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INFLUENCE OF VALVE-SEAT ANGLES TO OPERATION VALUES AND EMISSIONS OF MEDIUM-SPEED DIESEL ENGINES

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Abstract: For the development of gas exchange for large diesel engines, a compromise has to be found between efficient valve-flow and the time between overhauls. On the one hand, large effective flow areas, especially during valve-overlap, are demanded. On the other hand, there are limitations of cylinder bore regarding the maximum diameter of inlet and outlet valves and the minimum distance (dead space) between valves and piston, as well as wear-related smaller seat angles. For large medium-speed diesel engines, a valve-seat angle of $\beta = \mathbf{30}°$ for inlet and outlet valves is a standard application. For engine-operation with clean fuels, a valve-seat lubrication (gasoil) or smaller seat angles (natural gas) need to be applied. With this presentation, the basic influence of different valve-seat angles on the operation values and emissions will be considered for the example of the single-cylinder research engine FM16/24. Using a self-developed testbed, experimental investigations into effective flow areas as a function of valve-lift at inlet and outlet valves have to be executed. With this input, different cycle calculations including T/C have to be carried out to determine deviances in specific fuel-oil consumption, exhaust-gas temperatures, NOx emissions and air/fuel ratio. The results will be discussed critically.

Key words: medium-speed engine, gas exchange, valve-seat angle, flow coefficient, thermal load

1. PROJECT MOTIVATION

More than 95 % of the world's merchant ship fleets are powered by diesel engines. Currently about 71 % are burning heavy fuel-oil [1]. The market share of ≈1 % for gas-engines is mainly induced by special boil-off requirements of LNG-carriers. Directly driving 2-stroke engines with speeds up to 104 rev/min dominate the propulsion market for container ships, tankers and bulkers. Cruise liners, ferries, container-feeders and special-purpose vessels are mainly propelled with reduction gears by medium-speed four-stroke engines with engine speeds between 333 rpm and 1,000 rpm. Gensets are mostly medium-speed powered. Intensive competition in the propulsion market led to extreme concentration of manufacturing capacities.

As a result of aggressive competition, the remaining companies increased the specific output of their engines to ensure the resultant cost-attenuation that would enable them to retain themselves in this market. The basic requirement is a sufficient charge-air pressure, limited by the maximum circumferential speed of T/C wheels. Large four-stroke diesel-engines with optimised volumetric efficiency generate with every bar charge-air pressure ≈6 bar mean effective pressure, which are paid by customers, while parts of available charge-air pressure must be used for emission reduction by Miller-timing. Optimised gas exchange is a usable way for breaking up this trade-off between vendable power and limited T/C capacity.

For the development of gas exchange of large diesel engines, a compromise must be found between efficient valve-flow and sufficient time between overhauls of at least 5,000 h for valves. At one side, large effective flow areas, especially during valve-overlap, are demanded for low exhaust-gas temperatures. Otherwise, there are limitations of cylinder bore regarding the maximum diameter of inlet and outlet valves and the minimum distance (dead space) between valves and piston, as well as wear-related smaller seat angles. For large medium-speed diesel engines, a valve-seat angle of $\beta_V = 30°$ (Fig. 1) for inlet and outlet valves is a standard application.

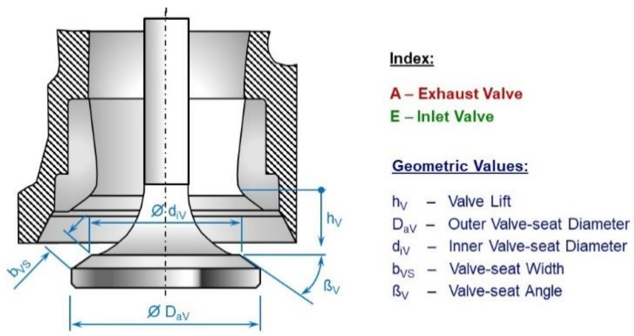

Fig. 1. Flow-relevant geometry of engine-valves

For engine-operation with clean fuels (i.e. all fuels without lubricating effects of combustion deposits, such as distillate fuels or fuel gases), a valve-seat lubrication (gasoil-operated MAN medium-speed) or smaller seat angles (natural-gas fired MaK engines) would be needed.

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Based on our study of the literature, we have gauged that there are no publications pertaining to the influence of valve-seat angles on operational values and emissions. By interchanges with development engineers of MaK and MAN, this result was confirmed. According to the authors' experience from industry, no advantages of CFD calculations versus experimental investigations regarding correctness of results and economic issues were found. With special consideration for financial limits and staff-potentials of a small university, it was decided to carry out only experimental investigations and cycle calculations related to this issue.

2. EXPERIMENTAL INVESTIGATIONS

2.1. Investigations at fluid-dynamic testbed

For experimental investigations of the behaviour of gas exchange valves in cylinder heads, a special testbed was designed and assembled (Fig. 2). The supply of compressed air was realised by a dry screw-compressor (for charging of test engines), calmed in a 400 L volume reservoir and measured by a thermal mass flow meter. As long as possible a constant difference-pressure of 50 mbar over the valves was used for measurements at smaller valve-lifts. For a valve-lift larger than $h_V \approx 6$ mm, the flow-capacity of the screw-compressor was not sufficient to keep this difference-pressure. The compressor was operated at maximum speed, and the resultant difference-pressure was considered for these measurements. At the maximum valve-lift of $h_V = 12$ mm, a pressure difference of ≈20 mbar was obtained. According to the measurement described in the literature [2, 3], this pressure difference used does not exercise any influence on the flow coefficients.

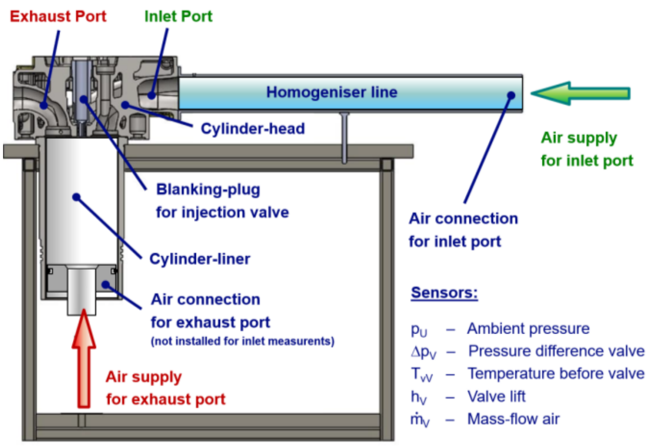

Fig. 2. Testbed for fluid-flow experiments

Every discrete valve-lift was adjusted and measured by an ordinary dial indicator. Pressure difference over valves was detected by a digital manometer. With a pressure calibrator, the ambient pressure was quantified.

This testbed represents valve-flow under steady-state conditions. Conception and design of this testbed for fluid-flow experiments orientated to experiences according to the testbed concepts at MAN Augsburg and MaK Kiel as well as the same model for cycle-calculation based on [4] is used. The results of these measurements had to be comparable to the flow-measurements at serial models to ensure representative results for different families of medium-speed engines. Up to now, no dynamic methods as described in the study of Szpica [5] are used at German manufactures of commercial medium-speed engines.

The jacket of a truncated cone normal to valve-seat was chosen as a reference surface for the geometric flow area (Fig. 3a) [4, 6 and source code of 7 without references to other authors]. Additionally, the significant effective flow area is calculated by multiplication of geometric area and flow coefficient, and the definition of the geometrical value is even. Published in the handbook for mechanical engineering of Beitz and Grote [8] is the alternative use of a cylinder's jacket, erected at the inner valve-seat diameter up to the seat ring. In the study of Heywood [9], a more complicated section-related version with three different cases in the order of valve-lift is published. All three variants are compared in the study of Swiderski [2]. Maximum deviation between the easier description as a truncated cone's jacket with just one equation as used for these investigations [4] and the complex formulas according to Heywood [9] is with 3.7 % at maximum valve lift for the test engine used quite low. A different formula for the same modelling of the geometric flow area as an area of a truncated cone's jacket is given in the study of Tanaka [10, p. 293].

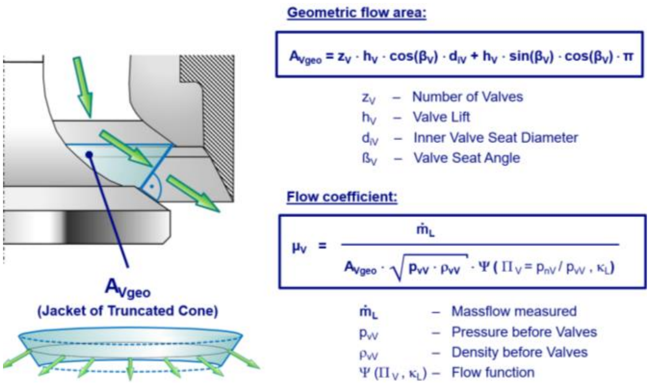

Fig. 3a. Calculation of flow coefficients

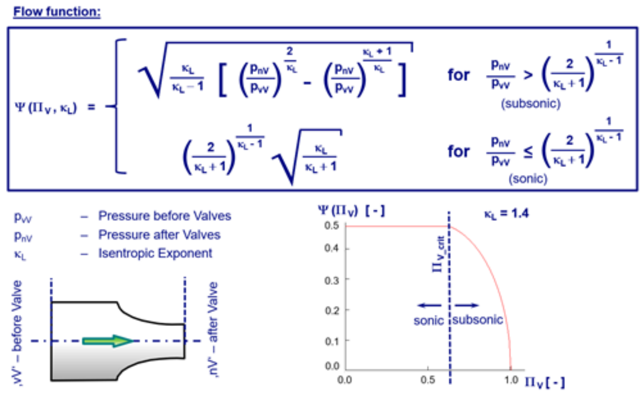

Fig. 3b. Definition of Flow-Function

Pressures before and after valves were calculated from ambient pressure and pressure difference according to the position of pressure sensors and flow direction at valves. Properties of air were calculated with formulas according to Zellbeck [4]. Definition of flow-function (Fig. 3b) was used according to Zellbeck [4] and Urlaub [11] with the fluid-dynamic model of a not-expanded orifice.

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A different formulation for flow-function is known from the Japanese manufacturer Niigata.

To ensure realistic values in engine field operation for all related measurements, a used cylinder head, grabbed out after ≈15,000 h at the main engines of SY 'Sea Cloud' (Ex 'Hussar'), was used instead of a new one. Reference measurements for inlet and exhaust were carried out several times to quantify their repeatability.

Diagrammed in Fig. 4 are the results of three measurements for inlet and two for outlet port versus valve-lift up to maximum value in the serial engine. Flow coefficients for zero lift are extrapolated with measured ones for 1 mm and 2 mm. In spite of the difficulties in correct adjusting small lifts at both valves and the resultant relative error in geometric flow area even at small valve lifts the repeatability is with a tolerance of max. 4.5 % at 1 mm lift of inlet valve quite well.

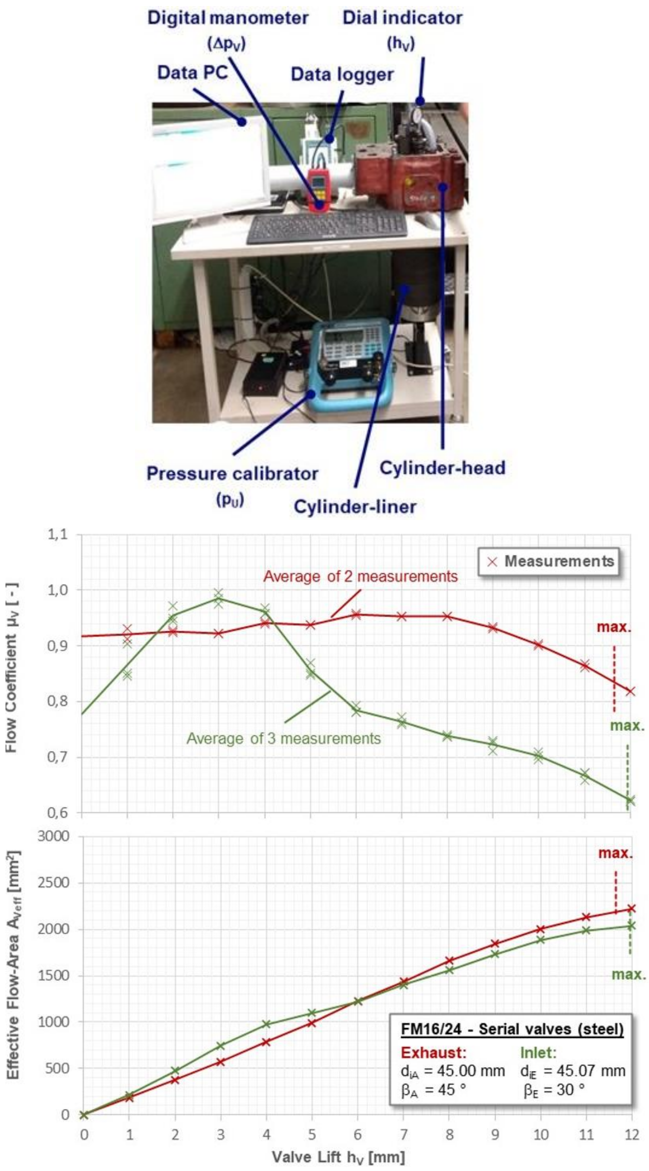

Fig. 4. Repeatability of fluid-dynamic measurements (serial valves)

As known from other medium-speed engines, the flow coefficients for the exhaust valves at higher valve-lifts are larger than the ones at inlet side due to diffuser effects in the exhaust-port. While at higher valve-lifts rising geometric flow areas are more compensated by falling flow coefficients, the increase of effective flow area becomes smaller. With maximum possible valve-lift inside the engine, the maxima of effective flow area were not reached. The valve-lift of the engine is not designed too large.

Several prototype-valves with different seat angles were designed, based on the geometry of standard valve for serial application (Fig. 5). For all prototype-valves, the diameter of valve-desk was kept constant for a sufficient mounting space inside the cylinder head. Equal heights of valve-desks ensured constant dead space between valves and piston inside an engine, as well as a constant compression ratio.

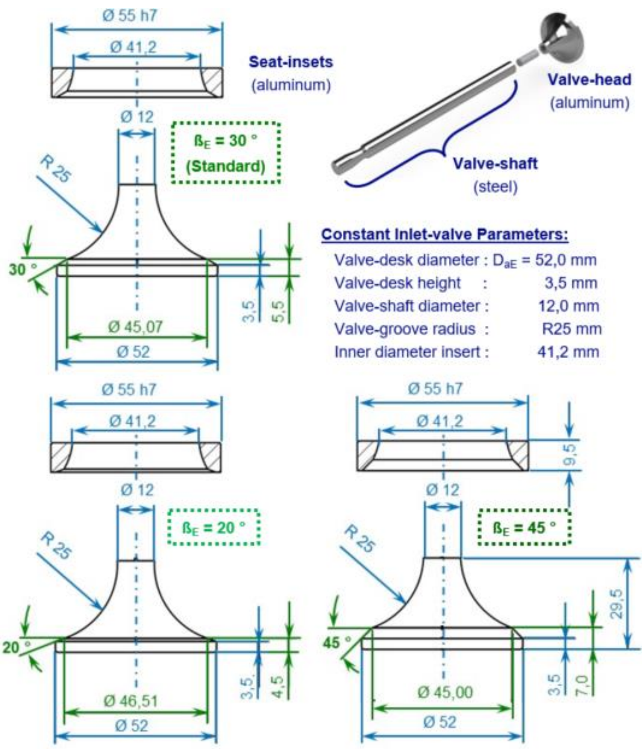

Fig. 5. Split-design inlet valves and valve-seats for fluid-dynamic tests

With a 25 mm radius of damageable valve-groove, it was ensured that sufficient safety would be available regarding mechanical stresses. For economic reasons, all prototype-valves for fluid-dynamic testbed were manufactured in split-design with steel valve-shafts and replaceable valve-heads made of aluminium. Seat-inserts for cylinder head (i.e. 'Seat-rings') were made of aluminium, too. Furthermore, prototypes with standard geometry were made to determine the influence of the manufacturing process. For flow-dynamic tests, the same cylinder head was used as for measurements with serial valves. Serial seat-inserts inside this cylinder head were ground out and replaced by the aluminium ones.

Fig. 6 shows the results of the fluid-dynamic test for the variation of valve-seat angle at inlet. With the split-designed valve in serial geometry, the influence of manufacturing process for prototype could be quantified.

Compared with flow coefficients of serial valves (dotted line and green line in Fig. 4), it is to be seen that the split-designed prototype valve with the same geometry as serial valves delivers slightly lower volume flows at higher valve-lifts and in that way larger flow-velocities. That can be explained by the manufacturing

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tolerance around the joint between valve-heads and valve-shaft. This deviation is about 7 % in volume-flow. All interpretations of influence of valve-seat angle are referred to prototype valve in serial geometry. Diagrams with flow coefficient in relation to the quotient of valve-lift and inner seat diameter (h_V/d_{Vi}) were not used, and comparisons to other engines were not considered.

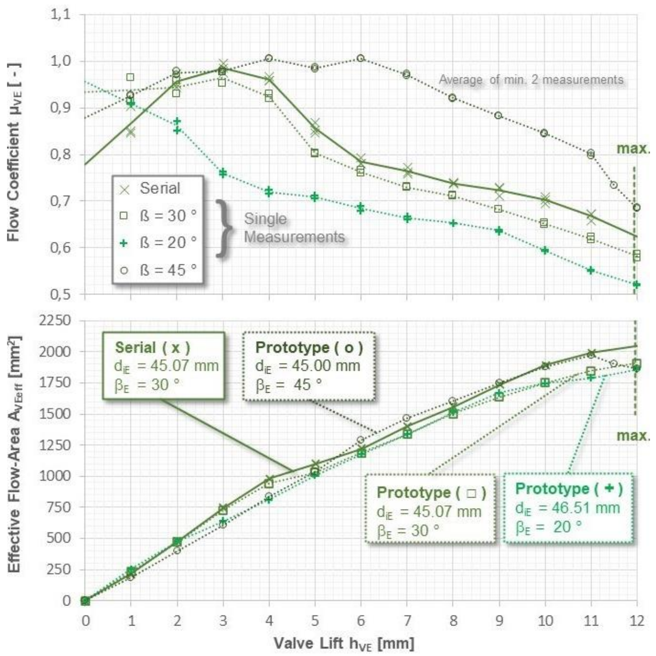

Fig. 6. Results of fluid-dynamic tests for variation of inlet valve-seat angle

As expected, the larger valve-seat angles deliver slightly larger effective flow areas. But the higher flow coefficients are partly compensated by smaller geometric flow areas, so that the effective flow area at higher valve-lifts is only slightly enlarged for a valve-seat angle of 45°. Up to a valve-lift of 5 mm, which is relevant for scavenging during valve-overlap, the valve-seat with 45° delivers worse flow characteristics compared to a standard geometry. The trend of both effects on the thermal load of the engine are opposing, so that an estimation seems to be difficult. As a next step, this impact had to be considered by real cycle calculations including the T/C.

2.2. Test-run at single-cylinder engine and calibration of calculations

Tests at constant-speed operation were carried out at a single-cylinder test engine, shown with its main technical data in Fig. 7. Charge-air pressure was controlled by an electronic frequency-converter of the dry screw-compressor. Back-pressure of T/C's turbine is simulated by throttle-valve after exhaust-gas vessel. Charge-air temperature remains at 45°C before the cylinder for all operating points. To keep comparableness to serial multi-cylinder engines in field operation, the same indicated power (not the effective output at crankshaft) was adjusted. Three attached pumps for lubrication oil and cooling water (fresh and sea water) were taken into consideration.

For this test-run, a 60-Hz constant-speed operation at 1,200 rpm between 100 % and 10 % load was chosen.

The commencement of delivery of the injection pump was set constant at all loads. This operation represents a typical medium-speed propulsion plant with controllable-pitch propeller (CPP operation) and shaft-generator and gensets for auxiliary use on board ships or stationary electric power-generation. Medium-speed engines driving fixed pitch propellers (FPP operation) are rare in commercial ships.

Fig. 7. Test engine for calibration of calculation model

Fuel consumption was measured by a mass flowmeter (Coriolis). All relevant exhaust-gas components were detected by classical physical methods (NDIR, MPA, FIA, CLA). The engine is full-indicated by pressure probes before, inside and after the cylinder as well as in the injection-line. Operational values and emissions according to ISO8178, start of injection, injection rate and heat release in cylinder are calculated by own-developed computer programs.

These measurements carried out were used for calibration and calculation of the cycle as well as for estimation of NOx emissions, as describe below. With additional experimental variations of charge-air pressure, charge-air temperature and start of delivery, the used cycle calculation models were proven regarding right estimation of operational values and NOx emissions.

3. CYCLE CALCULATIONS FOR ESTIMATION OF OPERATIONAL BEHAVIOR AT DIFFERENT SEAT ANGLES

As a next step, cycle calculations were carried out for the measured operating points described in Section 2.2 to validate the calculation models used for these considerations.

For the related cycle calculations the FORTRAN-code 'DYN', developed in a FVV working group [4] for all German medium-speed manufactures, was applied. This source code is open for members and therefore it was developed further and adapted for special needs of the users. For this consideration, the version V36.2 of MAN Augsburg was used [12].

Basis is a real cycle calculation with a single-zone model. Up to 20 cylinders with up to two turbochargers can be connected with up to 12 containers, so that even two-stage charging can be simulated. Calculation of gas exchange is done by simple quasi-static 'fill-and-relief'-method, without any considerations of gas-dynamic effects inside inlet manifold and exhaust-gas line.

All layouts for large 4-stroke MAN diesel engines were carried out with that program. Several comparisons with commercial software (e.g. GT power) showed its reliability even for predictions of back-flow into charge-air receivers at 5L engines. An example for such derated layouts is published in the study of Marquardt [13].

For calculation of the measured operational values, a model with one cylinder and two containers (before and after cylinder) was used (Fig. 8a). Both containers were set to the pressure values for charge-air and exhaust-gas as measured. From pressure indication at full load, the measured heat release inside the cylinder was replaced by a Vibe-function [14]. Vibe-parameters were converted for the other loads following the Woschni/Anisits-rule [15] (Fig. 8b). Therefore, the cylinder mass exponent [16] was set to X = 0.5. Heat transfers to cylinder walls were calculated by Woschni law, as modified by Gerstle/Eilts [6]. For NOx-calculation of medium-speed engines, the two-zone-model by Heider [17] has been well proofed. Its weakness regarding presumptions for variations of air/fuel ratio is known [16] and presented at test-engine FM16/24, too.

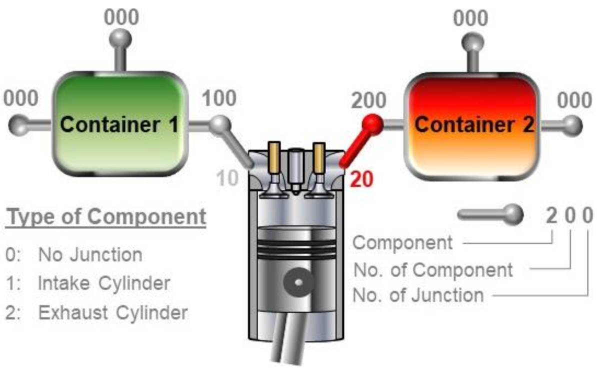

Fig. 8a. Model with two containers for comparison of measurement and cycle calculation

Vibe – Heat Release:

$$\frac{dQ_B}{d\varphi} = \frac{m_{Kz}\cdot \Delta h_u}{\Delta\varphi_{BD}} \cdot 6.908 \cdot (m_V + 1) \cdot \left(\frac{\varphi - \varphi_{BB}}{\Delta\varphi_{BD}}\right)^{m_V} \cdot \exp\left[-6.908 \cdot \left(\frac{\varphi - \varphi_{BB}}{\Delta\varphi_{BD}}\right)^{m_V + 1}\right]$$

$\frac{dQ_B}{d\varphi}$ – Heat Release $\Delta\varphi_{BD}$ – Combustion Duration
m_{Kz} – Fuel Mass per Cycle
φ_{BB} – Start of Combustion Δh_u – Lower Heat Value

Woschni/Anisits-Conversion for Shape Factor m_V and Combustion Duration $\Delta\varphi_{BD}$:

$$m_V = m_{V\_ref} \cdot \underbrace{\left(\frac{\Delta\varphi_{ZV\_ref}}{\Delta\varphi_{ZV}}\right)^{0.5}}_{\text{Ignition Delay}} \cdot \underbrace{\left(\frac{n_{M\_ref}}{n_M}\right)^{0.3}}_{\text{Speed}} \cdot \underbrace{\left(\frac{m_{z\_Es}}{m_{z\_Es\_ref}}\right)^{X}}_{\text{Mass at Intake Closes}}$$

Relation of:

$$\Delta\varphi_{BD} = \Delta\varphi_{BD\_ref} \cdot \underbrace{\left(\frac{\lambda_{V\_ref}}{\lambda_V}\right)^{0.6}}_{\text{Air/Fuel-Ratio}} \cdot \underbrace{\left(\frac{n_M}{n_{M\_ref}}\right)^{0.5}}_{\text{Speed}}$$

Relation of:

Fig. 8b. Modelling and conversion of heat release by Vibe-function [3, 16]

As mentioned before, the test-engine FM16/24 was operated at the same cylinder-indicated power (not with the same effective output at crankshaft) to ensure a comparison to a 9L serial engine. The mechanical efficiency of a MAN-Holeby 9L16/24 inclusive of three attached pumps was known by the authors. The mechanical losses of the crank drive of the single-cylinder FM16/24 test engine were determined before by indicator-method. The conversion of both effective outputs at crankshaft were rendered possible this way. All calculation results are diagrammed in relation to the equivalent output of the serial multi-cylinder engine.

Fig. 8c shows the satisfying precision of cycle calculation for loads between 100 % and 25 % at constant-speed operation. Fuel-oil consumption, peak-pressure in cylinder and air consumption are estimated well at loads above 50 %. Experience shows that differences become larger at low loads. Down to 25 % load the exhaust-gas temperature before turbine has a maximum failure of 10 K, which is quite well. A two-zone calculation model for NOx emissions was calibrated at full load and gives calculated values with a maximum error of 0.25 g/kWh at 25 % load. A valid calibration of engine-model as a precondition for estimations of the influence of valve-seat angle by cycle calculations is documented with this comparison of measurement and cycle calculation.

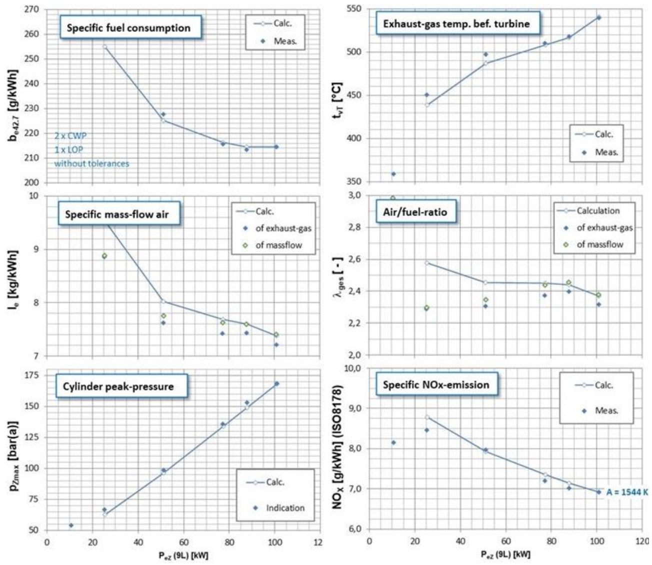

Fig. 8c. Comparison of measurement and cycle calculation at constant-speed operation

For estimation of operational values and emissions, with inlet-valve–seat angle dissenting from standard 30°, cycle calculations for 9L engines with T/C (MAN TCR14-41xxx) were carried out with a six-containers-model (Fig. 9). As is usual for modern engines, a single-exhaust pipe was chosen. The diameter of the exhaust pipe was set to the value of the cylinder bore to ensure constant-pressure charging as applied for field operations of this engine-type. Cooling-water pumps (CWP) for both temperature levels (HT and LT) were attached, as well as an engine-driven lub-oil pump (LOP). The power required for engine-driven pumps was estimated at 19.8 kW. For the sufficient trade-off between fuel-oil consumption and NOx emission for engines with restricted peak-pressure, an injection timing without relevant increase of cylinder-pressure after commencement of combustion was set as a constant for all calculations.

The engine is equipped with conventional valve-timing according to inlet-closing for maximum volumetric efficiency and an overlap of 105° crank-angle. With constant-pressure combustion, a charge-air pressure could be layout to 3.7 bar(a) without exceeding the peak-pressure limits (Table 1). According to the T/C performance of TCR14-41xxx, a maximum turbine-inlet temperature of ≈530°C could be achieved, which is quite high at an ambient pressure of 1 bar. Air/fuel ratio >2 at full load ensures acceptable thermal loads of combustion chamber. NOx emissions according to cycle E2 (ISO8178) for CPP operation ensures IMO-certification for Tier 3 outside Emission-controlled Areas (ECA).

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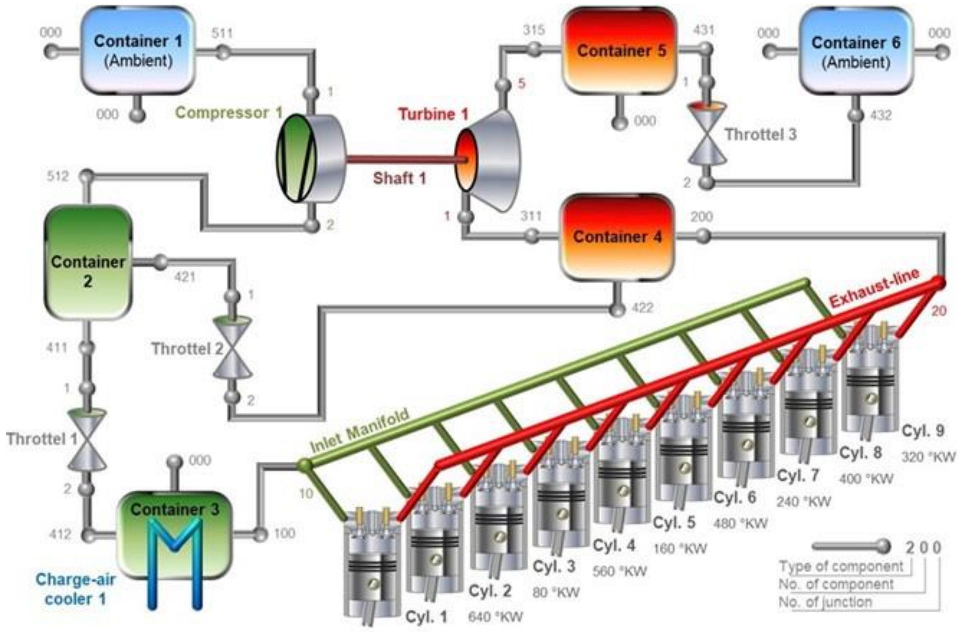

Fig. 9. Calculation model of 9L-engine with constant-pressure T/C and single-stage charge-air cooler

Tab. 1. Results of cycle calculation

Cylinder-output	P_{eZ}	[kW]	100				
Speed	n_M	[min^{-1}]	1200				
Valve-seat angle exhaust	β_A	[°]	45				
Type of inlet-valve	Serial/Proto.		Serial	Split-design			
Symbol in Fig. 6	x / + / □ / o		x	+	□	o	h_{VEmax} = 11 mm
Valve-seat angle intake	β_E	[°]	30	20	30	45	45
Charge-air pressure	p_{vZ}	[mbar(a)]	3700	+ 9	3712	± 0	- 2
Peak pressure	p_{Zmax}	[bar(a)]	170	± 0	170	± 0	± 0
Spec. air-flow	l_e	[kg/kWh]	7.50	- 0.01	7.48	- 0.02	- 0.02
Spec. fuel consumption	$b_{e42.7}$	[g/kWh]	213.7	+ 0.1	214.2	- 0.2	- 0.5
Air/fuel ratio	λ_V	[-]	2.08	± 0.00	2.07	+ 0.01	+ 0.01
Turbine-inlet temp.	t_{vT}	[°C]	531	+ 2	533	+ 1	± 0
Cycle-emissions E2	NO_x	[g/kWh]	7.5	± 0.0	7.6	- 0.1	- 0.1

Deviations to the calculated values with same valve-geometry in split-design are quite small. To compensate the slightly enlarged flow-resistance, the charge-air pressure had to be raised by 12 mbar. The resulting fuel consumption is enlarged by 0.2 % according to the higher losses in gas exchange. Exhaust-gas temperature is nearly the same. The chosen strategy of using split-designed valves for fluid-dynamic tests seemed to be useful.

From that layout, the deviations in operational values and emissions were estimated by further cycle calculations for the unconventional inlet valve-seat angles of 20° and 45°. In every case, charge-air pressure at full load was adjusted to achieve peak-pressures of 170 bar for all layouts. With valve-set angle of 20°, the exhaust-gas temperature at turbine inlet raises by barely 2 K, which seems to be a very small increase in thermal load.

By using the larger seat angle 45°, the thermal load of the engine inside the cylinder, mainly determined by its air/fuel ratio, is nearly the same as with standard geometry. Temperature at turbine inlet is slightly increased by 1 K, due to the lower effective flow area at full lift of inlet valve. No significant influence of valve-seat angles on NOx emissions is identified. This result is in conformity with the fact that usually no IMO-numbers are marked at valves, so that they are not classified as NOx-relevant engine components.

According to Fig. 6, for a valve-seat angle of 45°, the largest effective flow area is not reached at maximum valve-lift. The maximum effective flow area was reached 1 mm before maximum valve-lift, so that the valve-lift at inlet could be reduced with marginal thermodynamic advantages and a slightly improved fuel consumption. Higher dead-spaces between valves and piston or reduced deepness of cut-outs in piston-crown (i.e. 'valve pockets') could also be chosen. To this end, a newly designed inlet cam, inclusive of new NOx certification from IMO, would be necessary.

4. CONCLUSIONS

The influence of unconventional valve-seat angles at inlet of large medium-speed diesel engines was considered with experimental investigations at fluid-dynamic testbed. Before commencing this small research project, related literature studies were undertaken, and no comparable investigations were found.
The impacts of the resulting effective flow areas at the engine's inlet on the operational behaviour and NOx emissions were estimated by cycle calculation for turbo-charged multi-cylinder serial engines. For this use, calculation models were calibrated by experimental data, generated at the single-cylinder test-engine FM16/24.

By wear optimised seat-angles at inlet of β_E = 20° specific fuel consumption raised by 0.1 g/kWh compared to standard β_E = 30°, temperature deviation of 2 K at turbine inlet does not influence the thermal load of the T/C significantly. The air/fuel ratio during combustion is not changed, so that the temperatures at valves will not be higher than the standard design. No negative influences were identified for reduced valve angles at inlet. Given the expected larger valve-seat wear while operating medium-speed engines on fuels without lubricating combustion deposits, the reduction of valve-seat angles at inlet seems to be useful.

Fuel savings with enlarged inlet valve-seat angles β_E = 45° are with b_e = –0.2 g/kWh lower than the tolerances of testbeds for factory acceptance tests. Even with an optimised cam-geometry no thermal derating of the turbine could be detected by these investigations. Enlarged seat angles at the inlet seem to be unproductive.

Mechanical impacts to the wear of valve-seats and head-inserts must be considered by tests in field operation. With valve-seat lubrication, an easy and economical countermeasure was suggested, which can be applied if needed.

Investigations are continued by analogue experiments at exhaust-side as well as with studies of the influence of serial tolerances of casting processes for the cylinder heads [18].

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