GÖTAVERKEN TURBO-CHARGED CROSSHEAD DIESEL ENGINES

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Further Correspondence

Professor A. Sarsten wrote that the emphasis the authors placed upon matching air supply and turbocharging, and upon the detailed design of the thermally loaded components surrounding and forming the combustion chamber of the engine was interesting. His own institute was mainly concerned with computer calculation of temperature fields and stresses, and it was especially gratifying to see the work being done by one of the major engine builders in the continuous development of these thermally loaded components to meet the requirements of higher loads and increased component life and reliability.

In the Norwegian Large-bore Diesel Engine Research Project mentioned in the paper, his institute's computer calculations, based upon temperature measurements in service indicated that local temperatures in an area of the upper surface of the piston crown could become critical under abnormal operating conditions. Computer calculations of von Mises stresses showed that engine types having a general piston shape akin to the unribbed, oil-cooled type shown schematically in Fig. 21, displayed an area of initial yield near the upper surface of the piston (shown hatched). This was mainly due to the thermal expansion of the hottest portions and the influence of temperature on the yield point. The general course of the isotherms on such pistons was also indicated. The extent of this area of critical yielding depended upon the particular design and operating conditions, and could in some configurations extend completely across the bowl. In some designs a second, smaller area of initial yield developed at the transition radius on the inner surface (cross hatched). It must be noted that yielding of the piston did not in itself generally prove to be critical. However, the high surface temperatures did often lead to a burning away of the piston crown.

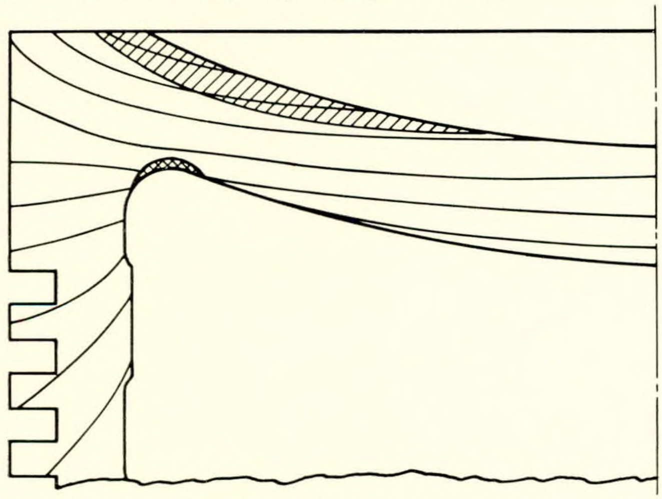

Fig. 21—*Area of initial yield and isotherms in typical piston (schematic).*

Computer calculations performed in the course of this research project indicated that thermal stresses were generally predominant in the critical areas of the thermally loaded components, and it was felt that further improvement ought to be possible in many cases by careful attention to section thicknesses. As a test run for subsequent research work in this field, regarding improvement of piston design, temperature fields and stresses were calculated for a number of variants of different piston designs and cylinder bores in order to establish trends and to study optimum section, thicknesses and design configurations. Finite element techniques (triangular elements) were employed for the computer stress calculations. As an example, Fig. 22 showed the upper part of a larger grid network covering a hypothetical super large bore size (1m bore) of oil-cooled piston. The heavier lines indicated the contours of the 7 variations of crown thickness and cutouts obtained by removing the data cards of the corresponding elements from the computer input deck. Due to the large number of variants covered, automatic computer-controlled plotting of the output was employed (Figs. 22 and 23).

Typical computer produced temperature fields were shown in Fig. 23 for a couple of variants, based upon a given set of boundary conditions. While calculated values at given points were dependent upon the engine size, load and the thermal boundary conditions specified, the trend as a whole did, however, agree well with the measured values presented in the paper for roughly similar configurations (see Fig. 9).

Research work on theoretical design of thermally loaded components was now being carried out under a government grant, and at a later date it might be possible to give more detailed results related to various components.

Dr. F. Ørbeck, in a written contribution, commented that in connexion with the matching of air supply and turbocharging some information from J-type Doxford Engines might be of interest for comparison purpose. The Doxford engines used longitudinal scavenging in conjunction with pulse-type turbocharging. With no scavenge pumps the possibilities of controlling the air flow over the engine speed (see Fig. 4a) were very limited. One would therefore aim for a satisfactory air flow at full load and accept what followed at the lower loads. The specific air quantities measured at full load for the 76J8 and 58J4 were 7·3 kg/bhp/h and 7·55 kg/bhph respectively and this was considered to be very satisfactory. The specific air quantity always decreased with increase in rev/min and for the 58J4 8·8 kg/bhp h was obtained at half load, i.e. similar to the top curve in Fig. 4a.

From the point of view of fuel consumption the Doxford engines ran satisfactorily with as low specific air quantities as 6 kg/bhph on the test bed. This, however, left inadequate margin for service owing to thermal loading. How far down in specific air quantity could the Götaverken engines go at full load before the fuel consumption was seriously affected? To reduce thermal loading, Doxford aimed at as high a specific air quantity at full load as possible without increasing the fuel consumption. Why the fuel consumption was affected adversely by increasing the airflow beyond a certain limit was not known in detail since it did not affect the mechanical work in the scavenge pumps as for the Götaverken engine. Regarding efficient combustion it should be expected that an increase in the excess air at full load over that at reduced load would be an advantage. In spite of the additional work for the scavenge pumps at full load one would, therefore,

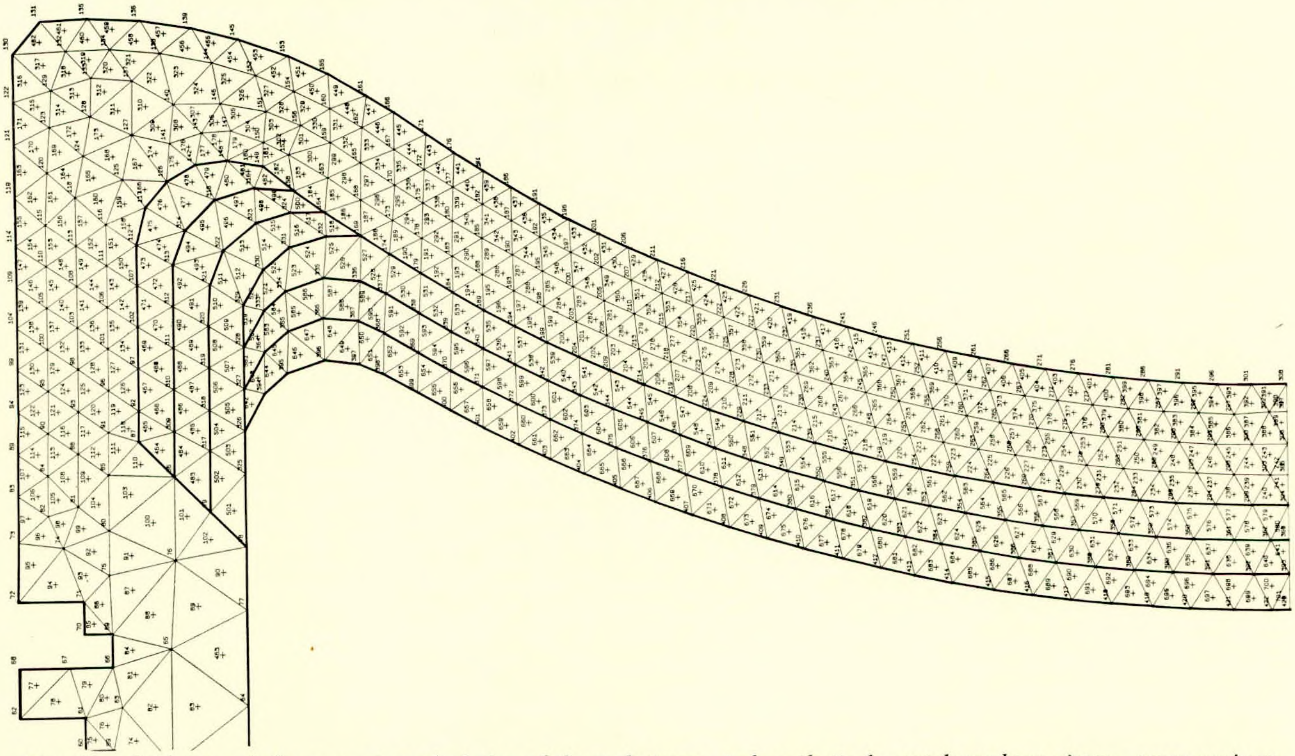

FIG. 22—*Grid network for computer calculation of thermal stresses in hypothetical super large-bore piston—seven variants.*

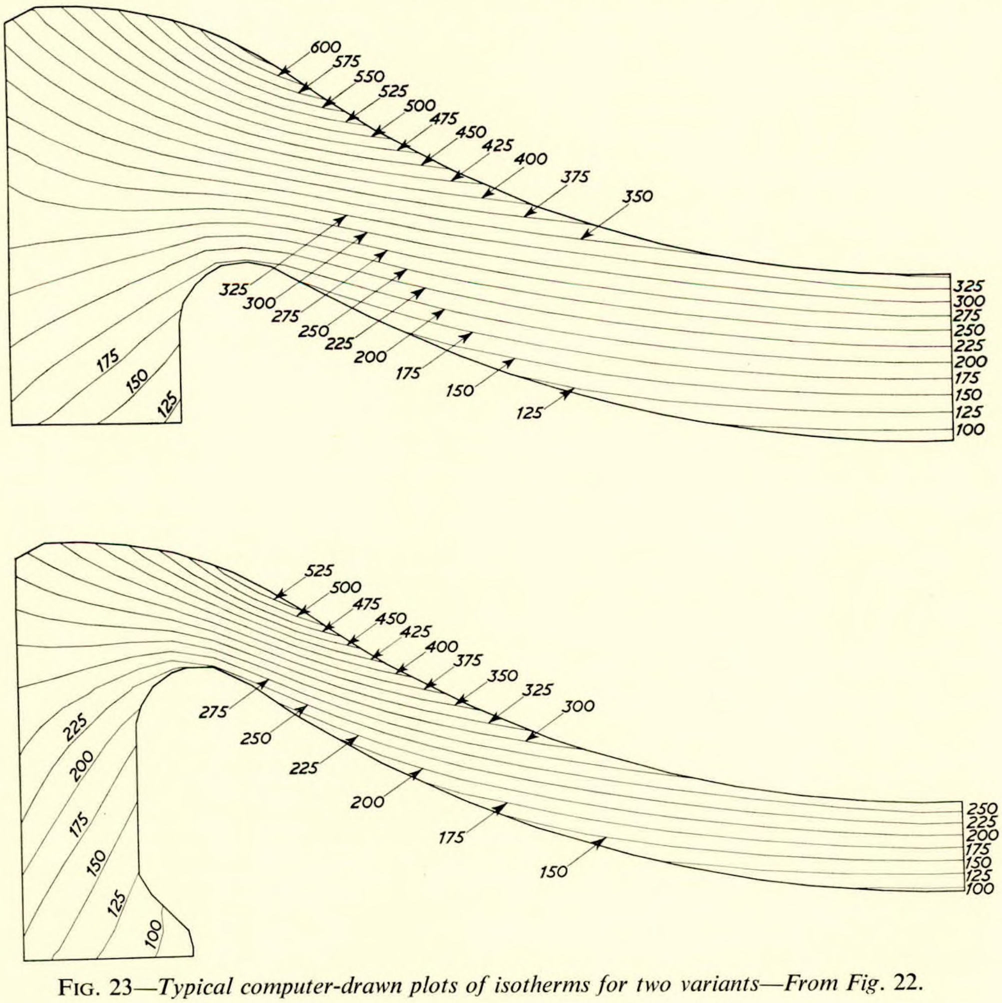

FIG. 23—*Typical computer-drawn plots of isotherms for two variants—From Fig. 22.*

have expected that Götaverken would have chosen specific air quantity increasing with speed.

The piston stresses shown in Fig. 8 demonstrated clearly the importance of the thermal stresses in comparison with the stresses due to pressure. Strain gauge measurements on Doxford engine combustion belts had given similar results, i.e. thermal stresses five-six times the stresses due to pressure.

Could the authors explain the method of calculation of piston stresses used by Götaverken and did the values given in Fig. 8 correspond to the maximum stress position?

MR. B. TAYLOR, B.Sc. (Member) in a written contribution endorsed the opinions put forward by the authors on the difficulties and cost of development work on large slow-speed engines. He pointed out that however long a prototype engine might be operated under controlled conditions on the engine-builders test bed, it was only in service that performance could be assessed. This made the collection of accurate performance data of the greatest importance and he congratulated the authors' company on introducing the method of coding of data for cable transmission.

The value of such information depended on the reliability of the instruments fitted to the engine for the measuring of temperatures and pressures. In Mr. Taylor's experience reasonably accurate measurements of temperatures by means of commercial instruments were difficult to obtain and, in this connexion, he advocated the use of local reading instruments wherever possible in preference to the distant reading type which might be mounted in the control room or at a centralized position.

The authors had stressed the importance of maintenance of high turbocharger efficiency in the constant pressure turbocharging system of the Götaverken engine; this was of course of equal importance in the pulse system as employed in the Doxford engine. It was now the practice to fit water washing equipment to both the turbine and blower ends of the turbochargers and Mr. Taylor assumed such equipment was employed on the Götaverken engine. Unfortunately, it was difficult to assess the effectiveness of water washing, mainly due to the wide variation in the quality of fuel oils. Could the authors give any results of their experience in this matter?

Several points referred to by the authors confirmed the findings of other enginebuilders. For example, the beneficial effect of streamlining of the exhaust system on the specific air flow through the engine, allowing higher outputs, had also been found on the Doxford engine; this had enabled the output to be increased by about 12 per cent with lower exhaust temperatures and the same fuel consumption.

Oil cooling of pistons was another matter where experience had been similar to that described by the authors. It had been found that reduction of the crown thickness of the pistons reduced thermal stresses and in the uniflow scavenged engine oil cooling was perfectly satisfactory. Tests had been carried out with water cooling of the lower pistons of Doxford engines but difficulties had been experienced in avoiding contamination of the lubricating oil with water and *vice versa*.

Mr. Taylor was interested to note the details of the de-aeration system shown in Fig. 12 and wondered whether the authors had considered the use of a cooling water system in which the water from the cylinders returned by gravity to a tank open to the atmosphere. This provided an effective method of allowing separation of entrained air from the water.

Finally, referring to the details of the camshaft chain drive shown in Fig. 13 he wondered whether a mistake had been made in the text where it was stated that the rubber-covered dampers were fitted with a clearance of only 0·5 mm between the chain and dampers. Under running conditions the position of the chain relative to the dampers must be different from the static position and it seemed that if the dampers were set so close when stationary there would be a risk of damage when running.

MR. E. P. CROWDY V.R.D., M.A. (Member) wrote that Fig. 9 showed how the temperature of that part of the piston crown exposed to combustion gases was reduced from 575°C, in the original design to 455°C in the second alternative. Though these developments resulted in a very desirable reduction in the temperature difference between the hot and cold surfaces (between 80° and 135°C) they also presumably resulted in more heat being rejected to the piston cooling oil. What increase in piston cooling oil outlet had been experienced and had fuel consumption been adversely affected?

Fig. 15a illustrated a Moment Compensator of the Lanchester type. Independent drive of a set of weights at each end of the engine must result in considerable snatch, due to the weights being driven up hill and down dale. Since the support bearings of the balanced weights would be subject to rotating load vectors, had any special precautions been taken to ensure a satisfactory service life of both the drive chain and support bearings?

MR. A. J. S. BENNETT, M.B.E. (Member), commented, in a written contribution on methods of silencing the turbo charging unit. The authors had reported that the compressor was responsible for a sound peak which had been measured to be about 15 dB higher than the mean noise level at the blade frequency of the compressor. What was the mean noise level at the time of the measurement?

Of the various methods of silencing which resulted in a reduction from 20 to 26 dB, the 90° bend in the pipe between the compressor and the air cooler appeared to be an important element. How much silencing was effected by the bend alone?

Did the authors consider that a silencer made of the absorbing materials used on the 90° bend but constructed for insertion in the straight length of pipe between the blower and the cooler would be of much benefit, assuming that there was plenty of room to accommodate a silencer in this position?

MR. P. JACKSON, M.Sc. (Member) wrote that the problems experienced in the operation of the large turbo-charged Götaverken engine were very similar to those experienced on Doxford and other engines. This was illustrated in the papers presented by the representatives of the manufacturers of large engines at the CIMAC Conference in Brussels last year. Mr. Jackson said he had studied the code system introduced by Götaverken to enable superintendents and chief engineers to send detailed data of the operating conditions of their engines at sea. He considered this a very clever and useful system which would save much money on cable and telex transmissions. He believed that Shell had a similar system for transmitting information from their tankers to head office, but he did not know of any other engine manufacturer having such facilities.

The section of the paper devoted to supercharging was very concise and explanatory and the problems of the size and efficiency of the reciprocating scavenge pumps assisting the constant pressure system of turbocharging were explained in detail. There were no such problems on the Doxford engine since scavenge pumps were not required with the more efficient pulse system of turbocharging now employed. Scavenging pumps were fitted to the first three turbocharged engines in the early 1950s before the full benefits of the pulse system were developed, and the size of the series scavenge pumps was determined as best at 1·25 times the volume of the cylinders, i.e. similar to the Götaverken findings. Doxford's never did well with scavenge pumps in parallel with the turbochargers.

Had Götaverken had any experience of scavenge pumps in parallel on their constant pressure system?

Mr. Jackson had been very interested to read of the research and experiments with various designs of piston heads, cylinder covers, and exhaust valves. The reduction of total stress as the surfaces exposed to combustion were thinned to reduce the thermal stress while increasing the mechanized stresses was to be expected, though few designers had the courage to do this until strain gauge tests showed the advantage. It was surprising to learn that the mechanical pulsating stresses were almost constant, irrespective of the thickness of the piston crown.

Chain dampers very similar to those shown in Fig. 13 had been employed on the Doxford engine for over forty years. They were faced with bronze pads and were now fitted about 3 mm clear of the chain which was in contrast with the ½ mm of the Götaverken practice.

In Mr. Jackson's experience with a small opposed piston engine having small upper pistons driving a camshaft connected to the main crankshaft by a chain, there had developed breakages of the chain and this had been replaced by the three-rod method of connecting a camshaft to a crankshaft. These rods broke and a flywheel had been fitted to the camshaft. In due course the camshaft broke and the flywheel ran up the shop. Subsequently, the flywheel was driven by friction plates and this was satisfactory until the clutch surfaces wore. Mr. Jackson wondered whether the flywheel connected to the camshaft of the Götaverken engine by a Holset damper system would be more successful (Fig. 14a).

Mr. Jackson said he was surprised to see the design of harmonic balancer driven by chain from the engine crankshaft (Fig. 15a). At least three companies in Britain had tried such an arrangement in the 1920s and the chains had failed in all cases. The problem was that the weights had to be lifted during one half of their revolution and then they dropped during the other half causing a variation of the driving torque from 100 per cent + to 100 per cent −, or in the chain, variation from a pull to a push of equal magnitude. Chains did not like a push and could not cope with such a variation of load.

In the case of the engine with which he was concerned, Mr. Jackson said that they next had tried a skew gear drive, but the gears pitted in about two months, so finally the drive was redesigned and a hardened and ground Maag high tensile steel gear was employed with a four inch face width. The weights were not very large being about 12 in radius and 12 in thick to give a balancing force of some eight tons on each end of the engine.

Another problem with harmonic balancers was in the bearings withstanding the swinging load of four tons for each weight. In the first design with white metal bearings, the white metal was scoured out in a few months and then plain roller bearings were fitted, but the rollers wore taper and ultimately Skefko heavy duty barrel roller bearings were fitted. These examples showed that harmonic balancers had problems and while they did provide an effective remedy to counterbalance free moments and forces, they introduced difficulties of their own.

Mr. J. R. B. Robertson (Member) wrote that he had found the paper extremely interesting since he had been associated with the maintenance of a five cylinder 760/1500 turbocharged engine for the last three years. Initially it was suggested that water washing of the turboblower turbine be fitted and this was carried out. However, it seemed that water washing unless carried out under completely clean conditions was of little use and he found it interesting that the authors suggested water washing once a week. Later it had been suggested that water washing should be applied to the compressor end of the turbine blower and Mr. Robertson was surprised that this was necessary as on each occasion that the turboblower had been opened up little or no deposit had been found either on the diffusers or the impeller itself.

A further point which was not stressed in the early stages was the necessity for maintaining the cleanliness of the scavenge air intercooler. Experience had proved that it was essential that this cooler be kept in a condition of maximum cleanliness, particularly on the air side where it would seem that quite a small quantity of oily deposit resulted in a very considerable fall off of efficiency. The situation of this cooler was such that it was difficult to work on although the water side might be dealt with, but the air side could only be sprayed with a solvent, washed and blown out which was not really effective. The only solution appeared to be a tank, and one was provided on this particular vessel which could be pushed below the cooler and the complete tube nest be lowered into same. A heating coil was provided in the tank and most satisfactory results had been obtained using a well known cleaning agent. It was however not easy for ship's engineers to carry out this work without providing special lifting supports and special lifting lugs for the cooler nest as the head room was most restricted. He felt that if all this equipment were necessary to keep the engine functioning efficiently, these water washing arrangements, the cleaning of the tank or some other means of cleaning the cooler on both water and air sides should be provided by the builder when the vessel was built. In addition to this, instructions regarding the recommended intervals of cleaning etc. should be given in the builder's hand book.

Mr. Robertson was interested in the one piece exhaust valve, as in his experience the built ones had given trouble through breaking up and causing damage to pistons liners, cylinder heads and blower. Had the builders developed the one piece valve for this reason? There was also the question of trouble with the valves fracturing in way of the retaining groove, although there was provision which was supposed to prevent the valve from dropping into the cylinder, which had also been experienced. Perhaps the authors would like to comment on this. It was also noticed that the new type of piston was reduced in thickness from 80 mm to 65 mm on the later designs and this would seem to minimize the safety factor and would result in less piston to burn away.

A piston from the vessel referred to, which had been severely burnt, was machine welded with special rods to the original thickness and this particular piston had now been in service for some time without any sign of deterioration whereas other pistons in the engine had deteriorated over the same period. Had the builders considered producing piston heads coated with some special material which would eliminate burning which appeared to take place on occasions when operating the engine on certain heavy fuels?

Authors' Reply

To Dr. Ørbeck's ideas concerning the scavenge air quantity, the authors answered that a reduction of the specific air quantity of the Götaverken engines on the test bed was a rather difficult matter, due to the constant pressure turbocharging system. An illustrative example was given. For one particular large-bore production unit 7·4 kg air/bhp h was reached at full load during normal test bed trial. The scavenge air pump volume was thereafter decreased by 25 per cent, the engine and the turbocharger being unchanged in other respects. The air quantity decreased by only a few per cent which meant that the volumetric efficiency of the air pumps increased by nearly 25 per cent. This meant, however, an inadequate margin for practical service. A decrease of the specific air quantity in service, to values as low as 6·0 kg/bhp h, indicated disturbances in the thermal process and in those cases the specific fuel consumption always increased. On the other hand, the authors underlined that the specific fuel consumption was not the only limiting factor when an increase of the specific air quantity was discussed. An increased air quantity for a fixed engine would only be reached by raising the scavenge air pressure which normally involved a higher maximum pressure in the cylinder. In that case the engine must be dimensioned to meet this higher pressure level. Raising the scavenge air pressure at maintained maximum pressure inevitably gave an increased fuel consumption. The fact that the fuel consumption increased with increased air quantity was also due to the higher flow resistance between the scavenging air belt and the turbine inlet, giving the same effect as if turbochangers with lower efficiency had been used. The turbocharging system—constant pressure type or impulse type—was of no importance in this connexion.

As to the influence of the load on the air delivery, if one wished to follow the bottom curve in Fig. 4a, one had to reduce the scavenge air pump volume and raise the scavenge air pressure because the specific air quantity at full load was the dimensioning factor. As already stated, this gave an inadequate margin for practical service. Besides, the turbocharger efficiency decreased due to higher pressure ratio on the compressor side.

Mr. Jackson had wondered whether Götaverken had had any experience of scavenge pumps working in parallel with the turbocharger. At Götaverken tests had been carried through on an older type of experimental engine (680 mm bore) and results from these tests had been reported previously*. It was the so-called constant-pressure—parallel system which was thoroughly tested, aiming at a free-running turbine without external additional energy. By a suitable combination of scavenge pumps swept volume, compression ratio, turbine area and diffuser this was reached and results were good. However, the system required a complicated arrangement of bypass and shunt pipes and different valves in order to make the blower work within the steady range. This was not convenient for practical service and the above tests were, therefore, not carried on.

Tests with a parallel scavenge pump had, furthermore, been made on a production engine to give supplementary air in a special case where the blower of the unit worked at its upper capacity limit. The best result was, however, in that case, obtained by reducing the series scavenge pump volume (i.e. entirely without parallel pumps), thus transferring the blower service point to a better efficiency range and compensating the volume reduction by increased scavenge air pressure to obtain the stipulated specific air quantity.

Mr. Robertson had given several points of view on water washing of the turbocharger compressor and turbine and had also discussed the importance of keeping the air cooler clean.

The authors could agree with his statement, that water washing was of little use unless carried out from an originally completely clean condition, as far as the compressor was concerned, while the turbine—in their experience—could be cleaned efficiently even if fouling had gone on for a longer time, provided there was adequate equipment and washing was carried out correctly.

The time interval for water washing of the compressor and the turbine was entirely dependent on the fuel oil quality and service conditions, and should be chosen accordingly. As an indicative value, Götaverken stated 250 hours for both compressor and turbine, but in certain cases shorter intervals might be more convenient. Furthermore, as far as the compressor was concerned, washing was done without loss of time for the ship, as it was to be carried out at full engine load. During turbine washing the engine load was to be so reduced that the turbine speed would conform to the instructions from the turbocharger manufacturer. The authors also referred to the reply given to Mr. Taylor, on page 202.

They fully agreed with Mr. Robertson regarding the importance of keeping the air cooler clean and the need of practical equipment to make this possible. For this purpose all Götaverken engines were now provided with lifting bars over all cooler elements to enable the engine staff to dismount, clean, or replace these easily. The new Götaverken engine (Fig. 1) gave, due to the position of the exhaust manifold over the engine top, more space for this work than the older 760/1500 engine referred to by Mr. Robertson. Furthermore, a washing tank for cooler elements was included in the standard equipment.

Regarding Mr. Robertson's requirements to be able to clean the air cooler in situ the authors stated that, for some years, this had been possible for the water side, through the detachable end covers on both sides of the coolers. For the air side, a stationary washing arrangement was under development and the first installation would go into service in autumn 1969.

Mr. Robertson had also touched upon time intervals for cooler washing and instructions for this.

The instruction book stated 2000–4000 hours as an indicative value but this must be considered in each particular case. The fouling of the air side mainly depended on the purity of the engine room air and was judged by observing the pressure difference over the cooler. The maximum value for the pressure difference was given in the instruction book. The fouling of the water side depended on the sea-water temperature, time in ports etc., and was best judged by calculating the air cooler temperature efficiency.

Regarding Dr. Ørbeck's question on the method used for the calculation of piston stresses, a number of such methods had been tried by the authors' company from time to time during past years. The trustworthiness of the results was, however, not too convincing. The great difficulty of finding a method which could give reliable results, without being too comprehensive or too sophisticated, was the main reason for the stress measurements in an experimental piston some years ago. The measurements were briefly reported in the paper (pages 190 and 191). The stresses in Fig. 8 were measured at the transition radius on the inner crown surface which, at that time, was believed to be the position of maximum stresses.

Later advanced computer calculation methods had shown that the stresses in the upper surface facing the combustion chamber, were somewhat higher, but these stresses, for very obvious reasons, were impossible to measure. The computer-calculated thermal stresses were in good accordance with the stresses reached by measurements.

An illustrative stress pattern for a piston crown of approximately the older Götaverken type was shown by Professor Sarsten in Fig. 21 and in his contribution the computer calculation method, based on the finite element techniques, was briefly described. The computer programme developed by Professor Sarsten and his collaborators at the Technical University of Norway had also been utilized for the most up-to-date cylinder liner design of the largebore engine 850/1700 VGS-U. The calculation revealed that the intended reduction of the combined stresses had been reached. The design was shown in Fig. 7.

Mr. Robertson was of the opinion that a reduction of the crown thickness from 80 to 65 mm gave "less piston to burn away" which was really a pessimistic point of view. The authors wished to emphasize that the dimensioning of pistons was, to a great extent, perhaps the greatest, a matter of temperatures. If incorrect fuel injectors were the only cause of burnt piston crowns, then Mr. Robertson's approach would be quite correct. However, the main cause for damage from burning seemed to be high surface temperatures. Measurements on two experimental engines and investigations on experimental units, as well as on production units, had shown that damage from burning could be expected when the surface temperature reached, or exceeded, 580°–600°C (1076°–1112°F), Tendency to slighter damage had even been found at 530°–560°C (986°–1040°F), but the development of this damage had ceased spontaneously after a time, without any preventive measures being taken.

What happens actually at these temperatures? Catalytic oxidation caused by the sodium and vanadium contents of the fuel should be mentioned in the first place, being a process commonly referred to. The authors also believed that, alternatively or simultaneously, the yield point of the material was exceeded during operation from time to time, thus giving rise to small, initially microscopic, cracks in the surface when the piston cooled down later, during service interruptions. This theory was confirmed by the fact that the shape of the burnt areas, in many cases, had a striking resemblance to the curves of constant stresses according to von Mise or, in other words, to the areas of critical yielding, given by Professor Sarsten in Fig. 21. Mr. Robertson had asked whether the authors' company had tried piston crowns coated with some protective surface material. The answer was that a welded-on chromium-steel layer was commonly used for the large-bore pistons with 80 mm thickness. In 1966, an automatic arc welding plant was installed, exclusively for performing this process on new pistons, as well as for reconditioning used pistons. The layer contained about 10 per cent chromium and 0·06 per cent carbon. This protective layer, giving excellent results for the pistons mentioned, had, however, not been utilized for the new large-bore piston with 65 mm thickness, the reason for this being simply the fact that the surface temperatures had been decreased to a level where no burning was expected.

* Collin, L. Th., 1962. "The Constant Pressure-charged Two Stroke Engine, its Present State and Development Possibilities". CIMAC, Copenhagen, paper A4.

Concerning Mr. Robertson's question about the one-piece exhaust valve, this design was developed after mishaps with valves of the older type consisting of a mild steel spindle and a valve disc of heat resisting steel. The one-piece valve was roughly 40–50 per cent more expensive, this fact being, however, well out-weighed by trouble free operation. In addition, the shrinking procedure for the built-up valve type was rather complicated, requiring an inconveniently high degree of accuracy.

On Mr. Jackson's comments on the camshaft drive, the authors underlined the fact that the Holset damper system used for the exhaust valve camshaft had so far given completely satisfactory operation. The first installation had to date attained about 9000 hours of operation. After all, it worked as a damper which meant that it attacked the very cause of the transmission problems originally experienced. The damper, by no means, worked as a flywheel, the built-in mass being freely carried in a bearing and surrounded by a thin, oil-filled space. The internal friction of this oil gave the damping action. As to Mr. Jackson's question on the harmonic balancer and its chain drive, reference was made to the previous answer to Mr. Crowdy, on page 204.

Mr. Bennett had made several questions relating to the section on "Silencing" and regarding his first question on the sound peak, the authors wished to underline the fact that this peak only occurred when the blower was not provided with a damper. When there was an absorbing silencer, the noise level fell evenly towards higher frequencies without any marked peak within the blower's own frequency range. This was due to the silencer being dimensioned mainly for that frequency. The total noise level measured at the control stand was, without silencer 108 dB(A) and with silencer 92 dB(A). The authors also referred to the earlier reply to Mr. Couchman on page 204. Mr. Bennett's question on the 90 degree bend in the silencer, its silencing effect and possibilities of replacing such a silencer by a straight one installed in the diffuser pipe between blower and cooler could be answered as follows: the 90 degree bend provided a very large part of the silencing, the actual part having, however, not been established at Götaverken, for which reason the authors were not in a position to give any more exact particulars. A straight silencer, of the type referred to by Mr. Bennett, had been designed and tested by a turbocharger manufacturer, but to date it had not proved capable of meeting the demands on an efficient silencer.