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Experimental Instantaneous Heat Fluxes in a Diesel Engine and Their Correlation

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THE DESIGN OF INTERNAL combustion engines is necessarily becoming more scientific as the standards of performance, economy, and pollution control are increased. One manifestation of this trend is the increasing use of mathematical simulation as a design and development tool. However, an engine design based on a mathematical model is no better than the assumptions in the model. Both Borman (1)* and McAulay (2) noted that, along with other information, a knowledge of the instantaneous heat transfer to the cylinder surface is necessary in order to formulate an accurate model of engine processes.

The instantaneous heat fluxes at the surface of the cylinder wall (either head, valve face, piston face, or cylinder sleeve) have historically been expressed with the use of surface heat transfer coefficients, that is,

$$q(t) = h(t) \cdot \left(T_g(t) - T_w(t)\right) \qquad (1)$$

*Numbers in parentheses designate References at end of paper.

Where:

t = time
q(t) = wall surface heat flux
h(t) = heat transfer coefficient
$T_g(t)$ = mass-averaged gas temperature
$T_w(t)$=wall surface temperature.

It should be noted that for a clean metal wall $T_w(t)$ is relatively constant with time and, therefore, is often considered constant. For simplicity the modifier (t) will be dropped for all of the quantities.

Nusselt (3), Eichelberg (4), Pflaum (5), Annand (6), Woschni (7) and numerous other authors, have presented expressions which can be used to determine h(t) in Eq. (1). In general, these expressions, or correlations, have been ones which related the surface heat transfer coefficient, h to the properties of the working fluid (that is, ρ_g, T_g, k_g, etc.).

The majority of the correlations proposed to date are not based on instantaneous data, either at a point or averaged over an area, but on time-averaged data often obtained

ABSTRACT

By the use of surface thermocouples to measure instantaneous temperatures, the instantaneous heat fluxes are calculated at several positions on the cylinder head and sleeve of a direct injection diesel engine for both motored and fired operation. Existing correlations are shown to be unable to predict these data.

An analysis of convective heat transfer in the engine leads to a boundary layer model which adequately correlates the data for motored operation. The extension of this motored correlation to fired operation demonstrates the need for instantaneous local gas velocity and temperature data.

from an energy balance on the engine. Eichelberg did have access to Hug's (8) data which was obtained using subsurface thermocouples from a large, low speed, naturally aspirated diesel engine.

Nusselt showed that the pressure-temperature term in his correlation was a free convection relationship. Eichelberg, Pflaum, and others have used modifications of this free convection form to describe the forced, convection heat transfer in an engine cylinder. The analyses of Annand and Woschni are examples of several attempts to characterize engine heat transfer by dimensionless parameters. However, the lack of experimental data has precluded the construction of conceptual models of instantaneous heat transfer in engines.

In light of the previous discussion and as part of a continuing program of research on the phenomena occurring in internal combustion engines, the authors have conducted a study of the instantaneous surface-heat transfer in a direct injection diesel engine. Since there was a severe lack of experimental data, instantaneous heat fluxes were obtained at several positions on the cylinder head and sleeve under both motored and fired operation. Part 1 of this paper summarizes the experimental program and the data obtained.

The next logical step in the program was to compare the experimental data with already proposed correlations. This phase of the program is covered in Part 2 of the paper. Since the agreement found in Part 2 was not satisfactory, a detailed theoretical investigation of the mechanisms of surface heat transfer was conducted with the experimental data serving as a guide and comparison.

Part 3 of the paper reviews the theoretical considerations involved in correlating surface heat transfer in diesel engines and emphasizes the areas where further study and data are needed.

EXPERIMENTAL PROGRAM AND RESULTS

The engine, instrumentation, and processing techniques used in this study have been described in detail by LeFeuvre (9). Two other publications, Shipinski (10, 11) describe the results of a study of engine heat release run concurrently and employing the same instrumentation system. A brief review of this system is given in the following paragraphs.

EXPERIMENTAL APPARATUS AND PROCEDURES - Instantaneous wall temperatures were measured using a Bendersky (12) type surface thermocouple. The instantaneous surface heat fluxes were then calculated using the measured instantaneous temperatures. Because of the spacial, temporal (during one engine cycle), and cycle-to-cycle variations in surface temperature (and hence surface heat flux), it was felt that a large quantity of data (over 10^6 points) had to be examined if meaningful and significant results were to be obtained. Thus a high speed, multichannel, data recording, reduction, and processing system was developed. The block diagram of this system is shown in Fig. 1. while LeFeuvre (13) gives complete specifications.

The engine was a 4-stroke, direction-injection, single-cylinder diesel engine with a 4.5-in. bore and stroke. Shipinski describes the engine, dynamometer, and systems for obtaining performance data such as speed, power, air and fuel consumption, heat balance, intake and exhaust temperatures and pressures, etc.

The primary transducers for the determination of surface heat flux are the surface thermocouples as illustrated in Fig. 2. The thermocouples are of the plated junction design used by Overbye (14, 15), Bennethum (I6), and Ebersole (17). The thermocouples used were in the form of a 2-56 screw. Since the major portion of the thermocouple probe was iron, the disturbance to the heat transfer pattern was kept to a minimum. When flush mounted, the thermocouple junction temperature was considered to be the true surface temperature.

During the course of the work, thermocouples were installed in a total of nine locations in the engine head deck and sleeve. These locations are shown in Fig. 3. For all of these locations, except at TC-9, the heat flux in the cylinder wall could, with good accuracy, be considered to be one-dimensional. A schematic of a thermocouple circuit used is shown in Fig. 4. Since data from up to eight thermocouples were recorded simultaneously, there are eight units in many of the blocks of Fig. 4. The temperature of the wall-coolant interface was constant during each test and recorded by the multipoint recorder ①.* The time-averaged value of the instantaneous temperature difference

*Numbers in circles designate components shown in block diagrams, Fig. 4.

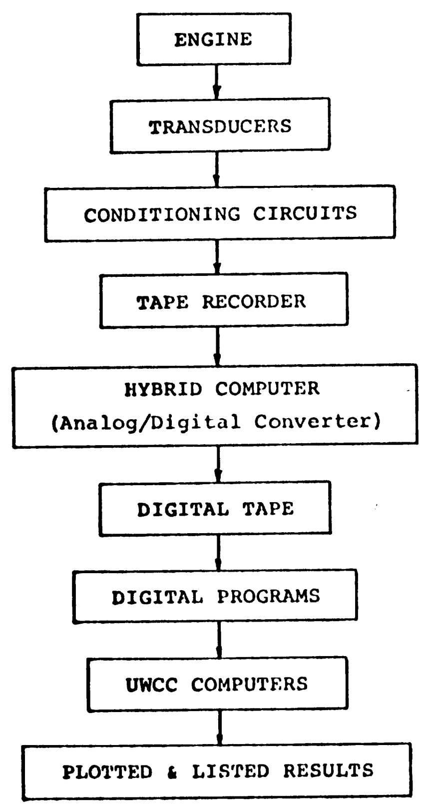

Fig.1 - Block diagram of experimental system

through the wall was measured by a light beam galvanometer ②.

The conditioning step of Fig. 1 consists primarily of amplification and biasing. The fixed gain amplifiers ③ had low noise, wideband high-gain amplification. It was desired to modulate the tape recorder with only the oscillatory component of the surface temperature. Thus the average value was biased out ④. The variable gain amplifiers ⑤ permitted optimum modulation of the tape recorder. A d-c calibration signal was recorded at the start of each data recording.

The magnetic tape recorder was a fully transistorized, 14-channel, 8-speed, high-performance machine of modular design and capable of frequency modulated (fm) or direct (dr) record /reproduce operation with a maximum tape speed of 120 in./sec. The data recorded during an engine run included cylinder pressure, crank angle (CA) indication at every degree of crank rotation, a pulse indicating piston top-dead-center (TDC), and eight surface thermocouple signals.

In most previous data handling systems it was necessary to manually scale the analog data. This drawback was overcome in the authors' system by using the analog-to-digital (A/D) conversion capability of a hybrid computer.

The analog signal representing pressure or temperature was played back from the tape recorder at 1/16 record speed to the hybrid computer and digitized at every CA for 50 cycles. An average cycle of pressure or temperature variation was then determined and written onto magnetic tape in digital form for subsequent processing. The averaging of

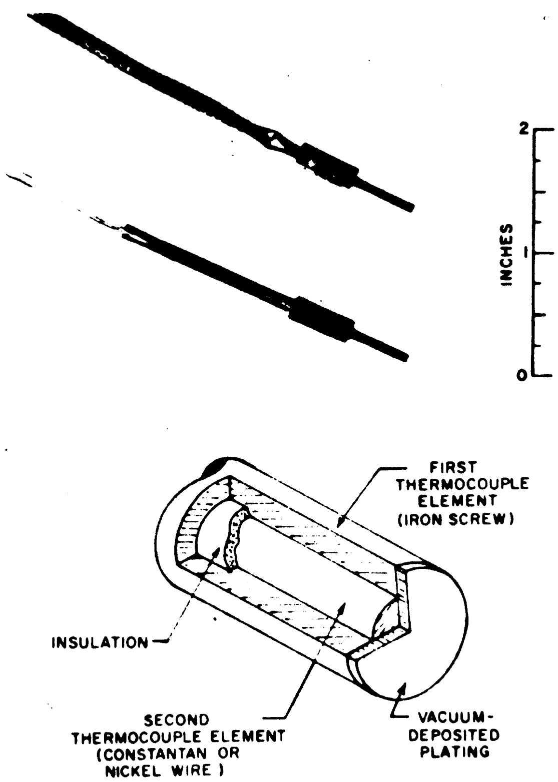

Fig. 2 - Surface thermocouple

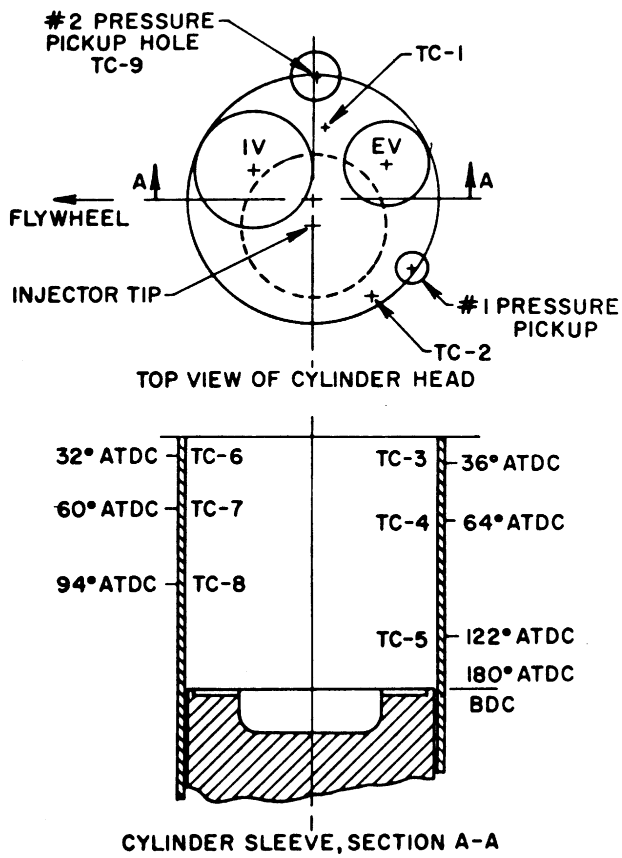

Fig. 3 - Cylinder head and sleeve geometry showing thermocouple locations

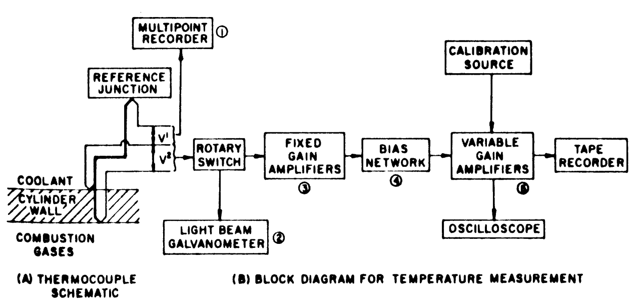

Fig. 4 - Thermocouple schematic and instrumentation for surface temperature measurement

a number of engine cycles served two purposes. First, cycle-to-cycle variation was eliminated to arrive at an average pressure-time or temperature-time curve. Second, random noise, introduced by the various electronic components, was attenuated by the factor $1\sqrt{N}$, where N is the number of cycles averaged from Bennett (18). This was particularly critical for the heat release studies as outlined in Shipinski (1967).

Since the heat transfer in the cylinder wall at the thermocouple locations was one-dimensional, the instantaneous heat flux was calculated as outlined by Carslaw (19) and Overbye (1961). The method involves the representation of the surface temperature by a Fourier series and the use of this series in a superposition solution of the partial differential equation governing heat conduction through the cylinder wall. This solution was carried out on the digital computer and the results obtained in both tabular and graphical form.

EXPERIMENTAL RESULTS - The following paragraphs contain a discussion of the experimental data. Some results are presented graphically in this section, typically with data from TC-1. However the data are quite extensive and space makes it impractical to put all the data in this paper. Table 1 shows the conditions studied. Mass-averaged temperature-time data have been computed for the runs marked with an asterisk (*). Copies of these digitized data, including instantaneous gas pressure and temperature and heat fluxes at five thermocouple positions, may be obtained by writing to the authors at the University of Wisconsin and paying a nominal reproduction fee.

One engine operating condition (Table 2) was defined as the standard operating condition (SOC) and repeated several times during the course of the experiments. For all subsequent figures, any variable whose value is not specified may be assumed to be at the value shown in Table 2. A comparison of the data obtained during these repeated SOC runs gives an indication of the reproducibility of the data. This is illustrated in Fig. 5. which shows instantaneous surface heat flux at one position on the cylinder head for five different engine tests at the same operating conditions. In general, the reproducibility of the data was very good on the cylinder head, whereas data from the cylinder sleeve indicated moderate scatter.

Fig. 6 shows the surface temperature-time curves for five

Table 1 - Summary of Operating Conditions

Run No.	Motored-M or Fired-F	Nominal Speed, rpm	Nominal Equivalence Ratio f	Nominal Injection Advance (deg CA BTDC)	Intake Density Ratio ρ/ρ_o
142*	M	1000	---	--	2.0
141*	M	1500	---	--	2.0
140*	M	2000	---	--	2.0
139*	M	2500	---	--	2.0
136*	F	1000	0.45	20.	2.0
137*	F	1500	0.45	20.	2.0
135*	F(SOC)	2000	0.45	20.	2.0
138*	F	2500	0.45	20.	2.0
144	F	2000	0.22	20.	2.0
134	F	2000	0.37	20.	2.0
143	F(SOC)	2000	0.44	20.	2.0
133	F	2000	0.53	20.	2.0
145*	F	2000	0.72	20.	2.0
150*	F	2000	0.45	10.	2.0
148	F(SOC)	2000	.045	20.	2.0
151*	F	2000	.045	30.	2.0
157*	M	2000	---	--	1.0
147*	M	2000	---	--	1.5
146	M	2000	---	--	2.5
152	F	2000	0.45	20.	1.0
153*	F	2000	0.45	20.	1.5
156	F(SOC)	2000	0.45	20.	2.0
132	F(SOC)	2000	0.45	20.	2.0
154*	F	2000	0.45	20.	2.5
155	F	2000	0.72	20.	2.5

different thermocouples for SOC operation. Note that the temperature axis is broken at several places to permit inclusion of all five curves on the one graph at a uniform scale, namely 10 F/div. The temperatures at TC's 3, 4, and 8 at CA = 0 are in parentheses. The surface heat flux histories corresponding to the temperature histories of Fig. 6 are shown in Fig. 7. Note that the peak instantaneous heat fluxes to the cylinder head can be 10 times the time-averaged values. The peak instantaneous flux at TC-1 was typically twice that at TC-2 for fired operation. With the exception of the time-averaged data from TC-3, the instantaneous and time-averaged heat fluxes decrease at lower positions on the sleeve for fired operation. The exception at TC-3 is attributed to the ineffectiveness of an improvised coolant passage near TC-3

It is evident from Fig. 6 that the passage of the piston rings over TC's 4 and 8 cause rapid temperature rises. This effect was particularly interesting at TC-4 as shown in Fig. 8. Note that five spikes are generated during the compression expansion process compared to six spikes during the exhaust-intake process. Theoretically the third piston ring should cover, but not go above, TC-4 at TDC. Six spikes occurred during the compression-expansion process only when the intake density ratio (ρ/ρ_o) was reduced to unity, either for motored or fired operation.

The effects of four operating parameters on the surface

Table 2 - Operating Conditions Defined as Standard Operating Conditions (SOC)

Compression	15.4:1
Speed	2000 ± 20 rpm
Dynamic Injection Timing	20± 1 deg CA BTDC
Intake Temperature	100 ± 3 deg F
Intake Tank Pressure	60 ± 1 in Hg abs
Exhaust Tank Pressure	60 ± 2 in Hg abs
Intake Valve Opens	520 deg CA
Intake Valve Closes	50 deg CA
Exhaust Valve Opens	310 deg CA
Exhaust Valve Closes	560 deg CA
Coolant Inlet Temperature	190 ± 10 deg F
Fuel	50 - 50 blend U-9 and T-16
	ASTM Secondary Cetane Reference Fuels
Equivalence Ratio	0.45
Fuel Flow	9.6 lb/hr

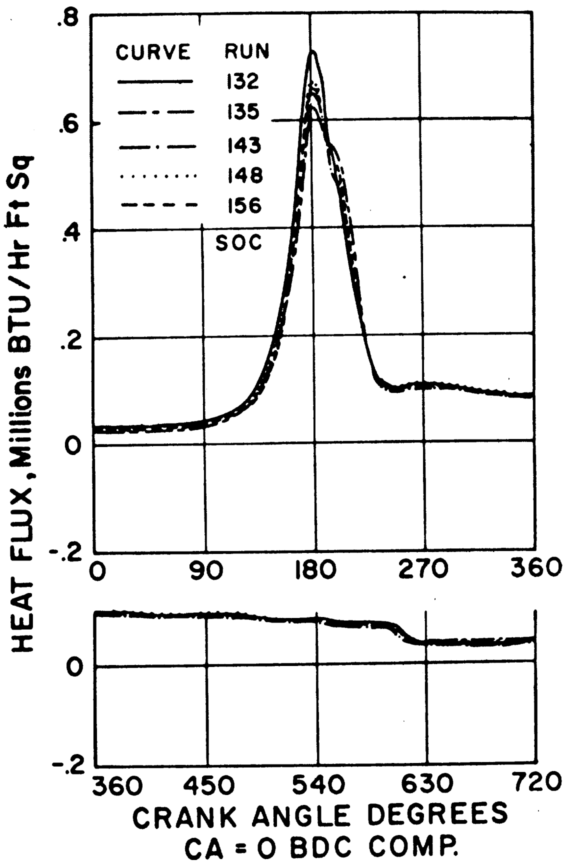

Fig. 5 - Cyclic surface heat flux at TC-2 for several engine runs at the standard operating conditions

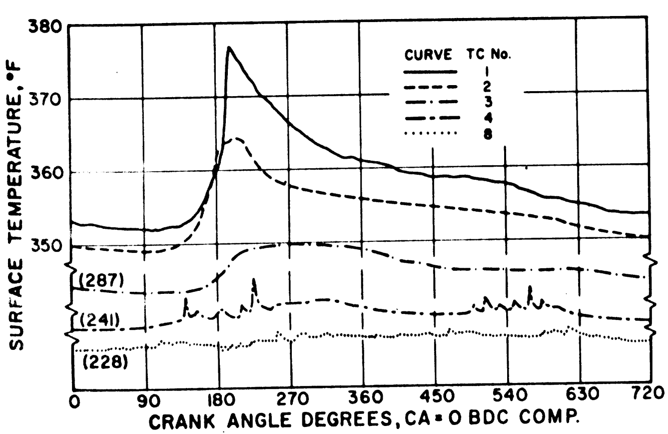

Fig. 6 – Cyclic surface temperature at five locations for SOC operation

heat flux were investigated, where applicable, under both motored and fired operation. These parameters are speed, equivalence ratio, injection advance, and intake density ratio. The authors do not wish to imply that surface heat flux depends only on the levels of these four parameters. These parameters were selected on the basis of the physical capabilities of the equipment available. The variations of surface heat flux with changes in each of the four parameters are discussed in subsequent paragraphs.

Over the speed range of 1000-2500 rpm, both motored and fired, the instantaneous and time-averaged surface heat fluxes generally showed an increase with increasing speed. This trend is most evident near TDC during the compression and expansion processes, as shown in Fig. 9. Note in Fig. 9 that the peak heat flux at TC-1 is higher at 1000 rpm than at 1500 rpm. This is an exception to the above generalization on the heat flux-speed trend. This exception occurred for both motored and fired operation but only at TC-1. The question as to whether this exception is physically significant, or whether it is due to experimental error, could not be answered.

The effect of equivalence ratio on the instantaneous heat flux at TC-1 is shown in Fig. 10. Both instantaneous and time-averaged heat fluxes were increased by an increase in equivalence ratio.

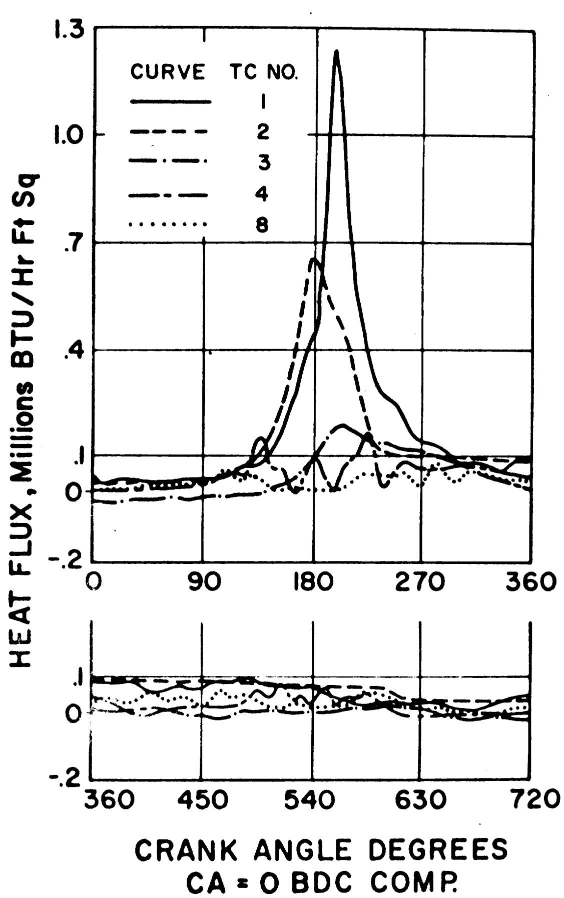

Fig. 7 - Cyclic surface heat flux at five locations in cylinder for SOC operation

The cylinder head thermocouples indicated increased heat flux at TDC with increasing injection advance, as shown in Fig. 11. However, the heat flux at about 20 deg CA ATDC decreased with increased injection advance, even though the mass-averaged temperature and pressure increased. The effect of injection advance on surface heat flux was shown to be small for the sleeve thermocouples. Most of the thermocouples indicated an increased time-averaged heat flux as injection advance was increased.

In general, an increased density ratio resulted in increased heat fluxes, as shown in Fig. 12. This generality applied for both instantaneous and time-averaged values at all thermocouples for several density ratios, with one interesting exception. At naturally aspirated (ρ/ρ_o = 1) operation, the instantaneous heat flux values at both TC-1 and TC-2 on the cylinder head were higher near TDC than those at higher density ratios (see Fig. 13). Yet the time-averaged values followed the above-mentioned general trend. Apparent pressure oscillations (Fig. 14), which may have been gas pressure oscillations, could account for the increased heat transfer, but the origin of the apparent pressure oscillations could not be proven.

OBSERVATIONS ON THE EXPERIMENTAL RESULTS - For fired operation the heat flux was greater at TC-1 than at TC-2. However. as shown in Fig. 15, this situation was reversed for motored operation. Notice that the thermocouple at the larger radius, TC-2 at r = 0.9 B/2, indicated a higher heat flux than TC-1 for motored operation. The existence of the relatively higher heat flux at TC-2 for motored operation is shown in Part 3 to be compatible with a swirl-initiated-boundary-layer model of cylinder head heat transfer.

Since the gas temperature and pressure histories on a CA basis are not significantly affected by a variation of engine speed, the increase in heat transfer with speed must be attributed to velocity and rate effects. Certainly the average gas velocity would be higher at increased engine speed. Moreover, the rate of compression of the thermal boundary layer is higher at the engine speeds. Since both of these factors contribute to surface heat transfer, one would expect higher fluxes at the higher speeds.

One would expect that, during the intake stroke when the bulk-gas temperature is less than the cylinder wall temperature, the direction of heat transfer would be from the wall to the gas, or, negative heat transfer. Generally the thermocouples on the sleeve show this effect. However, the cylinder head thermocouples show positive heat transfer throughout the cycle for fired operation. This indicates that the mass-averaged gas temperature may not be representative of the gas temperature for heat transfer calculations over at least part of the cycle during fixed operation.

PREVIOUSLY PROPOSED CORRELATIONS

As noted before, many previously proposed correlations were based on time-averaged data. The extent to which these correlations can be used to predict the instantaneous heat fluxes in a modern, high-speed, supercharged engine as used in this study is certainly of interest.

The cylinder head and piston surfaces are exposed to the cylinder gases throughout the complete cycle. During the compression and expansion processes, particularly near TDC, the cylinder sleeve is shielded from the high temperature gases by the piston. Thus the heat fluxes on the sleeve surface are expected to be less than those on the head and piston surfaces. This is confirmed by Fig. 7, which shows that the heat fluxes are generally higher on the cylinder head than on the cylinder sleeve. The correlations for surface heat transfer already proposed have considered cylinder head heat transfer almost exclusively. Thus the experimental data from the cylinder head thermocouples, that is TC's 1 and 2, are used in evaluating presently used correlations.

Annand gives a very good review of proposed correlations for surface heat transfer in engines. The extent to which several previously proposed correlations predict the authors' experimental data is shown In Figs. 16-19. The exact forms of the correlations used are given in Appendix A. Obviously it would be impractical to compare the experimental data with every correlation proposed heretofore. Those presented were thought to be the most popular ones.

Note that the existing correlations predict a single heat flux-time curve for the whole head area. The experimental data from TC-1 and TC-2 show that there is indeed considerable variation in the surface heat flux over the heat area. Fig. 19 shows that the Eichelberg's empirical relationship does a poor job of predicting motored heat transfer.

In general, none of the correlations used in Figs. 16-19 provides a good fit of the data from either of the thermocouples installed in the cylinder head. It is interesting to note that all of the correlations presented give a peak in the heat flux-time curve at about 190 deg CA for fired operation. The experimental data, particularly TC-1, show

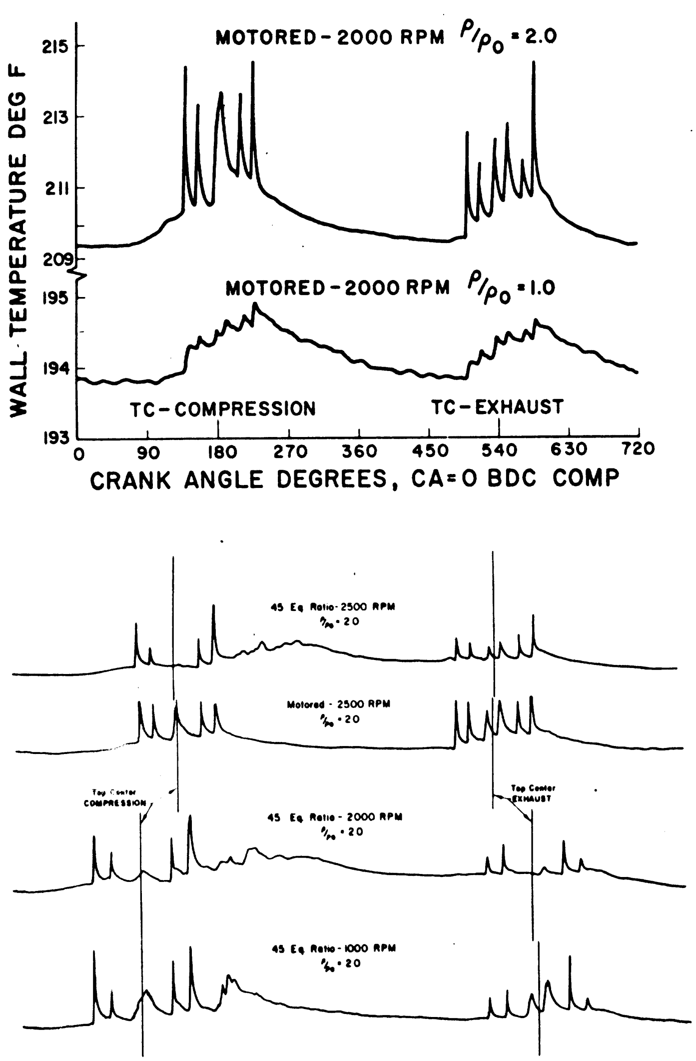

Fig. 8 - Cyclic surface temperature - time records from TC-4 on cylinder sleeve

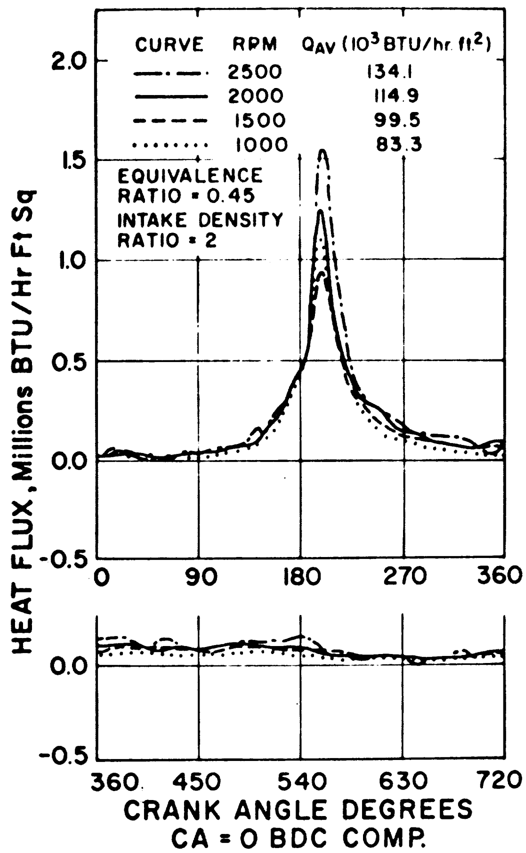

Fig. 9 - Cyclic surface heat flux at TC-1 for fired operation at several engine speeds

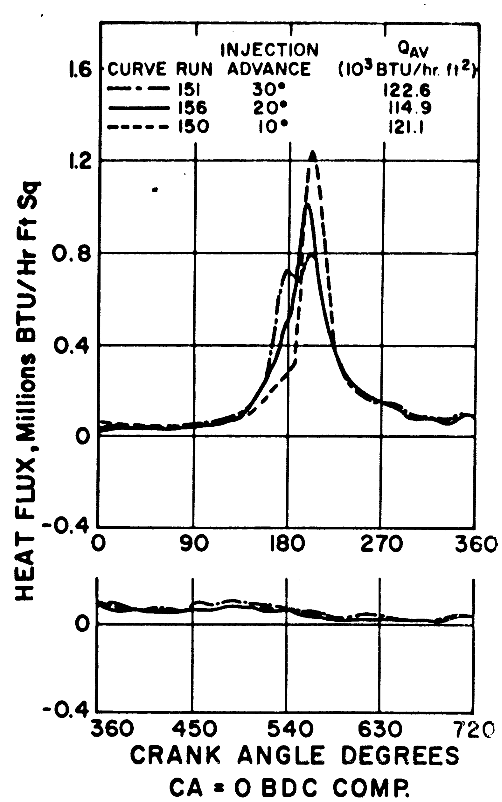

Fig. 11 - Cyclic surface heat flux at TC-1 for fired operation at several injection timings

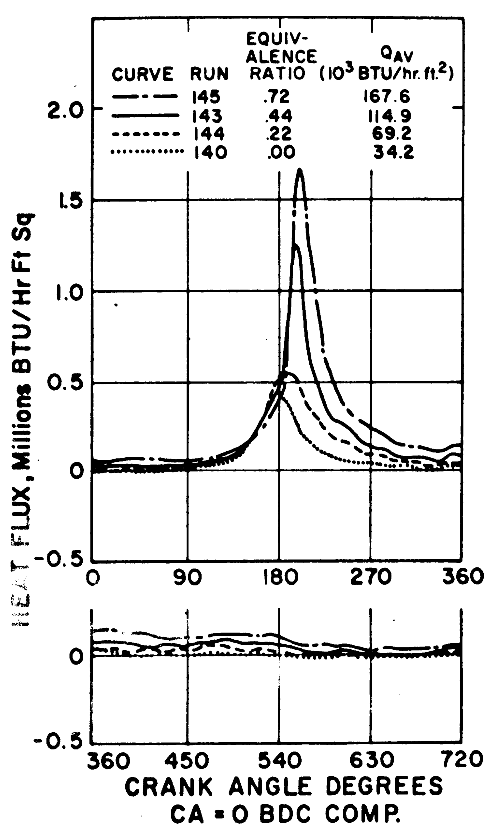

Fig. 10 - Cyclic surface heat flux at TC-1 for fired operation at several equivalence ratios

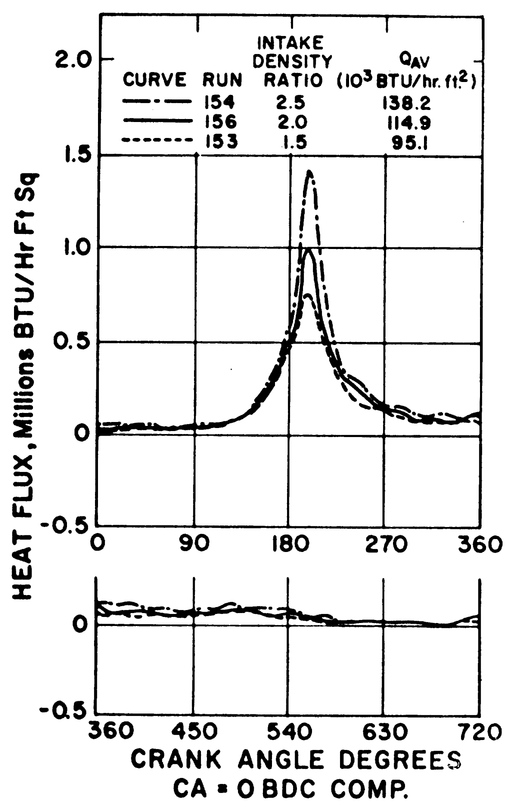

Fig. 12 - Cyclic surface heat flux at TC-1 for fired operation at several intake density ratios

a heat flux peak around 200 deg CA, over a wide range of operating conditions as seen in the figures of Part 1.

ANALYSIS OF ENGINE HEAT TRANSFER AND DATA. CORRELATION

Since none of the previously proposed correlations adequately predicted the experimental data. it was decided to next conduct a detailed, theoretical, study of heat transfer in an engine. At the least, such a study should delineate additional information needed and conceivably could lead to a new and better correlation.

Heat transfer in an engine is a complicated problem. Ebersole (17) presents data to show that radiant heat transfer is important, while Woschni (7) argues that it is negligible.

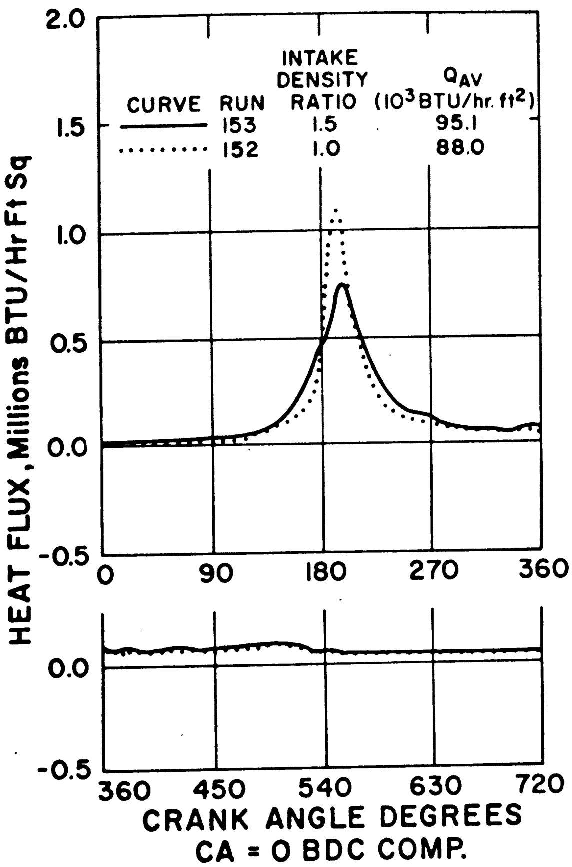

Fig. 13 - Cyclic surface heat flux at TC-1 for fired operation at intake density ratios of 1.5 and 1.0 (that is, N.A.)

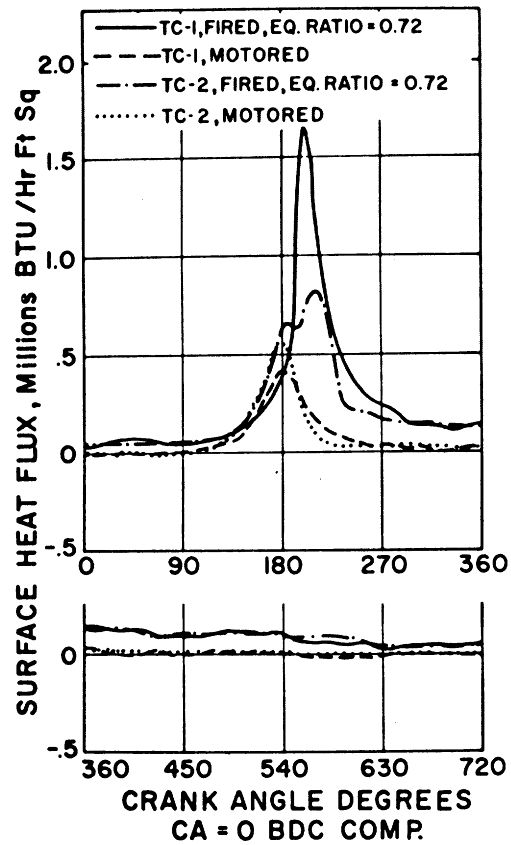

Fig. 15 - Cyclic surface heat flux at TC's 1 and 2 for motored and fired operation

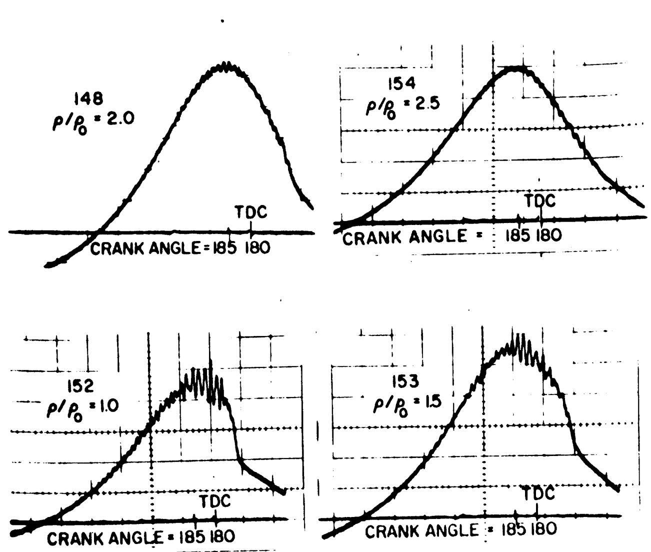

Fig. 14 - Cylinder pressure-time diagrams for several intake density ratios. Note that crank angle increases from right to left for each photograph

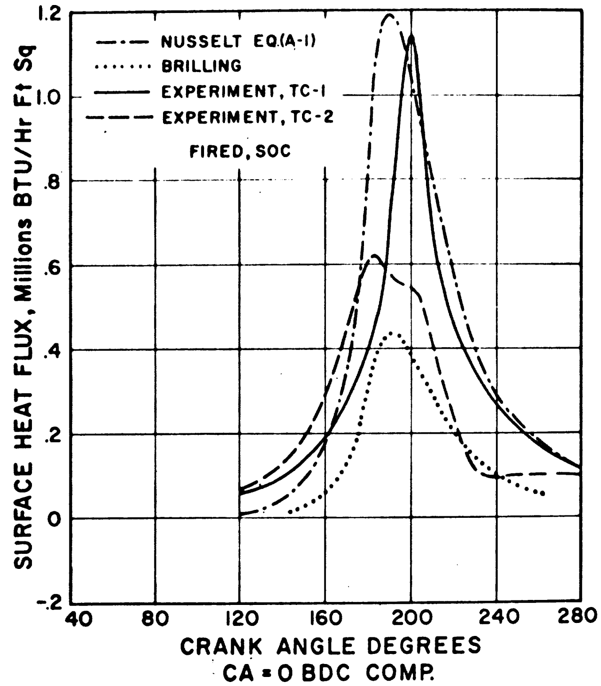

Fig. 16 - Comparisons of predictions of Nusselt and Brilling with experimental data from cylinder head for fired operation

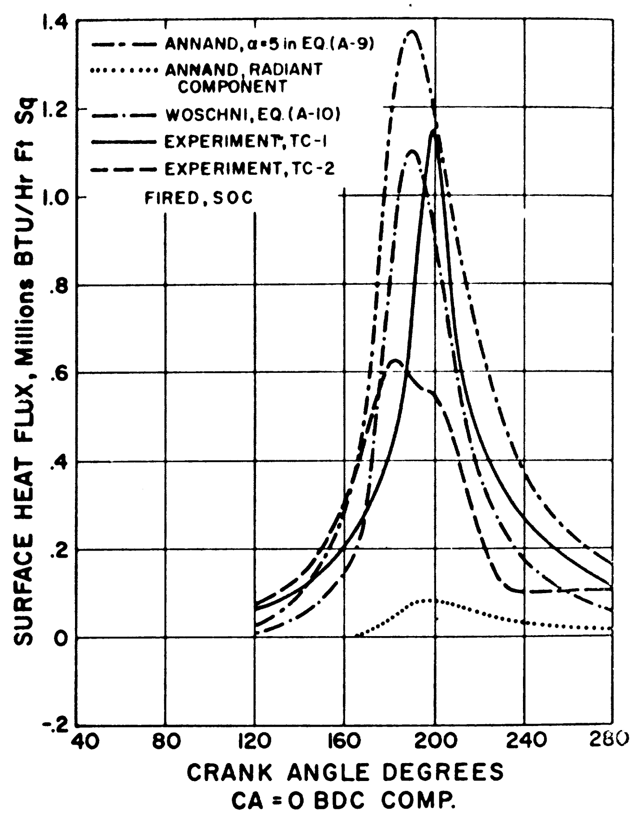

Fig. 18 - Comparisons of predictions of Annand and Woschni with experimental data from cylinder head for fired operation

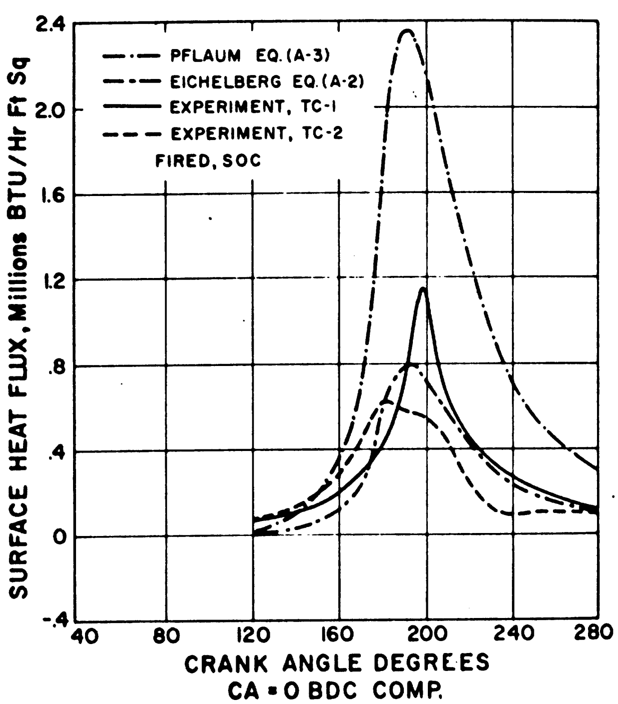

Fig. 17 - Comparisons of predictions of Eichelberg and Pflaum with the experimental data from cylinder head for fired operation

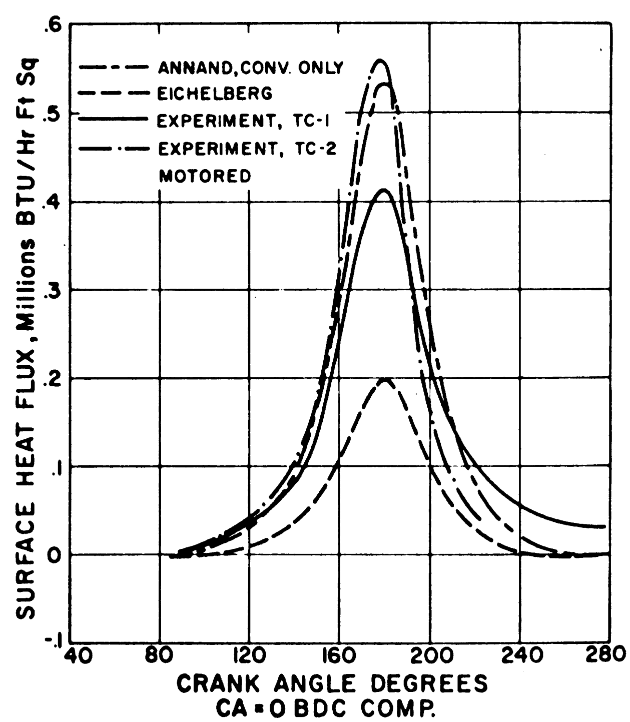

Fig. 19 - Comparisons of predictions of Annand and Eichelberg with experimental data from cylinder head for motored operation

Semenov (20) has experimentally demonstrated the existence of a boundary layer on the cylinder head but the relationship between this boundary layer and ordered and random gas velocities has not been established. Appendix B contains a discussion of ordered gas velocities in a motored engine and categorizes the velocities as either intake or piston related. The boundary layer has thermal capacity and in addition is compressed and expanded by the piston and the combustion processes with a consequent effect on heat transfer, Wendland (21).

An experimental investigation aimed at evaluating instantaneous radiant heat transfer in a diesel engine is currently in progress at the University of Wisconsin. Because of this and because of the disagreement as to the importance of radiant heat transfer, this study concentrated on the conductive and convective aspects of heat transfer in an engine.

The heat transfer for motored operation is relatively simple compared to the situation for fired operation. For motored operation, where there is no combustion, the radiant component is negligible and gas motions caused by combustion are not present. Thus the first step in correlating the experimental data is to formulate a correlation of the motored heat transfer. If this motored correlation can be expressed in a correct fundamental form, it should be adaptable to fired operation.

As already noted, Fig. 7 shows that the heat fluxes measured on the cylinder head are larger than those on the sleeve. In fact the time-averaged fluxes on the cylinder head are generally four times the time-averaged values on the sleeve even though there is friction heating of the sleeve by the piston rings. The convective heat fluxes to the piston should behave similarly to those on the cylinder head. Thus in the following discussion, heat transfer to the cylinder head will be considered almost exclusively. Comparisons between theory and experiment will utilize the experimental data from the thermocouples in the cylinder head, that is. TC's 1 and 2.

CONDUCTION-COMPRESSION MODEL OF HEAT TRANSFER IN THE MOTORED ENGINE - In an attempt to delineate the significance of the various factors, the authors used a one-dimensional conductive-compressive heat transfer model as investigated by Wendland (21). The basic assumptions made in the development of the conductive-compressive model, called the Adiabatic Plane model by Wendland, are:

1. The system is one-dimensional.
2. The gas is ideal.
3. The cylinder pressure is a function only of time and not of position.
4. A plane midway between the piston and the cylinder head is an adiabatic plane.

The gas mass between the head (or the piston) and the adiabatic plane is divided into a number of constant mass elements. Energy transfer between adjacent elements is by work or conduction heat transfer. A system of energy balances, one for each element, is solved to yield the temperature of all the elements at some point in time when the temperatures were known at the previous point in time.

A gas temperature profile was assumed at the intake valve closure (50 deg CA). This assumed profile, in conjunction with the known trapped mass, and the experimentally determined cylinder pressure were submitted to a computer program containing the governing equations of the Wendland model. The choice of any reasonable initial temperature profile was found to have a negligible effect on the resultant heat flux at the gas-wall interfaced.

The results of using the conduction-compression model to predict surface heat flux in the authors' engine are summarized in Fig. 20. The use of pure conduction energy transfer between adjacent mass elements resulted in approximately 20-25% of the peak experimental valve.

It was expected that in the engine the effect of free stream turbulence was to increase the effective conductivity of the gas through eddy conductivity, from Bird (22). The temperature gradient would be very low, not just at one plane (the adiabatic plane in Wendland's analysis) but over a region in the center of the gas. Thus the "adiabatic plane" could be considered to be closer to the wall than in the molecular conduction case. In an attempt to simulate free stream turbulence, the gas conductivity was increased by a factor of five everywhere but at the gas-wall interface. This resulted in a prediction of a peak flux 35-50% of the experimental value.

The conduction-compression model does not provide a good fit of the motored experimental data as seen in Fig. 20. Apart from the fact that the predicted heat fluxes are considerably lower than the experimental values, the model

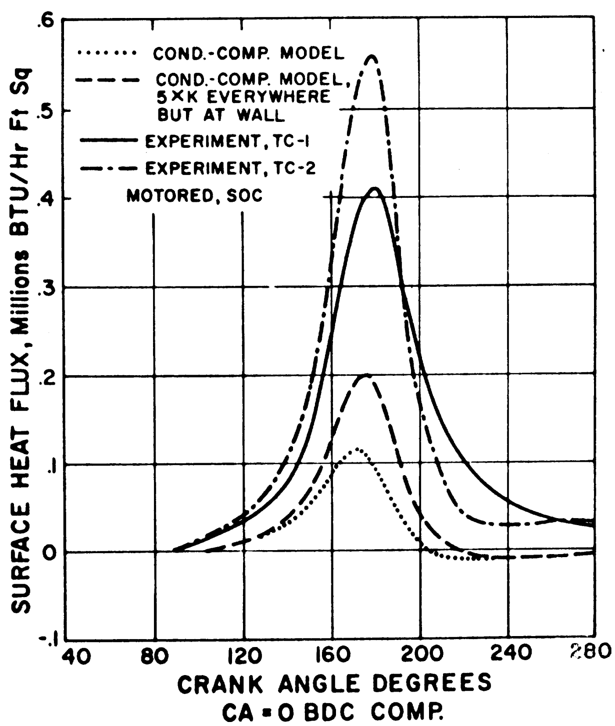

Fig. 20 - Comparisons of the results from the conduction-compression model with experimental data for motored operation

does not provide any means of predicting the different heat fluxes at TC's 1 and 2. Gas velocities parallel to the cylinder head surface are expected in the engine used in this study. Moreover, these velocities may not be the same at the two thermocouple positions. Thus it is felt that the weak point in the application of this model to the authors' engine is the ignoring of these gas velocities parallel to the head and piston surfaces. The model does incorporate the effects of pressure work and variable density in the boundary layer, but the thickness of the boundary layer is not controlled by a gas flow parallel to the surface as in the engine. Fig. 20 shows that the conduction-compression model does predict the rapid decrease in surface heat flux early in the expansion stroke, which is evident at TC-2. Negative heat flux early in the expansion process (30 deg ATDC) was measured experimentally by Wendland, and the conduction-compression model is the only one which predicts negative flux when the mass-averaged gas temperature is higher than the wall temperature.

Although the conduction-compression model predicted 50-75% of the experimentally measured heat flux in Wendland's study, an essentially identical model predicted almost 100% of the experimental value in a study by Goluba (23). Goluba measured the instantaneous surface heat flux at the stagnation point of a flow experiencing high-amplitude pressure oscillations. Using the same first three assumptions mentioned above, Goluba formulated the model in a different mathematical expression and achieved excellent agreement with his experimental data.

BOUNDARY LAYER MODEL OF HEAT TRANSFER IN MOTORED ENGINE - The concept of a boundary layer existing between the free stream fluid flow and some relatively stationary object was introduced by Prandtl in 1904. This concept has proven to be an extremely useful one for the study of both laminar and turbulent convective heat transfer. However, in general, the problems considered have been steady state ones, that is, where the boundary layer thickness at a point is constant with time. In the present case, the boundary layer thickness is expected to change throughout the engine cycle. Thus the first point to be considered is whether or not this steady state type of model is applicable to the unsteady heat transfer in engines.

Moore (24) shows that the time for a change in the free-stream conditions to diffuse through a laminar boundary layer is approximately equal to δ^2/υ. where δ is the boundary layer thickness, and υ the momentum diffusivity in the boundary layer. If this diffusion time is small, relative to other significant times in the problem, the boundary layer may be considered quasi-steady. That is, at any instant of time the boundary layer would be that associated with the conditions existing outside the boundary at that instant. For the present purpose, if this time is shown to be about 1 deg CA, the boundary layer was considered to be quasi-steady.

In Appendix B it is shown that the gas flow parallel to the head and piston surfaces is generally turbulent. Also, it is shown that the most significant gas velocity is probably a swirling one whereby the bulk of the cylinder gas can be considered to be in solid body rotation. Using this model, an analysis of Hartnett (25) may be applied to determine the turbulent boundary layer thickness. This calculation yields a turbulent boundary layer thickness on the head and piston of approximately 0.01 in. near TDC during the compression and expansion process. The thickness, which is representative of δ for this turbulent boundary layer, is determined by assuming a 1/7 power law velocity distribution and that the velocity achieves 50% of its maximum value in the region of the boundary layer which presents the greatest resistance to diffusion. Both of these assumptions are approximations, of course, but suffice for present purposes. Having determined this effective value of δ. the diffusion time is found to be approximately 0.5 deg CA at the SOC.

Alternatively, the conduction-compression model could be applied to furnish an estimate of the boundary layer thickness. From this model one obtains an estimate of the diffusion time of 10 deg CA at 2000 rpm. Since the model predicts less than 50% of the experimental heat flux, a reasonable estimate of the actual diffusion time from the conduction-compression model would be about 2.5 deg CA if the model predicted 100% of the experimental value.

On the basis of the above two estimates of the diffusion time, the authors feel that the assumption of a quasi-steady boundary layer is justified.

Either the partial differential energy equation or the Buckingham Pi theorem may be used to generate the significant dimensionless parameters to be used in a correlation for the surface heat transfer. The details of the former approach are given by LeFeuvre (13). Both approaches give rise to rate dependent parameters, which distinguish the unsteady situation in the engine from the classical steady-state situations. However, as shown above, the unsteady heat transfer in the engine may be considered to be quasi-steady. Thus. as a first approximation, the rate dependent, dimensionless parameters are not included and a correlation of the standard form,

$$Nu = f\,(Re,\ Pr) \qquad (2)$$

is appropriate.

From Eq. (2) it is seen that if a correlation containing some special variation is to be developed, the significant velocities and/or significant distances must be spacially dependent. The other significant quantities involved are essentially all functions of the gas temperature which must be determined from the cylinder pressure and density which we have assumed to be spacially independent.

For motored operation the instantaneous heat fluxes at TC-2 are higher than those at TC-1 (see Fig. 15). On the basis of a boundary layer concept this difference in heat fluxes could result from different boundary layer thicknesses since significant gas temperature gradients parallel to the cylinder head and piston surfaces are not expected in a motored engine. Thus, different boundary layer thickness resulting from different velocities appear to be the only reasonable explanation of the differences in the heat fluxes between TC-1 and TC-2.

A detailed discussion of the significant distances and velocities to be used in Eq. (2) in the correlation of the motored data is given in Appendix B. Briefly, for an open-chamber engine with moderate swirl, the significant distance for any position on the cylinder head (or piston) is taken to be the radial distance from the center of the bore. Also the significant velocity is considered equal to r ω, where ω represents the angular velocity resulting from intake-induced swirl. The assumption of a constant value for ω throughout the cycle is discussed in Appendix B. The Reynolds number in Eq. (2) is the same as that used in the correlation of friction factors and heat transfer coefficients in rotating flow systems, namely:

$$Re = \frac{r^2 \omega}{\nu} \quad (3)$$

where:

r = radius, here measured from the cylinder bore axis,
ω = angular velocity of the cylinder gases, and
ν = kinematic viscosity

For rotating systems, Dorfman (26) has shown that

$$Nu = a\, Re^{.8}\, Pr^{.33} \quad (4)$$

Eq. (4) may be rearranged to determine the film coefficient h, which may then be substituted into Eq. (1) to calculate the surface heat flux as:

$$q\,(t) = a\,\frac{k(t)}{r}\,Re\,(t)^{.8}\,Pr\,(t)^{.33}\left(T_g\,(t) - T_w\,(t)\right) \quad (5)$$

A least-squared error fit of the data from the motored engine using Eq. (5) predicted a value of 0.047 for "a." The fit was made at the SOC (motored) and the results are shown in Figs. 21 and 22. The gas properties were calculated at the average boundary layer temperature, the significant gas velocity was the swirl velocity, and the significant distance was the radius to the thermocouple position from the bore axis.

The agreement between the experimental data and the prediction from the correlation, Eq. (5) is quite good during the compression and expansion processes, although there is a small phase difference near TDC as indicated in Fig. 22. However, the fit is considered adequate since the motored correlation is not an end in itself, that is, the object in correlating the motored data is to describe the convective portion of the surface heat flux and then to apply this correlation to fired data.

The correctness of the speed dependency, $(r, \omega)^{0.8}$, of Eq. (5) is evident from Fig. 23 where the predicted and ex-

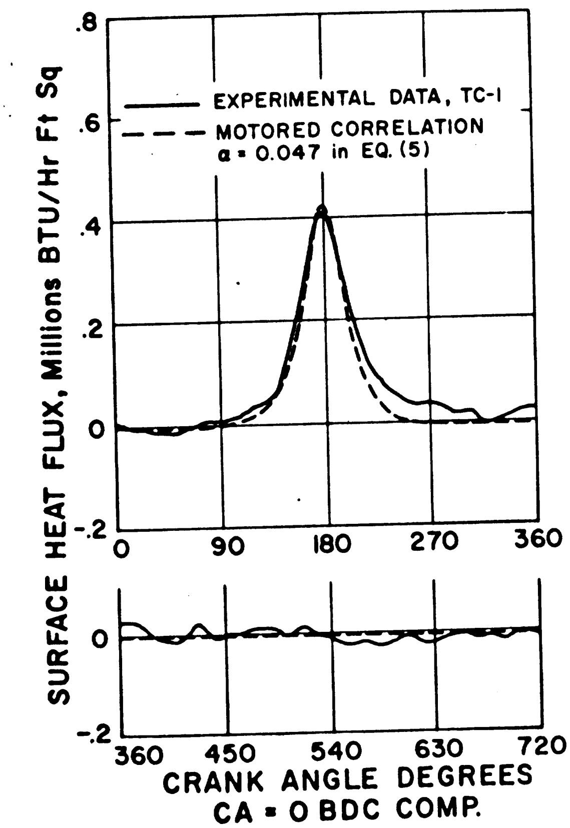

Fig. 21 - Boundary layer model fit of motored (SOC) data at TC-1

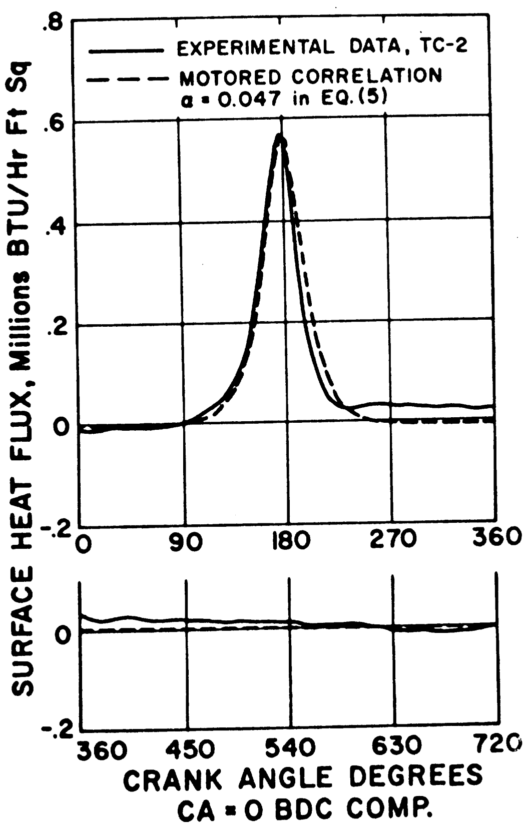

Fig. 22 - Boundary layer model fit of motored (SOC) data at TC-2

perimental fluxes arc shown for four engine speeds for TC-1. The agreement between the model and the experimental data is better at TC-2 as the 1000-rpm singularity (Fig. 9) was not evident at TC-2.

The mass-averaged gas temperature-time history is essentially independent of ρ/ρ_o. Thus at any instant for two different values of ρ/ρ_o, the ratio of the two heat fluxes predicted by Eq. (5) is the same as the ratio of the two values of ρ/ρ_o to the 0.8 power. Fig. 24 shows the experimental and calculated (Eq. (5) heat fluxes at TC-1 for four different intake density ratios. The calculated values at ρ/ρ_o = 1, 1.5 and 2.5 were found from the values at ρ/ρ_o = 2 by using the ratios of intake densities to the 0.8 power.

EXTENSION OF THE MOTORED BOUNDARY LAYER CORRELATION TO FIRED OPERATION - The correlation in Eq. (5) is shown to predict the motored data with moderate success. Gas velocities arising from combustion and radiation effects are expected to augment the predicted heat flux in the fired case. The correlation is for convective heat transfer and thus would not be expected to predict the total heat flux under fired operation. However, as an aid in furthering our understanding of heat transfer in the fired engine, the motored correlation, Eq. (5), was used to predict the fired heat transfer at TC's 1 and 2. and the results are presented in Figs. 25 and 26. The agreement between the correlation and the data is fair at TC-1 and poor at TC-2. The correlation predicts a convective flux greater than the total experimental heat flux for portions of the cycle, particularly at TC-2.

Note that the heat flux predicted by Eq. (5) is larger than the experimental value in Fig. 25 from 15 deg CA BTDC to 15 deg CA ATDC. Recall that many of the previous correlations reviewed (Figs. 16-19) show a similar tendency. From Fig. 27 it is seen that the motored and fired heat fluxes at TC-1 are approximately equal for CA < 185 deg. Note, however, that the mass-averaged gas temperature and pressure for fired operation, are significantly different than the motored values for CA > 170 deg. Thus the correlation, Eq. (5), as based on the mass-average gas properties, could not be expected to predict the apparent lag between the surface flux and the mass-averaged gas properties. These remarks apply equally to the flux at TC-2. In fact, due to the higher gas (swirl) velocity at TC-2 the flux prediction from Eq. (5) shows a greater error at TC-2 than at TC-1.

The combustion in the engine originates somewhere in the combustion chamber raising the temperature locally and the pressure uniformly throughout the cylinder. This is in-line with the assumption of uniform pressure in the cylinder which is valid for most operating conditions. The assumption of uniform gas temperature throughout the cylinder and the use

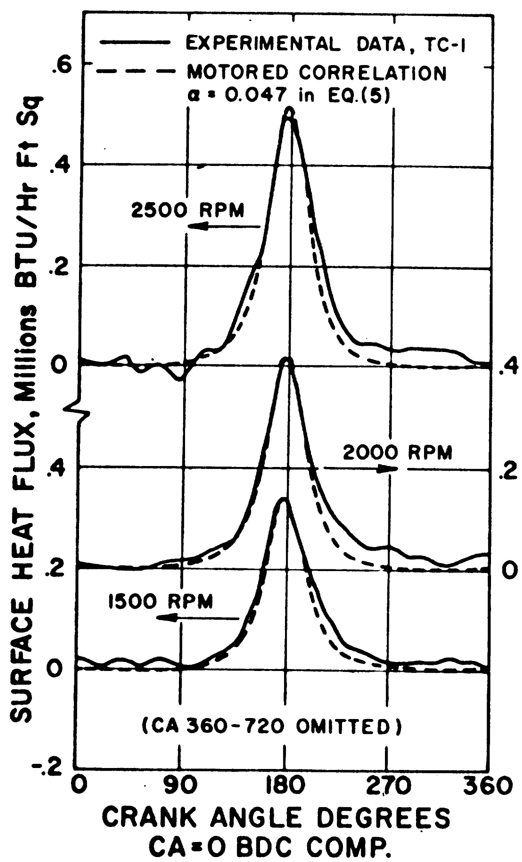

Fig. 23 - Cyclic surface heat flux variation with speed for motored operation, comparison between experiment and boundary layer model of Eq. (5)

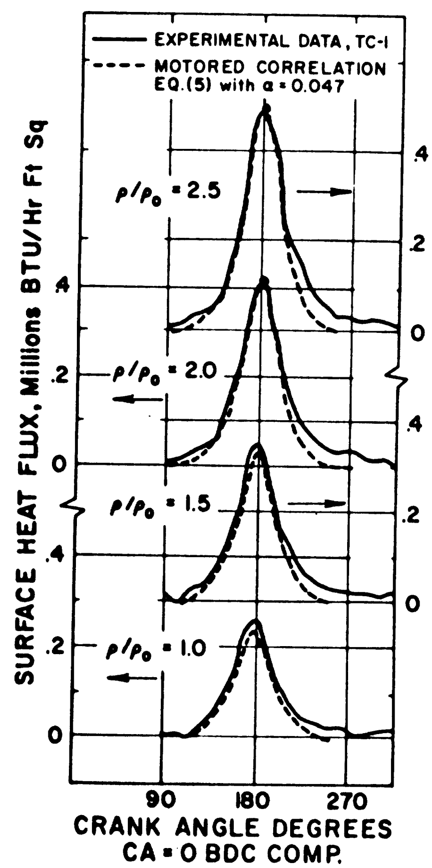

Fig. 24 - Cyclic surface heat flux variation with intake density ratio for motored operation. comparisons between experiment and boundary layer model

of this mass-averaged temperature as the source temperature for heat flux to the cylinder walls is questionable. In fact, until the flame actually reaches the boundary layer, the boundary layer temperature history is probably close to the temperature history under motored operation. However, the temperature gradient at the wall in the fired case would be greater than in the motored case because of the increased compression due to combustion.

In order to test this theory, the correlation from Eq. (5) was used to predict the surface heat flux for fired operation but the gas temperature used was that for motored operation. The results are shown in Figs. 28 and 29. Poor agreement will be noted for TC-1 and fair agreement for TC-1

Due to the offset position of the combustion chamber and to the valve cutouts in the piston, there should be considerable motion of the flame and unburned fuel into the area between TC-1 and the No. 2 pressure pickup hole (see Fig. 3). This, no doubt, produces appreciable gas velocities near TC-1. Fig. 3 shows that TC-l and TC-2 are approximately equidistant from the combustion chamber formed by the cavity in the piston. While the flame is concentrated in the cavity both TC's should receive equal heat flux by radiation. However, later burning

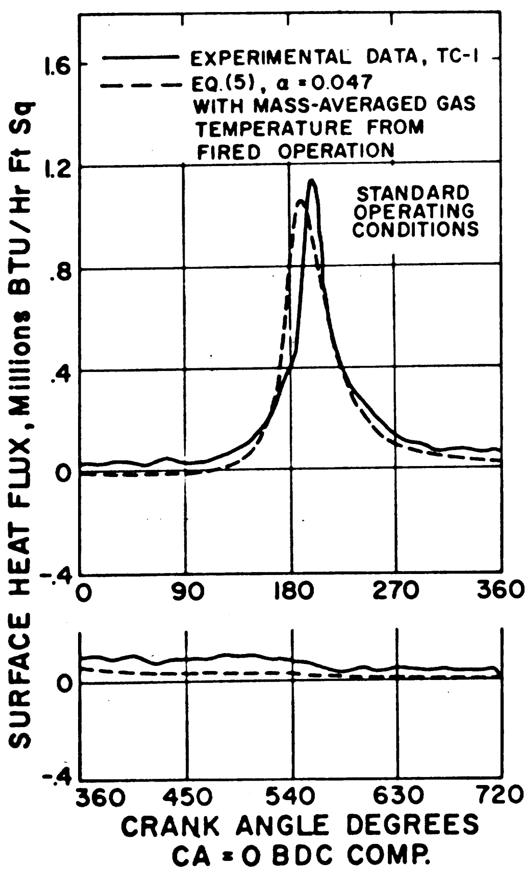

Fig. 25 - Extension of motored correlation to fired operation (SOC), at TC-1

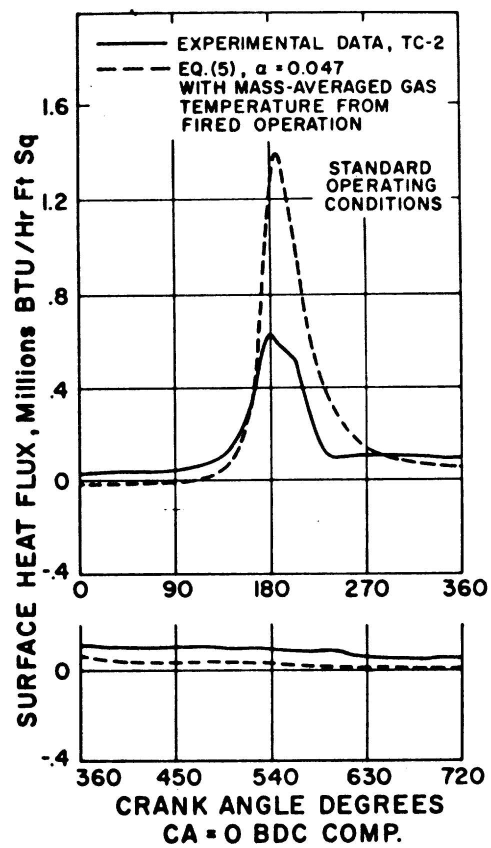

Fig. 26 - Extension of motored correlation to fired operation (SOC), at TC-2

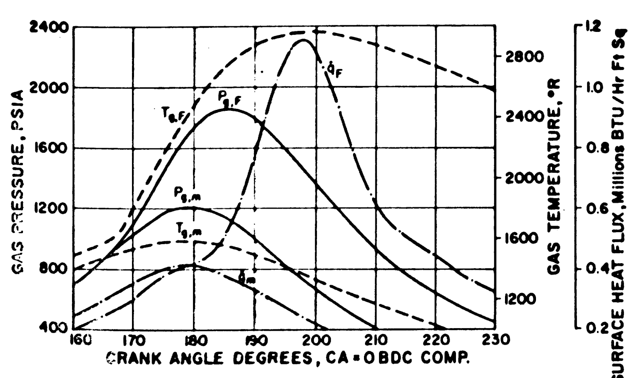

Fig. 27 - Gas temperature and pressure, and heat flux at TC-1, for motored and fired operation

is more likely to be symmetrical about the bore axis and TC-1 should receive a greater radiant flux than TC-2. Thus both radiation effects and gas velocity effects contribute to the higher peak flux at TC-1 at about 20 deg CA ATDC.

To be able to predict the total experimental heat flux for any position on the cylinder head or piston, it apparently is necessary to use the actual gas velocities and gas temperatures in Eq. (5). Furthermore, a separate term may be necessary to account for radiation. At present the necessary data are just not available to make any more than the roughest estimate of these influences and thereby achieve a fit of the data. A current study at the University of Wisconsin should provide the first experimental data on instantaneous radiant heat transfer in the engine. With this data one of the two presently unknown combustion related terms, that is, gas velocity and radiation, would be known. Then a more logical estimate of the gas velocity term should be possible.

CONCLUSIONS

The comparisons between the experimental data presented in this paper and the heat fluxes calculated using present correlations have shown that these correlations provide at best only an approximation to the data. The use of average piston speed, cylinder bore, and mass-averaged gas temperature in correlations for the instantaneous surface heat fluxes precludes the prediction of the spacial variation shown by the experimental data.

A study of the spacial and temporal variations in gas temperature and velocity (both motored and fired) is necessary if one wishes to improve on the boundary layer model as proposed by Sitkei (27), Annand, Woschni, and the present authors. Given the results of such a study, the authors suggest that this information should be incorporated into a boundary layer model as proposed in this paper. With an allowance for radiation (hopefully forthcoming from a current study at the University of Wisconsin) this improved correlation should furnish a better prediction of the data presented in Part 1 than is possible at present.

As mentioned in Part 3 the boundary layer models essentially ignore the effects of compression work in the boundary layer. Thus even an improved correlation incorporating instantaneous local gas velocities and temperatures cannot be expected to provide a complete picture. It may be necessary to combine the features of the conduction-compression and the boundary layer models. Some of the data of Part 1suggest this combination.

Recall that the conduction-compression model predicted a rapid decrease of surface heat flux early in the expansion stroke as seen in Fig. 20. Note from Fig. 11 that the flux

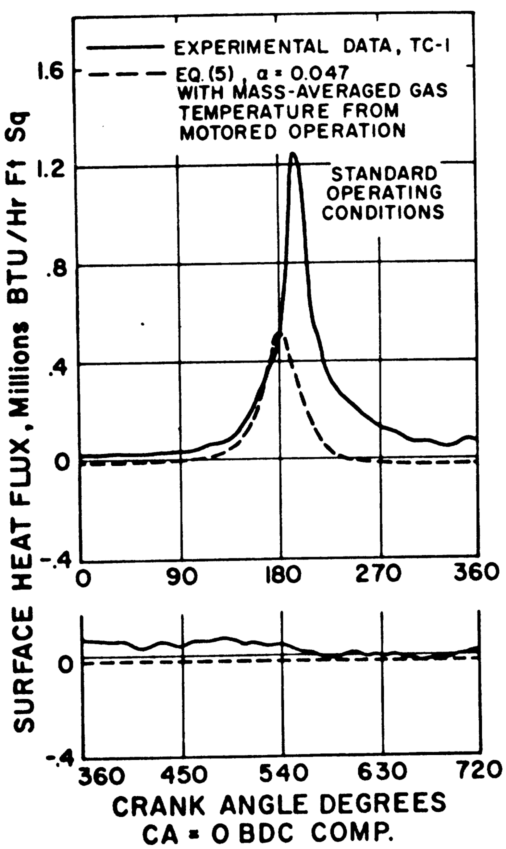

Fig. 28 - Extension of motored correlation to fired operation (SOC) at TC-1 but with use of motored gas temperature

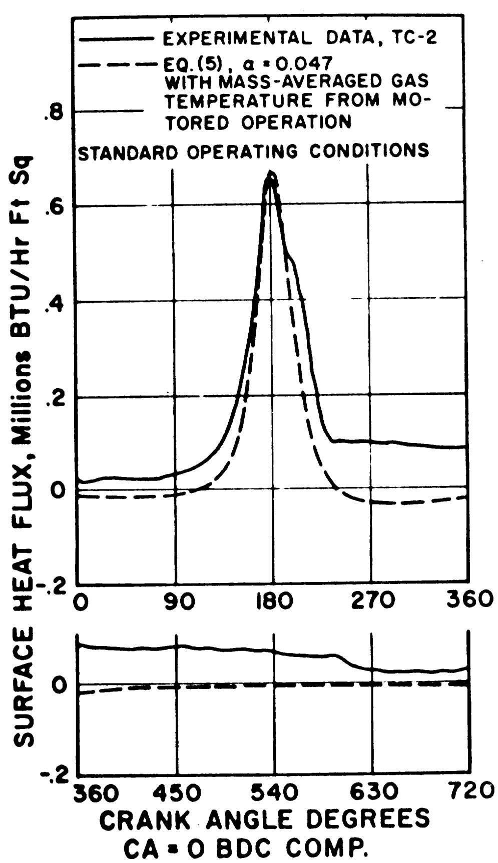

Fig. 29 - Extension of motored correlation to fired operation (SOC) at TC-2 with use of motored gas temperature

at 20 deg ATDC is much lower for the 30 deg injection advance run than for the 10 deg advance run, even though the gas temperature and pressure are higher at this point for the 30 deg advance run, LeFeuvre (1968-b). At 20 deg ATDC the 30 deg advance run has already experienced 35% of its pressure decrease (expansion) compared to 15% for the 10 deg advance case. Hence, by application of the conduction-compression model a lower heat flux might be expected.

The above reasoning follows from a conceptual combination of the conduction-compression and boundary layer models. Analytical work leading to a mathematical expression of this combination should prove to be interesting and profitable.

ACKNOWLEDGMENTS

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REFERENCES

1. G. L. Borman, "Mathematical Simulation of Internal Combustion Engine Processes and Performance Including Comparisons With Experiment." Ph.D. Thesis, Mech. Engr. Dept., University of Wisconsin, 1964.

2. K. J. McAulay, T. Wu, S. K. Chen, G. L. Borman, P. S. Myers and 0. A. Uyehara. "Development and Evaluation of the Simulation of the Compression-Ignition Engine." Paper 650451 presented at SAE Mid-Year Meeting, Chicago, May 1965.

3. W. Nusselt, "Der Warmeubergang in der Verbrennungskraftmaschine." VDI Forschungshaft, No. 264, 1923.

4. G. Eichelberg, "Investigations on Combustion Engine Problems." Engineering, Vol. 148, 1939, pp. 463, 547, 603, 682.

5. W. Pflaum, "Heat Transfer in Internal Combustion Engines." Presented at La Conference Internazionale di Termoteenia, Milan, November 1962.

6. W. J. D. Annand, "Heat Transfer in the Cylinders of Reciprocating Internal Combustion Engines." Proc. Inst. Mech. Engrs., Vol. 177, No. 36, 1963, p. 973.

7. G. Woschni, "A Universally Applicable Equation for the Instantaneous Hear Transfer Coefficient in the Internal Combustion Engine." Paper 670931 presented at SAE Combined National Meetings, Pittsburgh, October 1967.

8. K. Hug, "Messung and Berechnung von Kolbentem Peraturen in Dieselmotoren." Mitt. Inst. Thermodyn, Zurich, No. 1, 1937.

9. T. LeFeuvre, P. S. Myers, O. A. Uyehara, and J. S. Shipinski, "A Tape Recording and Computer Processing System for Instantaneous Engine Data." Paper 680133, Presented at SAE Automotive Engineering Congress, Detroit, January 1968.

10. J. H. Shipinski, "Relationships Between Rates-of-Injection and Rates-of-Heat Release in Diesel Engines." Ph.D. Thesis, Mech. Engr. Dept., University of Wisconsin, 1967.

11. J. H. Shipinski, P. S. Myers, and O. A. Uyehara, "An Experimental Correlation Between Rate of Injection and Rate of Heat Release in a Diesel Engine." ASME Paper 68 DGP-11, to be published in ASME Diesel and Gas Power Proc., 1968.

12. D. Bendersky, "A Special Thermocouple for Measuring Transient Temperatures." Mechanical Engineering, Vol. 75, 1953, p. 117.

13. T. LeFeuvre, "Instantaneous Metal Temperatures and Heat Fluxes in a Diesel Engine." Ph.D. Thesis, Mech. Engr. Dept., University of Wisconsin, 1968.

14. V. D. Overbye, "Variation of Instantaneous Wall Temperature, Heat Transfer, and Heat Transfer Coefficients in a Spark Ignition Engine." Ph.D. Thesis, Mech. Engr. Dept., University of Wisconsin, 1960.

15. V. D. Overbye, J. E. Bennethun, P. S. Myers, and O. A. Uyehara, "Unsteady Heat Transfer in Engines. SAE Transactions, Vol. 69 (1961), p. 461.

16. J. E. Bennethum, "Heat Transfer and Combustion Chamber Deposits in a Spark Ignition Engine." Ph.D. Thesis, Mech. Engr. Dept., University of Wisconsin, 1960.

17. G. D. Ebersole, P. S. Myers, and O. A. Uyehara, "The Radiant and Convective Components of Diesel Engine Heat Transfer." Paper 701C presented at SAE International Summer Meeting, Montreal, June 1963.

18. C. A. Bennett and N. L. Franklin, "Statistical Analysis in Chemistry and the Chemical Industry." New York: John. Wiley & Sons, 1954.

19. H. S. Carslaw and J. C. Jaeger, "Conduction of Heat in Solids." Oxford: Clarendon Press, 1959.

20. E. S. Semenov, "Studies of Turbulent Gas Flow in Piston Engines." in "Combustion in Turbulent Flow." ed. L. N. Khitrin, 1959. Translated from Russian by Israel Program for Scientific Translations, 1963.

21. D. W. Wendland, "The Effect of Pressure and Temperature Fluctuations on Unsteady Heat Transfer in a Closed System." Ph.D. Thesis, Mech. Engr. Dept.. University of Wisconsin, 1968.

22. R. B. Bird, W. E. Stewart. and E. N. Lightfoot, "Transport Phenomena." New York: John Wiley & Sons, 1960.

23. R. W. Goluba, "The Effect of Periodic Shock-Fronted Pressure Waves on the Instantaneous Heat Flux at the End-Wall of a Tube." Ph.D. Thesis, Mech. Engr. Dept., University of Wisconsin, 1968.

24. F. K. Moore, "Unsteady Laminar Boundary-Layer Flow." NACA TN2471, September 1951.

25. J. P. Hartnett, S. H. Tsai, and H. N. Jantscher, "Heat Transfer to a Nonisothermal Rotating Disk with a Turbulent Boundary Layer." Trans. ASME, Vol. 78, Series C, No. 3, August 1965, p. 363.

26. L. A. Dorfmann, "Hydrodynamic Resistance and the

Heat Loss of Rotating Solids." Translated by N. Kemmer, Edinburgh: Oliver and Boyd, 1963.

27. G. Sitkel, "Beitrag zur Theorle des Warmeuberganges im Motor." Konstruktion, 14, 1962, p. 67.

28. A. Stambuleanu, "Contribution to the Study of the Distribution of Heat Transfer Coefficients during the Phase of the Working Cycle of an Internal Combustion Engine." Third International Heat Transfer Conference, Chicago, 1966.

29. K. Lohner, E. Dohring, and G. Chore, "Termperaturschwingungen an der Innenwand von Verbrennungskraftmaschinen." MTZ, Vol. 17, No. 12, December 1956, p. 413-418.

30. W. Pflaum, "Warmeubergang bei Dieselmaschinen mit und ohne Aufladung." MTZ, Vol. 22, No. 3, March 1961; Translated in The Engineers Digest, Vol. 22, No. 7, July 1961.

31. N. A. Henein, "Instantaneous Heat Transfer Rates and Coefficients Between the Gas and Combustion Chamber of a Diesel Engine. Paper 969B presented at SAE Automotive Engineering Congress, Detroit, January 1965.

32. C. F. Taylor, "Heat Transmission in Internal-Combustion Engines." Proc.. General Discussion on Heat Transfer, Inst. Mech. Engrs., London, 1951, p. 397.

33. C. F. Taylor and T. Y. Toong, "Heat Transfer in In ternal-Combustion Engines. "ASME Paper No. 57-HT-17, 1957.

34. A. Nagel, "Heat Transfer in Reciprocating Engines." Engineering, Vol. 127, 1.929, pp. 59, 179, 279. 466.

35. K. Elser. "Der Instationaire Warnicubergang In Dieselmotoren." Mitt. Inst. Thermodyn, No. 15, Zurich, 1954.

36. B. Loeffler, "Development of an Improved Automotive Diesel Combustion System." SAE Transactions, Vol. 62, (1954), p. 243.

37. D. Fitzgeorge and J. L. Allison, "Air Swirl in a Road Vehicle Diesel Engine." Proc. Inst. Mech. Engrs. (A.D.), No. 4, 1962-1963, p. 151.

38. J. F. Alcock and W. M. Scott, "Some More Light on Diesel Combustion." Proc. Inst. Mech. Engrs. (A.D.), No. 5, 1962-1963, p. 179.

39. T. Okaya and M. Hasegawa, "On the Friction to the Disk Rotating in a Cylinder." Jap. Journal of Physics, Vol. 13, 1939.

40. S. L. Soo, "Laminar Flow Over an Enclosed Rotating Disk." Trans. ASME, Col. 80, (1958), p. 287.

41. J. W. Daily and R. D. Nece, "Chamber Dimension Effects on Induced Flow and Frictional Resistance of Enclosed Rotating Disks." Trans. ASME, Vol. 82, Series D, March 1960, p. 217.

APPENDIX A

PREVIOUSLY PROPOSED CORRELATIONS

Many different expressions have been proposed to correlate the surface heat fluxes in diesel engines. Attention is focused on three correlations upon which most other correlations have been based. The three correlations, which are considered in some detail, are those of Nusselt, Eichelberg, and correlations based on the Reynolds' analogy boundary layer theory. The modifications to these three basic forms, which have been suggested by various authors, are also considered. All the correlations discussed employ the mass-averaged gas temperature, $T_g(t)$, to represent the gas temperature.

NUSSELT'S CORRELATION - Nusselt's work was based on measurements of the heat loss from the combustion of quiescent, homogeneous mixtures in spherical bombs. He determined the influence of radiation by using gold-plated or blackened, inside-surface coatings on the bombs.

By incorporating a term to account for forced convection due to piston motion, Nusselt adapted results from the bomb experiments to the situation in an engine, and proposed that the surface heat transfer coefficient be expressed as:

$$h(t) = 0.0278\,(1 + 0.38\,V_p)\,[P(t)^2\,T_g(t)]^{1/3} + 1.275 \times 10^{-10}\,\frac{[T_g(t)^4 - T_w(t)^4]}{[T_g(t) - T_w(t)]} \qquad (A\text{-}1)$$

where:

$P(t)$ = instantaneous cylinder gas pressure in psia,

$T_g(t)$ = instantaneous cylinder gas pressure in R, and

V_p = mean piston speed in ft/sec.

The first term on the right of Eq. (A-1) represents convective transfer and the second term gives the radiative portion. The choice of 2/3 for the exponent of P(t) was actually an average of several values ranging from 0.5-0.8. Jaklitsch quoted by Stambuleanu (28) suggested values ranging from 0.44-0.90. Lohner (29) presents a linear temperature function for this exponent.

Brilling changed the piston speed term of Eq. (A-1) from ($1 + 0.38\,V_p$) to ($2.45 + 0.056\,V_p$) on the basis of tests on stationary Diesel engines. Fig. 16 shows the heat fluxes computed using the formulae of Nusselt and Brilling along with the experimental results from the cylinder head thermocouples.

EICHELBERG'S CORRELATION - Although Eichelberg's correlation is actually a modification of Nusselt's, it merits special consideration because of its wide usage and because of the related experimental work carried out by Eichelberg and his associates. Eichelberg summarized several years of research using subsurface thermocouples to study the instantaneous surface heat flux in large. low-speed diesel engines under NA operation. He proposed the correlation:

$$h(t) = 0.0565\,V_p^{1/3}\left(P(t)\,T_g(t)\right)^{1/2} \qquad (A\text{-}2)$$

for the surface heat transfer coefficient.

Elchelberg stated that he expected a small heat flux due to radiation. Yet he gave a relatively greater significance than Nusselt to gas temperature to account for radiation and increased gas velocity during the intake stroke. Eichelberg preferred to omit the separate radiation term and to express the influence of speed by the cube root of the mean piston speed, V_p.

Pflaum (30), on the basis of time-averaged heat flux data, has proposed modification of Eq. (A-2) to account for the effects of higher engine speeds and supercharged operation. His most recent proposal is to replace Eq. (A-2) by the expression:

$$h(t) = f_1 [P(t), T_g(t)] \cdot f_2(V_p) \cdot f_3(P_1) \qquad (A-3)$$

where:

$$f_1 [P(t), T(t)] = 0.0399 (P(t) \cdot T_g(t))^{1/2} \qquad (A-4)$$

$$f_2(V_p) = 6.2 - 5.2(5.7)^{-(0.0305 V_p)^2} + 0.00762 V_p \qquad (A-5)$$

$$f_3(P_1) = 1.175 (P_1)^{1/4} \text{ for the cylinder head} \qquad (A-6)$$

where P_1 is the intake pressure in psia. Note that at a fixed speed and intake pressure, the Pflaum correlation reduces to the same form as the Eichelberg correlation but with a different constant term.

Henein (31) obtained poor results in applying Eichelberg's correlation to a prechamber engine when the mean piston speed was used for V_p. When he substituted estimates of the swirl and squish velocities for V_p he obtained reasonable agreement between the experiment and the correlation for the compression stroke but not for the expansion stroke.

Fig. 17 shows the degree to which the Eichelberg and Pflaum correlations fit the authors' experimental results from the SOC.

CORRELATIONS BASED ON REYNOLD'S ANALOGY OF BOUNDARY LAYER THEORY A number of authors have used the Nusselt number-Reynolds number relationships of steady state systems to correlate engine heat transfer data. Professor C. F. Taylor (32. 33) advocated the use of dimensionless quantities such as Nu and Re in correlating time-averaged heat fluxes from several engines. However, apart from one brief reference, Herzfeld in Nagel (34), it is only recently that correlations of instantaneous surface heat transfer based on a Nu-Re relationship have been put forth in the literature.

Annand gave a rather extensive review of several correlations for h(t) and used dimensional analysis to arrive at the relation:

$$Nu = (\text{constant}) \cdot Re^n \qquad (A-7)$$

to correlate the convective heat flux. He suggested that the radiant heat flux be expressed by:

$$q_r(t) = c\left(T_g(t)^4 - T_w(t)^4\right) \qquad (A-8)$$

where "c" is a constant.

From a reanalysis of Elser's (35) data from a low-speed, 4-stroke diesel engine, Annand formulated the relation

$$q(t) = a\frac{k(t)}{D}(Re)^b\left(T_g(t) - T_w(t)\right) + c\left(T_g(t)^4 - T_w(t)^4\right) \qquad (A-9)$$

for the instantaneous surface heat flux, where:

$k(t)$ - gas conductivity, $\frac{\text{Btu}}{\text{hr ft deg R}}$
D = bore diameter, ft
a = 0.49
b = 0.7
$c = (1.03 \pm 0.37)\, 10^{-9}\ \frac{\text{Btu}}{\text{hr ft}^2 \text{ deg R}}$

Annand chose to select the bore diameter and the average piston speed as the significant distance and velocity to be used in Re of Eq. (A-7).

Woschni repeated some of the bomb experiments of Nusselt and concluded that the results of such experiments are not suitable for application to engine heat transfer. Woschni proceeded to formulate a correlation for the heat transfer coefficient using the well-known correlation of turbulent heat transfer in pipes. $Nu \propto Re^{.8}$ as his starting point. He chose bore diameter and mean piston speed as significant quantities in the Reynolds number but applied multiplying constants to the mean piston speed. Woschni's correlation may be summarized as:

$$Nu = 0.035\, Re^{.8} \qquad (A-10)$$

with cylinder bore as the characteristic length and the following expressions for the gas velocity in Re:

$V_g = 6.18\, c_m$ scavenging
$V_g = 2.28\, c_m$ compression
$V_g = 2.28\, c_m + (3.24)\, 10^{-3} \frac{V_s T_1}{P_1 V_1} (P_g - P_{g0})$
combustion and expansion

where:

c_m = average piston speed in m/sec

V_s = total cylinder displacement

T_1, P_1 and V_1 = cylinder gas temperature, pressure and volume at some convenient reference state

$(P_g - P_{g0})$ = instantaneous pressure difference between the fired and motored cycles.

Woschni determined the constants in the expressions for V_g by fitting the time-averaged results of his correlation to heat balance data from the engine.

Fig. 18 shows expermental data for the SOC compared to the heat fluxes calculated by Eq. (1) when the Annand and Woschni correlations are used for h(t). The term which Annand attributed to radiation is shown and seen to be quite small.

APPENDIX B

SIGNIFICANT VELOCITIES AND DISTANCES TO BE USED IN REYNOLDS NUMBER

To fix the functional relationship of Eq. (2) the significant quantities, particularly gas velocity and distance, are considered. The choices of a significant velocity and a significant distance are not independent, as pointed out in the following discussion.

SIGNIFICANT OR CHARACTERISTIC VELOCITY - The gas velocities in the motored engine are classified as being piston related or intake related.

Piston Related - The piston motion generates several gas velocities. One is a velocity perpendicular to the cylinder head and piston, which creates a stagnation-type heat transfer situation if the piston and head areas are flat and parallel. This is the gas velocity which is incorporated into the conduction-compression model. In reality, the piston and head surfaces are not flat but quite irregular due to protruding valves, valve cutouts in the piston, and a combustion cavity in the piston. Due to the presence of a combustion cavity in the piston (see Fig. 3) the piston motion introduces radial velocities parallel to the head and piston surfaces. During compression the flow is radially inward and during expansion it is radially outward. These velocities, termed squish velocities in the engine literature, have been considered by a number of authors, for example, Loeffler (36) and Fitzgeorge (37). Fitzgeorge shows the magnitude of the squish velocities to be highly dependent on the head-to-piston clearance at TDC. The authors have calculated a peak squish velocity in the engine at TC-1 of about 50 ft/sec based on a radial flow into the combustion cavity from the lip area. However, the actual squish velocity is significantly less than this value due to squish flow into the valve cutouts in the piston. More over, as reported by Alcock (38) several attempts to determine experimental evidence of squish in a motored engine have yielded inconclusive results. On this basis the authors consider squish velocities to be relatively insigniftcant in the motored engine.

Intake Related - During the intake process the intake port and valve combination can introduce significant gas velocities. In many engines, part of the design intent is to impart swirling gas motion about the bore axis. Shipinski (1967) has determined the mean angular velocity of the swirl motion in the engine used in this study to be approximately twice the angular velocity of the crankshaft. Strictly speaking, this swirl ratio, that is, swirl angular velocity divided by crankshaft angular velocity, applies only at the closing of the intake valve.

At intake valve closure (50 deg CA) the gas is swirling about the axis of the cylinder with diameter equal to the bore B. At TDC the major portion (about 80%) of the gas is in the combustion chamber which has a diameter of approximately 0.55 B. The axis of the combustion chamber is 0.5 in., or 0.11 B, from the bore axis. Thus, there is a movement of the center of mass of the swirling gas during compression and expansion. Because of the conservation of angular momentum, the angular velocity in the cornbustion chamber near TDC is approximately four times its value at BDC in the presence of negligible viscous dissipation.

Okaya (39) presents a method whereby the swirl deceleration due to viscous effects may be calculated. Using these results the authors have determined changes in swirl velocity during one engine revolution of approximately 10% and 20% of the value at intake valve closure for the BDC and TDC cases mentioned above.

The changes in the angular velocities in the lip area at TC-1 and TC-2 are influenced by the motion of the center of mass, the acceleration or deceleration due to the conservation of angular momentum, and the deceleration due to viscous effects. The viscous effects in the lip area near TDC are higher than in the combustion chamber but are counteracted by the acceleration necessary to conserve the angular momentum. As a first approximation the authors consider the swirl velocity to be constant throughout the engine cycle.

Semenov has published the only experimental results known to the authors on turbulence in engines. He used an 8 micron resistance wire in a 3.25 X 4.50 in. CFR engine under motored operation. The engine was an open chamber one with a flat piston and head. Semenov's results included: during intake considerable temporal and spacial variations in the gas velocity exist with peak values being as much as 10 times the mean piston speed, significant gas velocity gradients are found within 2-3 mm of the cylinder head; and the fluctuating component of the gas velocity decreases rapidly after intake valve closure. The turbulence which, exists throughout the compression stroke is essentially isotopic.

Semenov indicates that he did not use a shrouded intake valve and gives no indication of determining a swirl velocity. Since Semenov's engine was so dissimilar to the engine used in this project, many of the trends in Semenov's results may not be applicable in the present instance.

From the above discussion on gas velocities in the motored engine it is apparent that both squish and swirl velocities would be expected to vary with position in the cylinder. It is concluded that squish velocities are relatively insignificant in the motored engine. Moreover, the calculated squish velocity at TC-1 is greater than that at TC-2. This is opposite to the trend expected from the motored heat flux results. Thus the local squish velocity is not a good choice for the significant or characteristic velocity.

Thus swirl velocities are considered as the predominant velocities in the motored engine during compression and expansion. Certainly there are gas velocities related to the intake process, often referred to as jet velocities, which are significant for that portion of the cycle. However, for present purposes attention is focused on the velocities during the compression and expansion process.

SIGNIFICANT OR CHARACTERISTIC DISTANCE - Accepting the significant velocity as the swirl velocity, the selection of the significant distance must be compatible. The gas flow pattern due to swirl flow can be looked upon as having many similarities to the flow near rotating discs. An excellent review of the fluid flow and heat transfer problems associated with rotating systems is found in Dorfman. The flow pattern close to the cylinder head and piston would resemble that near a casing, which encloses a rotating disc, if the effects of the irregularities in the engine cylinder surfaces are considered small. That is, the tangential velocity increases with the radius and a secondary radially inward flow exists on the head and piston. This secondary flow is radially outward in the bulk gas in the engine whereas it is radially outward near the disc for the enclosed rotating disc. Soo (40) and Daily (41) present extensive studies of this type of flow. In general the flow Reynolds number characterizing the flow in rotating systems is defined as:

$$Re = \frac{r^2 \omega}{\nu} \tag{B-1}$$

As a result of the success achieved in correlating friction factors and heat transfer coefficients with this definition of the Reynolds number in rotating systems, the authors consider the local radius to be the significant distance in Eq. (2).

Using Eq. (B-1) flow Reynolds numbers are found to be in the range 10^5 to 6×10^5 . Dorfman gives $Re = 3 \times 10^5$ as a transition Re for rotating systems. Due to the irregularities in the cylinder head and piston surfaces and the turbulent nature of the intake process, the flow is taken to be turbulent.

DISCUSSION

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This extremely interesting paper contributes much to our understanding of the heat-transfer process in engines. There are some points on which I should like to comment.

1. Squish: In his Appendix B he mentions that Mr. Scott and myself could not find any squish in a motored engine. As regards inward squish during compression this is correct. But our paper (Authors' Ref. 38) showed, in our Fig. 9, a strong outward "unsquish" on the expansion stroke even when motoring, In a firing engine this "unsquish" would be even greater, due to the pressure rise in the bowl where combustion starts. From the photographs in Fig. 3 of our paper the radial "unsquish" velocity appears to be of the same order of magnitude as the tangential swirl velocity. In a firing engine this "unsquish" gas is hot, and its radial velocity must increase the heat transfer. This may account for some of the flux difference between couples 1 and 2.

2. Swirl: The authors say "Also the significant velocity is considered equal to rω, where ω is the angular velocity" and r is the radius, in other words a forced vortex. In near spherical prechambers we have found a "semifree" vortex, with linear velocity independent of radius. We have also found much the same relationship in other types of chamber.

Allowance for this would reduce the calculated heat transfer at TC2, and thus the discrepancy between theory and experiment shown in Fig. 26.

AUTHORS' CLOSURE
TO DISCUSSION

We appreciate Mr. Alcock's kind remarks and comments on our paper. Regarding our interpretation of his studies, Ref. 38 of the paper, our statement was that the results were inconclusive for the motored engine. Mr. Alcock's remarks above essentially enforce this statement since it is difficult to rationalize the existence, in the motored engine, of outward squish without the presence of inward squish.

Figs. 21 and 22 of the paper show that the heat fluxes measured in the motoring engine are essentially symmetrical about TDC at TC-1 and peak about 3-5 deg BTDC at TC-2. During motoring, one would not expect appreciable instantaneous property differences of the gas from one position to the other. Thus, the different heat flux-time curves may well be due to different gas velocities at these two points. The symmetry of the flux-time curve at TC-1 may not result from a constant gas velocity, $r\omega$ as assumed in the authors' model, but rather from a combination of velocities, one of which is the squish velocity. The experimental evidence gathered by Alcock and Scott (Ref. 38) on the pre-dominance of outward squish over inward squish might then explain the difference in motored fluxes between TC-1 and TC-2, since squish velocities would be higher at TC-1 than at TC-2. The authors do not feel that the experimental data currently available on instantaneous gas velocities in a motored engine warrant the use of this complex model of the gas motion.

The increased outward radial velocity in a firing engine (termed unsquish by Alcock) is certainly expected to contribute to the different heat fluxes measured at TC-1 and TC-2. However, Appendix B pertains to the selection of a significant velocity gas velocity for the motoring engine.

The authors' have some difficulty understanding the term semifree" vortex in a spherical prechamber. Thus, we are unable to speculate on its usefulness in a conceptual model of surface heat transfer in a motoring engine.