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## EFFECTS OF HUMIDITY AND AMBIENT TEMPERATURE ON ENGINE PERFORMANCE OF LEAN BURN NATURAL GAS ENGINES

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### ABSTRACT

High demands are placed on large gas engines in the areas of performance, fuel consumption and emissions. In order to meet all these demands, it is necessary to operate the engine in its optimal range. At high engine loads the optimal operation range becomes narrower as the engine comes closer to the knocking or to the misfire limit. The ambient conditions are of increasing importance in this range of operation.

Variations in humidity influence the engine's burn rate characteristics. An increase in humidity reduces the burn rate and increases the combustion duration. This increase in combustion duration has the same effect as retarding the time of ignition. Thus the thermal efficiency is reduced. Additionally, the engine is more likely to misfire as humidity increases. The cylinder temperature affects the engine fuel efficiency, knocking, exhaust gas temperature and particularly  $\text{NO}_x$  emission. An increase in manifold air temperature results in higher  $\text{NO}_x$  emission, heat transfer and knocking tendency. To avoid knocking, the time of ignition must be retarded resulting in lower engine efficiency.

In this paper the effects of changes in humidity and temperature of the intake air on engine performance were examined in a lean burn pre-chamber natural gas engine. Tests on a single cylinder research engine were carried out. Effects on knocking and misfire limit,  $\text{NO}_x$  emissions and fuel consumption were investigated depending on engine load. The interpretation of the results was supported by an extended analysis of losses.

### INTRODUCTION

The desire for a high degree of efficiency, low emission levels and high power output puts great demands on modern IC engines. In addition, reliability, durability and long service intervals are of great importance. In order to fulfill all the requirements the gas engine has experienced a renaissance within the last few years.

It is generally known that the specific humidity and the temperature of the intake air significantly affect the behavior of gas engines. To operate the engine in an optimal range it would be necessary to adapt the engine control settings to respective ambient conditions. The specific humidity of the intake air depends on the relative humidity, temperature and pressure of the ambient air. Of course, the level of humidity varies geographically and throughout the year.

This paper describes the effects of changes specific humidity has on engine performance of a lean burn natural gas engine. In particular, the influence on engine efficiency, emissions, air fuel ratio, knocking tendency and misfire limit was investigated by varying specific humidity of the intake air. At high engine loads (high boost pressures) and low intake air temperatures condensation may occur, leading to corrosion in the intake system. The influence of increased intake air temperatures (in order to avoid condensation) on engine performance is also shown.

### TEST PROCEDURE

#### TEST ENGINE

The experimental investigations were carried out on a natural gas fueled single cylinder research engine. For lean burn operation a gas-scavenged pre-chamber concept was used (Fig. 1). Therein charge stratification is achieved by the feeding of pure gas into the pre-chamber via a pressure-controlled valve. Due to this a significantly richer and easily ignitable mixture with high energy is available for igniting the lean mixture in the main combustion chamber.

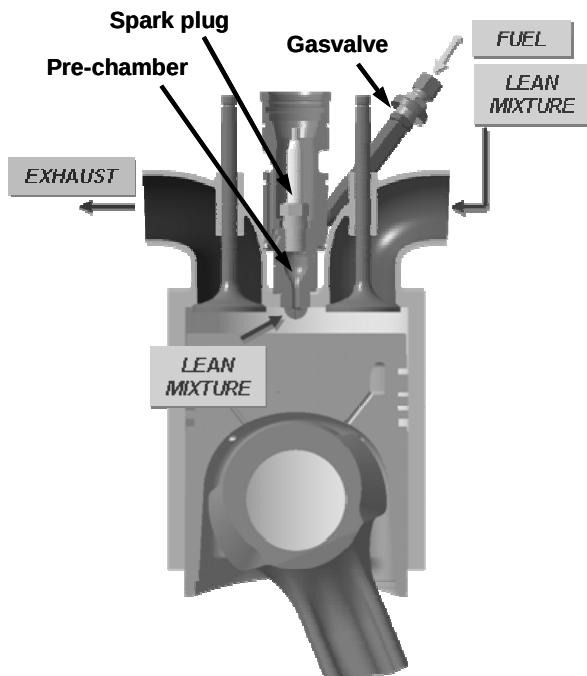


Fig. 1. Gas-scavenged Pre-chamber Concept [2]

The most important data of the single cylinder research engine (SCRE) and the corresponding full-scale engine (FSE) are summarized in Table 1.

|                                   | SCRE                      | FSE          |
|-----------------------------------|---------------------------|--------------|
| Number of cylinders               | 1                         | 12, 16 or 20 |
| Bore / Stroke [mm]                | 190 / 220                 |              |
| Displacement [dm <sup>3</sup> ]   | 6.2 / Cylinder            |              |
| Engine type                       | 4-stroke                  |              |
| Engine speed [min <sup>-1</sup> ] | 1500                      |              |
| Number of valves                  | 4                         |              |
| Pre-chamber volume                | ≈1-2% of clearance volume |              |

Table 1. Engine data

The single cylinder research engine was adapted to the intake and exhaust gas dynamics as well as the residual gas content of the full-scale engine using a one-dimensional gas exchange simulation program.

## EXPERIMENTAL SETUP

The set-up of the test bed used for these experiments is shown in Fig. 2. The intake air is compressed with a rotary screw compressor (C) including an integrated refrigerant dryer. After the dryer, the intake air is heated (electric heater) to the desired temperature. Pressurized water vapor is injected after the heater to avoid condensation in the pipe system. An electronically throttle (BC) is used to control boost pressure.

The conditioned (pressure, temperature and humidity) air and compressed natural gas (CNG) are mixed in a mixing device (Air/Gas Mixer). The humidity sensor is placed between the engine and the mixing device. The output signal of the humidity sensor corresponds to relative humidity. Based on this value the specific humidity is calculated by the data acquisition system in real time with the values of pressure and temperature of the intake air.

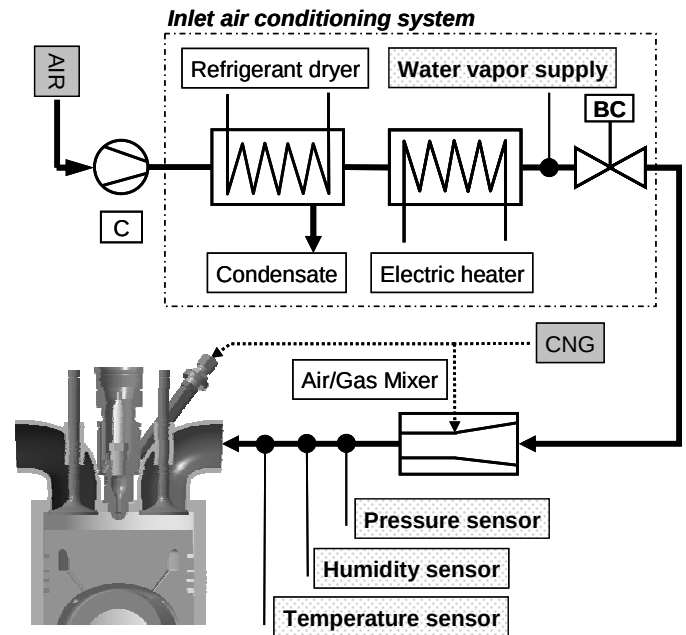


Fig. 2. Schematic of Intake Air Conditioning System for Humidity and Temperature Control

Specific humidity is varied by controlling the flow of water vapor via an electronic valve. The engine is instrumented to measure all relevant temperatures, flows and pressures, as well as cylinder pressure and emissions.

## TEST CONDITIONS

The tests were carried out at an emission level according to the *Technischen Anweisung* (Technical Instructions or TA-Luft; see Table 2).

| Exhaust gas component | TA-Luft limits                     |
|-----------------------|------------------------------------|
| NO <sub>x</sub>       | 500 mg/m <sub>n</sub> <sup>3</sup> |
| CO                    | 650 mg/m <sub>n</sub> <sup>3</sup> |
| Non-methane HC        | 150 mg/m <sub>n</sub> <sup>3</sup> |

Table 2. Emission Limits According to TA-Luft, referenced to 5% O<sub>2</sub> in the Exhaust Gas

For different operation points (defined by ignition timing, boost pressure, exhaust back pressure and air/fuel ratio) and specific humidity was varied between about 1 g/kg and 26

g/kg. In case relative humidity of the compressed intake air gets higher than 95%, the inlet air temperature is increased to a value which prevents condensation. The standard value for the intake air temperature was 50°C. For the tests with constant inlet air temperature, the intake air temperature was set to 60°C. The exhaust back pressure was adapted for each operation point corresponding to the full-scale engine.

Due to differences in friction between full-scale engine and single cylinder research engine indicated mean effective pressure (IMEP) was used to define the engine load. In general, the range of the IMEP was between 20 and 26 bar (corresponds to 18.5 and 24.5 bar BMEP). To assess the effects of humidity and inlet air temperature on the combustion process separately the engine was operated at an IMEP of 22 bar.

In order to analyze the test results, important engine parameters are shown against the specific humidity of the intake air. For a detailed interpretation of the results extended analyses of losses were carried out (compare also Fig. 3).

### ANALYSES OF LOSSES

An analysis of respective partial losses allows an allocation of losses to different originators [4]. The partial losses considered within this study are defined as follows and shown in Fig. 3:

$$\eta_i = \eta_{IRC} - \Delta\eta_{IC} - \Delta\eta_{RC} - \Delta\eta_{WH} - \Delta\eta_{GE}$$

- $\eta_i$ .....Indicated efficiency
- $\eta_{IRC}$ .....Efficiency of ideal engine with real charge
- $\Delta\eta_{IC}$ .....Losses due to incomplete combustion
- $\Delta\eta_{RC}$ .....Losses due to real combustion
- $\Delta\eta_{WH}$ .....Wall heat losses
- $\Delta\eta_{GE}$ .....Gas exchange losses

In order to determine the losses, a definition of the ideal operating process is necessary. The ideal engine process is defined as constant volume combustion with external mixture formation.

The losses caused by incomplete combustion are determined using an exhaust gas analysis. The losses due to real combustion take into account the difference of the rate of heat release compared to the ideal constant volume combustion at TDC.

Wall heat losses comprise a major element of total losses and can be calculated as the difference between efficiency with real combustion and efficiency of high pressure cycle. They depend mainly on temperature, pressure, turbulence and velocity near the combustion chamber walls.

The gas exchange losses are determined as the difference between the efficiency of the high-pressure cycle and the efficiency of the overall cycle. All measurements are

conducted on a single cylinder engine. Thus the results concerning gas exchange losses are not directly transferable to multi-cylinder engines.

Losses caused by the flow between pre-chamber and main combustion chamber and losses due to leakage are not considered in detail.

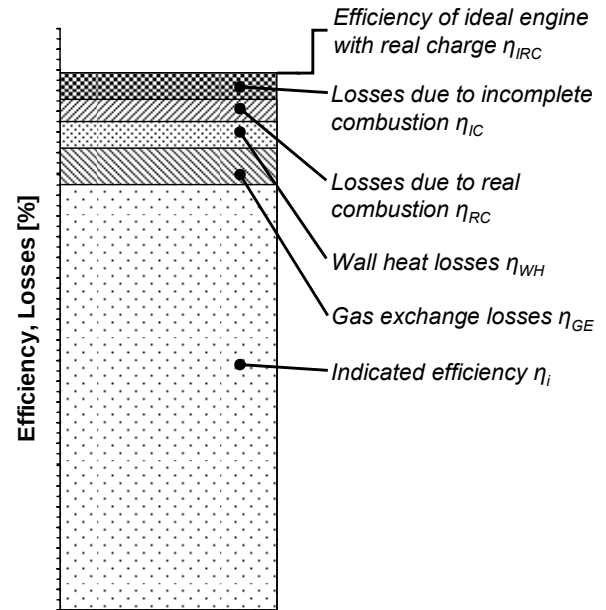


Fig. 3. Analyses of Losses

### EFFECTS OF AMBIENT CONDITION TO ENGINE PERFORMANCE AND EMISSIONS

The content of water (specific humidity) in the intake air is defined by the relative humidity, temperature and pressure.

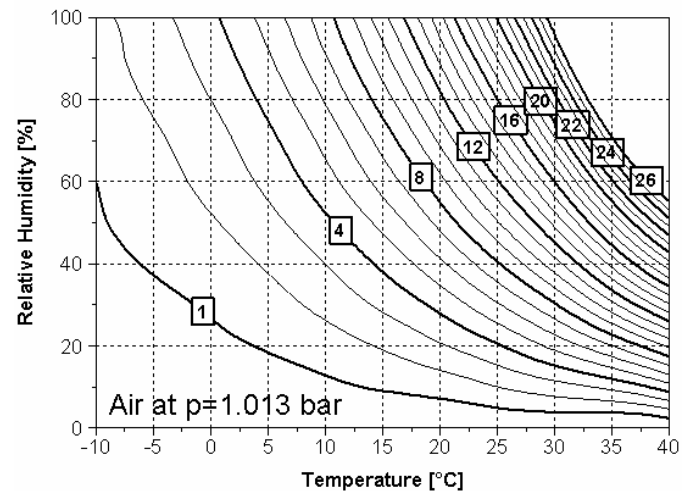


Fig. 4. Specific Humidity [g/kg] at 1.013 bar

In Fig. 4, the correlation between relative humidity, temperature and specific humidity is shown for the relevant range.

The values of specific humidity vary daily and throughout the year dependent on the ambient conditions and on the installation site of the engine.

## EFFECTS WITHOUT COMPENSATION

It is common knowledge that increased humidity levels alter the burn rate characteristics lead to lower  $\text{NO}_x$  emissions [1]. Increased spark delay and slower combustion as well as the increase in specific heat of the trapped charge due to the presence of increased levels of  $\text{H}_2\text{O}$  lead to lower  $\text{NO}_x$  emissions. Standard engine control systems which have no real-time information about  $\text{NO}_x$  emissions or humidity of the intake air are not able to compensate for changes in ambient conditions.

The first tests were carried out in order to assess the effect of variable specific humidity to  $\text{NO}_x$  production and to the rate of heat release without compensation by air/fuel ratio. The results of these measurements are shown in Fig. 5.

Throughout the paper the results are depicted relative to the values of standard specific humidity value of 8 g/kg.

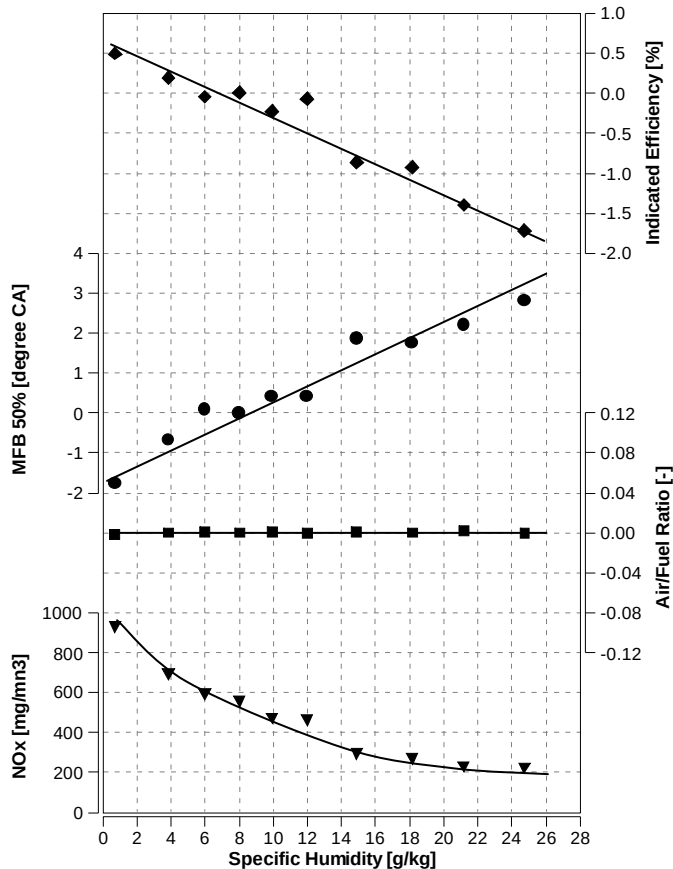


Fig. 5. Emission (@ 5%  $\text{O}_2$ ) and Efficiency at Constant Air/fuel ratio

The SCRE was operated at a constant speed of 1500 rpm and an IMEP of 22 bar (IMEP calculated from the measured cylinder pressure). Spark timing, air/fuel ratio and intake air temperature were held constant by the control system. Furthermore, IMEP was held constant by adapting boost pressure. Specific humidity was changed in the range from 1 to 26 g/kg. The intake air temperature was set to 60°C in order to prevent condensation at high humidity and pressure levels.

Starting from nearly dry conditions (0.8 g/kg) up to high humidity levels (24.8 g/kg),  $\text{NO}_x$  emissions decreased by approximately 75% due to slower combustion. The MFB 50% was retarded of about 4 degrees CA (see Fig. 5). As a result, the indicated efficiency was reduced by 2.2 %.

The detailed analysis of losses depending on humidity level is shown in Fig. 6. The efficiency of the ideal engine, essentially determined by the compression ratio and air/fuel ratio, is decreased by 0.2% due to higher boost pressure (about 0.4 bar) required for constant IMEP at high humidity levels. Losses due to incomplete combustion increase from 1.2% to 1.8%, caused by increased THC emissions of about 40%. In this case the loss of indicated efficiency is mainly caused by increasing the losses due to real combustion from 3.3% to 4.6%. The effect of increased humidity on the rate of heat release is shown in Fig. 7. It can be seen that due to rising humidity, the center of gravity of the rate of heat release (ROHR) is delayed (about 5 degree crank angle). Spark delay was not affected.

In contrast to the losses due to real combustion, wall heat losses due to delayed combustion are decreased by 0.4% during the tested humidity range. The decrease in indicated efficiency can be explained by the described specific losses.

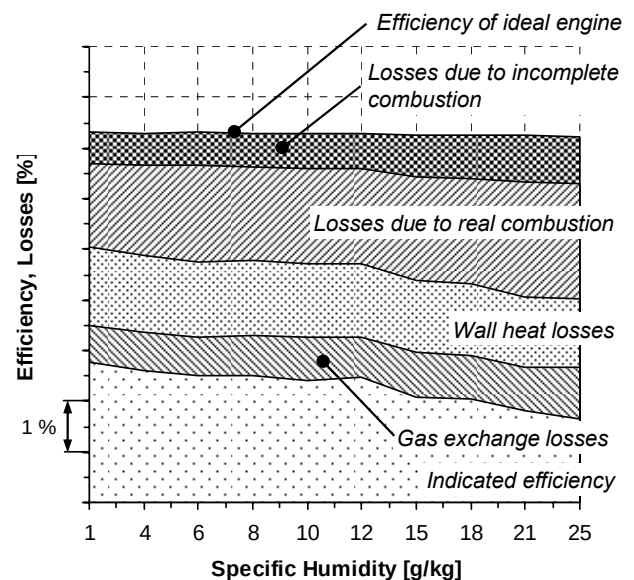


Fig. 6. Analysis of Losses against Specific Humidity

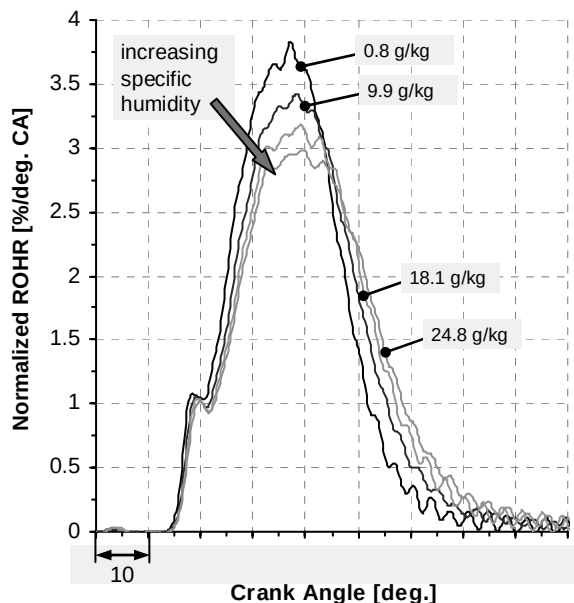


Fig. 7. Normalized Rate of Heat Release

Tests on the single cylinder research engine were performed with an exhaust gas pressure calculated under the assumption of constant turbo charger efficiency. In order to estimate the influence of the changes in ROHR caused by variable humidity on the gas exchange efficiency, one-dimensional engine cycle calculations were carried out. Fig. 8 shows a good correlation between the calculated and the measured indicated efficiency. From this the conclusion can be drawn that the efficiency of the gas exchange is hardly affected. This is also confirmed by Fig. 9 and Fig. 10 which show the efficiency of the turbine and the compressor respectively.

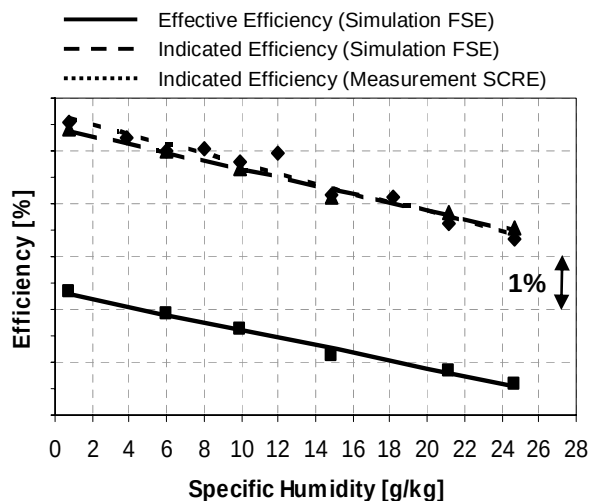


Fig. 8. Efficiency Simulation FSE and Measurement Result SCRE

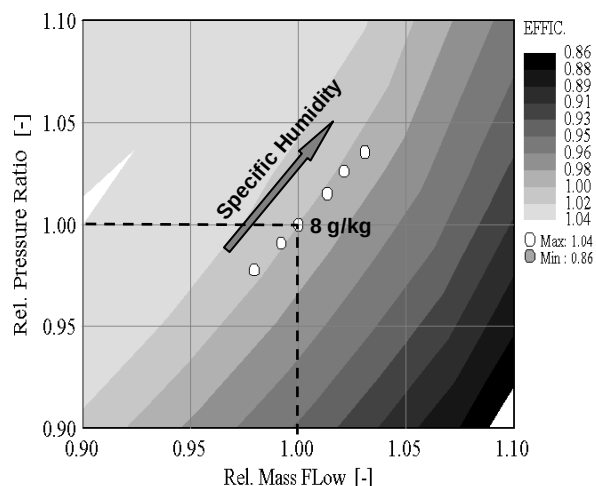


Fig. 9. Compressor Efficiency Map: Working Section

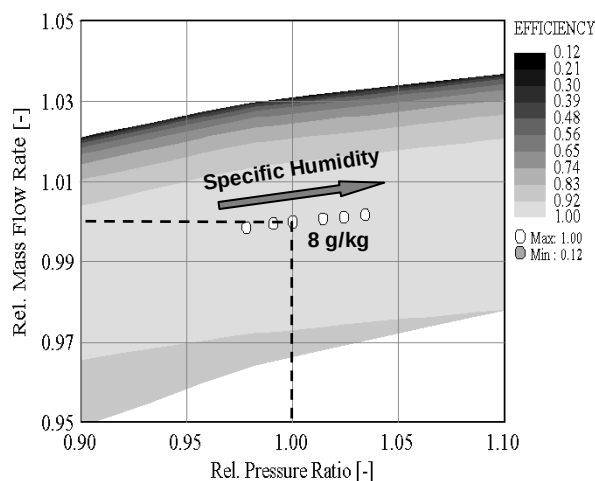


Fig. 10. Turbine Efficiency Map: Working Section

In order to ensure optimal engine performance, it is necessary to optimize the engine control settings based on the changes in ambient conditions (ambient relative humidity and temperature). Subsequently two strategies are described.

#### NO<sub>x</sub> COMPENSATION BY ADAPTION OF AIR/FUEL RATIO

One strategy to eliminate the decrease in NO<sub>x</sub> emission due to rising humidity levels is to adapt the air/fuel ratio in such a way that the NO<sub>x</sub> emissions will match the prescriptive emission limit. The feedback signal to vary the air/fuel ratio was the measured NO<sub>x</sub> emissions. These tests were done with the same engine configuration (compression ratio, piston shape, spark plug type, etc.) and at the same operating point as the tests without NO<sub>x</sub> compensation. Spark timing and intake air temperature were held constant. The air/fuel ratio was adapted by the control system to operate the

engine at a constant level of  $500 \text{ mg/m}_n^3 \text{ NO}_x$  emission @ 5%  $\text{O}_2$ . The results are shown in Fig. 11.

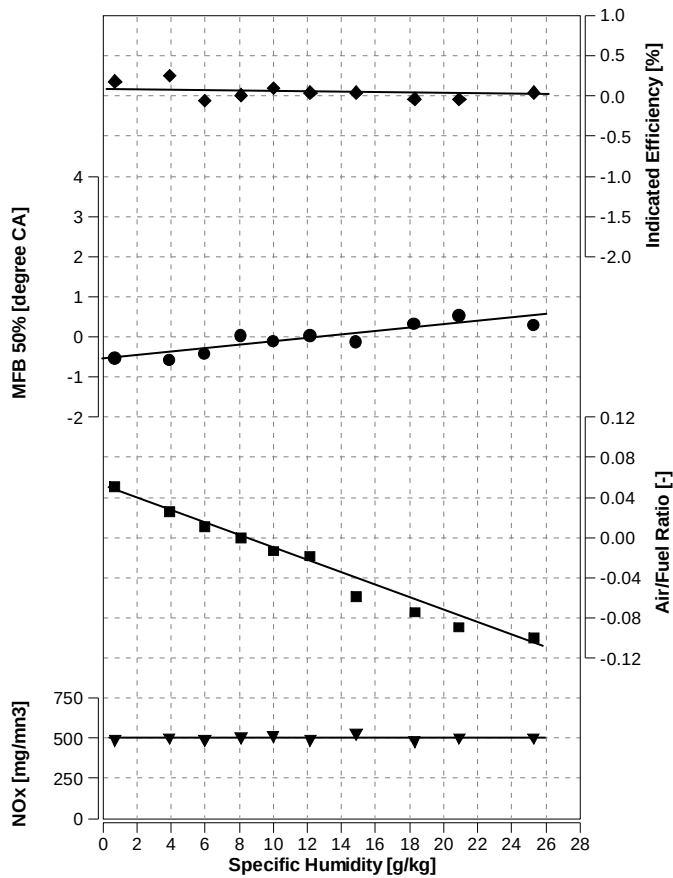


Fig. 11.  $\text{NO}_x$ =constant (@ 5%  $\text{O}_2$ ) and Air/Fuel Ratio=variable

Starting with a specific humidity of  $0.7 \text{ g/kg}$  to a value of  $25.3 \text{ g/kg}$ , an adaptation of the air/fuel ratio of approximately 0.15 was required to hold MFB 50% point and thus  $\text{NO}_x$  emissions constant. The reduction of about 0.6% in the efficiency of the ideal engine, see Fig. 12, is caused by the lower air/fuel ratio values at high humidity levels. Losses due to incomplete combustion, real combustion and wall heat losses are nearly unaffected. The lower efficiency of the ideal engine is compensated by lower losses due to real gas exchange because the required boost pressure for constant IMEP is less at lower air/fuel ratio values.

All in all, this strategy enables a constant indicated efficiency level over the examined specific humidity range.

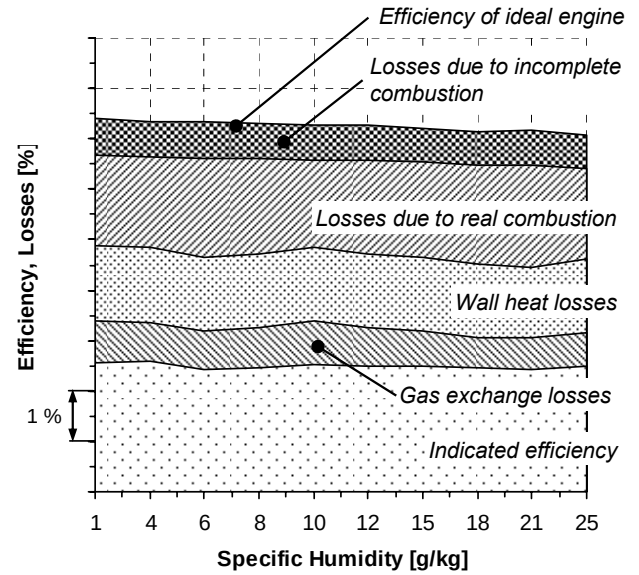


Fig. 12. Analysis of Losses against Specific Humidity

Fig. 13 shows the rates of heat release depending on specific humidity. Only slight differences can be seen.

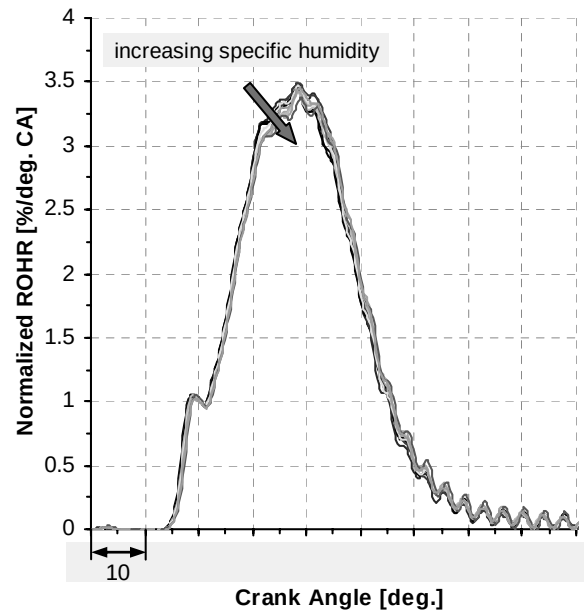


Fig. 13. Normalized Rate of Heat Release (IMEP=22 bar)

The measurements show that there is an approximately linear correlation between specific humidity and air/fuel ratio (see Fig. 11). An increase of  $4 \text{ g/kg}$  in specific humidity requires a reduction of the air/fuel ratio of 0.025.

#### $\text{NO}_x$ COMPENSATION BY ADAPTION OF SPARK TIME

The second strategy investigated was to adapt spark timing in such a way that the  $\text{NO}_x$  emissions will match 500

$\text{mg}/\text{m}_n^3$  @ 5%  $\text{O}_2$  at a constant air/fuel ratio. The results are shown in Fig. 14.

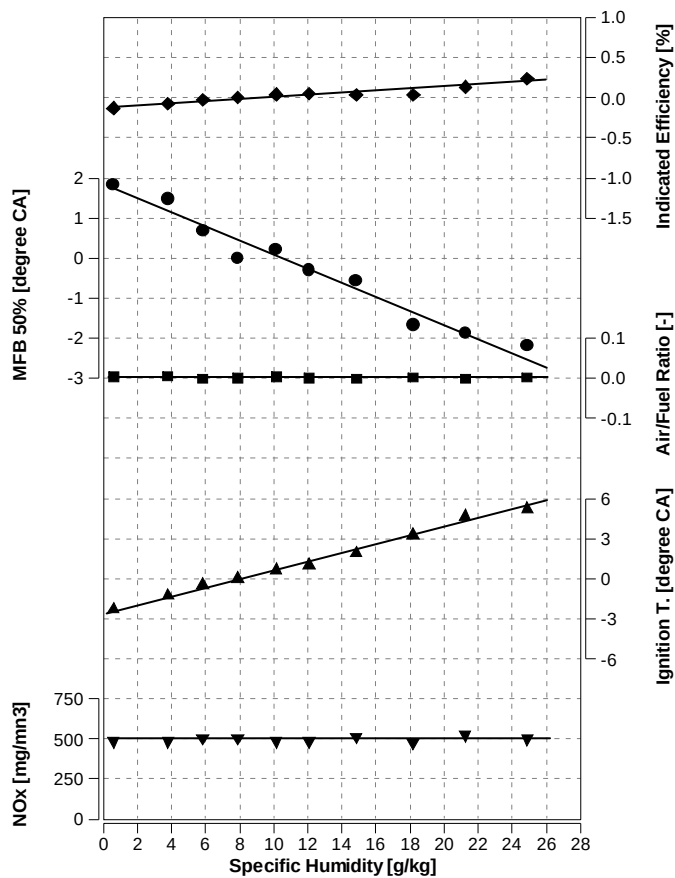


Fig. 14.  $\text{NO}_x$ =constant (@ 5%  $\text{O}_2$ ) and Ignition Timing=variable

Earlier spark timings result in higher combustion temperatures leading to a lower combustion duration and compensation of reduced  $\text{NO}_x$  emission. MFB 50% point requires advancing the timing 4 degrees CA. The analysis of losses is shown in Fig. 15.

Due to the constant air/fuel ratio level the efficiency of the ideal engine is not significantly influenced by increasing the humidity. The reduction of about 0.2% at maximum humidity is caused by the higher boost pressure required for constant IMEP. Losses due to incomplete combustion and losses due to real gas exchange are almost unaffected by humidity. The benefit in indicated efficiency of 0.4% at maximum humidity is caused by reduced losses due to real combustion. Fig. 16 shows the corresponding rates of heat release.

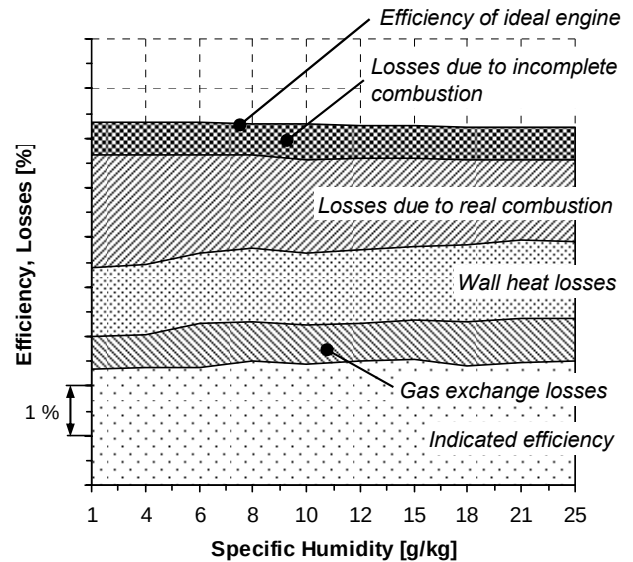


Fig. 15. Analysis of Losses against Specific Humidity

Similar to strategy one, there is an approximately linear correlation between specific humidity and spark time (see Fig. 14). An increase of 2 g/kg in specific humidity requires an advance of spark timing by 1 degree CA.

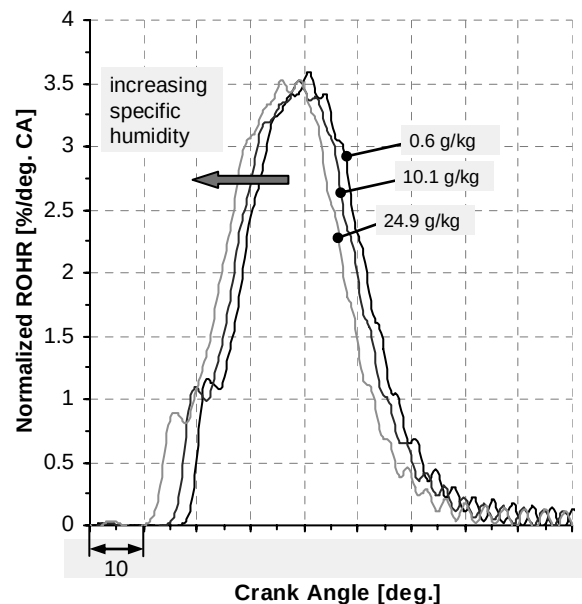


Fig. 16. Normalized Rate of Heat Release

#### EFFECT ON KNOCKING LIMIT WITH COMPENSATION OF CONDENSATION BY INCREASING INTAKE AIR TEMPERATURE

A very important parameter for gaseous fuelled engines is the knocking limit. For the assessment of the effect of changes in specific humidity on knocking, tests were carried

out at a constant speed of 1500 rpm, engine loads of 22, 24 and 26 bar IMEP at 500 mg/m<sup>3</sup> NO<sub>x</sub> emission @ 5% O<sub>2</sub>. The results are shown in Fig. 17.

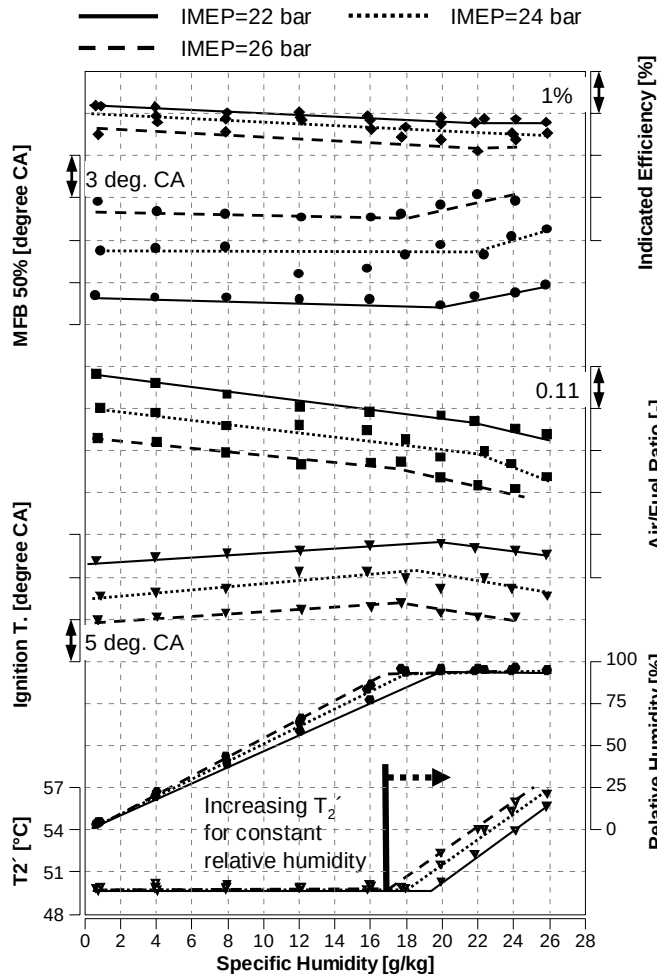


Fig. 17. Knocking Limit at 500 mg/m<sup>3</sup> NO<sub>x</sub> Emission @ 5% O<sub>2</sub>

Spark timing was advanced at different levels of specific humidity until knocking occurred. Measurements were done in such a way that the spark timing was set with a gap of one degree to the knocking spark time. The intake air temperature was set to 50°C. If the value of relative humidity in the intake system (sensor position see Fig. 2) rises over 95% (condensation limit) intake air temperature was decreased to hold relative humidity constant at 95%.

As described previously, the results in Fig. 17 show that higher specific humidity values requires lower air/fuel ratios to compensate for NO<sub>x</sub> emissions. In addition, the knocking tendency is decreased with higher specific humidity values. This effect can be exploited in order to operate the engine with earlier spark timings. Depending on engine load, (defined by boost pressure) the limit of specific humidity which affects 95% relative humidity at 50°C intake air temperature is

in the range of 18 to 20 g/kg. At specific humidity values lower than the condensation limit (relative humidity 95%) the combustion (MFB 50%) is not affected essentially due to adapting air/fuel ratio and spark time. If the intake air temperature has to be increased to prevent condensation, the knocking tendency rises. Thus spark timing has to be retarded. The results shows that higher intake air temperatures increase the knock tendency more than higher humidity levels decreases knock tendency.

In Fig. 18 the effect of rising humidity and increased intake air temperature to the combustion losses are shown.

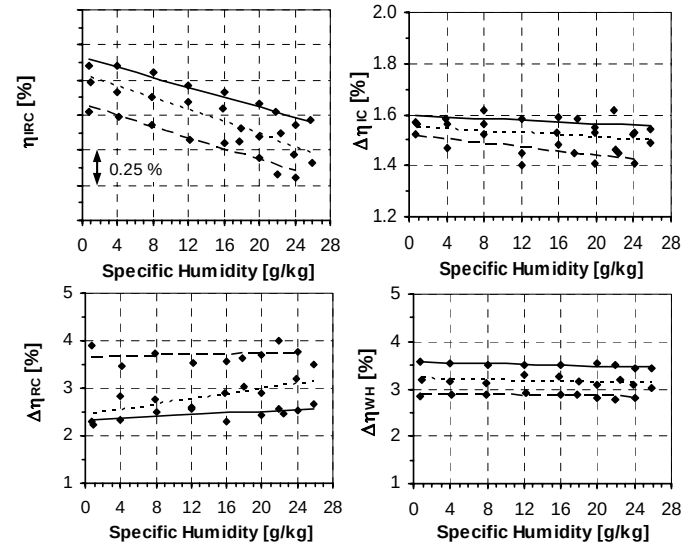


Fig. 18. Analysis of Losses against Specific Humidity

The efficiency of the ideal engine, essentially determined by air/fuel ratio, is decreased about 0.5% due to required lower air/fuel ratio values at high specific humidity levels. Losses due to incomplete combustion, due to real combustion, wall heat losses and losses due to real gas exchange are nearly constant over the examined specific humidity values. The indicated efficiency decreases about 0.5% from dry conditions to maximum tested humidity.

#### RANGE OF ENGINE OPERATION (MISFIRE LIMIT)

The experimental setup was also used to assess the operation range of the engine. The limits for the operation range were 500 mg/m<sup>3</sup> NO<sub>x</sub> emission @ 5% O<sub>2</sub> and the highest possible air/fuel ratio (misfire limit). Both limits were determined depending on specific humidity. Engine speed, spark time and boost pressure were held constant. The tests were done at an IMEP of 22 bar. The criterion for the misfire limit was COV ≥ 5 %. The results are shown in Fig. 19.

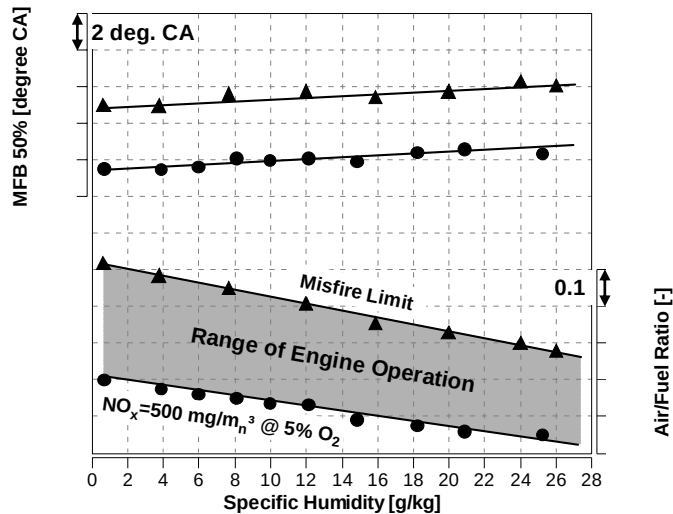


Fig. 19. Range of Operation Against Specific Humidity (IMEP=22 bar)

The characteristic of the air/fuel ratio is comparable to Fig. 7. The maximum air/fuel ratio for stable combustion is about 0.3 higher than that for 500 mg/m<sup>3</sup> NO<sub>x</sub> @ 5% O<sub>2</sub> emission. There is no essential influence to the burn rate (ROHR) in both cases due to NO<sub>x</sub> compensation from adapting the air/fuel ratio. The MFB 50% value increases about 1 degree CA, varying specific humidity from 1 g/kg to 26 g/kg. There is a parallel trend for MFB 50% at the boundaries of the engine operation range.

## CONCLUSION

To assess the influence of ambient conditions on engine performance, specific humidity was varied in a range of about 1 g/kg to 26 g/kg. Intake air temperature was increased to avoid condensation at high engine loads (high boost pressure) and high humidity levels.

Basically, increasing specific humidity levels affect the combustion process in such a way that the burn rate is retarded. Measurements at constant IMEP and constant air/fuel ratio show a quadratic correlation between NO<sub>x</sub> emission and specific humidity. NO<sub>x</sub> emissions decrease about 60% when increasing specific humidity from 8 g/kg to 25 g/kg. The losses due to incomplete combustion and the losses due to real combustion caused by the retarded burn rate are increased. This is also the reason for the decreasing indicated efficiency of 1.7% at specific humidity 25g/kg. One-dimensional engine cycle calculations confirmed that the effects on the efficiency of the gas exchange due to shifting of the operating point of the turbo charger are circumstantial in the present case.

To compensate for the reduction of NO<sub>x</sub> emission and to avoid disadvantages in efficiency, two strategies were examined: Adaptation of the air/fuel ratio and adaptation of spark time. The results show that the air/fuel ratio has to be decreased by about 0.1 to compensate an increase of specific

humidity from 8 g/kg to 25 g/kg. The adaptation of ignition timing at constant air/fuel ratio enables constant NO<sub>x</sub> emissions and simultaneous shifting of the burn rate in optimal range. In this case the indicated efficiency was increased by 0.4% due to a reduction of combustion losses.

There is a linear correlation between air/fuel ratio and spark time respectively to compensate for changes in specific humidity in both strategies. Increased humidity results in a decreased knocking tendency. Only slight effects on the operation range (misfire limit) could be seen.

## NOMENCLATURE

|                   |                                             |
|-------------------|---------------------------------------------|
| SCRE              | single-cylinder research engine             |
| FSE               | full-scale engine                           |
| $\eta_I$          | indicated efficiency                        |
| $\eta_{IRC}$      | efficiency of ideal engine with real charge |
| $\Delta\eta_{IC}$ | losses due to incomplete combustion         |
| $\Delta\eta_{RC}$ | losses due to real combustion               |
| $\Delta\eta_{WH}$ | wall heat losses                            |
| $\Delta\eta_{GE}$ | gas exchange losses                         |
| MFB 50%           | mass fraction burnt 50%                     |
| ROHR              | rate of heat release                        |
| TDC               | top dead center                             |
| BC                | boost controller                            |
| C                 | compressor                                  |
| CNG               | compressed natural gas                      |
| IMEP              | indicated mean effective pressure           |
| NO <sub>x</sub>   | nitrogen oxides                             |
| O <sub>2</sub>    | oxygen                                      |
| CO                | carbon monoxide                             |
| HC                | hydrocarbon                                 |
| THC               | total hydrocarbon                           |
| H <sub>2</sub> O  | water                                       |
| T2'               | intake air temperature                      |
| CA                | crank angle                                 |
| deg.              | degree                                      |

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