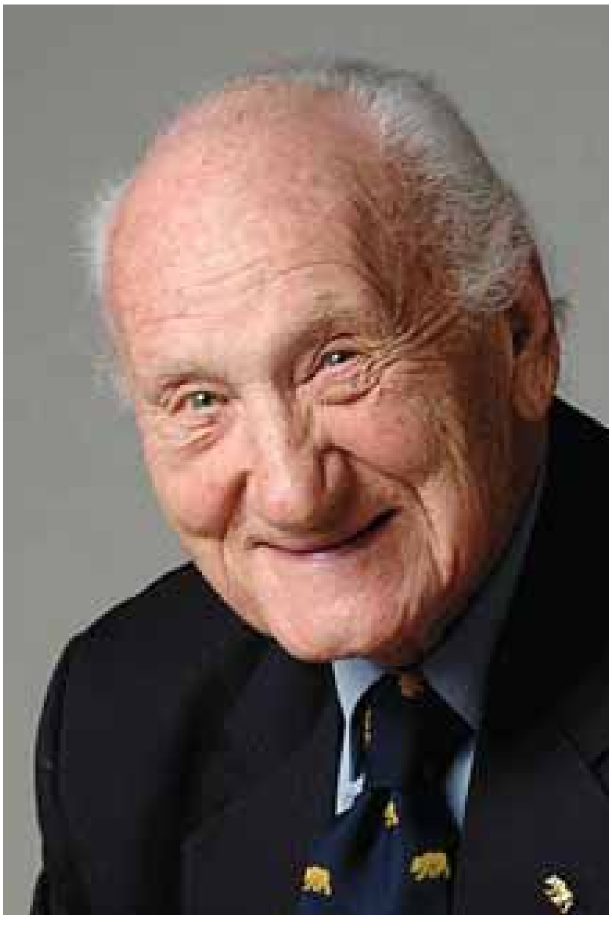
MEMORANDUM OF PROF. A. K. OPPENHEIM AND AN EXAMPLE OF APPLICATION OF THE OPPENHEIM CORRELATION (OPC)* FOR THE HEAT LOSSES DURING THE COMBUSTION IN IC-ENGINE

Antoni Jankowski

Institute of Aviation
Krakowska 110/114, 02-256 Warszawa, Poland
tel.: +48 22 8460011, fax: +48 22 846432,
e-mail: ajank@ilot.edu.pl

Jan Czerwinski

AFHB, Laboratory for IC-Engines and Exhaust Gas Control
Berne University of Applied Sciences Technics & Informatics, BFH-TI, Biel-Bienne, Switzerland
tel.: +41 32 3216680, fax: +41 32 3216681
e-mail: jan.czerwinski@bfh.ch



Antoni Oppenheim (Peg Skorpinski photo)

Antoni Kazimierz Oppenheim, professor emeritus of mechanical engineering at the University of California, Berkeley, and one of the world's leading experts on combustion and radiation heat transfer, died Saturday, Jan. 12, 2008 at the age of 92.

Oppenheim was born in Warsaw, Poland, on Aug. 11, 1915. He was home-schooled in French until the age of nine, when he attended local schools. After graduating from his high school in 1933, he entered the highly competitive Warsaw Institute of Technology, where he studied aeronautical engineering.

He passed the 2^{nd} war time in London.

In 1943 he graduated and continued with a doctorate at University of London. In 1945 he was admitted to the title of a doctor at Imperial College.

In 1948 professor A. K. Oppenheim moved to the USA where he got tenure at Stanford University which later helped him get the status of legal alien in the US. In 1950 he moved from Stanford University to University of California-Berkeley where in 1954 he got the position of associate professor and in 1958 the position of professor. He was affiliated with this university till the end of his life even when retired. This was also the place where his most important works were done.

His areas of interest were very wide: heat exchange, radiation, the nature of the initiation of gas detonation, development and structure of fronts of detonation, reciprocal influence of the fronts of detonation, blast, turbulent combustion and engines themselves. His interest in the combustion engines originated in Poland and was cultivated in Great Britain and the US.

At a time when the mechanics of detonations were largely a mystery because such events occurred at supersonic speeds, Oppenheim tackled the task of studying them by helping develop a type of high-speed photography that uses a laser light source to capture sub-microsecond exposures. As a result of this technique, Oppenheim was able to design experiments that led to groundbreaking descriptions of blast waves and of the process by which a detonation occurs.

Oppenheim is also credited with developing a method for quantifying radiation heat transfer - how heat moves through space.

Oppenheim's most recent research focused on improvements to the efficiency of the internal combustion engine powering most automobiles on the road today.

Oppenheim proposed and tested a pulsed jet combustion system in which the air-fuel mixture is ignited at multiple points throughout the cylinder. Experiments showed that this more efficient system allows for a leaner air-fuel mixture and lower operating temperatures. He calculated that this system could ultimately double the gas mileage while drastically reducing pollution in current internal combustion engines.

Among the numerous awards Oppenheim received throughout his career were: the Dionizy Smolenski Medal of the Polish Academy of Sciences for outstanding contributions towards advances in the knowledge of combustion and especially to the dynamics of explosions and reactive systems; the Alfred C. Egerton Medal of The Combustion Institute for distinguished, continuing and encouraging contributions to the field of combustion; and the Berkeley Citation, one of the highest honours bestowed by the university to those who have exceeded the standards of excellence in their fields.

Oppenheim received honorary doctorate degrees from the University of London, the University of Poitier and Warsaw's University of Technology. He was also a member of the International Academy of Astronautics, a fellow and honorary member of the American Society for Mechanical Engineers, a member of the U.S. National Academy of Engineering; and a foreign member of the Polish Academy of Sciences a fellow member of European Science Society of Powertrain and Transport.

Oppenheim officially retired from UC Berkeley in 1986, but he remained very active in his research until his death.

The whole community of scientists in the field of combustion engines, thermodynamics and combustion profoundly regretted the passing of Professor Antoni K. Oppenheim. For many of us he was known only from his publications that were always original and challenging. Many of us knew him personally, met him at the conferences and scientific symposia. He surprised his younger fellow engineers with his dynamics, inspired, made people ponder and discuss. He was charismatic, jolly, curious and open to other people.

We will keep in mind his teachings and we will miss his open-minded optimism, inspiration and understanding.

Oppenheim Correlation (OPC)

The present technical work was inspired by several discussions with Prof. Oppenheim in the years 1997 and 1998. In a collaboration between UCB and AFHB the data were exchanged and evaluations of the same data on both places were performed. The authors acknowledge the contributions of assistants: Mr. Slava R. Spector, UCB and Mr. P. Comte, AFHB.

The work on OPC was dedicated 1999/2000 to Antoni K. Oppenheim, our friend and professor to his 84 birthday, 57 years activity on combustion analysis and more than 50 years of SAE activity.

The authors

Abstract

The Oppenheim Correlation (OPC) is an empirical algorithm, which allows a simple estimate of heat losses to the wall during the combustion in IC-engine.

In present paper the results of different applications of OPC will be shown.

Even if there are still several needs and ideas for further research it can be stated, that the OPC is a promising possibility of modelling the wall heat losses and due to its simplicity it has to be recommended to the engine community.

The OPC can be used not only for didactics purposes, but also for quick simulation of wall heat losses and eventually for the on-line regulation of the cooling system.

In particular basic milestones of calculation of the working cycle, modelling of heat losses according to Oppenheim (OPC), engines and test procedure, examples of the OPC-application are presented in the paper. The OPC can be recommended to the engine community as a good alternative of modelling and calculation of the heat losses trough the combustion chamber wall.

Keywords: *combustion engines, Oppenheim Correlation OCP, heat transfer, heat looses, Vibe-Function*

1. Introduction

The analysis and investigation of the combustion process and of the heat losses of the IC-engines are very important tasks for the R & D engineers as well at the numeric as at the experimental level.

There is an extremely rich literature on these subjects, e.g. for combustion [1-15], for heat transfer to the wall [16-23]. A lot of technical meetings and symposia are organized only on the topics of pressure indication, evaluation and combustion analysis, [24, 25].

The development of the numeric modelling of the working cycle was marked particularly by the work of I. Viebe, [5], for the calculation of the heat release and by the work of G. Woschni, [18], for estimate of wall heat losses.

Oppenheim et. al. gives in his work a new look on the analysis of the exothermic process of combustion, as well as a new empirical algorithm, which allows a quick and simple estimate of heat losses to the wall, [26-29]. This algorithm, which is only a small part of the “Refinement of Heat Release Analysis”, [28], will be called, as in the title of this paper: *Oppenheim Correlation (OPC)*.

The subject of the present paper is to show the results of some applications of the OPC on the indicated pressure cycles of two engines. Only the high pressure part of the engine cycle and especially the combustion period will be treated.

It will be shown that the OPC is a very good alternative of modelling and quick calculation of the wall heat losses during the combustion.

2. Basic milestones of calculation of the working cycle

The objectives of calculation can be roughly represented by means of the following scheme, Fig. 1.

The Vibe-Function, [5], allows the analytical representation of the heat release part of the working cycle i.e. modelling, Fig. 2.

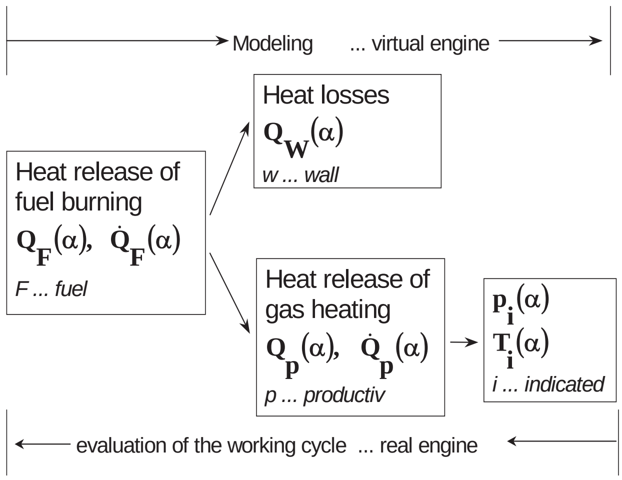


Fig. 1. Ways of calculation the HP-part of working cycle

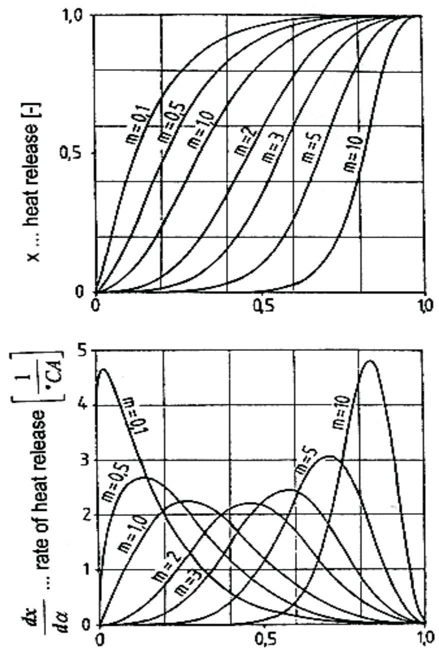


Fig. 2. Viebe function and its differentiation for different form factors "m"

$$x = 1 - e^{-k(\alpha/\alpha_C)^{m+1}}, \tag{1}$$

where:
x - part of the burned fuel energy,
k - factor for SI & Diesel engines k = 6.908,
α - time position of the combustion [°CA],
α_C - combustion duration [°CA],
$\alpha = 0$, $x = 0$ - start of combustion (SOC),
$\alpha = \alpha_C$, $x = 1$ - end of combustion (EOC),
m - Viebe factor.

By means of the variation of the Viebe-factor "m" different time-developments of combustion can be simulated. This function can be applied for modelling of fuel burning as well as for modelling of gas heating. There are applications with variable Viebe-factor, or "two-stage" Viebe-factor during a working cycle.

Another important step for the R & D of combustion in the IC-engines was the analytical estimation of the heat losses through the combustion chamber walls given by Woschni, [18], complemented and refined in numerous later works. This will be called a *Woschni Correlation (WOC).*

WOC gives a semiempirical expression for the instantaneous, locally averaged heat transfer coefficient "α_{WOC}":

$$\alpha_{WOC} = 130\, d^{-0.2} \cdot p^{0.8} \cdot T^{-0.53} \left[C_1 \cdot c_m + C_2 \cdot V_h \cdot \frac{T_1}{p_1 \cdot V_1} (p - p_0) \right]^{0.8}, \tag{2}$$

where:
d - cylinder bore,
C_1 - constant for the intake flow situation,
C_2 - constant for combustion chamber configuration,
c_m - average piston speed,
V_h - displacement per cylinder,
p_1, V_1, T_1 - thermic state at the beginning of compression,
p_0 - instantaneous pressure on the compression line.

The wall heat losses per computing interval result:

$$d\dot{Q}_W = \alpha_{WOC} \cdot A \cdot \Delta T \quad , \tag{3}$$

where:

A (α) - instantaneous wall surface,

ΔT(α) - temperature difference $\Delta T = T_{Gas} - T_{Wall}$.

The total wall heat losses in the crank angle interval (α_1 ... α_2) are:

$$Q_W(\alpha) = \int_{\alpha_1}^{\alpha_2} \dot{Q}_W(\alpha)\, d\alpha \, , \tag{4}$$

All these calculations give a global, locally averaged view on energy balance of the combustion chamber. It is so called *1-Zone-Model*.

For evaluation of the indicated pressure signals $p_i(\alpha)$, and for the statistical assessment of many working cycles the common indicating apparatus uses the gas equation (5) and 1st theorem of thermodynamics (6) to obtain the traces of $Q_p(\alpha), \dot{Q}_p(\alpha)$ and $T_i(\alpha)$ (eq·7)

$$T_2 = T_1 \frac{p_2 \cdot V_2}{p_1 \cdot V_1}, \quad 1 - \text{begin of interval}, 2 - \text{end of interval}\, , \tag{5}$$

$$dQ = Q_{1-2} = mc_V(T_2 - T_1) + p_m(V_2 - V_1)\, , \tag{6}$$

$$\dot{Q} = \frac{dQ}{d\alpha}, \quad Q(\alpha)\Big|_0^{\alpha i} = \int_0^{\alpha i} \dot{Q}(\alpha)\, d\alpha\, , \tag{7}$$

This calculation uses the function of crank mechanism and it needs usually an input or an assumption of the temperature at start of compression.

This type of quick, on-line-evaluations is very useful for diagnostics in the applied engine research.

3. Modelling of heat losses according to Oppenheim (OPC)

In the "Refinement of Heat Release Analysis", [28], the authors use a *2-Zone-Model* (2 components model) composed of reactants and products of the exothermic reaction, which is illustrated by means of Le Chatelier diagram.

With this model it is possible to calculate the energy fractions in consecutive CA-intervals according to Fig. 3.

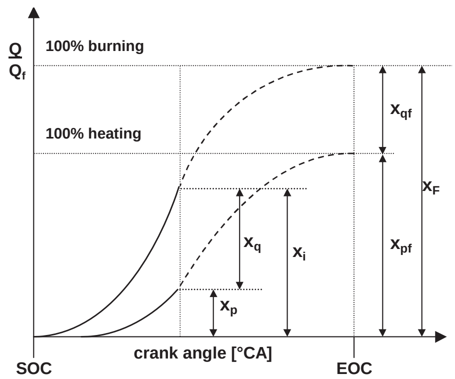


Fig. 3. Burning and heating during the combustion: x – energy fraction, F – fuel, f – final, p – productive (effective in gas, heating), q – nonproductive (wall heat losses, including blow-by losses), xi – fraction of burned fuel at successive computing interval

It was stated, that there is a correlation between the effective part of fuel energy x_p and its total amount x_i, which can be expressed by the equation:

$$\frac{x_i}{x_F}=1-\left(1-\frac{x_p}{x_{pf}}\right)^{\frac{1}{\sigma}}, \tag{8}$$

where σ is Oppenheim exponent.

The value x_F, the final fraction of fuel burned is always $x_F = 1$. Additionally to [28-30] x_F was introduced by the authors in the formulas (8), (9), (10) for easier translating to the absolute energy amounts by multiplying all energy fractions "x" with the final fuel energy Q_{Ff}.

As a result of several experimental studies on different engines the estimation of the exponent σ is given in [29] with the equation:

$$\sigma=\left(0.5-\frac{x_{pf}}{x_F}\right)\cdot\frac{x_{pf}}{x_F}+1.5, \tag{9}$$

After a further research this expression was changed in [30] to the following form:

$$\sigma=2-\left(\frac{x_{pf}}{x_F}\right)^{\frac{3}{2}}, \tag{10}$$

If the values x_p, x_{pf} and Q_{Ff} ($Q_{\text{Fuel final}}$) are known it is possible to obtain x_i at each computing interval and consecutively the wall heat losses:

$$x_q=x_i-x_p, \tag{11}$$

Two approaches of the Oppenheim Correlation (OPC) will be presented in this paper:

a) estimate of Q_{Ff} from measurement of the fuel consumption and estimate of x_p and x_{pf} from the standard heat release evaluation with the 1-zone-model. The results of this estimate will be marked as "OPC_{meas}" (measured).
b) estimate of x_p, x_{pf}, x_i and Q_{Ff} by means of the 2-zone-model from [28-30]. The results of this estimate will be marked as "$OPC_{calc\ UCB}$" (calculated at University of California Berkeley).

4. Engines and test procedure

The data of pressure indication of two engines were analyzed.

Engine 1

6 cylinder in line, IDI-Diesel, TCI, V_H = 21.7 dm^3, B/S = 160/180 mm, con rod 360 mm, compression ratio $\varepsilon = 16$, operating conditions: 1500 rpm / 1671 Nm, 1300 rpm / 1194 Nm.

With the data of this engine following calculations were performed:

- evaluation of the heat release $[p_i(\alpha) \rightarrow Q_p(\alpha)]$ with the 1-zone-model,
- evaluation of the wall heat losses $[Q_w(\alpha)]$ according to WOC,
- evaluation of the wall heat losses $[Q_w(\alpha)]$ according to OPC_{meas}.

Engine 2

6 cylinder in line, CNG, TCI, V_H = 7.8 dm^3, B/S = 115/125 mm, con rod 200 mm, compression ratio $\varepsilon = 11$, operating conditions: 2200 rpm / 810 Nm, 1600 rpm / 550 Nm.

The calculations carried out with the data of this engine were:

- evaluation of the heat release $[p_i(\alpha) \rightarrow Q_p(\alpha)]$ with the 1-zone-model,
- evaluation of the wall heat losses $[Q_w(\alpha)]$ according to OPC_{meas},
- evaluation of heat release and wall losses according to $OPC_{calc\ UCB}$.

5. Examples of the OPC-application

Engine 1

Figure 4 shows some results of the heat release evaluation at different operating conditions.

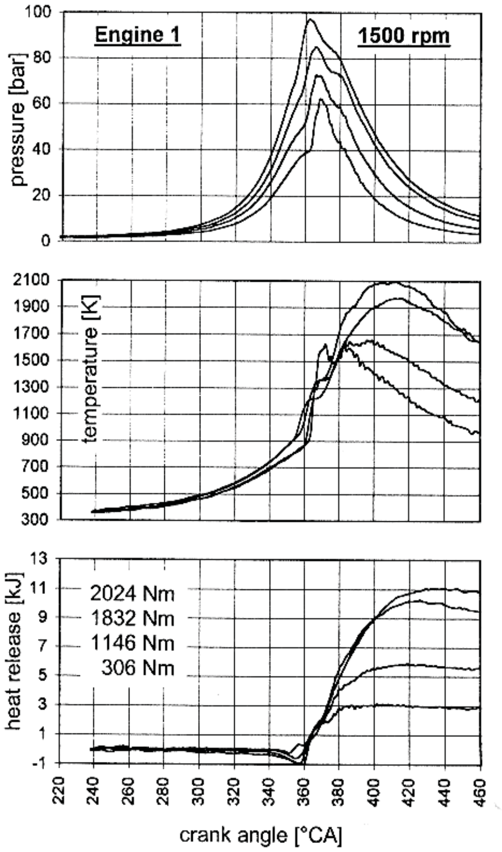


Fig .4. Pressure, temperature and heat release at different operating conditions

On the pressure traces of this engine there are different disturbances, which sometimes make the evaluation difficult.

There are pressure oscillations of the combustion as a result of the interference between the prechamber and the main combustion chamber.

Due to relatively long pressure indication orifice the presence and superposition of the "in pipe" oscillations may be assumed in certain cases.

Furthermore the presented pressure cycles are single cycles without filtering of parasite signals, or using smoothing algorithms.

Due to these pressure traces it wasn't possible to perform the calculation of heat release according to the 2-Zone-Model of UCB [28]. This calculation was performed with the 1-Zone-Model research program of the laboratory for IC-engines (Biel-Bienne), [31, 32], which will be designed with the abbreviation AFHB (see nomenclature).

In this program is included an evaluation of wall heat losses according to WOC, which needs a knowledge of a compression line at each operating point.

The calculation of wall heat losses according to OPC was also programmed, [33]. Some chosen examples of the results are represented in Fig. 5 and 6.

It can be stated, that OPC considerations only the combustion period, while the WOC allows also to calculate the heat, which is transferred to the wall during the compression and during the expansion stroke. Since the main purpose of OPC is the control of combustion process of engine, which is the reason why OPC does not take into account the other parts of the cycle.

The integrated values of the wall heat at the end of combustion which are calculated with both methods correlate very well.

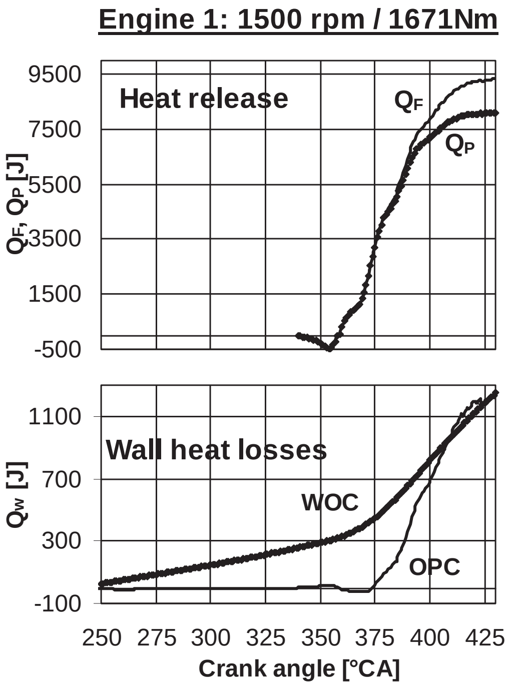


Fig. 5. Heat release and wall heat losses

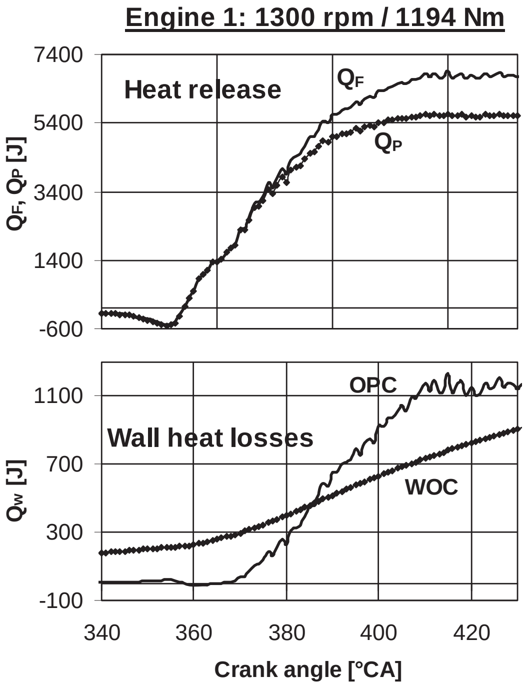


Fig. 6. Heat release and wall heat losses

By means of varying the TDC-mark, or the temperature at the beginning of compression it was stated, that the OPC, which is closely connected to the heat release function, is more sensitive to those varying parameters.

Engine 2

The combustion pressure indication system on this engine was a standard apparatus DATAC (Kistler-COM), which allows the averaging of several working cycles, statistical analysis and the heat release calculation.

The measured pressure data could be exported from the DATAC and treated with the AFHB-laboratory research program mentioned above. The calculation of wall heat losses according to OPC was included in this program. The introduction of WOC for this engine can be an objective of further study.

Figure 7 and 8 shows the results at two operating points of the engine. The original pressure cycles were very smooth comparing to the Diesel engine 1. In addition to that averages of 10 pressure cycles were taken into the calculation.

Again it can be remarked, that the OPC gives the information about the wall heat transfer only during the combustion period.

If necessary to estimate the heat transfer during the other phases of the engine working cycle different approaches have to be investigated:

- in the compression stroke, where $p - p_0 = 0$, see equation (2), this equation for heat transfer coefficient becomes very simple,
- in the expansion stroke it is possible to find a simplified approach for a definite family of engines - without necessity of the compression line (p_0), but with an experimental support.

If it would be a question of a quick on-line calculation of the working cycle and of the wall heat losses for the purposes of transient regulation of the engine, so it is most important to have more accurate information about the combustion period. The variations of the wall heat flow of the combustion period in the entire engine operation map are much bigger and have more importance for the total wall heat losses, than the heat transfer during the other periods of the working cycle.

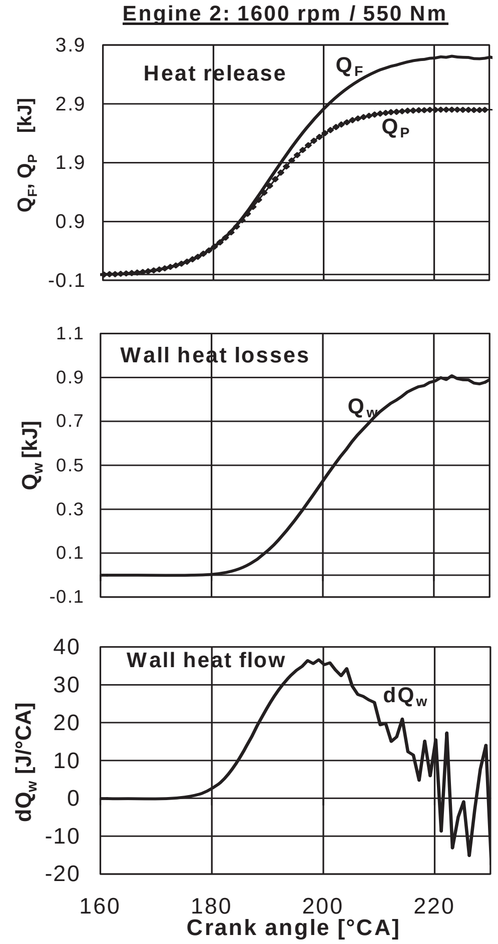


Fig. 7. Heat release and wall heat losses

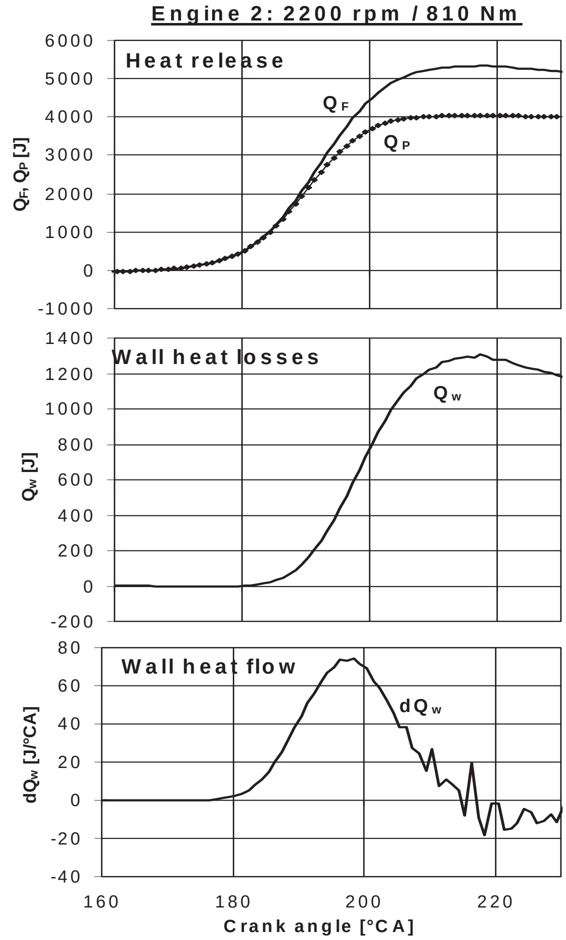


Fig. 8. Heat release and wall heat losses

What is the influence of the different estimates of the Oppenheim exponent σ, equation (9) σ_1 and equation (10) σ_2. Both functions are represented in Fig. 9.

In the range of utilization of those functions, where the quotient of productive final energy and final fuel energy has usually the values between 70 and 85%, the difference σ_1 and σ_2 is very little. This has also a very little influence on the results, which is shown in Fig. 11. The maximum values of heat release (x_{pf}) in this figure were taken from the standard evaluation of DATAC indication apparatus. It was stated during the work, that there are slight differences between the results of the 1-zone-model programs AFHB and DATAC. This is a topic for further research and development of those programs.

Some data of the engine 2 were evaluated at the UCB with the 2-zone-model of [28-30].

An example of the energy fraction of fuel burned x_F (x_{iF}) and of the productive energy part xp is given in Fig. 10. These energy fractions were represented as absolute values and compared with those resulting from DATC-evaluation. These comparisons, as well as the wall heat losses are depicted in Fig. 12.

There is a little difference between the maximum values of Q_p (gas heating function) calculated with the two methods, which obviously causes the difference of Q_w. Nevertheless the correspondence of those results is very good.

Another difference, which was compensated by applying the values of x_{iF} and x_p from Fig. 10, is the combustion duration.

For the engine operation point 2200 rpm / 810 Nm the combustion duration indicated by Q_p in Fig. 12 is about 41°CA, while in Fig. 10 results about 37°CA. A method of estimation of the initial and final point of combustion by means of the polynomial straight lines on the log p – log V – graph is shown in Fig. 13. The very exact finding, particularly of the EOC ("f"), is not possible. This finding is also not necessary, because point "f" is not the point where the combustion stops – it is simply the point where combustion stops doing any useful work – i.e. this is the end of *Exothermic Process of Combustion*.

This problem is commonly known and therefore the standard indicating apparatus defines usually the SOC and EOC as convention at certain percentage of converted energy.

Certain differences of combustion duration, as those mentioned above have to be accepted.

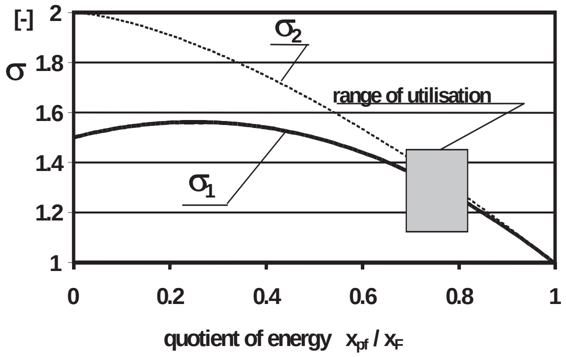


Fig. 9: Estimate of Oppenheim Exponent σ

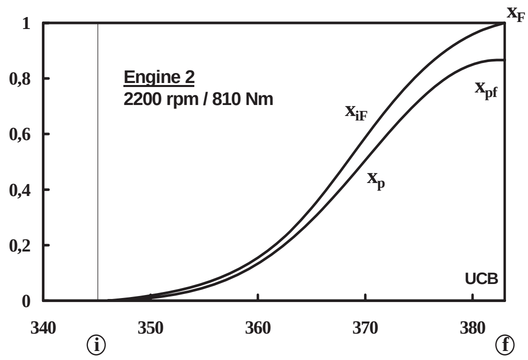


Fig. 10. Fuel burned and productive energy portion during the combustion

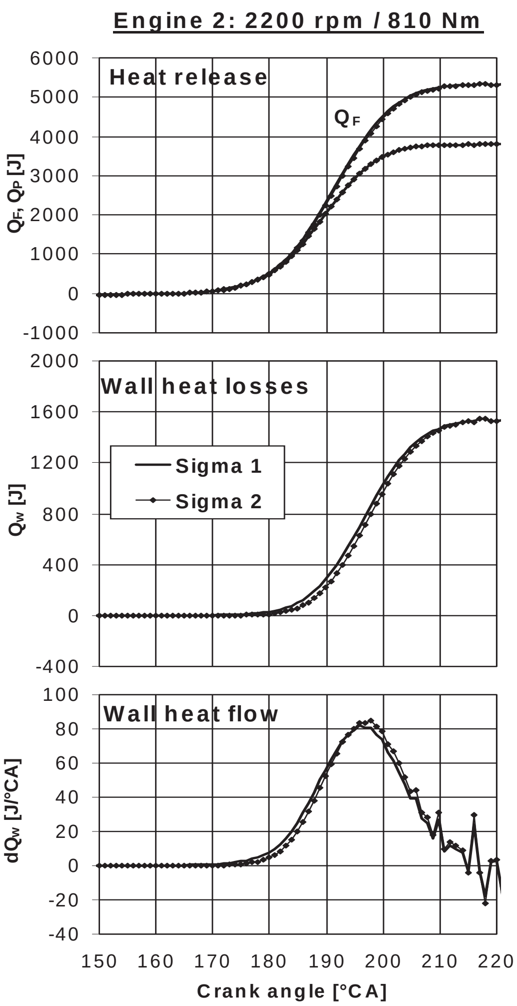


Fig. 11. Influence of the SIGMA algorithm on the results with OPC

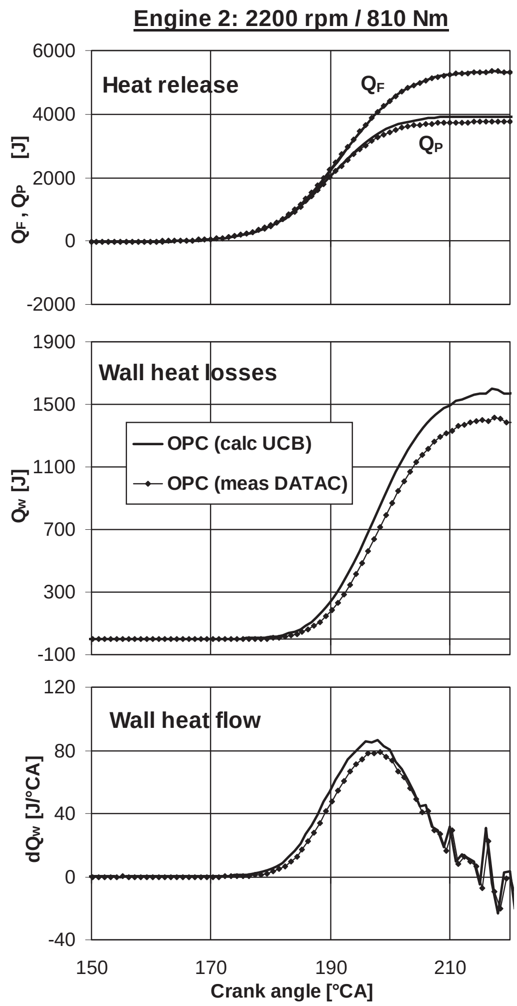


Fig. 12. Comparison of two evaluation methods of heat release with OPC

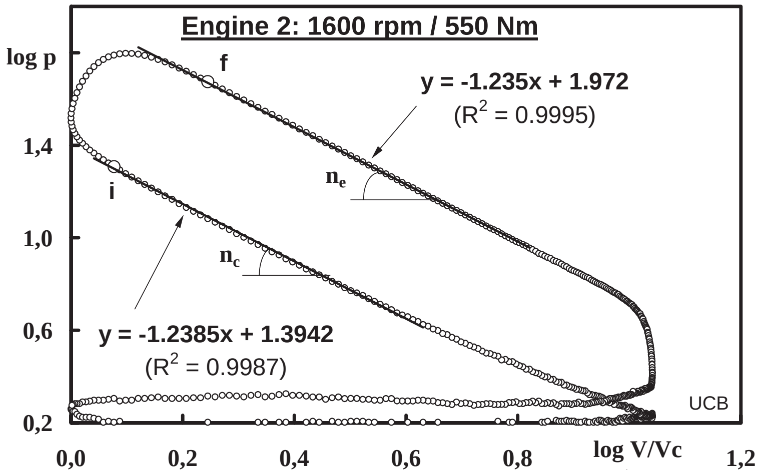


Fig. 13. Logarithmic p-V-diagram and estimate of the initial „i" (SOC) and final „f" (EOC) points of combustion

6. Conclusion

There are several ideas to use low cost pressure sensors, which can be included in the gasket, in the spark plug or otherwise, to control "on line" the combustion in each cylinder of the engine. The exceptional technological progress in the domain of electronic regulation makes these ideas ever more realistic.

The possibility of on-line-estimation of the heat flow from the combustion chambers to the cylinder head could be a very interesting input e.g. for the control of a variable cooling system. For this purpose the OPC would be much easier to handle and to calibrate, than the usual algorithm of WOC.

The integrated values of the wall heat at the end of combustion which are calculated with both methods correlate very well.

The OPC gives information about the wall heat transfer during the most important period, which is the period of combustion.

The different algorithms for exponent σ have no significant influences on the results.

The results of heat release from the 2-zone-model correspond very well with those from the standard 1-zone-model.

The OPC due to its simplicity and exponential character, has a similar sense and application for the wall heat losses, as the Viebe-function for the heat release.

The numerical precision of the wall heat losses estimation with OPC in the combustion period depends on the precision of heat release and fuel burning function. The wall heat transfer in the entire engine working cycle is believed to correlate with the wall heat of the combustion period.

The tentatives to find out this correlation can be pursuit on theoretical or experimental way. The work in this matter can be very useful to compare engine concepts, identify boundaries of operation and build a general understanding of engine system behaviour.

The OPC can be recommended to the engine community as a good alternative of modelling and calculation of the heat losses trough the combustion chamber wall.

Nomenclature

Parameters and abbreviations:

AFHB - Abgasprüfstelle der Fachhochschule Biel – Laboratory for Exhaust Emission Control of the University of Applied Sciences, Biel-Bienne, Switzerland,

α - crank angle [°CA], heat transfer coefficient [W/m^2·K],

α_c - combustion duration,

B - bore,

CA - crank angle,
C_1, C_2 - constants for the WOC,
c_m - average piston speed,
c_v - specific thermic capacity at constant volume [J/kg·K],
ε - compression ratio,
EOC - end of combustion,
HP - high pressure,
ILOT - Institute of Aviation (Instytut Lotnictwa), Warsaw, Poland,
m - Viebe factor, mass,
n_c - polytropic exponent of compression,
n_e - politropic exponent of expansion,
σ - Oppenheim exponent,
OPC - Oppenheim Correlation,
p_i - indicated pressure,
p_m - average pressure in the computing interval,
p_0 - compression line pressure,
Q, $\dot{Q}$ - heat [J], heat flow [J/deg],
S - stroke,
SOC - start of combustion,
TDC - top dead centre,
T_i - instantaneous combustion temperature,
UCB - University of California Berkeley,
V - volume,
V_H - engine displacement,
WOC - Woschni Correlation,
x - part of burned fuel energy,
$x_i = x_{iF}$ - fraction of burned fuel at successive computing interval,
$x_F = x_{Ff}$ - final fraction of burned fuel.

Indexes:
F - fuel,
f - final,
w - wall,
p - productive,
q - nonproductive,
i - initial; computing interval number.

References

[1] Müller, H., Almstadt, K., *Die Entflammungsphase im Ottomotor – Dauer und Streuung in Abhängigkeit vom Betriebspunkt*, MTZ 43, 4, S. 149, 1982.
[2] Bartlmä, F., *Gasdynamik der Verbrennung*, Springer-Verlag, Wien–New York 1975.
[3] Kort, R. T., Mansouri, S. H., Heywood, J. B., Ekchian, A., *Divided – chamber diesel engine, part II: experimental validation of a predictive cycle – simulation and heat release analysis.* SAE Techn. Paper 82 02 74, Sloan Automotive Laboratory, Massachusetts Institute of Technology.
[4] Sitkei, G., *Kraftstoffaufbereitung und Verbrennung bei Dieselmotoren.* Ingenieurwissenschaftliche Bibliothek, Springer-Verlag, Berlin 1984.
[5] Viebe, I., *Brennverlauf und Kreisprozess von Verbrennungs-motoren*. VEB Verlag Technik, Berlin 1970.
[6] Lang, K., *Berechnung von Druck-verlauf und Wirkungsgrad im Verbrennungsmotor,* MTZ 30, 5, S. 173, 1969.

[7] Müller, H., Berling, H., *Program-mierte Auswertung von Druck-verläufen in Ottomotoren*. VDI Fortschrittbericht Reihe 5, Nr. 30.

[8] Songo, S., Tadahide, T., Hidetaka, N., Toshiaki, K., *Statistical analysis of pressure indicator data of an internal combustion engine*. SAE Technical Paper 770882, Toyota Motor Co. Ltd.

[9] Müller, H., Bertling, H., Haahtela, O., *Die Entflammungsdauer und ihre Auswirkung auf den Verlauf der Energieumsetzung beim Otto-motor*, MTZ 39, 7/8, S. 333, 1978.

[10] Lange, W., Woschni, G., *Thermo-dynamische Auswertung von Indikatordiagrammen, elektronisch gerechnet*, MTZ 25, 7, S. 284, 1964.

[11] Woschni, G., *Elektronische Berechnung von Verbrennungs-motor – Kreisprozessen*, MTZ 26, 11, S. 439, 1965.

[12] Woschni, G., *Engine cycle simulation – an effective tool for the development of medium speed diesel engines*. SAE Paper 870570, SAE Trans 96.

[13] Pischinger, R., Krassnig, G., Taucar, G., Sams, T., *Thermodynamik der Verbrennungs-Kraftmaschine*, 4610, Springer-Verlag, Vienna 1992.

[14] Primus, R. J., *Visual Thermo-dynamics: Processes in Log(p)-Log(T) Space*, SAE Paper 1999-01-0516.

[15] Barba, C., Burkhardt, C., Boulouchos, K., Bargende, M., *Ein empirisches Modell zur Voraus-berechnung des Brennverlaufes beim PkW-Common-Rail-Dieselmotor*, MTZ 60, 4, S. 262, 1999.

[16] Woschni, G., *Universally applicable equation for the instantaneous heat transfer coefficient in the internal combustion engine.* SAE Paper 670931, SAE Trans. 76.

[17] Grossmann, D., *Beitrag zum instationären Wärmeübergang in einem Viertakt-Ottomotor*. Dissertation Technische Universität Braunschweig 1969.

[18] Woschni, G., *Die Berechnung der Wandverluste und der thermischen Belastung der Bauteile von Dieselmotoren*, MTZ 31, 12, S. 1970.

[19] Woschni, G., Fieger, J., *Experminentelle Betimmung des örtlich gemittelten Wärmeüber-gangskoeffizienten im Ottomotor*, MTZ 42, 6, S. 229, 1981.

[20] Woschni, G., Fieger, J., *Expermiental Investigation of the Heat Transfer at Normal and Knocking Combustion in Spark Ignition Engines*, MTZ 43, 2, S. 63, 1982.

[21] Stiepe, K., Polej, A., *Brennraum-seitige örtliche thermische Rand-bedingungen für Verbrennungs-motoren*, MTZ 59, 7/8, S. 500, 1998.

[22] Woschni, G., Klaus, B., Zeilinger, K., *Untersuchung des Wärme-transportes zwischen Kolben, Kolbenringen und Zylinderbüchse*, MTZ 59, 9, S. 556, 1998.

[23] Pivec, R., Sams, T., Wimmer, A., *Wärmeübergang im Ein- und Auslasssystem*, MTZ 59, 10, S. 658, 1998.

[24] AVL-Germany, Technical University Darmstadt, 3rd International Indicating Symposium. Proceedings, Darmstadt 1998.

[25] Institute for IC-Engines, Technical University Graz, Austria, 6th Symposium The Working Process of the Internal Combustin Engine, Proceedings, Graz 1997.

[26] Oppenheim, A. K., Kuhl, A. L., *Paving the Way to Controlled Combustion Engines (CCE)*, SAE Paper 95961, 1995.

[27] Oppenheim, A. K., Kuhl, A. L., Packard, A. K., Hedrick, J. K., Johnson, W. P., *Model and Control of Heat Release in Engines*, SAE Paper 960601, 1996.

[28] Oppenheim, A. K., Barton, J. E., Kuhl, A. L., Johnson, W. P., *Refinement of Heat Release Analysis*, SAE Paper 970538, 1997.

[29] Oppenheim, A. K., Kuhl, A. L., *Thermodynamics of a Closed Combustion System*, University of California, Berkeley and LLNL, Livermore.

[30] Oppenheim, A. K., Spektor, R., Kuhl, A. L., *Thermostatics and Thermo-kinetics of Closed Combustion Systems*, University of California, Berkeley and LLNL, Livermore.

[31] Weilemann, R., *Messungen und thermodynamische Analyse des MAN_Dieselmotors.* Diplomarbeit Verbrennungsmotoren, Ingenieur-schule Biel, August-September 1996.

[32] Stadler, J., *Analyse des Aufladesystems und der Wand-wärmeverluste im Kennfeld des MAN-Dieselmotors*, Projektarbeit Verbrennungsmotoren, Ingenieurschule Biel, 97/98.
[33] Prochazka, L., Rütter, J., *Untersuchungen des Motorbetriebes mit CNG*. Projektarbeit Verbrennungsmotoren, HTA Biel, 98/99.