THE NEW HIGH EFFICIENCY 1.5 MW ENGINE OF JENBACHER AG

HIGH EFFICIENCY CONCEPT - HEC

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Jenbacher AG

ABSTRACT

Jenbacher AG has more than 40 years of experience in the design and development of gas engines. In the mid-1980's the development of the LEA.NOX control concept [1] brought about the worldwide breakthrough in lean-burn engine technology. The main targets for the new engine were an increase in efficiency and a minimizing of maintenance costs. The success of the HEC gen-set represents a considerable step forwards in upgrading the efficiency of a gas engine to 44% - a level equal to that of a modern diesel engine. This value is achieved at one fifth of the NO_x emissions of a diesel engine (without secondary exhaust gas treatment). Furthermore, the specific power density of the new engine - 24.6 kW/dm³ - is certainly comparable to modern diesel engines. Newly developed control and monitoring strategies, as well as long-distance data transmission, allow preventive maintenance measures and an additional improvement of reliability.

INTRODUCTION

The engine represents a new development that was able to utilize the many years of experience with Series 2/3 again and again with regard to improvement of components. The development of combustion resulted on the basis of direct ignition usual for bore diameters under 170 mm. The important components were dimensioned for the full load of 60,000 operating hours typical for stationary engines. At that time a main overhoul is necessary. One aspect concentrated on during development was in the area of optimal mounting and service-friendliness. The development of the combustion was carried out on a special research engine, based on the long experience of the actual engine Series 2/3. After an intensive development phase, beginning with the end of the year 2000 the first pilot engines were delivered to customers.

OBJECTIVES

Jenbacher gas engines have always distinguished themselves through their high degrees of efficiency and power densities [2]. For that reason, the following primary objectives were fixed:

- an efficiency comparable with modern diesel engines
- a high power density for 50 and 60 Hz applications
- emission limits set by 1/2 TA-Luft
- adaptability of the engine to various gases
- reduction of maintenance costs
- utilization of state-of-the-art control and monitoring concepts

CONCEPT

The constructional features of the new engine correspond to a typical "long-stroke concept". Theoretical preparatory work showed clear advantages in the implementation of this approach both with regard to thermodynamics and component dimensioning. The maximum ignition pressure specified for the dimensioning of the engine was 19.0 MPa. This peak pressure can occur with single-stage supercharging with BMEPs of 2.6 MPa and very lean combustion. Moreover, proven constructional features were taken over from Series 3, e.g. the uncooled exhaust port, mixture formation upstream from the compressor and two-stage mixture cooling. The most important parameters of the engine are summarized in Table 1.

Bore/stroke:	145/185 mm
Cylinder displacement:	3.05 l
Number of cylinders:	20
Total displacement:	61.1 l

Cylinder-cylinder distance:	230 mm
V angle:	70°
BMEP:	1.8 – 2.6 MPa
Supercharging:	ABB TPS 57
Charge cooling:	two step intercooler

Table 1: Technical data of the engine

DESIGN FEATURES OF THE ENGINE

CRANKCASE

Figure 1 shows the cross-section of the engine. The 70° V angle was taken over from the existing series. The crankcase is designed to be very rigid and has a flange bordering on the oil pan. The engine is supported on this flange by means of steel/rubber elements on the gen-set frame. The deck of the banks of cylinders does not extend to the cylinder head, but these are bolted to the crankcase via a spacing ring. The advantage of this design is minimal liner deformations and a contribution additionally to less total weight. A cooling water passage has been cast on both sides of the crankcase, the amount of cooling water being dosed through a bore between the lower and upper deck of the cylinder head. All cylinders therefore have the same cooling conditions and no temperature gradients occur from the front to the rear cylinders of the 20-cylinder engine. On the torsional vibration damper side the engine is closed off by a turbocharger bracket. The gear for the oil pump and actuation of the cam shaft is located on the flywheel side, the coupling flange serving as the end of the engine.

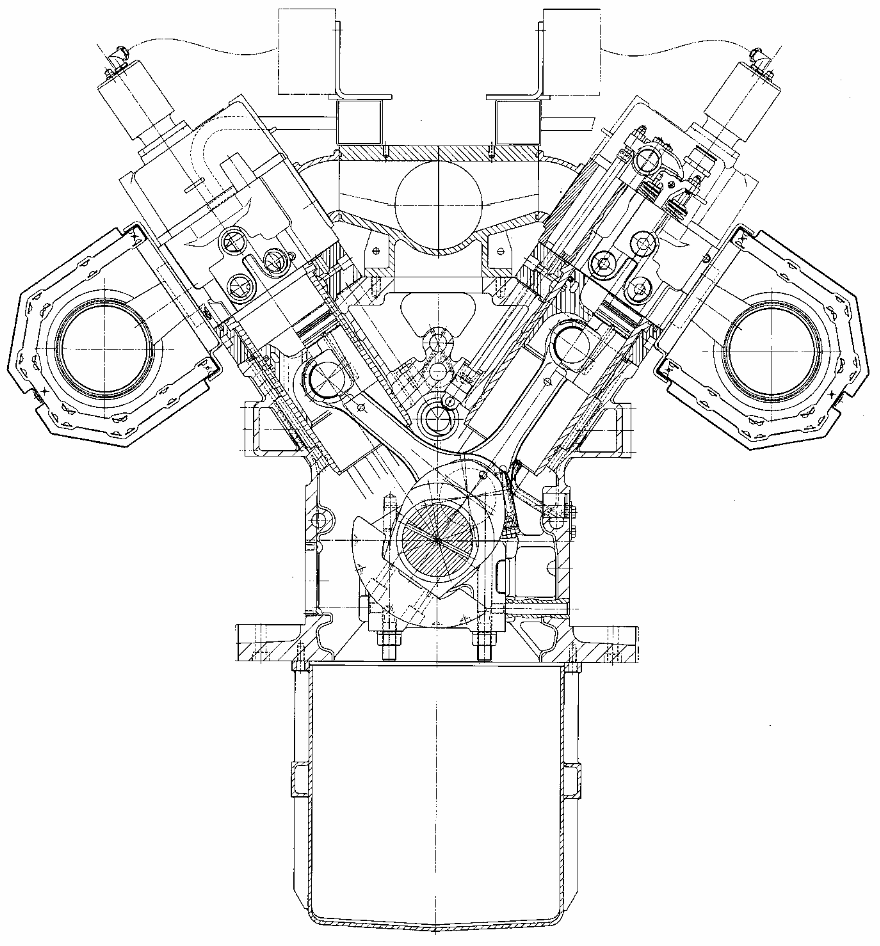

Figure 1: Cross-section of the engine

CYLINDER HEAD

The engine has 4 valve cylinder heads with equally large intake and exhaust valves to optimize the gas exchange. A "cross flow type" was chosen as the flow concept, where the exhaust gas side is located on the outside of the engine. Figure 2 shows the view of the cylinder head from the combustion chamber side. The spark plug is located centrally in a very well cooled sleeve. A maximum ignition pressure of 19.0 MPa was selected for the dimensioning. This allows mean pressures up to 2.6 MPa. The cylinder heads are connected to the crankcase with 4 stud bolts, the bolt forces being effected through the controlled tightening method. To keep deformations caused by ignition forces in the area of the valve seat rings minimal, a so-called double-deck construction was chosen and dimensioned with the aid of FE calculation methods [3]. The upper water jacket is separated from the lower water jacket by means of a conical partition, the passage of cooling water being effected by bore holes in the wall between the valves.

Figure 2: Cylinder head

INTAKE AND EXHAUST PORTS

Based on the existing Series 3, one of the development objectives was to optimize gas exchange. The existing 2-valve head showed increasing losses especially with the high mass flow rates of the lean-burn engine concept from BMEPs of 1.7 MPa onwards. From the perspective of the development of combustion, if the knock resistance (methane number) of the fuel is sufficient, BMEPs up to 2.6 MPa are possible. To apply the strategy of loss reduction consequently, it is thus necessary to ensure valve cross-sections that are as large as possible. Alongside this requirement, it is also necessary to produce the desired flow conditions in the combustion chamber through the intake ports. On the basis of experience with Series 3, development was oriented towards ports producing a swirling flow. As a potential solution, one selected the known version of the combination of a tangential port with a spiral port. This concept allows a relatively high degree of swirl with very good flow coefficients (mysigma). A comparison of the improvement potentials of the 2-valve version with the 4-valve version is shown in Figure 3. Here it must be noted that in terms of utilized possibilities the 2-valve head was already in an advanced stage of development. On the intake side, the 4-valve head has a 19% better flow (referring to the bore diameter) with a swirl level that is somewhat more than 20% higher. It was even possible to increase the bore-specific flow coefficient of the exhaust port by 37.5%.

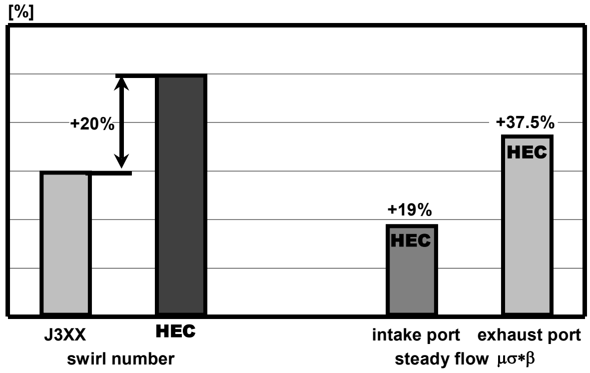

Figure 3: Improvement of the flow coefficients of the 4-valve cylinder head compared with the 2-valve head

VALVE TRAIN

The engine has only one camshaft located centrally in the crankcase that is gear driven on the flywheel side. Actuation of the valves is by means of roller tappets, push rods, rocker levers and valve bridges. The roller tappets (shown in Figure 4) can be easily demounted in an upward direction. The supply of oil to the joints and rocker levers is through two main oil channel in the crankcase. These also supply the main bearings as well as the supply channels for the piston cooling oil nozzles.

Figure 4: Roller tappet

CRANKSHAFT

In comparison to the existing Series 3, besides the higher ignition pressure the demands made on the respective components have increased additionally on account of the 10mm larger piston diameter. Due to the very conservative dimensioning of Series 3 it was possible to retain the same cylinder liner-to-liner distance. To increase reserves, the diameter of the main bearing was increased from 100 to 125 mm. The con-rod bearings have a diameter of 100 mm. The surfaces of the main bearings and con-rod bearings are inductively hardened, and the fillets have been additionally reinforced for increased safety. The structure of the crankshaft was calculated in detail using the FE method and optimized for the safety factors required for stationary engines. In this regard, Figure 5 shows the load curves in the area of the fillets from the con-rod bearing to the crank web. The used material is a heat treated 50 CrMo4 Ni V. Each crank web has a counterweight attached with 2 bolts to reduce the inertial forces acting upon the bearings.

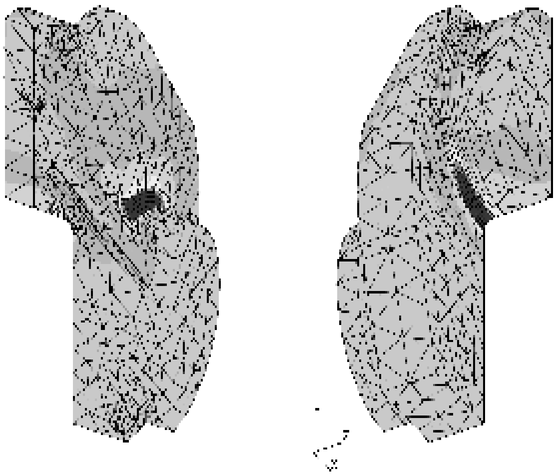

Figure 5: FE results from the calculation of the crankschaft

CON RODS AND BEARING TECHNOLOGY

State-of-the-art technologies are used for the production of the connecting rods. The con-rods are precisely forged and have an optimal weight. Regarding bolted cap and rod, FEM analysis was carried out to optimize structural shapes and the small end redesigned as a stepped variant. This adaptation pays greater attention to the higher ignition pressures. The big end is diagonally splitted (based on the laser grooved method) for maintenance and mounting reasons and is produced for the first time for an engine of this size using the crack technology known from the automotive industry.

Figure 6 shows a con-rod of this type in comparison to the conventional type of con-rod. The advantage of the crack type is the extremely high dimensional stability of the bearing diameters, with the consequence that so-called oversize con-rods can be avoided after engine overhauls and costs saved. The pressure-side bearing shells are, like those of Series 3, produced using the sputter technology and thus offer an optimal reserve of running time.

Figure 6: Con-rods produced by means of the crack technology (below) compared with the conventional technology (above)

MIXTURE AND EXHAUST GAS FLOW COMPONENTS

Since the components used for the flow of mixture and exhaust gas can also be responsible to a great extent for flow losses, they are optimized after the design phase by means of CFD analysis. The large collecting pipe for the gas/air mixture lies between the two banks of cylinders. The mixture is passed from this pipe via adapters directly to the cylinder heads. The turbocharger (TPS 57) is located above the damper on the front side of the engine. It allows pressure ratios up to 4.7 and is accordingly large enough to operate the engine with BMEPs up to 2.6 MPa. The mixture coming from the compressor is then conducted via a diffusor to the 2-stage intercooler, followed by the throttle.

GEARING

The gearing is located on the flywheel side and serves to drive the oil pump and the cam shaft. The cooling water pump is driven electrically and thus the engine can be warmed up prior to startup relatively easily by means of a heater located in the flow of cooling water.

PISTONS AND LINERS

The pistons used are of the mono-block type with a ring groove insert and a cooling gallery. Figure 7 shows a version of a piston used for tests. The following types of piston rings are used: a chrome ceramic-coated top ring, a chromed minute ring, and a D-ring with a coiled spring as a scraper ring. In spite of the BMEP of 2.03 MPa (during the first test phase), a high value for gas engines, the measured surface temperatures are low due to lean-burn combustion and lie at 240°C at the edge of the bowl of the combustion chamber. In comparison to highly loaded diesel engines, this otherwise critical piston area is about 100° C lower and thermal damage is not expected even at maximum BMEPs.

Figure 7: Pistons of the engine

The "wet" cylinder liner is centered in the spacing ring on the outside of the flange. To ensure low oil consumption over long running periods, a "Schabering" (anti-polishing ring) is located on the inside (Figure 8). With this concept, oil consumption can be guaranteed in a range of 0.1 to 0.3 g/kWh with more than 30,000 operating hours.

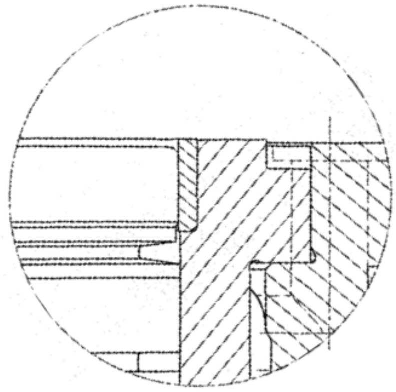

Figure 8: Jenbacher "Schabering" of the engine

THERMODYNAMIC ASPECTS FOR THE "HEC" COMBUSTION DESIGN

The fundamental correlations required to achieve the indicated degrees of efficiency are shown in Figure 9. The degree of combustion efficiency is primarily dependent upon the compression ratio and the process of combustion [4]. Theoretically, the best possible combustion is held to be isochore heat input (constant volume combustion); the lowest degrees of efficiency are given with isobaric heat input (constant pressure combustion). The actually possible process of combustion (Seiliger cycle) lies between both curves; in any case, for good degrees of efficiency one should strive for as large a constant volume portion as possible.

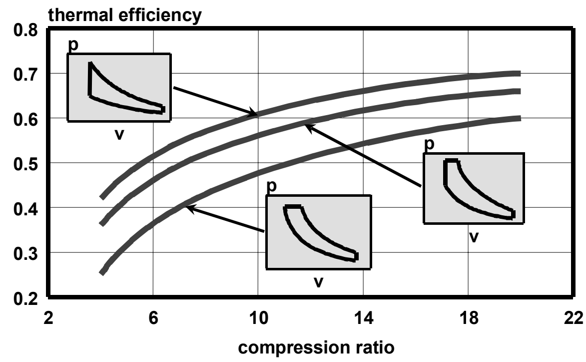

Figure 9: Relationship between compression ratio, type of combustion and degree of efficiency

Regarding gas engines, one must also pay attention to the fuel properties (knock resistance), as these also restrict the possibilities of the combustion process. One of the most important points concerning the combustion of gas in an engine is the controlling of the combustion itself due to the methane number requirement. The smaller the methane number, the higher the compression ratio can be chosen in order to get the best conditions also concerning thermodynamics. Besides the relation of the compression ratio, the design of the combustion presents another decisive influence concerning efficiency. Figure 10 shows the influence of the combustion duration with three different compression ratios compared with the theoretically found indicated degrees of efficiency. Within the range of present combustion times of 50 to 60° crank angle, approx. 47% indicated efficiency is attained with a compression ratio of 1:12. A faster combustion (40° crank angle) with the same compression ratio allows 2% points more.

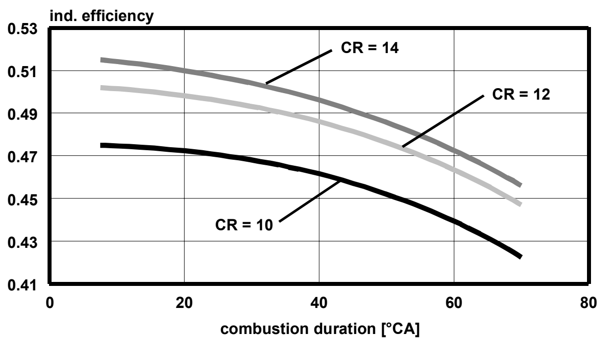

Figure 10: Influence of compression ratio and combustion duration on efficiency

REALIZATION OF HIGH EFFICIENCY COMBUSTION

One of the essential criteria for lean gas engines is as homogenous a mixture as possible. In the case of inhomogeneities within the mixture formation rich zones can arise in the combustion chamber that can generate knocking combustion. Therefore appropriate attention was paid to the mixture formation.

MIXTURE FORMATION

The Tec Jet System is used as the metering valve for the new engine. The principal function is shown in Figure 11. The concept of the Tec Jet System is based on an axial valve with closed loop measurement of the gas mass. The actual process of mixing with the combustion air is carried out in a mixer located in front of the compressor. This concept comes with the advantage that great heating value differences of the gas cause no problems in engine operation. All control interventions can be carried out in a very short time (100ms) and all interventions are done via a CAN bus supervised from the central control unit dia.ne.

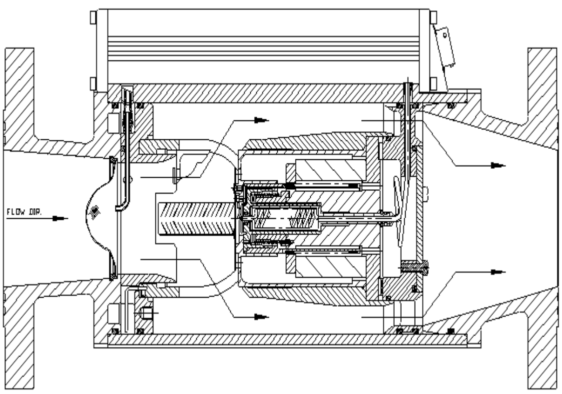

Figure 11: "Tec Jet" gas control valve

COMBUSTION CONCEPT

The combustion concept, which is designated as direct mixture ignition, is based on positive experience gained with Series 3. The stroke/bore ratio of the HEC engine was designed as a thermodynamically advantageous long-stroke engine with a value of 1.275. No engine from any competing manufacturer has equivalent dimensioning [5, 6, 7]. The swirl level was increased about 20% and the production of turbulence in the combustion chamber is achieved additionally through a special form design of the piston and the combustion chamber side of the cylinder head. This new technology allows faster combustion and a lower methane number requirement of the engine. Figure 12 shows a comparison of the Lambda of the two engine types (at the same NO_x emissions). With the same ignition conditions this new combustion concept made it possible to reduce the methane number requirement by 20.

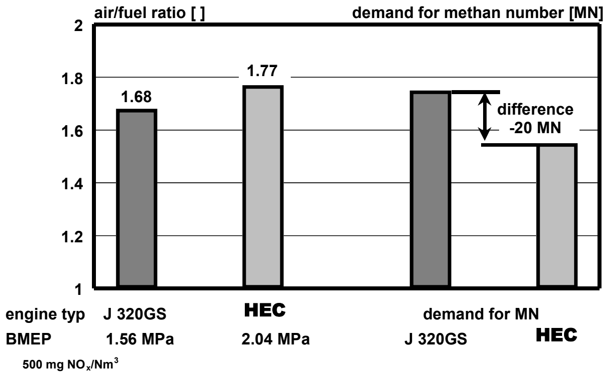

Figure 12: Comparison of Lambda and methane number requirement of Series 3 with the HEC engine

Figure 13 shows the pressure increase in the cylinder of Series 3 compared to the HEC concept. The considerably faster combustion of the load of 1.67 MPa (IMEP) is easily recognizable in the indicator diagram. The level of NO_x was adjusted at the 1/2 TA Luft NO_x standard (250 mgNO_x/Nm³, corresponds to 90 ppm).

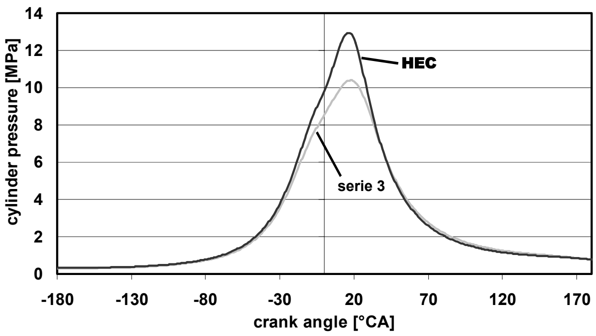

Figure 13: Comparison of the pressure curve at a IMEP of 1.67 MPa Series 3/HEC

With the help of the variation coefficient (AVL method) the level of the combustion development of the HEC combustion concept is very good comparable with the actual Series 3. Figure 14 shows the situation near the lean limit at an NO_x level of 250 mg/Nm³; furthermore, the variation of the maximum of the cylinder peak pressure is indicated.

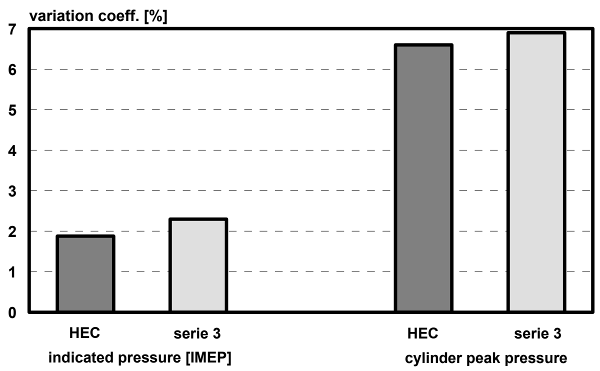

Figure 14: Comparison of the variation coefficient regarding IMEP and peak pressure of Series 3/HEC

As is shown in Figure 12, the Lambda of the HEC engine is 1.77 with 500 mg NO_x. In addition, through fast combustion the exhaust gas temperature after the turbine is reduced to 430°C (Figure 15). This tuning attains the efficiency optimum of the engine. Depending on market requirements, a different tuning can also result in higher exhaust gas temperatures with small losses of efficiency.

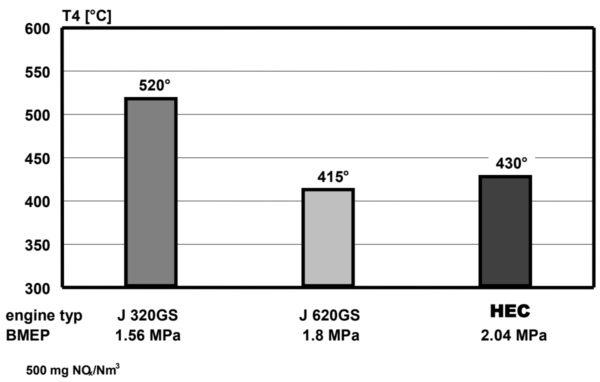

Figure 15: Comparison of exhaust gas temperatures of Series 3 and 6 with the HEC engine

HEAT BALANCE OF THE HEC ENGINE

The HEC engines concerning the pilot plants have a rated power of 1451 kW (BMEP=1.9 MPa). All the work in the R&D department concerning the engine components and combustion was carried out with up to 20% higher loads. An efficiency of 44% is reached with a BMEP of 2.1 MPa. Figure 16 shows the heat balance of the first delivered engines.

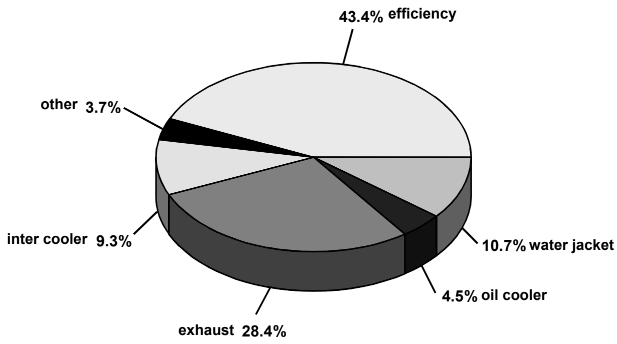

Figure 16: Heat balance of the HEC pilot engine

Through fast combustion and the low temperatures in connection with it there is less thermal stress on components in spite of the higher BMEP. In Figure 17 the combustion chamber bowl edge temperature of the HEC concept is compared with the temperatures dependent on the BMEP and combustion concept. The highest degree of component stress occurs at Lambda = 1 (BMEP = 1.17 MPa turbocharged); lean-mixture combustion is characterized by lower values. Principally, the temperature load on the HEC piston due to the smaller Lambda is, despite the higher BMEP, perhaps equal to or somewhat less compared with the present state of Series 3.

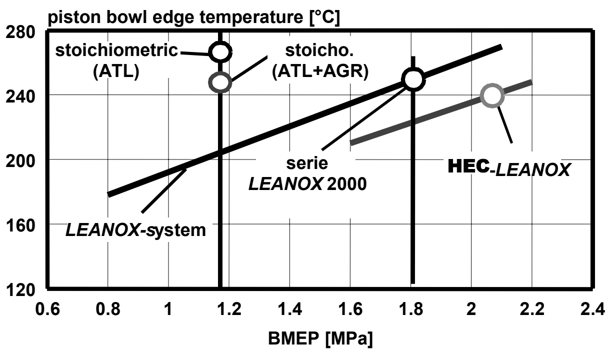

Figure 17: Thermal component stress of various combustion concepts

Regarding the Lambda = 1 concept it can be said that "apparent" potentials can be found again and again that, when considered superficially, can lead to interesting approaches to solutions [8]. However, hard reality first evinces itself in the customary running times of the CHP plants. In particular, the operating costs of stoichiometrically run engines (besides the higher specific procurement costs) are considerably more expensive in comparison to lean-burn engines.

Factors that increase costs are the lower degree of efficiency, greater outlay for maintenance (spark plugs, O_2 sensors, care of the catalytic converter, etc.) and the shorter service life of the affected components due to deposits of incineration of ash. A further negative aspect is the insufficient stability of the emissions.

EFFICIENCY OF THE ENGINE

The degree of efficiency of 44% attained through the HEC concept is a milestone for the technology of stationary gas engines. In this regard Figure 18 shows a comparison of the HEC engine to the competition as well as to several diesel engines [9]. To be able to make a comparison to modern diesel engines, the specific consumption values have been converted into MJ/kWh. What is particularly noteworthy is that the gas engines achieve the indicated degrees of efficiency at about 1/5 the NO_x emissions of the best diesel engines.

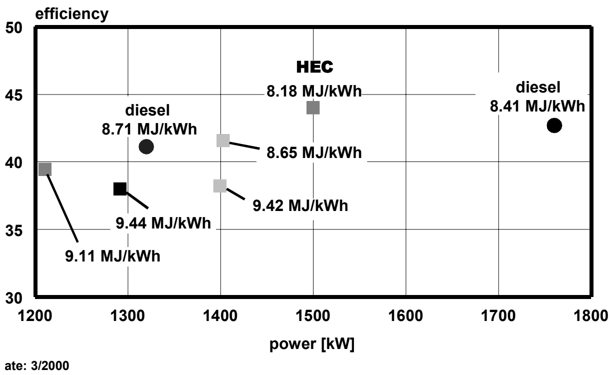

Figure 18: Comparison of the degrees of efficiency (HEC to competition and to modern diesel engines)

THE NEWLY DEVELOPED JW P4 SPARK PLUG

The new BMEPs and the peak pressures resulting from them make very great demands on the ignition system. To be able to fulfill these demands, parallel development of a new type of spark plug was initiated and has been completed in the meantime. This spark plug is designated as JW P4. Besides the improvement of the housing, the amount and the composition of noble metal were modified to produce a version with greater thermal resistance.

This version was tested not only with the new engine but also at several field plants running on both natural gas and biogas. Figure 19 shows the special designed electrodes.

Figure 19: Photo of the JW P4

CONTROL OF THE HEC ENGINE

For open- and closed-loop control of the HEC engine one employed a further development of the Jenbacher "dia.ne" engine management system based on a high-performance PLC (program logic control) system. The control of lean-burn combustion is based on the proven LEA.NOX concept. A particular feature that deserves mention is the automatic ignition voltage control system "monic" (monitoring ignition control) [10]. The concept of "monic" allows an online display of all ignition voltage values and thus a monitoring of the condition of the spark plugs. To do this, one need only press a button on the visualization unit of "dia.ne" (Figure 20).

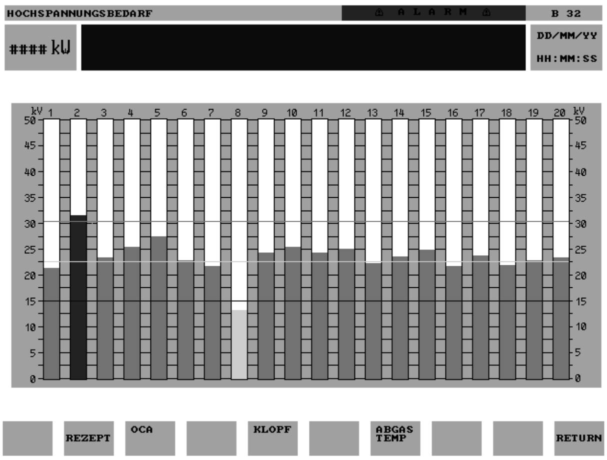

Figure 20: Visualization of actual ignition voltage values on the dia.ne

To be able to maximize the operational reliability of the engine also under difficult conditions, one developed a concept of optical recording and visualization of misfires or other irregular conditions in the cylinder. This concept was termed "oca" (optical combustion analysis).

Each cylinder head has a combustion chamber window to allow optical recording of the light conditions during combustion. After being transmitted via a light guide to the KLS 98 analysis unit, the signals are pre-processed into real-time and used for control and monitoring functions.

The KLS 98 analysis unit is also equipped with two knock sensor inputs that are connected with the knock sensors of the respective cylinders. As a result, besides its combustion chamber window, each cylinder has a knock sensor used for cylinder-specific control of ignition energy and knocking. Figure 21 shows the KLS 98 analysis unit with the respective knock sensors.

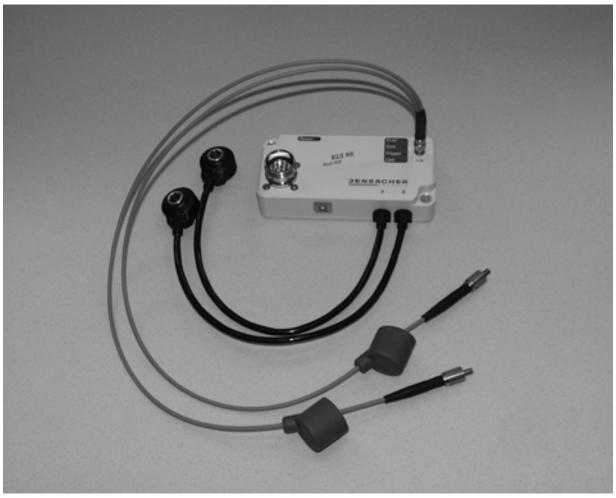

Figure 21: KLS 98 with the respective knock sensors

All information recorded about misfires and knocking are displayed in the customary way on the screen of the "dia.ne". The components mentioned above are already available for Series 3. The simplication of service and maintenance tasks is rounded off by "hermes", the Jenbacher long-distance data transmission concept. External sites have access to all information though direct communication with "dia.ne". "hermes" also allows selected data/events to be routed to a specific service center, resp. to have software updates and adjustment operations carried out from there.

SUMMARY AND FUTURE DEVELOPMENT

The HEC engine represents a concept achieving a degree of efficiency of 44%. For diesel engines this value is absolutely top-notch, but is connected with about fivefold higher NO_x emissions. Utilizing this concept means that in future calorically weak gases can also be burned with high degrees of efficiency.
This engine will be available in the 2002 product program. Before being released for full serial production it will be tested in constant operation at several pilot plants and adapted for various low heating value gases.

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