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Correlation of Exhaust Smoke to Significant Operation Variables in a Direct Injection Diesel Engine

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ABSTRACT. In this study a simple quantitative correlation for predicting exhaust smoke level from a direct injection diesel engine has been developed. The effects of different engine operating parameters on exhaust smoke level have been investigated experimentally. The experimental results showed that exhaust smoke is significantly affected by changes in the diffusion burning fraction, engine speed and overall equivalence ratio. As a result a simple quantitative formula which relates exhaust smoke to these variables is presented.

The diesel engine has a poor public image because of its visible soot (smoke) emission and sometimes objectionable odour. Many theories of smoke formation were hypothesized in the literature. A study of these theories revealed that no general agreement has been found as to a single particular mechanism by which smoke formation can be explained (Tan 1975). Sampling methods showed that soot particulates form at the period of rapid combustion (Fujiwara *et al.* 1984) in the combustion chamber and are further reduced by oxidation. An increase in the excess air ratio is found to result in a significant reduction in exhaust smoke density (Ishida *et al.* 1988).

Certain fuel properties have been known to affect the soot formation process in flames. Improvements of smoke emissions were obtained in high-speed diesel engines by fuel heating and blending with a range of low viscosity fuels (Murayama *et al.* 1988). The prevailing conditions of the burning process such as the pressure, temperature, and overall equivalence ratio also affect smoke formation. The tendency to liberate smoke has been found to increase with pressure rise (McArragher and Tan 1972). The equivalence ratio was found to be an important factor in relation to soot release as reported by Janota *et al.* 1977.

Early trials at diesel combustion modelling were concerned with the calculation of an apparent heat release rate. Some of these models have been developed and extended to include emission formation models (Hiroyasu and Kadota 1976, Khan *et al.* 1971). These models are concerned with modelling soot formation and combustion inside the combustion chamber through the use of kinetic data and semi-empirical correlations.

Bryzik (1976) developed quantitative relationships between diesel engine exhaust smoke emission and significant operating variables. He postulated a third-order model to describe exhaust smoke level as a function of air/fuel ratio, fuel injection timing and engine speed.

Materials and Methods

The experimental results of this paper were obtained during a study of the effect of different engine operating parameters (such as engine speed, overall equivalence ratio, inlet air temperature, inlet air pressure and dynamic injection timing) on exhaust smoke level. A description of the experimental setup, instrumentation and results of this study were reported by Abdalla (1983).

The engine used was a single cylinder, direct injection, 4-stroke diesel engine. The smoke intensity was measured by a Hartridge smokemeter in Hartridge Smoke Units (HSU).

Analytical Formulation of Heat Release Rate

The calculation of the rate of heat release from experimental cylinder pressure diagrams is based on the application of the first law of thermodynamics during the closed cycle period. The first law relates the rate of heat and work transfer to the change in internal energy. The computation of a realistic burning rate curve requires accurate experimental cylinder pressure data with knowledge of instantaneous cylinder volume, a heat transfer model, a knowledge of gas composition and properties and the total mass of gas in the cylinder (Marzouk and Watson 1976).

In order to express the shape of the actual fuel burning rate curves in a manageable analytical form, a substitute burning rate function is introduced. The most widely used analytical expression which describes the pattern of the rate of burning is the Wiebe (1956) function. This is expressed as follows:

$$\frac{m_{f,b}}{m_{f,t}} = 1\text{-}\exp\left(-CW_1 \cdot \tau^{cw_2}\right) \tag{1}$$

Marzouk (1976) developed a flexible mathematical function based on experimental data obtained from a turbo-charged diesel engine. The heat release period was considerd to take place essentially in two parts: the premixed combustion phase which is associated with the early part of the post-ignition process, and a diffusion combustion phase. The two modes of burning were assumed to start at the ignition point and to proceed simultaneously for a small part of the heat release period during which the combustible mixture prepared prior to ignition was totally consumed. Following this the fuel was assumed to burn as it entered the combustion chamber in accordance with a rate known as diffusion burning rate.

Consequently the instantaneous burning rate was expressed mathematically as the sum of two quantities represented by two mathematical functions:

$$\frac{dm_{f,t}}{d\theta} = \frac{dm_{f,p}}{d\theta} + \frac{dm_{f,d}}{d\theta} \tag{2}$$

The amount of fuel burning in a diffusion flame is thought to be an important factor as far as smoke release in a diesel engine is concerned (Khan 1970).

A similar analogy to that used by Marzouk (1976) is employed here to define a diffusion burning fraction, α, as the ratio between the cumulative fuel consumed by diffusion burning ($m_{f,d}$) to the total amount of fuel injected per cycle ($m_{f,t}$):

$$\alpha = \frac{m_{f,d}}{m_{f,t}} \tag{3}$$

Using this definition a non-dimensional form of fuel burning rate is obtained as follows:

$$R_t(\tau) = (1\text{-}\alpha).\ R_p(\tau) + \alpha.\ R_d(\tau) \tag{4}$$

where:

$$R_k(\tau) = \frac{\psi}{m_{f,k}} \cdot \frac{dm_{f,k}}{d\theta} \quad \text{where } k=t,p,d \tag{5}$$

$$\tau = \frac{\theta - \theta_i}{\psi}$$

In this way the rate of burning is expressed as the sum of two component distributions weighted by the diffusion burning fraction (α).

Extensive trials employing different mathematical functions were carried out before arriving at the following two functions which best represented the experimental data:

For premixed burning:

$$R_p(\tau) = C_1 \cdot C_2 \cdot \tau^{C_2 - 1} \cdot \exp(-C_1 \cdot \tau^{C_2}) \tag{6}$$

For diffusion burning:

$$R_d(\tau) = C_3 \cdot C_4 \cdot \tau^{C_4 - 1} \cdot \exp(-C_3 \cdot \tau^{C_4}) \tag{7}$$

From the processed heat release data, reliable burning rate curves were established for the different test conditions (see Fig. 1). The shape parameters (C_1, C_2, C_3 and C_4) and α in equation (4) were evaluated for each burning rate curve using a non-linear curve fitting routine.

The following relation was established from the experimental results and expresses the diffusion burning fraction as a function of equivalence ratio and ignition delay:

$$\alpha = 1.155 \frac{\phi_0^{0.461}}{(ID)^{0.449}} \tag{8}$$

Similarly the ignition delay was represented by the following relation:

$$ID = 3.227 \cdot \frac{\exp.(2100/T)}{P^{1.14}} \tag{9}$$

where

T : mean temperature during ignition
P : mean pressure during ignition

Exhaust Smoke Correlation

Exhaust smoke intensity is found (Abdalla 1983) to be influenced by engine operating variables such as overall equivalence ratio, engine speed, intake air temperature and pressure and dynamic injection timing. Alterations of dynamic

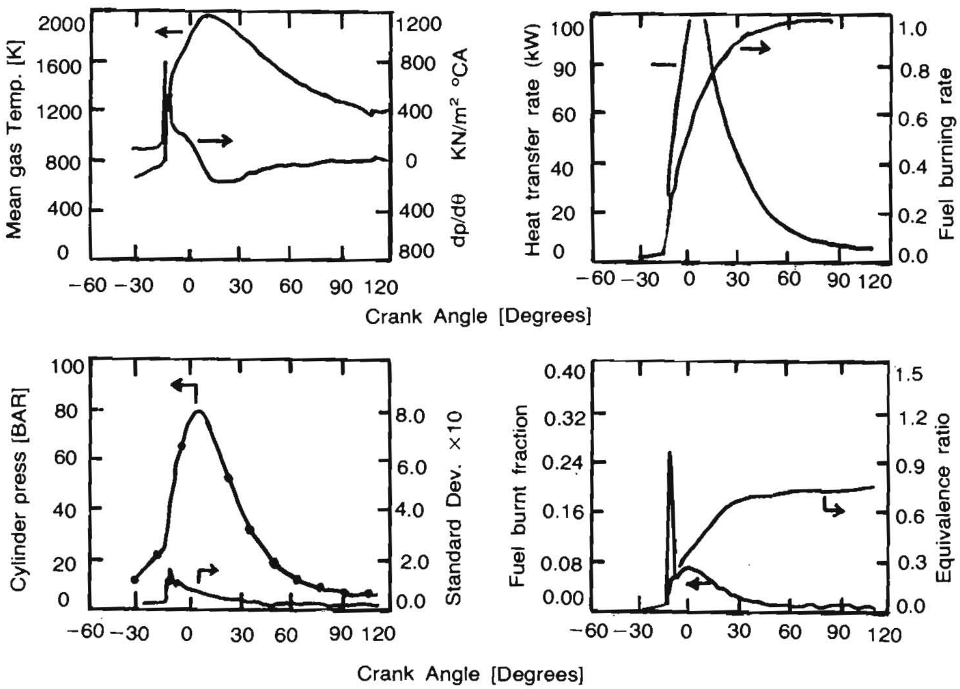

Fig. 1. Typical results from burning rate computations (N = 800 rev/min, $\phi_0 = 0.82$, injection timing = 28.21°BTDC).

injection timing, intake air temperature and pressure are found to produce significant changes in the ignition delay period which in turn is related to diffusion burning fraction (equation 8). However, the present study showed that exhaust smoke release is significantly influenced by the proportion of fuel burning as a diffusion flame (Fig. 2), equivalence ratio (Fig. 3) and engine speed (Fig. 4).

From the experimental results exhaust smoke is found to be best represented to the above variables by the following relationship:

$$\frac{S}{S_{ref}} = 0.2763 \left[\frac{N}{N_{ref}}\right]^{1.091} \cdot \left[\frac{\phi_0}{\phi_{0ref}}\right]^{1.832} \cdot \exp\left[1.1091 \frac{\alpha}{\alpha_{ref}}\right] \tag{10}$$

In the present work, the test condition that gave the maximum Break Mean Effective Pressure (BMEP) is chosen as the reference point. Comparison between calculated and measured exhaust smoke at different test conditions is presented in Figs. 3-7.

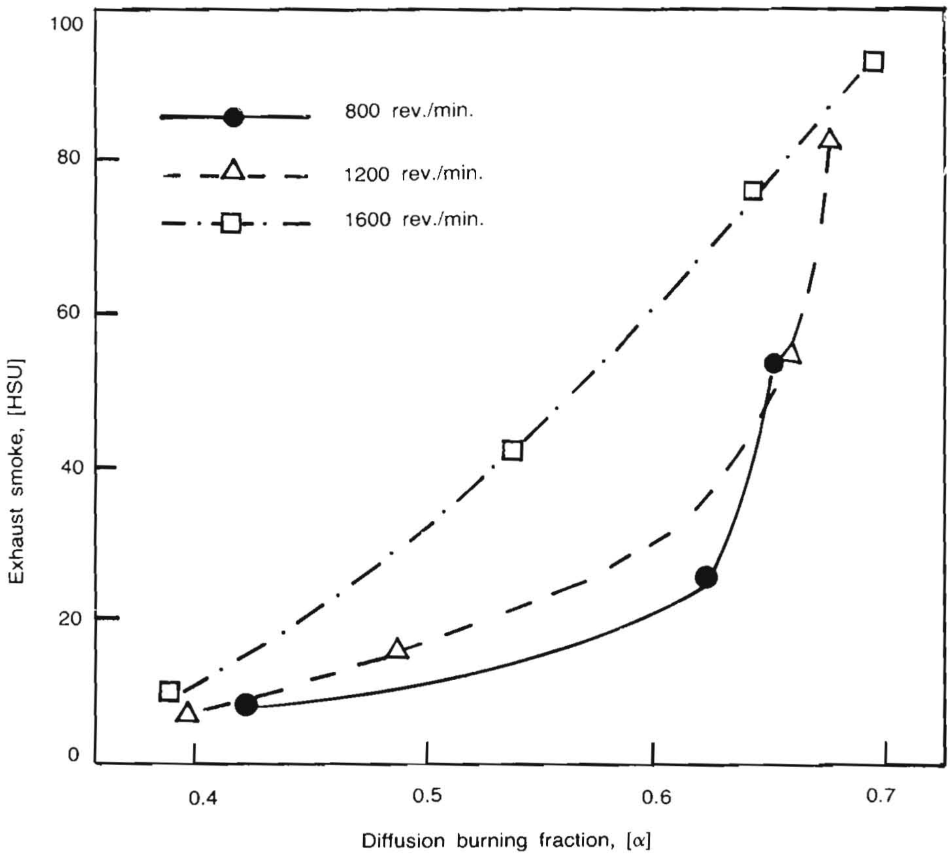

Fig. 2. Effect of diffusion burning fraction and engine speed on exhaust smoke at naturally aspired engine conditions.

The smoke correlation described above seems to predict changes in exhaust smoke level at different conditions reasonably well with an average error of $\pm$ 14%.

Conclusion

It may be concluded that the simple exhaust smoke correlation presented in this paper is useful in predicting changes in exhaust smoke level due to alterations in engine operational variables such as equivalence ratio, speed, inlet air temperature and pressure and fuel injection timing.

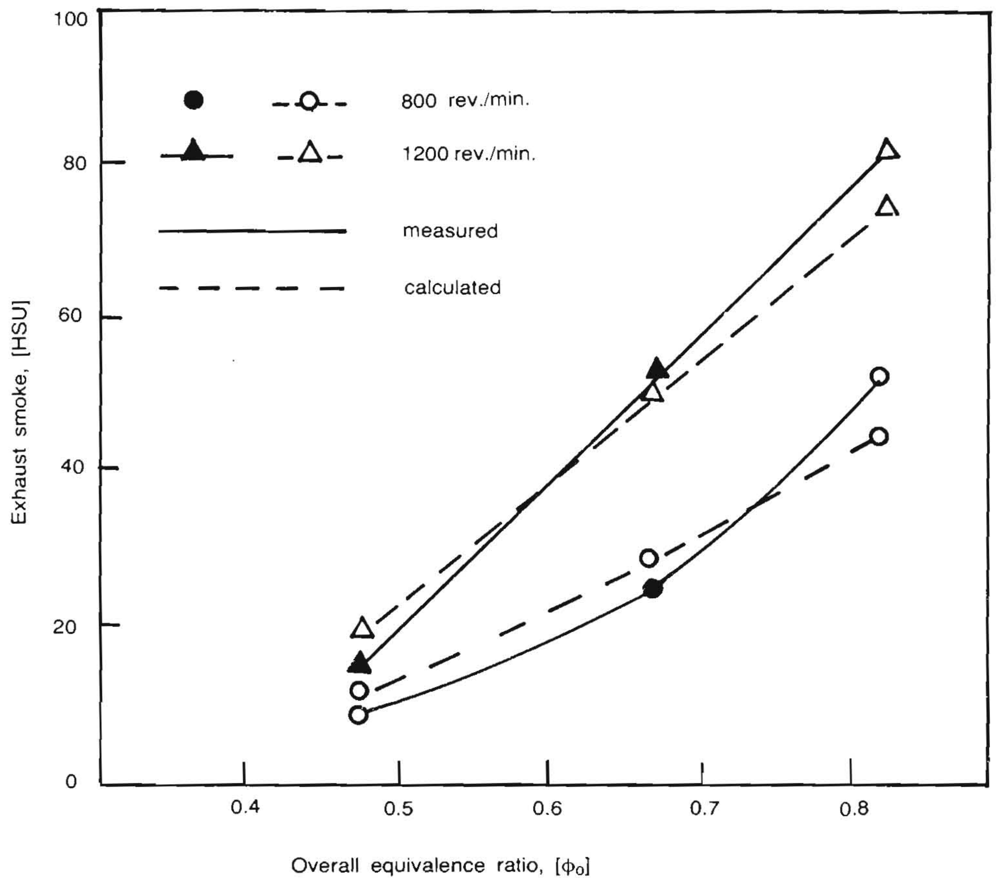

Fig. 3. Calculated and measured exhaust smoke versus equivalence ratio at different engine speeds.

Exhaust smoke emission is found to be significantly influenced by the proportion of fuel burning in a diffusion mode. A reduction in smoke level could be achieved if a greater fraction of the total fuel is injected and premixed before ignition occurs.

The derived smoke correlation could be incorporated into an existing performance simulation program to predict exhaust smoke level during transient as well as steady state operation of a direct injection diesel engine.

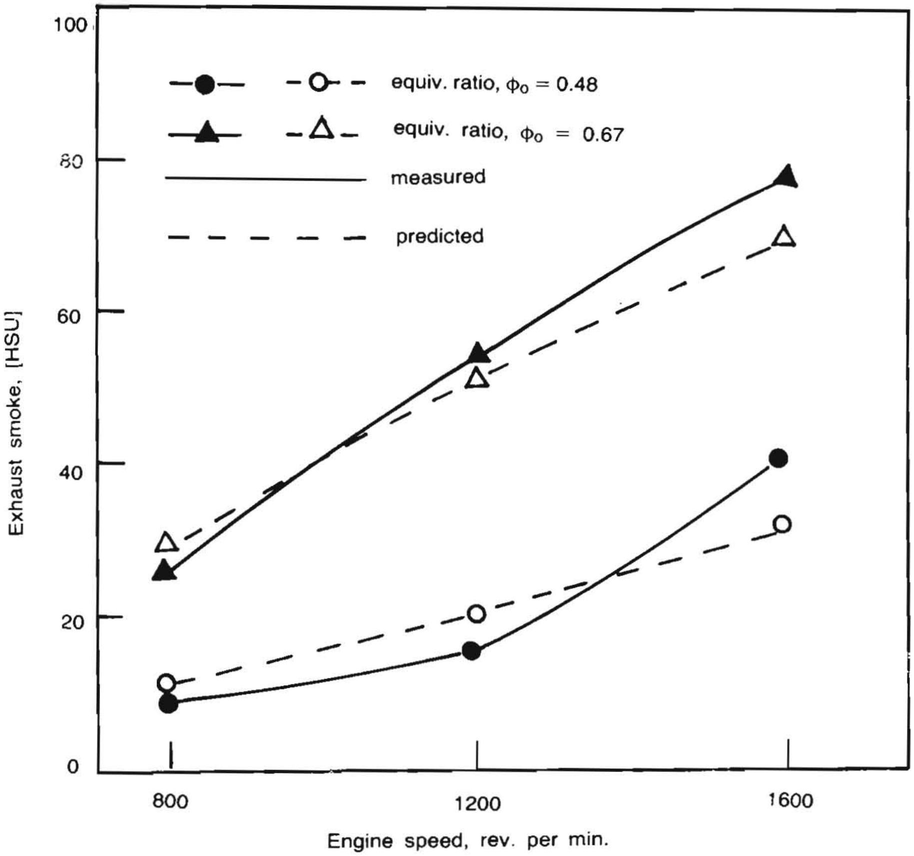

Fig. 4. Calculated and measured exhaust smoke versus engine speed at different equivalence ratios.

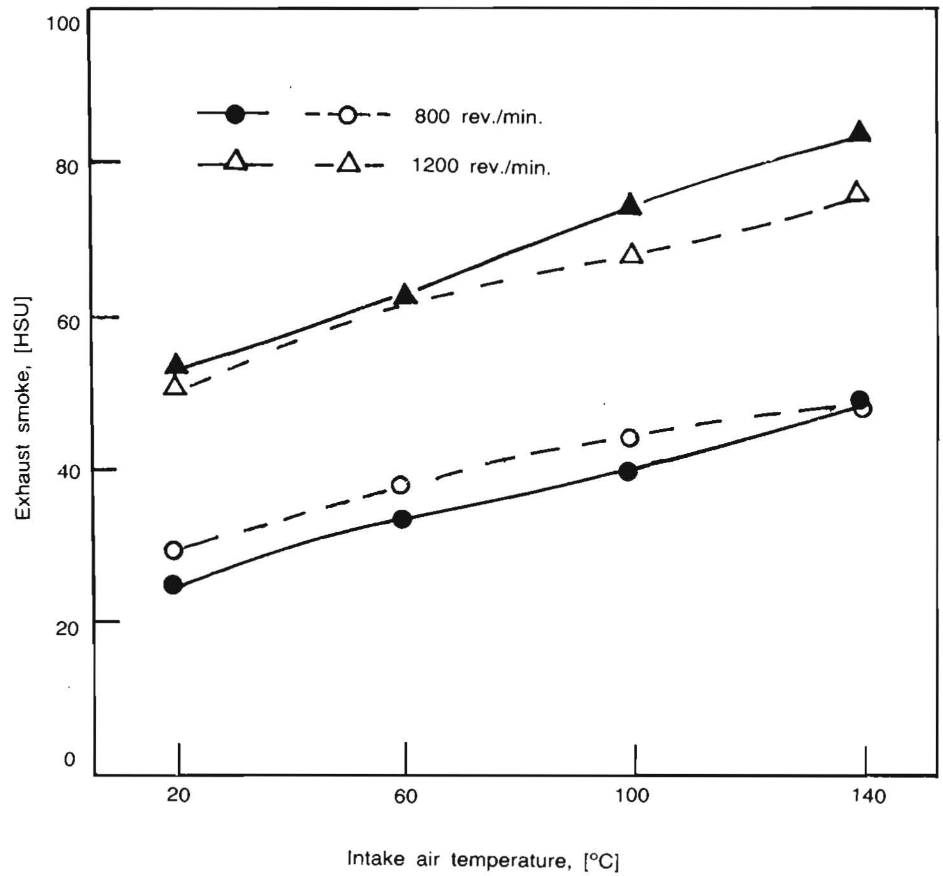

Fig. 5. Calculated and measured exhaust smoke versus air temperature. ϕ_0 = 0.67, injection timing = 28.21° BTDC.

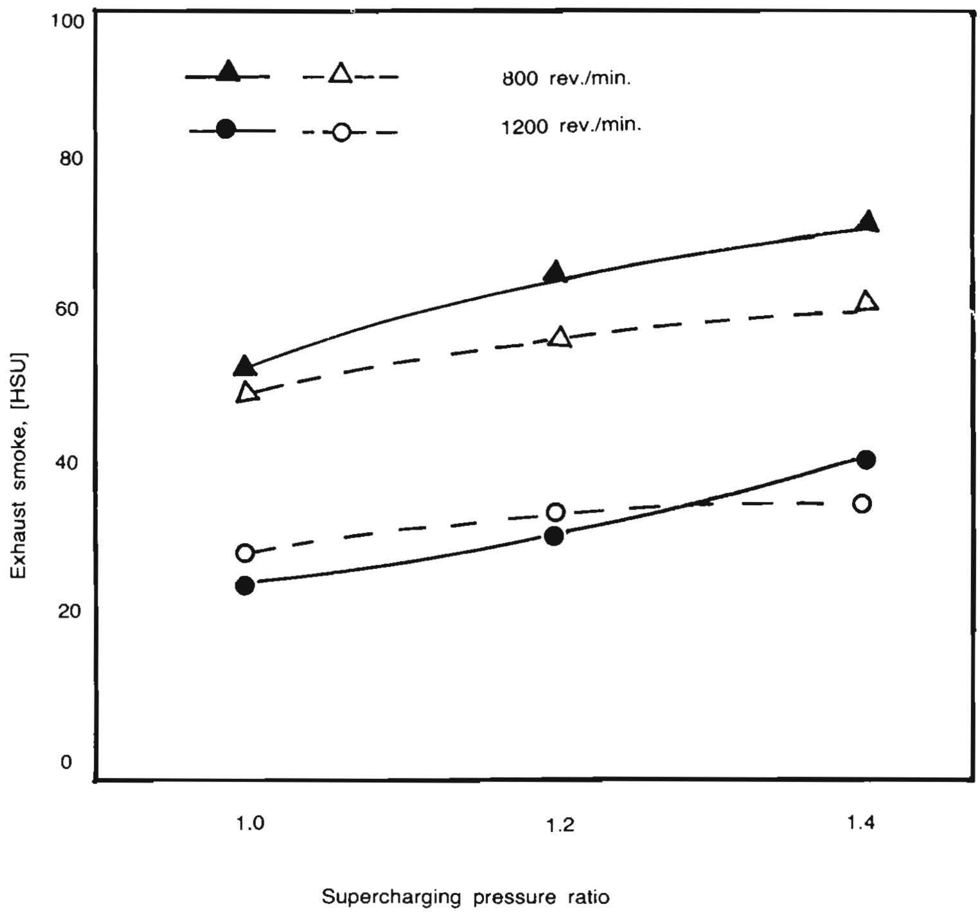

Fig. 6. Calculated and measured exhaust smoke versus supercharging pressure ratio. ϕ_0 = 0.67, injection timing = 28.21° BTDC.

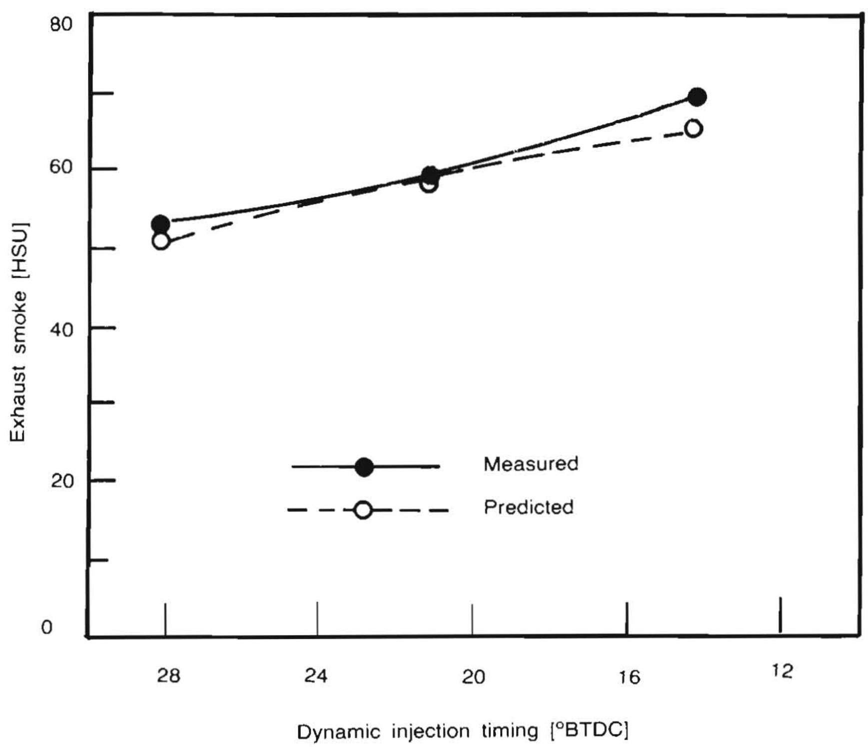

Fig. 7. Calculated and measured exhaust smoke versus dynamic injection timing. Engine speed = 1200 rev./min. and ϕ_0 = 0.67.

Notation

C_1, C_2	Wiebe shape factors for premixed burning
C_3, C_4	Wiebe shape factors for diffusion burning
CW_1, CW_2	Constants
R	Fuel burning rate
HSU	Hartridge Smoke Units
ID	Ignition delay
m	mass, kg
N	Engine speed, rev/min
P	Pressure, bar
S	Exhaust smoke, HSU
T	Temperature, K
α	Diffusion burning fraction
ψ	Burning duration, CA degrees
ϕ_0	Overall equivalence ratio
τ	Normalized crank angle
θ	Crank angle, degrees

Subscripts

b	burned
d	diffusion
f	fuel
i	ignition
p	premixed
ref	reference
t	total

References

Abdalla, K.N. (1983) Effect of significant operating variables on exhaust smoke in a direct injection diesel engine, *Sudan Eng. Soc. Journal,* **25:** p. 22.

Bryzik, W. (1976) *Relationships between exhaust smoke emissions and operating variables in diesel engines,* Doctoral Dissertation, Univ. of Detroit.

Fujiwara, Y., Fukazawa, S., Tosaka, S. and **Murayama, T.** (1984) Formation of soot particulates in the combustion chamber of a prechamber type diesel engine, *SAE* SP-578, p. 147-156.

Gaydon and **Wolfhard** (1960) Flames, Chapman and Hall Publishers.

Hiroyasu, H. and **Kadota, T.** (1976) Models for combustion and formation of NO and soot in a direct injection diesel engine, *SAE* Paper No. 760129.

Ishida, M., Izumi, S. and **Yoshimura, Y.** (1988) Studies on combustion and exhaust emissions in a high speed diesel engine, Nippon Kikai Gakkai Ronburshu, B Hen **54**(498): 506-512.

Janota, M.S., Crookes, R.J., Daie, S.J. and **Nazha, M.A.** (1977) Soot and gaseous pollutant formation in a burning fuel spray in relation to pressure and air/fuel ratio, *J. Inst. cf Fuel,* **50**(402).

Khan, I.M. (1970) Formation and combustion of carbon in a diesel engine, *Inst. of Mech. Engineers,* Diesel Engine Combustion Symposium, p. 56.

Khan, I.M., Greeves, G. and **Probert, D.M.** (1971) Prediction of soot and NO concentration in diesel engines exhaust, *Inst. of Mech. Engineers,* No. C142/71.

Marzouk, M. (1976) *Transient response of turbocharged diesel engines,* Ph.D. Thesis, Imperial College, London University.

Marzouk, M. and **Watson, N.** (1976) Some problems in diesel engine research with special reference to computer control and data acquisition, *Proc. Inst. of Mech. Engrs.* **190**(23).

McArragher, J.S. and **Tan, K.J.** (1972) Soot formation at high pressures: *A literature review, Combustion Science and Technology,* **5:** 257-261.

Murayama, T., Miyamoto, N. and **Tosaka, S.** (1988) Efficient and low smoke combustion of various Low-grade fuels in high speed small diesel engines, *ASME Transactions,* **110**(3): 385-392.

Tan, K.J. (1975) *Soot formation in diesel fuel spray,* Ph.D. Thesis, Queen Mary College, London University.

Wiebe, I. (1956) Habempirische Formel für die Verbrennungsgeschwinding Keit, Verlag Akademie der Wissenchaften der Vd. SSr, p. 51, Moscow.

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استنباط صيغة تجريبية لحساب كثافة الدخان (السخام) في عادم محركات الديزل ذات الحقن المباشر

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يسبب الدخان (السخام) الخارج من محركات الديزل أثراً سلبياً لدى معظم الناس ويفاقم من مشكلة التلوث البيئي. هناك الكثير من النظريات التي تتحدث عن كيفية تكون الدخان (السخام) الناتج عن حرق وقود الهايدروكربون ولكن إتضح عند تفحص هذه النظريات أنه لا يوجد اتفاق عام يمكن على ضوئه تفسير ظاهرة تكون الدخان.

في هذه الدراسة أجريت اختبارات متعددة على ماكينة ديزل ذات حقن مباشر لدراسة تأثير عوامل التشغيل المختلفة كسرعة الماكينة ونسبة التكافؤ الكلية ودرجة حرارة وضغط الهواء الداخل وتوقيت الحقن على كثافة دخان العادم.

حسبت معدلات حرق الوقود لكل حالة اختبار بعد تطبيق القانون الأول للديناميكا الحرارية وباستعمال المعلومات المتاحة عن ضغط الغاز داخل الاسطوانة عند كل زاوية من زوايا عمود المرفق خلال الفترة التي تكون فيها الصمامات مغلقة. أعتبر معدل حرق الوقود مكون من صيغتين: الأولى تمثل حرق الوقود المخلوط أثناء فترة تعوّق الاشتعال والثانية تمثل حرق الوقود إنتشارياً. ونتيجة لذلك تم التعبير رياضياً عن معدل الحرق كمجموع للصيغتين مع وزنها بكمية أخرى هي كسر الوقود المحترق إنتشارياً (α) والذي يعرف بأنه نسبة الوقود المحترق إنتشارياً إلى كمية الوقود الكلية المحقونة في الاسطوانة خلال الدورة.

أعـطت قراءات ضغط الغـاز المعالجـة، منحنيات مـوثوقـة لمعدلات الحـريق لمختلف حالات الاختبار واستنبـطت علاقـة تربط كسر الـوقود المحـترق إنتشاريـاً (α) مع نسبة التكافؤ الكلية وتعوّق الاشتعال.

لقد وجد في هذه الدراسة أن كثافة دخان العادم تتأثر تأثراً مباشراً بالتغـيرات التي تحدث في سرعة الماكينة ونسبة التكافؤ الكليـة وكسر الوقـود المحترق إنتشـارياً وكنتيجـة لذلـك استنبطت صيغـة تجريبيـة تربط كثـافـة دخـان العـادم مـع هـذه المتغيرات، وعندما طبقت هذه العلاقة لحسـاب كثافـة الدخـان لحالات الاختبـار المختلفة أعطت توافقاً مشجعاً. ويمكن القول هنا أن هذه العلاقة التجـريبية يمكن استخدامها في برامج محاكاة أداء الماكينة للتنبؤ بكثافة دخـان العادم خـلال تشغيل ماكينات الديزل ذات الحقن المباشر في الحالات الثابتة والعابرة.