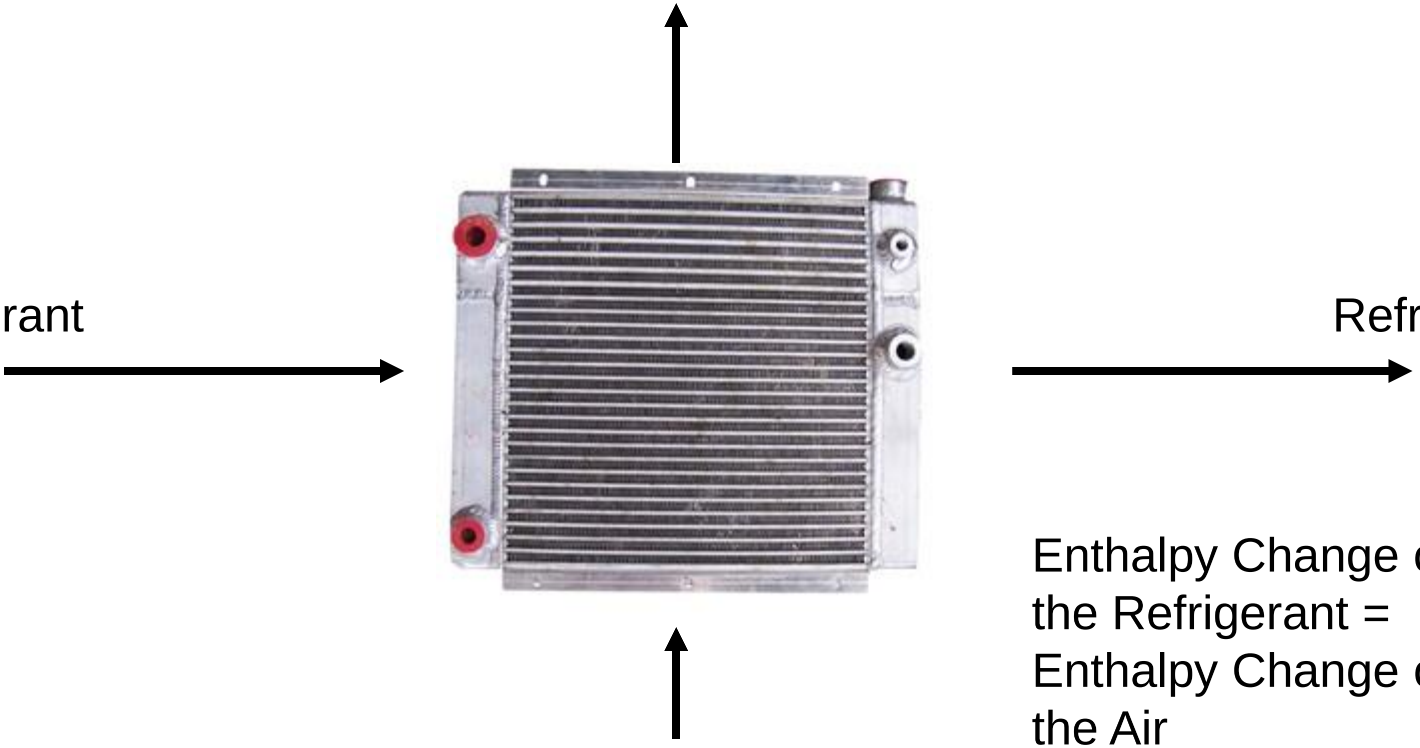


AC Thermodynamics

- 1-2 Isentropic Compression
  - Compressor
- 2-3 Isobaric heat transfer
  - Condenser
- 3-4 Throttling process
  - Expansion valve
- 4-1 Isobaric heat transfer
  - Evaporator



Evaporator Systems



Air Out

Refrigerant In

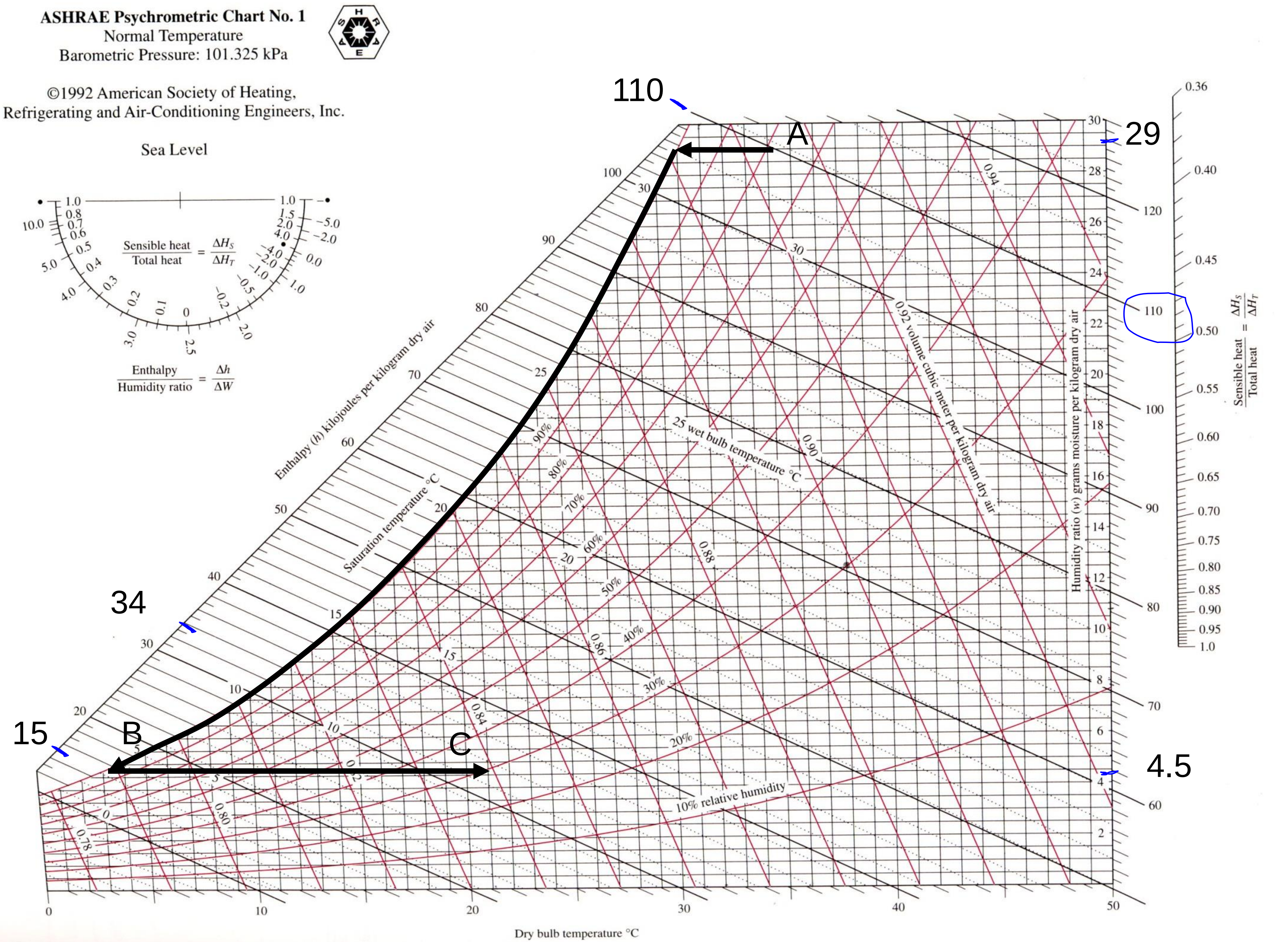
Refrigerant out

Air In

Enthalpy Change of the Refrigerant = Enthalpy Change of the Air

AC Systems

- Outside Air is 35˚C 80% relative humidity Comes across HTEX (AC system) exiting at 3˚C and is injected into a 21˚C Compartment
  - If Outside air is passed at 4 kg/min (assume dry air equivalence) and then exits the cabin what is the condensation removal required at the HTEX?
  - How big is the AC system in Tons?
  - What is the heat gain of the cabin?
  - If the AC system has a COP of 2.5 what is the power required in Watts?
  - What could be done to lower the power requirement and what is the minimum power requirement?


ASHRAE Psychrometric Chart No. 1
Normal Temperature
Barometric Pressure: 101.325 kPa
©1992 American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.
Sea Level
Sensible heat / Total heat = ΔHS / ΔHT
Enthalpy / Humidity ratio = Δh / ΔW
Enthalpy (h) kilojoules per kilogram dry air
Saturation temperature °C
25 wet bulb temperature °C
0.92 volume cubic meter per kilogram dry air
Humidity ratio (w) grams moisture per kilogram dry air
10% relative humidity
20%
30%
40%
50%
60%
70%
80%
90%
Dry bulb temperature °C
0
10
20
30
40
50
110
A
29
34
15
B
C
4.5

Solutions

- Condensation Removal (Change in Absolute Humidity)

$$\dot{m}(\gamma_A - \gamma_B) = 4kg_{Dry\,Air} / \min\left(29g_{H_2O} / kg_{Dry\,Air} - 4.5g_{H_2O} / kg_{Dry\,Air}\right) = 98g_{H_2O} / \min$$

- Size of the Refrigeration System (Q_l in the proper unit 211kJ/min/ton)

$$\dot{m}(h_A - h_B) = 4kg_{Dry\,Air} / \min\left(110kJ / kg_{Dry\,Air} - 15kJ / kg_{Dry\,Air}\right) = 380kJ / \min \qquad \frac{380kJ / \min}{211kJ / \min / Ton} = 1.8Ton$$

- Heat Gain (how does the air heat up?) (enthalpy change from evaporator exit to compartment exhaust)

$$\dot{m}(h_c - h_B) = 4kg_{Dry\,Air} / \min\left(34kJ / kg_{Dry\,Air} - 15kJ / kg_{Dry\,Air}\right) = 76kJ / \min$$

- Power required (Q_l/COP in proper Unit)

$$\dot{W} = \frac{Q_l}{COP_r} = \frac{380kJ / \min}{2.5} = 152kJ / \min = 2.53kW$$

- Minimum Power (Why are we getting rid of relatively cool dry air? Recirculate!) (Ql becomes heat gain and Power=Q_l/COP in proper Unit) (Note that some fresh air is required so this is overly optimistic)

$$\dot{W} = \frac{Q_l}{COP_r} = \frac{76kJ / \min}{2.5} = 30.4kJ / \min = 0.507kW$$

ASHRAE 1975 minimum value of 5 cubic feet per minute (CFM) of outdoor air per person in building design.