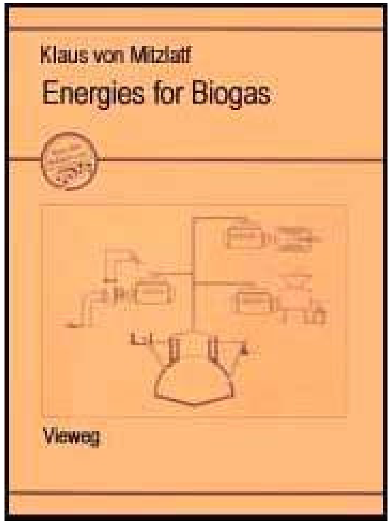


Engines for Biogas (GTZ, 1988, 133 p.)

3. Essential theory on internal combustion engines

3.1 Some Basic Definitions and Relations

The very basic description of an engine and its way of functioning is assumed to be general knowledge for a mechanic, technician or a person willing to engage in the modification and operation of a biogas engine.

3.1.1 Engine Volumina, Vd, Vc, Vtot

The "displaced volume" of one cylinder Vd,c $(1, cm^3)$ is the volume displaced by the piston between its lowest position, the "bottom dead center", BDC, and its highest position, the "top dead center", TDC. The total displaced volume of a

multicylinder engine, Vd,e , is the volume of one cylinder multiplied by the number of cylinders, i:

$V_{d,e} = V_{d,c} \cdot i$ (Equ. 3.1)

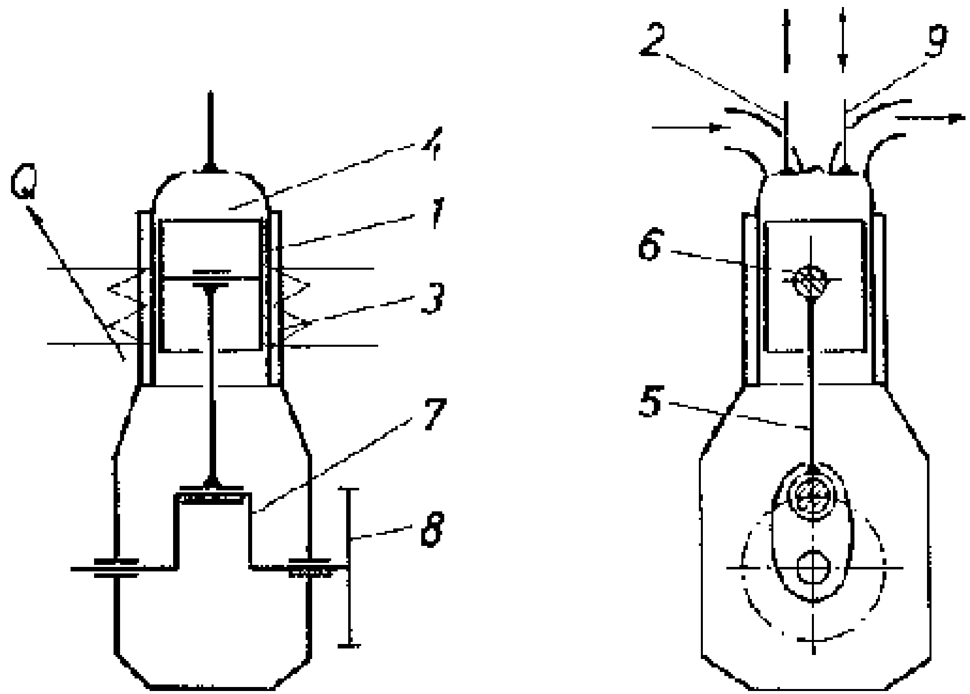


Fig.31:Principal scheme of a 4-stroke engine.1 piston, 2 inlet valve, 3 cylinder, 4 combustion chamber, 5 connection rod, 6 gudgeon pin, 7 crankshaft, 8 flywheel, Q head rejected (cooling).

The volume of the combustion or compression chamber Vc is the volume into which the air or an air/fuel mixture is compressed when the piston has reached TDC. The total cylinder volume Vtot is the sum of the displaced volume and the combustion chamber volume of one cylinder:

$V_{tot} = V_{d,c} + V_c$ **(Equ. 3.2)**

3.1.2 Engine Speed, n

The engine speed describes the number of total (360°) revolutions of the crankshaft in a certain period of time, usually per one minute, i.e. 1/min or rpm.

3.1.3 Power, P

In most cases the power specified for an engine is the mechanical power, which is the mechanical energy (here "torque") transmitted by the crankshaft or flywheel within a certain period of time:

$$P = \frac{\text{torque(kJ)}}{\text{time(s)}} = \text{torque} \times \text{speed(inkW)}$$ **(Equ. 3.3)**

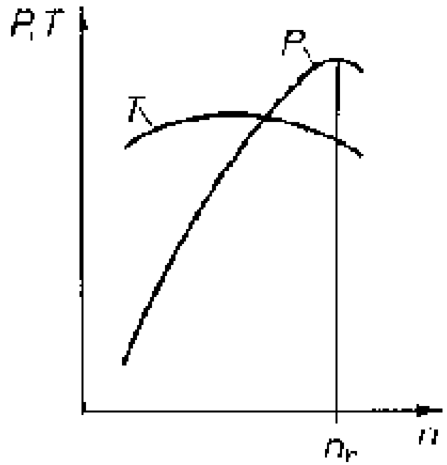


Fig. 3.2: Engine power output P and torque T as a function of engine speed n: nr marks the rated speed.

With a change in engine speed, i.e. the time for one cycle, the power output of the engine changes also. The diagram in Fig. 3.2 demonstrates in principle the course of the torque (i.e. work) and power as a function of engine speed.

Heat energy, delivered by an engine through its exhaust and cooling water/air (normally 60-70%), is often wasted but may also be used for heating or process purposes especially in stationary engines (see Chapter 8 on "cogeneration").

3.1.4 Compression Ratio,

$$\varepsilon = \frac{V_{tot}}{V_c} = \frac{V_{d,c} + V_c}{V_c} = 1 + \frac{V_{d,c}}{V_c}$$ **(Equ. 3.4)**

The compression ratio gives the relation between the total cylinder volume at BDC (Vd,c + Vc) and the volume left for the compressed fuel/air mixture at TDC (Vc). The compression ratio should not be confused with the pressure rise during the compression stroke.

3.1.5 Isentropic Exponent, γ

The isentropic exponent γ is a specific constant of a gas or a gas mixture and is defined as

$$\gamma = \frac{c_p}{c_v}$$ **(Equ. 3 5)**

The exponent describes the theoretical behavior of a perfect gas during a thermodynamic process, e.g. compression and expansion. The theoretical processes are however assumed to be reversible and adiabatic, i.e. have no losses

or other influences from out" side, unlike natural processes.

3.1.6 Polytropic Exponent, n

A technical process like an engine process involves losses. heat transfer and other irreversibilities and cannot therefore be described by the isentropic exponent γ. The polytropic exponent n is used instead. It is a function of the type of gas or gas mixture, the heat transfer from and to the cylinder walls, the mixture of fresh gas with the rest of the burnt gases, etc. Actual values for the polytropic exponent of air and air/fuel mixtures range from n = 1.30 . . .1.36.

3.1.7 Pressure after Compression, Pc (without ignition)

$p_c = p_s \cdot e^n$ **(Equ 3.6)**

The suction pressure Ps is the actual pressure in the cylinder at BDC and is not equivalent to the ambient pressure Pa due to pressure losses in carburetor throttle as well as the inlet channel and valve. As a mean value use

$p_s = 0.9 \cdot p_a \pm 0.05$ **bar**

Example:
- For a direct injection diesel engine with
= 17, p_s **= 0.9 bar, n= 1.3**

$p_c = p_s \cdot {}^n$

$p_c = 0.9 \cdot 17^{1.3} = 35.8$ **bar**

- For a standard Otto engine with = 8.5,

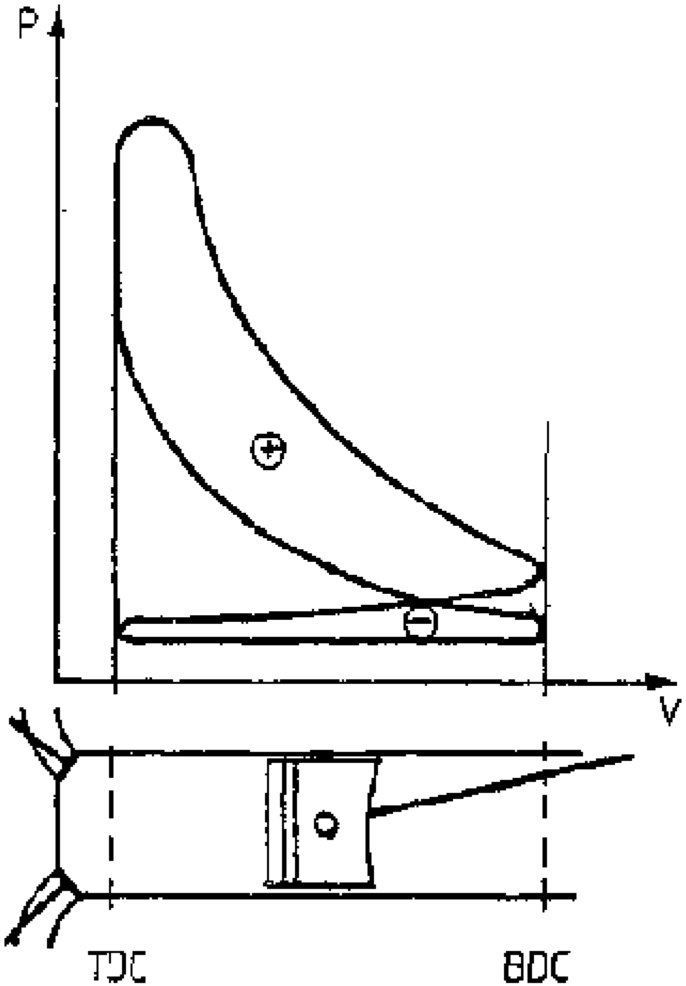
p_s = 0.9 bar, n = 1.35

$p_c = 0.9 \cdot 8.5^{1.35}$ = 14.9 bar

3.1.8 Temperature as a Result of Compression,T_c¹ (without ignition)

$T_c = T_s \cdot {}^{n-1}$ (Equ 3 7)

The suction temperature T_s is not equivalent to ambient temperatures, usually near 293 K (20 °C). The temperature of the air or air/ fuel mixture rises as a result of heat transfer from the inlet channel, cylinder walls and the mixing with the remaining, not exhaust" ed hot flue gas volume from the previous cycle which filled the compression chamber (Vc). As a mean volume: $T_s = T_a$ + 50 K = 323 K ± 20 K.



Figure

Example:

- Diesel engine, = 21, T_S = 330 K, n = 1.3

$$T_C = T_S \cdot e^{n-1}$$

$$T_C = 330\ K \cdot 21^{0.3} = 823\ K\ (= 550\ °C)$$

- Otto engine, = 8.5, T_S = 330 K, n = 1.35

$T_C = 330\ \text{K} \cdot 8.5^{0.35} = 698\ \text{K}$ (= 425 °C)

3.1.9 Necessary Compression Chamber Volume, V_C

$$V_c = \frac{V_h}{\left(p_c / p_s\right)^{1/n} - 1} = \frac{V_h}{\left(T_c / T_s\right)^{1/(n-1)} - 1} = \frac{V_h}{\varepsilon - 1}$$ **(Equ 3.8)**

The equation relates all necessary parameters to the volume of the compression chamber and will be useful in cases where a change of compression ratio is required.

3.1.10 Process Efficiency, η

The efficiency of a process is given by the relation between the useful result and the effort made. In the case of an engine the result is the mechanical power (and the heat flow from cooling water/air and exhaust gas if utilized²), and the effort is the fuel energy consumed by the engine.

$$\eta = \frac{\text{mech. power}(-\text{heat flow})}{\text{fuel energy consumption}} = \frac{P_m\left(+P_h\right)}{E_f}$$ **(Equ.3 9)**

whereby the fuel energy flow/consumptionis given as

$$\dot{E}_f = \dot{m}_f \cdot H_u (\text{in kW})$$ **(Equ.3.10)**

3.1.11 Specific Fuel Consumption, sfc

Another means of describing the efficiency of an engine is the specific fuel consumption, i.e. the fuel input on a mass or volume basis related to the mechanical energy output (P_m):

$$sfc = \frac{\dot{m}_f}{P \cdot t}\left(\text{in } \frac{g}{kWh}\right) \quad \text{mass basis}$$ **(Equ. 3.11)**

$$= \frac{\dot{V}_f}{P \cdot t}\left(\text{in } \frac{m^3}{kWh}\right) \quad \text{volume basis}$$

The specific fuel consumption is often used in engine specifications rather than the efficiency to show the fuel economy of the engine. It differs between engine types and point of operation and is a function of the mean effective pressure, excess air ratio, engine speed, and point of ignition.

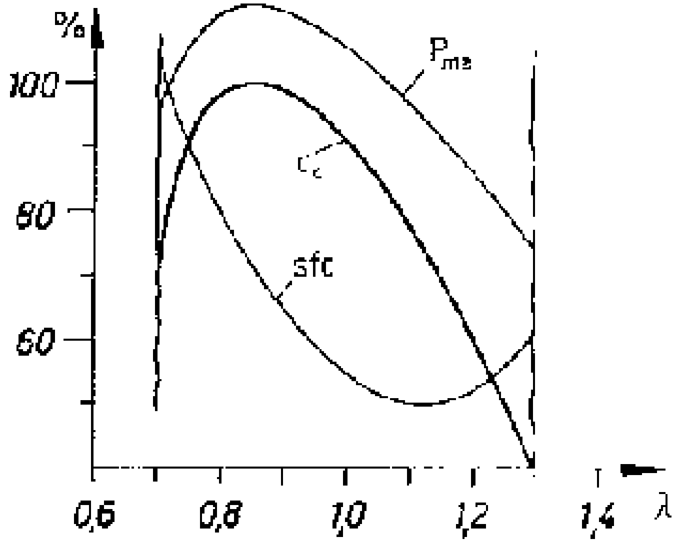


Fig. 3.3: Pressure, volume (p, v) diagram of a 4-stroke engine cycle

3.1.12 Mean Effective Pressure, $p_{m,e}$

**The mean effective pressure is a theoretical value, often used as a means to describe and compare engine performance and economy It is the theoretical pressure needed to be constantly effective onto the pistons on their way down from TDC to BDC to produce the actual
mechanical power of an engine:**

$$p_{m,e} = \frac{P(kW) \cdot 1200}{\Sigma V_d(dm^3) \cdot n(min^{-1})} (\text{in bar})$$ **(Equ. 3.12)**

As in a thermodynamic cycle process the theoretical efficiency rises with the pressure, the actual efficiency or fuel economy of an engine will rise as a function of the mean effective pressure, hence the compression ratio and the cylinder filling.

3.1.13 The 4-stsoke Cycle Process in a p,v-Diagram (Fig. 3.3)

The area marked (+) in the diagram shows the work transmitted from the burning and expanding air/fuel mixture to the piston. The area marked (-) is the work that the piston delivers while expelling the burnt flue gas and sucking in fresh air or air/fuel mixture The process is often shown without the negative work in an idealized form.

3.2. Variable Process Parameters

3.2.1 Combustion of a Fuel in Air

The combustion of a fuel in a mixture with air (or actually oxygen O_2) is an exothermal process in which the chemically bound energy of the fuel is released to generate heat energy while the chemical binding is changed and the combustion product remains at a lower level of energy. For the components of hydrocarbons (i.e. carbon C and hydrogen H) such as petrol, diesel fuel, methane, natural gas, etc. the combustion equations are given in the above table.

Compounds taking part in combustion		Combustion product	Heat energy released
Carbon:	$C + O_2$	CO_2	+406.9 kJ/kmol (Equ. 3.13)
	C + O	CO	+123.8 kJ/kmol (Equ.3.14)³
Hydrogen:	$H_2 + ½ O_2$	H_2O	+242kJ/kmol (Equ.3.15)

The calorific value of a fuel is the sum of the heat energy released from its components at complete combustion. For the calorific values of various fuels refer to table in Appendix II.

For complete combustion a certain relation between the amount of fuel and of oxygen or air is required, the "stoichiometric ratio". Should the air/fuel ratio in a mixture be different from the stoichiometric ratio the combustion will be either incomplete at air shortage, or unutilized "excess air" will be present in the process. A very helpful parameter to describe an actually given air/ fuel ratio is the "excess air ratio" l:

$$\lambda = \frac{\text{actual amount of air}}{\text{air necessary for stoichiometric combustion}}$$ **(Equ. 3.16)**

so that

l= 1 stoichiometric air/fuel ratio
l> 1 air excess (mixture lean)
l< 1 air shortage (mixture rich)

The best combustion performance will always occur at values near l = 1. Mixtures at values below l = 0.5 rich or above l = 1.5 lean usually do not properly ignite from an ignition spark. The supply of the right mixture of air and fuel is therefore of utmost importance for the performance of a spark ignition (Otto) engine. Diesel engines can however operate at high excess air ratios (l= 1.5 . . . 4.0) as the fuel is injected into the combustion chamber in a liquid form and the combustion takes place around the circumference of the fuel spray droplets.

The droplets evaporate and mix with the surrounding air. At a certain distance from the core a stoichiometric mixture will automatically be established. This is where the combustion takes place.

In a still or laminar flowing gaseous air/fuel mixture the burning velocity has a maximum at l = 0.9 but decreases when the mixture is richer or leaner.

In order to adapt the velocity of the combustion process to the velocity of the engine cycle the point or crank angle at which ignition is initiated needs to be varied in relation to the excess air ratio. Lean mixtures with a slower burning

velocity require an earlier (i.e. more advanced) point of ignition to ensure that the combustion pressure peak occurs at an optimum crank angle after the piston has passed TDC. Richer mixtures com-bust faster so that the ignition point should be retarded accordingly.

The further influences on the ignition timing are explained hereunder.

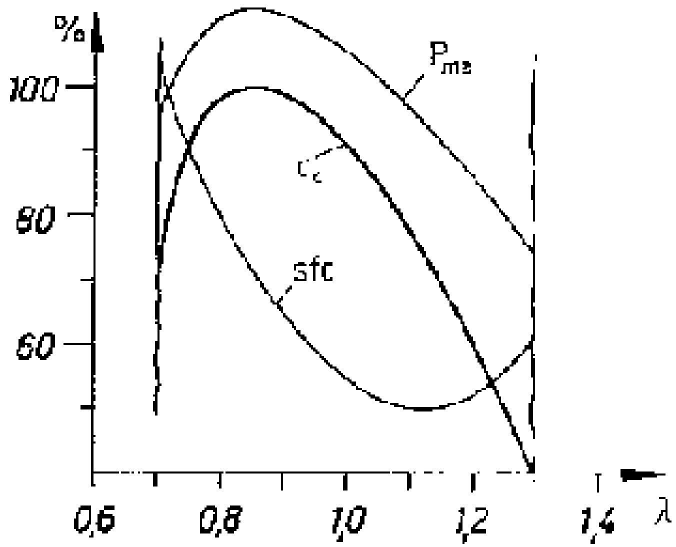


Fig. 3.4: Mean effective pressure Mme, combustion velocity c_c and specific fuel consumption sfc as a function of excess air ratio l

3.2.2 Combustion Velocity and Ignition Timing

The velocity of the combustion of the air/ fuel mixture during one combustion stroke is essential for the performance of an IC engine. The time available for the (complete) combustion of the air/fuel mixture is extremely short, e.g. for an engine operating at a speed of n = 3000 min^{-1} the time for one combustion stroke

is 1/100 s.

The combustion begins at its ignition source, either a spark-plug (Otto engine) or the spray droplets (diesel engine), and takes some time to fully develop. The pressure then develops in such a way that the pressure peak occurs shortly after the piston has reached TDC. The high pressure after TDC causes a high force onto the piston. The mean effective pressure, hence the work output, results from the course of the pressure between TDC and BDC. Premature ignition or too high pressure before TDC will consume extra work (or power) from the piston as it needs to compress against the burning and expanding gas mixture.

Delayed ignition or slow burning of the air/ fuel mixture will have the effect that the mixture still burns when the combustion stroke is finished and the exhaust valve opens. Not only will the valve get unnecessarily hot and may be damaged but a lot of fuel energy will be lost with the still burning exhaust gases. This part of the fuel energy cannot contribute to the production of mechanical energy.

The timing of spark ignition or injection of diesel fuel is found as a compromise between premature and delayed ignition, both resulting in a power loss. The timing as related to the burning velocity is however dependent on some operational parameters:

-engine speed n,
-engine load P,
-excess air ratio l (see Chapter 3.2.1),
-type of fuel used,
-pressure and temperature.

The combustion velocity of an air/fuel mixture rises significantly as a function of its actual temperature and pressure.

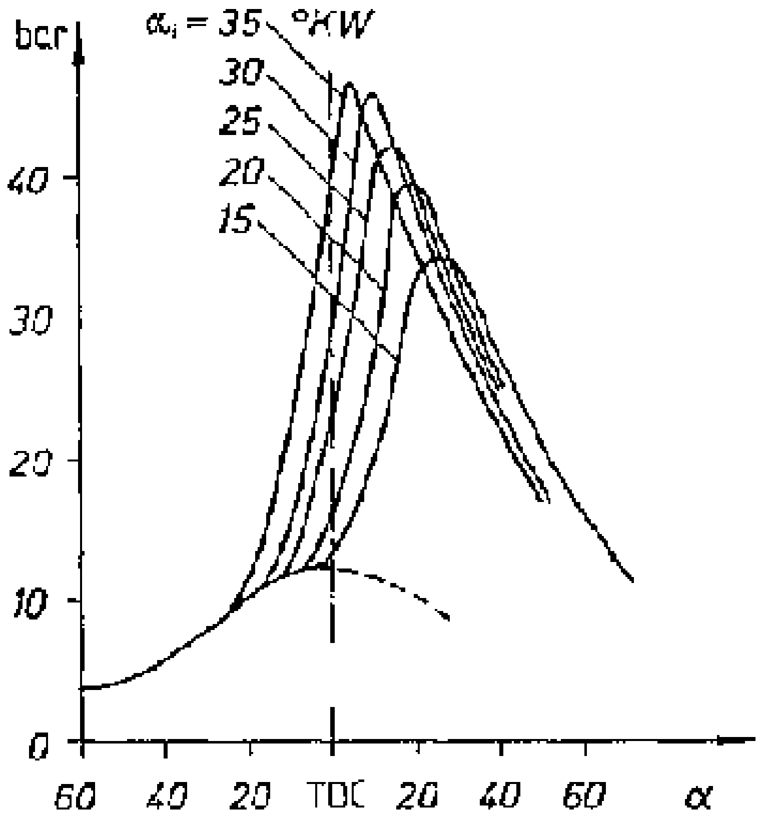


Fig. 3.5: Course of pressure as a function of the crank angle α (point) of ignition α_i

3.2.3 Engine Speed

With increased engine speed the time for combustion becomes shorter, but the time for development of combustion and pressure does not similarly shorten. In order to prevent the pressure peak occurring too far behind TDC (pressure and power loss) the ignition point is advanced. This is usually done by a centrifugal

force mechanism (not commonly used in stationary diesel engines).

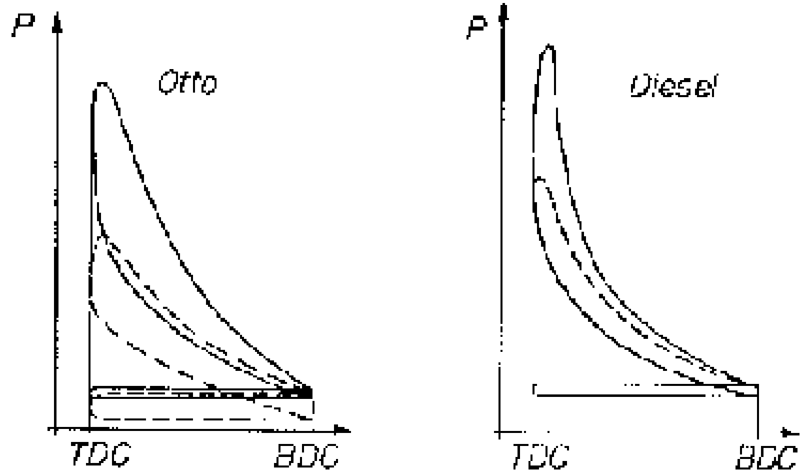


Fig 3.6: Partial load (- - -) and full load (____)p, v-diagrams for Otto and diesel engine

3.2.4 Partial Load (Cylinder Filling)

The suction pressure in the cylinder is usually lower than the ambient pressure due to flow resistance in air filter, inlet valve and, in the case of an Otto engine, the position of the throttle for power control. When the throttle is in a controlled position it provides an additional depression which causes the suction pressure p_s to decrease subsequently. The amount of air/fuel mixture on a mass basis filled into the cylinder will therefore be lower. This leads to a drop in mean effective pressure $p_{m,e}$ and power output.

The combustion velocity at lower cylinder filling rates and at lower pressures is also reduced so that in order to compensate the ignition timing needs to be further advanced in partial load (throttled) operation. This is done in relation to the

suction pressure behind the throttle. A simple diaphragm is used to operate the advancing mechanism accordingly.

Low ambient pressures at higher altitudes have an effect similar to throttling so that the power output of an engine drops at a rate of about 10% for each 1000 m in altitude.

The partial load behavior of an Otto engine is characterized by the larger negative work needed to overcome the additional resistance of the throttle in the suction stroke. Diesel engines are not air-throttled in partial load, and hence only their pressure and power output is reduced. Otto engines therefore have the disadvantage of a reduced efficiency in partial load operation because of the reduced cylinder filling.

The cylinder filling is however further influenced by the flow resistance of the inlet manifold, duct and the inlet valve itself. Even at a 100% opening of the throttle valve the cylinder usually only receives a reduced amount (on a mass basis) of what it can theoretically contain, i.e. the mass that can be filled into the cylinder volume V_h at ambient conditions. Each engine type has, by its original design, a built-in "supply efficiency" (sometimes called "volumetric efficiency") defined as

$$\eta_{vol} = \frac{\text{m actually supplied}}{\text{m theoretically suppliable}}$$ **(Equ. 3.17)**

where m = mass of air or air/fuel mixture.

Unless the actual value for η_{vol} is known, take η_{vol} = 0.85 as an average value. The supply efficiency is essential for the determination of the actual air or air/fuel

mixture sucked into the engine and the design of mixers for fuel gas and air. When an engine is operated at a lower than its rated speed the flow through the inlet is reduced, hence the flow resistance, so that the volumetric efficiency increases with an operational speed decrease.

3.2.5 Interdependence of Load and Speed

The actual point of operation of an engine is determined by the load (or power demand) and the power produced from the fuel input at a certain torque and engine speed. The point of operation is established as a balance of power supply and power demand.

The power produced is not only a function of the amount of air and fuel supply, hence of the resulting effective pressure after combustion, but also of the actual engine speed (see Fig. 3.2 and Equ. 3.3).

The power demand from a driven vehicle or a machine can be subject to changes. On the other hand the operator may wish to operate the vehicle or machine at another speed or power output. When the load rises, the speed of the engine will fall until the load also decreases and a new balance is. found. Should the load remain constantly high, the engine will further decrease speed and finally come to a halt. When the power demand decreases, the engine will increase its speed until an increase in power demand occurs. If the demand remains low, the engine can speed up and even be damaged unless the fuel input is reduced. Most driven machines however increase their power requirement with a speed increase and decrease it with a speed decrease. Subsequently with a change in power

requirement the engine will then find its new balance and continue its operation at a different speed.

Some driven machine types perform sufficiently well even at a speed different from the exactly specified one (see Chapter 7.4). Others however need to operate at one single speed only. Should the load on the engine and subsequently the speed change, a change of fuel input to the engine can compensate for the change in load so that the engine continues operation at the speed required. When the load rises, an increase of fuel (or air/fuel mixture) is needed to cause an increase of the power output until the former speed is reached again. A decrease in load must accordingly be compensated by a decrease in fuel input. Most engine control systems use the change in speed to sense a change in load and operate the fuel supply system accordingly.

3.2.6 Type of Fuel Used

The burning or flame propagation velocity of an air/fuel mixture largely depends on the type of fuel used. Some gases, especially methane, have a slow burning velocity. This becomes visible in biogas cookers where the velocity of a slight air draft may be faster than the burning velocity and carry away the flame from the burner ring. Even though the burning velocity of an air/methane mixture under higher pressure and temperature is much higher than in atmospheric conditions, it is lower than the velocity of gasoline or diesel fuel mixtures with air.

In order to fully utilize the fuel energy during the combustion stroke and to achieve a good combustion process with the pressure peak optimally after TDC, it will be necessary to advance the ignition timing in Otto (spark ignition) engines

when biogas is used. Changing the injection timing in diesel engines when operated with biogas requires a more difficult operation and can often not be done without modifying a few parts, e.g. gears. The operation would also have to be reversed in any case of biogas shortage where the proportion of diesel fuel increases accordingly.

3.3 Relevant Engine Types

In principle all internal combustion engines can be operated with liquid fuels (which are in vapor/gaseous form when they ignite) or with gaseous fuels. The given framework of this publication however calls for the narrowing of the scope of engines towards types that can be modified and operated with acceptable efforts:

-Power range to abt. 50 kW;
- Engines considered should be based on standard engine types produced in larger series;
-2-stroke engines, as the smaller types do not nave a very good reputation for long engine life and often use lubrication in a mixture with the liquid fuel. This excludes the use of a gaseous fuel. (Larger 2-stroke diesel engines range at power outputs of 500 kW and more and are usually individually projected and expensive units);

- No gas turbines as they are comparatively expensive and require sensitive operation and maintenance;
- No rotary piston (Wankel) engines because of generally bad reputation for reliability and engine life;

- No turbocharged engines because of their relatively sophisticated control systems.

The engine types to be considered here are therefore:

- Otto (gasoline) engines, 4-stroke;
- diesel engines, 4-stroke.

The specific features of these two engine types are explained in more detail in the following chapters. A comparative summary is given in Appendix III.

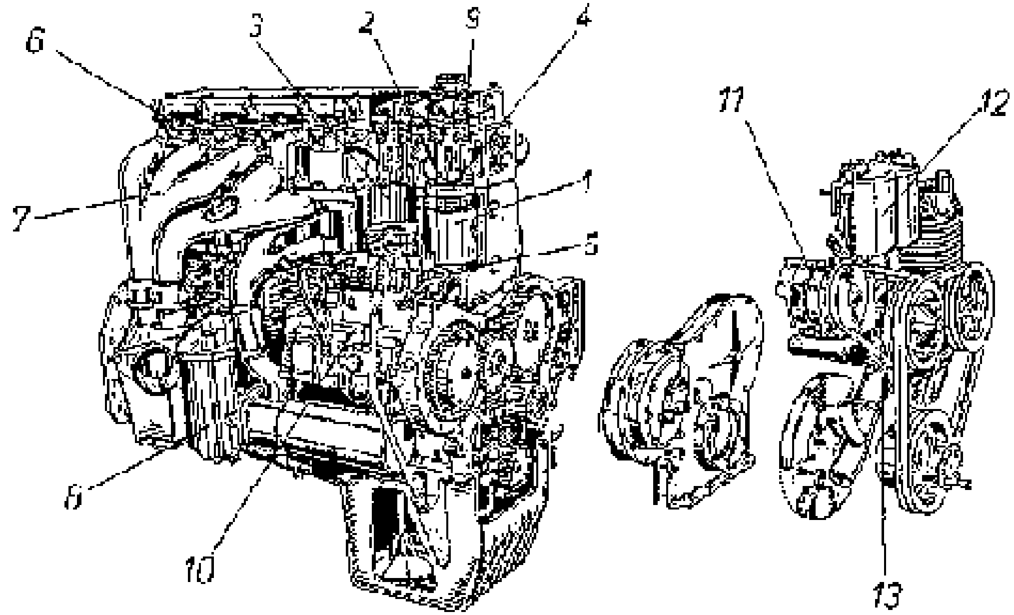


Fig. 3.7: 6-cylinder diesel engine, partly opened (MAN).

1 piston, 2 inlet valve, 3 cylinder, 4 combustion chamber, 5 connection rod, 6 injector nozzle, 7 suction mainfold, 8 oil filter, 9 outlet valve, 10 injector pump, 11 alternator,12 fuel filter, 13 cooling water pump.

3.3.1 Diesel Engines

3.3.1.1 The Diesel Process

The diesel engine and its process are shown in the diagrams Figs. 3.7 and 3.8. The engine sucks air at ambient conditions and compresses it to a pressure around 60 bar and above whereby the air reaches temperatures around 600°C. Shortly before the piston reaches TDC, fuel is injected and ignites immediately at these conditions. An external source for ignition is usually not necessary. Only at low ambient temperatures a"glow plug" is sometimes used to facilitate the start-up. The point or crank angle £i of injection is chosen (£i about 25°) considering that the pressure rise through combustion reaches a peak shortly after the piston has passed TDC.

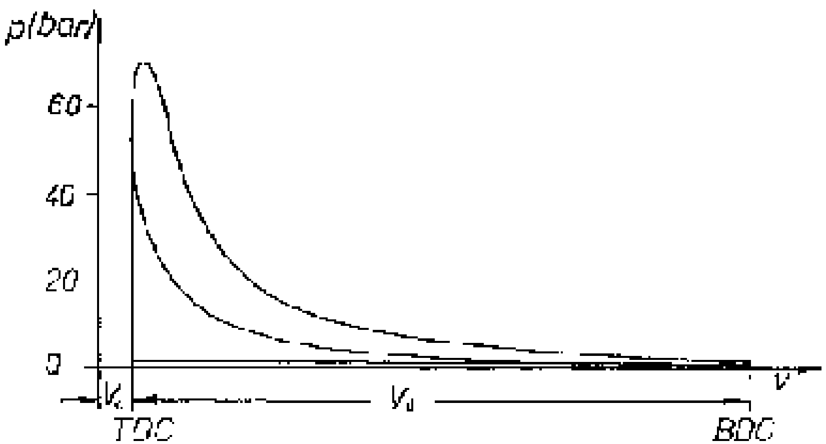


Fig. 3.8: Simplified p, v-diagram of a diesel process

3.3.1.2 Operational Parameters and Control

In a diesel engine the air/fuel mixture is prepared within the cylinder by the

injection of a certain amount of diesel fuel into the air during its compression by the piston. The spray droplets ignite immediately when they come into contact with the hot air. The point or crank angle £i of ignition is almost identical with the crank angle of injection, usually around 25° before TDC (Fig. 3.9).

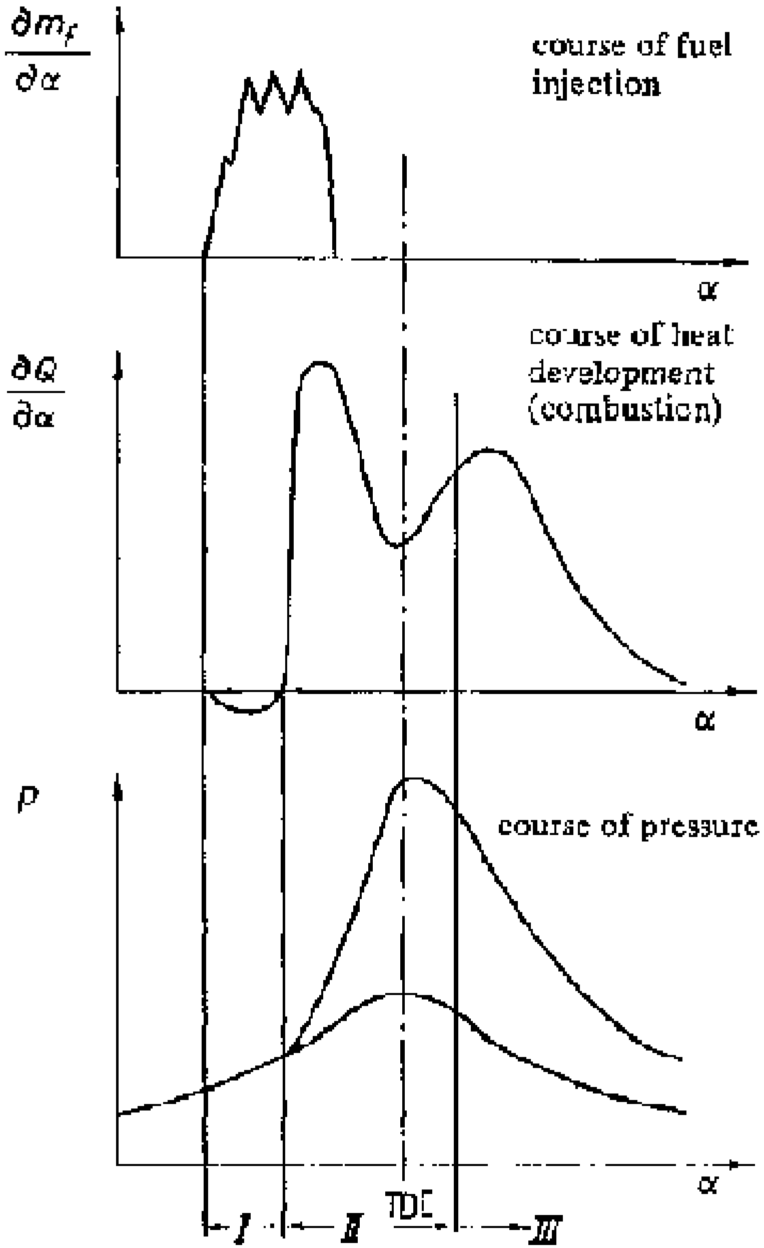


Fig. 3.9: Courses of fuel injection, combustion and pressure as a function of the crank angle α

In the first phase (I) injection of fuel begins but some time is needed for part of the fuel to evaporate and form a combustible air/fuel mixture. In the second phase (II) the fuel begins to ignite while the injection still continues. The start of combustion results in a sharp increase in heat and pressure. In the third phase (III) the combustion of the more slowly combusting parts, mainly the carbon components, takes place.

The diesel fuel is injected by the injection system, an example of which is shown in Figs. 3.10 and 3.11.

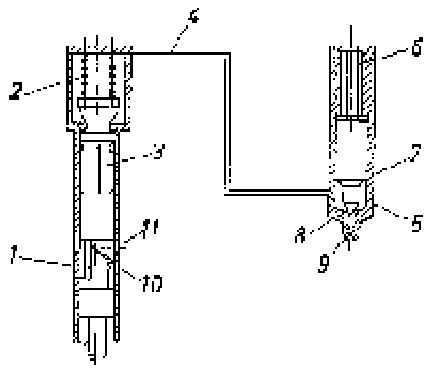


Fig.3.10: Principal scheme of injection system (numberscorespond to text).

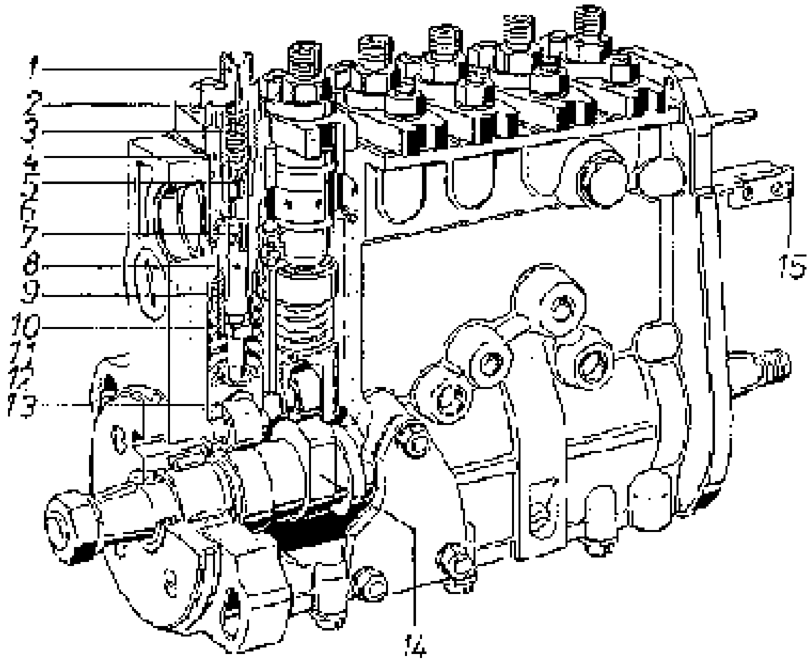


Fig. 3.11: Diesel injector pump (Bosch).

1 valve holder, 2 filling piece, 3 valve spring, 4 pump cylinder, 5 valve, 6 suction and control bore, 7 oblique control edge, 8 plunger, 9 control bush, 10 plunger lug, 11 piston sprig, 12 sprig holder, 13 roller shaft, 14 cam, 15 control rack.

The plunger (Fig. 3.10) is moved up and down by a camshaft which is in direct gear with the crankshaft of the engine in order to forward the fuel at the required crank angle. When the plunger (1) is pushed upwards, the fuel is pressed against a valve (3) which is springloaded (2) and moves against the spring to open, passes the injector pipe (4) and enters the injector (5). As pressure rises in the space underneath the injector needle (7), which is also springloaded (6), the needle moves upwards from its seat (8) and fuel passes the fine bores (9) to enter the cylinder in a well distributed spray.

Control of engine power is effected by variation of the amount of fuel injected. The plunger (1) can be turned so that when it is moved upwards the oblique pitch of the control edge will give way to the fuel intake bore (11) according to its axial and angular position. As soon as the pitch of the control edge has reached and opened the bore (11) the injection pipe and nozzle are rendered pressureless and the injector needle (7) immediately closes the spray jets. In a multicylinder engine all plungers are connected to a common rack and are turned simultaneously for control.

The speed control is effected using the above mechanism within the injection pump and a mechanically controlled governor (centrifugal weights). As long as the required engine speed is not yet reached, the plungers supply the maximum amount of fuel to the injectors so that the engine power, and hence the speed, increases (unless engine is overloaded). As it reaches the required speed the governor operates the rack, the plungers are turned and reduce the amount of fuel injected until power and speed are balanced as required. When the load increases further, the speed will automatically drop, but a small decrease in speed effects a change in the governor which operates the rack in such a way that more fuel is injected until the required speed is reached again. For a decrease in power the system works accordingly.

All diesel engines are equipped with governors. The governor can be tuned, modified or even disconnected from the injector pump when the engine shall be operated to run on other fuels. Such modification however requires careful handling and sufficient experience and expertise. A more detailed description of diesel engine modifications is given in Chapter 5.

In order to maintain the required conditions (p, t) after compression the airflow at the inlet to the diesel engine is not controlled, i.e. there is no throttle or choke. A throttling or decrease in suction pressure would lead to a decrease in pressure after compression and to a decrease in temperature (see Equ. 3.6/3.7). This would have a negative effect on the combustion, the mean effective pressure Pm,e and the control. In extreme cases it could even make the necessary self" ignition impossible. Diesel engines therefore always have unthrottled air inlets, also when operated with gas in "dual fuel" mode.

Due to the higher compression ratio (e = 16 . . . 22) diesel engines operate at a relatively high efficiency, i.e. ηtot = 0.3 . . . 0.4, and low specific fuel consumption, i.e. sfc = 250 . . . 300 g/kWh at rated conditions. Diesel engines, unlike Otto engines, enjoy a comparatively high efficiency in partial load operation also, i.e. the specific fuel consumption does not significantly increase in partial load. They are therefore very suitable for operation under conditions of varying power demands. They also enjoy long engine life such as 20 000 . . . 30 000 hours or even longer before-an overhaul is necessary and are found on the market in standard series and large numbers for stationary and vehicle purposes.

Diesel engines are designed according to different philosophies concerning the combustion and combustion chamber forms (Fig. 3.12).

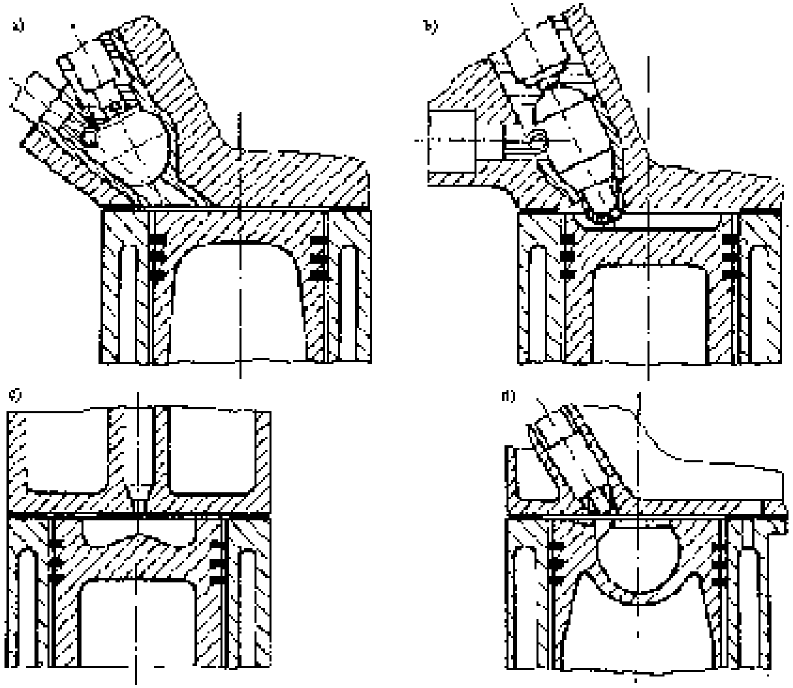


Fig. 3.12: Different combustion chamber forms and methods of fuel injection. a) swirl chamber, b) antechamber, c) direct injection, d) MAN method

The direct injection type can be best modified to use (bio)gas as
- the compression ratio is relatively low (e = 17); a higher compression ratio would lead to higher temperatures at which the gas/ air mixture could self-ignite in an uncontrolled manner at the wrong time and severely affect the performance and life of the engine,
- the even shape of the combustion chamber is optimal for gas/air combustion.
- conversion to Otto process is eased by an advantageous position for the spark plug (i.e. former position of injector nozzle) and by an easily executed reduction of the compression ratio to values of c - 10 . . . 12. For a more detailed description of diesel engine modification refer to Chapter 5.

3.3.2 The Otto Engine

3.3.2.1 The Otto Process

The Otto engine and its process are shown in Figs. 3.13 and 3.14.

**The Otto engine sucks a readily composed mixture of air and fuel.
The mixture is compressed to pressures around 20 bar and temperatures around 400 °C (see Equ. 3.7). At these conditions the mixture cannot selfignite. A spark plug is used to ignite the mixture at a suitable moment or crank angle before TDC for optimum performance.**

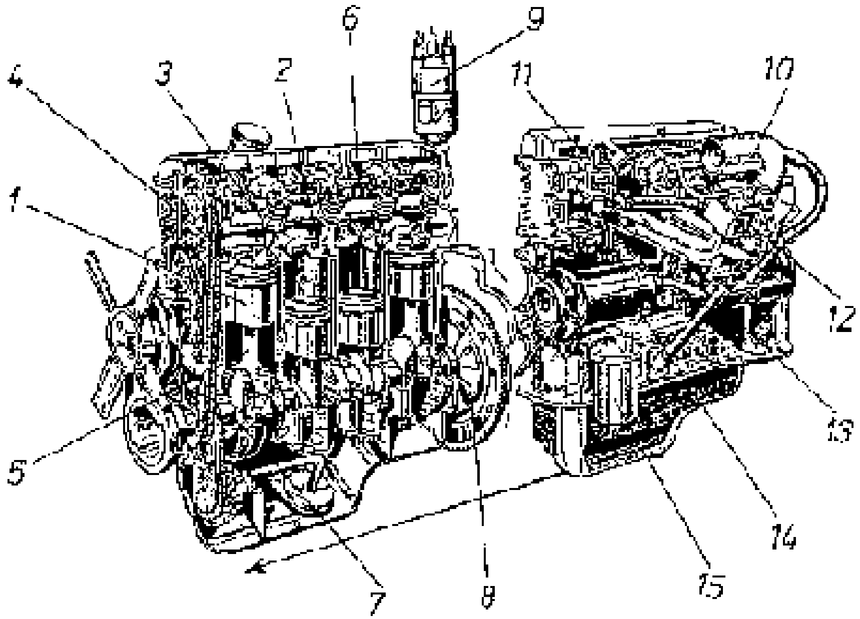


Fig. 3.13: Otto engine, partly opened (BMW).

1 piston, 2 inlet valve, 3 cylinder, 4 combustion chamber, 5 connection, 6 overhead camshaft, 7 crankshaft bearing, 8 flywheel, 9 distributor, 10 suction

from air filter, 11 suction mainfold, 12 carburetor, 13 starter motor, 14 generator, 15 oil filter.

3.3.2.2 Operational Parameters and Control

Otto engines in vehicles are usually operated at varying conditions of speed and load. In order to keep performance optimal at all conditions the point of ignition is changed in relation to engine speed and suction pressure. The crank angle can vary as much as from 7° before TDC to 40° before TDC according to the actual point of operation.

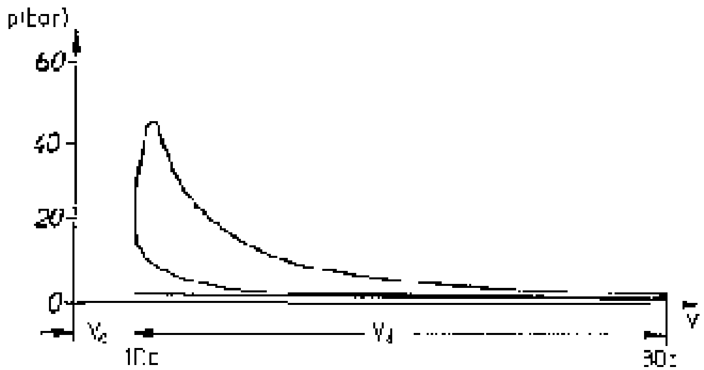


Fig. 3.14: Simplified p, v-digram of an Otto process

If a liquid fuel is used, the air/fuel mixture is usually prepared in a carburetor. The carburetor by its design ensures an almost constant air/fuel ratio at any airflow rate. The power and speed control of the engine is effected through a throttle valve integrated into the carburetor housing which allows the varying of the inlet mass flow of the mixture by its degree of opening. The throttle valve causes a certain pressure drop of the mixture due to which the cylinder filling is reduced on a mass basis (the volume flow remains constant). The pressure drop at the

throttle valve causes a subsequent drop in the suction and mean effective pressure, hence a drop in power and efficiency. Otto engines therefore have a higher specific fuel consumption in partial load than diesel engines where the airflow is not throttled and the Pm,e is only affected by the amount of fuel injected. Otto engines should preferably be operated in a slightly throttled or unthrottled mode for optimum fuel economy, especially in continuous service.

Another alternative for the preparation of the air/fuel mixture is the injection of liquid fuel into the suction channel where it mixes with the airstream before entering the cylinder. The amount of fuel injected is related to the amount of air sucked into the engine and electronically controlled. The airflow is controlled via an air throttle valve. Injection systems are more sophisticated than carburetion systems but provide a more accurate relation between engine operation and fuel mixture through electronic control' and hence better fuel economy. Direct fuel injection is not very common in Otto engines.

As mentioned before, the actual air/fuel ratio is an important parameter for the engine performance. Excess air ratios near λ = 1 are required whereby at
- λ = 0.9 the power produced is at a maximum but a certain percentage of incomplete combustion has to be taken into account
(i.e. formation of toxic CO).
- λ = 1.1 . . . 1.15 the fuel economy is at a maximum; the CO content in the exhaust gas is almost zero. Nitrogen oxide NO_X increases however (toxic).
- λ = 1.3 the mixture loses ignitability.

3.3.2.3 Design Parameters

The compression ratio of an Otto engine is a function of the fuel used. Higher compression ratios result in higher temperatures of the air/fuel mixtures. This may cause uncontrolled self-ignition and an uneven combustion process, both disadvantages for engine performance and life span. Usual compression ratios are

- for standard petrol: = 7 . . . 8.5
- for superpetrol: = 8.5 . . . 9.5
- for gas (CH4, LPG): = 10 . . . 12

Compression ratios higher than e = 12 are not recommended as

- accurate spark plug function cannot be assured and
- fuels such as LPG and natural gas tend to self-ignite at higher pressures, depending on their composition.

With the lower compression ratio than a diesel engine the mean effective pressure of an Otto engine is lower, as is its overall efficiency. Values of η = 0.25 . . . 0.32 are common. As an Otto engine for a vehicle can operate at higher speeds, its power output in relation to its displaced volume is however higher than that of a similarly sized diesel engine. While an Otto engine appears cheaper at first sight, it will have a shorter life expectancy.

3.3.2.4 Gas Otto Engine

Gas Otto engines are designed for a variety of gaseous fuels. Specific types with a higher compression ratio than e = 12 for the use of methane alone are not found on the market. The point of ignition and the mixing devices will however have to be adapted for the type of gas, mainly its calorific value.

Gas Otto engines receive their air/gas mixtures from gas mixing valves, venturi mixers or, in the simplest case, gas mixing chambers. Apart from the different type of air/fuel mixing system they follow the same criteria and parameters as Otto engines for liquid fuels.

Gas mixing valves coordinate the supply of both air and gas by a diaphragm that opens the air and gas inlets in relation to the pressure in the space between throttle valve and air inlet. The air/fuel ratio is determined by the size of the internal openings for air and for fuel respectively. Fine calibration can be achieved by changing the pressure, hence mass flow, with the help of an adjustable throttle at the gas inlet. The gas pressure at the inlet to the mixing valve is usually low (20 . . . 50 mbar). Gas supply from sources with higher pressure, i.e. Iiquid or compressed gas from storage cylinders, will have to be reduced by reduction valves before entering the mixing valve.

Venturi mixers utilize the velocity increase and subsequent pressure reduction in a flow through a tube with a contraction. The pressure at the smallest cross-section area is a function of the air velocity, hence the air volume flow. Fuel gas enters and mixes with the airstream at the smallest cross-section (the "bottleneck"). An almost constant air/ fuel ratio is thus achieved.
A more detailed description as well as design parameters are given in Chapter 6.

Mixing chambers are the simplest devices for mixing air and fuel. The chamber can either be a simple T-joint of two tubes or can be a chamber of a larger volume with one inlet each for air and fuel gas and an outlet for the mixture of both. However, air and fuel are not supplied in a constant ratio independent of the suction of the engine, but have to be controlled by external valves. Such mixing

devices can therefore not easily be used for automatic speed and power control but can function in a fixed setting if the engine is operated at one steady condition only.

In principle every Otto engine can be operated on gas. The conversion of a petrol engine into a gas engine is often done when gas, mainly LPG, is found to be cheaper than petrol or for lift trucks operated inside storage halls (fuel gas is less contaminated than petrol fuel and produces a less dangerous exhaust gas). The modification is described in more detail in Chapter 6.

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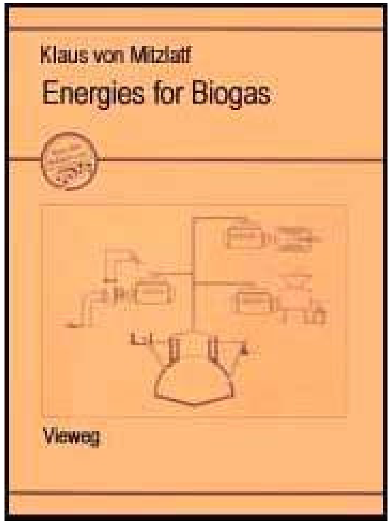


Engines for Biogas (GTZ, 1988, 133 p.)

4. Biogas and its Properties as a Fuel for Internal Combustion Engines

4.1 What is Biogas?

Biogas originates from bacteria in the process of biodegradation of organic material under anaerobic conditions. It consists of a varying proportion of CH_4 (methane) and CO_2 (carbon dioxide) and traces of H_2S, N, CO, O, etc. The content of CH_4 and CO_2 is a function of the matter digested and the process conditions like temperature, C/N ratio, etc. Methane is the most valuable component under the aspect of using biogas as a fuel; the other components do not contribute to the calorific ("heating") value and are often "washed out" in purification plants in order to obtain a gas with almost 100% CH_4. For further details of biogas production the use of the respective literature is recommended [3, 4, 5, 6].

4.2 Energy Content of Biogas

The useful part of the energy of biogas is the calorific value of its CH_4 content. The other components have strictly speaking an energy content also but they do not participate in a combustion process. Instead of contributing they rather absorb energy from the combustion of CH_4 as they usually leave a process at a higher temperature (exhaust) than the one they had before the process (mainly ambient temperature).

The following are the thermodynamic parameters of CH_4 at standard conditions[1] (i.e. 273 K, 1013 mbar=0.1013 MPa):

- specific treat c_p = 2.165 kJ/kg K,
- molar mass M = 16.04 kg/krnol,

- density ρ =0.72 kg/m³,
- individual gas constant R =0.518 kJ/kg·K,
- lower calorific value
H_u = 50000 kJ/kg,
$H_{u,n}$ = 36000 kJ/m³n.

The actual calorific value of the biogas is a function of the CH_4 percentage, the temperature and the absolute pressure, all of which differ from case to case. The calorific value of the biogas is a vital parameter for the performance of an engine, a burner or any other application using biogas as a fuel. The calculation of the calorific value can be done using the standard thermodynamic relations for gases:

-Standard gas equation
$p \cdot V = m \cdot \rho \cdot T$ (Equ. 4.1)

-isentropic exponent
$\gamma = c_p/c_v$ (Equ. 4.2)

-specific gas constant
$R = c_p - c_v$ (Equ. 4.3)

-constant volume process(v=constant)
$\frac{\rho_2}{\rho_1} = \frac{T_1}{T_2}$ **(Equ. 4.4)**

- constant pressure process (p = constant)

$$\frac{r_2}{r_1} = \frac{v_1}{v_2} = \frac{T_1}{T_2}$$ **(Equ. 4.5)**

- constant temperature process (T = constant)

$$\frac{p_1}{p_2} = \frac{v_2}{v_1} = \frac{\rho_1}{\rho_2}$$ **(Equ. 4.6)**

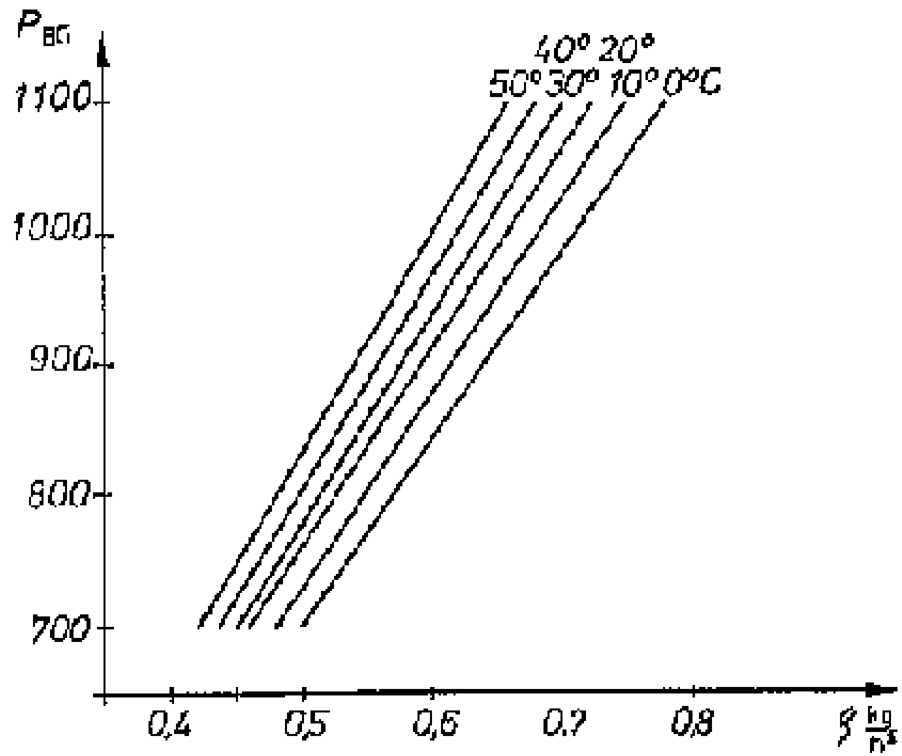


Fig. 4.1: Density ρ of CH_4 as a function of biogas pressure and temperature

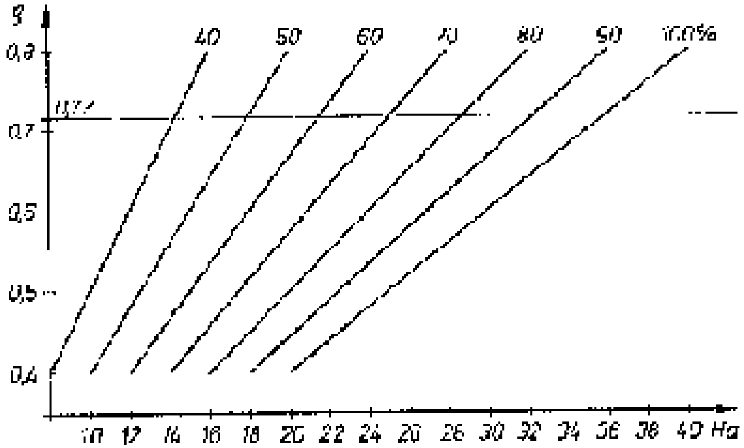


Fig. 4.2: Calorific value of biogas as a function of the density and volume %-age of its CH_4 content (ρ=0.72 is the density at a standard condition)

The graphs (Figs. 4.1, 4.2) will facilitate an easy determination of the density of the CH_4 component in a first step and the calorific value of the biogas in a second step. Use the diagrams as follows:

-Determine the actual density ρ of the CH_4 in the biogas using the actual biogas temperature and pressure (ambient pressure + biogas plant pressure (gauge) or pressure measured at inlet to the mixing device).

- Find the actual calorific value using the density and the percentage of CH_4 in the biogas mixture.

A precise calculation of the calorific value can be done following the example below.

Example:

Calculation of the calorific value of biogas at the following conditions:
-composition:

CH_4 = 60% Vol, i.e. VCH_4/V_{tot} = 0.6
CO_2 = 40% Vol, i.e. VCO_2/V_{tot} = 0.4

Traces of other components negligible

-temperature:	T = 298 K (= 25 °C)
-pressure, ambient:	P_a = 950 mbar
-pressure in biogas plant:	p_p = 20 mbar, gauge

Step 1: total pressure of biogas

$P_t = 950 + 20 = 970 \text{ mbar} \triangleq 0.97 \cdot 10^5 \text{ Pa}$

If humidity of biogas was not considered in the gas analysis so far, the value has to be corrected using the diagram in Fig. 4.3 and the related example.

Step 2:. density ρ of CH_4 in mixture at actual pressure p and temperature T, calculated on the basis of the table values at standard conditions

- temperature correction: $\rho_2 = \rho_1 \frac{T_1}{T_2}$

- pressure correction: $\rho_2 = \rho_1 \frac{p_2}{p_1}$

- $\rho_{CH4act} = \rho_{CH4,std} \cdot \frac{p_{act}}{p_{std}} \cdot \frac{T_{std}}{T_{act}}$ **(Equ. 4.7)**

$$= 0.72 \cdot \frac{970mbar}{1013mbar} \cdot \frac{273K}{298K} = 0.63kg/m^3$$

Step 3: actual calorific value of given biogas

- $H_{u,act} = \frac{V_{CH4}}{V_{tot}} \cdot \rho_{CH4,act} \cdot H_{u,n}$ **(Equ. 4.8)**

= 0.6·. 0.63 kg/m³· 50 000 kJ/kg
= 18900kJ/m³

Compare with value obtained when using the diagrams in Figs. 4.1 and 4.2.

Biogas emerging from the plant is usually fully saturated with water vapor, i.e. has a relative humidity of 100%. Depending on the course of the gas piping between plant and consumer, part of the water vapor will condense when the gas is cooled. The humidity can be reduced by cooling and warming again of the gas with a drain trap for the condensate at the cooler.

The gas analysis often either does not consider the humidity or it is done at the plant, not at the consumer. In those cases the humidity needs to be considered for the establishment of the calorific value. This can be done by subtraction of the partial pressure p' of the water vapor from the total gas pressure p_t. The remainder is the corrected pressure value p_c to be considered in the above calculations of the calorific value.

$p_c = p_t - p'$ (Equ. 4.9)

The partial pressure of water vapour itself is a function of the gas temperature and the relative humidity as given in Fig. 4.3.

Example:
given:

- gas temperature: t_g = 40ºC
- relative humidity: 100%
- total gas pressure: p_t = 970 mbar

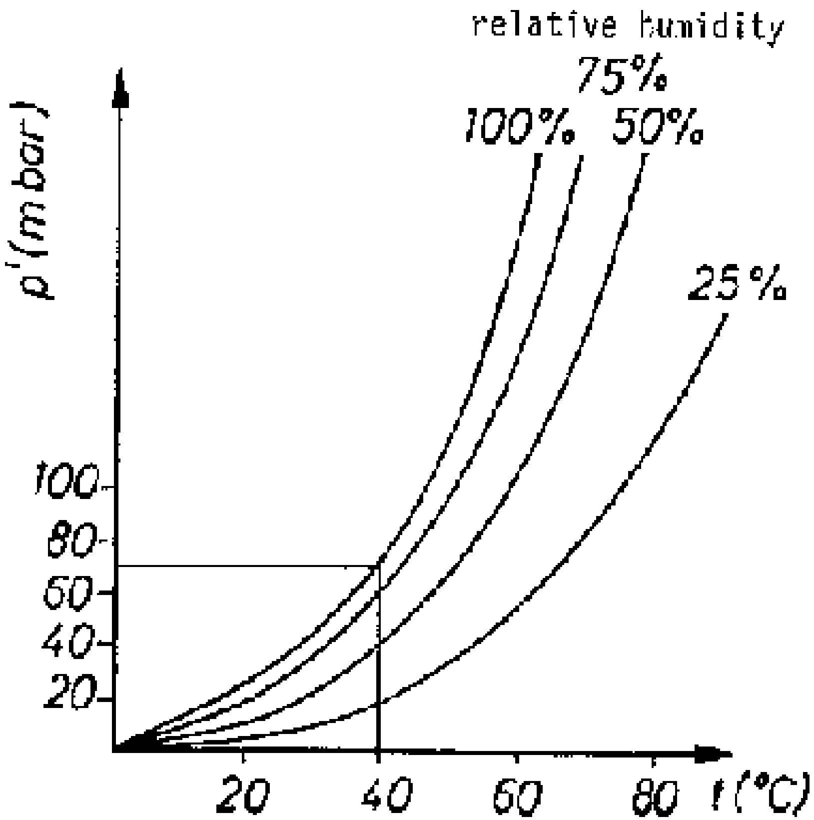


Fig. 4.3: Partial pressure of water vapor in a mixture with biogas as a function of a biogas temperature and relative humidity

Solution:

Step 1: partial pressure from diagram: p' = 70 mbar

Step 2: corrected gas pressure for calculation of calorific value from Step 1 in previous example onwards:

$p_c = p_t - p' = 970 - 70 = 900$ **mbar**

4.3 Biogas Consumed as a Fuel

The fuel consumption of equipment using biogas is often specified in m³n/h or m³n/kWh, i.e. standard cubic meters per hour or per kilowatt hour (sic) respectively. The standard cubic meter (m³ n) means a volume of 1 cubic meter of gas under standard conditions, i.e. at a temperature of 0 °C (273 K) and a pressure of 1013 mbar. The consumption of biogas in actual volume will differ from these data according to the actual conditions of the biogas as fed to the equipment (motor, burner, etc.) in terms of

-temperature,
-pressure,
-composition, i.e. CH_4 content.

The determination of the actual volumetric consumption of an engine operating on biogas fuel is of utmost importance for the dimensioning of biogas plant, engine,

mixing device and other equipment. A difference of 50% between actual volumetric consumption and specified consumption of a biogas engine can easily occur and could result in poor performance of the engine if not considered.

Using the diagrams Figs 4.1 and 4.2 the consumption of the specific biogas can easily be found:

Step 1:

Check how the fuel consumption fc is specified.

-If in m^3n/h, continue with step 2.
-If in m^3/h without biogas specification assume a calorific value of H_u = 20 000 kJ/ m^3 .
-If as specific fuel consumption at rated conditions use fc = sfc· P (in m^3/h). (Equ. 4.10)
-If only the efficiency η is specified use fc = $1/\eta \cdot p \cdot 1/H_u \cdot$ 3600 (in m^3/h). (Equ. 4.11)
-If no information is given use Equ. 4.11 with η= 0.3 for dual fuel and larger Otto gas engines and η = 0.25 for smaller Otto gas engines as well as H_u = 20 000 kJ/m^3.

Step 2:

Determine the calorific value of the biogas used for specification of the equipment by the manufacturer.

- If consumption is specified by engine supplier in kJ/h, use this value and continue further below in step 4.
- If calorific value of biogas is specified in kWh/m³ n transform this figure by multiplying by 3600 to obtain it in kJ/m³ n.
- If biogas is specified by its CH4 content in Vol % use diagrams in Figs. 4.1 and 4.2 to obtain the calorific value in kJ/m³ n.

Step 3:

Determine the required energy flow (calorific consumption) of the engine at rated performance in kJ/h by multiplying the specified consumption rate at standard conditions in m³ n/h with the calorific value of the biogas in kJ/m³ n, as-specified by the engine supplier (energy consumption = specified volumetric consumption x calorific value of biogas).

Step 4:

Determine the actual calorific value of your specific biogas in kJ/m³ using the procedure explained in Chapter 4.2.

Step 5:

Determine how much of your specific biogas will be consumed by the engine in m³/h by dividing the energy consumption (Step 3) by the calorific value of your specific biogas (Step 4):

$$\text{volumetric consumption} = \frac{\text{energy consumption (in kJ/h)}}{\text{specific calorific value (in kJ/m}^3\text{)}} \text{(in m}^3\text{/h)}$$ **(Equ 4.12)**

Example:
Manufacturer's engine specification:

- **power rating P = 20 kW**
- **fuel consumption at rated power fc = 10 m³ n/h**
- **biogas used 70% CH_4, 30% CO_2**

Specification of biogas from your plant (see Chapter 4.2)

H_u = 18 900 kJ/m³

Step 1:

No calculation needed as the fuel consumption is specified.

Step 2:

From diagram Fig. 4.2 calorific value of biogas used in specification of manufacturer:

$H_{u,n}$ = 25200 kJ/m³ n (at standard conditions).

Step 3:

Energy consumption (flow) of the engine at rated power

$$\dot{E} = fc \cdot H_{u,n} = 10 m^3 n/h \cdot 25000 kJ/m^3 n = 250000 kJ/h$$ **(Equ. 4.13)**

Step 4:

Calorific value of your specific biogas from plant (see Chapter 4.2)

H_u = 18 900 kJ/m³.

Step 5:

Actual biogas consumption fc of engine at rated power

$$fc = \frac{\dot{E}}{H_u} = \frac{250000 kJ/h}{18900 kJ/m^3} = 13.23 m^3/h$$ **(Equ. 4.14)**

The volumetric fuel consumption in this case would be 32% higher than specified by the manufacturer at standard ("n") conditions, which demonstrates that the above calculation should not be dispensed with.

4.4 The Technical Parameters of Biogas/Methane

Methane and gases having a considerable methane content have long been researched on to establish their physical properties and technical behavior.

Some of the properties, which have an effect on the combustion process in an engine, shall be explained hereunder:

- Ignitability of CH_4 in a mixture with air

CH_4: 5 . . . 15 Vol %

air: 95 . . .85 Vol %

Mixtures which are leaner, i.e. CH_4 content less than 5 Vol % or richer, i.e. CH_4 content more than 15 Vol %, will not properly ignite with spark ignition.

-Combustion velocity c_c in a mixture with air at a pressure of p = 1 bar

c_c = 0.20 m/s at 7% CH_4
c_c = 0.38 m/s at 10% CH_4
c_c = 0.20 m/s at 13% CH_4

The combustion velocity is a function of the volume percentage of the burnable component, here CH_4. The highest value is near the stoichiometric air/fuel ratio, mostly at an excess air ratio of 0.8 . . . 0.9. It increases drastically at higher temperatures and pressures.

-Temperature at which CH_4 ignites in a mixture with air

T_1 = 918 K. . .1023K(=645°C...750°C)

- Compression ratio of an engine, e, at which temperatures reach values high enough for self-ignition in a mixture with air (CO_2 content decreases ignitability, i.e. increases possible compression ratio)

e = 15...20

- Methane number, which is a standard value to specify a fuel's tendency to "knocking", i.e. uneven combustion and pressure development between TDC and

BDC

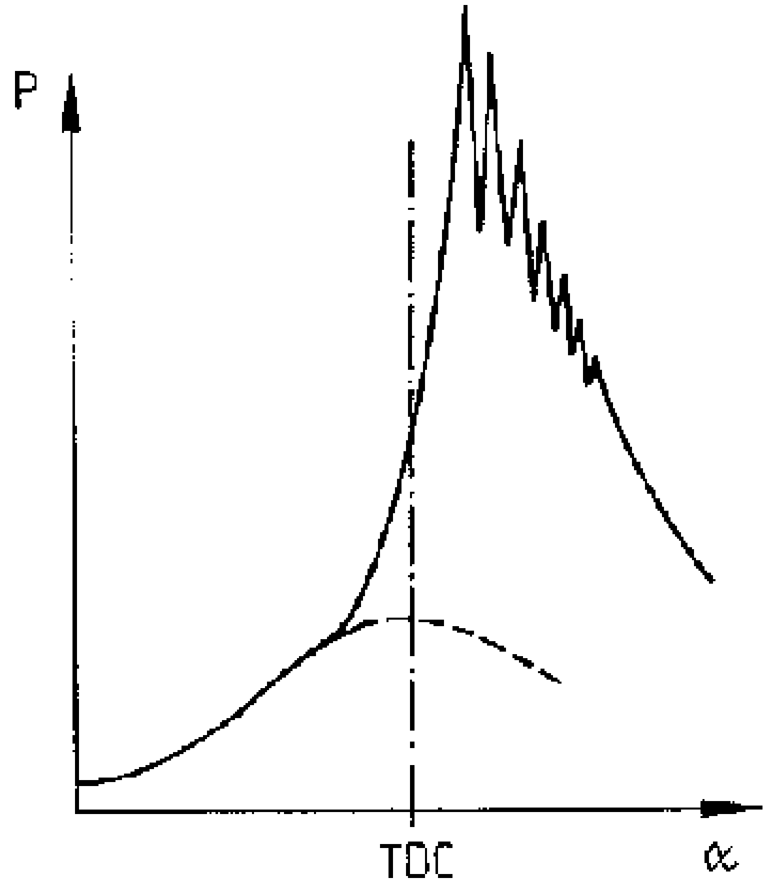


Fig.4.4: "Knocking" in a p, alfa-diagram of an engine

CH_4, 100%:	100
biogas (CH_4 70%):	130
for comparison:	
butane:	10
propane:	33.5

Methane and biogas are very stable against "knocking" and can therefore be used in engines of higher compression ratios than petrol engines. Fig. 4.4 illustrates the cause of the pressure and hence the force on the piston when the engine "knocks". Operation under such conditions will gradually destroy the engine.

- Stoichiometric air/fuel ratio on a mass basis at which the combustion of CH_4 with air is complete but without unutilized excess air

$$\frac{m_{CH4}}{m_{air}} = \frac{1 kg CH_4}{14.5 kg\ air}$$

4.5 Desulphurization and Filtering of Biogas

Biogases from different materials contain different percentages of hydrogen sulphide H_2S, i.e. 0.10 . . . 0.50% Vol (1000 . . . 5000 ppm). As H_2S is corrosive to metals especially in connection with water or humidity, its content should be as low as possible when used as a fuel in engines. Some engine manufacturers specify a maximum allowable value of 0.15% Vol; others allow more or give no data.

H_2S can be removed by filtering with earth or with iron oxide (e.g. filings) whereby the filters need to be regenerated or the material exchanged periodically [24]. Recent experiments in a large biogas plant in Ferkessedougou, Ivory Coast [25], have revealed that by purging a small amount of air into the gas holder or store and allowing a reaction time of about 25 . . . 30 hours, a substantial percentage, i.e. about 80%, of the H_2S is reduced to elementary sulphur which is

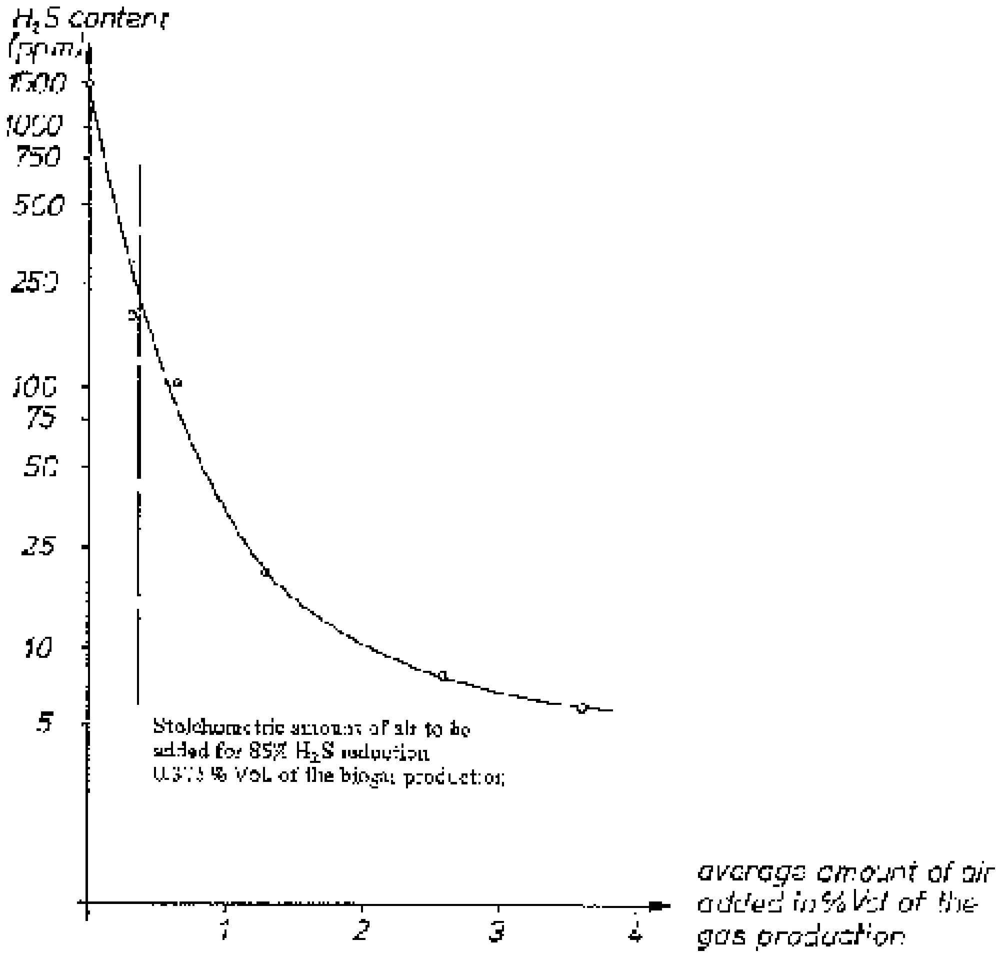
deposited on surfaces within the plant or on the floating scum. The amount of air allowed into the gas holder/store needs however to be well dosed, preferably with a small dosage pump. A mean value for the constant air supply is approx. 0.4 % Vol of the constant gas production for a reduction of approx. 80% of the H_2S, e.g. from 0.5% Vol H_2S to 0.1% Vol, which is adequate for engine operation.

Depending on the type of biogas plant and piping, some indispensable solids can be drawn with the gas to the mixer. A simple filter in the form of a larger container filled with washed rubble or a tissue filter with no measurable pressure loss is recommendable in any system.

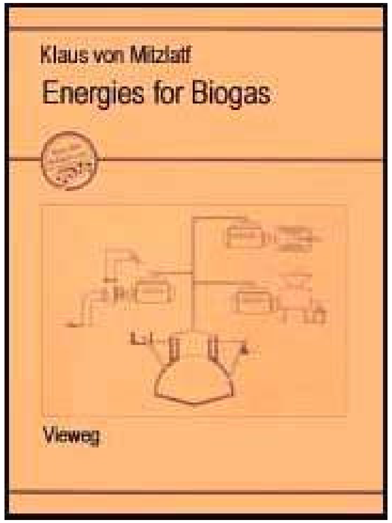
initial H_2S content		stoichiometric amount of oxygen as vol. % of biogas production	stoichiometric amount of air as vol % of biogas production
in ppm	in vol. %		
500	0.05	0.025	0.125
1000	0.10	0.050	0.250
1500	0.15	0.075	0.375
2000	0.20	0.100	0.500
2500	0.25	0.125	0.625
3000	0.30	0.150	0.750

a) Stoichiometric amounts of oxygen or air to be added for an 85% reduction of

the H_2S content for different initial H_2S content values.



b) H_2S reduction from initially 1500 ppm as a function of added air.



Engines for Biogas (GTZ, 1988, 133 p.)

5. The Gas Diesel Engine

Diesel engines can be modified to operate on gaseous fuels in two different ways:

-dual fuel operation with ignition by pilot fuel Injection,
-operation on gas alone with spark ignition.

5.1 The Dual Fuel Engine

5.1.1 What is "Dual Fuel Operation"?

As described in Chapter 3.3.1 on diesel engines, the fuel is mixed with air towards the end of the compression stroke of the engine by being sprayed into the combustion chamber with high pressure (about 200 bar). The fuel is immediately

ignited when it comes into contact with the hot compressed air.

In dual fuel operation the normal diesel fuel injection system still supplies a certain amount of diesel fuel. The engine however sucks and compresses a mixture of air and fuel gas which has been-prepared in an external mixing device. The mixture is then ignited by and together with the diesel fuel sprayed in.

The amount of diesel fuel needed for sufficient ignition is between 10% and 20% of the amount needed for operation on diesel fuel alone. It differs with the point of operation and engine design parameters.

Operation of the engine at partial load requires a reduction of the fuel gas supply by means of a gas control valve. The valve can be manually operated or automatically, using mechanical or electronic system. A simultaneous reduction of the air supply would however decrease the suction, hence the compression pressure and the mean effective pressure, and would lead to a drop in power and efficiency. With drastic reduction the compression conditions might even become too weak to effect self-ignition. Dual fuel engines should therefore not be throttled/controlled on the air side.

The air/fuel ratio of the sucked mixture varies by control of the fuel gas but even a very lean mixture (λ= 4.0) still ignites with the many well distributed spray droplets of diesel fuel.

All other parameters and elements of the diesel engine remain unchanged such as the compression ratio, the point or crank angle of injection, etc.

Modification of a diesel engine for a dual fuel process has the following

advantages:

- **Operation on diesel fuel alone is possible in cases where fuel gas is in short supply.**

- **Any contribution of fuel gas from 0 . . . 85 % can substitute a corresponding part of the diesel fuel while the performance remains as in 100% diesel fuel operation.**

- **Because of the existence of a governor at most of the diesel engines automatic control of speed/power can be done by changing the amount of diesel fuel injection while the gas fuel flow remains uncontrolled, i.e. constant; diesel fuel substitutions by biogas are however less substantial in this case.**

The limitations need to be mentioned also.

- The dual fuel engine cannot operate without the supply of diesel fuel for ignition.

- The fuel injection jets may overheat when the diesel fuel flow is reduced to 10 or 15% of its normal flow. Larger dual fuel engines circulate extra diesel fuel through the injector for cooling.

Self-modified diesel engines are often operated at higher diesel fuel rates than necessary for ignition purposes in order to facilitate sufficient cooling of the jet. In operation with scrubbed biogas, i.e. 95...98% CH_4, combustion temperatures are higher than for untreated biogas so that diesel fuel substitution is limited at about 60% maximum, i.e. an amount of 40% diesel fuel is necessary for ignition and for

cooling of the injector nozzle.

To what extent the fuel injection nozzle can be affected is however a question of its specific design, material and the thermal load of the engine, and hence differs from case to case. A check of the injector nozzle after 500 hours of operation in dual fuel is recommended.

5.1.2 Different Types of Dual Fuel Modification

The type of modification chosen is largely dependent on:

- **anticipated type of operation,**
- **available funds,**
- **available expertise/manpower,**
- **type of driven machine,**
- **biogas supply,**
- **availability/cost of engine,**
- **economic conditions.**

All parameters need to be well considered before a choice of engine is made. The alternative of an Otto gas engine or even no engine but an alternative solution is also worth being discussed (see Chapter 7).

5.1.3 Mixing Devices for Dual Fuel

For dual fuel operation a mixing device has to meet the following requirements:

- provide a homogeneous mixture of both air and fuel gas,

- vary the fuel gas flow according to performance required,
- be able to supply sufficient air and fuel for operation at maximum load and speed under consideration of the actual pressures of gas and air and the fact that the excess air ratio shall not be less than about λ = 1.5 because sufficient excess air is needed for combustion of the pilot fuel also,
- enable automatic control of operation in partial load by means of a governor or electronically controlled mechanisms if required.
There are several alternatives in meeting these requirements.

5.1.3.1 Simple Mixing Chambers

A simple mixing chamber consists of a container or even a T-junction of a tube or flow channel with an inlet for air and for gas each and an outlet for the mixture of both. The outlet is connected to the intake channel or manifold of the engine. For control of the engine power (partial load) the fuel gas supply is controlled by a valve. The valve may be hand-operated or can be connected to an automatic control, either mechanically by a governor or electronically.

The airflow into the mixing chamber is not controlled for reasons explained earlier. It may however be necessary to slightly throttle the airflow before it enters the mixing chamber or mixing zone in a channel in order to provide a slight depression. The depression may only be necessary in cases where the fuel gas is supplied at a low pressure (underdimensioned supply piping!) to create the necessary pressure drop for sufficient suction of the fuel gas. The position of the depression throttle will remain unchanged during operation. In most cases the depression created by the air filter provides sufficient suction for the gas. Any depression, however, lowers the performance. A marginal loss may be seen as

acceptable if control is eased on the other hand.

The gas flow however is also dependent on the dimension of the gas pipe. Pipes with small diameters create more resistance, hence more pressure drop than in pipes with larger diameters. The gas supply pipe from the plant shall therefore have a diameter which is not smaller than about 0.5 times the diameter of the air inlet to the engine manifold. An oversupply of fuel gas cannot occur as the gas flow will be controlled by the gas valve at inlet to the mixing chamber.

Mixing chambers with a larger volume than just a T-joint pipe provide a longer retention time of air and fuel inside the chamber and a more homogeneous mixture which becomes essential when the distance between mixing device and inlet manifold is short, hence the mixing time. The connection of the gas supply pipe into the suction chamber of a larger oilbath air filter may meet the requirements for a simple mixing chamber also.

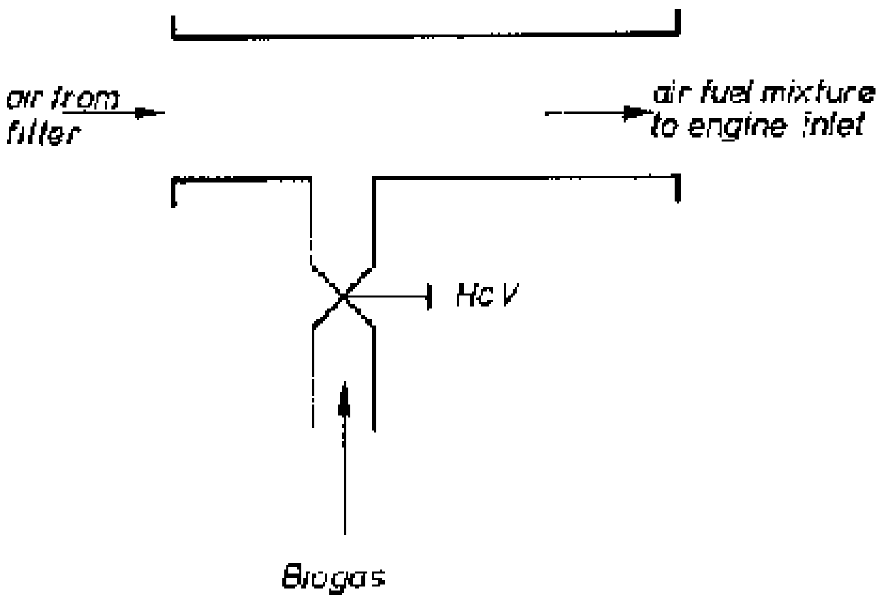


Fig. 5.1: T-joint mixer

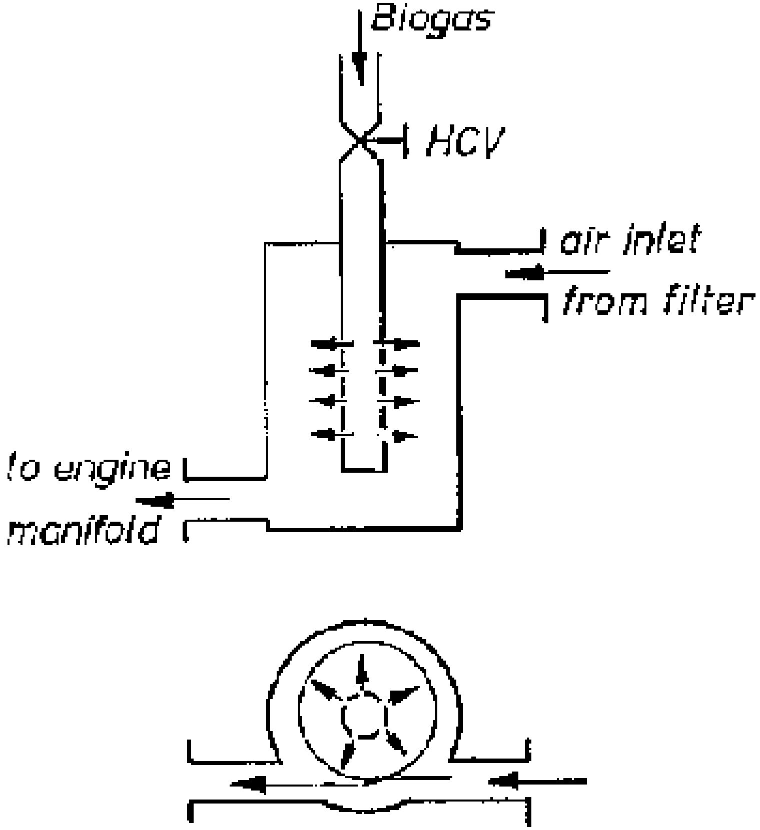


Fig.5.2: Simple mixing chamber with hand - controlled valve (HCV)

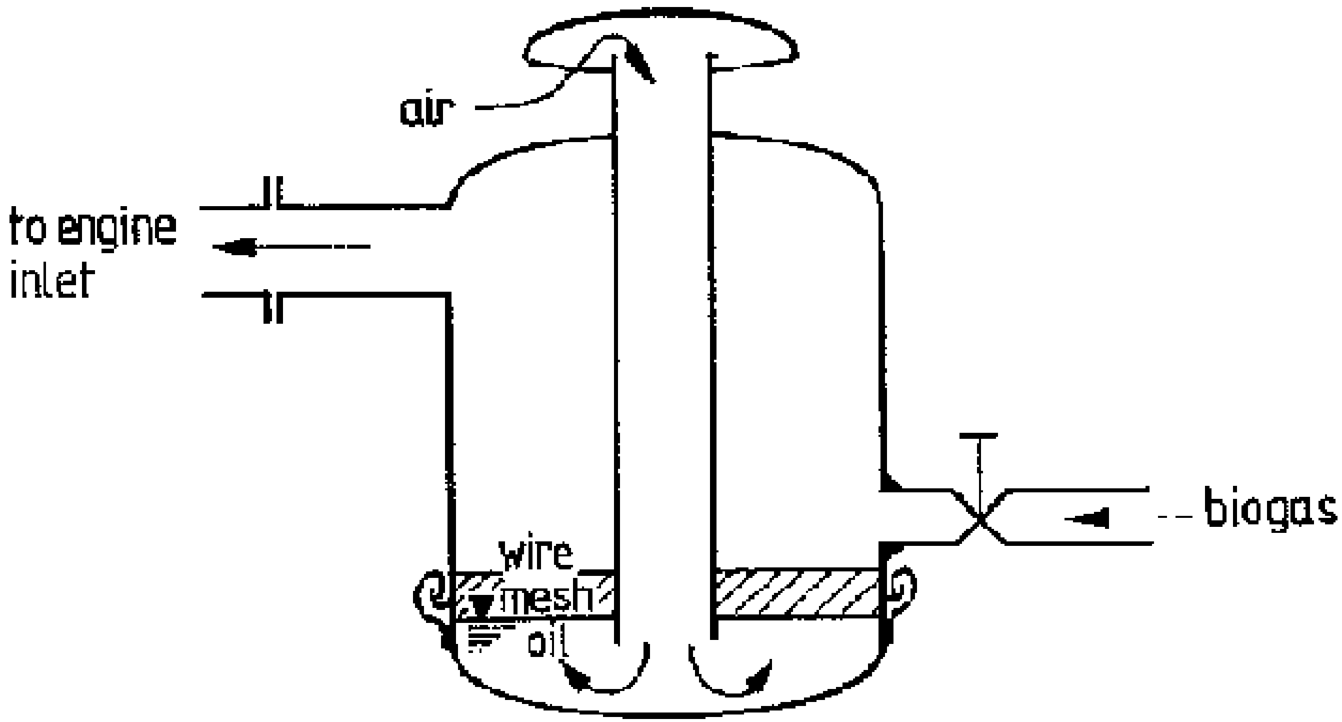


Fig. 5.3: Air filter modified into mixing chamber

A mixing chamber as described here will provide an individual air/fuel mixture according to its design and/or setting of its fuel gas valve. Once it is properly tuned, the engine operates well at constant speed and power output as long as the power demand from the driven machine is not varied. At higher load and hence lower speed, however, the intake of the engine will suck less air while the fuel gas flow remains almost constant. As a result the air/fuel ratio will change and the mixture becomes richer. If the engine finds a new balance at a lower speed which the driven machine can tolerate, operation may continue without adjustments as the usually high excess air ratio allows for more fuel. The governor - unless blocked - will also increase the amount of diesel fuel injected to maintain the

former speed. To save this additional diesel fuel consumption, the control of the gas flow should therefore always be adjusted when the engine is operated at considerably different conditions.

5.1.3.2 Venturi Mixer

A venturi mixer is shown further below in Fig. 6.2. The supply of biogas through several bores around the circumference of the "bottleneck" facilitates the homogeneous mixture of gas and air. The specific advantage of a venturi mixer, i.e. the constant air/ fuel ratio of the mixture, can hardly be utilized by a dual fuel diesel engine as a variation in power output is usually effected by a variation of fuel alone, hence by excess air ratio, not by a variation of the cylinder filling rate as is the case in Otto engines.

The design of a venturi mixer for diesel gas engines will have to consider a larger excess air ratio of about λ = 1.5 to ensure complete combustion of fuel gas and pilot fuel. They do not need a throttle valve for the control of the intake to the engine as this would lower the mean effective pressure, hence the efficiency of the engine. Should the venturi mixer used have a throttle, it should be kept fully open at any condition. Power and speed are to be controlled by variation of the fuel input (fuel gas and/or diesel fuel) only. For the design parameters of a venturi mixer refer to Chapter 6.

5.1.3.3 Mixing Valves

Mixing valves are designed to supply an engine with an air/fuel mixture at a constant excess air ratio while the flow rate of the mixture can be controlled by an

integrated throttle valve. For similar reasons as explained in the previous chapter on venturi mixers, the mixing valves have no special advantage compared to a mixing chamber in dual fuel operation.

5.1.3.4 Other Mixing Devices

In some larger specially designed diesel gas engines fuel gas is supplied through an extra gas inlet valve in the engine's intake which is opened and closed by the engine's camshaft in relation to the crank angle. A gas control valve in the gas inlet pipe/channel is connected to the engine's speed and power control. This control system provides better fuel economy as fresh gas is only sucked in when the outlet valve is already closed so that absolutely no fuel is wasted, i.e. uncombusted.

This system is usually not provided for engines within the scope of this publication as it involves more sophisticated mechanics and control and makes the engine more expensive. The system cannot be integrated into a normal diesel engine with reasonable efforts and is therefore not considered here.

5.2 Modification into a Dual Fuel Engine

5.2.1 Design and Dimensioning of Mixing Chamber

5.2.1.1 Volume of the Mixing Chamber

The mixing chamber types mentioned above basically provide good mixing of air and biogas. In the tube-type mixer the distance between gas inlet and the engine manifold should not be too short to allow sufficient time for the mixture to

become homogeneous.
This is essential for a multicylinder engine as the flow conditions in the manifold may cause an uneven distribution of fuel gas to the cylinders if air and fuel are not fully mixed before thy enter the manifold. As a minimum distance between the gas inlet and the inlet to the engine manifold one should consider twice the tube (inlet) diameter.

As an orientation value for the volume of a mixing chamber choose the cubic capacity of the engine, i.e. about 2 lifers for an engine with 2-1 capacity. The actual shape of the mixing chamber whether cubic or cylindrical may be chosen in accordance with the availability of space, material and the best mode of connection to the manifold.

5.2.1.2 Connection to Engine and Air Filter

Air filters are in most cases directly connected to the engine inlet manifold; in a few cases they are detached and connected with a flexible hose pipe. Usual ways of connection are

- clamps,
- flanges,
- threads.

The design and dimensions of the mixing chamber inlet and outlet need to match with the air filter and inlet manifold respectively. Tube-type mixers should have the same or larger diameters than the inlet manifold. In case of a larger diameter a reducer adapter is necessary with a maximum reduction angle of 10° to ensure

smooth flow without detachment. Mixing tubes with a diameter smaller than the manifold should not be used as they cause unnecessary flow restrictions and power reduction at higher speeds.

Adapters will also be necessary to connect square-shaped channels with circular channels. The cross-sectional area of the mixing device should in no case be smaller than the respective area of the engine inlet manifold.

5.2.1.3 Gas Inlet Pipe/Nozzle

The fuel gas inlet nozzle dimension is mainly dependent on:

- fuel energy required by the engine at maximum rated power and speed,
- calorific value of the biogas (per volume) under the actual conditions of temperature, pressure and its composition (CH_4 content), see Chapter 4.

The fuel energy required by an engine can be determined using its specifications, either the total efficiency or the specific fuel consumption at rated conditions. In cases where no information is available the following mean values can be assumed:

- total efficiency η_{tot}

= 0.25 for engines up to 1000 cm³ capacity

= 0.3 for engines from 1000 cm³ upwards

- specific calorific fuel consumption sfc_{cal}

$$= 3.3 \frac{\text{kWh (fuel energy input)}}{\text{kWh (mech. energy output)}}$$

The following diagrammatic example shall demonstrate the determination of the actual volumetric demand for biogas of an engine with the following data (see procedure in Chapter 4):

- rated power (mech.): P = 10 kW
- biogas volumetric calorific value:

$H_{u,vol}$ = 20 000 kj/m³

- specific calorific fuel consumption:

$$sfc_{cal} = 3.3 \frac{kWh}{kWh}$$

- proportion of biogas in total fuel: 80%

Step 1:

Find the total volumetric fuel demand (consumption)

$$fc_{vol} = \frac{sfc_{cal} \cdot (3600 s/h) \cdot P}{H_{u,vol}} = \frac{3.0 \cdot 10 kj/s \cdot 3600 s/h}{20000 kj/m^3}$$

$$fc_{vol} = 5.4 \frac{m^3}{h} = 0.0015 \frac{m^3}{s}$$

Step 2:

Consider proportion of biogas, i.e. 80%

$$fc_{volbg} = 0.8 fc_{vol} = 0.0012 \frac{m^3}{s} = 4.32 \frac{m^3}{h}$$

The volumetric fuel demand in this case is 4.32 m^3/h.

The diameter of the fuel gas inlet nozzle which is large enough to allow the calculated volume to pass into the mixing chamber depends on the following parameters:

- vacuum (or depression) in mixing chamber or manifold,
- pressure in biogas plant or piping respectively.

A volume flow through a pipe, orifice, nozzle or similar is described by

$V = c \cdot A$ **(Equ. 5.1)**

with V = volume flow in m^3/s, c = flow velocity in m/s, Across-sectional area in m^2.

From an energy balance for a tube flow[2] at two different cross-sectional areas (1 and 2) the velocity can be calculated:

$\frac{p_1}{\rho} + \frac{c_1^2}{2} = \frac{p_2}{\rho} + \frac{c_2^2}{2}$ **(Equ. 5.2)**

$c_2 = \sqrt{2 \frac{p_1 - p_2}{\rho} + c_1^2}$ **(Equ. 5.3)**

The density of the biogas, like the calorific value, varies with the pressure, temperature and composition. The velocity in the piping between the plant and the engine depends on the volume flow (as calculated), the crosssectional area of the pipes (see Equ. 5.1) and the flow resistance of pipe bends, valves, etc. The piping size shall always be large enough so that the flow velocity does not exceed c_1 = 2 m/s to reduce Cow friction and prevent a substantial pressure loss between plant and engine. Too narrow piping or restrictions can cause a throttle effect and insufficient biogas supply to the mixing chamber.

The active difference of pressure between the gas in the supply pipe before the mixer and the pressure of the airflow in the mixer is a sum of the

- biogas plant pressure, i.e. Δp = 0.005 ... 0.02 bar,
- depression in manifold/mixing chamber, i.e. dp = -0.01 ... 0.02 bar,
- losses in piping, filters, control valve and the nozzle or jet itself, i.e. dp = 0.01 . . . 0.05 (estimated).

It can therefore assume values between 0 and 60 mbar (0 ... 60 cm W.H.) depending on the actual conditions of plant, piping, engine suction, etc.

A simple and effective way to establish the actual pressure difference at maximum conditions is a connection of a water-filled U-tube, even from bent transparent plastic pipes. It should be connected to the manifold or mixing chamber on one side and the gas pipe before inlet to mixing chamber on the other.

A pressure difference of Δp = 50 mbar, an average biogas density of 1 kg/m³ and a flow velocity in the gas pipe of about 2 m/s result in a theoretical gas flow

velocity at the jet (or point of smallest diameter, i.e. orifice, control valve) of cg = 100 m/s (see Equ. 5.3). However, at high velocities as in this case the flow friction considerably reduces the velocity, especially when the gas is introduced through several small holes instead of one larger inlet.

The exact calculation of all parameters influencing the cross-sectional area of the nozzle would involve extremely precise and scientific measurements and manufacture of the mixing device as weld as constant gas conditions. It shall therefore, in line with the framework of this publication, be allowed to use a more practicable approach to establish the dimension of the jet. The fact that furthermore the gas conditions are subject to changes due to weather and biogas plant performance justifies the use of assumption which consider a variety of operational parameters and will allow the engine to be operated under more than only one specified condition.

It is therefore recommended to dimension the nozzle's cross-sectional area in such a way that sufficient biogas can be supplied to the engine even at "unfavorable" conditions, i.e. low volumetric calorific value of the biogas, low gas pressure, considerable flow resistance, etc. The gas inlet will thus be slightly oversized in some cases. However, an oversupply of biogas can easily be prevented by the control or calibration valve which after all acts as an additional resistance in the piping system and reduces the active pressure difference at the nozzle, i.e. the flow velocity and gas supply. Should the biogas supply at a later stage still be found too high at fully opened control valve, an additional fixed orifice or adjustable throttle can be installed in the gas pipe to limit the maximum gas flow and prevent operation with an oversupply of gas at the control valve in fully open position. A well adjusted pneumatic pressure regulation valve can serve

the same purpose.

The following parameters shall therefore serve for the dimensioning of the gas pipe:

- active pressure difference Δp = 0.02 bar (20 cm W.H.),
- velocity at gas nozzle c_g = 20 m/s,
- volumetric calorific value of biogas $H_{u,vol}$ = 17000 kJ/m³,
- specific fuel consumption of the engine sfc = 0.8 m³/kWh.

The example below shall illustrate the procedure:

Engine parameters:

- rated power: 25 kW
- cubic capacity: 3.5 liters
- engine speed: n = 1800 1/min
- volume efficiency: η_{vol} = 0.85
- manifold connection diameter: 60 mm
- substitution of diesel by biogas: 80%
- mixer type chosen: tube type

Step 1:
Volumetric air intake, V_{air} (4-stroke engine):

$$V_{air} = \eta_{vol} \cdot \frac{V_h \cdot n}{2000 \cdot 60} \text{ in m}^3/\text{s}$$ **(Equ. 5. 4)**

$$= 0.85 \cdot \frac{3.5 \cdot 1800}{2000 \cdot 60} = 0.0446\ m^3/s$$

Step 2:

Cross-sectional area of intake (and tube mixer), Ai:

$A_i = \frac{d^2 \cdot \pi}{4}$ **(Equ. 5.5)**

$$= \frac{0.06\,m^2 \cdot \pi}{4} = 0.0028\,m^2$$

Step 3:

Intake velocity, c_i:

$$c_i = \frac{V}{A} = \frac{0.0446}{0.0028} = 16\ m/s$$

Step 4:

Volume flow of biogas (fuel consumption, f_c) at rated power:

$$f_c = sfc \cdot P = 0.8 \frac{m^3}{kW \cdot h} 25\ kW = 20\ m^3/h = 0.0056\,m^3/s$$

Step 5:

Consideration of percentage of biogas in total fuel (for dual fuel only):

$$fc_1 = 0.8 \cdot fc = 0.8 \cdot 0.0056 = 0.0045 m^3/s$$

Step 6:

Cross-sectional area Ag and diameter dg of nozzle:

$$A_g = \frac{fc}{c_g} = \frac{0.0045 m^3/s}{20 m/s} = 0.000225 m^2$$

$$d_g = \sqrt{\frac{4 \cdot A_g}{\pi}} = \sqrt{\frac{4 \cdot 0.000225}{\pi}} = 0.017 m = 17 mm$$

The gas nozzle diameter of $d_g = 17 mm$ can thus be assumed to be sufficient for operation under the conditions specified above. However, should the engine be operated at a higher rate of power and speed the nozzle may be found to be too small. It is therefore essential to carefully anticipate all possible ranges of operation. In cases of doubt a 10% oversizing of the gas nozzle diameter is allowable. The total area of a multiple hole gas inlet shall also be about 10% bigger than the area calculated for a one-hole inlet to compensate for the increase in flow friction. The maximum (total) area of the nozzle(s) shall however not exceed one tenth of the intake manifold cross-sectional area.

The shape of the nozzle and the way it is connected or introduced into the mixing device is important for a good mixture of air and gas. The following methods are possible:

- Simple T-joint:
The gas pipe is butt-joined without protruding into the mixing device, effecting

only a little change of active pressure drop at higher engine suction (speed). The minimum distance of the gas inlet, i.e. two times the tube diameter from the engine manifold has to be observed for all T-joint mixer types (see Fig. 5.1).

-T-joint with the gas pipe protruding into the mixing device:
The gas pipe (nozzle) is cut oblique (30 ... 45°) with the opening facing the engine inlet. The protruding gas pipe slightly decreases the cross-sectional area for the airflow and causes a slight depression, thus increasing the active pressure drop for the gas to flow into the mixing device. The pressure drop rises with engine suction (engine speed), and hence sucks more gas also. The function is somewhat similar to the function of a venturi jet. The mixing performance is superior to that of a blunt T-joint (see Fig. 5.4).

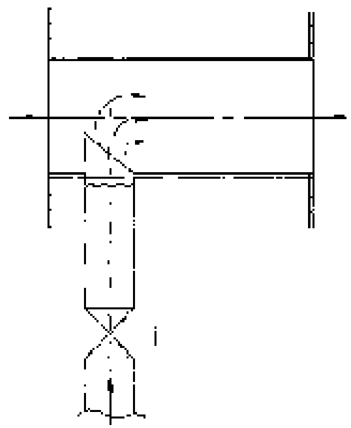


Fig. 5.4: T-joint mixer with oblique, protrunding gas inlet

- Venturi mixer:
This type is equipped with a ring channel and several small gas inlets around the circumference (see Fig. 6.2). With a ratio between the manifold inlet diameter and the venturi jet diameter of d_i/d_v = 1.5 . . . 1.7 the venturi provides an almost

constant ratio of air and fuel at any flow rate into the engine without adjusting the gas valve. However, when used for a dual fuel engine, at partial load operation the gas control valve needs to be operated (partly closed) for fuel reduction.

- Mixing chambers with larger volumes: Due to the relatively low flow velocities more time for mixing is available. It is, however, advantageous for the mixing if the gas pipe protrudes into the chamber and distributes the gas through several holes. The flows can also be further mixed with two or three layers of wire mesh (about 1 cm^3 mesh aperture) at a short distance (about 5 mm) between each other (see Fig. 5.5).

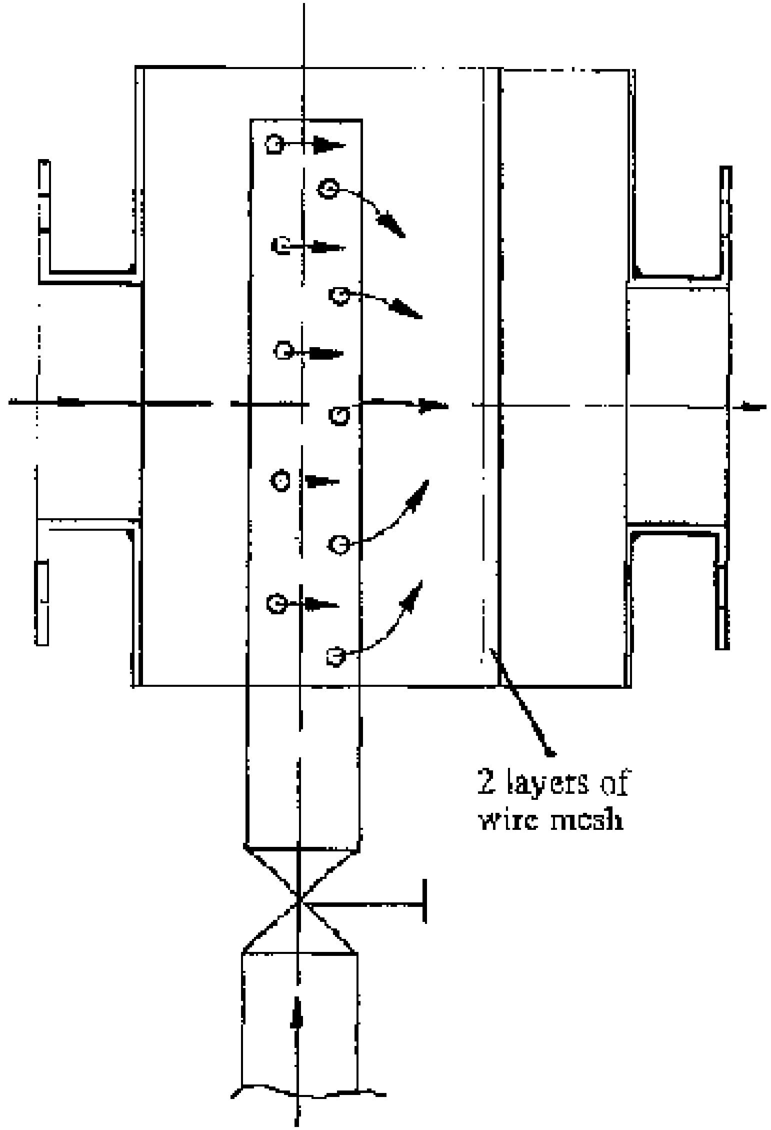


Fig. 5.5: Mixing chamber with gas distribution pipe and wire mesh for intensive mixing

5.2.2 Manufacture and Installation

5 2.2.1 Manufacture

Tube Type

A tube-type mixer can be manufactured from standard tube material, e.g. water pipes and other steel tubes. Plastic material may be suitable in cases where the tube is not directly mounted to a hot engine manifold or when heat-resistant material is available. The gas pipe/nozzle can be brazed, welded or glued with a two-component synthetic resin cement into a hole with a matching diameter. When the connecting flanges are being welded to the tube the final position of the gas inlet has to be observed in relation to the manifold to obtain good access to the control valve when mounted directly to the mixer. The mixer is to be installed between the air filter and the engine inlet. In cases where space does not allow direct mounting to the manifold the mixing tube/chamber can be installed nearby using a flexible hose pipe for connection to the manifold. In case of a connection flange at the manifold a short tube socket will have to be manufactured to connect hose pipe and manifold.

V-type engines or other engines with two air inlets require one common mixing device to secure the supply of all cylinders with a uniform air/fuel ratio. The mixing tube/chamber will have to be connected to the two inlets with a Y-pipe, two flexible hose pipes and two pipe sockets mounted to the engine inlets. The use of two individual mixers should be discouraged unless they are identical in all parameters including the setting of the gas control valve.

Their design parameters would then need to consider that they feed only one half of the engine, i.e. airflow and gas supply are one half of what the engine requires in total (see Fig. 5.7).

Mixing Chamber

Mixing chambers can be made of sheet material, larger tubes, hollow profiles, etc. The connectors or flanges and the gas inlet are brazed or welded, likewise the body itself. Should an oilbath air filter be used as a mixing chamber the gas inlet needs to be connected to the clean air chamber.

If it is necessary to install the mixing chamber separately from the engine due to scarcity of space or the existence of more than one engine inlet refer to explanations given above for tube-type mixers.

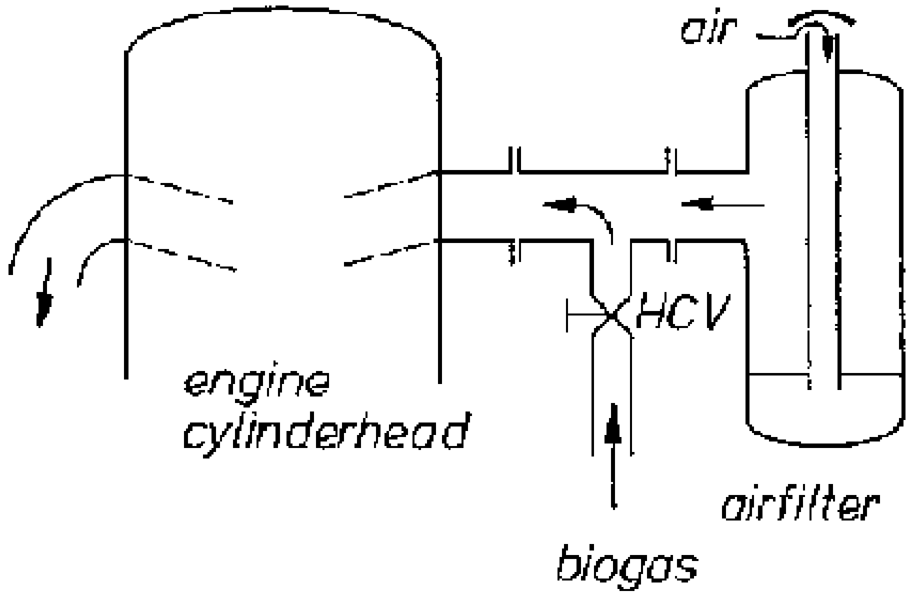


Fig. 5.6: T-joint mixer installed between air filter and engine inlet

5.2.2.2 Installing the Mixing Chamber

Diesel engines, whether of a stationary or vehicle type, are usually equipped with

an air filter/air cleaner connected to the inlet manifold or suction channel of the engine. The air filter can be fixed using

- a flange,
- threads,
- a clamp.

The installation of the mixing chamber is carried out as follows:

- Disconnect the air filter.
- Take measurements of the connecting flange, threads, clamp and manufacture flanges, threads, clamps in such a way that the mixing chamber can be connected to the manifold and the air filter can be connected to the mixing chamber with matching dimensions.
- Observe the flow direction in the mixing chamber.
- Observe easy accessibility to the biogas control valve cum piping.
- Observe the final position of the air filter (space!).
- Manufacture or buy additional gaskets or seals and bolts/nuts or clamps.
- Mount the mixing chamber to the manifold with gasket/seal.
- Mount the gas control valve cum seal and connect it to the biogas piping with flexible hose pipe and hose clip (engine vibrates!).
- Mount the air filter to the mixing chamber with gasket/seal.

With the mixing chamber properly inserted between air filter and engine manifold and the connection of a manual control valve the essential steps for a simple but practicable modification of a diesel engine have been taken.

For V-type engines distribute the air/fuel mixture with the help of a Y-pipe to the engine inlets. If two separate air filters were previously used, they must both be retained and connected to the mixing device possibly using another Y-pipe. One can also use a new air filter, which needs to be large enough for the total air volume flow rate of the engine, i.e. twice the volume flow rate of one of the previous air filters (see Fig. 5.7).

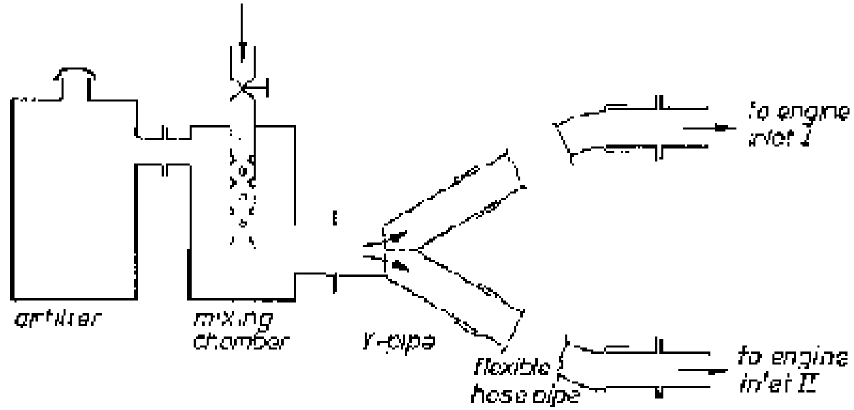


Fig.5.7: Mixing chamber connected to a V-type engine (engine)

5.3 Control in Dual Fuel Mode

5.3.1 Manual Control

There are two different ways to control the power and speed of a dual fuel diesel gas engine. As only the fuel flow (but not the airflow) is to be varied, one can control the supply of both

- **the diesel fuel, and**
- **the fuel gas.**

Almost every diesel engine is equipped with a speed governor. Governors may be different in their design and function. The main difference is determined by the original use of the engine, whether for a vehicle or for stationary purpose. The governor/injector system should be retained in order to facilitate operation on diesel fuel alone whenever required.

Stationary engines mostly have a manually adjustable lever to set the required speed.

The governor will act to vary the amount of fuel injected in order to maintain the required speed at any load. However, the speed will be constant within certain limits only, usually + 2 ... 5%. The control characteristics of the governor are usually very "steep", i.e. within a certain small variation of speed the control rack hence amount of fuel are varied from 100% to minimum (idling). For very precise speed control the lever therefore sometimes needs to be adjusted marginally by hand after a larger change of power demand unless a particularly accurate governor is employed.

When the engine is started on diesel fuel and the biogas valve is slowly opened the governor senses an increase of speed which results from the increase of total fuel. The speed increase effects a change in the centrifugal mechanism and the control rack is moved to reduce the injected fuel. With more biogas being introduced; diesel fuel is furtherly reduced. Should the governor have a minimum (idling) position, the diesel fuel amount cannot be reduced by the governor to less than the set idling amount, so that further biogas will cause a speed increase of the engine. The idling adjustment screw can be used to set the amount of pilot fuel needed, i.e. 15 . . . 20 % of rated power. The performance control is now effected

by variation of the biogas supply alone until the biogas supply itself becomes too low for the required power and the governor increases diesel fuel to a larger than the ignition portion only.

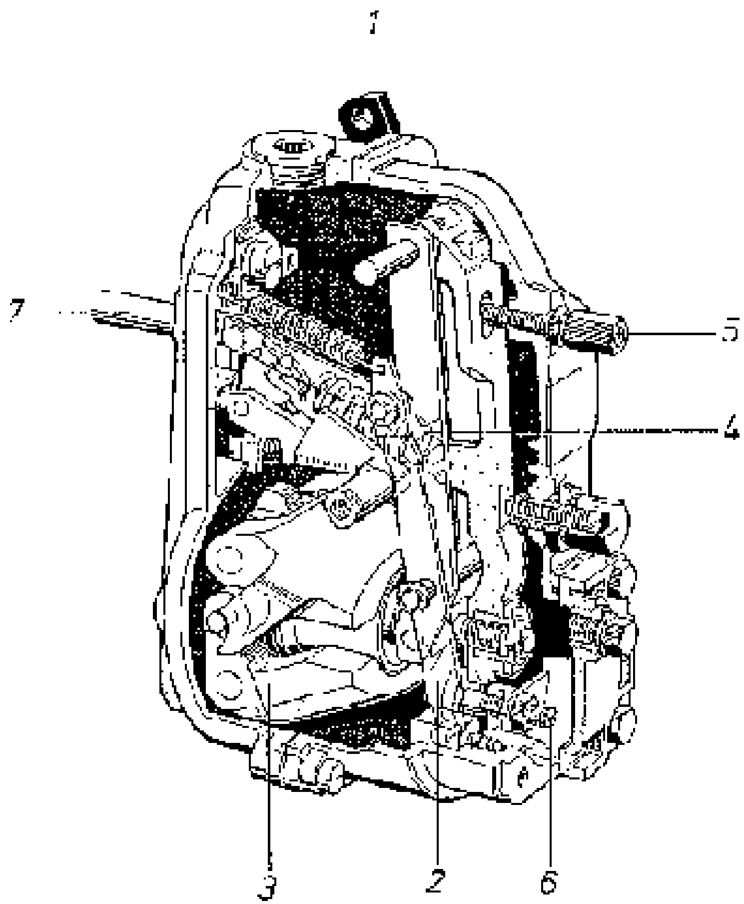


Fig. 5.8: Governor for diesel fuel injector pump (Bosch).

1 control lever, 2 governor lever, 3 centrifugal weights, 4 governor main spring, 5 idling adjustment, 6 full load adjustment, 7 control rack to injector pump

If the governor has no adjustable idling mechanism and too much biogas is

introduced, the injected diesel fuel is gradually reduced to less than about 10 ... 15% of its original amount. Sufficient ignition is no longer guaranteed, the engine will begin to stall and finally come to a halt.

The maximum possible biogas input is reached just before the engine starts to run unevently. The relevant position of the manual biogas valve should be marked or fixed to prevent a biogas oversupply. At any different speed or power required, however, the gas control valve position will have to be adjusted. The simple manual method of control therefore needs either a guaranteed continuous load on the engine or an operator nearby to adjust the gas flow according to the engine load.

Small variations of load will cause small changes in speed. The driven machine's operation or performance curves will determine to what extent such changes in speed are allowable, i.e. how far the engine/machine set can operate without constant supervision. At constant supply of biogas an increase of power demand will be automatically compensated by increase of diesel fuel injection, while a decrease in power demand may cause dangerous overspeeding if the governor had been blocked by the idling screw and cannot cut off the ignition fuel.

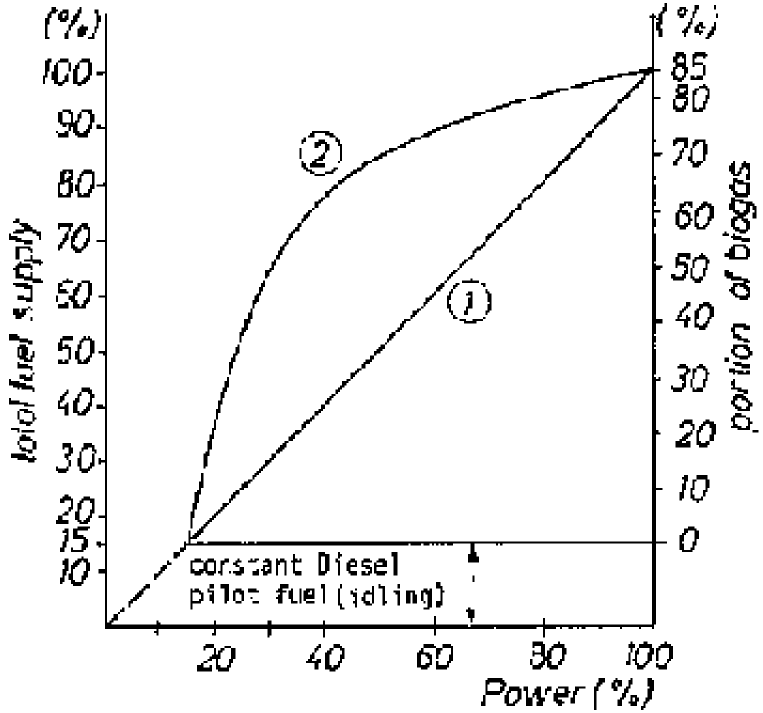


Fig. 5.9: Fuel supply vs. power output Diesel pilot fuel constant, biogas controlled (manualy or automaticaly) according to power demand.
1 total fuel, 2 portion of biogas in total fuel (simplifying assumption: sfc=constant).

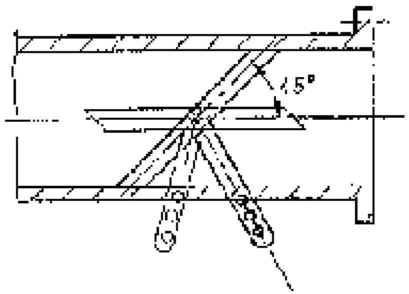


Fig.5.10: Butterfly gas control valve with elliptic butterfly for small angular movement (45º)

5.3.2 Automatic Control

For some applications automatic control is required, e.g. for electric generators unless the electric power is very stable or the electrically driven machines can tolerate the speed/frequency fluctuations. The gas flow needs to be controlled by a butterfly valve which is operated between fully open and fully closed position by short movements, i.e. a 90° or smaller angular movement, and with little force (Fig. 5.10).

The butterfly valve can be operated by a solenoid mechanism (positioner/actuator) which receives its impulse from an electronic control unit which again has a sensor for the engine or generator speed or frequency. The minimum diesel fuel for ignition is set at a fixed point in the injector pump whereby the control rack is blocked in the respective position, i.e. the fuel injected does not change with speed alterations. The idling adjustment screw on the governor can be used for setting the constant minimum pilot fuel injection.

This arrangement does not only require expertise in modification of injector/governor units and electronic equipment. It also needs a secure overspeed protection' as in case the load drops to zero (e.g. generator switch tripped) the engine can overspeed. The governor in this case cannot reduce the diesel pilot fuel injected anymore. If the control does not immediately close the gas control valve, the engine can be driven to selfdestruction. The overspeed device will have to act upon the air supply and/or the diesel fuel supply using solenoid valves.

In case the engine is needed to operate on diesel fuel alone the additional bolt

inserted into the governor housing to block the control rack can be turned backwards or removed. Even at any lower rate of biogas supply the engine will operate to its required performance. If the gas is not sufficient to produce the required power the governor will increase the amount of diesel fuel automatically. The speed droop of the electronic control unit will, however, have to be smaller than that of the mechanical governor so that the gas valve is opened to utilize all possible gas first before the diesel fuel is increased by the mechanical governor.

A possible alternative to the electronic speed control is a separately mounted mechanical governor which is driven with a V-belt from the engine's pulley on the crankshaft. Mechanical governors are usually reliable, less prone to maladjustments and comparatively easy to install. A separate mechanical governor also does not interfere with the function of the integrated governor acting on the fuel injection pump.

The mechanical governor of the engine can in principle also be used for speed control. However, this involves elaborate modifications as the governor movement needs to be transferred to outside its housing while the control rack for the injector pump is disconnected and fixed in the appropriate position for pilot (ignition) fuel injection. The governor movement and the movement of the gas butterfly valve lever need to be tuned upon each other. A sound knowledge of control mechanics and the characteristics of the governor is also necessary for such modification. Governor types from vehicle engines are usually not suitable as they often only control the low speed (idling) and overspeed range while the control within the normal operation range is done by the driver's pedal, i.e. by an operator. Last but not least a governor modification cannot easily be reversed in cases where biogas is not available at the full rate and the engine would have to

be operated on diesel fuel.

For most applications the electronic or separate mechanical governor should be given preference.

5.3.3 Semi-automatic Control

The normal self-governing mechanism of the diesel engine can however also be used without separate control of the gas supply. This is achieved when biogas is supplied at a lower rate than the maximum possible, i.e. as long as the diesel fuel portion is larger than what is necessary for ignition.

The larger portion of diesel fuel leaves room for the governor to control the engine's power/speed by increasing and decreasing the diesel fuel portion while the gas supply is set at a constant rate.
If for instance the gas portion of the total fuel supply is only 60% at rated power, the diesel fuel portion will be 40%, but can be decreased by 25% to the minimum necessary 15%. This reduction of the total fuel supply of 25 % can hence control the power output by about 25%.

The anticipated operations of the driven machine will determine the necessary changes in power demand. The fuel portion constantly suppliable by biogas is a function of these power changes. The diagram in Fig. 5.11 gives the maximum percentage of total fuel suppliable by biogas in relation to the anticipated fluctuation of the power demand.

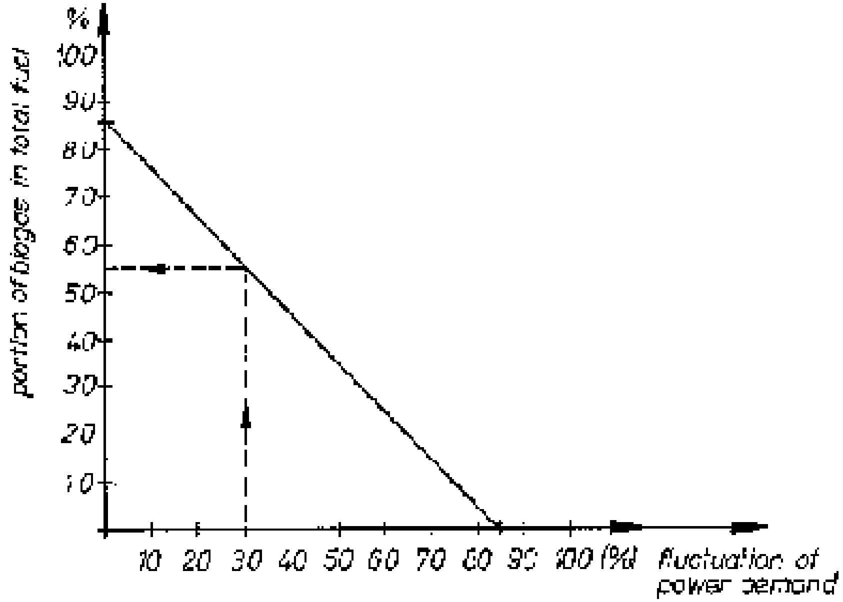
Example:

- Power range required by driven machine: 21 ... 30 kW;
- Speed: constant, 1500 1/min;
- Engine type: diesel gas engine;
- Control: gas manually set, uncontrolled, diesel fuel controlled by governor

Step 1:
Determine anticipated power variation in percent below maximum power required:

$$\%\ \text{Variation} = \left(1 - \frac{21}{30}\right) \cdot 100 = 30\%$$ **Equ. 5.6)**

Step 2:



Use diagram in Fig. 5.11: with load variation of 30% constant biogas supply = 55% of total fuel supply.

The diesel fuel supply will hence vary from 45 % at full load, i.e. 30 kW, to 15% at anticipated partial load, i.e. 21 kW. Should the load be reduced further the governor will reduce the fuel injected subsequently to less than 15% and stop the engine unless it is blocked by the idling screw. In this case the biogas supply should be manually reduced to a still lower constant admission rate. Operation with insufficient ignition fuel is to be avoided. An oversupply of total fuel which is possible as in automatic control needs to be safely excluded.

The "semi-automatic" method may be convenient for certain modes of operation. However, the possible load variations need to be carefully anticipated or tested. Last but not least the maximum possible substitution of diesel fuel by biogas cannot be fully utilized in this case.

For further information on operation of the engine with the driven machines refer to Chapter 7.

5.4 Performance, Operational Parameters

Diesel gas engines have been in use for a variety of purposes using gas such as natural gas, sewage gas, biogas, gas from waste disposal dumps and even carbon monoxide. The performance of diesel gas engines in dual fuel mode, i.e. using two fuels at a time, has been found to be almost equal to the performance using diesel fuel alone as long as the calorific value of the gas is not too low, i.e. as long as the fuel gas volume necessary for the power required is not too high.

The inlet channel and manifold of a diesel engine are dimensioned in such a way that at the maximum speed and power output of the engine sufficient air can be

sucked in to obtain an air/(diesel) fuel ratio which is optimal for operation at this point, i.e. excess air ratio λ = 1.2 ... 1.3. When the diesel fuel is reduced and an air/gas mixture is sucked in instead of air alone, part of the air is displaced by the fuel gas. With less air fed to the engine and an excess air ratio necessarily maintained at λ = 1.2 ... 1.3 the total fuel input (diesel and fuel gas in kJ/s) will be less than the fuel input in diesel operation. As a result of this reduction in both fuel and air, the maximum power output at high speed in dual fuel mode may be less than in diesel fuel operation.
This decrease is however less significant than in modified petrol engines.

For operation at lower and medium speeds, however, the air inlet is larger than necessary ("overdimensioned") and allows a relatively larger amount of air/fuel mixture to be sucked in. Hence the power output will not be significantly lower than in diesel operation. In some cases even more power can be obtained if the dimension of the inlet allows more air/fuel mixture in than required for the original power in diesel fuel operation. Operation at a higher power output than originally designed for may however be harmful to the engine and should in any case be avoided.

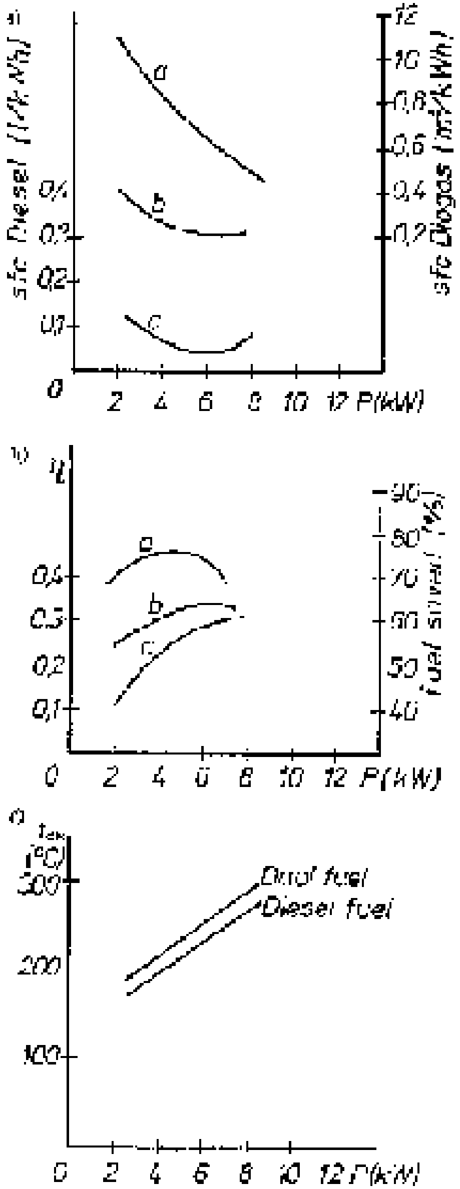


Fig. 5.12: Performance charts of a 10 kW single cylinder gas engine with biogas at n=1500 min^{-1} (from[11]).

a) a sfc biogas in dual fuel mode , b sfc diesel in diesel fuel mode, c sfc diesel in

dual fuel mode
b) a diesel fuel saved, efficiency dual fuel mode
c) exaust gas temperature at silencer outlet

Fig. 5.12 shows the performance of a single-cylinder diesel gas engine. Note that at high speeds the substitution of diesel fuel by biogas is reduced as a result of air being displaced by biogas at a rate too high to obtain complete combustion at full power. At higher biogas inputs the excess air ratio decreases to λ = 1.1 or less, causing smoke and a drop in power.

To predict the power output of a diesel engine converted into a diesel gas engine the following has to be observed:

- Operational speed: as long as the anticipated operational speed is less (< 80%) than the maximum rated speed specified for the engine it may be assumed that the engine will perform equally well in dual fuel mode as in diesel fuel mode. Substitution of up to about 80% of diesel fuel by biogas is possible without affecting the power output.

- Substitution of diesel fuel: the rate of substitution can be less than the maximum possible, i.e. less than about 80% (because of low availability of biogas or anticipated problems with injector overheating); the decrease in performance is insignificant.

- Operational power: for engines operating in continuous service, i.e. more than one hour at one time, the normal operational power should be at about 80 . . . 90% of the rated maximum power. The diagrams in Fig. 5.12 show that the

specific fuel consumption has the lowest value at between 70 . . . 90% of the rated power.

However, the operational power output of the engine is largely dependent on the power required by the machine or equipment being driven. The matching of both engine and driven machine requires careful consideration in order to ensure the optimum operation of the engine (see Chapter 7).

The exhaust gas temperature in dual fuel mode is higher than in diesel fuel mode as the combustion velocity is lower, i.e. the combustion process may not be completed when the exhaust stroke begins. It is therefore more important to be observed at high engine speeds and high rates of substitution by biogas. In order to prevent the exhaust valves from becoming overheated, the temperature measured at the outlet of the cylinder head should not exceed 550 °C. Reduction of temperature is achieved by a reduction of speed and/or biogas rate.

5.5 Modification of a Diesel Engine into a Gas Otto Engine

5.5.1 Necessary Alterations

The principal functioning of an Otto engine has been dealt with earlier in Chapter 3.3.2. The modification of a diesel engine into an Otto engine, i.e. spark ignition engine, involves a major operation on the engine and the availability of certain parts which will have to be changed (see Fig. 5.13). The main changes are the

- removal of the injector pump and injection nozzles,
- reduction of the compression ratio to ε = 10 . . . 12,
- mounting of an ignition system with distributor (cum angular gear), ignition coil,

spark plugs and electric supply (alternator),
- provision of a mixing device for the supply of an air/fuel mixture with constant air/fuel ratio (venturi mixer or pneumatic control valve).

5.5.2 Removal of Injection System

The removal of the injection system is the easiest part and does not require too much expertise. However, the gear drive for the injector pump (see Fig. 3.7) has to be carefully disassembled as it may be needed to drive the distributor of the spark ignition system. If this is not required, the engine housing needs to be closed off with a cover (to be manufactured accordingly) to prevent dirt from entering the crankcase and loss of engine oil.

5.5.3 Reduction of Compression Ratio

The reduction of the compression ratio to ε = 12 or less is essential because at higher pressures spark ignition does not always function effectively. The choice of the compression ratio also depends on the possible variety of gases to be used. Natural gas with a considerable percentage of early igniting components (butane) requires a relatively low compression ratio, and LPG (propane) also tends to self-ignite a lower temperatures (compression) than pure methane (see table of fuel properties in Appendix II). The compression ratio of industrially converted engines is therefore found in the range of ε = 10.5 ... 11.5 to facilitate the use of a variety of gases.

A change of the compression ratio is effected by enlarging the volume of the compression chamber Vc (see Equ. 3.8). It can be performed by:

- exchanging the piston(s) for one that effects a lower compression ratio,
- machining off material from the piston
- machining off material from the combustion chamber in the cylinder head,
- exchanging the standard cylinder head for a special low compression head,
- using a thicker cylinder gasket.

The shape of the combustion chamber also plays an important role.
While for the performance of a diesel engine an antechamber or swirlchamber arrangement is often advantageous for efficient combustion, an Otto engine requires an evenly shaped combustion chamber to facilitate even combustion propagation and pressure rise in the homogeneous air/fuel mixture. A direct injection-type diesel engine is therefore the best option for transformation into an Otto engine (see Fig. 3.12 diesel engine combustion chambers).

Exchanging the piston or the cylinder head is undoubtedly the most elegant method but it is restricted to engines for which manufacturers or suppliers offer such parts.

Machining off material from the piston top is usually possible but has an effect on the dynamic balance of the moving parts of the engine. It should be done in such a way that the material thickness of the piston top does not become critically low. (Diesel pistons usually have a strong top because of the high peak pressure, about 100 bar, active near TDC.) In machining off material from the combustion chamber in the cylinder head one needs to carefully consider the material thickness around the valve seats which should under no circumstances be weakened. A geometrical and even shape of the combustion chamber should be aimed for.

The use of a thicker gasket or insertion of a ring or spacer with the shape of the cylinder head gasket is only possible where appropriate material is available and where the joining surface, bolt length, etc. allows enlargement of the distance between cylinder block and head.

The additional volume to be created can be established as follows:

- Determine the previous volume V_{prev} of the combustion chamber by either calculating, using the previous compression ratio (Equ. 3.8), or by measuring the volumes of the cylinder head and the cavity in the piston (if any) with a liquid and adding the discshaped volume created by the distance between the piston at TDC and cylinder head plane (including the original gasket thickness). A disc-shaped or cylindrical volume is given by

$$V = \frac{\pi d^2}{4} \cdot h$$ **(Equ. 5.7)**

where h = (cylindrical) height of the disc.

- Determine the new volume V_{new} of the combustion chamber according to the required compression ratio (Equ. 3.8).

- Establish the additional volume /V to be created:

$\Delta V = V_{new} - V_{prev}$ **(Equ. 5.8)**

If the additional volume is created by increasing the gap between cylinder head and gasket the additional thickness Ah is found:

$$\Delta h = \frac{4 \cdot \Delta V}{\pi \cdot d^2}$$ **(Equ. 5.9)**

If material is machined off from the piston or cylinder head it may be easier to determine the new volume by filling the respective cavity with liquid, measuring its volume and working towards the final volume in steps. A uniform amount of volume addition and shape of the new combustion chamber is essential for every cylinder in multicylinder engines to ensure an evenly distributed performance.

5.5.4 Addition of Ignition System

The type of electric ignition system chosen depends on the number of cylinders of the engine. In a single-cylinder engine a transistor-type ignition system can be used. A magnet is attached to the flywheel of the engine and a pick-up is mounted on the casing so that when the magnet on the flywheel passes close to the pick-up a spark is initiated by a transistor and the ignition coil. This system will cause a spark at every revolution of the engine, i.e. one at the beginning of the working stroke and another one in the overlapping phase between exhaust and suction stroke where it is not utilized but does not do any harm. Such simple systems are available from various manufacturers and are widely used in single-cylinder motorcycles. The positioning of both the magnet and that of the pick-up have to be well synchronized with each other and with the position of the piston or its actual crank angle. Ignition timing is essential both for good combustion and optimum performance of the engine. Mounting the pick-up on a small plate with slots or long holes allows for fine tuning after recommissioning. Once properly set, this type of ignition does not need to be readjusted after a certain period of operation as it is not subject to wear and tear unlike systems using breaker points.

Unless the supplier of the ignition system stipulates a different method, fixing the magnet on the flywheel can best be done by drilling an appropriately sized hole into the flywheel from either the outer circumference in radial direction (towards the center) or in axial direction near the outer circumference (observe material thickness). The hole should not be wider or deeper than the magnet itself as it has to be exactly filled by the magnet for reasons of balance. The magnet is glued in with a two component epoxy resin and additionally secured with a horizontal pin in the case of insertion in radial direction (Fig. 5.14).

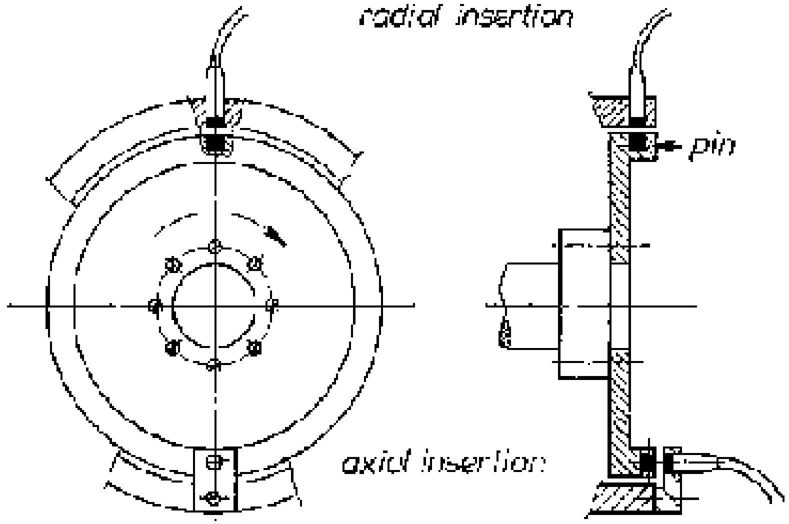


Fig. 5.14: Fixing the pick-up in the casing and the flywheel, two different versions; upper half: radial insertion, lower half: axial insertion.

The same system can also be used for a twocylinder engine if the crank angle between the two cylinders is 360°, i.e. if both pistons are at TDC at the same time. The transistor unit can then be connected to two ignition coils in series, each one working on half the voltage of the system. Both spark plugs will fire at the same time, one igniting the mixture in the respective cylinder, the other one firing

without effect during the overlapping phase of the other cylinder.

Diesel engines modified into Otto engines still require a disconnection of the injector pump. The pump would immediately be damaged when running dry, i.e. without diesel fuel, and can cause further damage to the engine. Should the pump camshaft be indispensable, ea. to drive the original governor which may be used for automatic control, at least the plungers cum roller and spring need to be removed.

Engines with more than two cylinders require an ignition distributor of the type commonly found in vehicle-type Otto engines. The key issue is the connection to the camshaft or the gear drive of the former injector pump as both provide the necessary speed, i.e. half the engine's crankshaft speed. Depending on the possible mode of connection and space a 90° angular gear drive with a transmission rate of 1: 1 may be needed. The distributor will have to be mounted in a way that it is free to be turned in its clamp holder, preferably by 360°. The ignition can then be set by choosing the most suitable position for the distributor. This is especially useful when a diaphragm for advancing the ignition by suction pressure from the manifold is attached as it requires extra space.

Distributors from vehicle Otto engines are usually equipped with centrifugal advancing mechanisms. which advance the ignition in relation to the engine speed as required. They therefore require one specific direction of rotation of the rotor, i.e. they need to be connected to a shaft rotating in the same direction. The opposite direction of rotation would cause a delay in the ignition and poor performance at higher speeds.

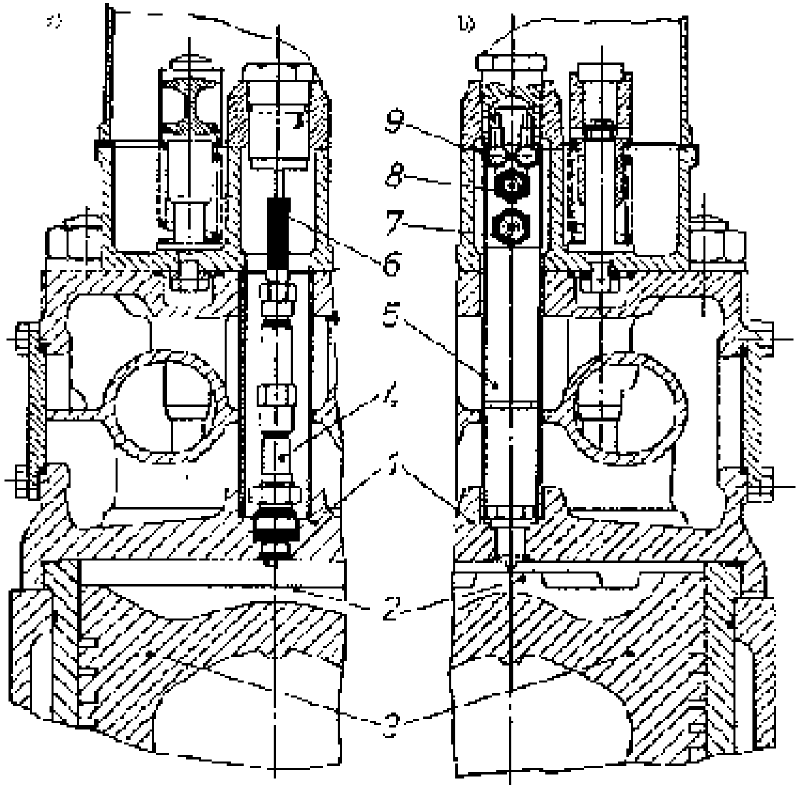


Fig. 5.16: Cylinder head modified with spark plug on increased combustion chamber volume (a) vs. original diesel version with injector (b)

1 cylinder head, 2 combustion chamber, 3 piston, 4 spark plug, 5 injector nozzle, 6 ignition cable conection, 7 fuel supply from main injector-pump, 9 cooling oil connections

Matching of the distributor model with the direction of rotation available from the engine is therefore essential.

The coordination of the distributor cable outlets with the engine cylinders must consider the "built-in" firing order of the engine. To find out the correlation

between the position of the piston and the stroke of the process for any cylinder and the respective position or angle of the distributor/ camshaft/crankshaft, one can open the cylinder head cover and carefully turn the engine's crankshaft in the normal operating direction. Use a thin screwdriver and insert it carefully through the hole of the spark plug of the first cylinder to sense the piston's movement towards TDC. If both the inlet and outlet valves are firmly closed at TDC and remain closed even when the crankshaft is turned to either side by about 90° this TDC position is the one where the working (combustion) stroke begins, i.e. where a spark is needed. This cylinder's spark plug will have to be connected to the respective contact in the distributor cap to which the distributor rotor points. If the rotor does not point to any contact the entire distributor will have to be turned in the opposite direction of the rotor's rotation until the breaker prints open. As at this position the respective cylinder will be ignited at TDC, the ignition cable of the cylinder concerned will now have to be connected to the distributor cap contact, to which the rotor points. The precise point of ignition before TDC will be finally set after all cylinders have been connected to the distributor.

As the next step the crankshaft will have to be turned in the direction of operation until another piston reaches TDC with valves closed. Connect the ignition cable for this cylinder to the next cable socket in the distributor cap following the rotor's direction of rotation. Continue the procedure until all cylinders are connected to their respective sockets on the distributor cap.

To obtain the required point of ignition, i.e. about 20 . . . 22° crank angle before TDC, the whole distributor can now be turned against the direction of rotation of the distributor rotor by about 10°. Should a stroboscope light be available, one center punch mark on the engine (flywheel) housing as well as one mark for 0°

TDC and 20° before TDC each on the flywheel itself will be useful for the precise tuning of the ignition after start-up.

Installation of the spark plug in the cylinder head requires careful craftsmanship. The removal of the injector jet leaves a hole which may be used if

- the hole is not bigger than the core diameter of a standard spark plug thread (three sizes available!),

- the cylinder head thickness corresponds with the length of the threaded part of the spark plug (two standard lengths available),

- the extension of the hole including threads does not considerably reduce the material thickness towards the valve seats; otherwise cracks can easily result and the seats may become loose.

The injector hole will have to be drilled to the size necessary to tap the threads (refer to respective standards). If the material of the cylinder head is too thick, the hole for the body (not threads) of the plug can be extended until the beginning of the thread is on a level with the combustion chamber surface and the electrodes protrude slightly (not more than 2 ... 3 mm into the combustion chamber' see Fig. 5.16a). Any protruding of the spark plug threads into the combustion chamber may cause damage to the valves or piston if they touch each-other. Furthermore removal of the spark plug can become almost impossible when the protruding threads are burnt and filled with hard combustion deposits.

If the spark plug thread diameter is smaller than the hole an appropriately sized bush, threaded internally and externally, has to be inserted, possibly with a collar

and screwed in from outside.
A possible leakage must be carefully avoided. Any liquid cylinder head gasket material or "loctite" may be applied when screwing the bush into the cylinder head, but keep the threads for the spark plug clean.

It should not be forgotten that the ignition system requires a source of electricity, i.e. an alternator cum batteries and regulator which can be adopted from any vehicle-type engine. Some diesel engines are equipped with alternators and batteries for the electric starter and other purposes anyhow.

Last but not least it must be clearly understood that the above modifications and the machining of the cylinder head as well as piston, etc. require a well equipped machine workshop, precision and associated expertise.

5.5.5 Addition of Mixing Device and Speed Control

The choice of the mixing device to be used follows the same criteria as for any other Otto engine modified for the use of gas. A venturi mixer, a gas mixing valve or even a simple mixing chamber for a limited range of operation can be used. The design and dimensioning of mixing devices for Otto engines are explained in more detail in Chapter 6.

In a case where it is possible to connect the ignition distributor to the camshaft, the original speed governor can be retained and utilized for speed and power control. The movement of the control rack may, via appropriate lever and rod, be connected to the butterfly valve of the gas carburetor or venturi mixer. The injector pump housing and its camshaft may have to be retained also in cases where the governor is attached to the outer end of the injector pump, using the pump shaft for its motion. External control devices as described for dual fuel operation can also be used.

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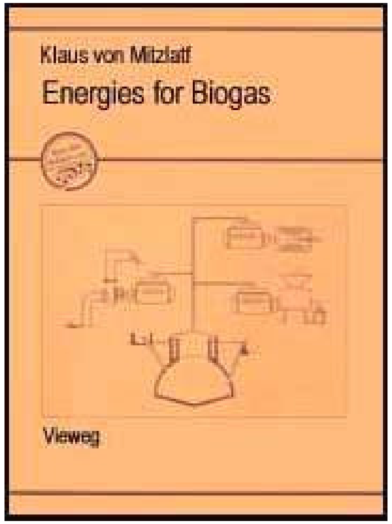

Engines for Biogas (GTZ, 1988, 133 p.)

6. The Gas Otto Engine

6.1 Necessary Modification

The modification of an Otto engine (spark ignition, petrol or gasoline engine) is comparatively easy as the engine is designed to operate on an air/fuel mixture with spark ignition. The basic modification is the provision of a gas-air mixer

instead of the carburetor. The engine control is performed by the variation of the mixture supply, i.e. the throttle valve position as has been the case with petrol fuel.

An increase in the compression ratio appears to be desirable as it provides an increase of the efficiency of the process from the mere thermodynamic point of view. A lower specific fuel consumption and a higher power output can be expected. The modification is however permanent and prevents operation on original fuel in cases of biogas shortage.

The adjustment of the point of ignition in relation to the slow burning velocity of biogas imposes no specific problem as a standard ignition system provides for adjustments in a sufficiently wide range.

Engines which cannot operate on unleaded fuel will miss the lubrication effect of condensing lead especially on their exhaust valves. They are therefore subjected to increased wear and tear in gas operation.

6.2 Performance and Operational Parameters

Gas Otto engines when modified from Otto engines using petrol are found to produce less power than in the petrol version. The reason is the decrease in volumetric efficiency as a gaseous fuel occupies a larger portion of the mixture's volume sucked into the engine than liquid fuel and displaces air accordingly. The liquid fuel has a higher volumetric energy content than gas and also cools the air/fuel mixture when evaporating in the intake manifold. The cooling effects an increase in density, and hence the amount of air/fuel mixture actually sucked into

the engine on a mass basis is higher.

A gas engine, especially when operating on biogas with a large proportion of useless carbon dioxide, can suck a reduced amount of air only to allow room for th necessary amount of fuel gas. As in Otto engines an excess air ratio of $\lambda = 1 \pm 0.1$ has to be maintained and the inlet ducts and manifolds are dimensioned for operation with petrol, the total fuel energy in a mixture of biogas and air is less than in petrol operation. With the decrease in the maximum possible supply of fuel energy or the energy density of the mixture (mixture heating value) the maximum power output consequently decreases in the same proportion. The rate of decrease in power is largely dependent on the volumetric heating value of the gas, e.g. biogas with 70% CH_4 has a higher volumetric calorific value than biogas with 50% CH_4 only. The power output of an engine is therefore higher in operation on gases with high calorific value than in operation on "weak" gases. Biogas (60% CH_4) with a calorific value of H_u = 25 000 kJ/nm³ ranges as a medium weak gas and causes power reductions of about 20% (purified methane or natural gas 10%, LPG 5%). The main effect of the reduction of power is that it needs to be well considered when selecting the power class of an appropriate engine for a given application with a specified power demand (see Chapter 7).

The engine's power and speed control is performed by a variation of the supply of the air/fuel mixture to the engine. This is achieved by the operation of a butterfly valve situated between the actual mixing device and the engine inlet. Closing the butterfly valve effects a pressure drop (throttling effect) in the flow of the mixture by which the cylinder is filled with a mixture at lower pressure p_s, hence with a lower amount of air/fuel mixture on a mass and energy basis. As a result the

power output, the mean effective pressure and the efficiency decrease in controlled (partial load) operation. The effect of the decrease in efficiency is realized in an increase of the specific fuel consumption in partial load operation (see Fig. 6.1). To compensate for the above-mentioned effects the engine should rather be operated at medium speeds but with open throttle. This requires an appropriate combination with the speed and power requirements of the driven machine as explained further in Chapter 7.

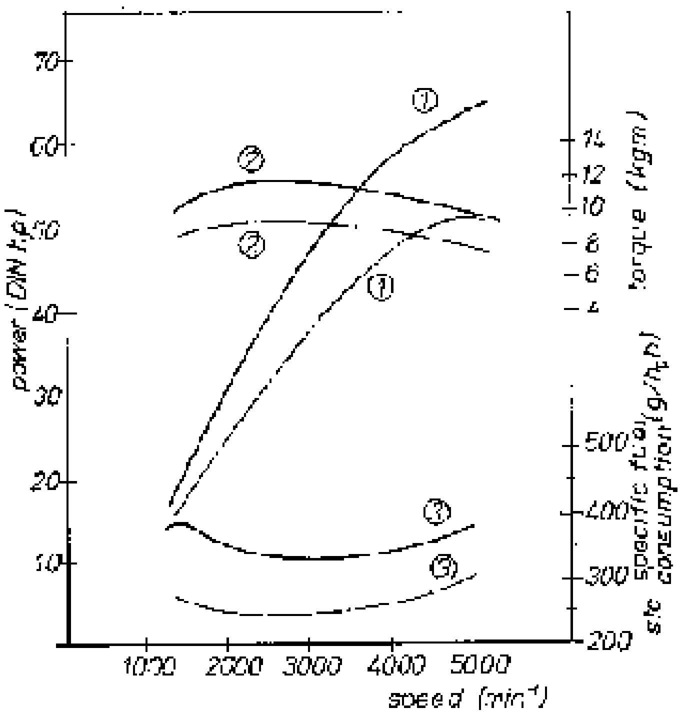

Fig. 6.1: Performance diagram of an Otto engine using liquid fuel (——) and methane(—·—) alternatively (Rodagas).
1 power, 2 torque, 3 specific fuel consumption.

The mixing device has to ensure the provision of a constant air/fuel ratio

irrespective of the actual amount sucked into the engine, i.e. irrespective of the butterfly valve position. This is achieved by adequate design of the mixing device, whether a venturi mixer or a suction-pressure controlled mixing valve. A simple mixing chamber however requires a control of the fuel gas flow together with the main butterfly valve, i.e. it cannot provide a constant air/fuel ratio by its design alone.

Before a specific mixing device is chosen, the necessity/possibility of another type of fuel for cases of insufficient biogas supply should be considered. The different fuels and their technical requirements are given below:

- LPG, natural gas: mixing valve or venturi, with pressure reduction valve (50 mbar) before gas inlet. Maximum compression ratio e = 11. Simple mixing device for biogas can be used with adjustment at gas inlet (for operation at constant conditions).

- Alcohol: carburetor, similar to petrol version but with main jet enlarged in the ratio of calorific values of petrol/alcohol. Petrol carburetor can be retained. Maximum compression ratio ε = 12.

- Petrol: previous petrol carburetor retained or remounted. Maximum compression ratio ε = 9.5 for premium, ε = 7.5 for regular.

6.3 Design of Mixing Devices

6.3.1 Venturi Mixer

A venturi mixer utilizes the same fluid-mechanic effect as a standard carburetor,

i.e. the change in airflow quantity and velocity causes a change in pressure at the channel contraction which in turn effects a change in flow of another medium (fuel) to join and mix with the main airflow in the required proportion.

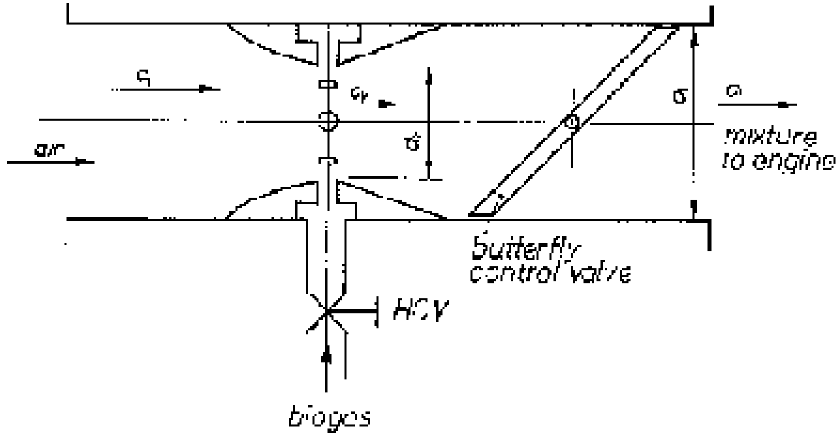

Fig. 6.2: Venturi mixer with gas supply through several bores.
c_1 velocity at mixer inlet, c_v velocity at venturi contraction, d_i diameter of mixer/engine inlet, d_v diameter of venturi contraction, c_i velocity of mixture at engine inlet.

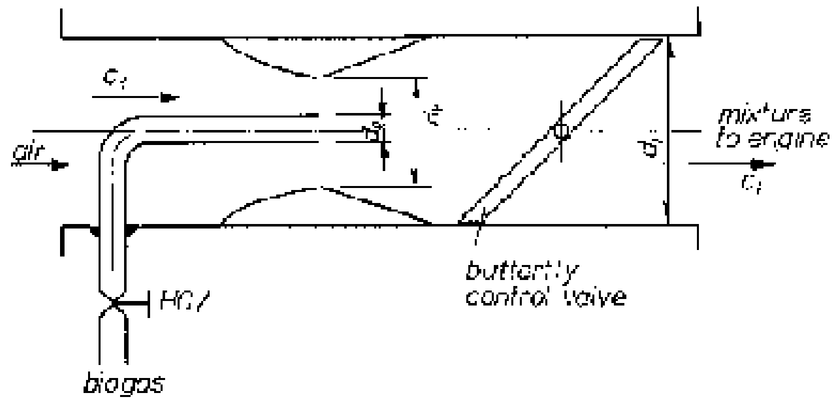

Fig. 6.3: Venturi mixer with a single gas inlet nozzle.

d_g diameter of gas inlet nozzle, other symbols as in Fig. 6.2

The venturi principle functions as follows:

For high air volume flow:

- Air velocity is high.
- Air pressure is low at the contracted cross-section.
- The pressure difference between fuel gas and airstream is high.
- Much fuel gas flows through the openings to mix with the airstream.

For low air volume flow:

- Air velocity is low.
- Air pressure is high at the contracted cross-section.
- The pressure difference between fuel gas and airstream is low.
- Little fuel gas flows through the openings to join the airstream

The following procedure shall give a general representation of the dimensioning of a(self-made) venturi mixer.

Step 1:

Determine the volumetric intake V, (in m^3/s) of the engine as a function of engine cubic capacity V_1 (in m^3/s) at rated or maximum operational engine speed n (in 1/min or rpm), see Equ. 3.17 and 5.4:

$$V_1 = \frac{V_h}{2000} \cdot \frac{n}{60} \cdot \eta_{tot}$$

Step 2:

Determine the mean intake velocity c; (in m/s) of the venturi mixer using the channel's cross-sectional area Ai (in m²), see Equ. 5.1 and 5 5 :

$$c_i - \frac{V_i}{A_i}$$

whereby $A_i = \frac{1}{4} \cdot d_i^2 \cdot \pi$

The cross-sectional dimension of the venturi mixer should be equal to that of the manifold. The intake velocity c_i almost equals the velocity c_i of the air coming from the air filter when the throttle is fully opened. In a controlled position the velocity before the butterfly valve (in flow direction) is reduced. The dimensioning of the inlets for fuel gas, however, needs to consider the fuel requirement at unthrottled operation for maximum performance, i.e. at maximum intake.

Step 3:

Determine the cross-section of contraction. The contraction in the venturi mixer will cause the airflow velocity to rise as a linear function of the change in the cross-sectional area. The velocity at the contraction or "bottleneck" of the venturi c_v should not exceed c_v = 150 m/s at maximum flow rate. The "bottleneck" or venturi area A_v is found by

$A_v - A_i \cdot \frac{c_i}{c_v} \geq A_i \cdot \frac{c_i}{150 m/s}$ **(Equ. 6.1)**

Its diameter d_v is found accordingly:

$d_v \geq \sqrt{\frac{4 \cdot A_v}{\pi}} \geq \sqrt{\frac{4 \cdot A_i \cdot c_i}{\pi \cdot 150}}$ **(Equ. 6.2)**

The shape of the contraction has an influence on the flow in a sense that the more abrupt the change in area is, the more extra friction and separation of the flow from the channel wall occur.
The venturi shall therefore be evenly shaped following the example given in Fig. 6.2. The aperture angle on the downstream side shall not exceed 10°. The contraction side upstream is not so sensitive and is often shaped in a roundish profile as can be seen in Fig. 6.2. Standard carburetors use similar venturi profiles.

Step 4:

Determine the required biogas fuel flow. The main parameters for the determination of the fuel flow are the

- **engine operational power,**
- **calorific value of the biogas as per volume ($H_{u,vol}$),**
- **specific fuel consumption of the engine or the efficiency respectively.**

The specific fuel consumption of the engine or the efficiency are not always known especially in second-hand or reconditioned engines. However, as a rough figure for Otto engines η = 0.25 or sfc = 4 kWh fuel/kWh mech. energy can be chosen.

The fuel and the gas volume flow required can be calculated in accordance with the procedure used in Chapter 5.2.1.3 for diesel engine mixing chambers. In an Otto engine, however, the fuel gas provides 100% of the required fuel as no other fuel (dual fuel mode) is supplied, i.e. Step 2 in the above-mentioned procedure is not required for Otto engines.

Step 5:

Determine the area of the fuel gas inlet, Ag. The fuel gas inlet at the bottleneck of the venturi jet can have different shapes (see Figs. 6.2 and 6.3):

- several openings around the circumference of the venturi jet being fed by a ring channel, or
- pipe with-one opening.

When the second alternative is chosen the area occupied by the fuel gas pipe Ag in the core of the venturi has to be subtracted when establishing the bottleneck cross-sectional area of the venturi.

The effective area A_v^* **is therefore**

$A_v^* = A_v - A_g$ **(Equ. 6.3)**

whereby $A_g = \frac{1}{4} \cdot d_g^2 \cdot \pi$

The flow velocity c_v **in the anular clearance at the area** A_v^* **shall also not exceed 150 m/s. The cross-sectional area of the gas inlet** A_g **is then established similar to**

the procedure in Chapter 5.2.1.3.

$$A_g - \frac{fc}{c_g}$$

whereby the flow velocity of the fuel gas in the jet/nozzle is

$$c_g = \sqrt{\frac{2 \cdot \Delta p}{\rho} + c_{bg}^2}$$

The active pressure difference Δp for the fuel gas flow is established between the pressure in the gas supply pipe before the mixer (i.e. the biogas plant pressure minus the pressure losses caused by the flow resistance in the gas piping system up to the connection at the mixer) and the pressure in the venturi bottleneck where the gas flow joins the airflow.

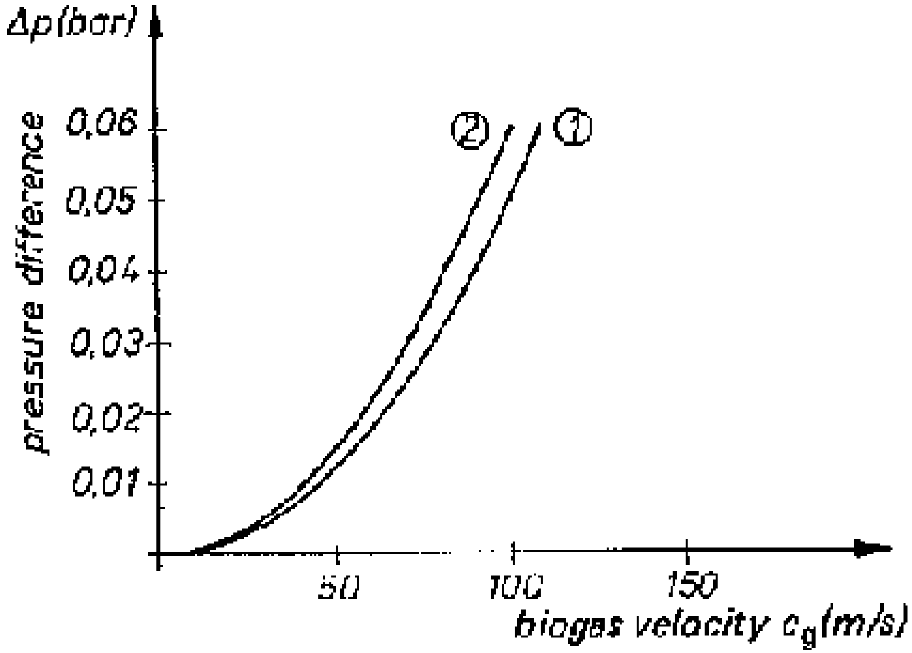

Fig. 6.4: Biogas flow through a nozzle cg as a function of the active pressure difference /p.
1 gas density p=1.2 kg/m³.

The pressure in the gas supply pipe ranges at 0.005 ... 0.02 bar gauge. The pressure in the venturi bottleneck is a function of the contraction of the venturi, the actual airflow rate and the pressure reduction caused by the air filter. It can be calculated
using Bernouilli's equation (see Equ. 5.2):

$$\frac{p_1}{\rho} + \frac{c_1^2}{2} = \frac{p_v}{\rho} + \frac{c_v^2}{2}$$

so that the pressure at the venturi bottleneck p_V is

$$p_v = \rho\left(\frac{c_i^2 - c_v^2}{2} + \frac{p_1}{\rho}\right)$$ **(Equ. 6.4)**

The velocity at the venturi bottleneck is found using the continuous flow equation (see Equ. 5.1) (for incompressible media)²:

$$V = c \cdot A = const$$

with the previously calculated parameters (see Equ. 6.2) for the intake velocity c_i at fully opened throttle

$$c_v = \frac{A_i}{A_v} \cdot c_i$$

The volumetric intake of the engine V_i can also be used to determine the velocity c_v:

$$c_v = \frac{V_i}{A_v}$$

when the venturi bottleneck area A_v is already known. As mentioned, the venturi bottleneck area is to be established in such a way that at maximum volume flow rate V_i the velocity at the bottleneck ranges between c_v = 100 ... 150 m/s. A smaller bottleneck diameter increases the venturi velocity while a larger one decreases it respectively.

As s rule; of thumb and for first calculations the diameter ratio for a venturi may range at d_v/d_i = 0.67 which would result in a velocity ratio of c_v/c_i = 2.25, e.g. a

velocity increase from c_i = 50 m/s to c_v = 112.5 m/s. Fig. 6.5 gives the relation between the diameter ratio and the velocity increase to some selected velocities at the venturi bottleneck.

A scientifically precise calculation of the fuel gas inlet area would require a precise determination of the pressure of the gas at the calibration valve of the venturi, the fuel gas temperature and its composition as well as a high precision manufacturing standard. However, in biogas applications the volumetric calorific value often differs with plant performance and ambient parameters. Furthermore building a venturi mixer should consider its applicability for more than only one specific engine operating at one specific biogas plant.

Due to these non-uniform boundary conditions the layout of the venturi shall be based on assumption of "unfavorable" conditions for the calculations of the calorific value and the pressure drop in the biogas system up to the venturi mixer. If this results in slight overdimensioning of the fuel gas inlet area Ag (whether single nozzle or several bores), the calibration valve can be partly closed, imposing an additional but controllable resistance in the fuel gas supply system. The venturi gas mixer can thus always be adjusted to the actual fuel gas conditions. The additional advantage is that it provides a possibility for manufacturing venturi mixers in small series for similarly sized engines and different gases if required by the market.

A similar approach is used by the commercial manufacturers of pressure-controlled gas mixing valves and venturi mixers, i.e. "Impco" and "Rodagas". All mixing valves and venturis are equipped with a fuel gas calibration valve for mixture adjustment.

The calibration of venturi mixers and gas mixing valves is done during operation at the maximum required power and speed. The gas calibration valve is at first kept fully open and the engine warmed up. It is then gradually closed until the engine begins to lose power/speed, and carefully opened again until the required set point is reached again. The calibration valve should be fixed in this position. An additional control of the CO content in the exhaust gas is recommended; the CO value is optimal at 1.0 + 0.5% Vol.

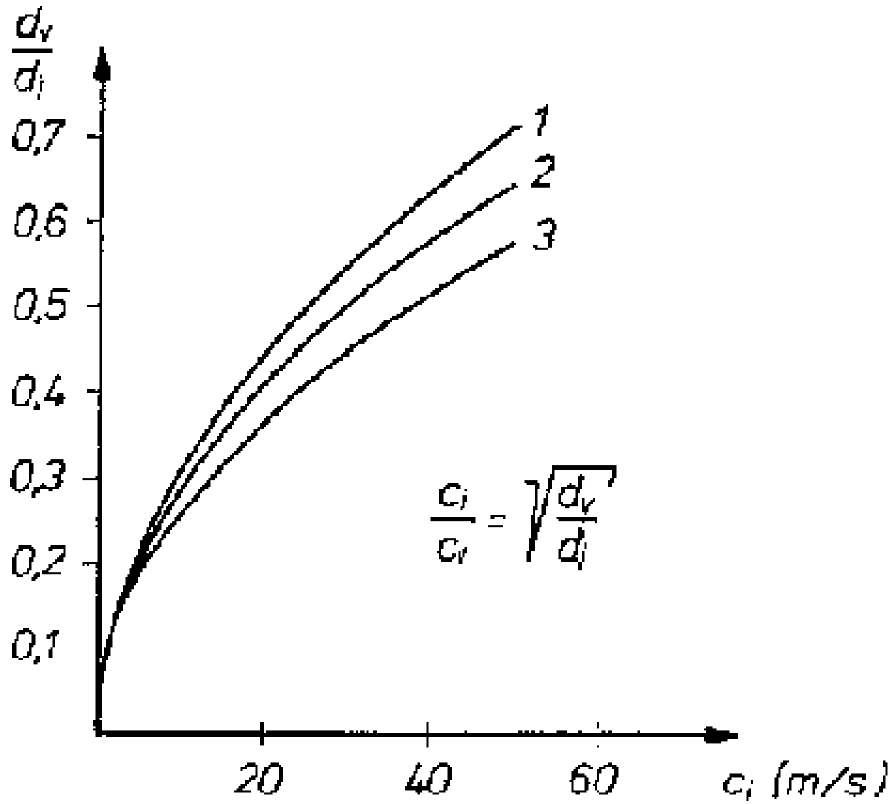

Fig. 6.5: Venturi diameter ratio d_v/d_i as a function of intake velocity c_i and the required velocity at the venturi bottleneck c_v

1 c_v = 100 m/s, 2 c_v = 120 m/s, 3 c_v = 150 m/s

Idling, if necessary, can be adjusted with the lever operating the butterfly valve in such a way that a small clearance is left for the idling amount. Some mixing valves have separate idling screws.

6.3.2 Pressure-Controlled Mixing Valves

Pressure-controlled gas mixing valves are in frequent use for motor vehicles which are driven by LPG. They are manufactured in large series and in different types and sizes for differently sized engines. As the manufacturing of these valves uses rather sophisticated methods and materials not everywhere available, it does not appear recommendable to try self-manufacture. The selfmanufacture of a venturi involves far less effort in terms. of material equipment and skills while it provides a technically sound solution as well.

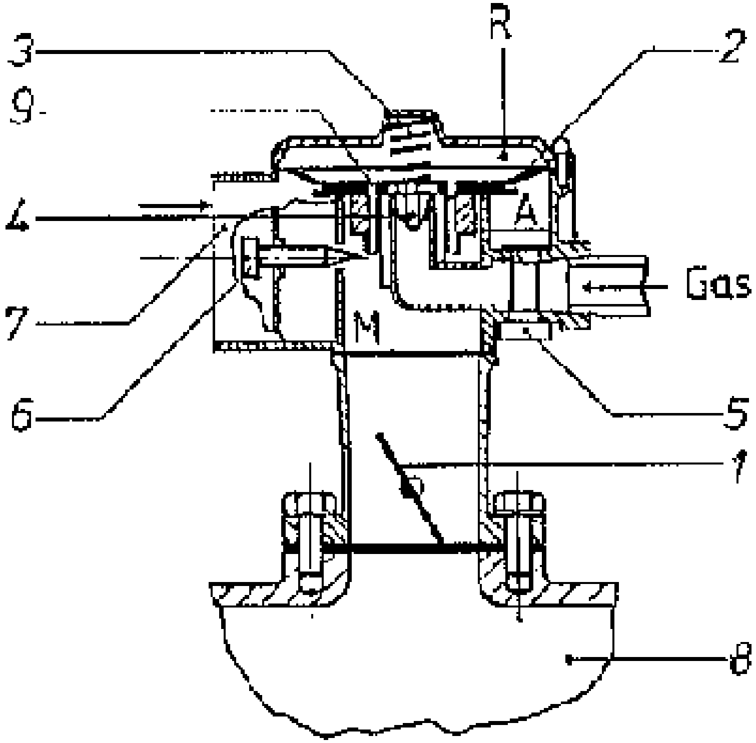

Fig. 6.6: Cross-sectional view of gas mixing valve.
1 butterfly throttle valve, 2 diaphragm, 3 spring, 4 gas valve cone, 5 mixture adjustment valve, 6 air bypass adjustment, 7 air inlet, 8 engine inlet, 9 bore for suction pressure, connects M and R, A space of air inlet before mixing zone, M space of mixture flowing to engine inlet, R space behind diaphragm, connected to M via bore (9).

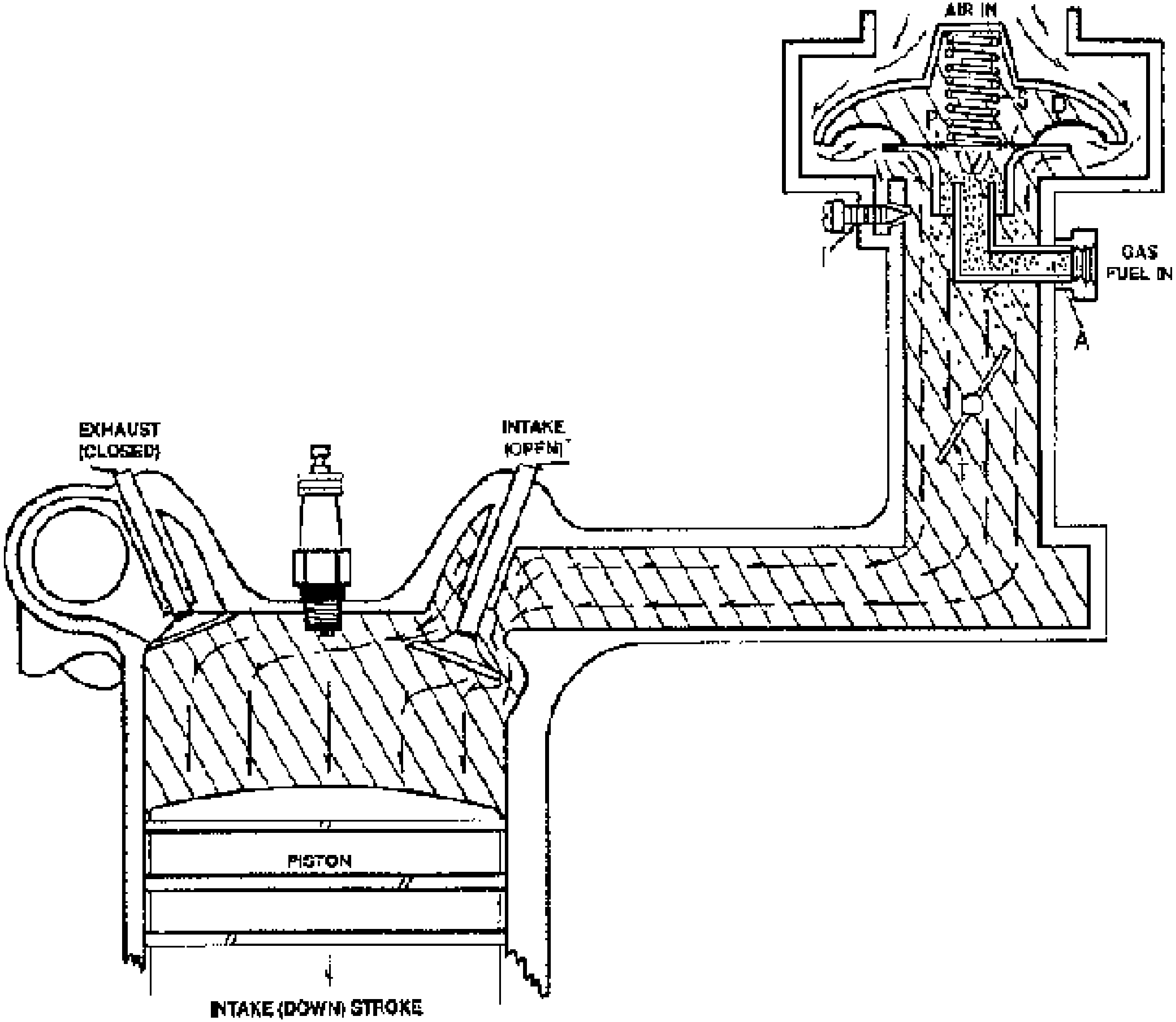

Fig. 6.7: Gas mixing valve in operation, schematic (Impco). S metering spring, D diaphragm, P vacuum transfer passage, V gas metering valve, I idle air bypass adjustment, A power mixture adjustment, T throttle valve.

The operation of the engine (Fig. 6.6) produces a suction pressure ("vacuum") in space M which is passed on to the space R behind the diaphragm [2] via a bore [9]. The space A is connected to the air intake and has almost ambient pressure conditions. The pressure difference between A and R forces the diaphragm to move against the force of the spring [3]. The valve now allows air to pass from A into M through a gradually opened, calibrated ring channel.

Simultaneously the fuel gas can now pass through an opening controlled by the valve cone [4]. The air and fuel mixed in space M are sucked into the engine intake [8] via the butterfly throttle [1]. (See Fig. 6.7 for demonstration of mixing valve in opened position.) The more the throttle is opened for more power, the more the vacuum from the engine intake becomes effective in the spaces M and R, and hence the more air and gas are allowed in through their increased openings. The air/ fuel ratio remains constant as required because both the ring channel and the valve cone have been shaped accordingly. Variations in gas quality (pressure, calorific value) can to a certain extent be compensated by the mixture adjustment or calibration valve [5] which acts as a throttle in the gas supply changing the active gas pressure at the opening, hence the amount of fuel mass entering (in other words, the calorific value of the fuel on a volume basis). A modification of the internal structure of the mixing valve is not practicable and should be avoided.

In places where these LPG mixing valves are easily available they may be used as long as the calorific value of the biogas is not lower than about 25 000 kJ/m³. The gas inlet opening inside the valve has been dimensioned for LPG with a much higher volumetric calorific value than average biogas. The gas inlet will therefore be too small for weaker gases and may produce an air/fuel mixture too lean for good performance.

6.3.3 Introduction of a Constant Pressure Control Valve

A constant pressure valve helps to provide a constant pressure in the biogas supply pipe from the biogas plant. Whenever the biogas pressure is likely to fluctuate in a range of more than 20 mbar or to become higher than 50 mbar before the mixing valve or venturi, a constant pressure (pressure reduction) valve should be introduced and mounted into the biogas pipe before the mixer. Higher fluctuations in biogas pressure would result in corresponding fluctuations of the volumetric calorific value and unbalance the setting or calibration of the mixer, hence the performance of the engine.

Constant pressure valves are always necessary when the biogas is supplied by means of a blower or when LPG is used as an auxiliary fuel in case of biogas shortage. Whenever it can be foreseen that the gas pressure will continuously be rather low (i.e. Iower than 5 mbar) a pressure reduction valve should not be introduced as it produces a small but disadvantageous extra pressure drop even if it is fully open.

Pressure reduction valves are commercially produced in many varieties and specified by their pressure, volume flow rate and type of gas. For more information refer to manufacturers' overview
in Chapter 10.

6.3.4 Simple Mixing Chamber

A simple mixing chamber or even T-joint tube-type mixer may provide an alternative for one special application. This is the case when the engine is

operated steadily at one load and one speed, i.e. when the driven machine guarantees a steady power demand. Equally important is the respective calibration of both air and fuel gas supply.

The mixing chamber can be designed in accordance with the criteria stipulated in Chapter 5.2. The control of the power or the point required by the operation is done with one valve each in the air and gas supply and requires experience in finding the required air/fuel ratio. So-called "feeling" is rarely reliable enough to assure operation at the required air/fuel ratio. Another possibility is the provision of a butterfly valve for the mixture, a hand-operated valve for the fuel gas and an uncontrolled air inlet from the air filter, in other words a mixer similar to the venturi type but without the venturi nozzle ring.

It should be borne in mind that even if the simple mixer is properly calibrated or set at one specific point of operation, a change in power demand from the driven machine will change the speed of the engine, hence the volume intake, and cause a disproportion in the air/fuel ratio, unlike in a venturi or gas mixing valve. Small variations may be acceptable as long as the driven machine tolerates speed fluctuations. In case of larger power demand fluctuations the control has to be readjusted in due course by operating personnel as the engine can be damaged by running on an improper mixture or at overspeed.

Only few applications may allow the use of simple mixing devices under the mentioned limitations. These are

- **an electric generator with a reliably controlled constant power output and a network with a corresponding demand, and**

- a centrifugal pump delivering a constant flow rate of water against a constant head.

6.4 Change of Compression Ratio

Standard petrol engines operate at compression ratios of ε = 7 ...9 so that self-ignition of an air/fuel mixture is impossible. The efficiency and power output can in principle be improved by an increase of the compression ratio to E = 11 . . . 12 for operation on gas. An increase from e = 7 to e = 10 will for instance result in a power increase of about 10%. One must, however, bear in mind that these engines have been designed for their original compression ratios with respect to the allowable load on the crankshaft bearings, etc. An increase of the compression ratio is furthermore an irreversible modification which does not allow operation of the engine with petrol any longer. Compression increase can be achived by machining off an appropriate portion of the cylinder head sealing surface. (For determination of the new compression volume refer to Chapter 5.5.3.) In some cases, however, the valves are very close to the piston and may touch the piston at TDC in the valve overlapping phase when the cylinder head is machined off.

With regard to the reasons given above the increase of compression ratio needs careful consideration and should rather be avoided with respect to engine life especially when the engine is earmarked for continuous operation. Otto (vehicle) engines are usually built for life spans of about 4 000 hours as opposed to diesel engines with life spans of 10 000 . . . 20 000 hours. The unavoidable power reduction in biogas operation should therefore be welcomed as a means to reduce wear and tear and increase the engine's life span.

6.5 Manufacture and Installation

6.5.1 Venturi Mixer

A venturi mixer in its details is given in Appendix IV. The body can be manufactured from a standard steel tube but should be somehow finished inside to obtain a smooth surface. The connecting flanges are made in accordance with the flange size of the engine's inlet manifold and air filter respectively. The venturi ring requires careful machining on a lathe machine and an extremely smooth surface. The ring groove around the circumference which forms the fuel gas channel to supply the gas inlet jets should have a free area of at least 1.5 times the total area of the jet bores to provide a slow flow with only little resistance.

The bore holes are to be evenly distributed around the circumference, the number of holes being chosen in such a way that the individual bore has a diameter of between 2 mm for smaller engines and 4 mm for larger engines respectively. The previously calculated fuel gas inlet area is divided by the number of holes to obtain the area of the individual bore A_b, its diameter d_b found by

$$d_b = \sqrt{\frac{4}{\pi} \cdot A_b} = \sqrt{\frac{4}{\pi} \cdot \frac{A_g}{\text{number of bores}}}$$ **(Equ. 6.5)**

The outer diameter of the venturi ring is to be machined to precisely match with the inner diameter of the mixer body to avoid uncontrolled air bypass. An extra O-ring in a groove will suffice to tighten the venturi ring against the tube body. The venturi ring is held in position by a setscrew fitting into a small hole in the center of the circumferential fuel gas supply channel. The setscrew should not block any

of the fuel gas bores and be positioned opposite the fuel gas supply pipe connection.

The fuel gas supply pipe from the plant should have a diameter large enough to keep the flow velocities lower than 2 m/s. In the normal case the use of a standard tube diameter is recommended, i.e. 3/8", 1/2", 3/4", etc., as the calibration valve can then be chosen from standard series also. The pipe can be brazed or welded into an appropriately sized hole drilled into the mixer body.

The choice of the calibration valve depends on the availability of technical equipment. Standard water valves made of brass may after some time show corrosion due to the H_2S traces in the biogas but may be used where there is no alternative. Ball valves with stainless chicks are specifically recommended, also because they open and close with a 90° movement of their lever only and the optimum position can later be fixed with a stop screw easily.

The butterfly valve needs to be carefully manufactured in such a way that it can totally close the venturi mixer's flow area in the "closed" position. In any position it shall not interfere with the flow through the venturi ring. This means that its downstream distance from the venturi ring end needs to be at least 0.5 times the main channel's (inlet) diameter d_i. Some carburetor manufacturers choose to shape the butterfly valve as an ellipse so that it closes the flow channel at an angle smaller than 90° from the "open" position (see Fig. 6.2). This shape, however, is more difficult to obtain in a self-made version.

The two bearings holding the butterfly shaft require some precision in manufacturing. They need to

- allow free and easy movement of the shaft, especially when the butterfly valve is to be connected to an automatic control system,

- be airtight to prevent uncontrolled air to be sucked in and thus unbalance the calibration of the air/fuel ratio.

If the butterfly valve is operated manually and rarely only, rubber seals as shown in the detailed drawing can be used. For frequent and fine movement like in automatic control a brass or bronze bush on either side is more appropriate: Standard carburetors provide good examples also.

6.5.2 Use of Petrol Carburetors or Components

There are some reasons to furtherly utilize the original petrol carburetors in the process of air/fuel mixing:

- If the engine is to be operated on its original fuel in case of gas shortage the original carburetor can be retained completely and the gas mixer is mounted onto the carburetor's air inlet. In case of operation on fuel gas, petrol is no longer fed to the carburetor while fuel gas is fed to the mixer. A further advantage is that the butterfly valve of the carburetor is still used and the (venturi) mixer does not need its own butterfly valve In case of biogas shortage the gas supply is closed and the petrol supply opened. The ignition timing needs however to be readjusted whenever the type of fuel is changed (about 10° ... 15° earlier for biogas operation). Operation on the two fuels at one time is impossible as each individual mixer i.e. carburetor and gas mixer, is calibrated for single fuel operation only. The air/fuel mixture would become too rich.

- If petrol fuel shall not be used any longer the carburetor can still be retained to make use of the butterfly valve. In order to reduce the flow resistance by the carburetor its original venturi ring may even be removed together with the petrol inlet nozzle.

- Another alternative is the modification of the carburetor itself to act as a venturi gas mixer. This can be achieved by replacing the original carburetor venturi by a new one for biogas which has been designed and dimensioned according to the procedure in the previous chapter. A hole will have to be drilled into the carburetor body at a suitable place to insert the biogas supply pipe in such a way that it meets the internal ring channel of the venturi ring. The biogas supply pipe or a short tube for connection to a flexible hose pipe will have to be threaded if the carburetor body's wall thickness allows for screwing in. Otherwise a two-component epoxy resin glue can serve the purpose unless aluminum welding facilities are available.

- The original carburetor can also be modified by simply removing the petrol inlet nozzles and drilling one hole from the outside through the body and the original venturi. The hole will have to meet the venturi at its bottleneck and be big enough to allow the required biogas to join the airstream. The calculations need to consider the actual size of the given venturi (measure!) and the fact that the fuel gas is supplied by one inlet only.

The last alternative may be easier to manufacture but may also show inferior mixing qualities in cases where the distance to the manifold is short. The fuel gas emerging from one inlet (off-canter) may not have mixed sufficiently well with air before the mixture is distributed to the different cylinders. Individual cylinders

may thus receive different mixtures which is unfavorable for uniform running.

The installation of the mixing chamber whether on an existing carburetor or air filter or in the place of the previous carburetor follows the same guidelines as given in Chapter 5.

6.6 Control

6.6.1 Manual Mode

The only control device of an Otto engine is the butterfly valve which varies the amount of combustible air/fuel mixture admitted to the cylinder. The speed or power output of the engine can therefore only be controlled by opening and closing the butterfly valve. If the valve is kept in one position and the load drawn from the engine drops, the engine will increase speed until speed and load have found a new balance. If the load is too low to find a new balance the engine overspeeds and can finally destroy itself. In case of load increase the speed of the engine decreases. If the load drawn from the engine does not decrease also the engine can finally come to a standstill. In case of a new balance of load and speed the engine continues operation at lower speed which may be hazardous at speeds below about 1300 1/min when operating at high load for longer periods.

Manual operation therefore requires the presence of an experienced operator to take care of load fluctuations and operate the butterfly valve accordingly. Unlike diesel engines Otto engines have no overspeed. safety device or governor in most cases. Some however are equipped with a simple centrifugal mechanism within their distributor rotor which cuts out the ignition at any speed above the

maximum. The engine is not shut off completely but continues with speed fluctuating around the maximum in an "on and off" mode which should not be tolerated for more than a few minutes.

6.6.2 Automatic Mode

The principal methods for automatic operation and control have already been dealt with in greater detail in Chapter 5 on diesel gas engines.

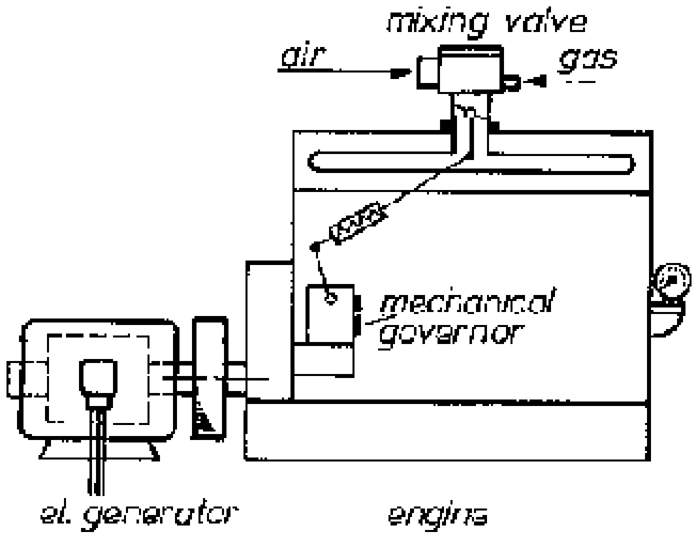

Fig. 6.8: Mechanical speed governor directly acting on the butterfly throttle in the mixing valve.

Some Otto engines originally designed for stationary purposes may have a centrifugal governor type of control mechanism which can of course be used. The motion of the governor rack or lever will have to operate the butterfly valve of the new mixer, whether venturi or mixing valve, or it simply continues to operate the former carburetor's butterfly if the latter is still used in a modified form. Electronic

systems which sense the engine speed and operate the butterfly valve with a positioner may be easier to install at an engine that has no shaft connection for a mechanical governor.

Electronic control systems are, however, sensitive to rough climate and handling and need suitable expertise for maintenance and repair. A separate mechanical governor, even if driven by a V-belt from the crankshaft pulley, may appear to be more appropriate in cases where a little deviation (+ 3%) from the set speed is permissible.

An additional and separately connected overspeed device is always recommendable whether for manual or for automatic operation. The device, e.g. similar to an engine speed tachometer, interrupts the ignition circuit or energizes a solenoid valve in the gas supply line to make sure that the engine never runs at a speed higher than allowable. The engine should only be restarted by an operator who has carefully checked the reason for overspeeding, rectified the fault, and manually reset the system. 7. Planning a Biogas Engine System