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DEVELOPMENT OF A RECIPROCATING COMPRESSOR USING WATER INJECTION TO ACHIEVE QUASI-ISOTHERMAL COMPRESSION

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ABSTRACT

A novel kind of reciprocating air compressor is being developed, in which quasi-isothermal compression is achieved by the injection of a large quantity of water through spray nozzles inside the compressor. The compressor is mainly intended as a part of a new thermodynamic cycle for efficient power generation, but it could be used in stand-alone applications as well. Due to the high density and heat capacity of the liquid water spray, the temperature increases only slightly during compression. The water is not consumed but separated from the pressurized air, cooled and re-used, offering options for heat recovery. A compression ratio of up to 1:30 can be realized in a single cylinder. Results from a test rig and a full-scale compressor prototype are presented.

INTRODUCTION

Compression of air and other gases to high pressures is commonly performed by the use of multi-stage reciprocating compressors with interstage cooling, which limits the gas temperature at each stage and reduces the compression work. The low gas temperature also permits the use of low temperature materials for sealing etc. The equipment is relatively expensive in view of the number of cylinders, valves, heat exchangers and pulsation dampers, which are required in a multi-stage unit. The efficiency is limited by the need to balance the theoretical benefit of multiple cooling stages against the practical effect of pressure losses and heat exchanger temperature differences.

A different kind of reciprocating compressor is being developed, in which quasi-isothermal compression is achieved by the pulsed injection of a large quantity of water through multiple spray nozzles inside the compressor. The atomized water is not evaporated but heated up and discharged with the air through valves in the cylinder head. Water and air are separated downstream. Discharge fluid temperatures are low because of the cooling effect of the injected water. The separated liquid water is cooled and then re-injected into the cylinder. The dry air from the separator is available directly as process air, since the separator not only removes condensed water (from the humidity of the intake air) together with the injected water, but also acts as a pulsation damper. The main advantage of this type of compressor is that high pressure ratios (up to about 30) can be achieved in a single stage with a significant reduction in the power consumption. It is expected that the equipment cost will also be reduced compared to state-of-the-art compressors. Additionally, the compressed gas is never in contact with lubrication oil since all cylinder lubrication is provided by water.

This new type of compressor, called “isocompressor”, is primarily intended to be used for a novel kind of large engine, the isoengine, see Coney et al. (2002a, 2002b). However, stand-alone applications for isocompressors appear to be beneficial over existing technology under certain conditions, depending e.g. on flow rate, inlet pressure, compression ratio and chemical properties of the gases to be compressed.

Further Novel Aspects and Benefits

The isocompressor has a new kind of discharge valve described in detail below that allows a safe and reliable discharge of the air-water mixture from the cylinder. Furthermore, the spray water serves as a lubricant, such that relatively high piston speeds can be achieved without oil lubrication. The injected water also functions as an "extra piston", filling crevice volumes and yielding a good volumetric efficiency. Recovery of the low-grade heat of compression becomes more attractive, since the heat is rejected from only one source (the spray water) and not from multiple intercoolers. Additionally, a single water/water heat exchanger can be designed to be more efficient (i.e. with a low driving temperature difference) than multiple air/water heat exchangers.

PHYSICAL MECHANISMS

The effect of heat transfer from the air to water droplets may be represented by a modified form of the well-known relationship for an ideal gas undergoing isentropic compression or expansion. The modified equation may be derived from the First Law of Thermodynamics in a similar way to the original equation, yielding

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\left(1-\frac{1}{\gamma}\right)} \exp\left[-\frac{ua}{c_p}\left(1-\frac{T_w}{T_1}\right)\Delta t\right] \qquad (1)$$

Equation (1) relates the change of air temperature and pressure during a time interval Δt, with the exponential function representing the effect of heat transfer to the droplets. The indices 1 and 2 denote the beginning and end of this short interval. T_w is the mean spray water temperature, c_p is the specific heat of the air in the cylinder, γ is the ratio of specific heats, u is the heat transfer coefficient between the air and the water droplets and a is the droplet surface area per kg of air. If $ua\,\Delta t$ is near zero then the exponential function tends to unity and the temperature ratio tends to the unmodified isentropic relationship. On the other hand if $ua\,\Delta t$ is large then T_2 rapidly converges on T_w. Since the parameters of the exponential function depend on time and may undergo rapid changes during compression, the above expression should only be applied to small time intervals and small pressure ratios.

The term c_p / ua can be interpreted as a thermal time constant τ for transient heat transfer, see e.g. Incropera and De Witt (1990, p. 228). The time constant τ represents the time necessary to reduce the temperature difference between air and water by the factor 1/e, which is a reduction of 63.2%.

Measured data allow the calculation of the value of ua as a function of crank angle during the compression stroke. A theoretical approach can be taken to derive estimates for ua/c_p (i.e. the reciprocal of the time constant τ) using a correlation for droplet heat transfer. A correlation by Ranz and Marshall (1952) indicates that for air the Nusselt Number is equal to 2 + 0.53 times the square root of the droplet Reynolds Number. This suggests the relationship

$$\frac{1}{\tau} \approx 6\left[2+0.53\left(\frac{\rho_g\, v_d\, d}{\mu_g}\right)^{0.5}\right]\frac{k_g}{\rho_l\, c_p\, d^2}\,\frac{m_l}{m_g} \qquad (2)$$

where k_g, ρ_g, μ_g are thermal conductivity, density and viscosity of the air, ρ_l is the density of water, d is droplet diameter, v_d is the droplet velocity relative to the gas, m_l is the mass of injected water and m_g is the mass of air. However, this equation assumes a uniform distribution of droplets, which is difficult to achieve in practice. A reasonable estimate of droplet sizes can be obtained by measurements on nozzles operating under similar conditions, but droplet velocities are a strong function of time and position within the compressor. Typically the droplet velocities are high immediately after injection, but the drag of the air causes a rapid deceleration of the droplets, which in turn causes a reduction in the rate of heat transfer. Typically the mass of the injected water is around 3 times the mass of the air, and with injection velocities of around 30 m/s, there is a substantial amount of momentum transfer from the water to the air. The motion of the compressor piston also affects the air motion, which has an effect on the motion of the droplets.

To understand the order of magnitude of the time constant τ, values may be inserted appropriate to compression of air to 30 bars with a droplet diameter of 75 μm, a constant water/air mass ratio of 3 and assuming a relative velocity of 0.1 m/s. This indicates that $1/\tau$ increases from about 200 s^{-1} to 300 s^{-1} as the pressure increases. Normally, however, water is injected continuously throughout the compression so that the water/air mass ratio is low at first but increases with the pressure. This factor and the non-uniform distribution of water droplets at the start of the compression means that in practice $1/\tau$ generally starts at a lower value but increases significantly as the water content increases. The intended operating speed of the compressor is 600 rpm. If $1/\tau \sim 300\ s^{-1}$, then $\tau \sim 3.3$ ms corresponds to a crank angle movement of 12° at 600 rpm. This is low enough to give a significant reduction in the compression work at 600 rpm. However, if a lower value of τ can be achieved it would be beneficial.

Optimization of Water Injection to Maximize Heat Transfer

The optimization of the water injection into a cylinder with a moving piston involves consideration of several factors, and particularly the limited time for penetration of the droplets into the cylinder. Introducing water spray early on in the compression, or even during the down-stroke, gives more time for injection and dispersion of the droplets. However, most of the compression work (and hence heat generation) occurs at the end of the compression stroke. If all the water is injected early in the compression stroke, the droplets are slowed to a low velocity by the time that a high heat transfer rate is most needed. On the other hand, since the density of the air rises during compression, droplets, which are injected late in the process, have difficulty in penetrating to the center of the cylinder. To overcome this, nozzles can be selected, which direct droplets of larger size and higher velocity towards the central axis of the cylinder. In this case, the important atomization may not occur in the nozzles but as a result of the interaction between the fast moving droplets and the high pressure air. To optimize the process, it is possible to use a combination of nozzles, with different injection timings, and this is the approach that has been adopted in the present work.

CFD simulations have been carried out to get a better understanding of the process. These show clearly that the temperature is low in the wet zones, which are well penetrated with water droplets, and high in the dry zones of the cylinder. Furthermore, they indicate that dry air that is pushed upwards by the piston has a tendency to drive the droplets back towards the cylinder walls, preventing adequate cooling of the core region. The right choice of spray nozzles, their position, injection timing and the injected water mass is one of the key factors for the success of isothermal compression.

PROOF-OF-CONCEPT TEST OF ISOTHERMAL COMPRESSION

Proof-of-concept tests to determine the feasibility of quasi-isothermal compression of air by means of water injection were done in 1995 using a converted large bore diesel engine. The test results were then used for the design of a purpose-built full-size prototype isothermal compressor, integrated into a novel type of engine for high efficiency power generation.

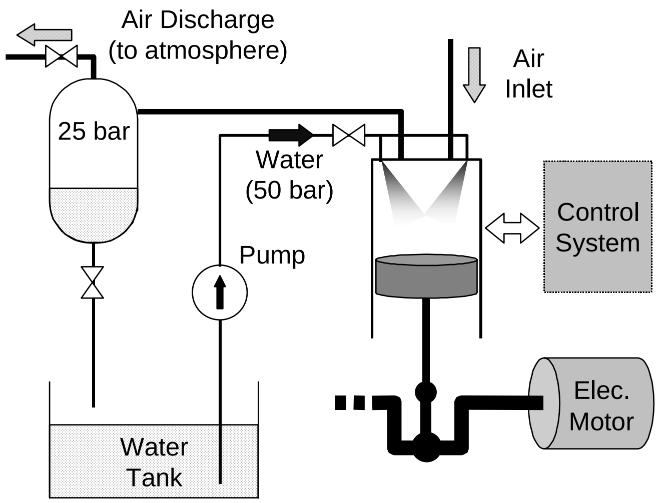
Test Rig Design

One cylinder of a Bolnes three-cylinder low-speed two-stroke marine diesel engine with a 200 mm bore and a 350 mm stroke was converted into a compressor. A variable speed electric motor was used to drive the compressor. The cross-head arrangement of the engine permitted a design, which prevented any water that leaked past the piston, from contaminating the oil in the crankcase. The cylinder head contained one inlet valve (suction valve) in the center and four discharge valves, all hydraulically operated. Additionally, two relief valves were accommodated on the cylinder head as a safety feature. Six cylindrical inserts (each containing three commercial hollow-cone atomizers) were mounted in the cylinder head. The piston had cutouts to avoid contact with the nozzles. The cylindrical inserts were mounted at an angle of 45° to the vertical and were oriented at 20° to the cylinder radius in order to direct part of the hollow cone spray towards the center of the cylinder.

Air was taken from the test shop though a filter. The compressor discharged the air/water mixture into a pressurized water separator, which was normally maintained at 25 bar pressure using a control valve. A purpose-designed hydraulic piston pump capable of up to 50 bar injection pressure was used to inject de-ionized water. The hydraulic pump was fed from a supply loop at 10 bar, which could also provide some low pressure injection at an early stage of compression. It was possible to control the start and end of the injection period and to adjust the ramp rate of the high pressure injection pump. The high pressure piston pump could inject up to 65 grams per stroke. It was possible to achieve 100 grams per stroke if both high and low pressure injection was used. After discharge of the compressed two-phase mixture air to the pressurized separator, the water was fed back into the main atmospheric water tank as shown in Figure 1.

Measurements and Analysis

A high-speed system recorded transient data including cylinder pressure and atomizer pressures at half-degree crank intervals. The atomizer pressure allowed calculation of the transient water flow into the cylinder. A low speed data recording system was used for measurements, which changed slowly. More than 350 tests were performed, with the key operational parameters listed in Figure 1.



Rig Test Parameters

- Rig running speed (50 - 200 rpm)
- Atomizer type and set up
- Timing of main water addition
- Mass of water added
- Cylinder clearance volume
- Intake and discharge valve timings

Figure 1: Rig test schematic diagram and test program

One aim of the analysis was to determine the experimental work of compression and compare it with the ideal adiabatic compression work, yielding a percentage work saving. This was determined using the measured pressure with the air volume calculated from the crank angle measurement. The calculated work included the work of pumping water from atmospheric pressure up to the injection pressure, but it was assumed that 80% of the pressure energy of the separated high pressure water would be recoverable. The calculated work of compression included the work required to expel the compressed gas at 25 bar minus the smaller amount of work done on the piston during the air intake. The practical effect of discharge pressure losses was not included in this analysis. The mass of air being compressed was determined from the cylinder pressure at suction valve close and the air inlet temperature. The values of the ideal adiabatic and isothermal compression work were then calculated for this same mass of gas to compare with the experimental compression work.

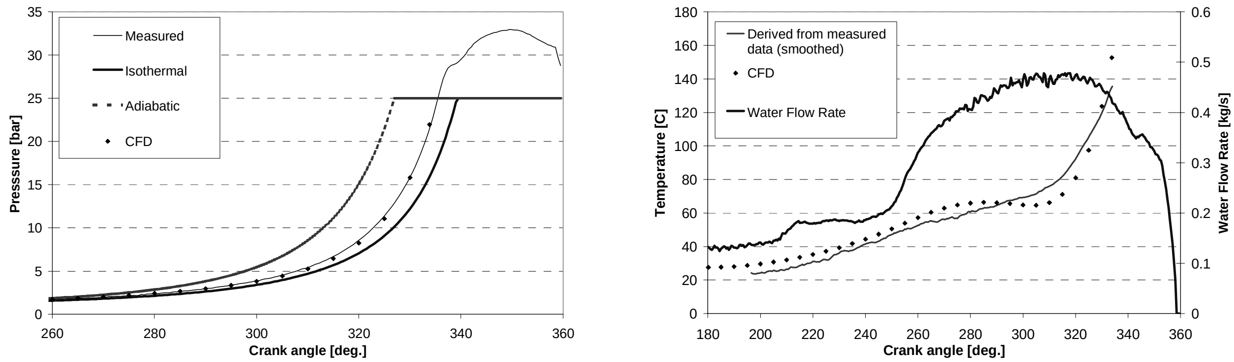
It was also possible to determine the average air temperature from the transient pressure measurement, which is helpful in understanding the physical processes. The alternative of a direct measurement of transient air temperature in the presence of a large amount of colder water is not considered to be feasible in practice.

Test Results

The best experimental results were obtained at 100 rpm rig running speed using Spraying Systems N4 nozzles, which have a flow number of $2.5\times10^{-7}m^2$. In this case the calculated air temperature was held below 100˚C for the whole transient and the work saving was 28% relative to the reference adiabatic case. Figure 2 shows some results at 200 rpm using N10 nozzles, which have a flow number of $6.3\times10^{-7}m^2$. Figure 2a shows the plot of the in-cylinder pressure together with the pressure traces calculated for an ideal isothermal and an ideal adiabatic compression, both calculated with the same mass of gas and initial conditions as in the actual test. As expected, the experimental trace lies between the two ideal traces but is closer to the isothermal condition. The actual cylinder pressure exceeds 25 bar at the end of compression, but in calculating the work saving only the part of the compression up to this pressure was examined.

Figure 2b shows the calculated gas temperature and water flow rate for the same test run. In this example, the plot shows the gas temperature rising slowly as compression begins with a low level of water addition. When the main water spray begins, the gas temperature rises more slowly in spite of the increasing heat generation. Towards the end of the stroke, the heat generation rate increases rapidly and the water spray is unable to remove all of the heat. The maximum temperature occurs when the delivery valve opens.

In spite of some concerns about the possible effects of injecting significant amounts of water into a reciprocating compressor, no problems were encountered with "hydraulic locking" in any of the tests.



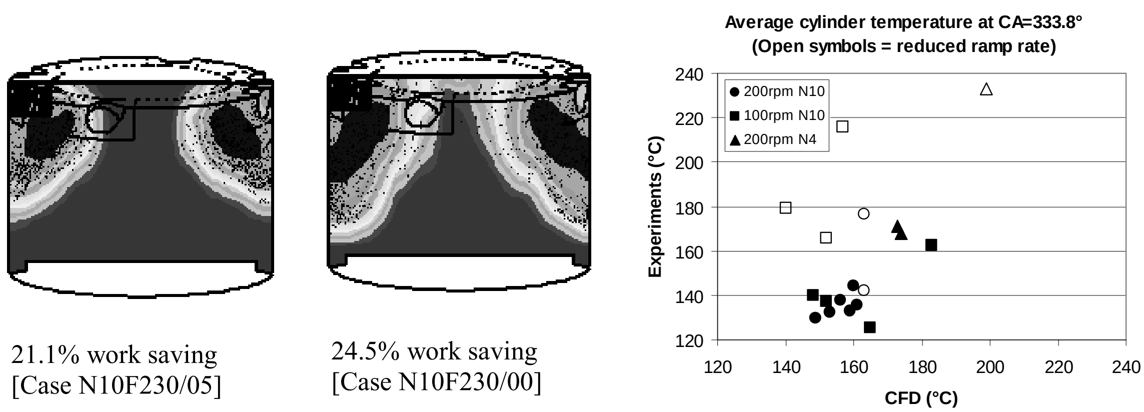
a) b)

Figure 2: Pressures and temperatures during compression (rig test)

CFD Analysis

The development and verification of a detailed CFD model of the single hollow cone spray used in the Proof of Concept rig is described by Stephenson et al. (2001). It has been successfully extended to model the cylinder in the test rig compressor, with 18 sprays in a moving geometry, see Abdallah et al. (2001). This model was created in the Star CD computer code and consists of 160 000 computational nodes in a moving mesh with droplets modeled using the Lagrangian approach. It represents the smaller of the two cylinder head clearances investigated experimentally. The transient simulations were carried out from a crank angle of 180° (bottom dead center) to 333.8°, with 0.2° crank angle per step, i.e. a total of 769 steps for one case. The cases were chosen with the intention to cover the following parameters:

- Nozzles provided by Spraying Systems Inc. (type N10 and N4), producing hollow cone sprays
- The widest possible range of different water flow profiles through the nozzles.
- Two running speeds, i.e. 100 rpm and 200 rpm



a) b) c)

Figure 3: CFD plots of the temperature and droplet distribution in the cylinder at a crank angle of 291° and a comparison between average cylinder temperature predicted by CFD and obtained experimentally.

The type of simulation reported here is near the limit of what is possible with today's CFD-codes. A number of simplifications, assumptions and omissions made in CFD models affect the precision of the simulations. Despite this, the simulations were able to predict the final pressure within 14% and the final absolute temperature within 12% of the experimental data in all 18 simulated cases and much better in many of them. One example of predicted variations in cylinder pressure and temperature with crank angle is compared to experimental data in Figure 2, showing good agreement. The experimental average cylinder temperatures at a crank angle of 333.8° are plotted against the predicted temperatures in Figure 3c. It can be seen that the agreement between predictions and experiments is not as good for cases with reduced ramp rate (open symbols) than for the full ramp rate cases (solid symbols). The reason for this will be studied further in the continuing work.

Figure 3a and b also show results from CFD calculations of two different cases. The one to the left has a smaller work saving compared to adiabatic compression than the other case, which is the same case as shown in Figure 2. It is obvious that there is a strong correlation between the temperature distribution and the distribution of droplets (small black dots). It is also noted that a higher work saving is achieved in the case where the droplets have penetrated further. The central area without droplets is the area where the temperature is high.

PROTOTYPE ISOTHERMAL COMPRESSOR

The research on isothermal compression by water injection is being carried out in conjunction with a development program for a large high-efficiency reciprocating engine. The novel isoengine cycle, described by Coney et al. (2002a, 2002b), has advantages over existing power cycles mainly because the isothermal compressor is able to deliver compressed combustion air at high pressure but low temperature, which allows recovery of heat from various parts of the engine. The fuel saved by the recovery of engine heat together with the reduced compression work is predicted to give an electric efficiency of up to 60% in the fully developed engine. A 3 MW_e prototype engine, with a single 385mm × 400mm (bore × stroke) compression cylinder operating at 600 rpm design speed, is nearly complete. Testing of the compressor has already begun. The compressor is able to carry out quasi-isothermal compression of up to 1.7 kg/s of air with an inlet pressure of 4 bar (abs) and discharge pressure of 100 bar (abs). A maximum operating pressure of up to 170 bar (at a compression ratio of up to 1:30) is intended for a 7MW commercial version of the isoengine having two compression cylinders.

Design Concept

The Proof of Concept tests demonstrated that achieving a large heat transfer area (i.e. a small droplet size) and a high mass of injected water together with a good penetration and good distribution is essential. The prototype compressor has a displacement volume of 46.6 liters compared to 11 liters in the Proof of Concept tests. The maximum speed has been increased from 200 rpm to 600 rpm. Furthermore, a turbocharger and electric blower can be used to boost the air intake pressure to a maximum of 8 bar (abs). This implies a maximum air flow, which is 100 times larger than in the Proof of Concept tests. To achieve a corresponding increase in water flow and to improve the distribution, the number of nozzles was increased to a total of 360 (compared to 18). The average nozzle flow number was also increased to a value of 2.85×10^{-6} m^2 (4.5 times larger than the N10 nozzles). Fan spray nozzles were chosen in place of hollow-cone nozzles in order to achieve a good penetration towards the center of the cylinder. These nozzles are not part of the cylinder head but are inserted into two spray rings located at the upper end of the cylinder. Each spray ring has a separate water supply and feeds two circumferential rows of 90 nozzles. The piston rings remain below the lowest nozzles at all times.

Three inlet valves and four discharge valves are mounted into the cylinder head. Extra relief valves to prevent damage by hydraulic locking are not required since the novel discharge valves provide inherent safety features, as described below. The air-water mixture is discharged into a conventional separator with an efficient mechanical gas dryer. The water is cooled and used for re-injection while the compressed air is fed to the combustor part of the engine. The separator also acts as a pulsation damper for the delivery air.

Inertial Pumping System

A schematic diagram of the air path and the spray water loop is shown in Figure 4. The system allows water to be injected into the compressor and then recirculated through a cooler without consuming power in injection pumps. Instead the water is driven by the separator pressure and controlled by valves and by making use of the inertia of the water in the pipework. The system includes an accumulator that receives pressurized water from the separator and isolates the pressure and flow fluctuations from the spray water cooler. The inertial pipework consists of a high inertia loop and a low inertial leg. Two independently controlled, fast acting spool valves are positioned adjacent to the upper and lower spray rings and are opened over precise periods during the compression stroke to deliver spray water into the cylinder. A further valve is positioned in the low inertia leg. At the start of the compression stroke, the cylinder pressure rises slowly and only a small flow of spray water is required. At this point, the water in the high inertia loop is accelerated from rest, steadily increasing the flow rate into the cylinder. Although the pressure in the cylinder eventually rises up to and above the separator pressure, the fluid momentum built up in the high inertia loop is sufficient to overcome the resistance and spray water continues to be injected into the cylinder. If a boost to the flow is required earlier during the compression, the valve in the low inertia leg is opened as well but this reduces the flow later in the stroke.

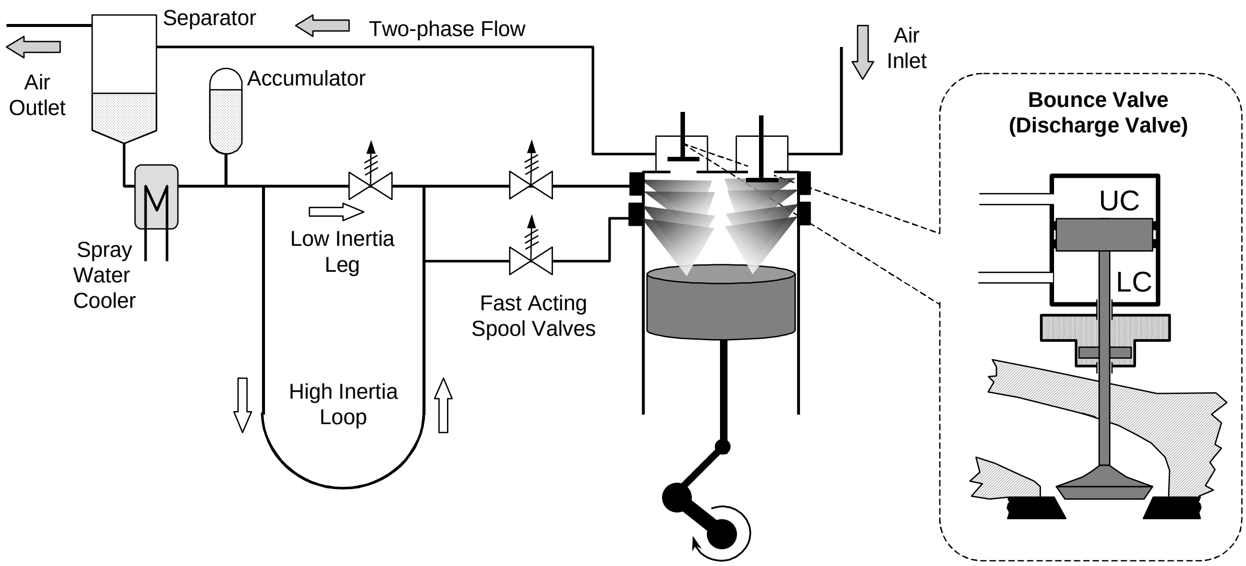


Figure 4: Prototype reciprocating isothermal compressor schematic diagram

Novel Discharge Valves

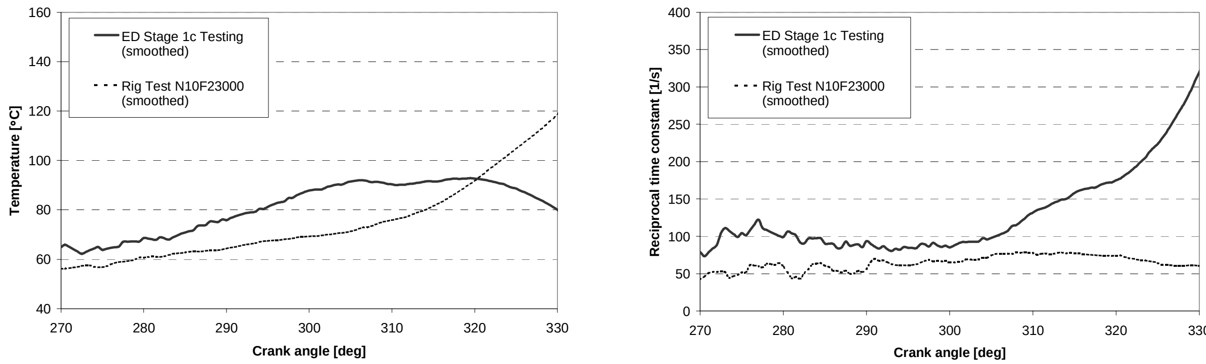
A schematic of one of the four discharge valves (bounce valves) used to release the compressed air/water mixture from the cylinder is shown in the enlarged section on the right side of Figure 4. The top of each valve stem is connected to a piston, which can move freely in a small cylinder. The air filled spaces above and below the piston are referred to as "upper chamber" (UC) and "lower chamber" (LC), respectively. Both chambers are pressurized by air reservoirs. When the pressure rises during compression, it becomes large enough at some point to overcome the force exerted by the port pressure, the differential pressure between UC and LC and the valve's weight. The valves start opening and release the air/water mixture during the remaining part of the compression stroke before the piston reaches TDC. The pressure in the upper chamber rises during the valve lift while that of the lower chamber falls, causing a reversal of the net force on the valve, which returns it into its closed position. An oil damper ensures that each valve is slowed down before closing, avoiding damage to the valve seats.

The pressure difference between the upper and lower chamber can be adjusted to vary the valve opening time. It is preferable to operate the bounce chambers with an initial bias towards early opening in order to minimize the over-pressure in the cylinder, which results from the finite time to open the valves. The absolute pressures of the two chambers determine the natural frequency of the bounce valve, which can be used to adjust the valve closing time. At low compressor speeds, the valves bounce twice while discharging the flow.

First Test Results

The compression cylinder underwent first testing in June 2001. Degassing of dissolved air caused delays in the timing of water injection, but this was overcome by a re-design of the spray rings. Some improvements to the sealing and air pressure control of the bounce valves were also implemented. Resumed testing in early 2002 showed significantly improved performance in both the above areas. Reliable operation of the isocompressor at compression ratios of up to 1:25, including the bounce valves and the inertial pumping system, has been demonstrated at 200 and 380 rpm. Later this year when the engine is completed, compressor testing will continue up to the design speed of 600 rpm.

First test results for the prototype (Engineering Demonstrator, ED) show that significant improvements compared to the Proof of Concept rig tests can be achieved. A plot comparing the temperatures during the most important part of the compression for the rig test at 200 rpm (case N10F23000) and data for the prototype at 380 rpm is shown in Figure 5a. It can be seen how an increase in air temperature towards the end of the compression stroke can be avoided with the prototype design, even though both the speed and the cylinder diameter are almost doubled. Very high values for the reciprocal time constant are reached here, see Figure 5b. This improvement is the result of a better spray water distribution achieved with 360 nozzles, better penetration of the fan sprays into the high pressure air and an overall increase in the water/air mass ratio.



a) b)

Figure 5: Comparison of derived temperatures (a) and reciprocal time constants (b) for rig test and first ED test results during the most important part of compression stroke vs. crank angle

Once the prototype engine is fully completed, further compressor tests will be carried out to optimize the power saving. Further development is anticipated to improve the performance of the bounce valves to reduce discharge pressure losses, optimize the timing and maximize volumetric efficiency.

CONCLUSIONS

It has been demonstrated that it is possible to achieve substantial reductions in the compression work of a reciprocating compressor by injecting large amounts of water directly into the cylinder. This has been done at speeds up to 380 rpm and pressure ratios up to 25 in a 46 liter cylinder. The tests show that the transient air temperatures can be maintained below 100°C compared to an adiabatic temperature of about 500°C. Development of the isocompressor with its innovative inertial pumping system and bounce valves is continuing.

This novel compression technology will be part of a new high efficiency thermodynamic cycle for power generation. The technology can also be used for stand-alone compressors in industrial processes where gases at high pressure are required. In addition to savings in power consumption it is expected that costs can be reduced compared to conventional reciprocating compressor technology.

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