Trans.I.Mar.E., Vol.100, pp.133–143

The potential of ceramics and insulation in marine diesel engines

N. P. Kyrtatos, BSc, DIC, PhD
National Technical University of Athens

SYNOPSIS

The potential benefits from the use of high-temperature ceramic materials and insulation in four- and two-stroke marine diesel engines were investigated with the aid of computer simulation. The characteristics of engineering ceramic materials for high-temperature applications are briefly described and the engine process simulation models used are outlined. If the engine combustion space is insulated, the heat rejected to the coolant is reduced. This allows a reduction in the cooling required by the engine but also leads to increased combustion chamber wall temperatures. The ability of some engineering ceramics to withstand higher temperatures than conventional metals, combined in some cases with higher thermal resistance, makes them attractive for such applications. A small part of the energy formerly rejected to the surroundings is transformed into some extra reciprocator work, thus improving the engine's efficiency, but most of it appears as higher exhaust gas temperatures, thus increasing the energy available for waste-heat recovery. Four-stroke engines suffer from volumetric efficiency loss because of charge air heating from the hotter combustion space components. Two-stroke uniflow engines are much less affected by this problem and thus they are more suitable for the use of insulation. The performance of an insulated two-stroke uniflow-type large marine engine was investigated in more detail using the engine simulation model by varying the level of insulation of individual components, optimizing the turbocharger match and including a power-recovery turbine.

INTRODUCTION

In the last decade, the continuous development of high-temperature ceramic materials with excellent heat and corrosion resistance properties has provided the impetus for extensive research and development work in low-heat-rejection internal combustion engines.[1, 2] The aim of this has primarily been to improve the total energy efficiency of the engine, but other possible benefits have emerged such as the ability of such an engine to burn a wide range of fuels and, in the longer term, the potential of improved engine reliability through reductions in friction and component wear.

Relevant research work is aimed on the one hand at improving the thermal and mechanical properties of ceramics, as well as the processes of bonding and fastening ceramics and other materials, and on the other hand at investigating the performance of low-heat-rejection engines through experiments and computer simulations.

It has been shown[3] that insulating the combustion space of an engine results in increased temperatures of the walls adjacent to the hot gas, with most of the heat formerly lost to the coolant being redirected to increase the energy content of the exhaust gases. In general, only small improvements in the engine power or efficiency were observed, partly because of the detrimental effect of the hotter walls on engine volumetric efficiency. It was concluded that the main potential for improving the overall energy efficiency of the powerplant lies with 'waste energy' recovery methods. However, the capital cost of energy-recovery equipment may be unacceptable for relatively small engines.

For marine engines, whether two or four stoke, the relative cost of additional waste-heat recovery equipment becomes more acceptable as the engine rated power increases. In addition, with reference to deteriorating and variable fuel quality, the potential for enhanced wear and corrosion resistance of ceramics, as well as the possible improvements in the combustion process because of hotter engine walls, make ceramic materials attractive for applications in heavy-fuel engines.

Nicholas Kyrtatos is Assistant Professor of Marine Engineering at the National Technical University of Athens, Greece. He obtained a B.Sc. in Marine Engineering from the University of Newcastle upon Tyne in 1975 and a Ph.D. in Mechanical Engineering from Imperial College, London in 1979. He was Postdoctoral Research Assistant at Imperial College from 1979 to1980 and Visiting Professor in the Department of Mechanical Engineering at McGill University, Montreal from 1980 to 1982. He then served at the Greek Airforce Research Centre, Athens, until 1984 when he took up his present position.

HIGH-TEMPERATURE ENGINEERING CERAMICS

Engineering ceramics, also known as fine ceramics or new ceramics, are man-made materials consisting of metallic elements such as aluminium, silicon, magnesium and boron, chemically combined in precise proportions with non-metallic elements such as carbon, nitrogen and oxygen. They can be generally classified into ceramic oxides such as alumina, zirconia, magnesia and beryllia, non-oxidized ceramics such as carbides and nitrides, glass ceramics such as lithium aluminium silicate (LAS) and aluminium magnesium silicate (AMS), and composite materials such as fibre-toughened ceramics.[4]

Each class generally designates a broad family of alloy systems with wide property variations. Since the physical properties of ceramics are dependent on chemistry, microstructure and fabrication techniques, these must be defined exactly in order to specify these materials. Test data on ceramic mechanical properties exhibit wide scatter and so design strength cannot be firmly established. This variability is best dealt with by using statistical concepts, in which failure is attributed to flaws randomly distributed throughout the material.

Designing with ceramics requires special considerations, since these materials are generally brittle and any localized stresses are not relieved by local plastic deformation. Moreover, ceramics usually have high moduli of elasticity and poor impact strength. Therefore highly refined stress analysis is needed in ceramic design.

Ceramic materials in the form of monolithic components, tiles and coatings have been used in combustion chambers of experimental engines. Since ceramic coatings can be used on existing engines, there have been numerous demonstrations of the use of coatings as thermal barriers on conventional metal engine components. Some of this activity is directed towards marine engines.[5, 6, 7] One of the main problems is establishing lasting adhesion of the ceramic to the metal substratum.[8]

Although in general the thermal conductivity of ceramics is lower than that of metals, in most cases additional insulation backing the high-temperature ceramic is required for the combination to act as a thermal barrier. Manufacturing techniques giving variable cross-sectional density ceramic components or the development of new families of ceramic materials may solve these design problems.

Regarding the potential use of high-temperature ceramic component combustion chambers in heavy-fuel engines, it has been reported[9] that ceramics do not remain inert in the presence of combustion. Their microstructure may be affected by combustion products, especially sodium and vanadium compounds; ceramic–metal bonds of porous coatings may fail by the ingress of corrosive products; local hot spots may be caused by carbon deposition on relatively rough or porous ceramic surfaces; rough surfaces may also increase the local turbulence in the gas/wall boundary layer, increasing heat transfer; and certain ceramics may act as catalysts and affect combustion. Extensive additional work is required in these areas.

INSULATED ENGINE PERFORMANCE PREDICTION

In this paper, the performances of two thermally insulated marine diesel engines, a four-stroke high-speed engine and a two-stroke uniflow low-speed engine of similar power output, are compared by using computer simulation. The engine simulation model is then used to examine in more detail the effect of thermal insulation on the performance of a two-stroke uniflow large marine engine.

Methodology

The approach followed was purely thermodynamic, implying that no engine dynamics or structural aspects were considered, apart from the definition of the insulated combustion chamber models, as described below.

The engine simulation model used in the present work is a general-purpose program, which can be used to simulate various types of engines. The engine model is of the 'control volume' or 'filling and emptying' type.[10] This model treats a multi-cylinder engine as a series of thermodynamic control volumes interconnected through valves or ports. The use of the control volume concept implies two basic assumptions. First, spatial uniformity of fluid properties is assumed within any control volume at any instant (or within a 'time step' in the computation), i.e. no property gradients, perfect mixing of fluids at different states and homogeneous mixtures at thermodynamic equilibrium at each instant. Secondly, processes are assumed to be quasi-steady, i.e. for any time interval, the rate of change of any parameter is assumed constant.

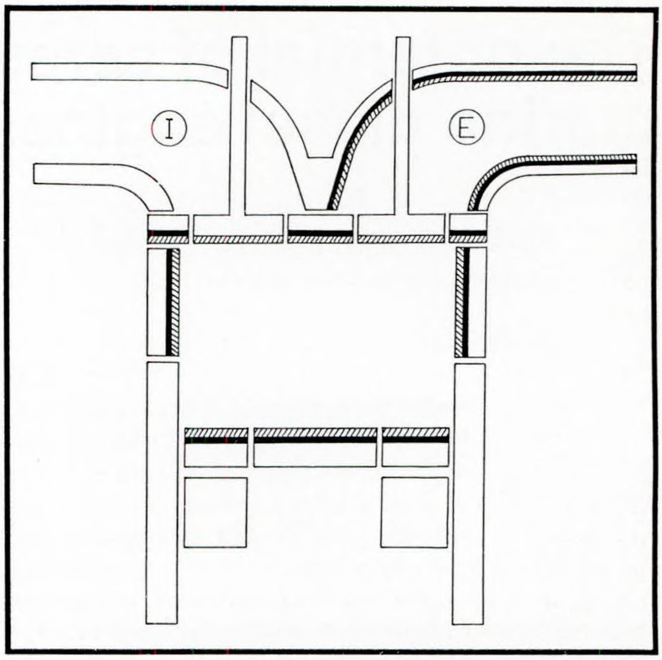

Fig. 1. Insulated cylinder model (four-stroke engine)

In order to evaluate the work, heat and mass transfer taking place across the boundaries of the control volumes, the conservation equations are applied in appropriate forms to each control volume. A number of supplementary models for the various processes inside the engine are also required. These include models for combustion, heat transfer, friction, gas exchange as well as gas property relationships. All these are in the form of subroutines appended to the basic program.

The elementary form of the engine system is considered to be composed of the turbocharger compressor, the inlet manifold, the reciprocator (cylinders), the exhaust manifold, and the turbocharger turbine. Models for various engine components and appendages such as additional turbochargers, power turbines, air coolers and exhaust gas boilers may also be included. For simulating a particular engine, the versions of the subroutines containing the appropriate models have to be selected. Full geometric data for the engine are required and complete performance data for the turbocharger and other turbomachinery components. The basic engine modelling computer program used is described in refs. 3 and 11.

The process models used in the present work are briefly described below with emphasis on the model for heat transfer.

Heat transfer. In the heat-transfer sub-model, the heat flow path between the hot gas inside the cylinder and the surroundings is divided into three parts.

(1) Heat transfer from gas to combustion chamber wall.
(2) Conduction through the wall.
(3) Heat transfer from wall to coolant, assumed at constant temperature.

For the first part, for the four-stroke engine, a Woschni-type[12] bulk heat-transfer coefficient correlation based on turbulent flow in pipes was used with appropriate calibration carried over from previous investigations on the conventional metal engine. For the two-stroke engine a correlation proposed by Bulaty[13] was used. This correlation modifies the Woschni[12] correlation to account for the dependence of the Prandtl number on the Nusselt number for the case of large stroke/bore ratio engines. The calibrations of the correlations for the reference (metal) condition were retained for the subsequent insulated engine runs.

For the detailed investigations of the two-stroke engine, the Bulaty[13] correlation was further modified to include an additional term proposed by Woschni *et al.*[14] for the case of high surface-temperature walls of insulated engines. This correlation is based on experimental results for insulated engines, where it was deduced that the bulk gas-to-wall heat-transfer coefficient increased in the first part of combustion as the surface temperature of the combustion chamber walls increased. The heat transfer at the exhaust valve seat was based on Hausen's equation,[13] while the heat transfer at the exhaust port was based on a correlation suggested by Zapf[13] accounting for a variable Prandtl number.

For the second part of the heat flow path, a one-dimensional heat flux model based on the equivalent electrical circuit concept was used. The geometries of the combustion chambers are shown in Fig. 1 for the four-stroke engine and in Fig. 2 for the two-stroke engine. In the model a composite wall design is used, with the option of using up to three layers with different thicknesses and material properties. The liner and piston crown can be subdivided to obtain a temperature distribution. The surface temperature of each component part is assumed to be constant over an engine cycle and is updated every engine cycle by integrating the instantaneous heat flows and applying the heat flux equation.

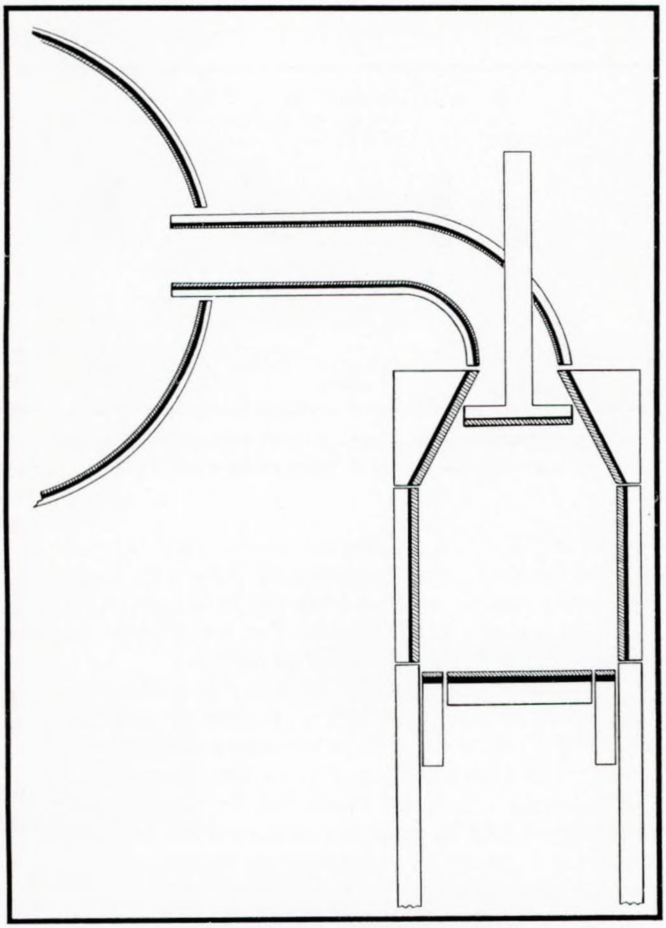

Fig. 2. Insulated cylinder model (two-stroke engine)

For the third part of the heat flow path, available empirical metal-to-coolant bulk heat-transfer coefficients were retained throughout the investigations.

Scavenging. For the two-stroke uniflow engine a separate scavenging model was used during the gas-exchange process. In this model, the engine cylinder can be subdivided axially into three zones. A burned gas zone fills the cylinder until the inlet ports open and a fresh air zone and a mixing zone are then established. The intrusion of the air zone into the burned gas zone and hence the growth of the mixing zone, is governed by two time-varying coefficients. Each zone exchanges heat with its adjacent zones and chamber walls.[15]

Combustion. A Wiebe[16]-type heat-release curve was adapted so as to match available pressure data for the four-stroke engine. For the two-stroke engine, experimentally derived fuel burning rate diagrams for diesel fuel were used. The heat-release curves for the reference condition were retained for the insulated engine runs, assuming no changes in the ignition delay or the shape of the heat-release curves.

Friction. The formula of Chen & Flynn[17] was used for predicting the frictional losses for the four-stroke engine. For the two-stroke engine experimental values of frictional losses were available. The values for the reference condition were retained for the insulated engine runs.

Performance prediction results

For the first part of the study two marine engines were chosen as reference engines: a high-speed four-stroke 20 cylinder V engine developing 2650 kW at 1700 rev./min and a slow-speed two-stroke uniflow scavenged six-cylinder engine developing 2980 kW at 200 rev./min. The reason for the choice of these two engines was mainly the availability of experimental data for calibration purposes. Program calibration was performed at the MCR point of each engine and the set of predicted values was taken as reference for the other runs.

A set of results was obtained for each engine, simulating an 'adiabatic cylinder' engine, with no heat transfer between the working fluid and its adjacent walls. Performance predictions for such a hypothetical engine made of zero conductivity/zero heat capacity combustion chamber materials can easily be obtained by setting the gas/wall heat-transfer coefficient to zero. The results are included for comparison purposes, although probably certain sub-models of the engine simulation program would have been operating beyond their useful range.

As shown in Fig. 1 for the four-stroke engine and in Fig. 2 for the two-stroke engine, a ceramic/air gap/metal sandwich was used as the basic insulated engine wall design. The original overall wall thickness of the various component parts of the reference (metal) engine was retained. The high-temperature first layer was assumed to be partially stabilized zirconia (PSZ) ceramic with thermal conductivity 0.0025 $kW/m\cdot K^{-1}$, assuming no variation of thermal conductivity with temperature. The insulator second layer was assumed to be stagnant air at atmospheric pressure. The load-bearing material third layer was assumed to be metal, with properties dependent on the component part. In both engines only the top one-third of the

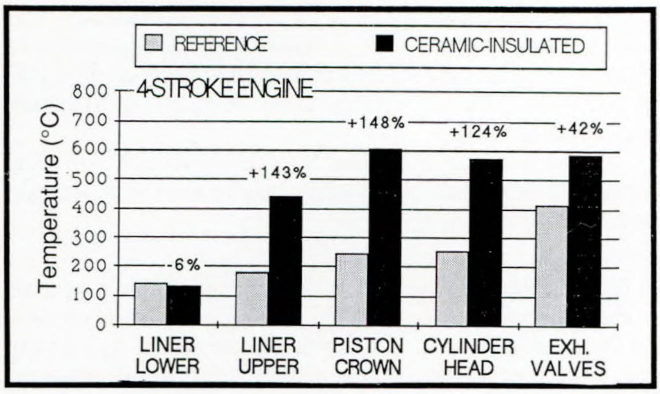

Fig. 3. Component temperatures in combustion chamber (four-stroke engine)

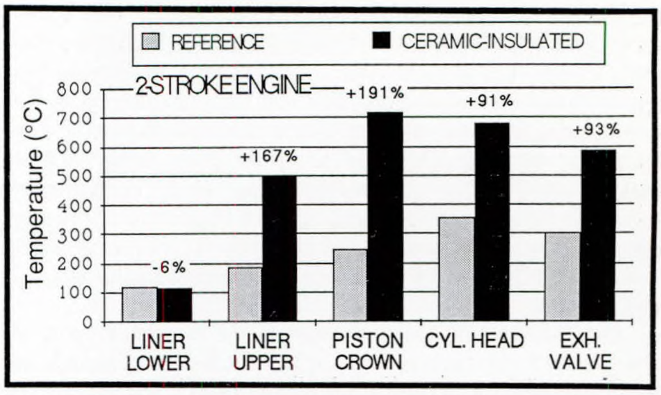

Fig. 4. Component temperatures in combustion chamber (two-stroke engine)

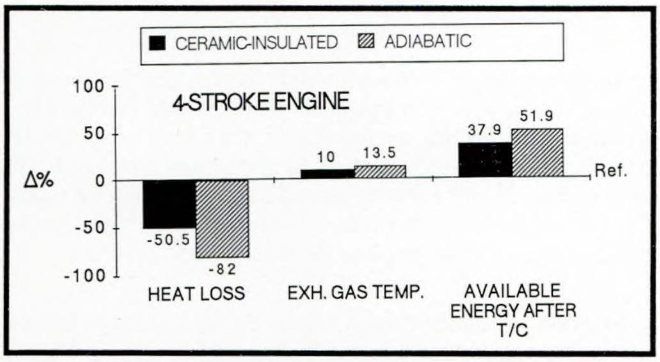

Fig. 5. Percentage changes in various parameters (four-stroke engine) from reference values

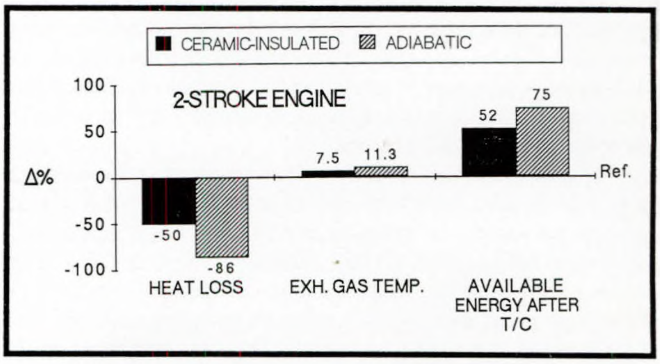

Fig. 6. Percentage changes in various parameters (two-stroke engine) from reference values

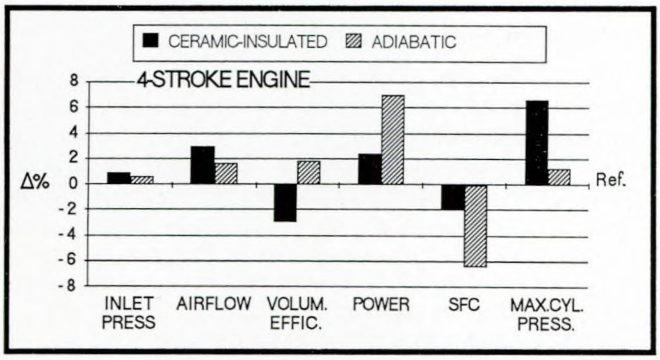

Fig. 7. Percentage changes in various parameters (four-stroke engine) from reference values

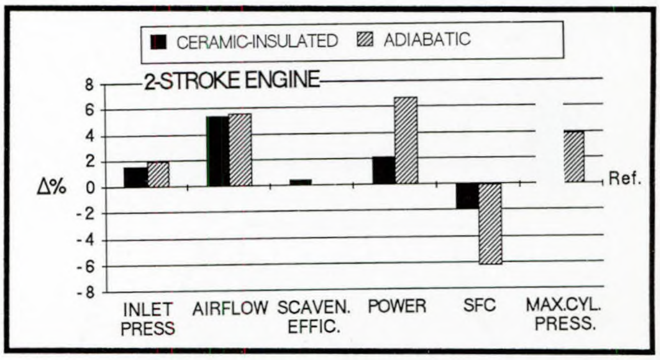

Fig. 8. Percentage changes in various parameters (two-stroke engine) from reference values

liner was insulated, in addition to the piston crown, cylinder head, valve faces and exhaust port. Material properties were assumed to remain constant with increasing temperature.

Assuming the first layer to be a thick ceramic coating of up to 25% of the total thickness of the original (metal) component, the insulator layer thickness was adjusted until a 50% reduction in the total heat loss in relation to the reference engine was obtained. In general, this was achieved with insulator thicknesses of not more than 5% of the total component thickness. These figures can be regarded as medium-term design targets.

All the results presented are to the engine MCR point with the fuelling and engine speed kept constant. All the runs were performed under ISO ambient conditions and with diesel fuel of LCV 42 500 kJ/kg.

The immediate consequence of insulation was an increase in the temperature of the combustion chamber components. As shown in Fig. 3, a substantial temperature increase was observed at the ceramic/insulated top part of the liner of the four-stroke engine, with no temperature change of the non-insulated bottom part of the liner. This was more pronounced for the two-stroke engine, as shown in Fig. 4.

This can be explained by the fact that in the two-stroke uniflow engine there is no inflow of relatively cool charge air at the upper part of the liner and therefore this part remains in contact with high-temperature gases for most of the engine cycle. Similar trends are shown for the temperature of the piston crown and the mean temperature of the exhaust valve, where again the values and percentage increases for the two-stroke engine are higher.

The percentage reduction in total heat outflow for the ceramic/insulated and 'adiabatic cylinder' engine configurations in relation to the reference engine are shown in Fig. 5 for the four-stroke engine and in Fig. 6 for the two-stroke engine.

Apart from the increased component temperatures, the main effect of insulation was the increase of the exhaust gas temperature, as shown in Figs. 5 and 6. The results presented were obtained after the turbocharger turbines were rematched by increasing the swallowing capacity so as to obtain approximately the same inlet pressure as in the reference case. This rematching was necessary because of the increased exhaust gas energy of the insulated engines.

An undesirable effect of the hot chamber walls was the heating of the incoming charge air. This was more pronounced in the four-stroke engine and resulted in a drop in volumetric efficiency. This could have been partly offset by the additional boost provided by the turbocharger due to the increased exhaust gas energy, if the turbocharger was not rematched. On the other hand the rematched (larger) turbine imposed a lower back pressure to the engine, hence the drop in volumetric efficiency for the four-stroke engine was not substantial (see Fig. 7), and in fact the airflow was increased.

With the two-stroke uniflow engine there was no inlet air charge heating problem, hence with a rematched larger turbine the scavenging efficiency increased slightly (see Fig. 8), and the airflow also improved. The increased airflow resulted in better purity of the trapped air and this contributed to the increased power in both engines. Equivalent trends were observed for the specific fuel consumption. The higher values of trapped air mass and temperature resulted in higher maximum cylinder pressures (see Figs. 7 and 8). It is again noted that, since the effect of ceramics and insulation on combustion cannot be readily predicted quantitatively, the reference heat-release diagrams for each engine were kept unchanged for all runs.

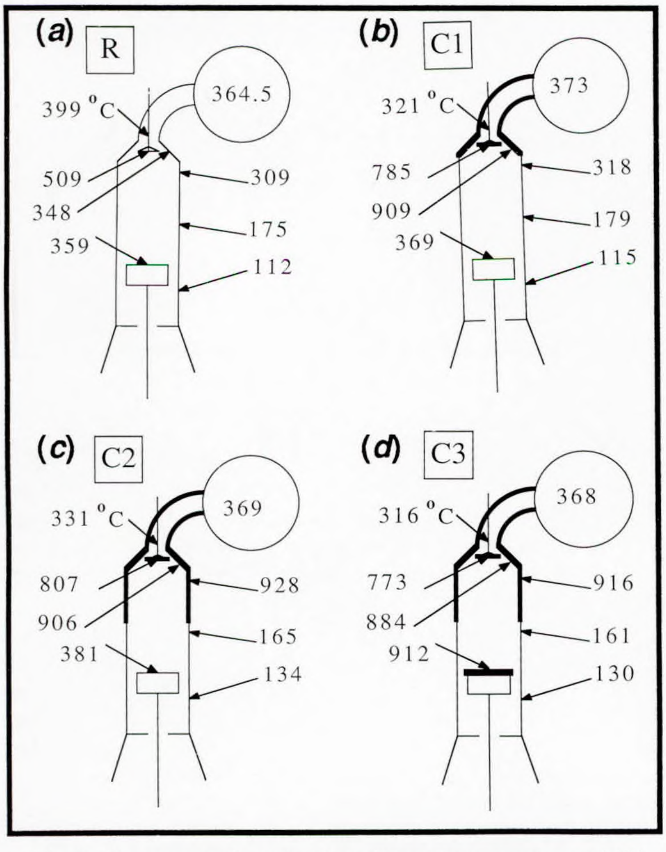

Fig. 9. Component and exhaust gas temperatures (°C) in (*a*) reference engine, (*b*) configuration C1, (*c*) configuration C2 and (*d*) configuration C3

The exhaust gas energy available after the turbocharger turbine is shown in Fig. 5 for the four-stroke engine and Fig. 6 for the two-stroke engine. In both cases the final condition of the exhaust gas was assumed to be at a temperature of 180 °C and ambient pressure. It can be seen that although the absolute values of the 'waste energy' available are higher in the four-stroke engine because of the higher exhaust gas temperatures, the percentage change is larger in the two-stroke engine, hence the relative benefit from the application of insulation can be larger for this type of engine.

Additional detailed parametric investigations were undertaken for various configurations of a large two-stroke uniflow marine engine. In order to allow the inclusion of a power turbine in the system, a larger six-cylinder engine developing 9000 kW at 110 rev./min was taken as reference. For all the configurations examined, all runs were performed at the MCR point, with the same fuelling and the same heat-release diagram for diesel fuel as for the reference engine. The injection timing and the engine compression ratio were also kept unaltered for all runs.

Three insulated engine configurations were examined, denoted C1, C2 and C3. The combustion chamber geometry shown in Fig. 2 and the insulation material properties of the postulated PSZ/air/metal sandwich used for the two-stroke engine in the previous comparative investigations were retained.

In configuration C3, the cylinder head, the valve face, the exhaust port, the top one-third of the liner and the piston crown were insulated. The insulation was adjusted to obtain approximately 40% reduction of the total heat loss. The resulting insulation thickness for each insulated component was kept unchanged in all configurations.

Configuration C2 was like C3 but with no insulation on the piston crown.

Configuration C1 was like C3 but with no insulation on the piston crown and the liner.

In each configuration the turbocharger was rematched in order to restrain the increase in maximum cylinder pressure to below 3%.

The temperatures at various parts of the cylinder of the reference engine (denoted R) in Fig. 9(*a*) can be compared with the temperatures for the insulated engine configurations C1, C2 and C3 in Fig. 9(*b*–*d*). It must be again noted that each configuration was individually optimized by rematching the turbocharger. Apart from the expected increase in temperature of the insulated parts, it is interesting to note the increase in temperature of the metal components of the partly insulated engine configurations.

This can be explained by referring to Figs. 10(*a*)–10(*d*), showing the heat flux through the various parts of the cylinder to the coolant for the configurations R, C1, C2 and C3. As the engine is progressively insulated, and although the total heat flux out of the combustion chamber is reduced, more heat flows through the non-insulated metal parts, increasing their temperature. This effect may be generally undesirable, although it may prove beneficial for the liner walls.

The exhaust gas temperatures of the insulated engines also increase in relation to the reference engine (Figs. 9*a*–9*d*), although with the rematching of the turbocharger the resulting increases are quite small.

Fig. 11(*a*) shows the engine power W_{E}, the total heat lost from the cylinder to the coolant Q_{L}, and the exhaust gas energy available after the turbocharger turbine E_{a} for the reference engine. The equivalent parameter values for the engine configurations C1, C2 and C3 are shown in Figs. 11(*b*)–11(*d*). It can be observed that despite the progressive reduction in the total heat loss, as more component parts are insulated, the resulting benefits in engine power are very small. The accompanying increases in available exhaust gas energy are larger but not very significant.

Another set of performance prediction results was obtained by coupling a power turbine to the reference engine (configuration R/PT) and to engine C3 (configuration C3/PT). The power turbine was parallel to the turbocharger and geared to the main engine. Its size and speed (gear ratio) were optimized so that the turbine would operate at its design point with approximately 10% of the total exhaust gas flowing through the power turbine at the available expansion ratio. In each configuration the turbocharger was rematched to retain the original charging pressure.

As shown in Fig. 12(*a*), the addition of a power turbine to the reference engine (configuration R/PT) improves the total power extracted from the engine ($W_E + W_{PT}$), with some penalty to the exhaust energy available since no engine setting was altered apart from the rematching of the turbocharger. When a power recovery turbine is attached to the fully insulated engine (configuration C3/PT) (see Fig. 12*b*), the engine power and the total power increase by a few percent, and the exhaust gas energy available also increases, following the reduction in heat loss.

FINAL COMMENTS

Before evaluating the results presented in this paper, it is instructive to review briefly some problem areas and possible shortcomings of the simulation code and the various submodels used for predicting the insulated engine performance.

1. Combustion. This is a major problem area *per se* because of the complexity of the phenomenon and its profound influence on all other processes. In this study, the reference heat-release diagrams were used for all insulated engine runs for each engine, although this may not be realistic in practice. The influence of the higher wall temperature on fuel characteristics will probably affect ignition delay and combustion, especially for heavy fuels.
2. Friction. The effects of ceramics on frictional losses were not accounted for in the simulation. These will be relatively more significant in the case of the four-stroke engine, where the piston ring friction is a larger proportion of the total friction, than in the two-stroke engine. Tribological problems, such as the deterioration of performance of conventional lubricants at high temperatures through evaporation or carbonization, operational problems, such as the effect of ceramics and insulation on deposit formation and engine component wear patterns, and structural design problems, such as the effect of the part-ceramic liner on piston rings, were not included in the present study.
3. Heat transfer. An inherent feature of the control volume models is the spatial uniformity of the gas temperature. Moreover, the heat-transfer correlations used are global empirical formulations for the convective coefficient that can be calibrated to fit engine data and were used unchanged for the insulated engine runs. Additionally, changes in radiation heat transfer, which may be of some importance in the ceramic component engines, were not accounted for.

The assumption of constant wall surface temperatures will

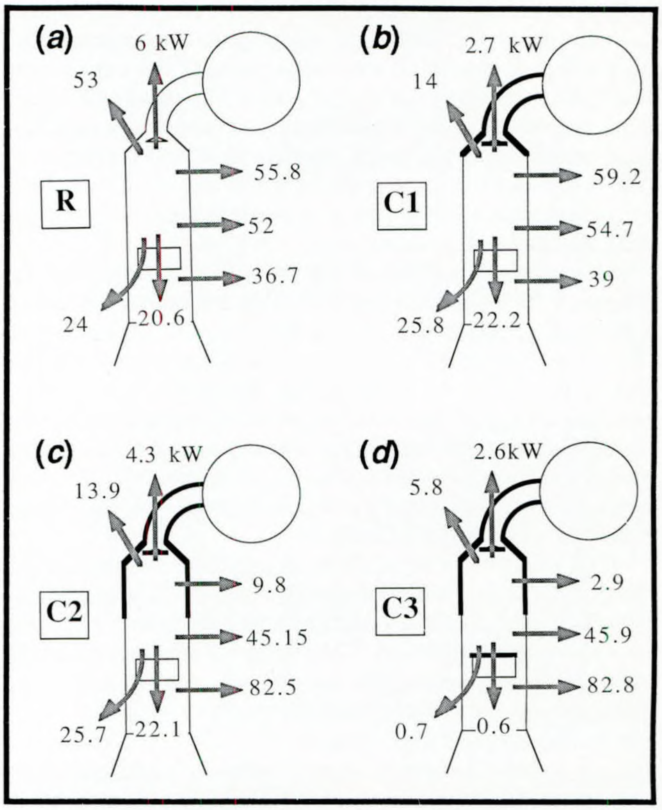

Fig. 10. Heat flux (kW) through various parts of the cylinder in (*a*) reference engine, (*b*) configuration C1, (*c*) configuration C2 and (*d*) configuration C3

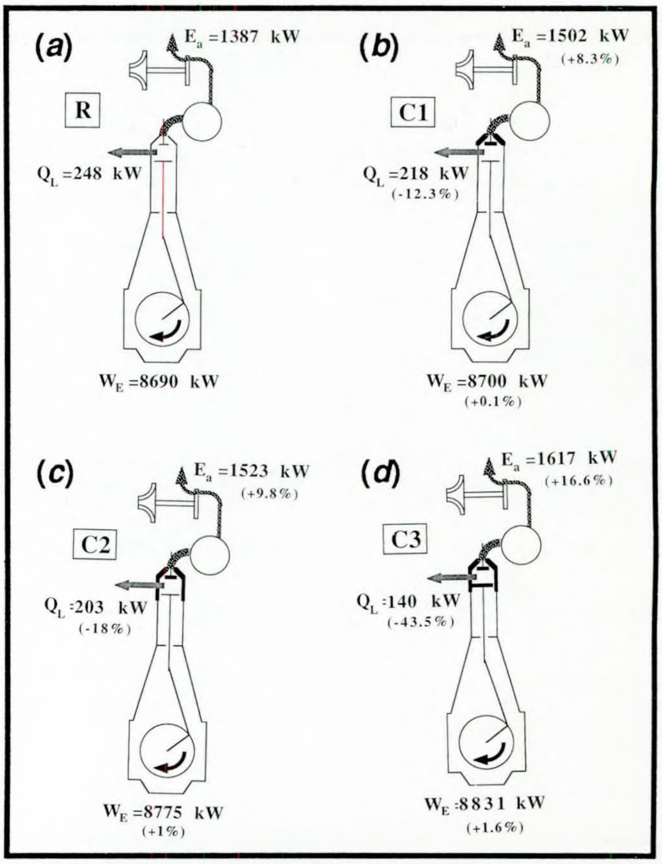

Fig. 11. Engine power (W_E), total heat lost (Q_L) and exhaust gas energy available (E_a) in (*a*) reference engine, (*b*) configuration C1, (*c*) configuration C2 and (*d*) configuration C3

not be valid for low-conductivity ceramic materials. However, the effect of this on engine performance is not significant, if the degree of insulation is not excessively large.[3, 18]

The problems referred to above would probably be more pronounced if performance predictions for the whole operating range of each engine were undertaken.

CONCLUSIONS

The following conclusions may be drawn from this study.

1. Engine insulation reduces the heat flow to the coolant and increases the temperature of the engine combustion chamber components. This was more pronounced in the two-stroke uniflow engine. Ceramic materials are able to withstand such high temperatures.
2. Most of the energy thus saved appears in the form of higher-temperature exhaust gases. Higher boost can therefore be obtained if the turbocharger is not rematched or the same boost can be obtained with a larger turbine, which also reduces the back pressure on the engine.
3. A small part of the energy appears as increased expansion work, which together with the reduced back pressure and increased airflow contribute to a small improvement in engine sfc. Similar improvements were predicted for both engines. A small additional gain will be the reduction in power absorbed by the cooling system.
4. Insulated four-stroke engines suffer from charge air heating and loss of volumetric efficiency. This can be largely compensated for by the increased turbocharger boost using the higher-energy exhaust gases.
5. Insulated two-stroke uniflow engines do not suffer from the above problem and because of the flow direction of charge air and exhaust gases during gas exchange, the increase in component temperatures is generally higher. This may be advantageous for heavy-fuel combustion. Furthermore, the percentage increase in exhaust 'waste energy' available after the turbocharger is also larger. From this point of view, the application of high-temperature ceramics and combustion chamber insulation can be relatively more beneficial in this type of engine.

From the detailed investigations of the two-stroke marine engine it was reconfirmed that the only significant effect of engine insulation is the increase in exhaust gas energy available after the turbocharger, which can be utilized for increased steam production, heating purposes, etc. Even so, it would require a fully insulated engine (C3) to achieve a relatively modest increase (17%) in the available 'waste heat'. Partly insulated engine combustion chambers do not make much sense thermodynamically, since at least some of the heat flow is redirected through the non-insulated parts with a consequent increase in their temperature, although this may not be a disadvantage in the case of the liner. The overall performance benefits from various insulated engine configurations are at best very small.

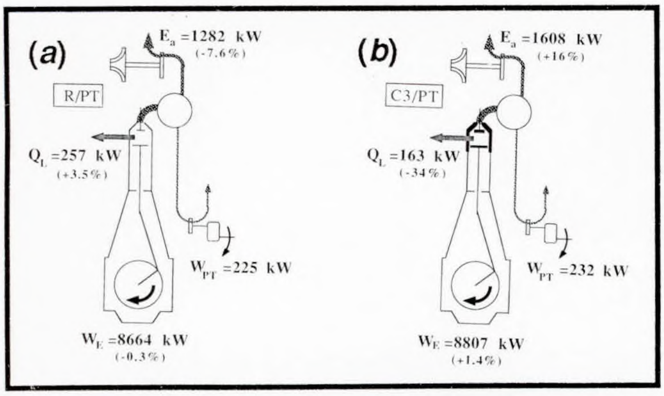

Fig. 12. Engine power (W_E), power recovery turbine power (W_{PT}), total heat loss (Q_L) and exhaust gas energy available (E_a) in (*a*) reference engine and (*b*) fully insulated engine

The addition of a power turbine to the reference engine increases the total power extracted and hence the efficiency of the engine, but these effects are not substantially augmented by insulating the engine.

In view of the above it may be concluded that the thermodynamic advantages gained by insulating large marine engines are generally quite marginal. The use of ceramics inside engine combustion chambers may prove advantageous if the prospects for improved heavy-fuel burning ability and reduced wear are realized, and provided that the numerous design problems are solved. This may prove to be the most profitable area for further research.

ACKNOWLEDGEMENTS

The author wishes to acknowledge the contributions of his research assistants I. Koumbarelis and J. Tsolkas. Work on applications of ceramics to engines was partly supported by the Ministry of Industry/R&D Secretariat contract No. 5955/EPE/479. Development work on engine simulation is also supported by Lloyd's Register of Shipping, London, and the A.S. Onassis Foundation, Liechtenstein, as part of the research project 'Diesel Engine Design and Operation for Variable Quality Fuels'.

REFERENCES

1. W. Bryzik & R. Kamo, eds., *'The Adiabatic Engine. Past, Present and Future Developments'*, SAE/PT-84/28 (1984).
2. W. Bryzik, D. Siegla & R. Kamo, eds., *'Advances in Adiabatic Engines'*, SAE/SP-85/610 (1985).
3. N. Watson, N. Kyrtatos & K. Holmes, 'The performance potential of limited cooled diesel engines', *Proc. I. Mech. E.*, vol. 197A, 45/83 (1983).
4. 'Ceramics and engines', *The Motor Ship*, p. 55 (May 1984).
5. G. Grosshans, 'Theory and applications of adiabatic diesel engines', *Alsthom Review*, no.1, pp. 21–30 (1985).
6. M. B. Smavik, 'An experimental study of the effect of thermal barriers on heat transfer and material temperature in a two-stroke diesel engine', paper D-19, CIMAC '85 Congress, Oslo (1985).
7. 'Test rig discovers exhaust valve durability', *MER*, p.18 (March 1984).
8. D. H. Harris, 'Practical aspects of ultra-thick thermal barrier coatings', ASM 8512-018 (1985).
9. J. M. Buckley-Golder, 'An overview of ceramic applications in engines', *The Motor Ship*, 8th International Marine Propulsion Conference, London (1986).
10. N. Watson & M. Janota, *Turbocharging the Internal Combustion Engine*, chapt.15, Macmillan, London (1984).
11. N. Kyrtatos, 'Engine simulation and performance prediction' RINA/IMarE/NTUA Seminar Appl. High Tech. Shipping, Athens, Greece (1986).

12. G. A. Woschni, 'A universally applicable equation for the instantaneous heat-transfer coefficient in the internal combustion engine', SAE Paper 67O931 (1967).
13. T. Bulaty, 'Beitrag zur Berechnung der Warmeuberganges inbesondere in langespulten, langhubigen Dieselmotoren', *MTZ* 46/2, pp. 67–70 (1985).
14. G. Woschni, K. Kolesa & W. Spindler, 'Isolierung der Brennraumwande - Ein lohnendes Entwicklungsziel bei Verbrennungsmotoren?' *MTZ* 47/12, pp. 495–500 (1986).
15. N. Kyrtatos & I. Koumbarelis, 'A three-zone scavenging model for two-stroke engines', Congress on Modelling of IC Engines, CMT87O7, University of Valencia (1987) .
16. I. Wiebe, 'Brennverlauf und Kreisprozess von Verbrennungsmotoren', *VEB* , Verlag Technik, Berlin (1970).
17. S. K. Chen & P. Flynn, 'Development of a compression ignition research engine', SAE paper 65O733 (1965).
18. D. N. Assanis & J. B. Heywood, 'Development and use of a computer simulation of the turbocompounded diesel system for engine performance and component heat transfer studies', SAE paper 86O329 (1986).

Discussion

K. J. MOLLISON (Bauteil Marine Installations Ltd.): Does the author anticipate that, should ceramic insulating material find an economic place in marine engine technology, it will cause:

(*a*) the requirement of additional monitoring equipment to be designed and installed to adequately monitor the engine and if so what does the author foresee regarding instrumentation requirements over and above traditional engine monitoring?

(*b*) the ceramic-furnished engine to be more predictable in wear and failure than traditional engines by application of suitable predictive algorithms, and would algorithms currently applied to traditional engines be suitable?

The paper considers rated-load engine simulation. In conducting the described research, did the author conduct any part-load application simulation, and if so were any anomalies observed?

Does the author expect ceramic technology to reduce heat losses from engine cylinders to an absolute minimum, thus allowing air-cooled free-ventilated marine engines to be used, and if so what major hurdles are forseen?

D. TSALAPATIS (MAN B7W Diesel A/S): We should like to congratulate the author on his investigations of the potential of ceramics and insulation in marine diesel engines.

We agree with the author that coating the combustion chamber walls with a heat insulation material increases the thermal resistance, which cannot be advisable solely for reduction of the SFOC, but should be considered as part of the total thermal efficiency improvement to be obtained by energy systems, such as recovery of coolant waste heat and exhaust gas waste heat, or by a turbocompound system.

We should appreciate your comments on the two following points.

(1) The heat transfer model is essential for the possibility of accurately predicting the engine performance for insulated or partly insulated engines. Therefore, we would be interested in your experience with the modified Woschni heat-transfer model in connection with calibration of the model and its universal use for the engine when calibrated.

(2) With reference to Fig.10, we ask you to kindly comment on the heat fluxes (real heat flow) to the cylinder liner. According to our experience from testing and calculations, the heat flow in the lower third of the liner is negligible and the heat flow in the middle third is small. In your results, there is almost equal heat flow to all three parts of the liner on the reference.

We would also consider the heat flow of 82.8 kW through the lower (uncooled) part of the liner for case C3 (Fig. 10*d*) to be rather unrealistic.

Seen from an engine designer's point of view, we do not consider it advisable to evaluate the effects on engine performance of using ceramic insulating layers in the engine when $P_{max.}$, MEP, $P_{scav.}$, and T/C efficiency are not kept constant, as changes in these parameters can be responsible for the major part of the changes of SFOC, air flow, heat flow, and available energy after T/C. Have you carried out any calculations where these parameters were kept constant and if so would you please comment on the results?

Professor N. WATSON (Imperial College): You started the work using Woschni's heat-transfer correlation for conventional engines, then changed to use his revised correlation for insulated engines. Given the controversy in the U.S. over this revised correlation, what difference did the change make to the calculated performance?

Could you comment on the lower liner temperature reducing in Fig. 4 but increasing in Fig. 9? What is the thermodynamic reason for this?

J. T. STANSFELD (Lloyd's Register of Shipping): I would like to congratulate Professor Kyrtatos on his paper which illustrates very well the use of a cycle simulation program to investigate the potential of a new area of engine technology without expensive laboratory developments.

In the predication of performance the author uses a ceramic/air gap/metal sandwich as the basic insulated engine wall design. The insulating air layer was assumed to be stagnant air at atmospheric pressure backed up by the load-bearing material which was assumed to be metal. If the pressure loading is to be transmitted in practice by some sort of partially porous ceramic, presumably the heat flow characteristics would be different. Perhaps the author would comment on the choice of 'sandwich' construction and the likely performance results without the insulating air layer.

One of the advantages mentioned in the paper for the use of ceramics is the increase in component temperatures that can be achieved. While this may be desirable to increase the waste heat available, the component temperatures will enter the range in which corrosion from heavy fuel combustion products can occur. The author reports that ceramic materials are susceptible to this type of corrosion at present. Given that marine engines will continue to burn heavy fuels, perhaps the author would comment on the likely practical future of ceramics in marine engines.

Author's reply

With reference to **Mr. Mollison**'s first question, the possible introduction of ceramic materials in marine diesel engines may require novel engine component designs as well as associated developments in the area of lubrication, such as the possibility of using solid lubricants, or the integration of the lubrication and cooling systems. This may in turn require new auxiliary equipment but will not directly influence the monitoring techniques and equipment. On the other hand, thermal barrier ceramic coatings rather than monolithic ceramic parts will probably be used in commercial engine applications. It has been reported[1], that the durability of ceramic coatings may be affected by the thermal environment of individual cylinders. This calls for more detailed information on the in-cylinder process to maintain the design environment and avoid coating failure. If local failure occurs, it would be difficult to diagnose unless additional instrumentation is available. Extensive failure would probably influence some of the overall performance parameters and could be located by accurate engine performance monitoring combined with fault diagnosis systems. The data acquisition and processing facilities in the engine management systems envisaged in the various 'Ship of the Future' projects, will probably be adequate for any engines with insulated combustion chambers.

With regard to the behaviour of wear and the failure modes of ceramic components in engines, the assumption that the ceramic materials have found an economic place in marine engine technology would imply that these components have demonstrated acceptable reliability. Significant advances have been made in ceramic materials research, with new materials and fabrication techniques giving improved properties being continually announced. Additionally, a number of hardware applications have been demonstrated, but considerable further effort is required before these materials are accepted by commercial operators. Standard engine design algorithms are currently used for design of ceramic applications in engines, such as finite element methods for stress and thermal analysis, but in all cases, account has to be taken of the variability in ceramic material properties. With reference to low-load simulation, most of the various process models used in the insulated engine simulation were carried over from the reference (metal) engine, although formally their applicability in such an extrapolation can be disputed. The models were also tuned by using reference engine performance data. For part-load calculations, as well as calculations at very high levels of insulation, the models have to be further 'stretched', which leads to added uncertainty in the results. Hence, part-load calculations were not included in the paper. Regarding the last question, I believe that there will be some time before an uncooled marine engine becomes possible. Reducing the cooling would require materials that can withstand the increased thermal load. To reduce the heat losses to the surroundings, the engine would have to be further insulated, which would add to the increase of component temperatures. Apart from the charge air heating problems, which can be balanced by the increased exhaust gas energy available to the turbocharger, and the material reliability problems mentioned above, the major obstacle will be the associated tribological problems.

I thank **Mr. D. Tsalapatis** for his kind comments, which also provide me with an opportunity to clarify some points. In the paper, two sets of results of simulating insulated engines are presented. Firstly, two different types of engines, a four-stroke and a two-stroke, are compared, and secondly, a large two-stroke marine engine is investigated in more detail.

For the comparison between the four- and the two-stroke engines, the engine cylinder liner was divided into two parts, with the top part being 30% of the total liner length in both cases. In addition, the temperature distribution along the liner in these cases was based on a linear equation fitted to the temperature mean values of the upper and lower part, which were assumed to act at the mid-point of each part. The large increase in temperature of the upper part when insulated resulted, in some cases, in and underestimation of the heat flux of the lower part.

On the other hand, in the heat transfer model used in the detailed investigations, the engine liner of the large two-stroke engine was divided into three (unequal) parts. The top part (which was insulated in configurations C1, C2 and C3) was 9% of the total liner length, the middle part was 25%, and the bottom part 66%. The liner heat fluxes of Fig. 10, as well as the mean temperatures of Fig. 9, correspond to this division and are considered to be realistic.

For the three-part liner used in the detailed investigations, a higher degree polynomial was fitted to the mean values of the temperatures of the three parts. Evidently, this gives better accuracy and would make the results in Fig. 9 more reliable, although it was found that in certain cases the temperature of the bottom tip of the liner was slightly overestimated.

The above would also explain the inconsistency in temperature noticed by **Professor Watson**. Both **Mr. Tsalapatis** and Professor Watson enquire about the use of the additional temperature-dependent term of the modified Woschni correlation for heat transfer in the detailed investigations on the performance of the insulated large marine engine. The additional term in the heat transfer equation would only affect high-temperature components and thus it did not make much difference to the performance of the reference (metal) engine at high loads, although some of the component temperatures were overestimated.

For the reasons explained above, low-load simulation results were not considered to be reliable, so I cannot comment on the applicability of the equation throughout the engine operating range. A comparison of the use of the original and modified heat-transfer correlations in the fully insulated engine simulation (configuration C3) did not change the overall performance predictions appreciably, although both the surface temperatures and the heat flux of the insulated high-temperature parts were altered (by up to 33% for the heat flux of the top 9% of the liner in configuration C3).

In response to **Mr. Tsalapatis**'s last question, in the insulated engine runs the turbocharger was rematched and inevitably the turbocharger efficiency was slightly changed. As a result of the rematching, the changes in scavenge pressure, maximum cylinder pressure and bmep, in relation to the reference engine, were small (maximum 1.1% for the scavenge pressure, 2.3% for the P_{max}, 1.6% for the bmep, for the fully insulated engine configuration C3). Undoubtedly, such changes will influence the overall engine performance parameters, but since the changes are small, the influence on the overall trends would be small. The purpose of the study was to provide an overall appraisal of the effects of insulation in large marine engines. In running the simulation program, when moving from the reference to the insulated engine, all sub-models, where there was no specific information of how the process will be affected by insulation, were left unchanged.

In the Final Comments section of the paper, the shortcomings of the method are presented. The most important effects are the influence of insulation on the heat-release curve and the heat-transfer process. However, a number of runs performed with presumed changes in the heat-release pattern and increases in the radiation heat-transfer did not change the qualitative appearance of the overall results for the performance of the insulated engines. I thank **Mr. Christensen** for the interesting information contained in his contribution. Regarding his first question, refs. 1–3 (below) report some of the recent work on ceramic coatings.

The figures for fuel consumption improvement quoted by Mr. Christensen probably refer to some early reports on the TACOM–Cummins adiabatic engine which was based on the NHC 250 Cummins commercial engine. However, recent experimental results by Cummins for a turbocharged V-903 engine, which was progressively insulated at fixed maximum cylinder pressure showed no improvement in bsfc[4]. The power increase or sfc reduction results presented in Fig. 11 in the paper are actually worse than the results shown in Figs. 7 and 8. Since more detailed modelling techniques were used for the former case, these results, showing very small changes in sfc with insulation, are considered more reliable, although the combustion process will most probably be influenced by insulation and high wall temperatures, this is unlikely to have a dramatic effect on performance.

Finally, with reference to **Mr. Christensen**'s last question, I do not expect that the insulated ceramic engine would find an application as a gas generator for a power turbine for marine propulsion. If such an application domain can be envisaged, and provided the ceramic materials reach the required stage of maturity, advanced gas turbines with high-temperature ceramic components to increase the turbine inlet temperature, would be more likely candidates. They are much simpler in layout and, incidentally, they are also much simpler to model. I thank **Mr. Stanfsfeld** for his remarks on the usefulness of engine process simulation programs, to which I would hasten to add that the appreciation of the limits of a simulation program by the user, is often as important as the quality of the models.

Regarding the other points raised by Mr. Stansfeld, the sandwich construction assumed for the insulated wall model was chosen because it provided a convenient way to alter the wall thermal properties by changing the relative thicknesses of the various layers, as, for example, the wall thermal resistance, which was controlled by adjusting the width of the air gap. In practice, a layered coating of ceramic on metal design, with alloying of ceramic and metal in various proportions depending on the layer location to reduce the coating internal stresses, seems to be, at the moment, the best approach for thermal barriers[2]. Such a coating, if sufficiently thick, can provide the required thermal resistance for reductions of the total engine cylinder heat rejection of about 40%. Up to that point, an almost linear relationship exists between the performance differences from the non-insulated reference engine and the level of insulation.

Fuel ingredients and combustion products have been found to attack the ceramic–metal bond and destabilize the structure of some thermal barrier ceramic coatings. Sealing with ceramic slurry, top coats of corrosion-resistant materials, and other methods have been used to cover and protect the underlying coating. Future practical applications of ceramics in marine engine combustion chambers will depend mainly on advances in materials technology permitting further improvements in wear and corrosion resistance.

References

1. A. V. Levy & S. Macadam, 'Durability of ceramic coatings in 14,000 hours service in a marine diesel engine', ASME paper 88-ICE-19 (1988).
2. K. L. Hoag *et al.*, 'Modelling of Thick Thermal Barrier Coatings', ASME paper 88-ICE-20 (1988).
3. R. A. Miller, 'Current status of thermal barrier coatings: an overview'. Surface and coatings technology, vol. 30, Elsevier Sequoia, Lausanne (1987).
4. K. L. Hoag, Contribution to the discussion of the ASME paper by G. Woschni & W. Spindler, 'Heat transfer with insulated cylinder walls and its influence on the performance of diesel engines', in 'Basic processes in internal combustion engines', ICE (T. Uzkan & J. M. Bailey, eds.), vol. 3, ASME (1988).