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Special Edition MTZ | October 2013

HYBRID Tugboat Propulsion System

ENERGY EFFICIENT Hydraulic Systems for Large Engines

LEAN BURN Concept for Gas Engines

/// GUEST COMMENTARY

Christoph Teetz
MTU

/// INTERVIEW

Christian Poensgen
MAN

OFFICIAL MAGAZINE OF CIMAC
The International Council on Combustion Engines

MEDIUM SPEED ENGINES
DEEP AND DEEPEST INSIGHTS

CALENDAR | **SHANGHAI 2013 – HELSINKI 2016**

The 27th CIMAC World Congress in Shanghai was a great success and CIMAC takes this opportunity to acknowledge the efforts of all who made the event work so well – the voluntary work of the session organisers; the authors submitting and presenting excellent technical papers; the efficient organisational skills of our hosts, the Chinese Society for Internal Combustion Engines; our generous sponsors; the media for its supporting editorial and advertising; and many others.

The next Congress takes place in June 2016 in Helsinki, Finland. A short retrospective of Shanghai appears below and some visual impressions of the Congress can be seen at: www.cimac.com/congress_2013/congress_2013.htm.

RETIRING PRESIDENT YASUHIRO ITOH AT THE 2013 CONGRESS OPENING CEREMONY

CIMAC CIRCLE 2013 | **MARINTEC 2013**

The next CIMAC Circle with presentations and panel discussion will be held on 5th December 2013 during the Marintec marine trade show in Shanghai, China. Precise timing and venue are 14:00 to 16:00 hours in Room N2-M42, Hall N2, at the Shanghai New International Expo Centre (SNIEC). Local registration begins at 13:30.

Participation is free-of-charge but advance online registration is required.

The topic will be Integrated Marine Systems for the Future with presentations from:

- Stefan Müller, MTU Friedrichshafen GmbH, Germany (panel chair)
- Yasuhiro Itoh, Niigata Power Systems, Japan
- Joo Tae Kim, Hyundai EMD, Korea
- Andrei Ludu, AVL List GmbH, Austria
- Christoph Rofka, ABB Turbo Systems Ltd., Switzerland
- Willie Wagen, Wärtsilä Propulsion Norway AS
- Prof. Zhongmin Yang, China Classification Society, China.

Details are available at www.cimac.com/congress_events/events-1.asp.

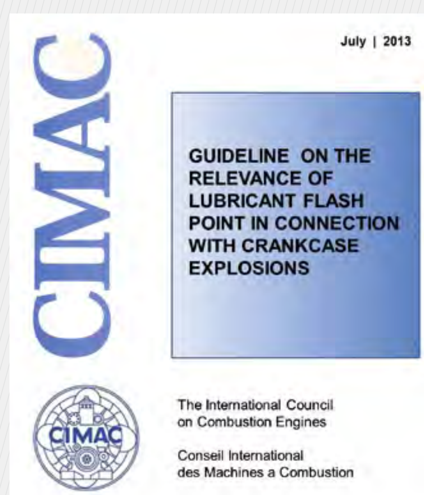
WORKING GROUPS | **PUBLICATIONS**

CIMAC Working Groups have published new documents. All are available for download on the CIMAC webpage:

- Influence of Fuel Quality on Black Carbon Emissions in the Arctic Region Caused by International Shipping by CIMAC WG 5 EEC – Exhaust Emissions Control, available at: www.cimac.com/workinggroups/wg_5_d_ex.asp
- Guideline for Ship Owners and Operators on Managing Distillate Fuels up to 7.0 % v/v FAME (Biodiesel) by CIMAC WG 7 F – Fuels, available at www.cimac.com/workinggroups/wg_7_hf.asp
- Guideline on the relevance of Lubricant Flash Point in Connection with Crankcase Explosions' by CIMAC WG 8 ML – Marine Lubricants, available at www.cimac.com/workinggroups/wg_8_ml.asp.

Presenting the Working Groups

In forthcoming editions of CIMAC NEWS in MTZindustrial, the CIMAC working groups will present their aims and outline current activities.


CIMAC

July | 2013

GUIDELINE ON THE RELEVANCE OF LUBRICANT FLASH POINT IN CONNECTION WITH CRANKCASE EXPLOSIONS

The International Council on Combustion Engines

Conseil International des Machines a Combustion

CIMAC WORKING GROUP 10 | **USERS**


Working Group
ENGINE USERS
WG 10

The CIMAC Working Group (WG) 10 constitutes one of the largest associations of international engine users in the world. Most of the members of this Working Group come from the maritime (shipping) sector, but there is also a considerable number of operators of diesel and gas engine in electrical power plants. The WG would also be a suitable forum for operators of diesel locomotives, engines in the offshore sector and gas compression applications.

CIMAC Working Group 10 meets regularly at six monthly intervals to discuss topical problems and their possible solutions, based on the practical knowledge of the engine users themselves. The experience of longstanding fleet technical managers assists the rapid identification and definition of issues and, in most cases, the development of suitable solutions. This saves engine operators time and money, as well as being a valuable input for engine builders and Classification Societies in their search for answers. For it is no exaggeration to say that it can often take months to find the right contacts at large engine companies or Classification Societies and to get them all round a table to discuss problems which are becoming more and more complex.

GROUP TO MEET IN EUROPE IN NOVEMBER | **INVITATION**

The next meeting of Working Group 10 is scheduled to take place in Europe in November, and the WG is presently able to accept new members. If you are an engine end user operating a large number of engines, have longstanding experience and would like to become active in this valuable global forum, please contact Jörg Erdtmann, the Chairman of CIMAC Working Group 10 Users, at jerdtmann@reederei-nsb.com. WG 10 looks forward to hearing from you!

CONGRESS RETROSPECTIVE | **SHANGHAI SHINES**

The 27th CIMAC World Congress on Combustion Engine Technology for Ship Propulsion, Power Generation and Rail Traction was held from 13th to 16th May in Shanghai, China. The choice of China as the host country for the prestigious Congress reflected the ongoing dynamism of the Chinese market for engines and their applications.

The Technical Programme reflected many current hot topics, such as CO_2 emissions. It consisted of 193 high quality, cutting edge presentations, flanked by 68 poster presentations. Contributions came from 20 countries, with strong participation from host country China.

In total, 889 participants from 33 countries attended the technical sessions while 82 companies exhibited their products and services at the accompanying exhibition.

"All in all, we can look back on a very successful event," notes Axel Kettmann, CIMAC's Vice-President Communication. "This year's Congress was once again a meeting place for all the major players from the engine community and offered extraordinary opportunities for networking with participants from all over the world."

Appropriately, the Congress began and closed with – literally – burning issues. The keynote speech by Professor Wanhua Su of Tianjin University expounded the Development of High Efficiency and Low Emissions Diesel Combustion Technology in China, while the Final Panel Discussion, chaired by Karl Wojik of AVL List GmbH, covered Large Bore Engines in the Light of Changing Fuels. Similarly, the subject of the Special Collin Trust Lecture was Sources of Energy from a Chinese Viewpoint, from Professor Dr. Li Jinghai, Vice President of the Chinese Academy of Sciences, who received the Collin Trust Lecture Award from its Vice Chairman Kurt Olsson.

VIEW OF THE CIMAC CONGRESS EXHIBITION AREA

THE IDEAL PLACE FOR RELAXED ENGINE INDUSTRY NETWORKING: THE TRADITIONAL ABB EVENING

FINAL PANEL DISCUSSION ON LARGE BORE ENGINES IN THE LIGHT OF CHANGING FUELS

PAPER AWARDS | **CONGRESS HONOURS**

At the Gala Dinner the President's Award went to Lieven Vervaeke and Tim Berckmoos, of Anglo Belgian Corporation, Belgium, and Sebastian Verhelst, University of Ghent, Belgium, for their paper The CRISTAL Engine: ABC's New Medium Speed Diesel Engine Developed to Comply with IMO III.

The Best Paper Award went to I. Calvert and A. Zucchelli, GE Jenbacher, Austria, and B. McCully and M. Krajicek, Ricardo, UK, for their paper: Integrated Design, Analysis & Development Processes Applied to the Design of a High Specific Output Gas Engine Cylinder Head.

PRESENTATION OF THE PRESIDENT'S AWARD: LEFT TO RIGHT: KARL WOJIK, FORMER PRESIDENT, CO-WINNER LIEVEN VERVAEKE, ANGLO BELGIAN CORPORATION AND RETIRING PRESIDENT YASUHIRO ITOH

A TEAM FROM GE JENBACHER AND RICARDO RECEIVE THE BEST PAPER AWARD FROM SESSION CHAIRMAN PATRICK FRIGGE (LEFT) AND NEW CIMAC PRESIDENT CHRISTOPH TEETZ (RIGHT)

CIMAC OFFICIAL STATEMENT | **IMO TIER III POSTPONEMENT**

Even as the congress was taking place at the 65th session of the International Maritime Organisation's Marine Environment Protection Committee (MEPC 65), held in London from 13th to 17th May, a narrow majority voted to propose a delay in the implementation, possibly by five years, of stricter limits on emissions of oxides of nitrogen (NO_x) from ships operating in NO_x Emission Control Areas (NECAs). CIMAC's response is published below

TIGHTER NO_x EMISSIONS FOR SEAGOING SHIPS | **RISK OF FIVE YEAR DELAY**

CIMAC is concerned about the decision of IMO's Marine Environment Protection Committee (MEPC 65) to propose a postponement by five years in the implementation of stricter limits on emissions of oxides of nitrogen (NO_x) for ships operating in NO_x Emission Control Areas (NECAs). If the proposed delay is adopted at MEPC 66, it would lead to stranded investments, jeopardise jobs in the shipbuilding industry, and weaken the reputation of the IMO.

The decision was taken despite the fact that, after 18 months of thorough investigations, an IMO experts group concluded that the technologies needed to comply with the Tier III NO_x values are available. The possible delay stems from a Russian proposal to amend MARPOL Annex VI in order to postpone the implementation of Tier III NO_x standards from 1st January 2016 to 1st January 2021, against the votes of the USA, Canada, and some EU member states. The decision still needs to be adopted at the next session of the Marine Environment Protection Committee (MEPC 66), to be held at the beginning of April 2014.

The international shipbuilding industry strongly depends on adequate, absolutely certain lead times prior to the implementation of new environmental standards, in order to adapt its business strategies and related design and production processes appropriately. Thus, a deferral like this, just over two years before its entry into force, has severe consequences for vessels currently being planned or on order for construction from 2016 onwards. Significant investments by manufacturers as well as their customers are threatened, in particular of those companies playing a leading role as pioneers of environmentally friendly Tier III technologies.

Moreover, due to a possible independent decision by the USA regarding its national NECA, an undesirable legislative patchwork may result. And even if the decision were to be withdrawn at MEPC 66, the result is still a year of great uncertainty, as well as the potential undermining of the dependability of IMO legislation, in principle. Finally, the primary proposal of stricter limits for NO_x emissions was initiated as a substantial contribution to the conservation of the environment on a global scale – an ambition which should not be lost sight of. CIMAC therefore strongly requests that IMO members implement Tier III as planned all along, i.e. with effect from January 2016.

THE NEXT THREE YEARS | **CIMAC FOCUS UNDER THE NEW BOARD**

With the end of the CIMAC Congress 2013 in Shanghai, the new Board Members took up their three year tenure. As reported below, all have clear visions of what they wish to achieve before the next Congress. The specific views of the new President Christoph Teetz, MTU Friedrichshafen, can be found in the regular Guest Commentary feature in this issue of MTZindustrial. "I am very proud to have been elected the CIMAC President," Teetz stresses.

The association's agenda is, of course, driven by the hot topics of the industry sector CIMAC represents – for instance, emissions and future ways to reduce them further will continue to influence CIMAC events and Working Groups. This includes gas as a fuel for ships and stationary applications as well as dual-fuel engines.

"We consider gas to be a really viable option to reduce significantly a ship's environmental impact,", says Paolo Tonon from Wärtsilä, Vice-President Technical Programme, while Marco Dekena from AVL, Paolo's counterpart Vice-President Technical Programme adds: "Gas is of course not a new area of combustion engine technology, but the current situation with lower prices and increasing ecological requirements urges the industry to develop in this direction."

The question of developing new technology brings us to another matter: That the success of this industry as a whole – and not only the manufacturers – depends on reliable political and legal parameters is clearly stated in the CIMAC statement which forms part of this CIMAC News. "This statement was prompted by the possible delay of IMO Tier III but is, of course, a heartfelt plea for a firm and dependable framework for each industry sector," states Peter Müller-Baum, the new CIMAC Secretary General. A delay like the proposed IMO Tier III deferral has severe consequences for vessels currently on order or under construction, with a huge impact on manufacturers, shipyards and ship owners. "The uncertainty threatens, in particular, those companies that have accepted a pioneering role in the development of environmentally friendly technologies," he adds.

For the next three years, however, other objectives also share high priority on the CIMAC agenda, including stronger cooperation with ship owners. As Derek Walford from Teekay Shipping, Vice-President Users notes: "I am very glad to be in this position for the next three years and I intend to do all I can to promote stronger participation by engine users at CIMAC. Actually, the Working Group Users is a very active one and I am convinced, also, that the various events CIMAC stages are attractive forums for ship owners all over the world."

CIMAC events are a major part of the association's "backbone", whether we are talking about CIMAC Circles at trade fairs (e.g. at Marintec in Shanghai on 5th December 2013) or, of course, the CIMAC Congress.

The next Congress takes place in June 2016 in Helsinki, Finland, and in a new departure from past practice, the 2016 event will not be completely organised by the hosting National Member Association, but from the CIMAC Central Secretariat with the strong support of the local organisation. As Robert Ollus from Wärtsilä, the current Congress President observes: "This allows us to focus on the regional aspects of such an event, while CIMAC Central will take care of finding partners, internationally, for long term co-operation, notably with regard to the accompanying exhibition."

A further major element in the Association's sustained success is, of course, the nine active Working Groups, which form the large engine industry's interface to other industry stakeholders – end users, regulatory authorities and suppliers – based on a deep knowledge of technology and industrial practice. In a pre-competition environment, they develop and propose standards and start initiatives which help all involved. "This is a win – win exercise for all parties and CIMAC provides the perfect platform for the process," points out the Vice-President Working Groups Christian Poensgen, MAN Diesel & Turbo.

Finally, the association will intensify its public relations activities in order to better reach all stakeholder groups worldwide. As Axel Kettmann of ABB Turbo Systems, Vice-President Communication reports: "We are about to relaunch our website, forming a state-of-the-art platform which will allow all CIMAC members and other interested parties to readily find what they need to know."

Taking all these new initiatives and ongoing activities into account, CIMAC is in a good shape to network within the large engine industry among manufacturers, customers, suppliers, and regulators all over the world, and to enable the collaboration all large engine industry stakeholders need to advance future engine technology in the right directions. As the Past President Yasuhiro Itoh, Niigata Power Systems notes, concluding his Presidency: "I am very proud to have been the President of this extraordinary network for the past three years. And I am happy to support the new Board members as well as my successor Christoph Teetz, for the current tenure."


YASUHIRO ITOH


CHRISTIAN POENSGEN


PAOLO TONON


MARCO DEKENA


DEREK WALFORD


AXEL KETTMANN


PETER MÜLLER-BAUM

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DUAL-FUEL ENGINE WITH CYLINDER PRESSURE BASED CONTROL

Cylinder pressure sensors were initally used to detect knocking and misfiring on spark ignited gas engines. On its latest MaK brand dual-fuel engine, Caterpillar Motoren is harnessing the deep insights into combustion and engine condition that can be derived direct from the origin of engine power in sophisticated control, monitoring and diagnostic systems.

AUTHOR

DIPL.-ING. (FH) BERT RITSCHER
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GASEOUS FUELS IN EMISSIONS REDUCTION

Gas and especially dual-fuel engines, with their ability to rapidly switch between gaseous and liquid fuels, are seen as a viable solution to the 80 % reduction in emissions of oxides of nitrogen (NO_x) required in coastal regions (Emission Control Areas) under IMO Tier III exhaust emissions legislation.

As a result, medium speed engine builder, Caterpillar Motoren, based in Kiel, Germany has designed the new MaK M 46 DF dual-fuel engine, pictured in ❶. This article gives an overview of the relevant design considerations, focussing especially on its electronic control concept which employs cylinder pressure signals as a source of key combustion data for improving the combustion process when operating on gaseous fuel, i.e. in the Gas Mode.

M 46 DF ENGINE CONTROL SYSTEM CONCEPT

The Main development targets regarding performance for the Caterpillar M 46 DF dual-fuel medium speed engines were:

: IMO Tier II compliance in the diesel operating mode
: IMO Tier III Emission Control Area compliance in the gaseous fuel operating mode
: Invisible smoke under all operating conditions
: Minimised methane slip
: Compliance with the latest SOLAS, IACS and MCS requirements.

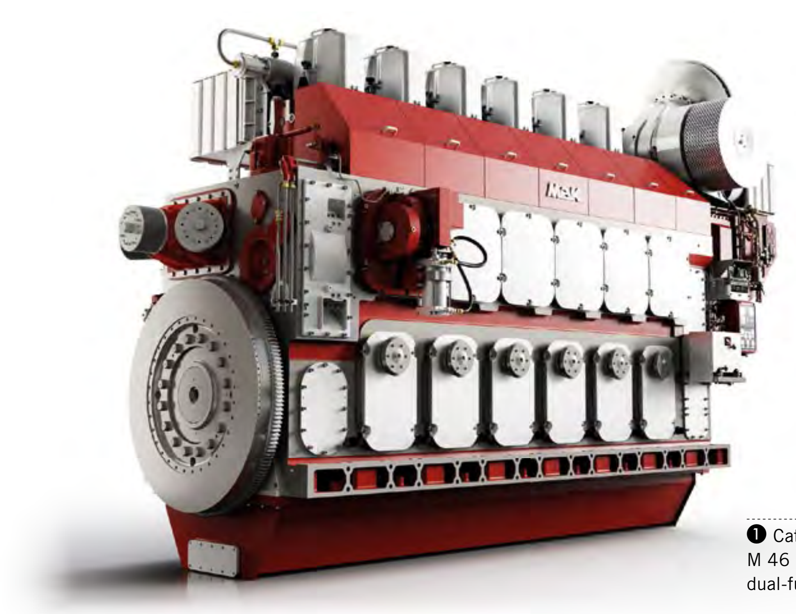
Basic engine data is shown in ❷.

CONTROL SYSTEM DEVELOPMENT

Software development in complex projects is often time consuming and a critical factor to the project timeline. The automotive industry has used model based software development technologies for several years and demonstrated its potential for reducing development times.

Model based software development technology was a key factor in accelerating the development of the M 46 DF. A realtime engine model (plant model) was created early in the project and with its interfaces defined, connected to the control system model while still in the development phase. This enabled the software development team to test control strategies in a simulated closed loop mode even before the first test engine was physically available on the testbed.

Initial gain and performance calibration were completed and validated using this closed loop tool. With these measures the first engine startups and runs went smoothly. Overall, a great deal of testbed time and fuel costs was saved. During the course of the engine's performance and calibration development, a rapid software prototyping tool gave engineers the flexibility to test alternative strategies "on the fly" by deploying a runtime executable for the Engine Control Modules (ECM), based on the control model.



❶ Caterpillar type MaK 6 M 46 DF six cylinder inline dual-fuel engine

		M 43 C	M 46 DUAL-FUEL	
		DIESEL	DIESEL-MODE	GAS-MODE
Emission		IMO II	IMO II	IMO III
Bore	mm	430	460	460
Stroke	mm	610	610	610
Speed	rpm	500/514	500/514	500/514
Power	kW/cyl. (MN ≥ 80)	1000	900	900
BMEP	bar	27.1	21.3	21.3
Liquid fuel consumption	g/kWh at 100 %	177	184	1.9
Gas fuel consumption	kJ/kWh at 100 %			7200
Efficiency	%	47	45.8	50

❷ Basic engine data

A Hardware-in-the-Loop (HiL) test rack is available on which the engine ECMs run with the production software connected to the realtime engine (plant) model. Production sensors and actuators are connected to this test rack to enable realistic, closed loop test capability of the electronic control system and its performance. All basic software validation work starts with this HiL test rack and it has increased the delivered software quality as well as reducing test cell time and fuel costs.

To reach the ambitious engine performance data shown above, it was decided early in the New Product Introduction (NPI) programme to apply in-cylinder pressure based strategies in the control system. This relatively new technology prescribed collaboration with an established supplier with production level experience of this technology.

ELECTRONIC CONTROL CONCEPT

With their twin operating modes comprising 100 % liquid fuel operation and ignition of an air/gas mixture using a liquid fuel pilot, dual-fuel engines require a high number of sensors and actuators under the unchanged requirements of reliability and availability, leading to a completely new, integrated electronic control concept. Since, in the past, new functionalities always required a stand-alone control box, it was decided to have a flexible, integrated system with scalable Programmable Logic Control (PLC) functionality to address all functional needs. Thus, it was possible to reduce the number of sensors, cables and boxes, and with it the number of failure modes.

The electronic control system should be separated into subsystems. First, the engine control system, to control engine speed, load, ignition fuel pressure, gas pressure, the air-fuel ratio (AFR), and the combustion process. The engine control system includes the cylinder pressure subsystem with the cylinder pressure sensors and its own corresponding subsystem, treated as a smart sensing unit. Second, the Marine Classification Societies require an independent protection, monitoring and alarm system with additional PLC functionality, here called the Modular Alarm and Control System (MACS).

ENGINE CONTROL

To manage all essential engine parameters using closed loop control (speed, load, AFR, gas pressure, ignition rail pressure), the Cat Electronics A4 Engine Control Modules (ECMs) are used. The system compromises two ECMs for inline engines and three for vee configuration engines. ECM 1 acts as the master control and is necessary to manage the engine. ECM 2 and 3 are used to operate the gaseous fuel admission valves and operate in conjunction with ECM 1.

To simplify the handling of production software files in the factory and aboard the ship, all ECMs share an identical software file. Identification is handled via a harness code. In practice all ECMs are interchangeable, leading to a reduction in onboard spare parts inventories and addressing the general customer requirement for easy and understandable software files and update processes.

Sensors and actuators that are necessary for engine control are hardwired to the appropriate ECM. Other sensors or actuators are indirectly connected via a CAN J1939 data link. This connection method is generally used to minimise overall system costs by choosing which system requires direct connection to the hardware. In some cases the MACS engine protection system has the connected hardware and the information is made available to the engine control system.

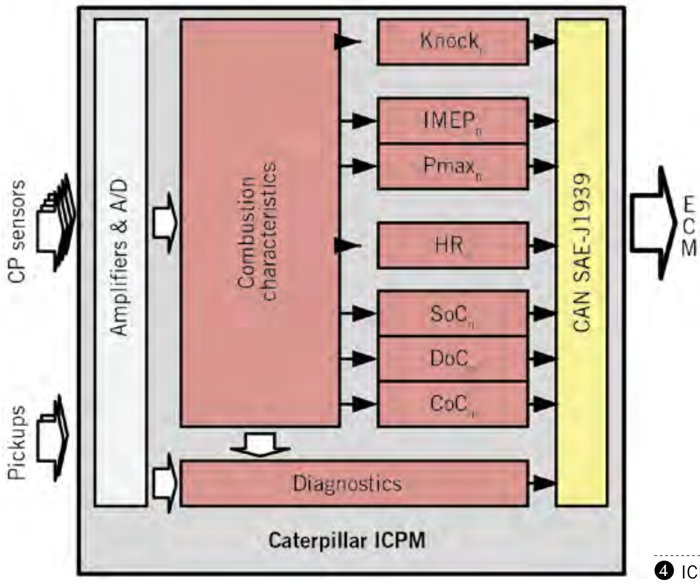
Engine speed and power output is controlled by closed loop (PID) control. The desired engine speed is determined based on a combination of switches and analogue inputs, determined by the MACS. Engine speed control uses three speed/timing position sensors for feedback. These sensors provide information regarding the angular position, rotational direction and speed of the engine. Three sensors are deemed necessary for redundancy, which is a carry-over concept from the MaK M 32 C Common Rail engine system. If at least one camshaft sensor is delivering healthy data, the position of the engine's pistons can be determined on the correct engine stroke of the four-stroke cycle, in order to operate the ignition injectors. The main controlled outputs are the fuel rack actuator that controls the flow of liquid fuel to the cylinders, and the gas admission valves (GAV) that control the flow of gaseous fuel-air mixture to the engine. Engine speed and power are controlled regardless of the type of fuel currently being

❸ ICPM In-cylinder pressure module

used. In steady state, either liquid or gaseous fuel will be used and the control system manages the transitions between the fuel types.

In Gas Mode, AFR is controlled based on turbocharger turbine blow-off and wastegate valves as the actuators. Feedback sensors include intake manifold air pressure and gas pressure. It was decided to apply electrical three-phase servomotors for the dual-fuel engine's wastegate and blow-off valves. These electrical actuators provide high torque and fast response with additional diagnostic capability. Onboard storage of additional hydraulic oil is eliminated, in addition to having a cleaner engine design without hydraulic pipes, pumps, tanks, and filters. Three-phase power supply and control of the actuator is achieved via embedded servo amplifiers. The servo amplifiers receive the absolute positioning command from the engine control system and transmit diagnostic information. Failsafe positioning in case of loss of electrical power or other failure modes is included. If the three-phase rotating electrical field is not energised, an internal actuator spring positions the actuator to a predefined position with the air valves closed.

Ignition of the combustion process in the Gas Mode is achieved using a micropilot common rail fuel system. It consists of a high pressure pump, high pressure fuel jumper lines and common rail injectors. The high pressure pump is mounted on the engine at the crankshaft's free



❹ ICPM block diagram

end and is driven by the crankshaft gear. The pump pressurises and delivers the required amount of fuel to the injectors and provides the desired pressure under closed loop control via two rail pressure sensors. It is also equipped with an inlet metering valve to control the pump flow and, with it, the pressure in the system.

The high pressure pump is also equipped with a safety valve which provides two functions. One is preventing pressures inside the high pressure system exceeding the maximum allowable system pressure. Once the maximum allowable pressure is exceeded, the valve opens and drops the pressure to below 600 bar and continues with the second function, which is the "limp home" function. This safety function is also active in the case of electronic control failure, which causes the fuel pump to deliver its full flow. Engine ignition timing and pilot fuel volume are controlled by the control system in both the Gas Mode and the Diesel Mode. The variation potential, combined with the cylinder pressure data feedback, is employed for diagnostic purposes.

Flex Cam Timing (FCT) positions (valve timing actuator) are now controlled by the MACS and commanded by the engine ECMs. FCT determines the required intake and outlet valve position based on detected engine conditions. The FCT positions are detected by the engine protection system (MACS) and communicated via the CAN J1939 data link. The system continuously monitors input data in order to detect fault/failure mode conditions. A detected failure related to the safe running of gas combustion would result in the engine reverting to diesel mode or, under certain defined conditions, in engine shutdown.

CYLINDER PRESSURE SUBSYSTEM

Each cylinder is equipped with a cylinder pressure sensor. The sensors' operating conditions were considered at an early stage in the cylinder head design in order to maximise sensor reliability and lifetime. All sensors are connected directly to the In-Cylinder Pressure Module (ICPM), depicted in ❸ and shown as a block diagram in ❹.

The system architecture follows a Smart Sensor Concept leading to a clear functional separation among the devices on the engine:

ICPM functions comprise:

: Sensor power supply and monitoring
: Signal processing and computation of combustion characteristics and signal pattern plausibility checks.

ECM functions comprise:

: Parameter handling
: Control algorithms
: Monitoring of combustion characteristics
: Error handling
: Service interface and diagnostics.

Among its functions the ICPM is used for anti knock control (AKC). Deviating from classical AKC concepts, the ICPM only computes knock levels for every cylinder and each combustion. Classification and cylinder individual control actions are performed in the ECM.

Clearly separated functions lead to lean and well tested interfaces. This not only facilitated the development process, but is also beneficial for the Marine Type Approval process and the associated Failure Mode and Effects Analysis (FMEA).

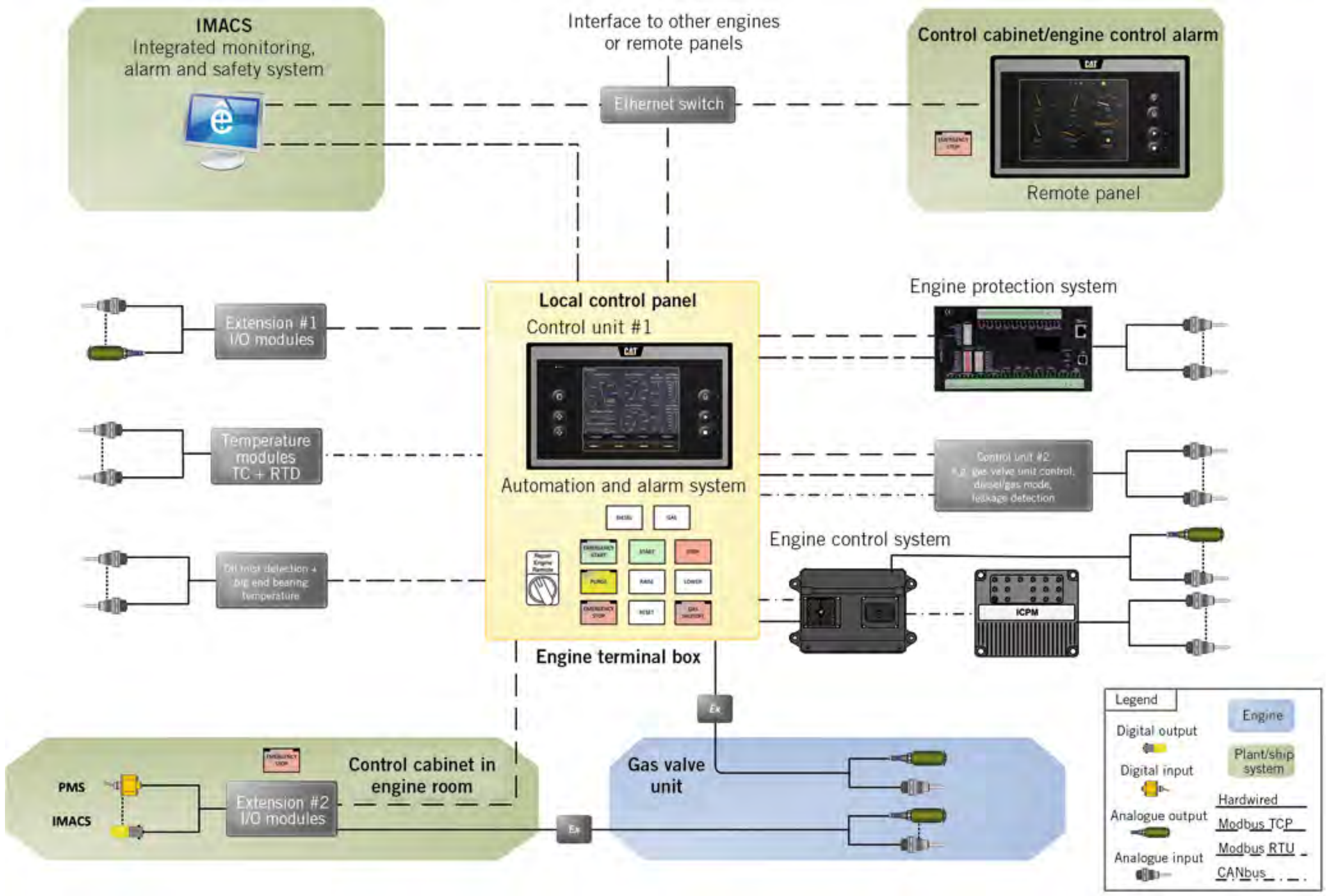
The ICPM computes combustion characteristics for each cylinder. The results are transmitted once per combustion cycle via CAN interface to the ECM.

The SAE J1939 protocol did not so far include parameter groups for combustion characteristics. To accommodate the growing interest in combustion control, the SAE J1939-71 standard (Vehicle Application Layer) was extended by PGNs EC2, ECPMSI, ECCI and ECCAI following a proposal from Caterpillar and AVAT [1].

MACS

A new Modular Alarm and Control System will replace the existing protection system and is the key component for engine monitoring, protection and control, as well as the key interface to the vessel.

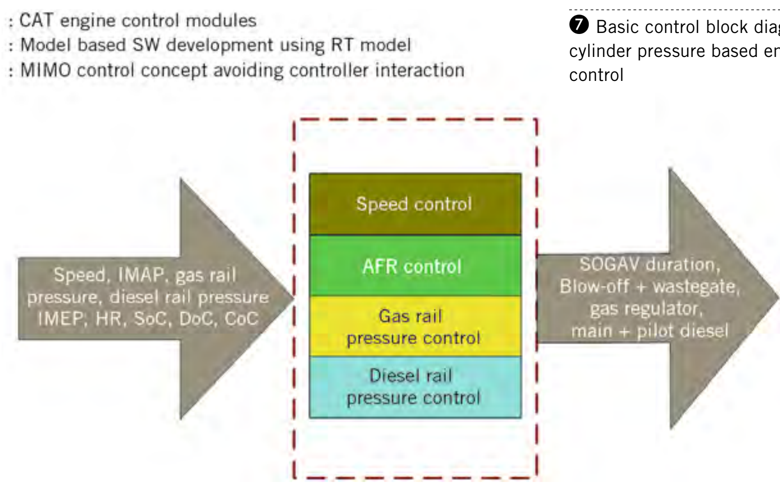
❺ Local control panel



❻ MACS system block diagram

To address the need to monitor all key engine data, not only at the vessel's integrated systems but also locally, a new operator panel/human machine interface (HMI) was designed. As evidenced in ❺, a key requirement was a common look and feel with other Caterpillar or MaK marine engines. Flexibility and expandability of the system were also realised. The local control panels consist of the familiar hardwired pushbuttons and mode selector. For the user, operating this electronic engine thus involves classic, hardwired pushbuttons and is hence identical with earlier mechanically controlled engines. The local display with softkey control helps the engine operator to visualise all key engine data on predefined pages.

Moreover, alarm and plant data can be displayed. Likewise, in multiple engine



❼ Basic control block diagram for cylinder pressure based engine control

installations the values from other engines are viewable on demand. At the customer's request other types of remote panels are available for installation in engine room cabinets, the engine control room or elsewhere. Different display versions and sizes are offered, as is touch input control. For single main engine plant applications, several critical engine sensor values will be displayed redundantly in order to comply with Classification Society requirements. Small backup displays are available to address this requirement. This display can be installed either in the engine mounted terminal box or other engine room control cabinets.

All required automation functions are included in a flexible PLC. This removes the need to have dedicated control boxes for each function (e.g. FCT, Slow Turn). Functionalities in the MACS are as follows:

: Engine start/stop, Local remote control
: Fuel Mode Control (gas/diesel)
: Blackout start/Ready to start
: Gas Valve Unit Control
: Flex Cam Technology (FCT)
: Inert gas flushing control
: Slow turn control
: Charge air preheating.

The complete MACS system is a shown as a block diagram in ❻, with all its hardwired or bus system connections. A colour code is used to show the location of the components. A blue background signifies on-engine mounting, or at least a terminal box mounted on the engine. The green background highlights components in the engine control room. This system features a monitoring and alarm sub-system, located in the number one control unit (yellow background in ⑥) and the remote panel. Measurement values are monitored for critical thresholds and are displayed on the local and remote displays. They are transmitted on bus connections to the vessel's monitoring, alarm and safety system.

CYLINDER PRESSURE BASED ENGINE CONTROL

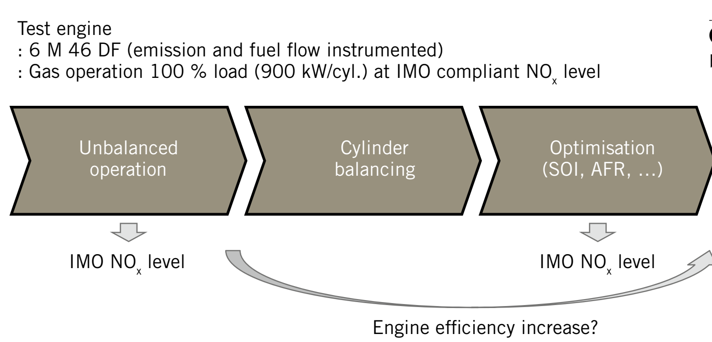
The basic control block diagram for cylinder pressure based engine control is shown in ❼. Three fuel systems are available in this dual-fuel engine configuration. Best-in-class engine performance, speed control and load acceptance behaviour can only be achieved using an integrated control and governor concept. One integrated control system controls all fuel actuators for gas and diesel ignition and the main fuel injection rack system. A fuel apportionment strategy distributes the fuel to the different systems, based on engine operating modes, state machines and the current load profile. In Diesel Mode, the fuel injection system is fired continuously to prevent injector clogging. But the flexibility to control the timing and fuel quality in the pilot ignition fuel system also makes that system useful in the Diesel Mode, to improve combustion characteristics and to detect possible faults.

It is necessary to have several control loops on complex engines and especially on gas engines. This always brings with it the risk of unwanted interactions between the different governors. Loop interaction makes it difficult to model and control the system with multiple Single Input Single Output (SISO) controllers. For example, increasing fuel flow will raise engine speed but also make the AFR richer. The AFR governor will add airflow to regulate the AFR and, with this, engine speed drops slightly. So the speed governor reacts and the total effect is that both speed and AFR governors are fighting each other. A Multi Input Multi Output (MIMO) controller takes account of these loop interactions and was thus selected for the M 46 DF engine.

Several special strategies are included to fulfil market requirements, like lowest possible load in the Gas Mode. One of the limiting parameters in low load operation is individual cylinders having very lean combustion. Cylinder skip firing was introduced to solve this issue. Below a certain load value, cylinders are sequentially cut out, so that the remaining, firing cylinders have higher individual load and thus a richer AFR, leading to stable operation.

ICPM COMBUSTION CHARACTERISTICS

Due to experience from a previous Caterpillar New Technology Introduc-



❽ Cylinder balancing procedure

tion (NTI) project where the potentials of cylinder pressure based engine control strategies were evaluated and quantified, it was decided not only to process direct pressure characteristics such as the knock level, peak firing pressure (P_{max}) and indicated mean effective pressure (IMEP) but also to incorporate heat release characteristics for engine control.

The ICPM uses precise engine data to calculate the pattern of the Heat Release Rate (HRR) from the cylinder pressure pattern. From this, Heat Release (HR) per combustion event and the characteristic crank angles for Start of Combustion (SoC), Duration of Combustion (DoC) and Center of Combustion (CoC) are derived. These values allow fast and precise assessment of the combustion processes taking place in the cylinders [2].

The model based controller design facilitated the integration of online combustion characteristics. From the simplified block diagram with all available input data shown in ⑦, it can be seen that the cylinder pressure subsystem delivers all the combustion characteristic data needed for advanced individual cylinder balancing and control processes.

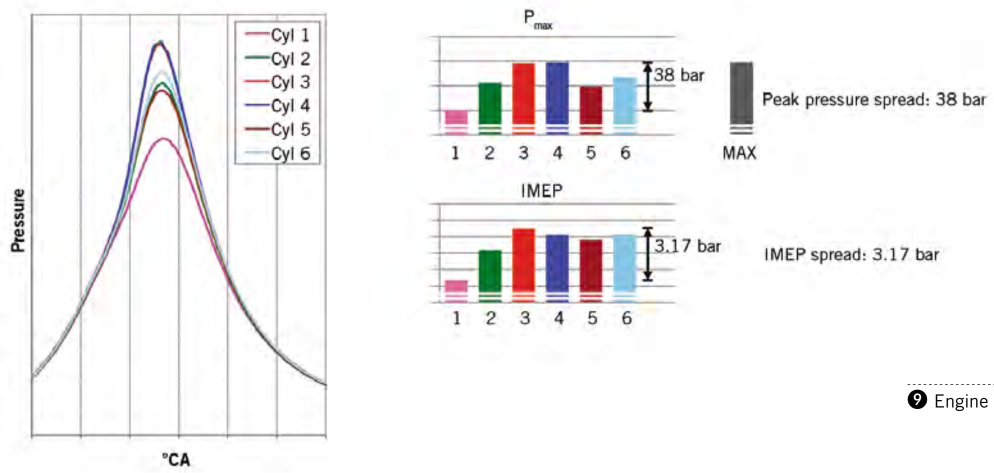
INDIVIDUAL CYLINDER CONTROL

With this online cylinder pressure data available, an obvious step was to improve cylinder equalisation in the Gas Mode. In any reciprocating combustion engine the individual cylinders vary slightly with respect to dimensional tolerances, wear levels, boost pressures, gas rail pressures, fuel injection rate and other parameters. Consequently, the cylinders exhibit differences in combustion, efficiency and knock margin. Integrated balancers in the ECM can compensate some of the differences by adjusting individual injection timings and GAV trims.

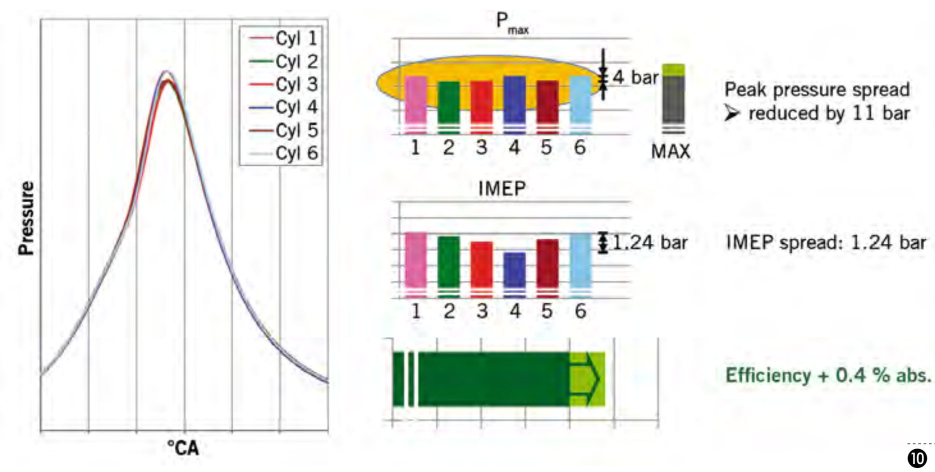
It should not be ignored that balancing has side effects. While the characteristic under control is usually well balanced other characteristics might diverge. A smart combination of two characteristics can mitigate the side effects. Many different strategies can be implemented and their evaluation is currently in progress.

BALANCING PERFORMANCE TEST RESULT

The high availability of realtime combustion data implies a high degree of freedom to design for the most appropriate and effective balancing method. The process for measuring the efficiency gain is shown in ❽. In detail, an MaK 6 M 46 DF test engine with full instrumentation (emissions and fuel flow) was tuned in the classic way to reach the NO_x values required under IMO Tier III in ECAs, with the balancing function disabled. Unbalanced cylinder pressure traces can be seen in ❾. After enabling the balancer and letting the emission values establish a reduction, the NO_x level was measured. Subsequently, with the IMEP balance function enabled, the engine was retuned to regain the IMO Tier III NO_x levels while measuring the reduced fuel flow. Results for the balancing variations with P_{max} and IMEP are documented in ❿ and ⓫. IMEP



❾ Engine operation without cylinder balancing

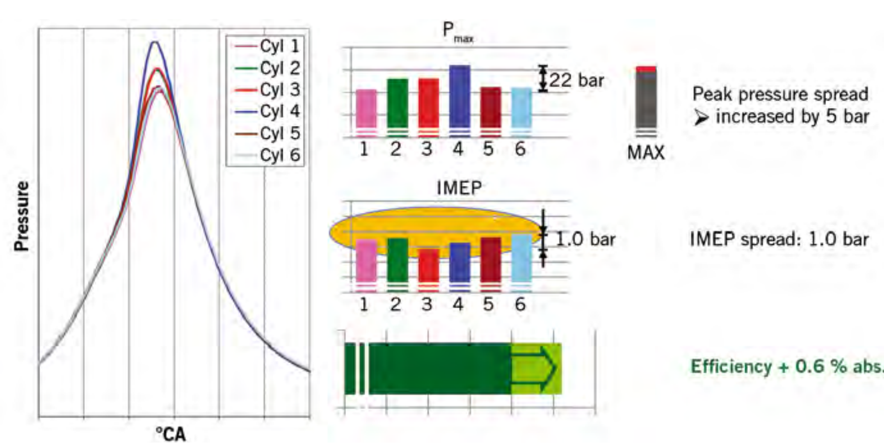


❿ Engine operating with P_{max} balancing

balancing in its present stage of development increases efficiency by 0.6 % but, as mentioned already, work to establish the optimum balancing control method is still ongoing and it is fully expected that higher efficiency gains will be attained.

CONCLUSIONS

In a relatively short development time the first M 46 DF engine was realised by Caterpillar Motoren. An inline six cylinder test engine was built and tested, as was a twelve cylinder vee configuration commercial engine which has already been delivered to a first customer from the cruise line industry. The main enablers were both the new, model based software development technologies and the robust Caterpillar NPI processes, together with Advanced Product Quality Planning (APQP) control of supplier integration. Not only the new cylinder pressure based control system is a technological improvement on this engine platform, but also the MACS system, which helps to achieve enough flexibility to address future needs and simplify automation functions, combined with easy integration into ships' systems. The cylinder pressure based control system offers many more functionalities and customer benefits for future upgrades. One example



⓫ Engine operating with IMEP balancing

would be condition based maintenance capabilities. Long time trend analysis of cylinder pressure data will give an indication regarding cylinder head and combustion chamber condition or wear.

Other Caterpillar engine platforms will receive a replication with this control system, such as the upcoming M 34 DF. Current Caterpillar gas engines employ the extended J1939 protocol in combination with classical, acoustic sensor based knock detection devices. The availability of the protocol itself will make an upgrade to a cylinder pressure based device readily achievable with only marginal modifications to the gas engines' control systems.

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THANKS

We would like to thank the teams involved from Caterpillar Mechatronics, Caterpillar Electronics and AVAT Automation for their contributions to this project.

Nomenclature

AFR	Air-Fuel Ratio
APQP	Advanced Product Quality Planning
CAN	Controller Area Network
CCR	Cat Common Rail
CoC	Center of Combustion
DF	Dual Fuel
DoC	Duration of Combustion
ECM	Electronic Control Module
FCT	Flex Cam Technology
FIS	Fuel Injection System
FMEA	Failure Mode and Effects Analysis
GAV	Gas Admission Valve
HiL	Hardware-in-the-Loop
HR	Heat Release
IACS	International Association of Classification Societies
ICPM	In-Cylinder Pressure Module
IMEP	Indicated Mean Effective Pressure
MACS	Modular Alarm and Control System
MCS	Marine Classification Society
MGPP	Multi Generation Project Plan
NPI	New Product Introduction
NTI	New Technology Introduction
PGN	Parameter Group Number
PLC	Programmable Logic Control
P_{max}	Maximum Peak Pressure
SoC	Start of Combustion
SOLAS	Safety of Life at Sea
TF	Transfer Function
TRL	Technology Readiness Level

"MARINE ENGINES NEED TO OPERATE AT OPTIMUM EFFICIENCY WHILE COMPLYING WITH LOCAL LAWS"

After a false dawn, the shipping sector, which accounts for well over half of all large engines, is only slowly recovering. And of all the market segments, the most stubbornly stagnated is marine medium speed. Dr. Christian Poensgen, Head of Engineering at MAN Diesel & Turbo SE in Augsburg, Germany gives us insights into how a leading manufacturer is addressing the technical challenges of the present market.

Dr. Christian Poensgen came into the large engine industry in 2008, following a successful career developing aero engines and industrial gas turbines, another form of combustion engines but a distinctively different industry. However, as he points out, the similarities far outweigh the differences in terms of applications and contexts, and going right back to basics, the physics is essentially the same. In the meantime, while MAN Diesel is now MAN Diesel & Turbo (MDT) and also encompasses gas turbines and other turbomachinery, Poensgen can be said to have made his mark in the piston engine industry. MDT has launched new high and medium speed diesel, gas and dual-fuel engines during his time in charge of four-stroke technical matters. And, personally, he has gone on to be elected Vice President of Working Groups at CIMAC.

MTZindustrial _ Dr. Poensgen, what knowledge have you gained while in the large engine industry and what have you been able to pass on from your time developing gas turbines?

POENSGEN _ There are a lot of common issues in aerospace and the large engine industry. Engineering and validation processes are very similar. Both industries are subject to very strong regulatory requirements to ensure the safety of the product, either Class controlled or State controlled. To be succinct: a ship in stormy weather or close to the coast with a loss of propulsion power is as endangered as an airplane with a loss of thrust. For both industries, safety ranks first. Unlike the automotive sector, both industries work with small to medium series production numbers. This has a big impact on supply chain strategies and management. The aerospace and the large engine businesses have generally small to medium sized suppliers. Qualification procedures, supplier transition processes and general supplier chain management is very similar. This leads one-to-one into the practices and processes for new product introductions.

You also took over at a rather critical time. MDT's IMO Tier II four-stroke engines were launched, but like everybody else's, they featured an increase in specific fuel consumption (SFC) in the interests of lower NO_x emissions. And in the meantime shipowners are really suffering financially. How much progress has been made in reducing the penalties and what technologies are you investigating and implementing to gain further improvements on Tier II engines?

For Tier II, we have been able to keep SFC neutral through additional measures. Like our competitors, we introduced stronger Miller timing. This caused us to modify cam shaft timings, change turbochargers to newer, higher pressure ratios and revise engine control logics. Some older engines have now got the DNV clean design package, which increases SFC. At the same time, we also used the step to Tier II to take some older engines out of our portfolios.

Again, as with our competitors, we are pursuing engine efficiency rigorously. The main drivers are increases of firing pressures, reduction of internal friction and more intelligent engine control. Within the next few years we will see introduction of two-stage turbocharged engines for marine propulsion.

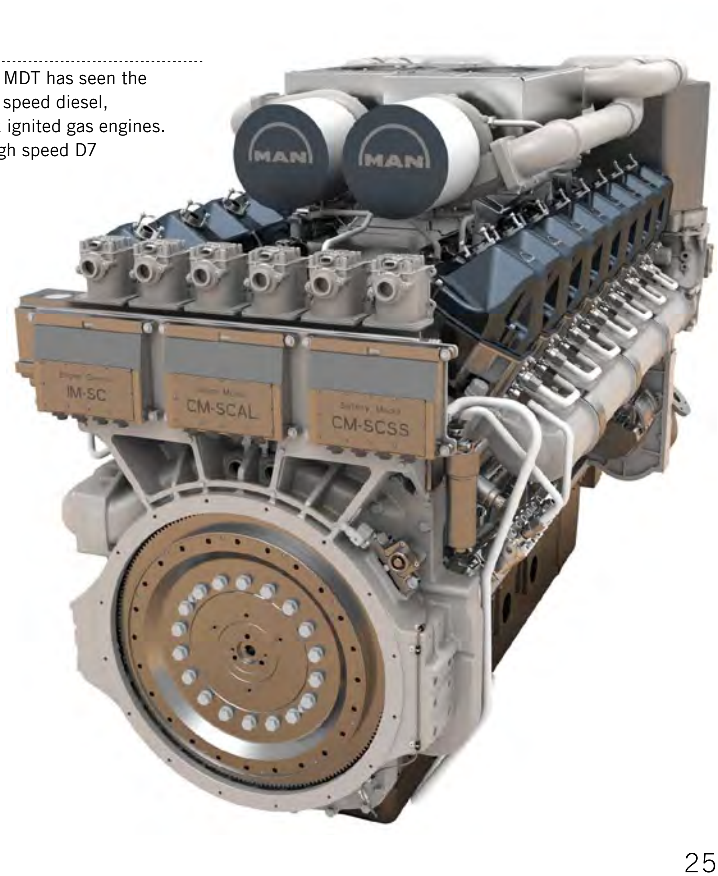
However, the exam questions have changed! It is today practice in both the marine and automotive industries to the target the region of use of the engine when optimising engine controls, for example for Europe, North America, Latin America, India or Canada.

In the marine sector this will become even more pronounced with the new SO_x regulations in 2015 and with IMO Tier III, whenever it comes. Ships travel around the world passing areas with different regulatory requirements. Regardless of where they are, marine engines need to operate at optimum efficiency while complying with local laws – and the latter could proliferate if IMO Tier III is delayed to 2021 because I cannot see, for example, the USA and Canada rescinding the ECA status of their coasts. But whatever happens, compliance will be delivered through more sophisticated engine control and aftertreatment systems.

With respect to economics, whether we like it or not: environmental friendliness is not for free. It always manifests itself in higher SFC, urea usage, scrubbing media and higher acquisition costs due to the need to implement aftertreatment systems or gas storage, handling and safety systems.

The candidate technologies for IMO Tier III seem to be distilling down to those that have stood the test of effectiveness and cost effectiveness. Can you tell me which technologies are crystallising at MDT?

There are three options open to the customer: Like everybody we see the SCR as the backbone system to fulfil IMO Tier III requirements in Emissions Control Areas (ECAs). We have invested in an in house large scale SCR test facility. Results are promising. The first field tests are running and demonstrating stable system operation. The aim is keep the sizing of the SCR package minimal, as space is usually a scarce and costly resource on ships. The second option is to operate with gas. Our 51/60DF and the recently developed 35/44 DF dual-fuel engines are the right candidates for those operators who have access to a gas supply network, or those who operate outside of the ECA zones with HFO fuel and switch to gas within the ECA zones. These engines will not require any aftertreatment systems, neither SCR nor



Poensgen's time at MDT has seen the launch of new high speed diesel, dual-fuel and spark ignited gas engines. Seen here is the high speed D7 scrubbers. Dual-fuel engines are designed to the Otto principle. Whilst they are capable of running with both diesel fuels and HFO, they are optimised for lowest gas consumption values. Once the gas supply market develops further, dual-fuel engines can be easily offered with spark ignition. This technology is well proven in our powerplant business. Once the marine market starts demanding spark ignited engines, we are well able to deliver. Finally there will be applications where EGR delivers the best value to customers. However, for four-stroke engines this requires distillate fuels and a multi-shot capable injection system to suppress soot. MDT is developing such injection systems together with Bosch.

Besides the recent discussion around the introduction of the IMO Tier III, the first step of the IMO SO_x reduction regulation will become effective January 2015. Gas engines give customers a choice either to invest in $DeSO_x$ aftertreatment equipment or into gas installations to fire dual-fuel engines. There is a distinct fuel cost saving from using natural gas compared to HFO or distillate fuels. Depending on the application, gas is the preferred choice from a total life cycle cost point of view.

Yes, gas engines are making headway in marine applications, and one would have to say that MDT was a little slow off the mark. How are measures progressing to rectify this perceived deficit?

The tendency for gas supply growth rates to outperform those of oil and coal started as early as 40 years ago. In the marine sector the market analysts are predicting a ratio of gas to diesel engines for ship newbuildings in a range between 10 and 30 % by 2020. Hence, the bulk of this market is yet to come.

Within the last couple of years MDT has increased investments into gas engine technology. We have a clear roadmap to back up our marine and stationary diesel engine portfolios with dual-fuel and spark ignited gas engines respectively. We have a 10.5 MW spark ignited engine, the 35/44G, on the test bed. Testing of this engine was very smooth, with no hiccups at all thanks to very intensive engineering work at the early design stage. The first plant installation will enter service at the end of this year.

Moreover, we have already announced the L35/44 DF engines to the market for marine propulsion and marine gen-set applications. This engine is on test. Within the next few years we will increase our gas engine portfolio with both larger and smaller bore sizes. A spark ignited gas engine in the 51 cm class came on test in the middle of this year, with further engines to follow.

MDT also recently announced a high speed engine, the D7, in the four to five litre per cylinder class, which both widens Augsburg's offering and fills a longstanding gap between the products of Augsburg and your colleagues at MAN Truck & Bus in Nuremberg. Can you put the new engine into its context within MAN and in the global market, and comment on the potentials and synergies of having a full line of engines to develop and further develop?

Yes, there is a clear strategy at MAN to have within our industrial engine portfolio all engines sizes to serve customers in off-highway applications as well as those who need propulsion systems for ships as large as the A. P. Møller-Mærsk 18,000 TEU class container vessels.

Before we launched the D7 we undertook a very thorough market analysis. Wherever we talked to potential customers we received strong encouragement to move forward. We announced this engine at the SMM in Hamburg in 2012 and got very positive feedback from the market. The development of this engine is challenging. We are using it to transfer knowhow from MAN's truck and bus engines into our medium speed segment and vice versa. Technically there are very interesting synergies to be realised.

Are there any other significant new engines in the pipeline at MDT and what market demands would they address?

We have a clearly formulated development roadmap. New products will be announced to the market in due time.

While everyone talks of specialisation – and your competitors practice it rigorously – MDT has a relatively high level of vertical integration, building its own turbochargers and fuel injection equipment, for example. Is this sustainable long term?

The degree of specialisation is a result of the individual strategies our competitors are following. Some of our competitors follow a strategy to be full system supplier in the power plant and marine businesses. They have acquired companies making winches, stabilisation flaps, tow cables, propulsors, water-jets, generators – right up to full ship design capability. Hyundai itself owns some of the world's largest shipyards. Others are focusing on the propulsion system, like us.

MDT regards some key technologies as so important that we decided to keep them in-house. Among them are the turbocharging system, engine controls, the injection system, aftertreatment and gas systems. It goes without saying that we do not control a core technology unless we have design, production of key parts and full supply chain control in-house.

Those competitors who are focused on propulsion are actually sailing in exactly the same direction as MDT, whilst others have secured their needs through partnerships. In any event, the availability of suppliers capable of delivering those technologies we regard as our core competences is narrow. Our turbocharger business is healthy and profitable, and this opens up the opportunity for us to sell our turbochargers to third parties.

Poensgen sees no reason to deviate from what Rudolf Diesel and Sir Frank Whittle taught us to do

Yes, it certainly seems to work on the turbocharging side, where you were pioneers of both two-stage turbocharging and variable geometry turbocharging on medium speed diesels, with the former culminating in the launch the first two-stage turbocharged engines to use the technology commercially. Can you give us insight into the thinking behind this engine and an update on the impact of the two technologies on the challenges of contemporary engine ownership?
Two-stage turbocharging gives us the opportunity to exploit higher compression ratios whilst enjoying wider operating maps. The advantage in SFC is roughly speaking 3 to 5 % depending on the application. This gives those operators an advantage who run their engines at full load round-the-clock. Operators of marine gen-sets will find engines fitted with a two-stage turbocharging system economically less attractive. Higher turbocharger pressure ratios enable engine operation with the most aggressive Miller Cycles. This is key to reducing NO_x formation beyond what is needed for IMO Tier II.

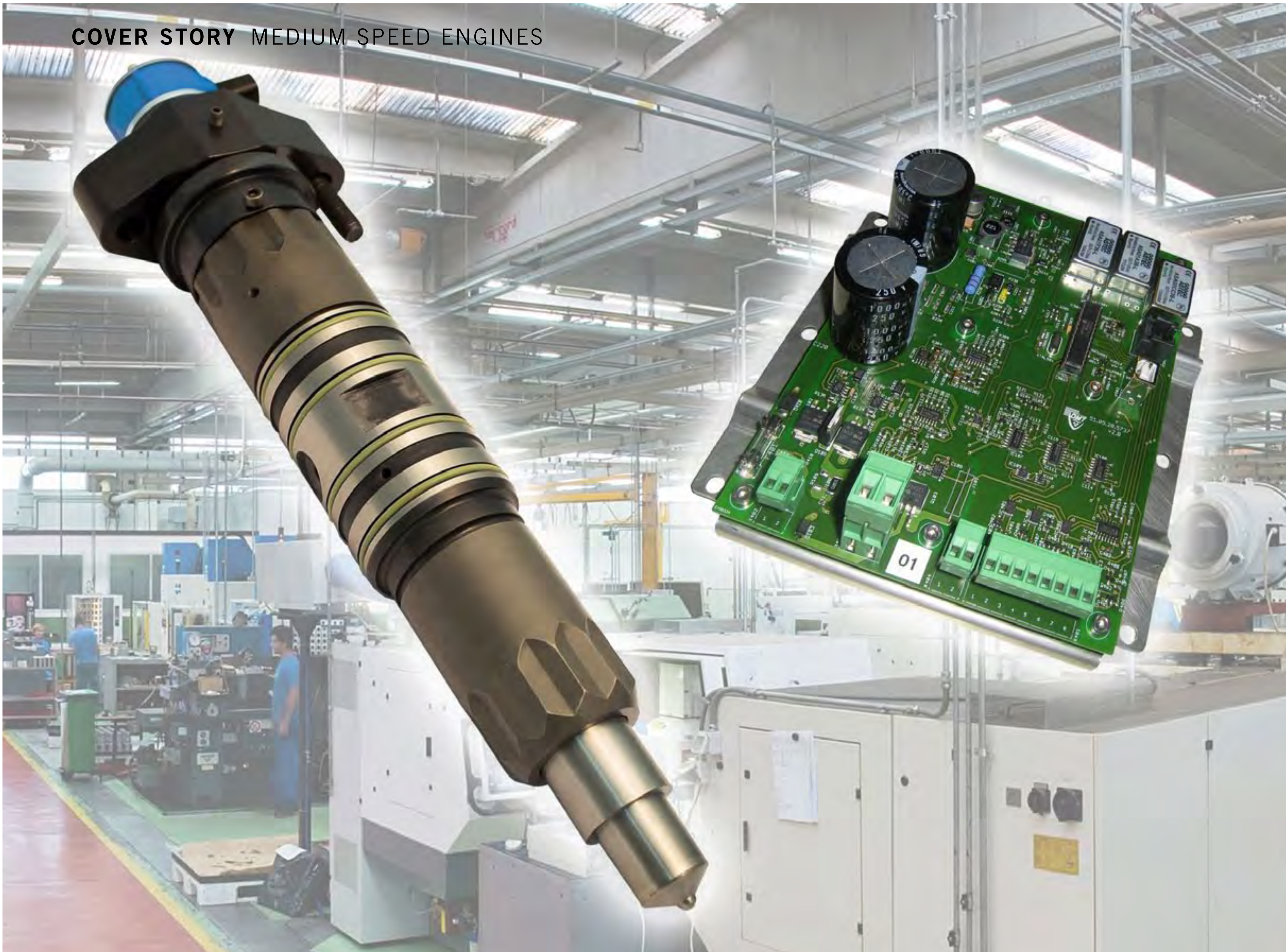
In gas applications, two-stage turbocharging helps us to run at higher mean effective pressures. In a laboratory environment mean effective pressure values up to 26 bar have been demonstrated. Two-stage turbocharging helps to run efficient EGR systems. Looking into the operating costs of engine owners, there is a clear advantage in using engines with a two-stage turbocharging system. So I would not exclude, in the long run, that engines with two-stage turbocharging systems running in power generation might replace today's combined cycle processes.

You were also recently elected CIMAC Vice President of Working Groups. How are you enjoying this role and what specifically do you hope to achieve?
The objectives of the CIMAC Working Groups is to bridge industry developments and industry needs to Classification Societies and other institutions in rule making in a pre-competition environment. I think this is a win – win exercise for all parties involved.

Being CIMAC VP Working Groups gives you a unique overview of the directions engine R&D is taking and of the industry's relations to other stakeholders. Can you share with our readers some thoughts on the general trends regarding these aspects?
Being the excellent platforms they are, all the CIMAC Working Groups are well placed and deliver good results. I see some issues with the maturity of the IMO International Code on Safety for Gas-Fuelled Ships (IGF Code) where there is still some considerable work to do. We need a more common understanding on the handling of gas within the marine industry. This is a key enabler for industry and investors to develop business plans for a denser gas supply network.

Finally, at Rolls-Royce and now at MDT, you have worked for companies that can claim direct links to the men who first invented and patented the gas turbine and the diesel engine – Sir Frank Whittle and Rudolf Diesel. Is this pure chance or are tradition and a sense of history and continuity something that you feel has validity in a modern global business?
Certainly I feel a sense of history here in Augsburg, given that the works is still on the same location where Rudolf Diesel developed his engine – marked by a small monument. But within the context of your question there are further names which come to my mind, like James Watt, Carl Linde, Nicolaus August Otto and Hans von Ohain. Like Diesel and Whittle, all of them were brilliant engineers who transformed industry and even society at the time they lived. Once people start to develop new piston engines and turbomachines, or seek to improve their efficiency, their common starting point and common thread are thermodynamics and fluid dynamics. I see no reason to deviate from what Rudolf Diesel and Sir Frank Whittle taught us to do.

INTERVIEW: Jonathan Walker
PHOTOS: MAN Diesel & Turbo



ADVANCED COMMON RAIL INJECTION FOR MEDIUM SPEED ENGINES

Emissions regulations come in discrete packages enacted on a set date but engine development is a continuous process. Since IMO Tier II, engine builders and component suppliers have worked consistently to eliminate fuel consumption penalties on the first compliant engines and to reduce NO_x emissions to the absolute minimum at source. At OMT the development focus is multi-shot fuel injection with heavy fuel oil from common rail systems where injection pressure is also freely selectable.

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COMBUSTION PROCESS OPTIMISATION

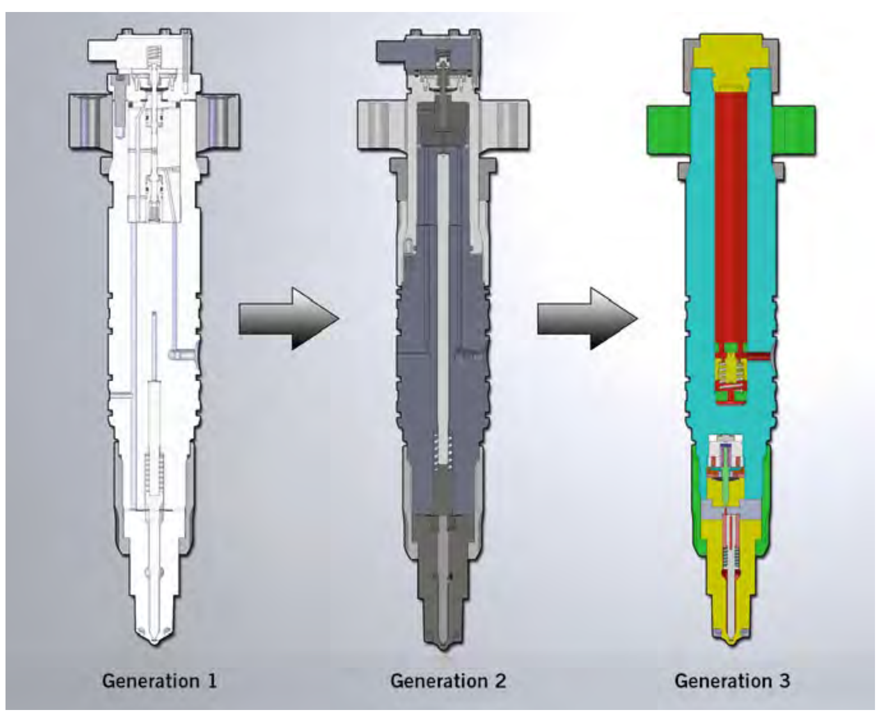
During the past decade, new technologies for large diesel engines in marine, power generation and railway applications have been mainly driven by environmental friendliness, especially in connection with heavy fuel oil (HFO). New emissions legislation requires considerable development efforts from engine designers and system suppliers to produce solutions which fulfil not only that aspect. At the same time traditional customer demands like high engine efficiency under all operating conditions, low life cycle costs and high reliability should not be compromised by measures for attaining low emissions.

Hence, there is continuing demand for more flexible and advanced solutions allowing the optimisation of engine performance to specific application needs, variable operating profiles, the range of available fuels and, last but not least, customer preferences.

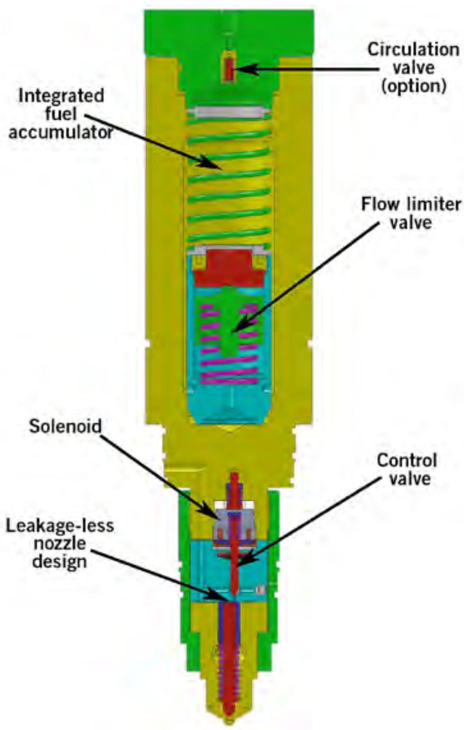
As well as exhaust gas aftertreatment or the use of essentially clean fuels like natural gas, optimisation of the combustion process towards lower emissions via internal measures is in the engine designers' focus. In combination with other key components influencing the combustion process, like turbocharging and variable valve actuation, the fuel injection system plays a leading role in the development of overall engine performance concepts.

In 2005 OMT, Turin, Italy, began developing its first electronically controlled common rail injector for a medium speed engine operating on HFO at a rail pressure of 1500 bar [1], shown in ❶ (left). After successful completion of performance tests on a rig, as well as on the customer's test engine, the injector specification was changed to a higher injection pressure of 1800 bar, in order to fulfil the increased engine performance requirements in terms of emissions of oxides of nitrogen (NO_x). As shown at ① (middle) the redesign included, among other changes, a new solenoid valve developed by OMT.

A further increase in the performance requirements of modern marine engines, implying higher rail pressure and advanced multi-shot capability, then led to a complete redesign of the common rail injector concept in 2009. This third generation injector, concept, shown at ① (right), is suitable for rail pressures over 2000 bar and incorporates an integrated fuel accumulator as well as a flow limiter valve and a new solenoid actuator [2].



❶ From left to right: the three generations of OMT common rail injectors



❷ Latest generation OMT common rail fuel injector with intregral fuel accumulator

HFO INJECTOR DESIGN

The main advantages of a common rail injection system compared to traditional pump-line-nozzle systems are, first, the possibility to perform multiple, electronically controlled injection events and, second, to freely select the most suitable injection pressure level regardless of engine load. Multi-injection capability allows wide combustion optimisation margins, as pilot injections can be used to reduce ignition delay while close coupled multiple injections can be used for rate shaping, thus optimising the heat release rate for each engine load point.

Higher injection pressures generally lead to better fuel atomisation and hence improved combustion, reducing emissions of soot and unburnt hydrocarbons.

From an injector designer's standpoint, increasing flexibility results in additional challenges. Ever higher pressures are more difficult to contain both in terms of material fatigue strength and fluid tightness. Multiple injection capability requires accurate and fast control stages to enable opening and closing of the injector many times per engine cycle, as well as suitably wear resistant moving parts, given the far higher number of cycles than in conventional systems.

In addition, a good multi-shot injector should be able to inject the same amount of fuel for a given command duration regardless of its injection history. In other words, the pressure fluctuations generated in the system by the first injection should not alter the quantity injected by subsequent injections when varying the dwell time between them.

Finally, common rail systems pose additional safety issues, e.g. the potential for overfuelling the engine if an injector needle seizes in the open position. These constraints strongly affected OMT's design choices and led to the creation of the injector layout shown in ❷.

The main innovations in the new design compared to previous OMT injector generations can be summarised as follows:

- Accumulator volume integrated into the injector body
- Integrated circulation and flow limiter valves
- Control valve close to the injector nozzle to precisely control HFO injection
- Zero static leakage to ensure high hydraulic efficiency
- Custom made solenoid to fit inside the injector body.

The integrated accumulator was introduced as a barrier against pressure fluctuations occurring in the rest of the high pressure fuel distribution system and, in particular, to those generated by the rail to injector pipe. These affect injection rate stability and are especially detrimental to multi-shot operation. By making this volume large enough, it is also possible to eliminate the need for a main accumulator (rail) in the system, simplifying overall system layout.

An integrated flow limiter valve restricts the maximum amount of fuel that can be injected in one shot; for instance, in the case of a faulty duration request by the electronic control unit (ECU) or the injector failing in the open position, the valve would limit the maximum injected quantity to a value that the engine can handle. Once closed, the valve can be reopened only by reducing system pressure below 100 bar. This ensures that the faulty injector is isolated until the engine can be shut down.

The valve piston design was optimised towards reducing the influence of fuel viscosity on the maximum permissible injection quantity and on the piston reset time, so as to guarantee correct operation even during engine cold starting on HFO. The valve was designed as a serviceable cartridge that, if needed, can be exchanged directly by a ship's crew during injector overhaul.

The injector design also allows the option of fitting a circulation valve to the injector flange to allow a constant flow of heated fuel through the integrated accumulator during engine standby. This avoids injector performance issues related to operation with the excessively high fuel viscosity typical of cold HFO.

THE HEART OF THE INJECTOR

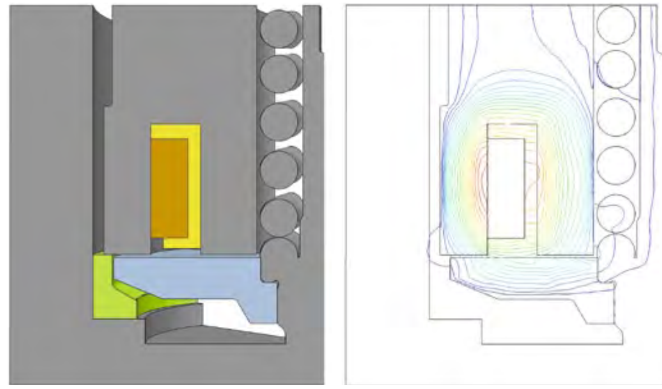
The heart of any electronic injector is its control valve. In order to reduce the dwell time between injections and the injector switching time, the control valve was placed inside the injector as close as possible to the injector needle. This translates into no intermediate mechanical parts between the control volume (the chamber in which the control valve modulates the pressure for pilot injector opening and closing) and the needle itself. It required the design of a custom made solenoid, placed in an area of limited space, under stress from high fuel temperatures and yet able to quickly generate the high forces needed to switch the valve. In addition, to ensure reliable operation on HFO, the solenoid runs in a dry cavity, i.e. air filled and hermetically sealed from the fuel by means of a membrane that ensures the fluid tightness of both the valve piston and body. In this way, the absence of relative motion between the piston and the seal eliminates wear issues on the sealing element and detrimental effects on piston dynamics due to friction forces.

The control valve arrangement described above allows the adoption of a zero static leakage design, i.e. when the control valve is closed no high pressure fuel drains to tank. This improves the hydraulic efficiency of the injector and reduces component overheating due to fuel expansion. The selected concept for the control valve is the two-way principle which confers the advantages of higher switching speed, more precise injector control and less complexity than three-way valves.

As a drawback, these valves involve a pressure reduction in the control volume via the draining of pressurised fuel during the entire injection event. This affects the hydraulic efficiency of the injector. Hence, to reduce the power consumption of the high pressure fuel pump, it is important to minimise the fuel flow rate discharged by the control valve.

As is well known [3] [4], the control volume pressure needs to be significantly lowered to destabilise the forces on the nozzle needle and initiate its upward motion, since the needle area below the seat does not contribute to the generation of a significant lifting force (sac volume not yet pressurised). On the other hand, once the needle has lifted enough to pressurise the sac, the force acting in the opening direction becomes much larger, thus allowing higher control volume pressures without causing needle closure. It follows that a large discharge flow rate is needed to initiate injector opening, but a much smaller one is sufficient to keep it open.

Exploiting this effect, the control valve was sized to discharge the minimum flow rate required to keep the injector open. Likewise, the injector was designed so that the flow restrictor supplying the control volume (allowing needle closure once the control valve is closed) could be machined on the needle, on a surface close to the sealing diameter so as to partly obstruct it when the needle is resting on its seat. Consequently, a smaller discharge flow rate is sufficient to initiate needle opening, since the control volume is fed through a choked flow restrictor. This principle is the subject of an OMT patent application [5].



❸ 3D model for the simulation of the magnetic circuit and results

SIMULATION

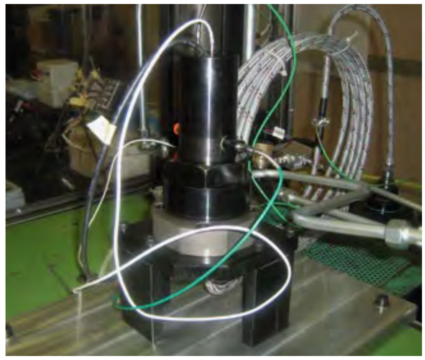
To determine the dimensions and the functional characteristics of the parts composing the injector presented here, accurate analytical investigations were carried out, followed as necessary by appropriate validations using numerical simulations.

In order to support its injector design activities, OMT has worked for many years to develop, implement and refine mathematical and numerical models suitable for accurately predicting the operation of its fuel injection systems. In particular, account is take of issues related to the use of HFO. This work, performed in cooperation with the Politecnico di Torino [6], has resulted in the development of a library of injection system component data that can be used to predict the behaviour of the component or system under development.

HYDRAULIC MODEL

For analysis and simulation of the dynamics of the fuel contained in the cavities of the injector and the feed lines, the fuel was modelled as a nearly incompressible fluid with a constant thermal expansion coefficient and a constant speed of sound. Further, to take account of fuel cavitation phenomena, in accordance with information from a previous publication [7], when fuel pressure decreases to values close to its vapour pressure, it was assumed that the fluid could develop into a two-phase mixture consisting of liquid, vapour and released gases, to which thermodynamic laws of conservation were applied.

❹ OMT injector driver module



❺ Control valve test equipment

After this first analysis it was possible to identify the best ways to simulate the various parts of the hydraulic circuit and, in particular, to define feasible simplifications, whose introduction would speed up the simulations without losing prediction accuracy. Specifically, all the cavities that presented substantially homogeneous fluid conditions were considered as zero-dimensional chambers, i.e. chambers with fluid conditions that vary as a function of time only. Similarly, any narrowing or sudden change of cross sectional area in a fluid passage were modelled as a flow restrictor, i.e. assuming the fluid flow rate to be an algebraic function of the conditions in the upstream and downstream environments. On the other hand, those parts of the circuit where the propagation time of the pressure waves was not negligible in comparison with the total system dynamics were simulated as monodimensional ducts by applying a semi discrete, finite volume approach having second order accuracy.

MECHANICAL MODEL

A similar approach was followed when modelling the dynamic behaviour of the mechanical parts [8]. In particular, the elasticity and the density of the metal parts were initially considered constant, homogeneous and isotropic and the dissipation due to the hysteresis in the material was disregarded.

It followed that all the metal parts presenting negligible deformation or characteristic oscillation frequencies not relevant to the time constants of the system under examination, were considered as parts in rigid motion. Most of the injector parts were treated as such.

The Hertz theory was applied to those components which, ideally, would be in contact only at a point or on a line and, in case of impact, a partial dissipation of the energy accumulated in the deformation was taken into account.

ELECTROMAGNETIC MODEL

The electrical dynamics of the control valve solenoid is significantly faster than the mechanical system it actuates. However, the build-up and decay of eddy currents generated during solenoid switching – and hence the increase of solenoid force – and the motion of the valve piston occur on similar timescales. In addition, since the eddy currents result from changes in a magnetic field, these depend also on valve piston position. Hence, the system was modelled as a variable reluctance magnetic circuit and the equivalent electrical circuits were solved on the basis of the current in the solenoid and the eddy currents.

The dedicated 3D model shown in ❸ was used for the estimation of reluctances and the equivalent inductances resulting from different valve piston positions. Once the characteristic parameters of the electromagnetic device were known, it was possible to simulate the entire electromechanical system.

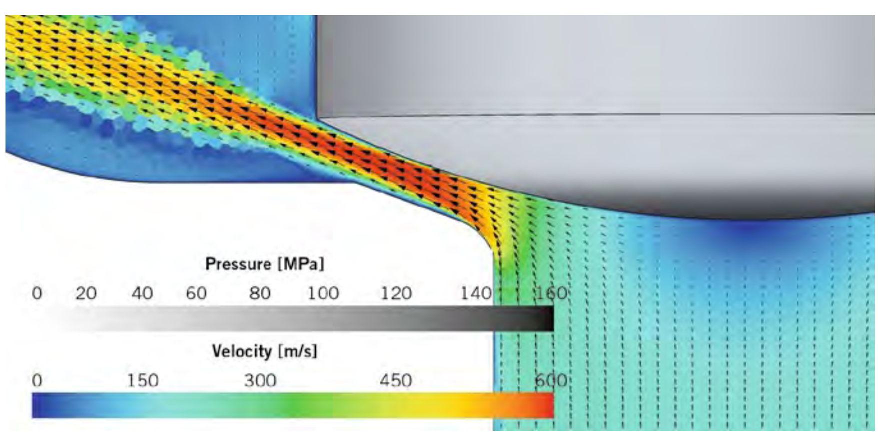
SIMULATION TOOL

The differential equations representing the models described above were implemented and solved using Simulink [9]. In the particular case of solving the partial differential equations which represent the behaviour of the one-dimensional models requiring a discretisation in space not natively available in Simulink, OMT has developed a specific library of components. Further details on the method applied and the definition of appropriate coefficients can be found in [2].

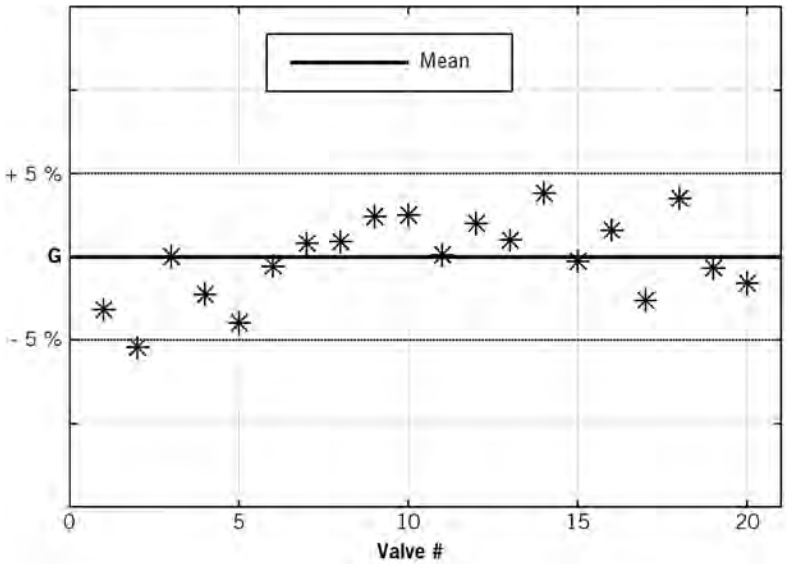
EXPERIMENTAL ANALYSIS

In a first step the basic injector design was optimised on the basis of the results of the numerical simulations. Subsequently two test rigs were set up to evaluate injector performance experimentally: one to record performance data and one to carry out endurance tests.

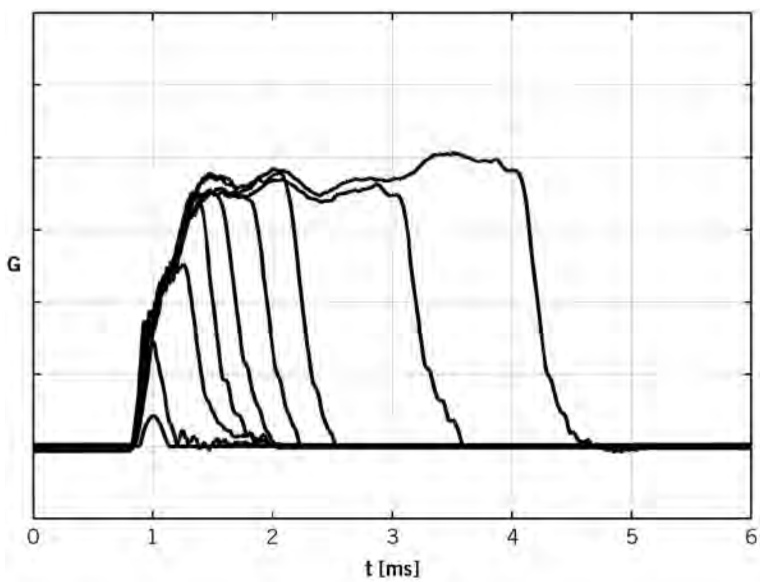
Solenoid drivers developed by OMT are shown in ❹, and were used to supply the current needed to rapidly actuate the control valve and thus obtain single and multiple injections of varying number and duration. Performance tests included the measurement of the injection rate by means of a custom made rate tube [10], since this allowed evaluation and optimisation of the injection rate shape, in



❻ CFD analysis of the restricted flow area through the control valve seat



❼ Measured valve-to-valve discharge flow rate variation (y-axis schematic)



❽ Measured injection rate shape for different injection durations (y-axis schematic)

order to yield the desired heat release rate when mounted on the engine. By using two pressure sensors, one near the injector nozzle and one close to the end of the rate tube, it was also possible to continuously measure the speed of sound in the rate tube.

It was, hence, possible to directly correlate the overpressure read by the first sensor during injection to the injection rate as described in [10]. By integrating this signal, the injected mass per shot could also be calculated and used to derive the injector delivery curves and shot-to-shot delivery variations. A similar device, shown in ❺ was developed to measure the flow rate discharged by the control valve. This allowed acquisition of useful data for validation of the injector simulation model and the CFD analysis results, as shown in ❻.

In addition, this device was used to evaluate the valve-to-valve flow rate variation, as this characteristic influences injector-to-injector variations in terms of injected quantity and rate shape. It should be noted that the control valve discharge flow rate is directly responsible for determining the opening speed of the injector needle.

With the aim of attaining early feedback regarding injector operation in conditions as close as possible to engine operation on HFO, an endurance test rig was designed for unmanned, continuous operation. It minimised the test time and enabled operation with both low and high viscosity oils at temperatures up to 100 °C.

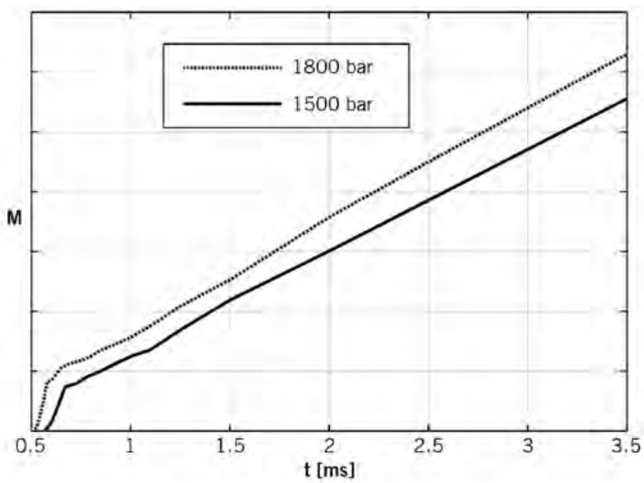
RESULTS

The performance of the control valves was tested prior to their assembly into the completed injectors. It was thus possible to verify correct operation of each valve and, if necessary, to perform required tuning procedures. This preliminary operation is very important because it limits injector-to-injector delivery variations and thus allows simple injector exchange on the engine without modifications to the ECU injector map. The discharge flow rate, in particular, determines the injector needle opening speed and hence affects injection rate shape. Results of tests performed on 20 valves are shown in ❼, demonstrating that it was possible to limit flow rate variability to less than 5 % up or down.

The performance of the complete injectors was then investigated in detail. A first screening was performed with the aim of evaluating single injection behaviour under several operating conditions. The collection of injection rate profiles obtained by varying the duration of the solenoid current signal is shown in ❽. The effect of the integrated accumulator is visible in the limited fluctuations in the injection rate trend when the injector is fully open. It is also possible to see that the desired rate shape was achieved (slower opening, faster closing) and that even very small injections – less than 1 % of the rated injection quantity – could be satisfactorily controlled.

A complete description of injector operation in both small and large injection areas is given in ❾, in the form of delivery curves. They are measured at two system pressure levels typical of operation on the customer's engine and plotted as functions of solenoid current signal duration. The curves demonstrate good linearity and are free from slope inversion areas which would pose problems for the engine control system.

Given the encouraging results in single-shot mode, injector performance was



❾ Measured delivery curves (y-axis schematic)

investigated in multi-shot mode. Shown in ❿ (left) are the solenoid control signal, I, the pressure in the integrated accumulator, P_a, and the injection rate, G, for the case where early, small pilot injection is used to warm up the combustion chamber and reduce ignition delay in fuel reaching the cylinder during main injection. The injector's capability to control small injections allows the engine designer to optimise the combustion process whilst minimising the detrimental effects that larger pilot injection would have on engine efficiency due to increased compression work.

A more complete example of multi-shot operation is shown in ⑩ (right). In this case, pilot injection is followed by pre-injection, closely coupled to the main injection, i.e. separated by a short dwell time – in this case 650 µs. The injector's ability to generate close coupled injections allows further scope to optimise the heat release rate in the combustion chamber by shaping the fuel injection rate.

A further investigation was performed in order to evaluate the influence of dwell time between pre-injection and main injection on the quantity injected during the latter, without varying solenoid current signal duration. For simplicity and ease of correlation, the same duration was chosen for the pre-injection and main injection, yielding injections of about 13 % of the rated quantity.

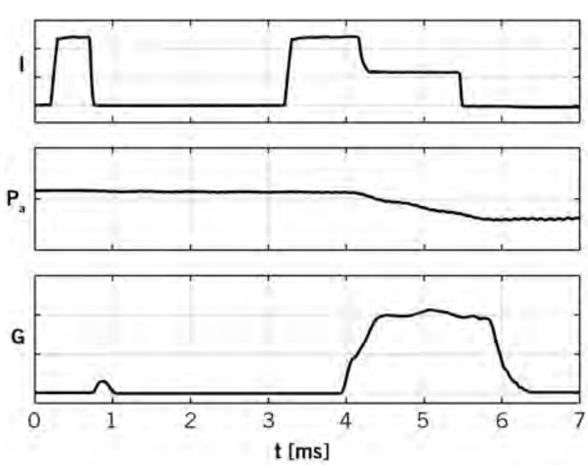
As shown in ⓫ below, the measured injected quantity varied within a band of 5 % up or down, centred on the average value of the recordings – i.e. the expected variation in engine fuelling due to dwell time variation was in the order of less than 1 % up or down. This result was possible due to the use of an integrated accumulator volume, which promptly damped pressure fluctuations generated in the injector during pre-injection. It represents one of the main advantages of the latest generation of electronic injectors in relation to predecessors which showed much larger variations. Such an injector is easier for the ECU to control, since there is no need to compensate the effect of dwell time on the injected quantity, given the low fuelling variations measured on the test rig.

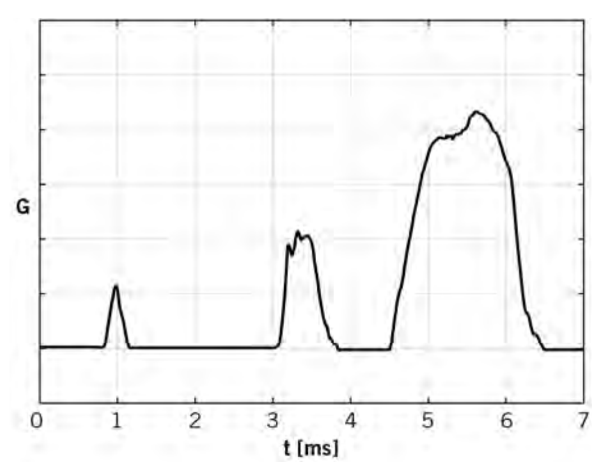
In addition to performance testing, the injector design was validated by means of endurance tests. In particular, three injectors were simultaneously installed and run on the endurance test rig for more than 22 million cycles, equal to 1000 hours of operation on the engine at rated speed. The bench was stopped at the 100, 200, 500 and 1000 hour milestones and performance recorded. No significant variation in injected quantity was determined throughout the test procedure, i.e. the delivery curves shown in ⑨ did not change appreciably, even after 1000 operating hours.

As confirmation of this observation, the inspections performed on the parts after the endurance test revealed no significant wear marks, even on the most stressed parts like the control valve piston seat, as shown in ⓬.

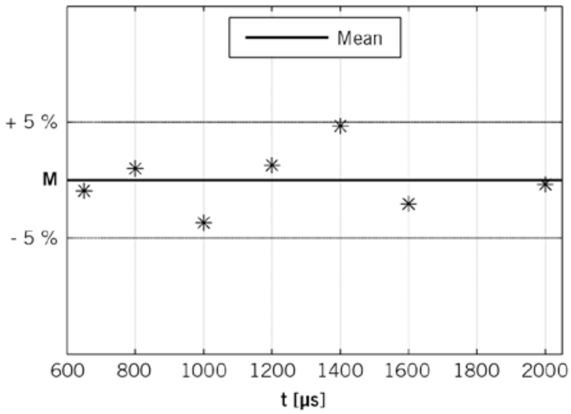
CONCLUSIONS

OMT has successfully developed a new common rail injector representing an evolution from its previous injector generations towards greater flexibility and higher injection pressures.

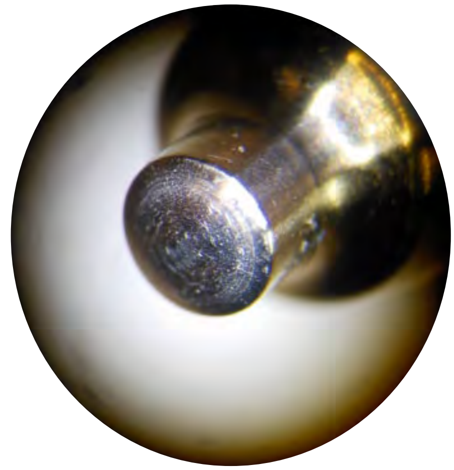




❿ Left: pilot and main injection; right: multi-shot operation (y-axis schematic)



⓫ Measured influence of dwell time between pre and main injection on main injection quantity (y-axis schematic)



⓬ Control valve piston sealing surface after 1000 hours of endurance testing

The measured results obtained from an extensive rig test programme confirmed that the main performance targets have been achieved in terms of injection rate shape, narrow delivery variations, multi-shot capability and high repeatability of the smallest possible injected quantity. In particular, the minimum controllable injected quantity was found to be very low and very stable. This makes this design ideal for multi-shot injection patterns and for application in dual-fuel engines, which will benefit from the ability to inject the smallest possible pilot fuel quantities. The new injector design has proven its durability in a 1000 hour endurance rig test without showing any significant wear on key components.

A first series of the new common rail injector has already passed extensive performance tests on a customer's lab engine. After completion of another test programme and an in-depth evaluation of performance and inspection results in autumn 2013, the new injector is due to be released for a pilot field installation.

The injector concept is scalable to fit a wide range of different engine sizes. Dimensions and detailed design can be tailored to specific customer requirements.

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Nomenclature

3D	Three-dimensional
A	Area
c_d	dDischarge coefficient
c_f	Force coefficient
CFD	Computational Fluid Dynamics
ECU	Electronic Control Unit
F	Force
G	Mass flow rate
HFO	Heavy Fuel Oil
I	Electric current
l	Length
M	Mass
NO_x	Oxides of nitrogen
P_a	Accumulator pressure
r	Radius
t	Time
UHC	Unburnt HydroCarbons
x_v	Valve piston position with respect to seat
Δp	Pressure difference
ρ	Density

THANKS

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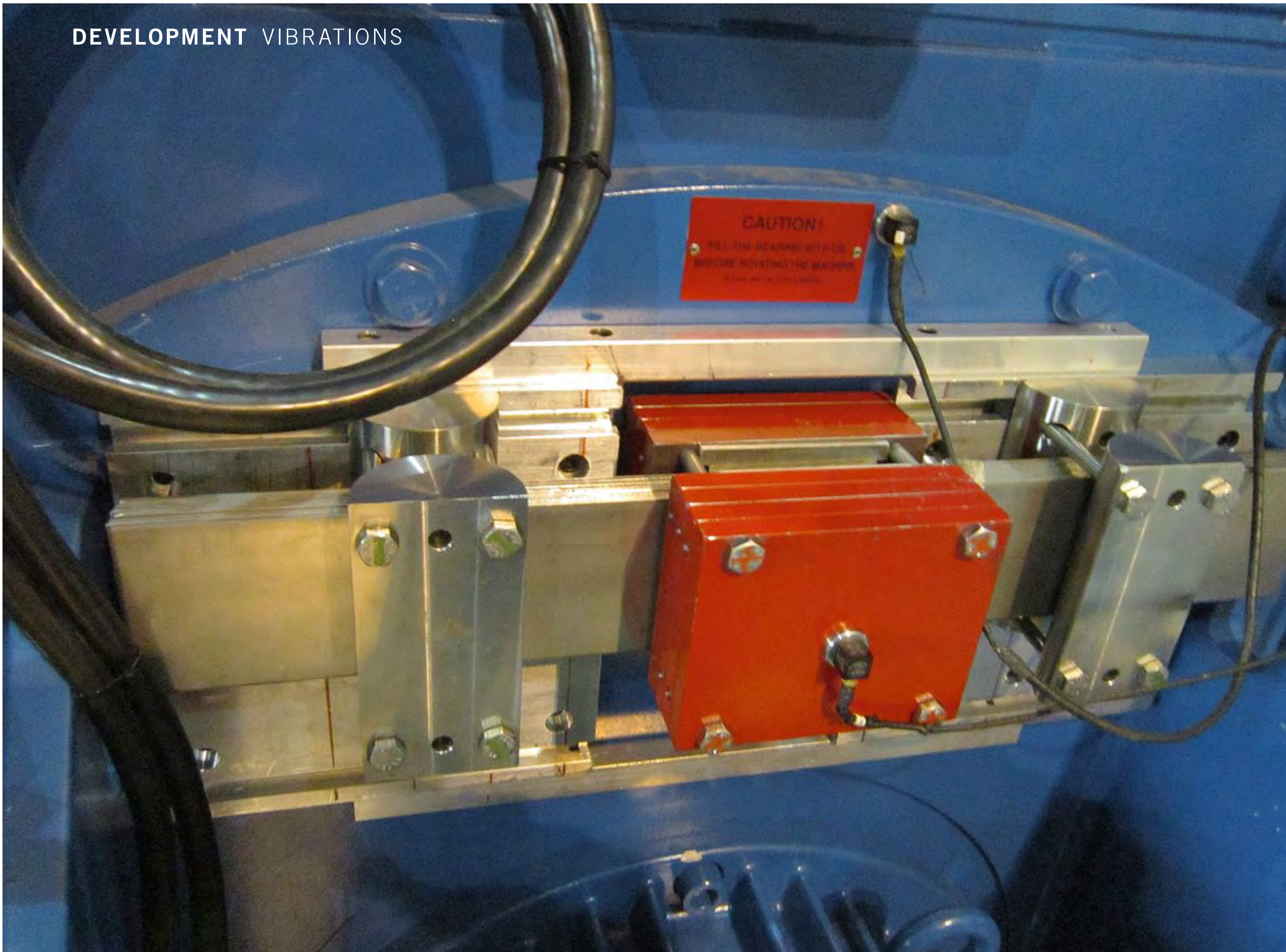

ADJUSTABLE TUNED MASS DAMPER CONCEPT FOR A DIESEL GENERATOR

Large piston engines produce power in short pulses and from as many as 24 cylinders. At worst, when they excite resonances, the resulting vibrations can be highly destructive. Attenuating vibrations when mounting engines and driven equipment is the special challenge taken up by scientists at the VTT Technical Research Centre of Finland.

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MASS DAMPING FOR VIBRATION CONTROL

A tuned mass damper is a well known concept that can be used to reduce undesired oscillations in structures. However, for structures where the dynamic properties are difficult to estimate, a traditional mass damper needs to be designed very carefully to make it work effectively [1, 2, 3, and 4]. For these kinds of structures, an adjustable tuned mass damper (ATMD) is an effective vibration control tool.
The reliability of diesel generators can be increased if their vibration levels can be reduced. Tuned mass dampers have been used to reduce the vibration of diesel engines and generators [5, 6]. Moreover, more complex adaptive tuned mass dampers have been developed [7] for industrial engines in a range of applications.

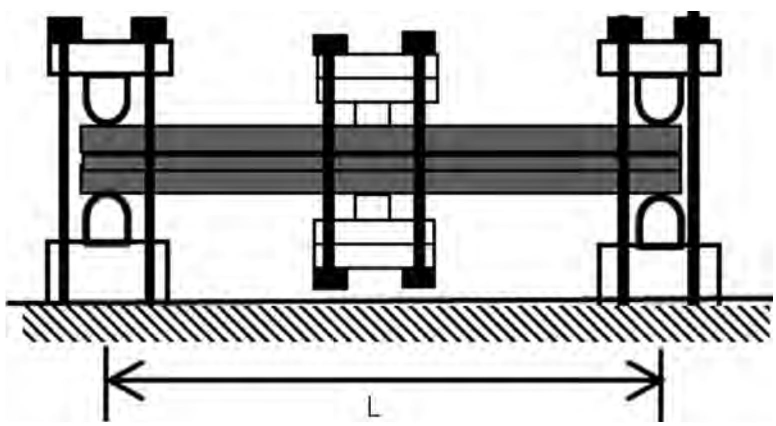
TUNED MASS DAMPING

In this study a simple concept for an ATMD was studied based on a leaf spring and a moving mass, as shown in ❶. The moving mass was located in the middle of the leaf spring packet and the span length was adjusted using semi-cylindrical supports. The ATMD was constructed using many thin plates that were stacked in the manner of a multiple leaf spring.

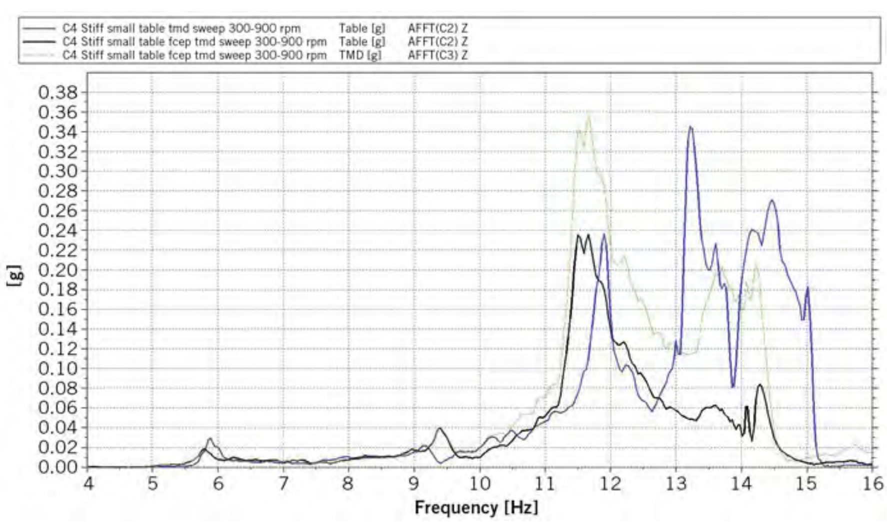
The adjustability of the tuned mass damper was achieved by variation of:

: The span length of the leaf springs
: The moving mass (in a range of 10 to 40 kg)
: The number of thin plate leaf springs
: The length of mass attachment plates.

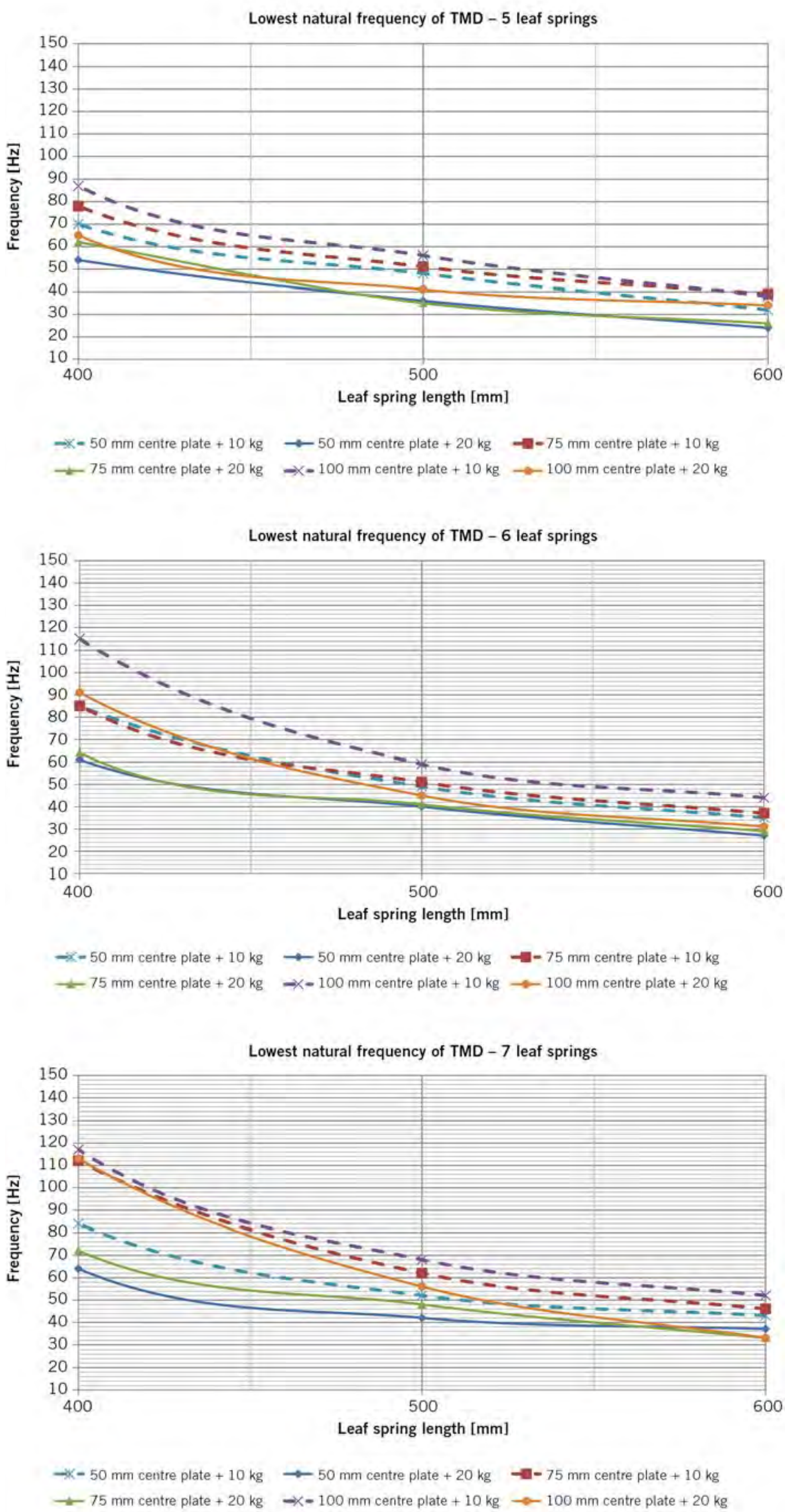
The frequency range of the subject ATMD was approximately 20 to over 100 Hz. The first prototype ATMD was designed



❶ Principle of the adjustable tuned mass damper ATMD; L = leaf spring span length



❷ The sweep test results: blue line = acceleration of original test table, black line = test table with ATMD and green line = moving mass of ATMD



❸ Natural frequency of the ATMD using thin plates

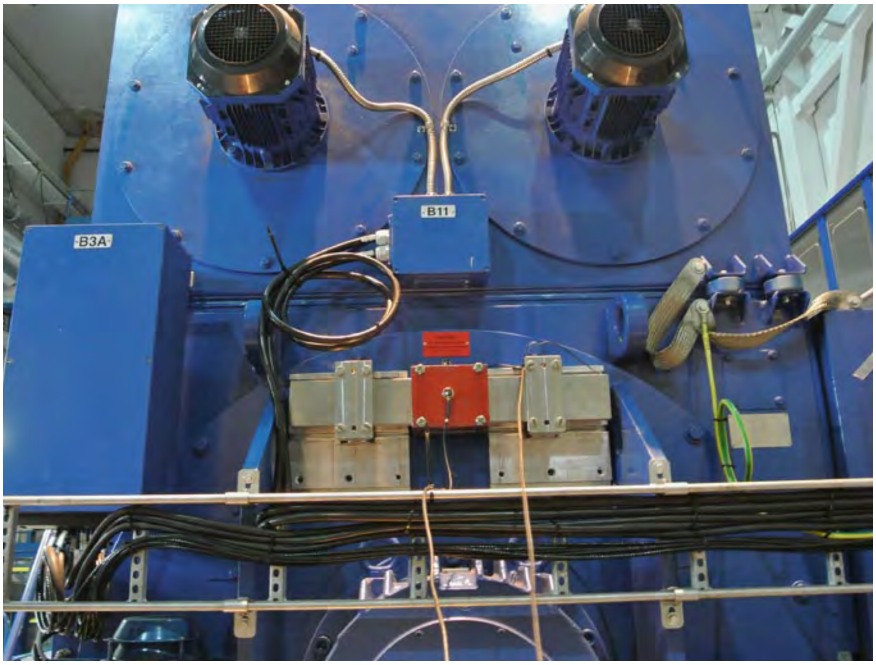
to be used on a self-excited resonance table with a frequency bandwidth of 10 to 20 Hz. The second prototype was designed to be used on a diesel generator set with a frequency bandwidth of 50 to 100 Hz.

TESTING

First, the ATMD was tested on a resonance test table whose total mass was 250 kg. The moving mass of the ATMD was 10 kg. The sweep test result using 300 to 900 rpm is presented in ❷.

The ATMD for the diesel generator was first tested with a modal hammer using a half-power point method to estimate the natural frequency and damping of the ATMD. The span length, moving mass, number of thin leaf springs and length of the mass attachment plates were varied during the test. The natural frequencies of the ATMD are presented in ❸. The degree of damping varied from 8 to 12 %.

The ATMD concept was tested in an ABB generator (IEC size 900, Power 4 MW and 750 rpm), as shown in ❹. The span length and mass were varied during the tests. The test results are presented in ❺. With the generator used in the mass damper tests, the challenge was the generally low vibration levels. The highest vibration level was found near 75 Hz, using 100 % power. The vibration level was approximately 0.4 to 0.5 g and vibration velocity 8 to 10 mm/s. The vibration levels of the generator's structure decreased from 0.48 to 0.13 g in connection with the ATMD using a moving mass of 40 kg. In addition, the vibration levels of the generator's structure decreased to 0.23 to 0.25 g when the ATMD was used with moving masses of 10, 15 and 20 kg. The vibration amplification of the moving mass was approximately 5 to 10 compared to the generator structure. The damping of the ATMD was approximately 5 to 10 % in generator tests. This correlates with the laboratory study of the ATMD, where damping was approximately 8 to 12 %. The ATMD did not increase the vibration levels of the generator at other frequencies. The vibration of the generator equipped with the ATMD, both in



❹ The ATMD concept installed to a generator

normal operation and locked, is presented in ❻, with a 0 to 250 Hz frequency bandwidth. There are no clear amplifications.

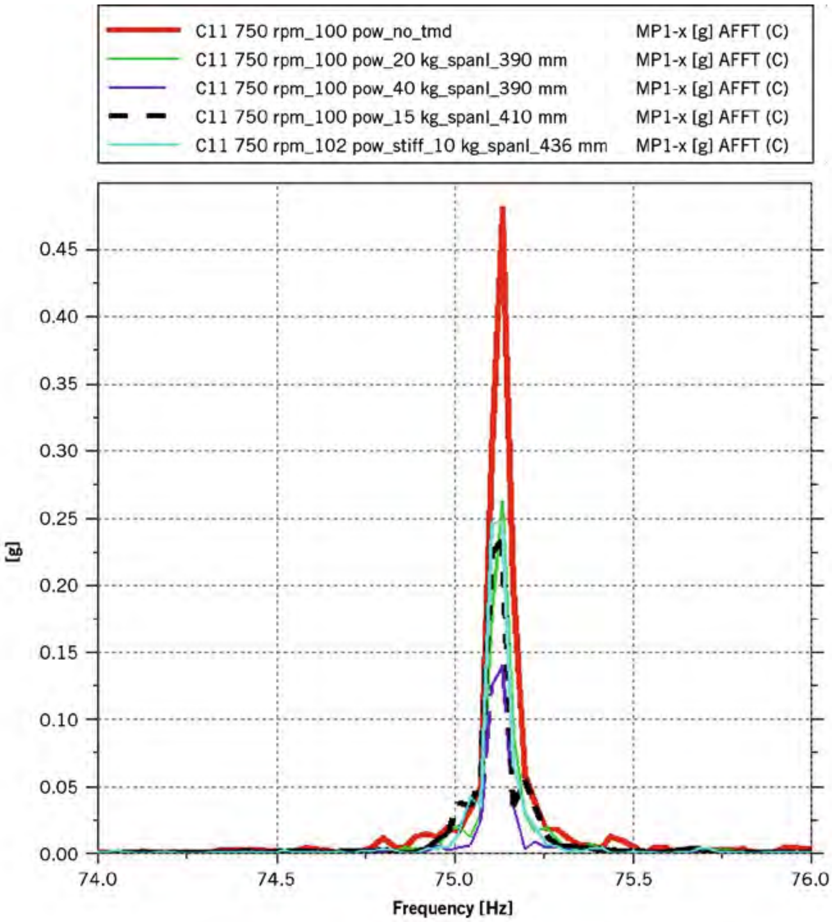
FOLLOW-UP TASKS

The long term properties of the ATMD will be studied using finite element analysis (FEA). The stress of the leaf springs will be calculated using measurement results. Excitation of the ATMD will be derived from generator vibration frequencies and amplitudes measured in the field over several years. Dynamic laboratory tests for the ATMD concept will be conducted with variation in the frequencies and amplitudes on four ATMD units using a total moving mass of 60 kg. Fretting phenomena will be studied in laboratory tests with variation of the dynamic amplitude. To achieve a degree of in-situ realism, environmental conditions will be varied as far as possible. The preliminary studies show that the modified endurance limit for a stainless spring steel is approximately 400 MPa and the maximum alternating stress is less than 20 MPa, where the ATMD moving mass is 20 kg and vibration velocity is 150 mm/s. At the time of writing the dynamic laboratory tests have been underway for two months using a 200 mm/s vibration velocity and no signs of fretting have been detected.

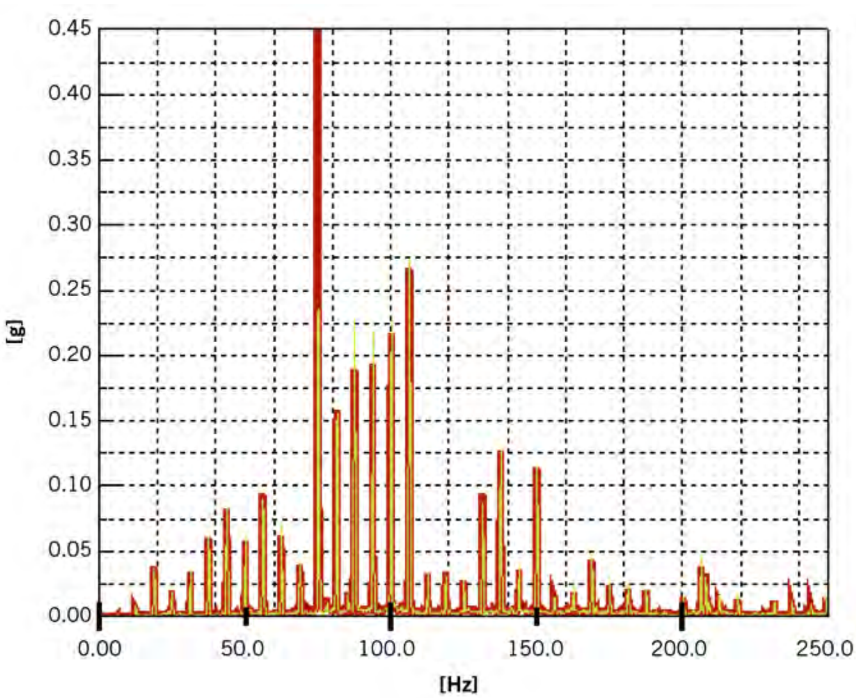
CONCLUSIONS

The ATMD concept has been proven to work effectively in a true scale laboratory test setup. Response of the self-excited 250 kg test table was decreased by approximately 70 to 80 %. The ATMD was also tested in a diesel generator, targeting reduced generator vibration levels. The generator vibration levels decreased by approximately 50 to 70 % using the ATMD compared with the original structure, or the original structure with dummy mass similar to that of the ATMD.

The aim of the project was to make a simple, low cost solution capable of achieving a wide control range. The ATMD can be used as a troubleshooting device with structures that have resonance problems, without extensive pre-study of the structure in question. The moving mass and span length can be readily varied, allowing adjustment of the ATMD in-situ. All the parts of the ATMD are made of steel, so its operating temperature range is wide and its reliability and robustness



❺ Vibration of the generator in the frequency domain: red line = original structure (without ATMD), purple line = generator with ATMD using moving mass of 40 kg, black line = 15 kg, green line = 20 kg and light blue line = 10 kg



❻ The vibration of the generator in the frequency domain: red line = original structure (ATMD locked) and yellow line = generator with ATMD using moving mass of 20 kg

favourable – according to preliminary results, the durability of the ATMD concept is very good.

The ATMD concept studied was designed for use in the generator part of a diesel generator set. However, the ATMD also has the potential to be used, for example, in other engine driven equipment as well as in manufacturing and transport applications in general.

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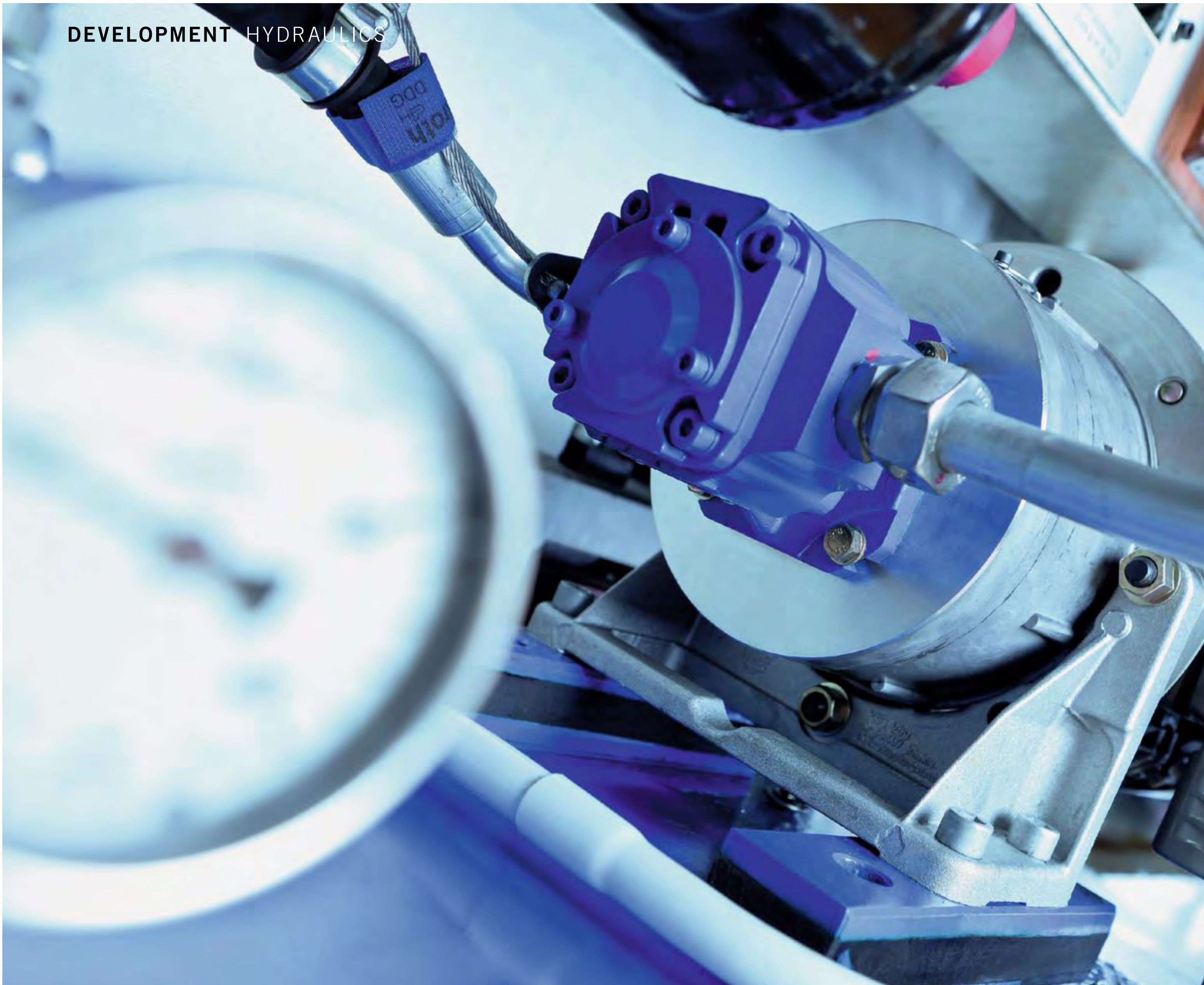
Torsional Vibration Symposium, Salzburg, 21 – 23 May 2014

Following an excellent call-for-papers, the Vibration Association reports that the first Torsional Vibration Symposium is set to comprise some 30 papers from a wide application spectrum. For example, already scheduled are themes as diverse as Torsional vibration simulation under ice impact and Vibration behaviour of wind turbines.
For full details visit torsional-vibration-symposium.com or mail info@torsional-vibration-symposium.com.



ENERGY EFFICIENT HYDRAULIC SYSTEMS FOR LARGE ENGINES

Hydrostatic power offers considerable advantages in terms of transmission flexibility, controllability and power density. There are many potential uses for hydrostatic power on large engines, of which several are standard on low speed two-strokes. Bosch Rexroth and a partner company have developed hydrostatic systems for converting surplus energy in engine exhaust gases into interesting fuel saving options, including power-take-in at the crankshaft and driving auxiliary systems.

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FUEL SAVINGS FOR ECONOMY AND ECOLOGY

As global energy demand increases, costs rise and supplies become scarcer, so innovations that can provide savings are increasingly sought after. An "off the shelf" technology that addresses these challenges and can be rapidly implemented is advanced hydraulics for energy recovery and engine system automation.

Saving fuel has both noxious and greenhouse gas emissions reduction benefits. Hence, a major technology driver is global regulations to reduce the greenhouse gas carbon dioxide (CO_2), and noxious emissions of oxides of sulphur (SOx) and oxides of nitrogen (NO_x). Increasing overall engine efficiency limits the effect of rising fuel costs for the operator, and provides a measurable reduction in environmental impact.

As well as large engines up to 100 MW for marine markets, other applications should be considered within this broad engine power range:

- Stationary power generation
- Mining and special vehicles
- Rail traction applications
- Mechanical drives e.g. pumps and compressors.

Key factors for optimum engine performance are:

- Low operating and maintenance costs
- Flexible alternative fuel capability
- Low engine and equipment investment cost
- Cleaner exhaust emissions
- High reliability and operational safety
- Proven, straightforward technology.

There are various measures available to reduce emissions, which are divided into primary (on-engine) and secondary (aftertreatment) types.

Primary engine emissions reduction measures comprise:

- Variable fuel injection (pressure, quantity, timing) dependent upon load conditions.
- Variable intake and exhaust valve actuation to reduce both fuel consumption and emissions.
- Turbochargers with variable geometry to optimise combustion air intake with engine demand.

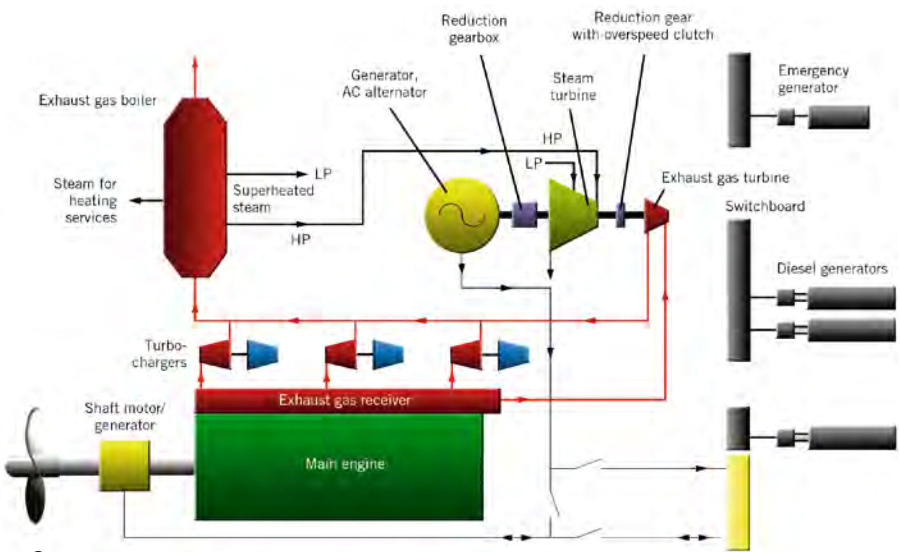
Secondary engine emissions reduction measures comprise:

- Selective catalytic reduction
- Exhaust gas recirculation
- Exhaust gas scrubber
- Fuel/water emulsions
- Humidified intake air.

Given the severity of emissions regulations, it is widely predicted that compliance with future emissions standards will require a combination of primary and secondary measures on diesel engines. The recent introduction of dual-fuel engines, two-stroke engines with ultra long strokes, camshafts replaced by hydraulic actuation of exhaust valves and various waste heat recovery systems all serve to both increase engine efficiency and reduce emissions.

ON-ENGINE HYDRAULICS

Two hydraulic solutions for large engines can be introduced to increase overall engine efficiency. They can readily be installed on large engines during vessel construction or retrofitted to existing engines. Both systems enable significant improvements.



❶ Principle of a waste heat recovery system

TURBO HYDRAULIC SYSTEM

The Turbo Hydraulic System (THS) is similar to other waste heat recovery (WHR) systems – an example is shown in ❶ in that it helps to lower fuel costs and reduce emissions to meet the IMO Energy Efficiency Design Index (EEDI) rules more easily.

Looking at the overall energy balance of a large engine, as shown in ❷, at around 25 % exhaust gases are the most attractive source of waste heat. A key component of WHR systems is the turbocharger. As shown in ❸, turbocharger technology has developed to the point where, particularly in the case of large two-stroke diesels, high efficiency turbochargers need only a specific portion of exhaust gas energy to fulfil their turbocharging function. The remaining energy is available for WHR systems.

The THS system for large engines, shown in ❹, has been developed by Mitsui Engineering and Shipbuilding (MES), a major Japanese shipbuilding company. Standard hydraulic components are used in the THS system. The key component is a compact, reliable, highly efficient hydraulic motor connected directly to the engine crankshaft.

At the end of 2012 a prototype THS system was installed on a vessel for testing. The THS performed well during the test phase and it is intended to install this system on production vessels. MES began this project in 2007 in cooperation with a Japanese government programme to reduce CO_2 emissions from ships.

As depicted in ❺ and ❻, THS comprises hydraulic pumps installed at the turbocharger to generate hydraulic energy out of the exhaust gas flow and drive an hydraulic motor connected to the crankshaft of the propulsion engine. THS thus reduces fuel consumption by supplementing the engine's normal torque output with the hydraulic energy converted from the energy in exhaust gas flow.

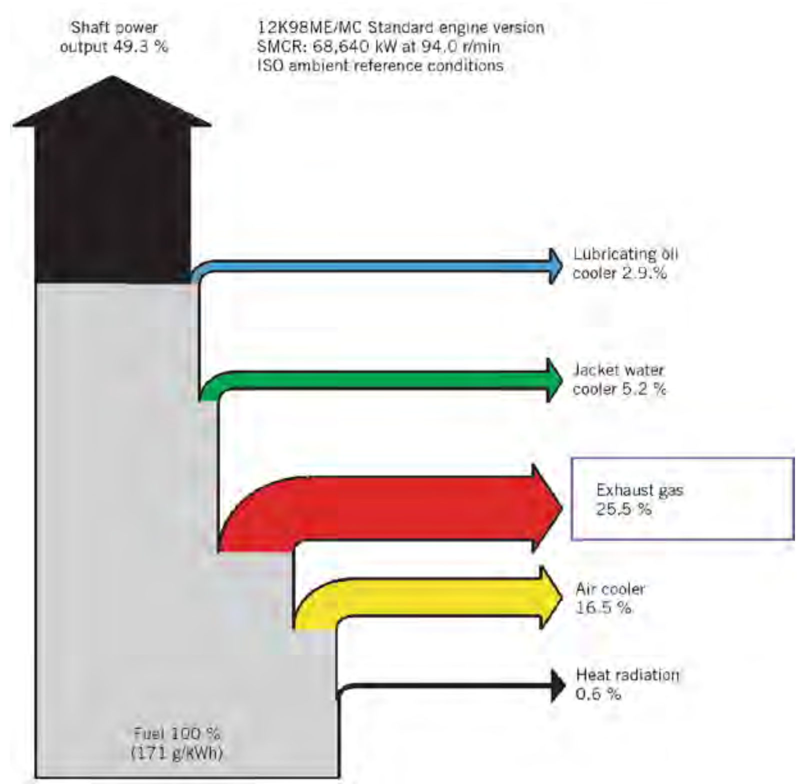
Compared to WHR systems utilising steam and power turbines, the design of THS is less complex, since standard hydraulic components are used, and the auxiliary component count is reduced.

THS benefits verified in the MES vessel test programme comprised:

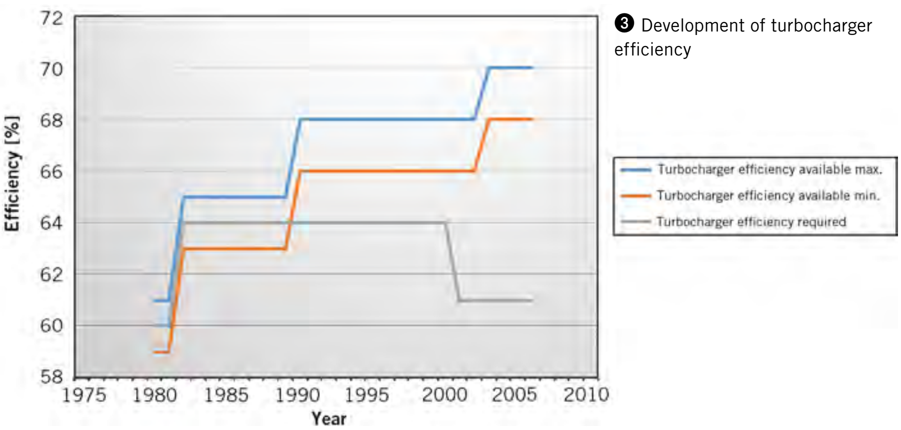
- Reduction in fuel consumption
- Reduction in emissions of CO_2, SO_X and NO_X
- Ability to retrofit older engine designs without existing hydraulic systems
- Use of standard hydraulic components
- High reliability and compact installation dimensions
- Favourable installation costs compared with other WHR methods
- Accelerated return on investment
- Equally applicable on four-stroke engines.

Overall fuel consumption can be reduced by 4 % at 100 % engine load and 3 % at 85 % engine load.

With lower installation costs than other WHR variants, Bosch Rexroth and MES expect that the THS system will be popular on the engines of smaller and medium sized vessels. In addition, retrofits are



❷ Heat balance example for a large two-stroke marine engine



❸ Development of turbocharger efficiency

❹ THS installed on a two-stroke engine

easily accommodated on older engines and engines equipped with multiple turbochargers. THS hence provides a ready opportunity for ship owners and operators to reduce fuel consumption and emissions. In addition, THS is not limited to feeding supplementary energy back into the engine crankshaft. Power taken off by the hydraulic pump may be used for the hydraulic functions of large engines, like fuel injection and exhaust valve actuation, or to drive generators for onboard electrical power.

PUMP DRIVE CONCEPTS TO INCREASE HYDRAULIC SYSTEM EFFICIENCY

Hydraulics can be used for a variety of applications on large engines where the inherent power density of hydrostatic power transmission is advantageous. Hydraulic energy is used to power cylinder liner lubrication, fuel injection and exhaust valve actuation. The design challenge is to find the application's optimal technical/economic balance, especially since component costs are an important driver. The equipment amortisation period has to be short to get rapid payback for the end-user.

Hydrostatic power transmission offers a wide spectrum of drive concepts, courtesy of an expansive array of components and modular variants. Configuration and component sizing play a crucial role in the final cost versus performance analysis of any drive concept.

A classic hydraulic drive consists of an electric motor and pump, generating hydrostatic energy for different engine functions. Large engine systems are constant pressure systems. Common variants are: fixed displacement pump with pressure relief valve; pressure-controlled variable displacement pumps; and fixed displacement pumps with variable speed motors.

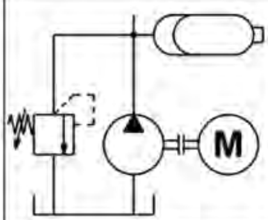
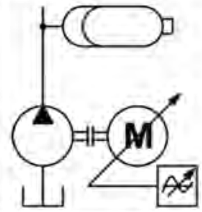
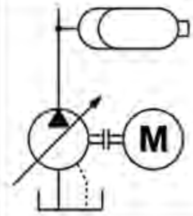
The evaluation matrix in ❼ compares the economic factors such as investment costs, energy efficiency, planning and capital expenditure. It also illustrates critical technical parameters like pressure consistency and zero pump flow. Energy efficiency is of crucial importance for life cycle cost assessment. Differing demands over the pump's complete duty cycle must be considered in order to assess total system flow requirements. Detailed cycle information is required to design an efficient pump system that is not overdimensioned. Drawing power out of the electric motor to satisfy the needs of the hydraulics when, and only when, actual work needs to be done is best accomplished by a variable speed (controlled) motor system.

Due to the nature of hydraulic systems, pressure drops occur (even in constant

❺ Hydraulic pumps installed at the turbocharger

❻ Hydraulic motor installed at the engine crankshaft

❼ Evaluation matrix for pump drives

Criteria	Fixed displacement pump with pressure relief valve	Fixed displacement pump with speed controlled electric motor	Pressure controlled variable displacement pump
			
Energy efficiency	--	++	-
System pressure stability	++	+	-
Noise	-	++	--
Design & comissioning effort	++	--	+
Investment costs with accumulator	++	-	+
without accumulator		-	+
Zero pump flow	yes	partially	yes
Drive design Motor-pump-group (MPG) PTO pump	MPG/PTO	MPG	MPG/PTO

pressure systems) due to actuators being energised and deactivated. Multiple actuators connected to the same pressure supply will naturally interact. As these actuators may control fuel injection timing, interaction must be kept to a minimum. The faster the system can make up for any loss in pressure, the better. The magnitude of the maximum permissible individual pressure drop is defined by the engine builder. A system's pressure stability is inversely proportional to the number of actuators that will be energised simultaneously and their distance from the pump.

To compare design and commissioning costs, a summation is needed of the engineering effort required to integrate single components into a complete and field serviceable pump system. The exact duty cycle profile, the maximum pressure/flow, system layout and service points and procedures must be known.

When evaluating hardware investment costs, the fixed displacement pump with pressure relief valve is the most cost effective option. It can be used as a baseline when comparing other drive concepts. The hardware costs for variable speed pump drives is approximately 60 % higher since an electric motor with frequency converter is required. An example is depicted in ❽.

FIXED DISPLACEMENT PUMPS

A fixed displacement pump and pressure relief valve system is also the simplest to design and service, as well as having the best pressure drop response characteristics. But these advantages can be outweighed by such a system's relative inefficiency. An analogy would be controlling a room's temperature by having the heating full on and opening the windows by varying amounts to keep the room from getting too hot. A variable displacement pump, on the other hand, is not as wasteful as a fixed pump and relief valve, but has inferior response characteristics to pressure drops.

A good illustration of drive concept evaluation would be the operational case of zero pump flow, when actuators have no demand for fluid. The pump's only task at these times is to maintain system pressure. For example, a fixed displacement pump would have to dump all the fluid coming out of the pump back to the tank, via the relief valve. This causes a great deal of fluid heating and the cooling system must be dimensioned to cope with this condition. Variable speed drives are the other extreme, where pump speed is reduced to almost zero, turning only fast enough to make up for leakages and to maintain pressure. A minimum amount of fluid heating occurs and the cooling system can be downsized accordingly.

VARIABLE DISPLACEMENT PUMPS

Variable displacement pumps provide only the amount of fluid needed to maintain constant pressure, but consume some energy even in a zero-flow operational case. However, they have the advantage of being able to be driven by the engine power-take-off (PTO) or by a separate electric motor. Motor pump units can be located nearer to the actuators or where it is most convenient. Naturally, care must always be taken to ensure the correct direction of pump rotation to avoid pump damage.

Other characteristics like noise emissions and hydraulic cooling loops are not discussed here, but these factors also could have an impact on the selection of the pump drive concept. The evaluation matrix shown in ⑦ gives the critical parameters needed for optimising hydraulic drives. However, detailed information and cost analysis of all operational modes are required to select the best pump drive.

Nomenclature

CO_2	Carbon Dioxide
EEDI	Energy Efficiency Design Index
IMO	International Maritime Organisation
MES	Mitsui Engineering and Shipbuilding
MPG	Motor-pump group
NO_X	Oxides of nitrogen
PTO	Power-take-off
SO_X	Oxides of sulphur
Sytronix	Speed variable pump drive
THS	Turbo Hydraulic System
WHR	Waste heat recovery

❽ Example of a speed variable pump (Sytronix)

ADDITIONAL POTENTIALS

Besides the two areas described, there are further hydraulic solutions which can increase overall engine efficiency. For example, variable hydrostatic fan drives offer an important option in four-stroke engine cooling. Compliance with emissions standards such as EPA Tier 4 has resulted in the need for more precise coolant temperature control, which requires increased cooling capacities. On its own, optimising engine combustion temperature is not enough to meet today's stricter emissions standards. Selective catalytic reduction, cooled exhaust gas recirculation, two-stage turbocharging and engine coolant temperature control strategies need to interact precisely to achieve optimum results. Hydrostatic fan drive systems are able to fill this niche, offering variable fan speed performance and the ability to pack more cooling capacity into confined spaces. The hydraulic pump draws only the power required by the fan. Hydrostatic fan drives represent a proven technology with demonstrated long service life in mobile engine applications.

CONCLUSIONS

Standard components and systems have been applied in configurations that increase the overall efficiency of large engines in attractive, innovative ways.

The THS system is a smart and compact hydraulic solution targeting reductions in fuel consumption and lowering emissions for small and medium sized vessels. Fuel savings of between 3 and 4 % are achievable according to engine load. THS can also be retrofitted on installed large engines.

The application of hydraulics for engine functions is not limited to any one single drive concept. Investigating different approaches often reveals possibilities to increase system efficiencies. Variable speed pump drives provide energy efficient solutions over a wide range of configurations, can create efficient new systems and are be readily retrofitted to modernise older hydraulic circuits.

THANKS

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