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Theoretical Calculation of the Z-process

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Symbols

Roman letters:

c_p	Specific isobaric heat capacitance, J/kgK
c_v	Specific isochoric heat capacitance, J/kgK
k	Polytropic constant
P	Power, W
p	Pressure, Pa
L_0	Stoichiometric air/fuel ratio
M	Molar mass, kg/kmol
m	Mass, kg
n	Engine speed, 1/s
Q	Energy, J
q_l	Lower heat value of liquid fuel
q_m	Mass flow, kg/s
R	Universal gas constant, kJ/kgK
r	Crack shaft radius
S	Piston stroke
T	Temperature, K
V	Volume, m^3
v	Specific volume, m^3/kg
W	Work, J
w	Mass fraction of species, (-)
x	Molar fraction of species, (-)

Creek letters:

η	Efficiency
κ	Adiabatic constant
λ	Air / fuel equivalence ratio

Subscripts:

air	Air
ad	Adiabatic
exh	Exhaust gas
f	Fuel
m	Mean, mass
res	Residual gas
sca	Scavenging
st	Stoichiometric
tr	Transport

1 Introduction

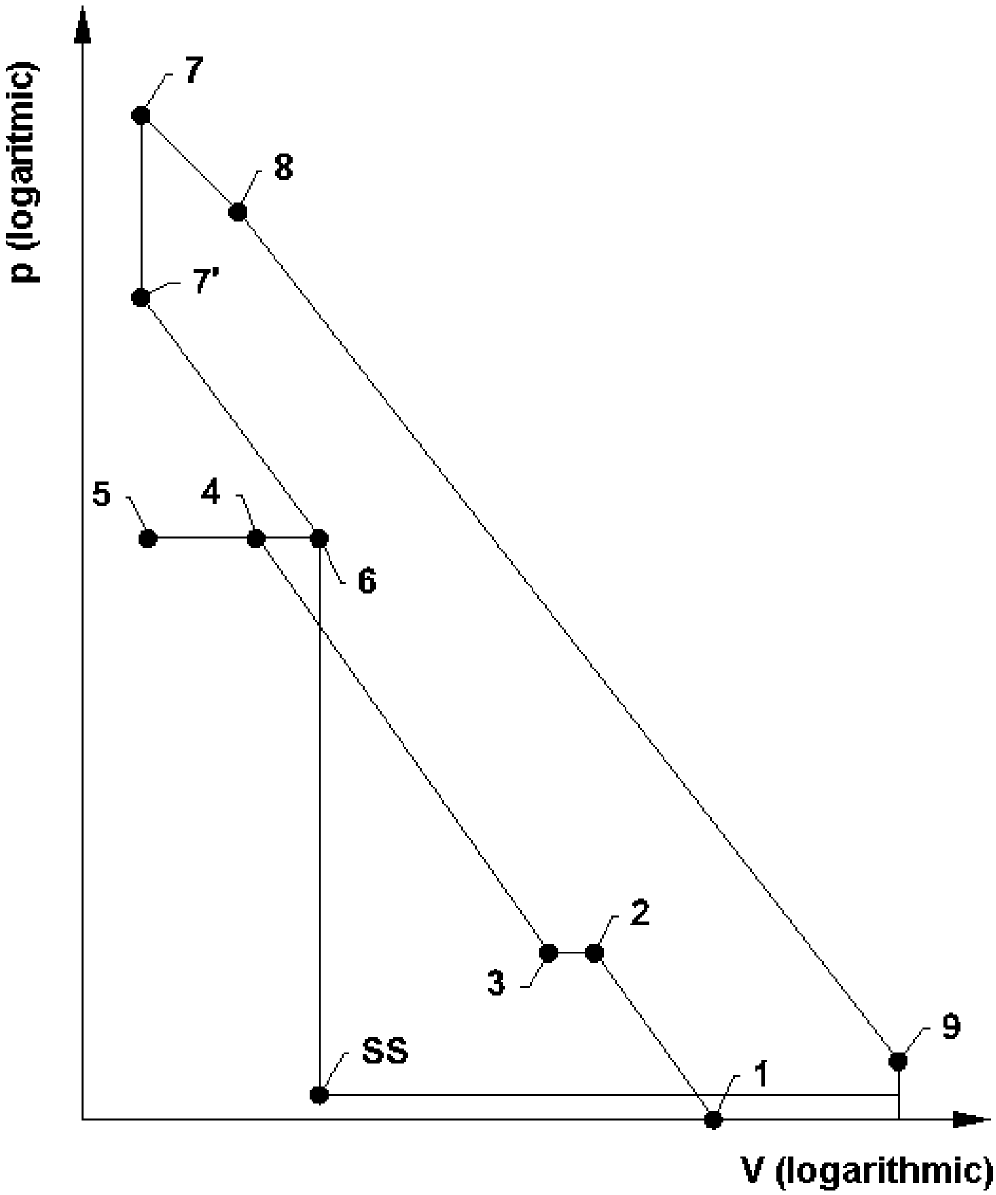
The purpose of this article is to introduce a simple theoretical calculation method for the Z-process. The calculation method is meant to be comparable to other theoretical processes like the theoretical Diesel process. However, the change of material properties and the chance of mass during cycle are considered in calculation.

A limited temperature -type combustion is used in the calculations. Also other types of combustion could be used in the Z-process.

For simplicity, a mechanical pre compressor is used in the calculations in this article. Use of an impulse turbo charger would lead to better efficiency.

2 Theoretical Calculation

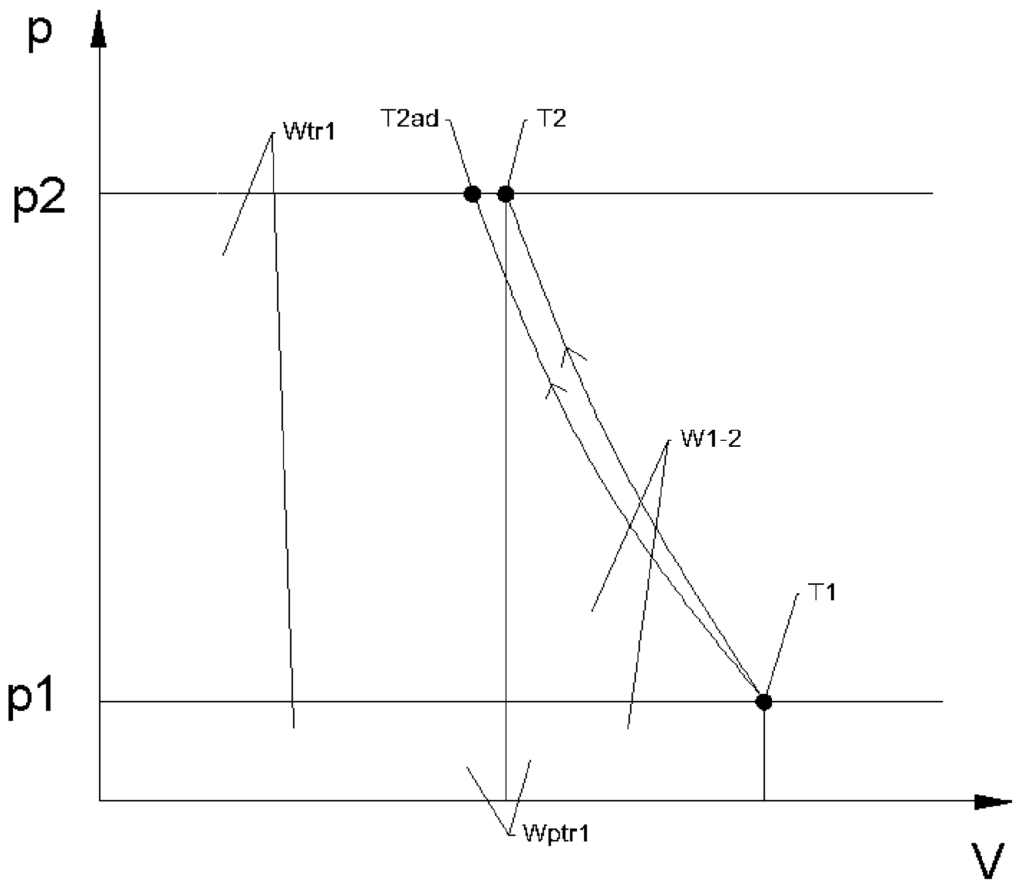
The Z-process is far more complex to calculate than common theoretical Diesel- or Otto process. The Z-process is explained in picture 1.



Picture 1

Compression in pre compressor occurs between points 1 and 2. First stage intercooler cools the fluid from point 2 to 3. Adiabatic compression occurs from 3 to 4 in the piston compressor. The fluid is cooled from 4 to 5 in the second stage intercooler. During scavenging, the intake air mixes with residual gas (SS) in the cylinder. The gas status after the scavenging is defined in the point 6. Adiabatic compression occurs in the working cylinder from point 6 to 7'. Isochoric combustion is done in the cylinder from point 7' to 7. Isothermal expansion is done in the cylinder from 7 to 8. Adiabatic expansion occurs from point 8 to 9.

2.1 Compressor Work



Picture 2

For adiabatic compression, we have

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\kappa} = \varepsilon^{\kappa} \tag{1}$$

Where

$\kappa \equiv$ Adiabatic constant

The temperature after compression is

$$T_2 = T_1 \varepsilon^{\kappa-1} = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}} \tag{2}$$

For adiabatic work, we have

$$W_{1-2} = \frac{mR}{(\kappa-1)M_{air}}(T_1 - T_2) \tag{3}$$

Where

m ≡ mass of the compressed fluid [= kg]

R ≡ universal gas constant [= kJ/kgK]

$M_{air} \equiv$ molar mass of air [= kg/kmol]

Adiabatic efficiency is usually announced for Lysholm type screw compressors and kinetic compressors. The definition for the adiabatic efficiency:

$$\eta_{ad} = \frac{T_{2ad} - T_1}{T_2 - T_1} \tag{4}$$

Where

$T_1 \equiv$ initial temperature [=K]

$T_2 \equiv$ final compression temperature [=K]

$T_{2ad} \equiv$ corresponding adiabatic temperature [=K]

The compression temperature can be calculated by substituting equation 2 to equation 4

$$T_2 = T_1 + \frac{T_{2ad} - T_1}{\eta_{ad}} = T_1 + \frac{T_1 \varepsilon^{\kappa-1} - T_1}{\eta_{ad}} = T_1 \frac{\left(\frac{p_2}{p_1}\right)^{\kappa-1} - 1 + \eta_{ad}}{\eta_{ad}} \tag{5}$$

To calculate the compression work, the polytropic constant (k) is defined. Equation for polytropic compression temperature

$$T_2 = T_1 \varepsilon^{k-1} = T_1 \frac{\varepsilon^{\kappa-1} - 1 + \eta_{ad}}{\eta_{ad}} \qquad (6)$$

Polytropic constant is solved from the equation of polytropic compression

$$\varepsilon^{k-1} = \frac{\varepsilon^{\kappa-1} - 1 + \eta_{ad}}{\eta_{ad}}$$

$$\Rightarrow k = 1 + \frac{\log\left(\frac{\varepsilon^{\kappa-1} - 1 + \eta_{ad}}{\eta_{ad}}\right)}{\log(\varepsilon)} = 1 + \frac{\log\left(\frac{\left(\frac{p_1}{p_0}\right)^{\frac{\kappa-1}{\kappa}} - 1 + \eta_{ad}}{\eta_{ad}}\right)}{\log\left(\left(\frac{p_2}{p_1}\right)^{\frac{1}{\kappa}}\right)} \qquad (7)$$

The polytropic compression work is

$$W_{1-2} = \frac{m_{air} R}{(k-1) M_{air}} (T_1 - T_2) \qquad (8)$$

Transport work of the compressor:

$$W_{tr1} = V_2 p_2 = m_{air} v_2 p_2 \qquad (9)$$

Where

m_{air} ≡ mass of intake air [= kg]

Specific volume is got from the ideal gas equation

$$v = \frac{V}{m} = \frac{RT}{Mp} \qquad (10)$$

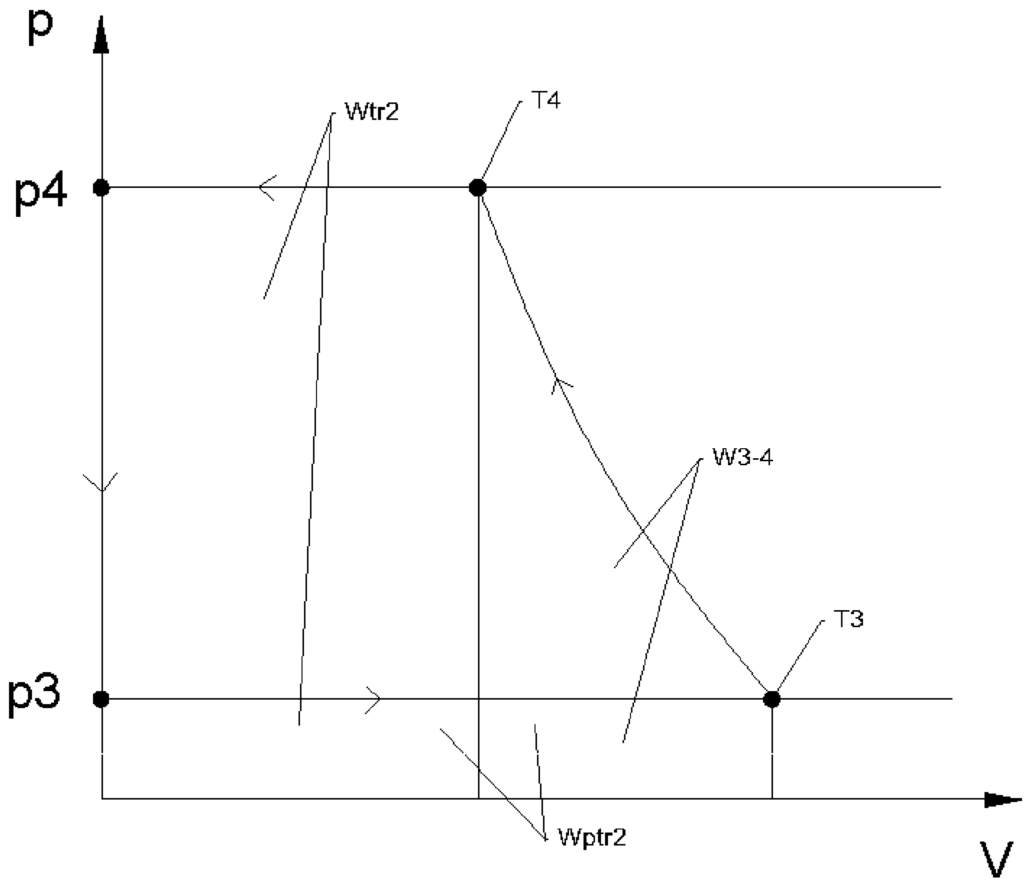
The pre compressor receives positive transport work from the environment

$$W_{ptr1} = V_1 p_1 = m_{air} v_1 p_1 \qquad (11)$$

The total work done by the pre compressor is

$$W_1 = W_{1-2} + W_{tr1} - W_{ptr1} \qquad (12)$$

2.2 Piston Compressor Work



Picture 3

If assumed adiabatic compression, the piston compressor does work (W_{3-4}) to the gas

$$W_{3-4} = \frac{mR}{(\kappa - 1)M_{air}}(T_4 - T_3) \tag{13}$$

For adiabatic compression, the compression temperature is

$$T_4 = T_3 \varepsilon_1^{\kappa-1} = T_3 \left(\frac{p_4}{p_3} \right)^{\frac{\kappa-1}{\kappa}} \tag{14}$$

Transport work of the compressor is defined same way as for the pre compressor

$$W_{tr2} = V_4 p_4 \tag{15}$$

By substituting the specific volume equation (10), we get

$$W_{tr2} = m_{air} \frac{RT_4}{M_{air}} \tag{16}$$

The piston compressor receives positive transport work made by pre compressor

$$W_{ptr2} = m_{air} v_3 p_3 \tag{17}$$

By substituting the specific volume equation to the positive transport work equation, it follows that

$$W_{ptr2} = m_{air} \frac{RT_3}{M_{air}} \tag{18}$$

For the total piston compressor work, we have finally

$$W_2 = W_{3-4} + W_{tr2} - W_{ptr2} \tag{19}$$

2.3 Cylinder Process

2.3.1 Exhaust Gas Removal

In this calculation, a constant pressure is assumed in the cylinder when piston rises from BTC to the start of scavenging. The work done by piston is

$$W_{p,exh} = p_{exh}\left(V(BDC) - V(SS)\right) \tag{20}$$

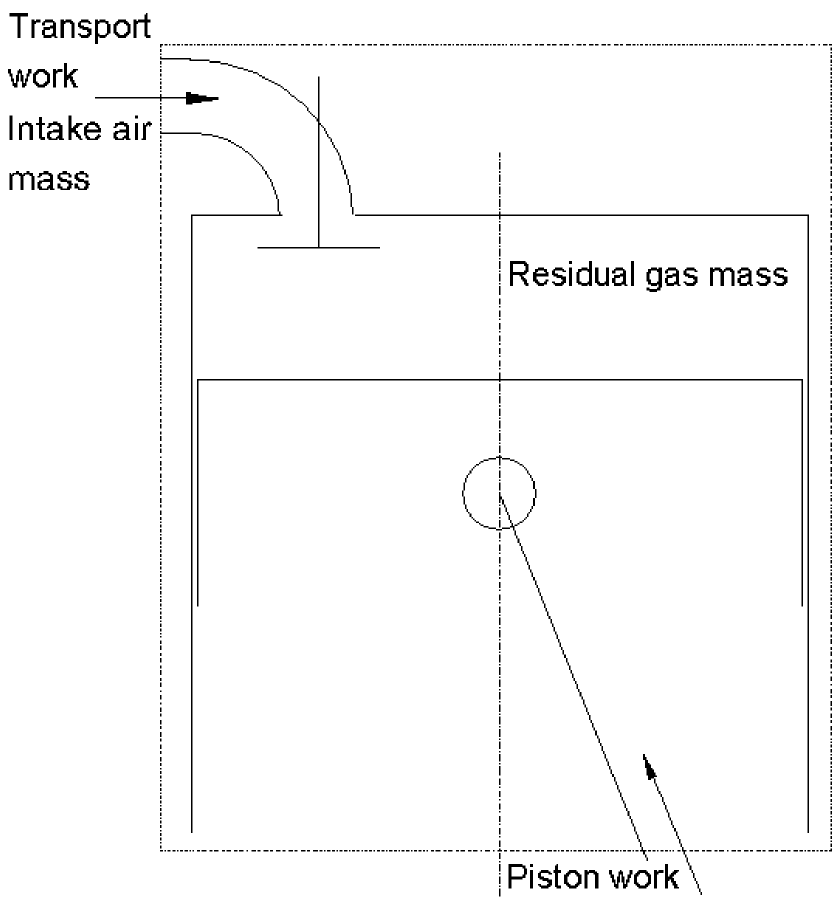
Where:

V(TDC) ≡ Cylinder volume at bottom dead center

V(SS) ≡ Cylinder volume at the scavenging start

2.3.2 The Scavenging

Reliable manual calculation of the scavenging is not possible. The phenomenon is far too complex. For example, back flow and pressure decrease in the intake duct during scavenging should be taken into consideration. In the following, the scavenging is estimated in the base of flow simulations. The scavenging duration is estimated according to knowledge learned from flow simulations. The pressure degrease in the intake duct is not considered in the calculation.



Picture 4

The pressure is assumed to rise linearly from p_{SS} (pressure at the start of scavenging) to p_6 in the cylinder during scavenging. The end of the scavenging occurs when the cylinder pressure has risen to scavenging pressure (p_6). The residual gas temperature ($T_{SS,res}$) is estimated / guessed. The mass of the residual gas is

$$m_{res} = \frac{p_{SS} V_{SS}}{R_{res} T_{SS}} \tag{21}$$

Where:

R_{res} ≡ gas constant for residual gas

p_{SS} ≡ pressure at the start of scavenging

T_{SS} ≡ temperature at the start of scavenging

Pressure rising is assumed to be linear in the cylinder during scavenging. The rising piston work during scavenging is

$$W_{p,sca} = \frac{p_{SS} + p_6}{2}\left(V(SS) - V(ES)\right) \tag{22}$$

Where:

V(ES) ≡ cylinder volume at the end of scavenging

Temperature of the intake air is T_5 at the boundary. Transport work $W_{air,sca}$ is done during scavenging

$$W_{air,sca} = p_5 v_5 \tag{23}$$

Where:

v_5 ≡ specific volume after second intercooler

The cylinder temperature after scavenging is

$$T_6 = \frac{m_{air} c_{v,air}\left(T_{m,air}\right) T_5 + m_{res} c_{v,res}\left(T_{m,res}\right) T_{SS,res}}{m_{air} c_{p,air}\left(T_{m,air}\right) + m_{res} c_{p,res}\left(T_{m,res}\right)} + \frac{W_{p,sca} + W_{air,sca}}{m_{air} c_{v,air}\left(T_{m,air}\right) + m_{res} c_{v,air}\left(T_{m,res}\right)} \tag{24}$$

Where:

$c_{v,air}$ ≡ specific volumetric heat capacitance of air

$c_{v,res}$ ≡ specific volumetric heat capacitance of residual gas

$c_{p,air}$ ≡ specific volumetric heat capacitance of air

$c_{p,res}$ ≡ specific volumetric heat capacitance of residual gas

T_m ≡ mean temperature of gas component (air or residual gas)

The cylinder volume must correspond the volume used in the calculations. The cylinder volume is

$$V_{ES} = \frac{(m_{air} R_{air} + m_{res} R_{res}) T_6}{p_6} \quad (25)$$

By rearranging, we get the air mass in cylinder

$$m_{air} = \frac{\frac{V_{ES} p_6}{T_6} - m_{res} R_{res}}{R_{air}} \quad (26)$$

The mass of stoichiometric exhaust gas in residual is needed for calculation of corresponding equivalence ratio of the gas mixture in cylinder. The mass fraction of stoichiometric exhaust gas as a function of equivalence ratio

$$w_{st,res} = 1 - \frac{1}{1 + \frac{L_0 + 1}{(\lambda_{res} - 1) L_0}} \quad (27)$$

From the mass of stoichiometric exhaust gas, the corresponding equivalence ratio in the cylinder after scavenging is

$$\lambda_6 = \frac{m_{air} + (1 - w_{st,res}) + \frac{L_0}{L_0 + 1} w_{st,res} m_{res}}{\frac{L_0}{L_0 + 1} w_{st,res} m_{res}} \quad (28)$$

2.3.3 Compression in the Working Cylinder

The final compression from point 6 to point 7’ is assumed to be adiabatic. The compression temperature from the adiabatic compression equation is

$$T_{7'} = T_6 \varepsilon_2^{\kappa-1} = T_6 \left(\frac{V_6}{V_{7'}} \right)^{\kappa-1} \tag{29}$$

The compression pressure is correspondingly

$$p_{7'} = p_6 \varepsilon_2^{\kappa} = p_6 \left(\frac{V_6}{V_{7'}} \right)^{\kappa} \tag{30}$$

The compression work from 6 to 7’ is

$$W_{6-7'} = \frac{mR}{(\kappa - 1) M_{6-7'}} \left(T_{7'} - T_6 \right) \tag{31}$$

2.3.4 Isochoric Combustion

Isochoric combustion occurs from 7’ to 7. Temperature is allowed to rise to T_7. The fuel mass burned from 7’ to 7 is estimated

$$m_{f,7'-7} = \frac{m_{7'} c_{v,7'-7} \left(T_7 - T_{7'} \right)}{\left(q_l - c_{v,f} \left(T_7 - T_f \right) \right)} \tag{32}$$

Where

q_l ≡ lower heat value of fuel for liquid fuel

T_f = fuel temperature during injection

$c_{v,7'-7}$ ≡ mean isochoric specific heat capacitance in cylinder

$c_{v,f}$ ≡ specific heat capacitance of fuel vapor

The specific heat capacitance is defined

$$\mathrm{c_p} = \mathrm{c_v} + \frac{\mathrm{R}}{\mathrm{M}} \quad \Rightarrow \quad \mathrm{c_v} = \mathrm{c_p} - \frac{\mathrm{R}}{\mathrm{M}} \tag{33}$$

For the mass fraction of air as a function of equivalence ratio, we have

$$w_{air}(\lambda) = \frac{(\lambda - 1)L_0}{\lambda L_0 + 1} \quad (34)$$

Where

$\lambda \equiv$ Equivalence ratio $\left(\equiv \frac{m_{air}}{L_0 m_f} \right)$

$L_0 \equiv$ Stoichiometric air to fuel ratio [= kg/kg_{fuel}]

For the molar fraction of air as a function of equivalence ratio, we have

$$x_{air} = \frac{\frac{w_{air}}{M_{air}}}{\frac{w_{air}}{M_{air}} + \frac{w_{st,exh}}{M_{st,exh}}} \quad (35)$$

Where

$w_{st,exh} \equiv$ mass fraction of exhaust gas

$M_{st,exh} \equiv$ Molar mass of stoichiometric exhaust gas

$M_{air} \equiv$ Molar mass of air

The molar mass of the gas in cylinder

$$M = x_{air} M_{air} + x_{st,exh} M_{st,exh} \quad (36)$$

The specific heat capacitance as function of equivalence ratio is

$$c_p(\lambda) = \frac{c_{p,st,exh} + (\lambda - 1)\frac{L_0}{L_0 + 1} c_{p,air}}{1 + (\lambda - 1)\frac{L_0}{L_0 + 1}} \quad (37)$$

The pressure from the ideal gas equation

$$p_{7'} = \frac{m_{7'} R T}{M_{7'} V_{7'}} \quad (38)$$

2.3.5 Isothermal Expansion

Isothermal expansion occurs while 7-8. Equivalence ratio is allowed to go to λ_8 (= λ_{res}). The corresponding air mass fraction is got from equation 27 substituting $w_{air} = 1 - w_{st,exh}$. The fuel mass injected during 7-8 is

$$m_{f,7-8} = \frac{(1 - w_{air,8})m_7 - m_{st,exh,7}}{L_0 + w_{air,8}} \tag{39}$$

Where

$w_{air,8} \equiv$ air mass fraction in the cylinder corresponding to λ_8

$m_7 \equiv$ cylinder mass in the point 7

$m_{st,exh,7} \equiv$ stoichiometric exhaust gas in the cylinder

For isothermal expansion, energy from the fuel during isothermal expansion and work got from the expansion are same

$$Q_{8-9} = Q_{in2} = m_{f,7-8} q_l \tag{40}$$

From the equation of isothermal expansion, we get the end volume of the expansion

$$Q_{in2} = m_{7-8} R_{7-8} T_7 \ln\left(\frac{V_8}{V_7}\right) \Rightarrow V_8 = \exp\left(\frac{Q_{in2}}{m_{7-8} R_{7-8} T_7} + \ln(V_7)\right) \tag{41}$$

2.3.6 Adiabatic Expansion

The final expansion from point 8 to point 9 is assumed to be adiabatic. The temperature after expansion from the adiabatic expansion equation is

$$T_9 = T_8 \left(\frac{V_8}{V_9}\right)^{\kappa_{8-9} - 1} \tag{42}$$

The final pressure is correspondingly

$$p_9 = p_9 \left(\frac{V_8}{V_9}\right)^{\kappa_{8-9}} \tag{43}$$

The expansion work from 8 to 9 is

$$W_{8-9} = \frac{m_8 R_8}{\kappa_{8-9} - 1}(T_9 - T_8) \tag{44}$$

2.4 Efficiency and Power

The theoretical work can be calculated from equation

$$W_{th} = Q_{in2} + W_{8-9} - W_{1-2} - W_{3-4} - W_{p,exh} - W_{p,sca} - W_{6-7'} \tag{45}$$

The theoretical efficiency is

$$\eta_{th} = \frac{W_{th}}{Q_{in}} = \frac{Q_{in2} + W_{8-9} - W_{1-2} - W_{3-4} - W_{p,exh} - W_{p,sca} - W_{6-7'}}{Q_{in1} + Q_{in2}} \tag{46}$$

For total efficiency, we get

$$\eta = \eta_{ind} \eta_g \eta_m \tag{47}$$

Where

$\eta_g \equiv$ Process efficiency (German gütegrad, defines the difference between real and theoretical process)

$\eta_m \equiv$ Mechanical efficiency

The brake power per cylinder is got from equation

$$P_{cyl} = \eta (Q_{in1} + Q_{in2}) n \tag{48}$$

Where

n ≡ speed of the engine (1/s)

The specific fuel consumption is defined

$$sfc\left(\frac{g}{kWh}\right) = \frac{3600\left(\frac{kJ}{kWh}\right)}{\eta(-) q_l \left(\frac{kJ}{g}\right)} \tag{49}$$

3 Material Properties

Specific heat capacitance at constant pressure can be calculated for each species (i) by using following polynomial

$$\frac{c_{p,i}(T)}{R_i} = a_{i1} + a_{i2}T + a_{i3}T^2 + a_{i4}T^3 + a_{i5}T^4 \tag{50}$$

[T] = K, [c_p] = kJ/kgK

Table 1: The coefficients for polynomials /1/ p.131

	H_2O		N_2		O_2		CO_2	
T range	300-1000	1000-5000	300-1000	1000-5000	300-1000	1000-5000	300-1000	1000-5000
a_1	4,07010E+00	2,71680E+00	3,67480E+00	2,89630E+00	3,62560E+00	3,62200E+00	2,40080E+00	4,46080E+00
a_2	-1,10840E-03	2,94510E-03	-1,20820E-03	1,51550E-03	-1,87820E-03	7,36180E-04	8,73510E-03	3,09820E-03
a_3	4,15210E-06	-8,02240E-07	2,32400E-06	-5,72350E-07	7,05550E-06	-1,96520E-07	-6,60710E-06	-1,23930E-06
a_4	-2,96370E-09	1,02270E-10	-6,32180E-10	9,98070E-11	-6,76350E-09	3,62020E-11	2,00220E-09	2,27410E-10
a_5	8,07020E-13	-4,84720E-15	-2,25770E-13	-6,52240E-15	2,15560E-12	-2,89460E-15	6,32740E-16	-1,55260E-14

The corresponding adiabatic constant is

$$\kappa = \frac{c_p}{c_v} = \frac{c_p}{c_p - \frac{R}{M}} \tag{51}$$

For a gas mixture (for example air) the specific heat capacitance is

$$c_p = \sum_{j=1}^{N} c_{p,j} w_j \tag{52}$$

Where

w_j ≡ mass fraction of species

N ≡ number of species

The molar mass of gas mixture is

$$M = \sum_{j=1}^{N} x_j M_j \tag{53}$$

Where

x_j ≡ molar fraction of species

For the molar fraction of species in gas mixture, we have

$$x_i = \frac{\frac{w_i}{M_i}}{\sum_{j=1}^{N} \frac{w_j}{M_j}} \tag{54}$$

4 Calculation Example

4.1 Assumptions and Boundary Values

In this example, the calculation of the Z-process is carried out for partial load.

The engine:

2 work cylinders

Cylinder diameter * Stroke = 95 x 90 mm → V_{stroke} = 638 cm³ /cyl.

Connecting rod ratio = 0.326 $\left(= \frac{Camshaft\ radius}{Connecting\ rod\ length} \right)$

Geometrical compression ratio: ε = 50

Engine speed: n = 2100 1/min = 35 1/s

The corresponding air / fuel equivalence ratio is set to 1.4. The environment temperature is 300 K and pressure 1 bar. The absolute humidity of the intake air is assumed to be 0.59%. The maximum cylinder temperature is limited to 2200 K. The pressure after the pre compressor is 2.4 bar. The pressure after second stage compressor is 10 bar. The average cylinder pressure is assumed to be 1.3 bar from bottom dead center to the start of scavenging. The exhaust gas temperature is guessed to be 800 K in this pressure (point SS). The exhaust gas temperature should be lower than the calculated adiabatic temperature because of the heat loss from the cylinder. The process efficiency is assumed to be 85 % and mechanical efficiency is estimated to be 85 %. The scavenging is assumed to occur while 125 – 145 ° BBDC.

The lower heating value of liquid fuel is 43.2 MJ/kg. Stoichiometric air/fuel ratio is 14.5. /1/ p. 915

4.2 Analyses

The adiabatic compression temperature after the pre-compressor is calculated first (equation 2). The adiabatic constant is calculated using polynomial coefficients. The results are focused by iterative method in spreadsheet calculation. The polytropic constant is calculated with equation 7 and work with equation 8, 9 and 11. At this moment, the intake air mass is not yet known, but the specific work can be calculated.

The piston compression temperature and adiabatic constant are calculated iteratively. Compression work is calculated by adiabatic work equation. Transport work and positive transport work are got from equations 16 and 18.

The cylinder average temperature and intake air mass are calculated iteratively. The temperature is calculated first by equation 24 using guessed mass of air and temperature for material properties. The air mass is then calculated with equation 26.

The corresponding equivalence ration is calculated using equation 28. Using equations 34, 35, 36 and 37, a spreadsheet computation method is developed for easy calculation of needed material properties (molar mass, specific heat capacitance and adiabatic constant). Material properties are presented in table 2.

Temperature and pressure after final compression are calculated iteratively in the manner described in chapter 2.3.3. The fuel mass injected during isochoric combustion is calculated according to the equation 32 iteratively. The iterated factor is the equivalence ratio, which affects to the material properties. While the temperature is known, the pressure can be calculated using ideal gas equation.

The fuel mass injected during isothermal expansion is calculated using equation 39. The adiabatic expansion is calculated according to chapter 2.3.6. The efficiency, power output and specific fuel consumption are calculated according to chapter 2.4. If the guessed exhaust gas temperature does not correspond the calculated value, it can be chanced and the calculation can be done again. The temperature, pressure, cylinder volume, equivalence ratio and mass per cylinder are presented in the table 2. In the table, the temperature in point SS is calculated adiabatic temperature (real temperature is lower).

Table 2

Point	T (K)	p (bar)	V_{cyl} (cm^3)	Mass (g)	Equivalence ratio (-)	Molar mass (kg/kmol)	Specific heat capacitance, isobaric (kJ/kgK)
1	300.0	1		0.336	infinite	28.86	1.016
2	420.3	2.4		0.336	infinite	28.86	1.028
3	310.0	2.4		0.336	infinite	28.86	1.017
4	463.2	10		0.336	infinite	28.86	1.035
5	310.0	10		0.336	infinite	28.86	1.017
6	665.1	10	87.12	0.455	5.53	28.88	1.097
7'	1238.3	124.6	13.02	0.455	5.53	28.88	1.219
7	2200.0	226.3	13.02	0.466	1.88	28.92	1.372
8	2200.0	101.0	29.50	0.471	1.4	28.94	1.401
9	946.0	2.0	650.96	0.471	1.4	28.94	1.227
SS	887.4	1.5	182.80	0.119	1.4	28.94	1.211

The work balance per cylinder is presented in the table 3.

Table 3

	Work / rotation (J)
Pre compressor, W_{1-2}	21.7
Pre compressor positive transport work. W_{ptr1}	29.0
Pre compressor transport work. W_{tr1}	40.7
Piston compressor work to gas. W_{3-4}	37.8
Piston compressor positive transport work. W_{ptr2}	30.0
Piston compressor transport work. W_{tr2}	44.8
Piston work from BTC to SS. $W_{p.exh}$	60.9
Piston work during scavenging. $W_{p.sca}$	45.4
Compression in the cylinder from 6 to 7. W_{6-7}	229.6
Fuel heat to cylinder during isochoric burn. Q_{in1}	467.6
Fuel heat during isothermal expansion. Q_{in2}	243.7
Work during isothermal expansion. W_{7-8}	243.7
Expansion from 8 to 9. W_{8-9}	622.7
Theoretical work	444.5

Substituting the work values to the equation 46, the theoretical efficiency becomes 62.5 %. For total efficiency we get 45.1 %. Corresponding specific fuel consumption is 184.6 g/kWh. For a 2-cylinder engine at speed 2100 RPM, the brake power is 22.5 kW. The fuel injection rate is 16.5 mg/stroke.

References

1. Heywood. John B. Internal Combustion Engine Fundamentals. Singapore. II Series. McGraw-Hill Book Company. 1988. 930 p. ISBN 0-07-028637-X