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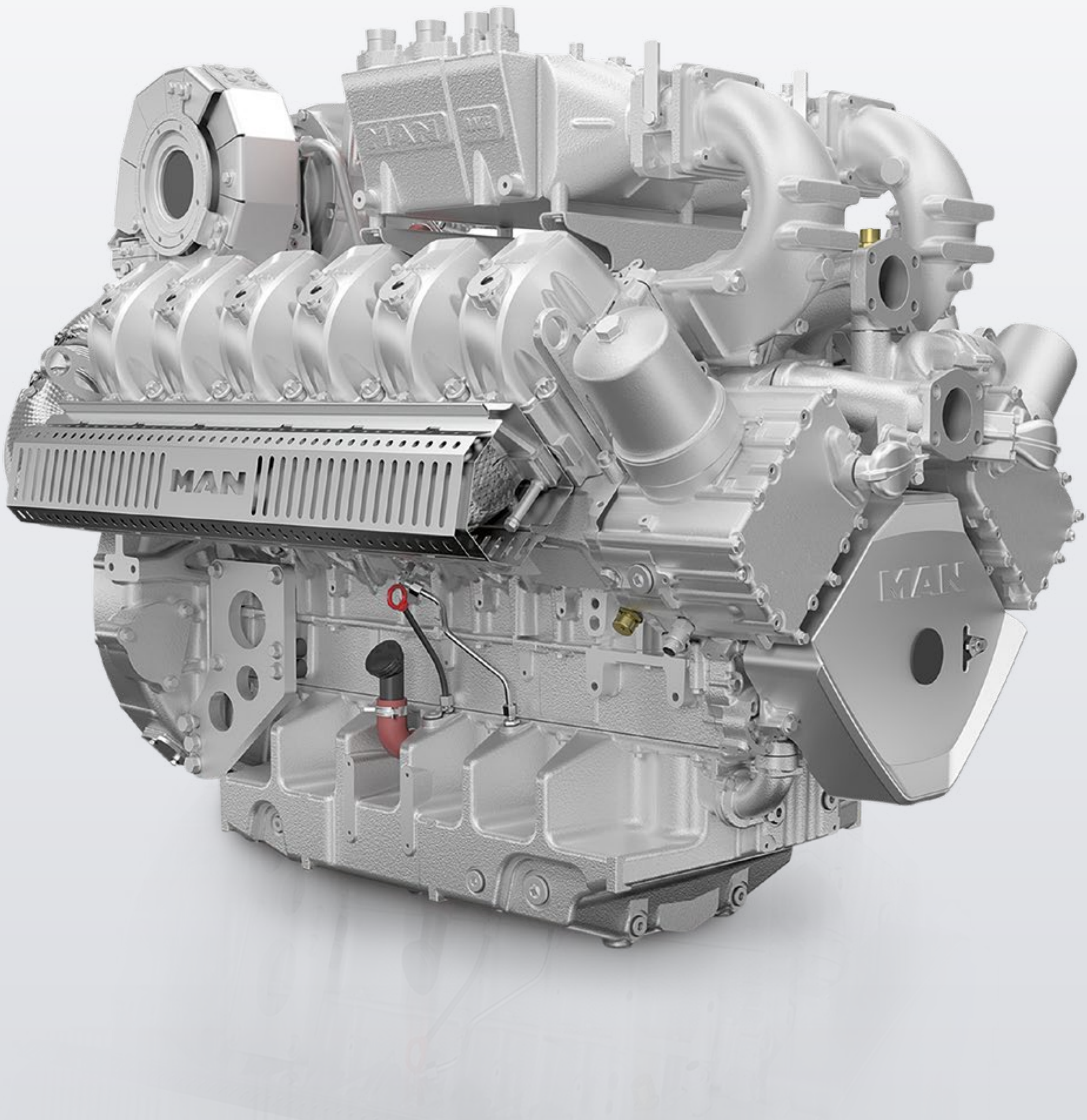
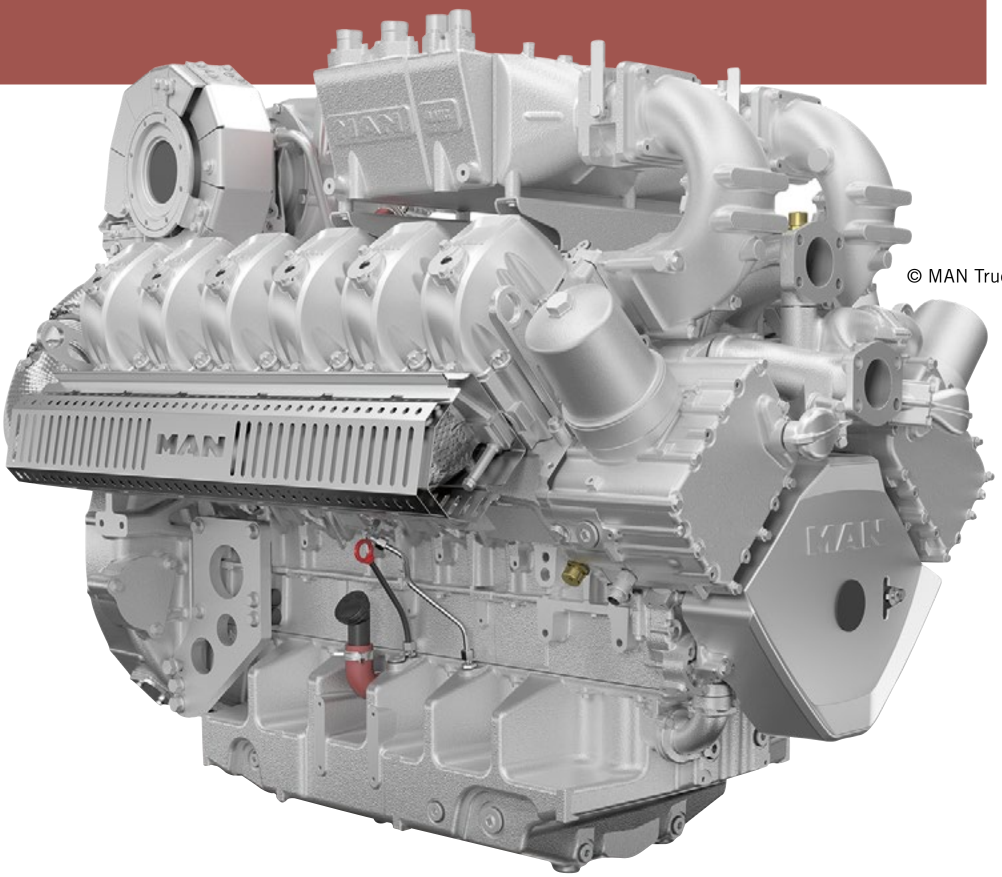


MAN E38

GAS ENGINE with High Medium Pressure and Efficiency

The New Twelve-cylinder MAN E38 Gas Power Engine



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The development goals for the E38 V12 gas power engine series from MAN were market-leading characteristics in terms of mechanical efficiency and mean effective pressure combined with a stable low-NO_x capability. In addition to fundamental changes in the turbo charging system, Atkinson camshaft timings proved to be the best solution, fulfilling all the restrictions and premises specified in this development project. With the help of these measures, a high effective efficiency of 44 % and a brake mean effective pressure of 20 bar could be achieved.

MAN has introduced a new twelve-cylinder stationary gas engine, the E38 V12. Compared to its predecessor, the MAN E32, the single turbocharger concept is probably the most prominent innovation of the new unit [1]. As a result, the design of the exhaust system had to follow the so-called 2-in-1 principle. While the exhaust pipes could nearly stay the same – despite of a slight increase of the gas flow cross-section due to higher power output – a new Y-shaped elbow was modeled, directly mounted to the turbocharger but connected and mechanically as well as thermally decoupled from the exhaust system by two expansion joints. Unlike in the case of the E32 gas engine, the turbocharger with a weight of about 100 kg is not borne by the exhaust gas system itself, but is placed on a solid supporting structure integrated into the flywheel housing. To reduce assembly complexity and even more important the risk of oil-leakage in a hot temperature zone to a minimum, the turbocharger oil supply is not implemented by an external pipe system (oil lines plus fittings). There are rather two oil bores in above mentioned supporting structure connecting the crankcase oil gallery with the in- and outlet ports of the turbocharger bearing housing.

To the circumstances of a next generation oil-module and the aim to leave the concept of an integrated engine breather system, the same needed to be redesigned. It now consists of a pre-separator placed in the engine-V-room between the two cylinder benches and a main separator (two times fleece trap), that is directly integrated in the intake manifold mounted on the turbocharger inlet and thus maintaining the easy access in case of servicing.

With respect to the cylinder head the premise was that a simple rocker arm assembly would still be used and the tappet passage in the crankcase would remain unchanged. The required transmission ratio and the already defined coolant supply further restricted the design options for the rocker arm assembly. Under these restrictions, the final design was developed in several iteration loops, resulting in a 20° rotation of the valve star and the introduction of a second intake port to further reduce pressure losses in the cylinder head, **FIGURE 1**. An innovative and patented bottom-up cooling system allows an effective cooling of the highly stressed coolant channel webs.

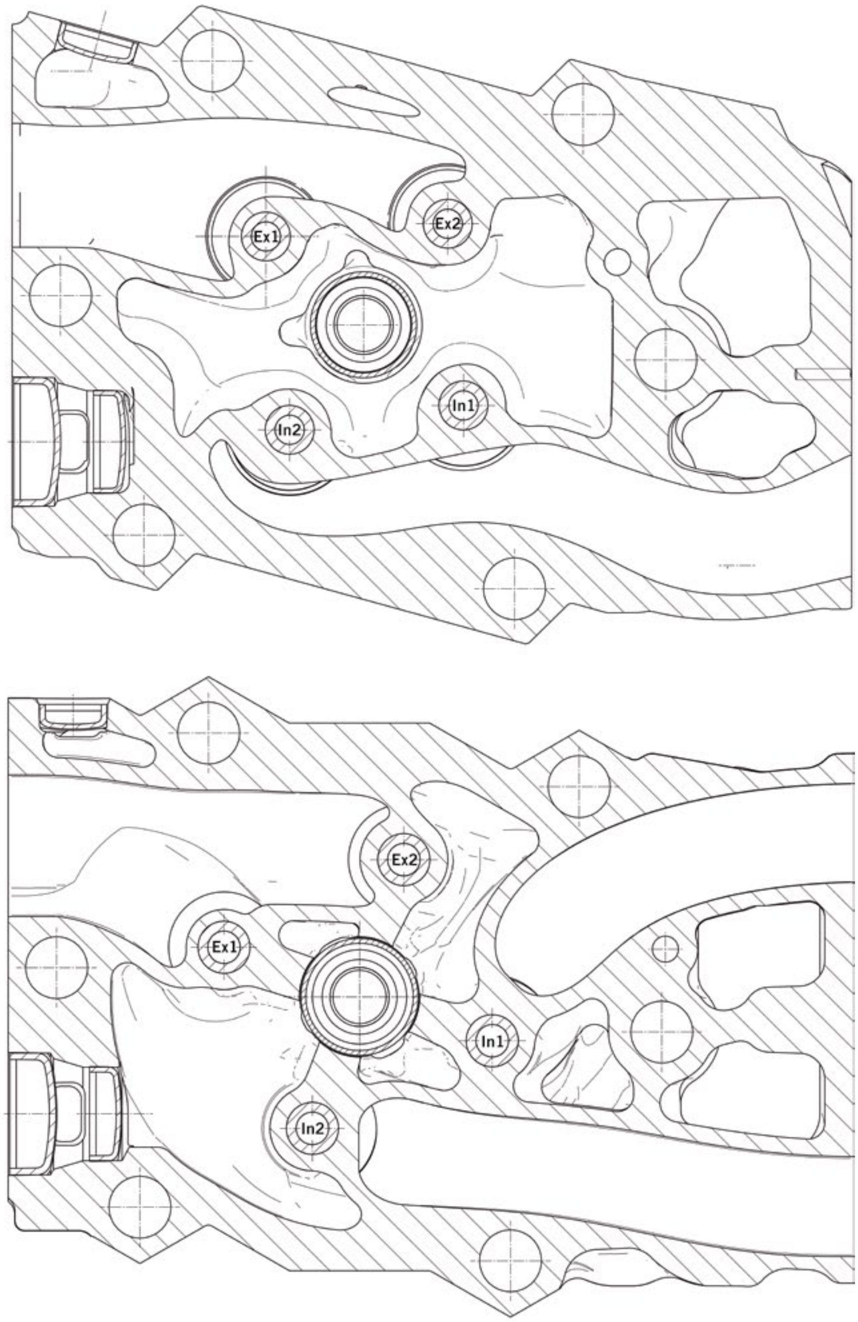

FIGURE 1 Sectional comparison of the E32 (top) and E38 (bottom) cylinder head and position of the intake (In1, In2) and exhaust valves (Ex1, Ex2) (© MAN Truck & Bus)

CHARGE CYCLE AND COMBUSTION ANALYSIS IN THE SIMULATION

In order to achieve the engine efficiency target of 44 % for this new development, detailed simulation studies were carried out in advance. By means of extensive simulation variations, it became clear that fundamental changes in comparison to the predecessor engine will be necessary. In detail, the following criteria were evaluated:

- increase of the compression ratio
- compact/more fast combustion process
- minimization of the heat transfer during expansion
- improvement/reduction of the charge cycle work.

At first glance, these requirements are partially contradictory. Compact combustion can for example be achieved with a high swirl, but this worsens the heat transfer in the expansion and also the charge cycle work. Thus, a way had to be found to improve all of the above factors. In the forementioned example this was achieved by a fairly low base swirl in combination with a piston having a high squish factor. As a result, the turbulence at combustion Top Dead

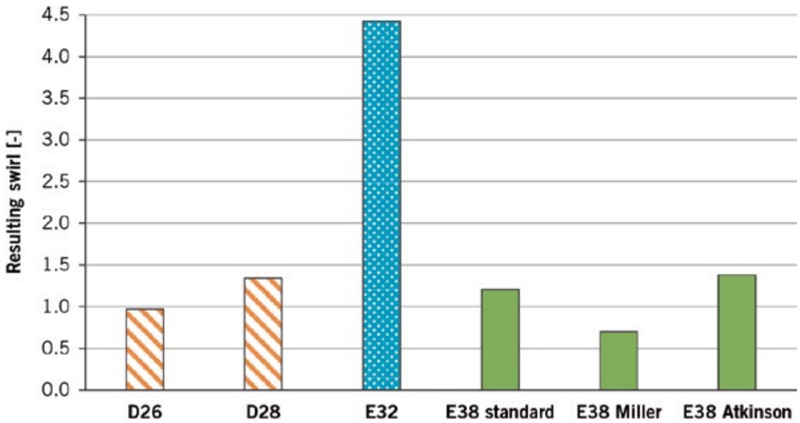

FIGURE 2 Swirl of various MAN engines (D: diesel; E: gas) versus the E38 with different valve timings (© MAN Truck & Bus)

Center (TDC) is high, but significantly lower during expansion.

COMMON PORT DESIGN FOR THE DIESEL AND THE GAS APPLICATION

Utilizing 3-D Computational Fluid Dynamics (CFD) methods a steady state measurement of the gas exchange ports on the component test bench was simulated and the swirl number was determined as a function of the valve lift.

For cost reasons, it was a goal from the beginning to use an identical cylinder head cast part for both, the new gas engine (E38) and the new diesel engine (D38), belonging to the same modular system. The challenge here was that modern diesel combustion layouts use a very low level of swirl to minimize heat transfer, shown for the D26 and D28 diesel engines in **FIGURE 2** as an example. However, Otto engines (engines with flame front combustion, respectively) require a higher degree of charge motion and turbulence to enable a compact rate of heat release, at least if not equipped with a scavenged (active) pre-chamber. The E32 gas power engine serves as an example for a high swirl setup.

The conflict of objectives was resolved by paying attention to the swirl number in relation to the valve lift during port design. Due to the fact that swirl is generated using two tangential ports, it increases more along with the valve lift than this would be the case with a spiral port design. Due to the almost linear increase in swirl over a wide range, it was possible to influence the swirl level of the cylinder charge motion via different maximum valve lifts and valve timings. The resulting swirl number for different valve lift profiles can be estimated in a simplified way by weighting the swirl at support points with known valve lift with the piston speed and integrating it via the crank angle. Even with this simplified method, it can be seen clearly that the swirl for the diesel application can be significantly reduced by means of a Miller profile. A slight increase in the mean swirl can be seen for the E38 Atkinson version with Late Intake Valve Closing (LIVC) compared with the E38 standard (filling optimized) timings. However, the resulting swirl for the Atkinson profile is underestimated in this calculation, since valve lifts at crank angles above 540° are not taken into account for the integration.

Due to the lower level of swirl compared to the predecessor E32 gas engine and the introduction of a second intake port, it was also possible to achieve a lower pressure drop, resulting in a >40 % higher flow coefficient.

SYSTEM DESIGN

The choice of the turbo chargers compressor is largely dependent on the engine's volumetric efficiency. In order to improve the charge cycle work, the Intake Valve Closing (IVC) time in particular was adjusted. In simple terms, a certain air mass flow is required for a given power at a desired lambda (air/fuel ratio). If the volumetric efficiency is subsequently lowered, the boost pressure requirement increases. If this can be achieved with a reasonable

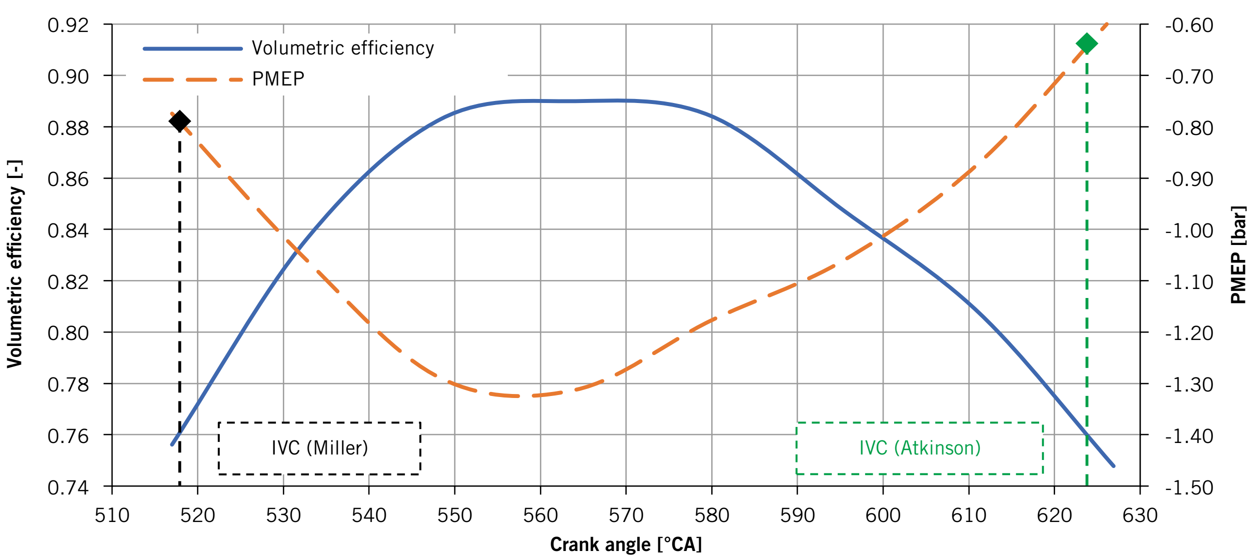

FIGURE 3 Pumping mean effective pressure versus volumetric efficiency for different IVC times (© MAN Truck & Bus)

small increase in exhaust backpressure (effective flow area and efficiency of the turbine) and the precondition that the compressor efficiency remains constant/high, the charge cycle work will improve.

However, there are limits to this process. The lower the volumetric efficiency, the higher the boost pressure that needs to be reached and that components like the gas mixture cooler need to withstand. The compressor outlet temperature also rises significantly, especially in case of single stage turbo charging. These two variables subsequently determine the minimum volumetric efficiency possible at a desired power output.

An over expansion cycle, like the Miller or Atkinson cycle, can be used to reduce the volumetric efficiency. As described above, one objective is to achieve a compact combustion process. For this purpose, the swirl should be maximized for the specified gas/diesel cylinder head respectively the common intake ports. As described for the port design, the higher the valve lift, the more swirl is generated. This means that with the Miller cycle significantly less swirl can be generated than with the Atkinson cycle due to its shorter valve opening time.

A 1-D CFD study with respect to the IVC times has been carried out to visualize the influence on the charge cycle work. The results are shown in **FIGURE 3**. If one compares the Miller timing (black, dashed) and the Atkinson timing (green, dashed) at a volumetric efficiency of 0.76 each, it can be observed that the Pumping Mean Effective Pressure (PMEP) is less negative for the Atkinson timings which increases the engines efficiency.

A further advantage of the over expansion cycles is the reduced effective compression ratio and by that a lower cylinder compression end temperature, which helps to mitigate abnormal combustion phenomena like engine knocking or pre-ignition.

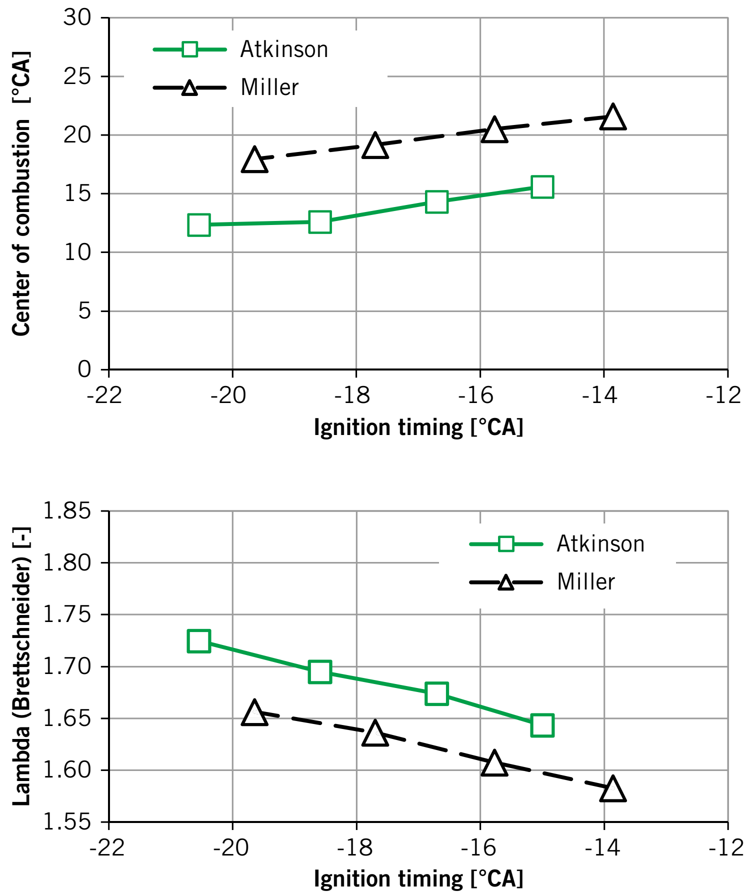

FIGURE 4 CoC and lambda for the SCE with different valve timings and constant NO_x emissions (© MAN Truck & Bus)

MEASUREMENT CAMPAIGN ON THE SINGLE- AND MULTI-CYLINDER ENGINE

To validate the studies of the CFD simulation, various test bench runs with a Single-Cylinder Engine (SCE) have been carried out. Especially the results with respect to the valve timings and the compression ratio were of interest. Besides the piston-cylinder-unit also the boundary conditions like charge pressure/temperature and exhaust backpressure were set according to the charge cycle simulation results to properly approximate multi-cylinder conditions. In the following a selection of the SCE results is discussed briefly.

Fulfilling the same NO_x emission limit, it can be observed in **FIGURE 4** that with Miller timings it was not feasible to reach the same Center of Combustion (CoC) as with Atkinson timings. The higher charge motion and turbulence with Atkinson timings therefore overcompensates for the leaner air/fuel ratio, which slows down the combustion process. Even if one compares the most early ignition timing with Miller and the latest with Atkinson timings, bringing both to a comparable lambda level, the CoC with Miller timings is still delayed about 3 °CA.

With respect to the Coefficient Of Variation of the Indicated Mean Effective Pressure (COV_{IMEP}) up to 0.8 %-points lower values have been observed using the Atkinson timings despite running on a higher air/fuel ratio which underlines the favorable charge motion and ignition conditions.

Eventually the SCE results already narrowed down the hardware for the Multi-Cylinder Engine (MCE) tests to one single compression ratio and one piston contour. Atkinson valve timings performed superb thus defining the final setup for the first prototype of the MCE.

For the MCE tests pressure indication on all twelve cylinders, various ignition systems and engine controls were at disposal. The gas consumption measurement was carried out redundant, using two different measurement systems, to guarantee a high reliability of the calculated engine efficiency.

The MCE tests verified the performance with the pre-selected components from SCE thus setting a solid starting point to adapt and optimize the passive pre-chamber spark plugs that have

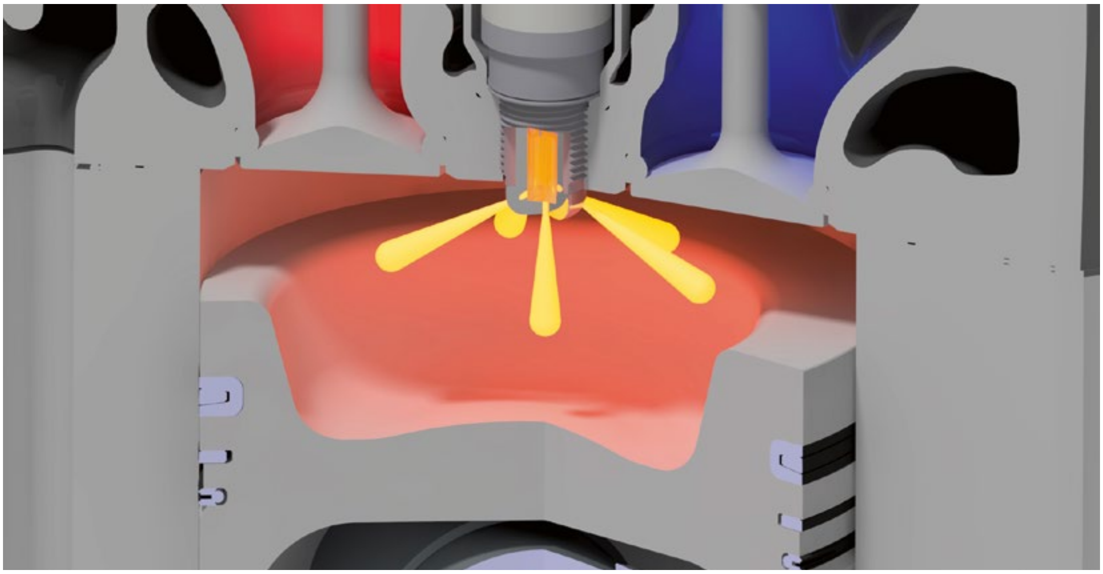

FIGURE 5 Pre-chamber spark plug with flame jets (© Multitorch)

already been in use for the E32 predecessor engine, **FIGURE 5**. After several optimization runs an increase in engine efficiency of >1 %-point and a decrease of the COV_{IMEP} of about 45 % was reached in comparison to a standard so-called J-Gap spark plug. The minimal NO_x emissions, the engine could run with a defined COV_{IMEP}, could be halved.

CONCLUSION AND PERSPECTIVE

The E38 gas power engine series is not only a E32 series with increased displacement but has undergone fundamental changes in terms of turbocharging, the charge cycle and the combustion process. The knowledge gained over many years of operation of the E32 series has been incorporated and further robustness-enhancing measures have been taken into account for the higher maximum power and medium effective pressure. Moreover, by means of the Atkinson timing, it has been possible to use the same cylinder head cast part for the diesel and gas engines without compromising the gas engines efficiency.

Due to the large number of common parts, key aspects of the diesel engine validation could be carried over to the gas engine, which not only consolidated the gas engine validation process, but also expanded its scope and requirements.

The extensive use of simulation methods and the validation of the results on the single-cylinder engine were key to achieve an effective engine efficiency of 44 % with the multi-cylinder engine. The increased displacement of 29.6 l in combination with a Brake Mean Effective Pressure (BMEP) of 20 bar results in a power output of 735 kW at 1500 rpm and 840 kW at 1800 rpm. The E3872 LE thus represents a major advance in the twelve-cylinder class of MAN gas power engines and provides an excellent foundation for future derivatives, whether for other power classes or alternative fuels such as hydrogen, methanol, or ammonia.

REFERENCE

[1] Wöhner, P. et al.: The new E38 MAN stationary gas engine for power and heat generation. 18th International MTZ Conference Heavy-Duty, On- and Off-Highway Engines, Nuremberg, 2023

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