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(54) GAS COMPRESSOR

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(57) ABSTRACT

A reciprocating gas compressor in which a piston **(3)** reciprocates within a compression chamber **(1)** in order to compress gas. Water is injected through nozzle **(6, 7)** into the compression chamber to cool the gas during compression. A source of pressurised liquid **(13)** is arranged to accelerate liquid through the nozzle **(6, 7)**. At least one duct **(14, 17, 19)** connects the pressurised source **(13)** to the nozzles **(6, 7)**. The dimensions of the duct **(14, 17, 19)** are sized to define the inertia of the liquid to control the acceleration of the mass flow through the nozzles during compression such that the cooling capacity of the liquid in the compression chamber increases as the pressure therein approaches its final value.

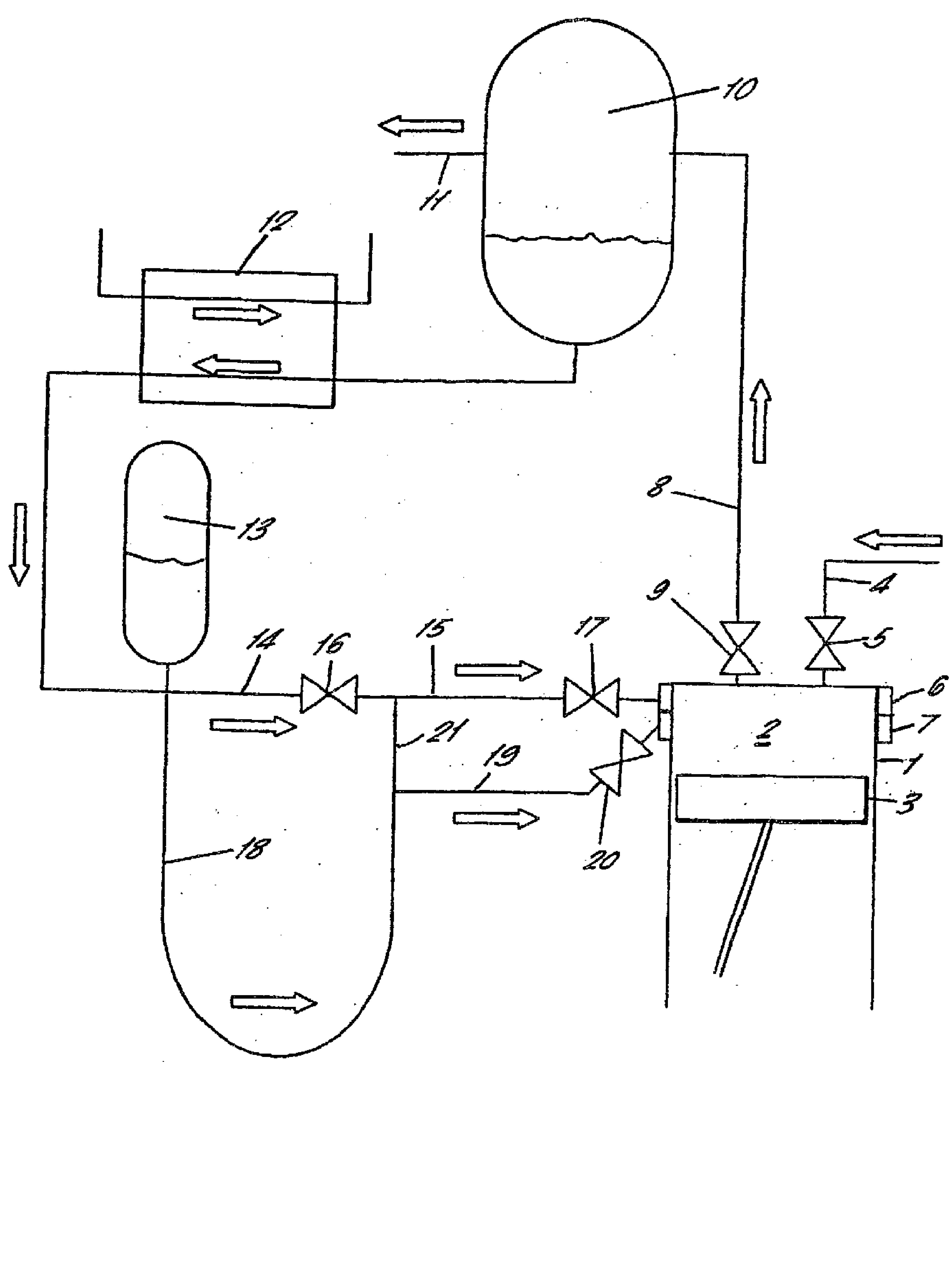


FIG. 1. LIQUID FLOW RATE

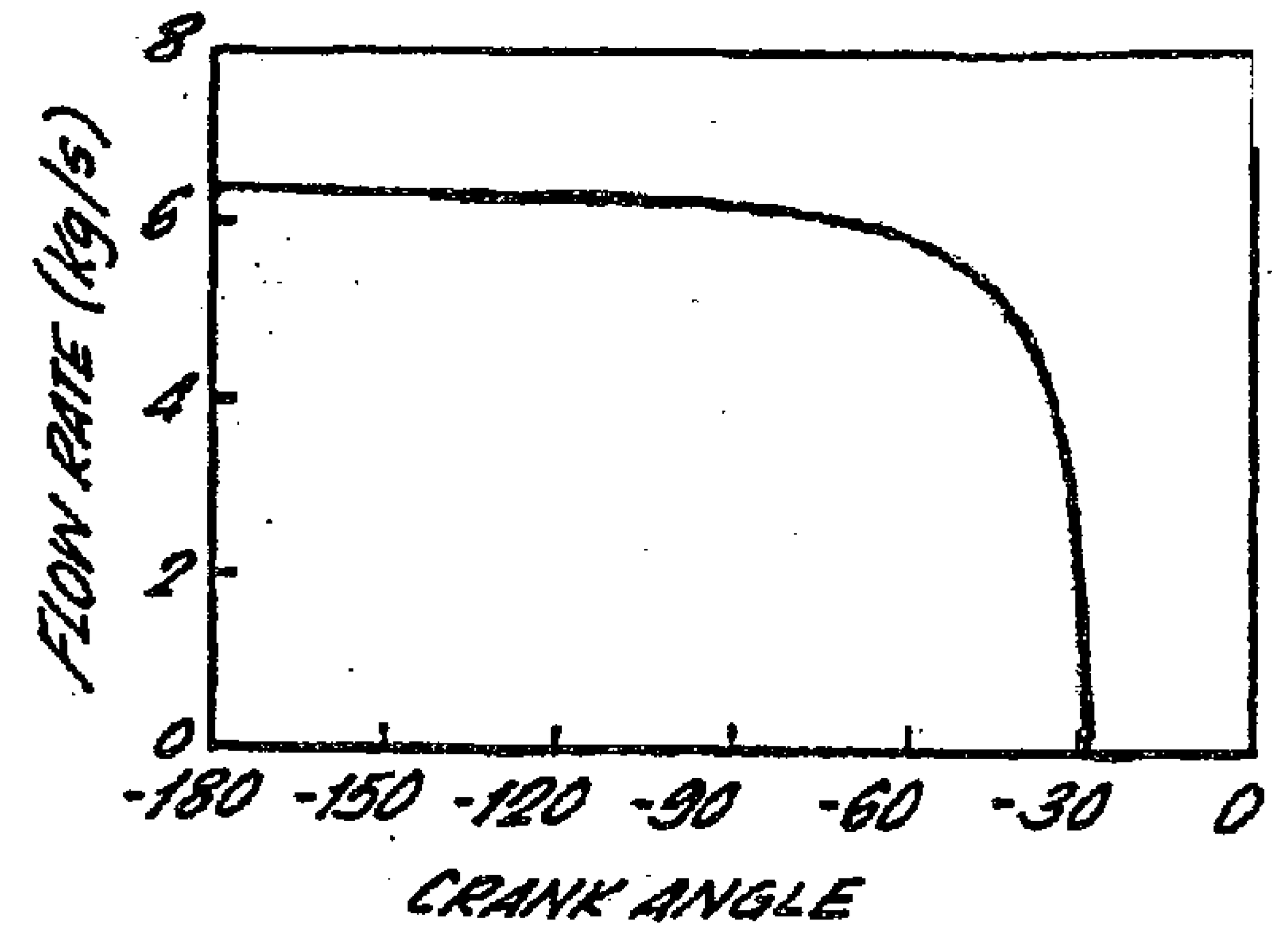


FIG. 2. NOZZLE dP

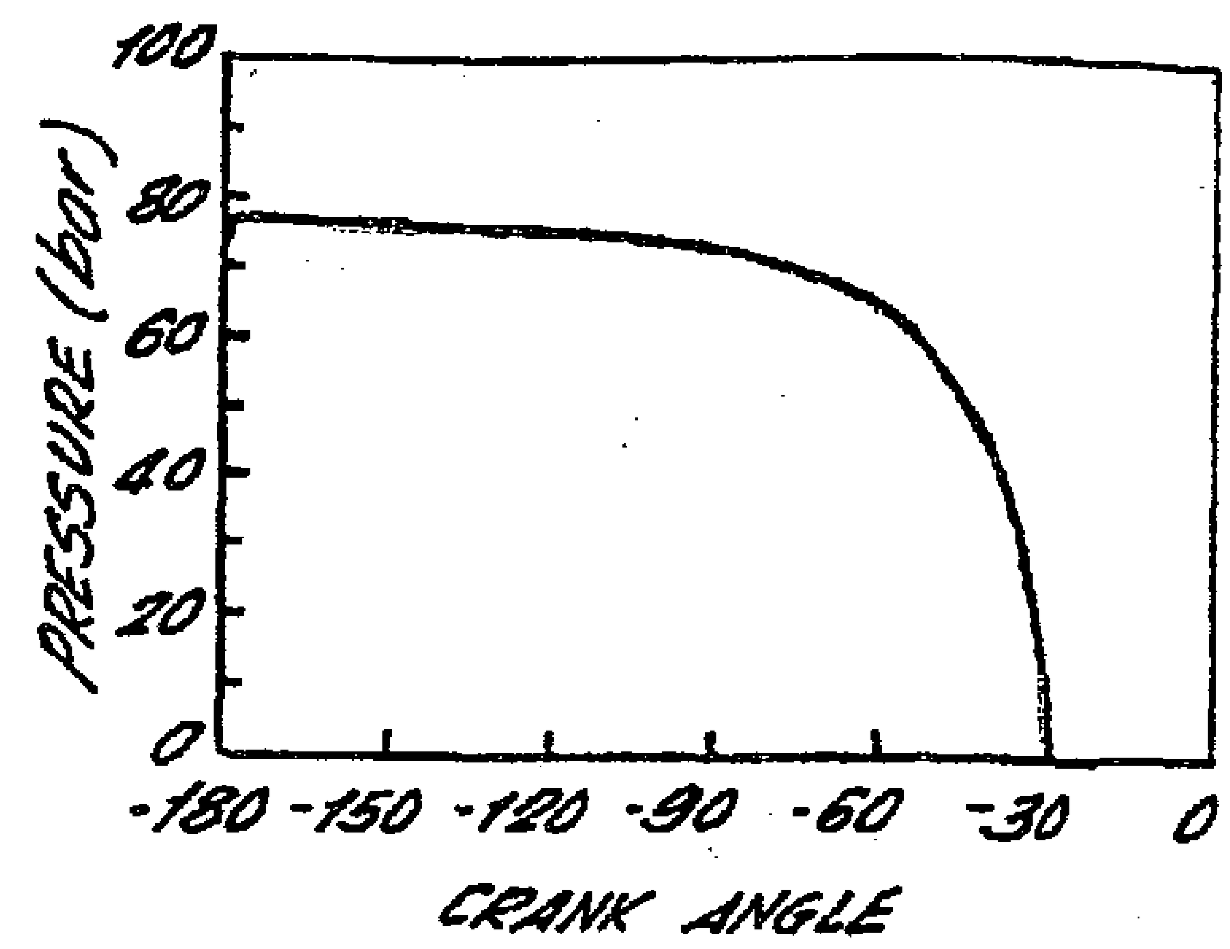


FIG. 3 LIQUID FLOW RATE

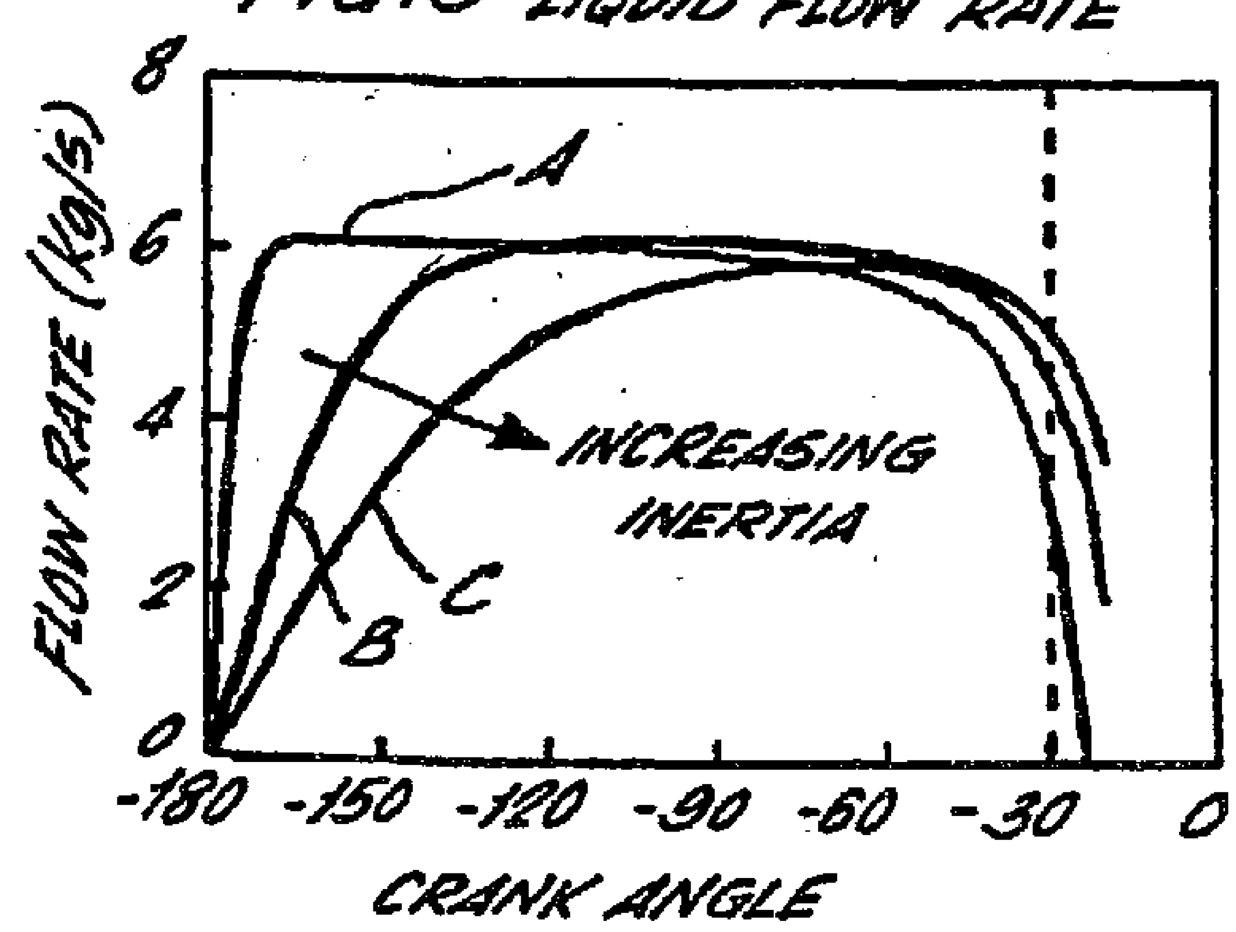


FIG. 4. NOZZLE dP

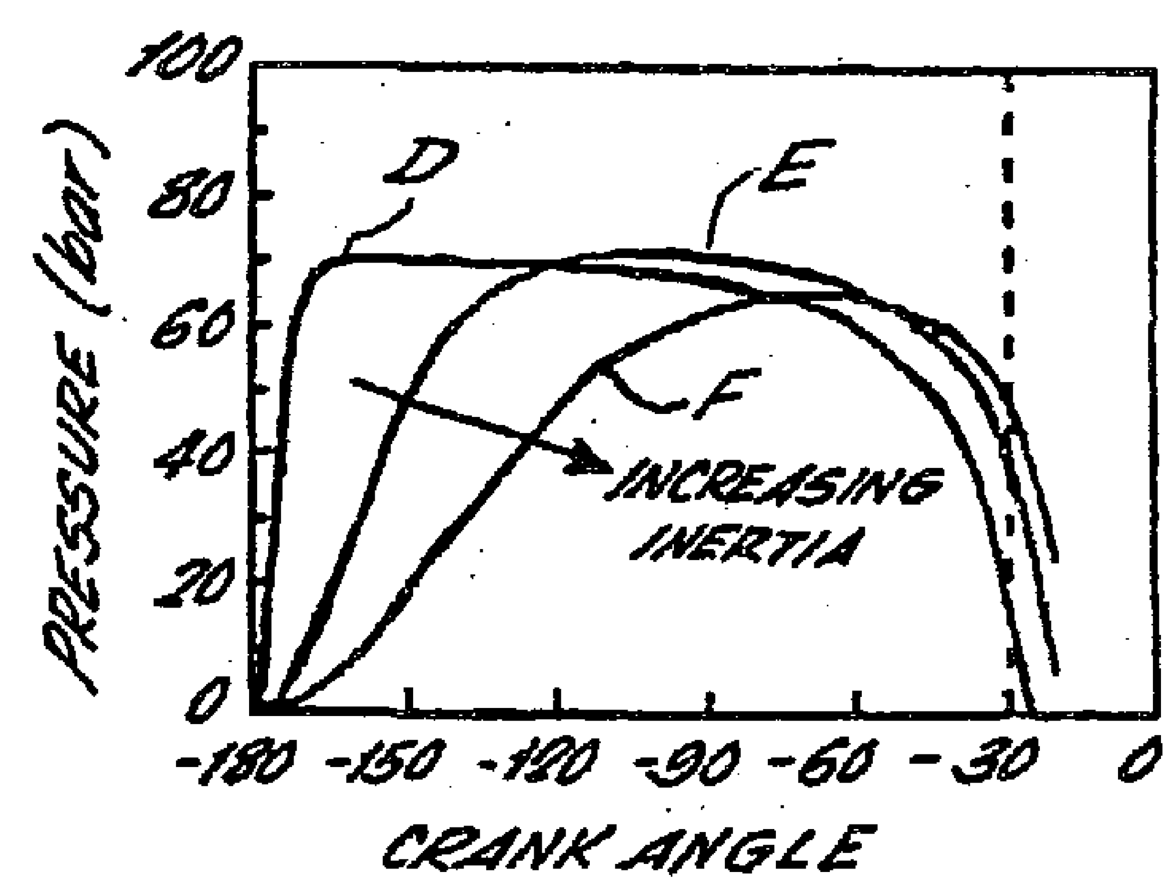


FIG. 5. NOZZLE dP

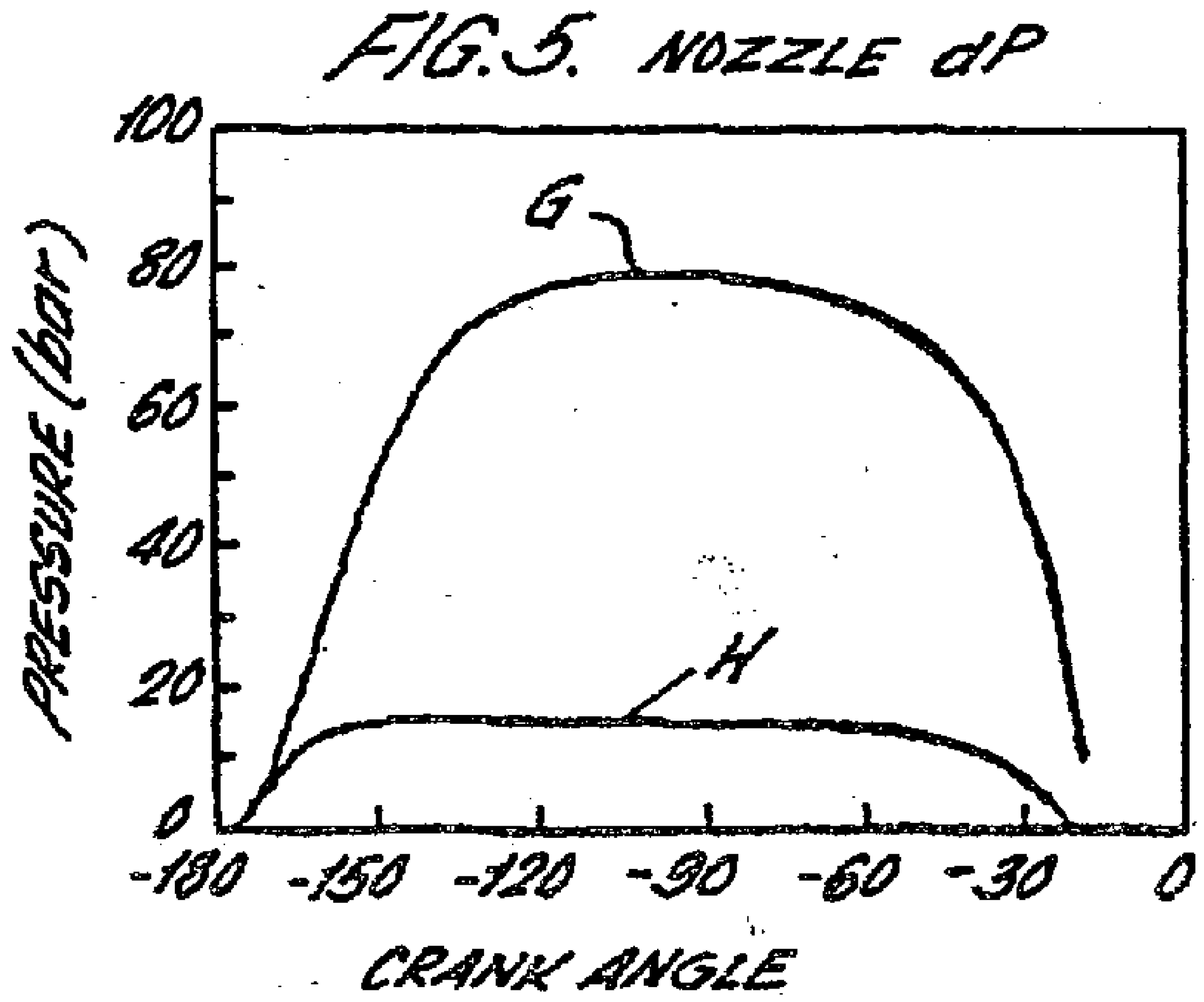


FIG. 6.

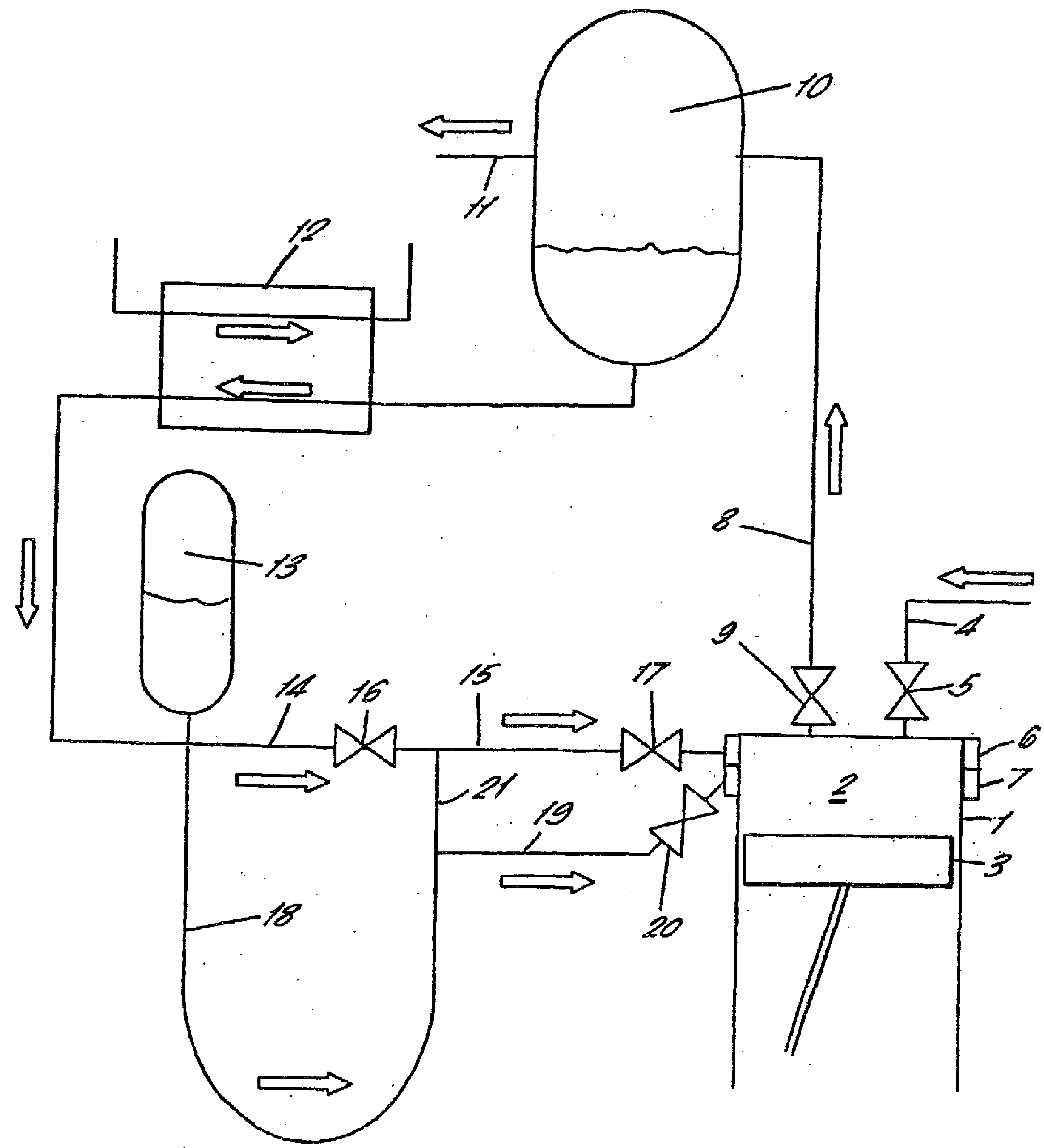


FIG. 7A.

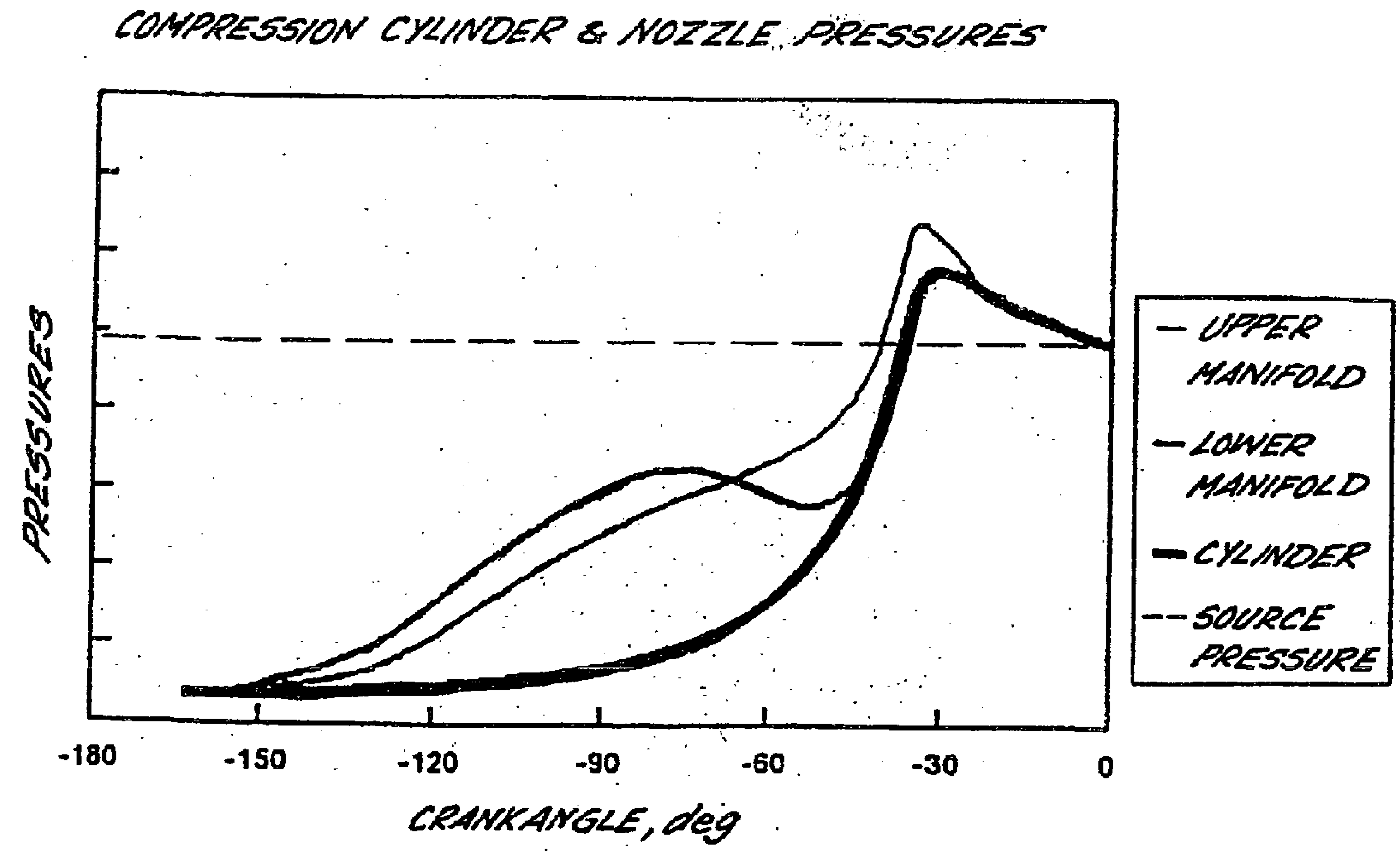
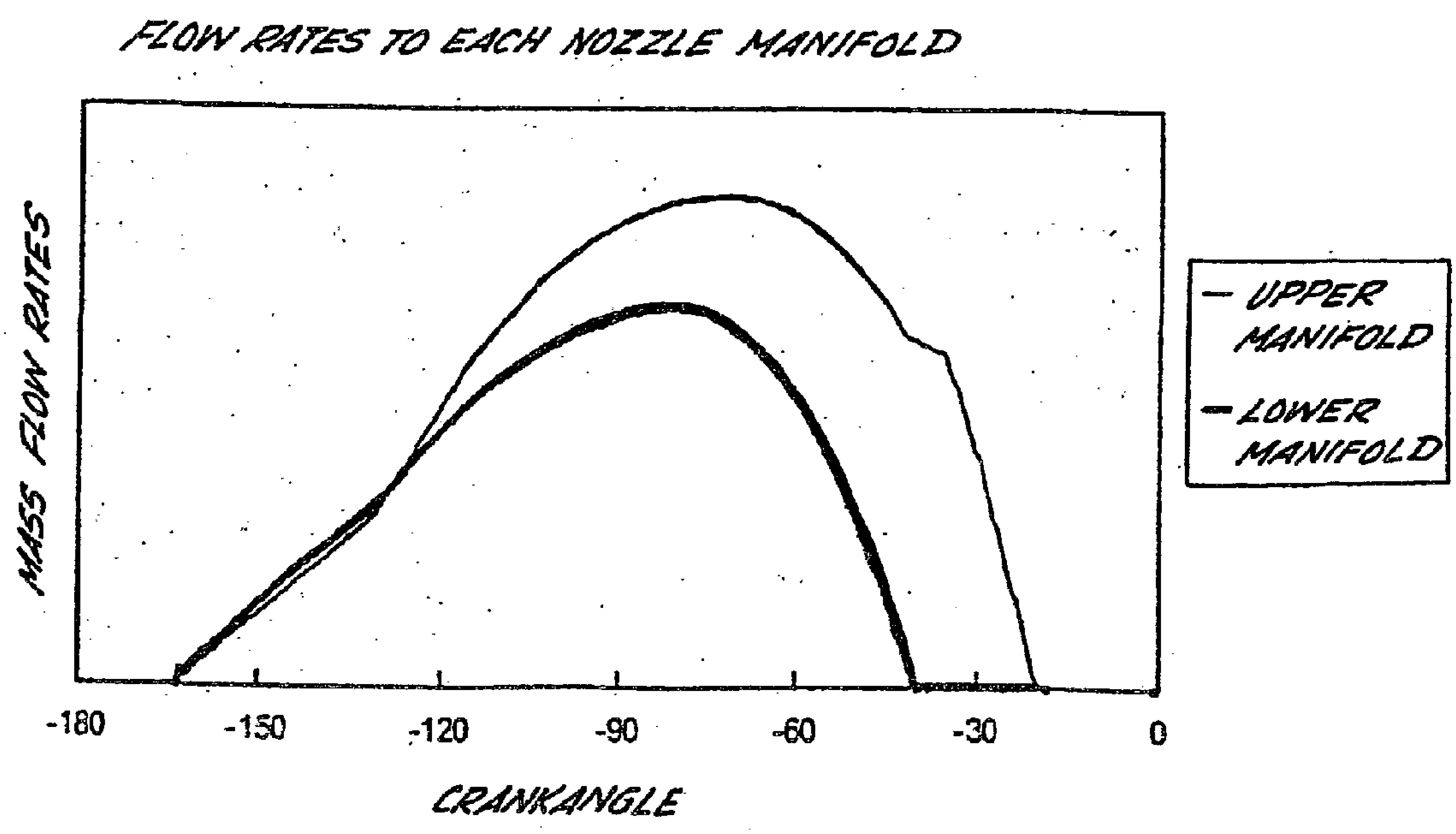

COMPRESSION CYLINDER & NOZZLE PRESSURES
PRESSURES
-180
-150
-120
-90
-60
-30
0
CRANKANGLE, deg
— UPPER MANIFOLD
— LOWER MANIFOLD
— CYLINDER
-- SOURCE PRESSURE

FIG. 7B.


FLOW RATES TO EACH NOZZLE MANIFOLD
MASS FLOW RATES
-180
-150
-120
-90
-60
-30
0
CRANKANGLE
— UPPER MANIFOLD
— LOWER MANIFOLD

GAS COMPRESSOR

[0001] This invention relates to gas compressors and in particular to reciprocating gas compressors in which liquid is sprayed into the compression chamber to control the gas temperature during compression.

[0002] The concept of spraying liquid into a compression cylinder as a means for absorbing the heat of compression is described in a number of publications and referred to in the art as "wet compression". The technique involves spraying liquid into the cylinder through a nozzle which divides the liquid into a mist of fine droplets. The droplets travel through the gas space and eventually impinge on the cylinder surfaces. While in the gas space, the droplets provide a heat sink which is in intimate contact with the gas being compressed and which has a large surface area allowing heat to be drawn efficiently from the gas, thereby limiting the rise in gas temperature and reducing the work required for compression.

[0003] The spray liquid may be driven through the nozzle or nozzles by a mechanical, positive displacement pump, as, disclosed for example in GB 722524, FR 903471 and WO 98/16741. On the other hand, the spray liquid may be driven by the compressed gas from the compressor, as disclosed in DE 357858. The compressor described in this document includes an accumulator which temporarily stores compressed gas and spray liquid drawn from the compression cylinder. The lower part of the accumulator is connected to a heat exchanger for cooling the spray liquid, and a reservoir containing cooled liquid below the heat exchanger is connected to a spray orifice in the compression cylinder by a duct. The liquid spray is driven and controlled by the pressure in the accumulator without any active control mechanism. Liquid is sprayed into the cylinder during both the air intake and compression strokes and ceases when the pressure in the compression cylinder reaches the pressure in the accumulator.

[0004] While compressed gas from the compressor offers an attractive driving means for injecting liquid into the compression cylinder by removing the need for additional mechanical pumps, a drawback of this method is the diminished control it provides over the flow rate into the cylinder compared with the control provided by a positive displacement pump. For example, the pressure in the accumulator is limited to the final pressure in the compression cylinder. Also, the pressure available to drive the spray liquid is the difference between the pressure in the accumulator and the pressure in the cylinder and this difference diminishes over the compression stroke and falls off rapidly to zero as the cylinder pressure approaches its final value. Thus, the flow rate, which depends on the pressure difference across the spray nozzle also decreases over the compression stroke and falls off rapidly as the cylinder pressure approaches its final value. However, most of the heat is evolved towards the end of compression, and therefore the flow rate needs to increase during compression to meet the increased cooling demand. Although this requirement can be met by using a positive displacement pump, since it allows the injection pressure and flow rate to be controlled independently of the cylinder pressure, this solution involves additional equipment and a power source.

[0005] In diesel engines, liquid injection systems are used to inject controlled quantities of liquid fuel into the combustion cylinder at a point in the cycle when the combustion air in the cylinder is at high pressure. Fuel injection is commonly driven by a com-operated mechanical pump to achieve the high injection pressures required. Another method is to use "common rail injection" in which the fuel is pumped into a common pressurised reservoir and then admitted by small very fast acting valves into each cylinder in turn. More recently, it has been proposed to drive the fuel injection using the "water hammer" effect. C. Stan and E. Hilliger "Pilot Injection System for Gas Engines using Electronically Controlled Ram Tuned Diesel Injection", CIMAC Congress 1998, Copenhagen, pages 1429 to 1438 describes a pilot fuel injection system for a gas engine comprising a closed loop circuit through which diesel fuel is circulated, a fuel injector whose input is tapped into the circuit and an electronically controlled valve placed in the circuit, downstream from the connection to the injector, for controlling the fuel flow. To operate the injector, the valve is opened and fuel is driven through the circuit by a pressure source which includes a fuel pump and pressure accumulator. In this phase, no fuel is injected since the pressure of fuel in the circuit is below the injector threshold pressure. To operate the injector, the valve is rapidly closed, abruptly blocking the flow through the circuit and generating a pressure wave and rapid increase in pressure sufficient to drive fuel through the injector. The peak pressure depends, inter alia, on the final velocity of the liquid in the circuit just prior to closing the valve which can be varied by varying the period over which the valve is open. As the pressure wave duration is constant and independent of pressure amplitude, the system allows the injection volume to be conveniently controlled by changing the pressure in the hydraulic circuit. However, this system does not allow the injection duration nor the injection profile to be easily controlled and requires the operation of a mechanical pump.

[0006] According to the present invention there is provided a gas compressor comprising a compression chamber to contain gas to be compressed, a compression piston to compress the gas by movement of the compression piston in the compression chamber, valve means for allowing compressed gas to be drawn from the compression chamber, an atomiser for spraying liquid into the compression chamber to absorb heat from the gas during compression, a pressurised source of liquid and a duct arranged to feed liquid from the pressurised source to the atomiser, wherein the source is arranged to accelerate the liquid through the atomiser into the compression chamber and the dimensions of the duct are sized to define an inertia of the liquid therein to control the rate of acceleration of the mass flow through the atomiser during compression such that the flow rate of liquid into the compression chamber is substantially reduced when the pressure difference between the source and the compression chamber is high and substantially enhanced when the pressure difference between the source and the compression chamber is low.

[0007] Advantageously, the present invention uses the inertia of liquid in the flow path between the pressurised source and spray atomiser to control the flow rate of liquid into the compression chamber. For example, the inertia can be defined to control the rate of acceleration and deceleration of liquid through the atomiser during compression. Thus, in the initial stages of compression, when the pressure difference between the pressurised source and compression cylinder is large but the rate of heat evolution is low, the inertia of the liquid can be defined to moderate or limit the rate of acceleration of liquid through the atomiser and limit the flow rate of liquid into the compression chamber. Advantageously, this allows the flow rate to be controlled so that only a sufficient quantity of liquid is present in the gas to provide the modest heat absorption capacity required in the early stages of compression. Advantageously, the inertia provides a means of reducing the mass flow rate without consuming and dissipating energy, the energy being stored in the liquid and being used to continue to drive liquid through the atomiser against the increase in pressure in the compression chamber. The inertia also provides a means of delaying the highest flow rates through the atomiser until the pressure in the compression chamber and therefore the heat of compression is rising rapidly as the pressure approaches its final value. The increase in flow rate is limited to some extent as the flow resistance increases sharply at high flow rates. Furthermore, the inertia also controls the rate of deceleration of liquid through the atomiser. For example, a high inertia causes the flow to continue at a relatively high rate near the end of compression in spite of the diminished pressure difference between the pressurised source and the compression chamber.

[0008] As mentioned above, a degree of control over the acceleration and deceleration of the liquid is provided by defining the inertia. However, in order to enhance the flow control, a valve is preferably provided to control the timing of the flow through the atomiser. Thus, the valve can be used to prevent water from being injected into the compression chamber during the air intake period. During this time, no heat is generated and the injection of water which does not perform efficient cooling may simply increase the parasitic power loss as work has to be done by the compressor piston in order to expel this water from the cylinder at high pressure. With the valve, the start of the injection can be delayed until the temperature of the air under compression has risen to a point where efficient heat transfer can take place. In compressors running at high speed, it may be advantageous to begin the injection of water during the air intake period, in order that the water is distributed within the cylinder when the temperature starts to rise. In this case also, the valve can be used to control the start of injection in order to maximise the beneficial cooling effect with minimum usage of water. The valve may also be used to terminate the flow quickly at the end of the compression period. This may cause some of the kinetic energy of the flow in the pipe to be lost. However, it will prevent the injection from extending unnecessarily into the air discharge period.

[0009] The inertia of liquid in a duct is proportional to its length and inversely proportional to the cross-sectional area of the flow passage defined thereby and therefore the inertia can be conveniently controlled according to those parameters.

[0010] Preferably, the dimensions of the duct are sized to define the inertia such that the mass flow increases through the atomiser in a period of at least 30°, more preferably at least 45° and most preferably at least 60°of crank angle of a nominal crank shaft driving the piston.

[0011] Advantageously, the present invention allows the flow rate to be controlled during compression without the need for a positive displacement pump and without the need for a pressure source of greater pressure than the compressed gas drawn from the compression chamber. Conveniently, the pressurised source may comprise a reservoir of liquid pressurised by compressed gas and conveniently, the compressed gas may be supplied by the compression chamber.

[0012] Preferably the reservoir is an accumulator, and the compressor further comprises a separator upstream of the accumulator which receives gas and liquid from the compression chamber, means to feed the liquid in the separator to the accumulator, and a cooler to cool the liquid being fed from the separator to the accumulator. The use of the accumulator isolates the pumping system from the upstream circuit. Without it the inertia of the upstream pipework would affect the pumping through the atomisers. Further, by providing a separate accumulator and separator, space is available for the cooler to be provided to extract from the liquid the heat absorbed in the compression chamber.

[0013] Preferably, a further duct is provided to feed liquid from the reservoir to the atomiser. The dimensions of the further duct are preferably sized to define an inertia of liquid therein different to the inertia defined by the dimensions of the first duct. A further valve is preferably provided for controlling the flow of liquid through the further duct. With this arrangement, under the control of the valve and the further valve, the duct with the lower inertia can be used to contribute a higher mass through the atomiser sooner and the duct with the higher inertia can be used to contribute a higher flow rate through the atomiser later.

[0014] The gas compressor may comprise a further atomiser for spraying liquid into the compression chamber, the flow through the atomiser being controlled by a further valve. This arrangement may be particularly advantageous when the atomisers are arranged to direct liquid towards different positions within the compression chamber displaced in the direction of movement of the piston. For example, one atomiser may be arranged to spray liquid into the volume adjacent the end of the compression chamber and the other atomiser may be arranged to spray liquid into the volume spaced from the end of the compression chamber. The use of ducts having different inertias feeding different atomisers allows the flow rate of liquid through the atomisers and into the compression chamber to be controlled individually so that the flow rate through individual atomisers may be different during the period of compression.

[0015] As described above, the duct and the further duct may be provided entirely separately from one another. However, they are preferably connected by a branch duct. This allows the flow through one of the atomisers to be shut off, whereupon the flow to the shut-off atomiser is directed through the branch duct to the other atomiser. Thus when one of the atomisers is covered by the compressor piston, this atomiser can be shut off without losing the benefit of the kinetic energy of the water upstream of the branch duct. If a branch duct is used, a third valve is preferably provided in one of the ducts upstream of the location where the branch duct joins the duct, while the valve and further valve are downstream of the branch duct. Normally the third valve would be placed in the upstream duct with the lower inertia. In this case, the third valve takes over the function of controlling the point at which flow begins in the duct with the low inertia.

[0016] Also according to the present invention there is provided a method of controlling the temperature of gas in a gas compressor during compression, the gas compressor comprising a compression chamber to contain gas to be compressed, a compression piston to compress the gas by movement of the compression piston in the compression chamber, a pressurised source of liquid and a duct arranged to feed liquid from the pressurised source to the compression chamber, the method comprising the steps of spraying liquid into the compression chamber by allowing the source to accelerate liquid through the duct and controlling the rate of acceleration of the mass flow into the compression chamber during compression by sizing the duct to define an inertia of liquid therein such that the flow rate of liquid into the compression chamber is substantially reduced when the pressure difference between the source and the compression chamber is high and substantially enhanced when the pressure difference between the source and the compression chamber is low.

[**0017**] Alternatively, the invention can be defined in its broadest sense as a gas compressor comprising a compression chamber to contain gas to be compressed, a compression piston to compress the gas by movement of the compression piston in the compression chamber, valve means for allowing compressed gas to be drawn from the compression chamber, an atomiser for spraying liquid into the compression chamber to absorb heat from the gas during compression, a pressurised source of liquid and a duct arranged to feed liquid from the pressurised source to the atomiser, wherein the source is arranged to accelerate the liquid through the atomiser into the compression chamber and the dimensions of the duct are sized to define an inertia of the liquid therein to control the rate of acceleration of the mass flow through the atomiser during compression such that liquid can be injected into the compression chamber when the pressure in the compression chamber is greater than the pressure of the source.

[**0018**] By allowing injection of liquid against a negative pressure gradient between the source and the compression chamber, the invention allows the liquid to be injected towards the latter part of the piston stroke when the compression chamber pressure and hence the heat of compression is at its highest.

[**0019**] This aspect of the invention also extends to a method of controlling the temperature of gas in a gas compressor during compression, the gas compressor comprising a compression chamber to contain gas to be compressed, a compression piston to compress the gas by movement of the compression piston in the compression chamber, a pressurised source of liquid and a duct arranged to feed liquid from the pressurised source to the compression chamber, the method comprising the steps of spraying liquid into the compression chamber by allowing the source to accelerate liquid through the duct and controlling the rate of acceleration of the mass flow into the compression chamber during compression by sizing the duct to define an inertia of liquid therein such that liquid is injected into the compression chamber when the pressure in the compression chamber is greater than the pressure of the source.

[**0020**] Preferably, the inertia is such that the period during which liquid is injected while the pressure in the compression chamber is greater than the pressure of the source is at least 5°, preferably at least 10° and more preferably at least 15° of crank angle of a nominal crankshaft driving the piston.

[**0021**] The preferred features defined above in relation to the first aspect of the invention will apply equally to the alternative definition of the invention given above.

[**0022**] Examples of embodiments of the present invention will now be described with reference to the drawings, in which:

[**0023**] **FIG. 1** shows a graph of the variation of liquid flow rate as a function of crank angle with negligible liquid inertia;

[**0024**] **FIG. 2** shows a graph of the variation of pressure difference across a spray nozzle as a function of crank angle with negligible liquid inertia;

[**0025**] **FIG. 3** shows a graph of the variation of flow rate as a function of crank angle with increasing liquid inertia;

[**0026**] **FIG. 4** shows a graph of the pressure across a spray nozzle as a function of crank angle with increasing liquid inertia;

[**0027**] **FIG. 5** shows the variation of pressure across a spray nozzle as a function of crank angle for two different values of duct resistance defining similar liquid inertias;

[**0028**] **FIG. 6** shows an embodiment of the present invention;

[**0029**] **FIG. 7A** shows a graph of a variation of pressure as a function of crank angle for the embodiment shown in **FIG. 6**; and

[**0030**] **FIG. 7B** shows a graph of the mass flow rate as a function of crank angle for the embodiment shown in **FIG. 6**.

[**0031**] **FIGS. 1 and 2** illustrate, respectively, an example of the flow rate through and pressure across a spray nozzle as a function of crank angle where the source of liquid is at the exhaust pressure of the compressor and the inertia of liquid in the flow path between the pressurised source and the spray nozzle is negligible. The compression piston is at bottom dead center at −180° and at top dead center at 0° crank angle and in this example, the discharge valve opens at −30° crank angle. As shown in these Figures, the maximum liquid injection rate occurs at the start of the compression process, i.e. at −180° crank angle as the pressure in the compression chamber rises during compression. The compressor reaches the desired pressure at about 30° before top dead center and from this point onwards, the pressure in the compression chamber is the same as pressurised liquid source. Therefore, the differential pressure between the pressurised source and the compression chamber and hence the flow rate fall to zero and remain zero from that point to top dead center.

[**0032**] However, this type of injection profile may not be the optimum for ensuring near-isothermal compression. In a reciprocating compressor, the majority of compression work is done towards the end of the compression stroke, when the pressure is rising rapidly and is accompanied by a rapid increase in the rate at which the heat of compression is evolved. Therefore, it is important that the cooling capacity of spray liquid and hence the mass of cool liquid in transit through the compression chamber is sufficient to absorb the rapid increase in heat energy as the pressure in the cylinder approaches its final value. However, in **FIG. 1**, the liquid injection rate rapidly falls off towards the end of compression. Also, in **FIG. 1**, the flow rate is high over the initial stages of compression when the compression work and hence heat evolution is relatively low and the large cooling capacity provided by the spray droplets is not required. Although a valve could be used to restrict the flow rate during the early stages of compression, the valve would cause energy to be dissipated which is undesirable. Also the use of a valve in this way would not help the injection rate later in the process, when the pressure difference between the source and cylinder becomes much smaller.

[**0033**] **FIGS. 3 and 4** illustrate, respectively, examples of flow rate through and pressure across a spray nozzle as a function of crank angle where the flow path between the pressurised source and the nozzle has an inertia which is sufficient to modify the flow rate profile. Again, the pressurised source is at the same pressure as the final pressure in the compression chamber and the discharge valve is opened at about 30° before top dead center. In **FIG. 3**, the inertia of the liquid in the flow path increases between curves A to C and in **FIG. 4**, the pressure difference across the spray nozzle in the latter stages of the compression increases from curve D to F. Referring to **FIGS. 3 and 4**, as the gas pressure acts on the liquid, the liquid in the flow path is accelerated at a rate which decreases with increasing inertia. Therefore, as the flow begins, in the early stages of compression, the inertia causes an effective pressure drop by reducing the flow and differential pressure across the spray nozzle. Thus, the inertia can be used to control the rate of increase of liquid flow through the atomiser during compression.

[**0034**] Resistive losses due to the ductwork and spray atomiser will also produce a pressure drop between the pressurised source and compression chamber. This will cause the liquid to begin to decelerate before the compression chamber reaches the source pressure, and this effect is shown in **FIG. 3** by the decreasing flow rate before the discharge valve is opened at 30% before top dead center. However, during deceleration, caused by the increase in pressure in the compression chamber, the liquid inertia reduces the rate of deceleration, producing an effective increase in differential pressure across the atomiser, relative to the case without inertia, thereby prolonging and sustaining a relatively high mass flow through the atomiser as the pressure in the compression chamber reaches its final value and sustaining a mass flow into the compression chamber after the gas pressure has reached its final value.

[**0035**] The inertia, I, of liquid in a duct of length L and internal cross-sectional area A is given by the expression I=L/A.

[**0036**] The pressure drop along the duct due to the inertia is:

$$P_2 - P_1 = I\ dW/dt\ m^{-1}$$

[**0037**] where W is the mass flow rate in kg/s and P is pressure (N/m^2)

[**0038**] Thus, when the liquid is being accelerated by the pressurised source, the inertia causes an effective pressure drop and reduction in acceleration of the mass flow and when the liquid is being decelerated by the rising pressure in the compression chamber and frictional losses in the ductwork, the inertia causes the liquid to react against the decelerative forces causing an effective pressure increase.

[**0039**] Changing the duct length or bore will also affect the duct resistance, so whereas the inertia effect will change the shape of the injection profile, the resistance affects both the magnitude of the atomiser pressure drop and the flow rate.

[**0040**] **FIG. 5** shows two cases where the liquid in the connecting duct between the pressurised source and compression chamber has a similar inertia in both cases, but the lower curve H has a much higher flow resistance and therefore lower flow and lower atomiser pressure drop than curve G.

[**0041**] The resistance is proportional to the duct length and inversely proportional to the fifth power of the duct diameter and so varies in a different way to the inertia. Thus, by choosing appropriate duct cross-sectional areas and lengths, and by choice of valve opening and closing times the desired profile can be obtained. The flow rate can also be reduced by throttling the pressure from the water reservoir which provides a way of raising the flow resistance without affecting the inertia characteristic.

[**0042**] Throttling the flow or having a high flow resistance leads to losses in the liquid supply system. At times this may be acceptable, since the liquid supply reservoir will be at a high pressure and it may not be desirable to use this full pressure across the atomisers all of the time. However, towards the end of the stroke, the losses should be minimised and the inertia effect used to ensure a good pressure drop and flow rate. To minimise the losses, the cross-sectional area of the duct should not be too small, and consequently to ensure a high inertia, the ductwork and duct length will need to be relatively long.

[**0043**] An example of the present invention will now be described with reference to **FIG.6**. A cylinder **1** defines a compression chamber **2** in which gas is compressed by a reciprocating compression piston **3**. The gas to be compressed is fed into the compression chamber **2** through gas inlet line **4** under the control of gas inlet valve **5**.

[**0044**] During the compression operation, water is injected as a spray of fine droplets through upper **6** and lower **7** elevationally spaced annular nozzle manifolds. The arrangement of nozzles is described in more detail in WO 96/16741.

[**0045**] The cool compressed gas together with the injected water are drawn out of the compression chamber **2** through gas outlet line **8** under the control of gas outlet valve **9**. The compressed gas and water are fed to a separator **10** where the gas and water are separated. The gas is removed through line **11** for subsequent heating and expansion. The water is fed to a heat exchanger **12** where it is cooled. The recovered heat may be used in another part of the cycle, or for space heating, or may be dumped. The water leaving the heat exchanger **12** is fed to an accumulator **13**. The accumulator **13**, which contains a compressible gas, can deliver a high flow rate for a short duration into the inertial pipe system. The accumulator acts as the upstream limit of the pipework which provides the inertial flow.

[**0046**] The liquid is fed to the nozzle manifolds **6**, **7** through a system of ducts and valves as described below. A first duct **14** connects the accumulator **13** to a second duct **15** which leads to the upper nozzle manifold **6**. Flow through the first duct **14** is controlled by first control valve **16** and flow through the second duct **15** is controlled by second control valve **17** downstream of the first control valve. The first duct **14** has a relatively low inertia, such that, when the valve **16** opens, the flow in the first duct **14** accelerates rapidly. A third duct **18** consists of a large U bend having a relatively high inertia leading from the accumulator **13** and a fourth duct **19** of shorter length than the third duct **18** connects the third duct **18** to the lower nozzle manifold **7**. The flow through the fourth duct **19** is controlled by a third control valve **20**. No valve is required in the third duct **18** since the flow in this duct begins earlier than the flow in the first duct **14** and can therefore be controlled by second **17** and third **20** control valves. A branch duct **21** connects the first **14** and second **15** ducts to the third **18** and fourth **19** ducts.

[**0047**] The first to third control valves **16, 17, 20** are fast acting valves such as hydraulic spool valves. On the other hand, the valves **5, 9** on the cylinder **1** are typically poppet valves.

[**0048**] Before the injection begins, all three control valves **16, 17, 20** are closed. Injection is normally initiated shortly after the start of compression by opening both second **17** and third **20** control valves. This is shown beginning about at −165° in **FIGS. 7A and 7B**. The control valves **17, 20** can be opened at the same time, or one can be opened before the other. For example, third control valve **20** could be opened sooner since this would allow more time for the water to penetrate to the lower part of the cylinder **1**. Thus, water flows to both nozzle manifolds **6, 7** during the early part of compression, with the flow rate rising as shown in **FIG. 7B**. Since the first control valve **16** is still closed at this time, the flow builds up relatively slowly because of the high inertia of third duct **18**.

[**0049**] As the piston **3** rises up the cylinder **1**, the work rate and therefore the heat input to the cylinder **1** increase. It therefore becomes desirable to increase the water injection rate more rapidly, and therefore the first control valve **16** is opened at about −130°. Since the first duct **14** upstream of the first control valve **16** has a low inertia, the flow rate to both of the nozzle manifolds increases rapidly. **FIG. 7B** illustrates the resulting increase in flow rate, particularly in the upper manifold **6**, to which there is a more direct path from the accumulator **13**. The boost to the flow provided by the opening of the first control valve **16** helps to overcome the increasing flow resistance from the nozzles and the associated pipework.

[**0050**] Near the end of the compression stage, when the work and heat input is near its maximum, the piston **1** begins to obstruct or completely block flow from the lower nozzle manifold **7**. This is typically at around −50° crank angle. Further injection of water from the lower manifold **7** is wasted since most of it simply hits the piston and does not contribute significantly to the cooling. At this time, the third control valve **20** is closed to block the flow to the lower manifold **7**. This also has the effect of forcing all of the water to flow to the upper nozzle manifold **6** providing a boost to this pressure as shown in **FIG. 7A**. As is also apparent from this figure, the pressure in the lower manifold **7** beyond this stage will simply follow the pressure in the cylinder. Since there is still significant kinetic energy in the upstream ducts **14, 15, 18, 19, 21** at this time, the closure of the third control valve **20** provides a further boost to the flow rate to the nozzles of the upper manifold **6** which are arranged to direct the flow into a relatively narrow space at the top of the cylinder **1**. The pressure rise caused by the closing of the third control valve **20** is sufficient to overcome the rapidly rising cylinder pressure and thus the high rate to the upper manifold is extended until the end of the compression at −30° as shown in **FIG. 7A**. The pressure is boosted to such an extent that for crank angles between 40° and 20° before top dead center, water is injected even though the cylinder pressure is above the source pressure.

[**0051**] When the discharge valve **9** opens, it is desirable to shut off the water flow as quickly as possible without causing unacceptable pressures due to water hammer. This may be done by closing the second control valve **16** over an appropriate time scale. Finally, first control valve **16** is also closed in readiness for the next cycle.

[**0052**] It should be noted that this embodiment uses the compressed air in the separator **10** as the source of pressure to drive the injection of the liquid. However, it would be equally possible to derive the pressure for the pumping system from a different source, such as a conventional pump. The function of the accumulator **12**, on the other hand, is to smooth out the pressure fluctuations, so that the inertia of all upstream pipework between the separator and the accumulator does not affect the behaviour of the inertial pumping system. It is likely that an accumulator would still be required in such an alternative system using a conventional pump.

[**0053**] As an alternative to the cooler **12** and accumulator **13**, it is possible to provide a cooler in the bottom of the tank **10**. This could take the form of a coiled pipe through which a coolant liquid flows. As well as eliminating the need for the cooler **12**, this arrangement could also remove the need for the accumulator **13**. In this case, the separator **10** would have to be close enough to the cylinder **1** that the pipework could still be arranged to provide the required inertia.

[**0054**] Although **FIG. 6** indicates that the inertia required by the third duct **18** could be achieved by a single U-bend, this might not be sufficient in some applications. Alternative means of achieving the desired pipe inertia within a confined space include having a number of coils in the pipe, or connecting a number of U-bends and straight lengths together in series to form a serpentine configuration similar to that used in some heat exchanger designs.

[**0055**] Additional nozzle manifolds could be provided with additional valves to control the flow through them. Also additional ducts could be connected to the accumulator **13**. These additional ducts may have different inertias and may also have valves to control the flow in those ducts to start at different times. A single branch duct could connect all the ducts leading from the accumulator **13** to all the ducts leading to the various atomisers. Alternatively, connections could be made selectively, for example involving more than one branch duct.

1. A gas compressor comprising a compression chamber to contain gas to be compressed, a compression piston to compress the gas by movement of the compression piston in the compression chamber, valve means for allowing compressed gas to be drawn from the compression chamber, an atomiser for spraying liquid into the compression chamber to absorb heat from the gas during compression, a pressurised source of liquid and a duct arranged to feed liquid from the pressurised source to the atomiser, wherein the source is arranged to accelerate the liquid through the atomiser into the compression chamber and the dimensions of the duct are sized to define an inertia of the liquid therein to control the rate of acceleration of the mass flow through the atomiser during compression such that the flow rate of liquid into the compression chamber is substantially reduced when the pressure difference between the source and the compression chamber is high and substantially enhanced when the pressure difference between the source and the compression chamber is low.

2. A gas compressor according to claim 1, wherein a valve is provided to control the timing of the flow through the atomiser.

3. A gas compressor according to claim 1 or **2**, wherein the dimensions of the duct are sized to define the inertia such that the mass flow increases through the nozzle in a period of at least 30°, preferably at least 45° and more preferably at least 60° of crank angle of a nominal crank shaft driving the piston.

4. A gas compressor according to any one of claims 1 to 3, wherein the pressurised source comprises a reservoir of liquid pressurised by compressed gas.

5. A gas compressor according to claim 4, further comprising means arranged to feed compressed gas from the compression chamber to pressurise the liquid in the reservoir.

6. A gas compressor according to claim 4 or claim 5, wherein the reservoir is an accumulator, and further comprising a separator upstream of the accumulator which receives gas and liquid from the compression chamber, means to feed the liquid in the separator to the accumulator, and a cooler to cool the liquid being fed from the separator to the accumulator.

7. A gas compressor according to any preceding claim comprising a further duct arranged to feed liquid from the reservoir to the atomiser.

8. A gas compressor according to claim 6, wherein the dimensions of the further duct are sized to define an inertia of liquid therein different to the inertia defined by the dimensions of the duct.

9. A gas compressor according to any preceding claim, comprising a further atomiser for spraying liquid into the compression chamber, the flow through the further atomiser being controlled by a further valve.

10. A gas compressor according to claim 9, wherein a branch duct connects the duct with the further duct.

11. A gas compressor according to claim 10, wherein a third valve is provided in one of the ducts upstream of the location where the branch duct joins the duct, while the valve and further valve are downstream of the branch duct.

12. A method of controlling the temperature of gas in a gas compressor during compression, the gas compressor comprising a compression chamber to contain gas to be compressed, a compression piston to compress the gas by movement of the compression piston in the compression chamber, a pressurised source of liquid and a duct arranged to feed liquid from the pressurised source to the compression chamber, the method comprising the steps of spraying liquid into the compression chamber by allowing the source to accelerate liquid through the duct and controlling the rate of acceleration of the mass flow into the compression chamber during compression by sizing the duct to define an inertia of liquid therein such that the flow rate of liquid into the compression chamber is substantially reduced when the pressure difference between the source and the compression chamber is high and substantially enhanced when the pressure difference between the source and the compression chamber is low.

13. A gas compressor comprising a compression chamber to contain gas to be compressed, a compression piston to compress the gas by movement of the compression piston in the compression chamber, valve means for allowing compressed gas to be drawn from the compression chamber, an atomiser for spraying liquid into the compression chamber to absorb heat from the gas during compression, a pressurised source of liquid and a duct arranged to feed liquid from the pressurised source to the atomiser, wherein the source is arranged to accelerate the liquid through the atomiser into the compression chamber and the dimensions of the duct are sized to define an inertia of the liquid therein to control the rate of acceleration of the mass flow through the atomiser during compression such that liquid can be injected into the compression chamber when the pressure in the compression chamber is greater than the pressure of the source.

14. A gas compressor according to claim 13, wherein the inertia is such that the period during which liquid is injected while the pressure in the compression chamber is greater than the pressure of the source is at least 5°, preferably at least 10° and more preferably at least 15° of crank angle of a nominal crank shaft driving the piston.

15. A method of controlling the temperature of gas in a gas compressor during compression, the gas compressor comprising a compression chamber to contain gas to be compressed, a compression piston to compress the gas by movement of the compression piston in the compression chamber, a pressurised source of liquid and a duct arranged to feed liquid from the pressurised source to the compression chamber, the method comprising the steps of spraying liquid into the compression chamber by allowing the source to accelerate liquid through the duct and controlling the rate of acceleration of the mass flow into the compression chamber during compression by sizing the duct to define an inertia of liquid therein such that liquid is injected into the compression chamber when the pressure in the compression chamber is greater than the pressure of the source.

16. A method according to claim 15, wherein the inertia is such that the period during which liquid is injected while the pressure in the compression chamber is greater than the pressure of the source is at least 5°, preferably at least 10° and more preferably at least 15° of crank angle of a nominal crank shaft driving the piston.

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