# Intake-Throttling for Trap Regeneration
# A Cursory Engineering Estimate

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## 1. Introduction.

Initiating soot trap regeneration, i.e. starting to burn off accumulated black carbon, constitutes a well known problem due to low exhaust temperatures at part load of engines which is addressed in a recent paper by A. Mayer et al.[1], submitted to SAE. It reports on both, experiments and their confirmation by detailed computations at the Department of IC Engines at the Federal Polytechnic Institute Zurich: Intake throttling is investigated as a means to raise exhaust temperatures for soot trap regeneration.
Mayer's paper presents as well a history of developments of this method and all combinations of alternate designs already tested. Therefore, it is referred to this paper without repeating details. However, old fashioned engineers are rarely satisfied with sophisticated computer analyses, they require some engineering estimates, which is done in the following

Basically, particle traps are loaded with soot after a few hours of operation; many times, probably in the vast majority of applications, engines are operated at very low part load; hence, exhaust temperatures are far from levels necessary to initiate burning off of soot. Occasionally, temperature levels are as low that even the use of catalysts may fail. Reason for this phenomenon is that power control of combustion engines is done by reducing fuel only which leaves the combustion process with an high air excess factor: While $\lambda$= 2 at full load, it equals up to about $\lambda$ = 8 at low part load. In the following an approximate analysis of the effect of controlling temperature by reducing air flow is presented.

## 2. Notations

| | |
|---|---|
| $H_u$ | calorific heat release of fuel, minus heat of condensation of water generated (i.e. lower heating value) [kJ/kg fuel] |
| V | Volume at standardised conditions (i.e. at 273K/0°C; 1013mbar) |
| $V_{as}$ | stochiometric volume of air required for burning 1kg of fuel, [$m^3_N$/kg] |
| $V_{ac}$ | stochiometric volume of combustion or flue gas generated, [$m^3_N$/kg] |
| $V_a$ | effective volume of air used for burning 1kg of fuel, [$m^3_N$/kg] |
| $V_c$ | effective volume of combustion or flue gas generated, [$m^3_N$/kg] |
| T; t | temperature K; °C. |
| c | specific heat,[kJ/kgK] |
| $c_a$ | specific heat of air at constant pressure, [kJ/$m^3_N$K] |
| $c_c$ | specific heat of combustion gas/air at constant pressure, [kJ/$m^3_N$K] |
| $c_f$ | specific heat of fuel, [kJ/kgK] |
| $h_a$ | specific enthalpy of air, [kJ/$m^3_N$] |
| $h_c$ | specific enthalpy of combustion gas, [kJ/$m^3_N$] |
| m | exponent of isentropic change of state, [-] |
| $\gamma$ | ratio of specific heat of gas,[-] |
| $\lambda$ | air excess factor, [-] |
| $\Pi$ | pressure ratio, [-] |

## 3. Methodical Approach

As an approximation a hybrid cycle is chosen, temperatures are determined due to:
1. Isentropic compression of air (plus some allowance for polytropic change of state).
2. Isobaric combustion of liquid fuel (hydrocarbon) injected, combustion temperature
3. Isochoric pressure increase of combustion gas due to temperature increase.
4. Isentropic expansion of flue or combustion gas

Finally, temperatures are compared for two different air excess factors,
$\lambda = 8$ and $\lambda = 2$.
"Hybrid cycle" because we do not idealise with pure air and due to isochoric pressure increase an element of the Otto cycle is incorporated.
Crucial is the estimate of the combustion temperature as a function of $\lambda$, which is accomplished by means of the approximate method of Rosin and Fehling. It is based on the balance of enthalpies.

## 4. Compression.

At first glance the isentropic compression from intake to turbo-/supercharger to compression within the engine's cylinder does not constitute any mental problem at all. However, it is quite advisable to take a closer look because just for approximate computations one should remain aware of simplifying assumptions. Compression of air is done in two stages, by turbo- or supercharging as a first stage, compression ratio $CR_S$ and then within the engine's cylinder, compression ratio $CR_Z$ . In case of intercooling, compression work of the piston within the cylinder is less, work of both compressions is to be determined separately; if there is no intercooling, work is lumped into one single compression, total pressure ratio is just the product of both ratios,

$$\Pi_t = \Pi_s \cdot \Pi_z \qquad (1)$$

Further, we consider throttling of the air flow prior to the cylinder and behind the turbo: this requires a look at the temperature depression due to the Joule-Thomson effect, because an isenthalpic throttling would require slow air flow, hence large volumes, we do not have isenthalpic throttling. Just for the sake of simplification this paper disregards both effects, of temperature reduction due to throttling and of temperature increase due to omission of intercooling, expecting some, at least partial, cancellation of deviations accepted. Further, reference temperature for enthalpies etc. and ambient air temperature are chosen to be $T_o$ = 273K or $t_o$ = $0^oC$.
Hence, temperature increase of the air prior to ignition is obtained to

$$\Delta T = T_0 \cdot \left(\Pi_t{}^m - 1\right) \qquad (2)$$

with m denoting

$$m = \frac{(\gamma - 1)}{\gamma} \qquad (3a)$$

and $\gamma$ being the ratio of specific heats, i.e.

$$\gamma = \frac{c_p}{c_v} \tag{3b}$$

## 5. Isobaric Combustion, Balance of Enthalpies.

A cursory estimate of combustion temperatures, based on the balance of enthalpies is obtained as follows:

$$V_c \cdot h_c = H_u + c_f \cdot t_f + V_a \cdot h_a \tag{4}$$

At left is the enthalpy of the flue or combustion gas, at right there are the contributing terms, as there are:

- Lower calorific value of the fuel, also called –not correctly, fuel heat release,
- Enthalpy of fuel, i.e. specific heat of fuel times temperature-difference to reference temperature and
- Enthalpy of air i.e. specific enthalpy of the intake air times volume of air standardised to reference conditions.

The first term, determined by the lower calorific value of Diesel-fuel, approximately $H_u$ = 40.000 kJ/kg is the predominant one.
Note: The lower calorific value is defined by the heat release measured by means of the calorimetric bomb and corrected by the heat of condensation of water generated, i.e. the heat of condensation of the water generated is deducted from the heat release determined by the calorimetric bomb.
The second term, enthalpy of fuel, is negligible compared to the other two; hence, it may be left away.
The third term, the enthalpy of the air after an approximate isentropic compression (plus an allowance of about 10-20% for the deviation from the ideal isentropic to polytropic change of state) is easily obtained. Temperatures are still low enough for considering specific heat of air to be constant. Both compressions, of supercharging and within the cylinder may be lumped together if there is no intercooling, which should be bypassed anyway if rise of temperature is the aim. As has been mentioned, one may argue temperature decrease due to Joule-Thompson and increase are compensating By simplifying the enthalpies of ideal gases to specific heat times temperature difference to reference temperature and the total enthalpy being the product of specific enthalpy times standard volume one obtains by means of simple transforming of equ. (4):

$$h_c \approx \frac{H_u}{V_c} + \frac{V_a}{V_c} \cdot h_a \tag{5}$$

By choosing $T_{ref}$ = 273K resp. $t_{ref}$ = 0°C a further simplification arises,

$$h_c = c_c \cdot t_c \tag{6a}$$
$$h_a = c_a \cdot t_a \tag{6b}$$

such that equ.s(5 and 6) yield a temperature of combustion

$$t_c \approx \frac{H_u}{c_c \cdot V_c} + \frac{V_a}{V_c} \cdot \frac{c_c}{c_c} \cdot t_a \qquad (7)$$

Combustion temperature $t_c$ depends on effective volume of air $V_a$, its temperature $t_2$, just proir to inset of combustion, and on effective volume of flue gas generated $V_c$ . These volumes are determined by means of the approximation of Rosin and Fehling; stochiometric values first:

$$V_{as} = \frac{a \cdot H_u}{1000} + b \qquad \left[\frac{m^3{}_N}{kg(fuel)}\right] \qquad (8a)$$

$$V_{cs} = \frac{c \cdot H_u}{1000} + d \qquad \left[\frac{m^3{}_N}{kg(fuel)}\right] \qquad (8b)$$

Coefficients of Rosin and Fehling for liquid hydrocarbons being given ( ref. [2]) to

$$a = 0{,}203 \quad \left[\frac{m^3}{kJ}\right]$$

$$b = 2{,}0 \quad \left[\frac{m^3}{kg}\right] \qquad (9a)$$

$$c = 0{,}265 \quad \left[\frac{m^3}{kJ}\right] \qquad \text{I)}$$

$$d = 0{,}0 \quad \left[\frac{m^3}{kg}\right] \qquad (9b)$$

Consequently, stochiometric volume requirement of air per kilogram fuel is obtained as

$$V_{as} = 10{,}12 \; \frac{m^3{}_N}{kg} \qquad (10a)$$

while volume of flue gas generated is

$$V_{cs} = 10{,}60 \; \frac{m^3{}_N}{kg} \qquad (10b)$$

yielding the ratio

$$\frac{V_{as}}{V_{cs}} = 0{,}955 \qquad (10c)$$

*I).: Do not confuse with specific heat .*

In course of further development the relative increase in stochiometric volumes of combustion will be required, which is defined as

$$\Delta V_{as} = \frac{V_{cs} - V_{as}}{V_{as}} \tag{11a}$$

Numerically, $\Delta V_s$ is obtained as

$$\Delta V_{as} = 0{,}0474 \tag{11b}$$

Allowing for an air excess factor lambda we obtain generally

$$V_a = \lambda \cdot V_{as} \tag{12a}$$

$$V_c = V_{cs} + (\lambda - 1) \cdot V_{as} \tag{12b}$$

and in particular with $2 \leq \lambda \leq 8$, $V_a$ and $V_c$ vary between

$$20{,}24 \leq V_a \leq 81 \quad \left[\frac{m^3{}_N}{kg}\right]$$

$$21{,}72 \leq V_c \leq 81{,}44 \quad \left[\frac{m^3{}_N}{kg}\right] \tag{13}$$

while the ratio of these volumes as required by equ.(5) is varying between

$$0{,}932 \leq \frac{V_a}{V_c} \leq 0{,}995 \tag{14a}$$

the corresponding reciprocal values are found to be between

$$1{,}073 \geq \frac{V_c}{V_s} \geq 1{,}005 \tag{14b}$$

Hence, this ratio is very close to unity, of which one may make use when transforming such that the dependence on lambda is elucidated:
First constant parameters are to be separated, for the effective volume of flue gas one may set according to equ.(12b)

$$V_c = \lambda \cdot V_{as} + V_{cs} - V_{as} \tag{15}$$

If a deviation of less than 2.5% is permissible, i.e.

$$\frac{V_{cs}}{V_{as}} \approx 1$$

equation (15) may be replaced by

$$V_c \approx \lambda \cdot V_{as} \qquad (16)$$

otherwise, one may set

$$V_c = \lambda \cdot V_{as} \cdot \left[1 + \frac{\Delta V_{as}}{\lambda}\right] \qquad (17)$$

Thus the first term of equ. (7) may be rewritten by using the polynomial expansion for $(1+x)^{-1}$ if x<<1:

$$(1+x)^{-1} = 1 - x$$

$$\frac{H_u}{c_c \cdot V_c} = \frac{H_u}{c_c \cdot \lambda \cdot V_{cs}} \cdot \left(1 - \frac{\Delta V}{\lambda}\right) \qquad (18)$$

Specific heats may be regarded constant for the temperature range considered (mean value of the interval $0^{o}C < t < 400^{o}C$; even increasing that interval is of minor importance); we take from tables, ref.[3]

for flue gas: $c_c = 1.45 \left[\frac{kJ}{Nm^3K}\right]$ (19a)

and for air: $c_a = 1.32 \left[\frac{kJ}{Nm^3K}\right]$ (19b)

The second term on the right hand side of equ.(7) is then obtained as

$$\frac{c_a}{c_c} \cdot \frac{V_a}{V_c} \cdot \Delta T = \frac{c_a}{c_c} \cdot \left(1 - \frac{\Delta V_{as}}{\lambda}\right) \cdot \Delta T \qquad (20)$$

which in turn leads finally to the following expression for the temperature of combustion:

$$t_c \approx \frac{1}{\lambda} \cdot \frac{H_u}{c_c \cdot V_{cs}} \cdot \left(1 - \frac{\Delta V_{as}}{\lambda}\right) + \frac{c_a}{c_c} \cdot \left(1 - \frac{\Delta V_{as}}{\lambda}\right) - \Delta T \qquad (21)$$

The factor $(1 - \Delta V_{as}/\lambda)$ is slightly less than unity, depending on lambda,

$0{,}977 \leq (1 - \Delta V/\lambda) \leq 0{,}994$ for $2 \leq \lambda \leq 8$.

Further,

$$\frac{H_u}{c_c \cdot V_{as}} = 2726 \qquad [^oC] \qquad (22)$$

and

$$\frac{c_a}{c_c} = 0{,}91 \qquad (23)$$

Accordingly combustion temperature may be obtained according to simplification chosen either as

$$t_c = \frac{2726}{\lambda} \cdot \left(1 - \frac{\Delta V_{as}}{\lambda}\right) + 0{,}91 \cdot \left(1 - \frac{\Delta V_{as}}{\lambda}\right) \cdot \Delta T \qquad (24a)$$

or

$$t_c = \frac{2726}{\lambda} + 0{,}91 \cdot \Delta T \qquad (24b)$$

**Just the latter expression (24b) elucidates the influence of the air excess factor $\lambda$: Reducing it constitutes the determining parameter for increasing the exhaust temperature.**

In the following, both an isochoric pressure increase and the final expansion are considered consecutively.

## 6. Isochoric Pressure Increase

Determination of isochoric pressure increase due to the rise of temperature is no doubt the simplest task of all:

$$p_3 = p_2 \cdot \left(\frac{T_3}{T_2}\right) \qquad (25)$$

note: due to defining reference temperature t =0$^o$C (= 273K), $T_2$ is identical with $\Delta T$ of equ.(2).

## 7. Expansion

More or less, isentropic expansion is now used to determine the final exhaust or tail-pipe temperature, which had been our aim from the beginning. It is reasonable to expand to a final pressure $p_e$, consisting of the ambient air pressure plus some increment for tailpipe or particle-filter loss $\Delta p$, we set arbitrarily $\Delta p$ = 200mbar.

Pressure ratio of expansion defined as

$$\Pi_e = \frac{p_3}{p_e} \qquad (26)$$

results then into $T_e$, the final temperature, as aimed for, which is obtained

$$T_e = T_3 \cdot \Pi^{-m} \qquad °K \qquad (27a)$$

and

$$t_e = T_e - 273 \qquad (27b)$$

By determining these temperatures numerically for minimum and maximum $\lambda$, the reasoning behind the methods of Mayer and Pattas is apparent.

## 8. Numerical Example.

This approximate method may now be tested against an example of ref.[1], which is based on detailed computations and the verification by maens of a test bench engine.

Change of state

No.1 Compression of air

a) $\lambda$ =2, $t_1$ = 273 K , $\Pi$ =17,2 (throttling and supercharging are cancelling out each other)
result:
isentropic temperature $T_2$ = 616K,
$\Delta T = t_2 = 343^oC$

as there is substantial heat transfer between the walls an internal friction, we set arbitrarily a higher temperature ( 10%)
$T_2^*$ = 680K
$t_2^* = 407\,^oC$

b) $\lambda$ = 8, no throttling, supercharging, compression ratio $\Pi_s$ =2, hence, total compression ratio $\Pi_t$ = 34,4.
Result:
Isentropic end temperature is obtained to
$T_2$ = 751K,
$\Delta T = t_2 = 478^oC$
and we set in analogy $T_2^*$ = 800K
respectively $t_2^* = 527\,^oC$

No.2 Isobaric Combustion; $t_c$ according to equ.(24b):

a) $\lambda$ =2:

Result:

$$t_c = \frac{2726}{2} + 0{,}91 \cdot 527$$

$t_c$ = 1733 $^oC$ => $T_c$ = 2006 K

b) $\lambda$ = 8:

$$t_c = \frac{2726}{8} + 0{,}91 \cdot 527$$

$t_c$ = 820 $^oC$ => $T_c$ = 1048 K

No.3 Isochoric pressure increase

a) $\lambda$ = 2:
$p_3$ = 50,4 bar

b) $\lambda$ = 8:
$p_3$ = 45,1 bar

No.4 Expansion to 1,2bar:

a) $\lambda$ = 2

$$\Pi_{ex} = \frac{1{,}2}{50{,}7} = 42{,}3^{-1}$$

$T_4$ = 687K => $t_4$ = 414$^oC$

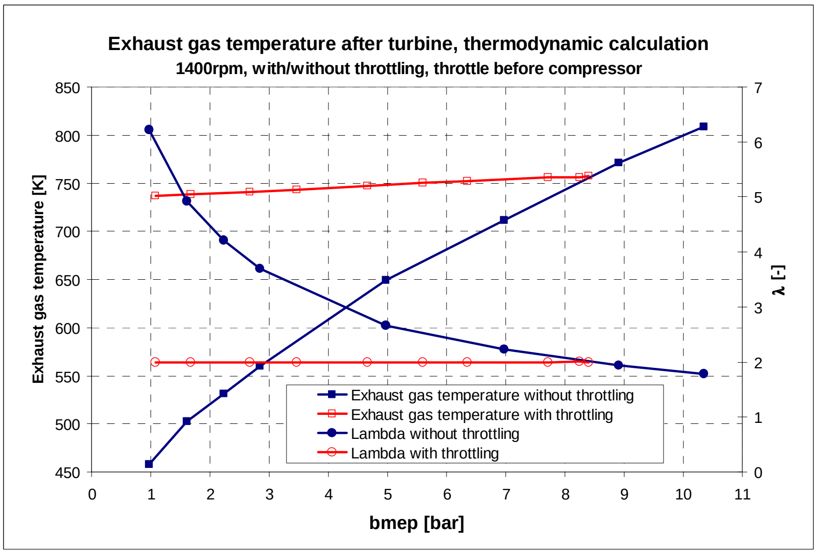
b) $\lambda$ =8

$$\Pi_{ex} = \frac{1{,}2}{50{,}7} = 37{,}5^{-1}{}_{-1}$$

$T_4$ = 372K => $t_4$ = 98,7$^oC$

The agreement of this very cursory estimate with results of detailed computations of ref. [1] is amazingly good; ref. [1] may be quoted:

*“At customary operations $\lambda$ increases from $\lambda$ = 2 to $\lambda$ = 6 – 8 with decreasing load, which brings about a reduction of the exhaust temperature from more than 800K to less than 450K”.*




The above fig from ref.[1] presents temperatures of exhaust gas with and without pre-throttling at 1400 rpm according to detailed computations.

## 9. Concluding Remark

In our case the good agreement may be accidental for the particular example chosen; however, an understanding for the basic consideration of reducing λ parallel with fuel for increasing exhaust temperature, in order to trigger filter regeneration is clearly achieved, even by means of elementary thermodynamics, which in turn due to some minor algebraic operations could be presented in a somewhat elegant way. The basic finding is: Exhaust temperature runs inverse to air excess factor, if exhaust temperature at part load has to be increased for initiating trap regeneration at part load, a simple means is to throttle intake air of the engine.

## 10. References

[1] A. Mayer et al.: Throttling of Diesel Engine Intake as a Means for Initiating Active Regeneration of Particle Filters SAE Paper Proposal No 03FL160

[2] Recknagel/Hörmann, Taschenbuch für Heizung, Klima und Lüftung, 65.Aufl.90/91, p.173, Oldenbourg

[3] H. Hiedl, Technische Mechanik, Bd.III, pp314/315, Oldenbourg, Wien 1972

[4] F.J. Wallace, Diesel engine reference book, second ed. 1999, B. Challen,R. Baranescu ed., Butterworth, pp 9 - 15