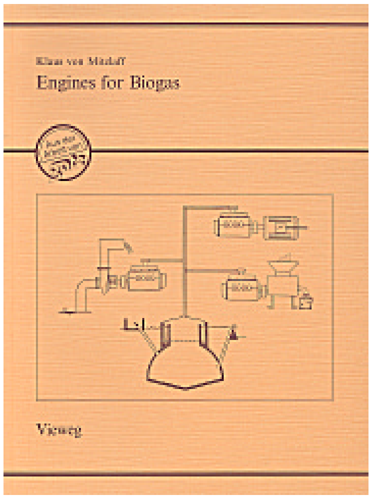

Engines for biogas

Klaus von Mitzlaff

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Deutsches Zentrum für Entwicklungstechnologien - GATE in: Deutsche
Gesellschaft für Technische Zusammenarbeit (GTZ) GmbH
P. O. Box 5180
D-6236 Eschborn
Federal Republic of Germany
Telephon: (06196) 79-0 Telex: 41523-0 gtz d
Fax: (0 61 96) 79 48 20

The Author: Klaus von Mitzlaff, born 1949 in Göttingen, West-Germany, studied mechanical engineering and graduated in 1975.
He was working as a lecturer in mechanical engineering at the University of Dar es Salaam, Tanzania until 1985. Now he is working as an advisor and coordinator for renewable energy and energy saving projects within a cooperation programme between Tanzania and the Federal Republic of Germany.

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Preface

The world's energy situation, whether in developing or industrialized countries, is an issue frequently discussed under economic, technical and political aspects. While it has meanwhile become common knowledge that today's main resources of energy such as coal, crude oil, natural gas and even nuclear energy will become scarce within the next generation the renewable sources such as hydro-, wind- solar- and bioenergy are gaining more and more importance in terms of research and development as well as implemented systems. A common feature of renewable energies is that they are mainly available through a decentralized, sometimes even individual approach. This generates a chance of having energy at one's own disposal but creates a problem of management and network when large energy quantities are required.

Some developing countries find themselves under considerable energy constraints. While the growing demand for household energy decreases the fuelwood reserves and increases desertification, their foreign exchange earnings do not allow for sufficient importation of energy. Their potential for other renewable energies may be large but is not sufficiently exploited for reasons like lack of capital and expertise. Industrialized countries, though still in a position to import energy, are feeling the burden of ever-increasing energy cost while their renewable energy potential is not tapped for, amongst others, political reasons.

The issue "biogas" tends to initiate adverse reactions, ranging from blind enthusiasm and belief via critical openmindedness or sympathy to total rejection. Critical sympathy appears to be a good precondition for coming to terms with biogas issues and for a successful development of biogas-related projects.

The biogas technology has been steadily developed within the last fifty years from small individually designed units to industrial plants with sophisticated boundary technology. The development, however, has largely taken place on the side of biogas production and anaerobic waste treatment. The utilization of the gas has only recently been given more attention as larger and more sophisticated biogas systems require or depend on a sensible utilization of the larger gas quantities. Transforming the energy from biogas into the thermodynamically higher valued mechanical energy marks one of the sensible options wherever appropriate.

The aim of this publication is to build a bridge between the elaborate literature and information on the biogas production side and the existing technical and scientific know-how on the side of internal combustion engines. An engine fuelled by biogas shall become understandable as a core module in a system of energy supply, energy transformation and a demand of energy for a useful purpose. This publication attempts to provide a source of essential information for decision-making, planning, modification and operation of biogas engines within this system.

The author hopes to contribute to the better understanding and the further development of the utilization of biogas for motive power. As this book is written while a large number of experts are working on and further developing similar issues in the field as well as in the laboratories, the author wishes to encourage the readers of this book to come forward with discussions, criticism and suggestions for further improvement on its contents and the form of presentation.

I wish to express my sincere gratitude to the GTZ and GATE for graciously helping to produce this publication. The cooperation with the corresponding department, especially with Dr. P. Pluschke, Mr. M. Homola and Ms. H. Mende, was agreeable, stimulating end marked by mutual understanding. Likewise the author is indebted to suppliers and manufacturers of engines, biogas ancillaries and modification equipment who provided data and specifications of their products as well as much useful discussion. Many thanks go to Ms. K. Pfeiffer who drew most of the figures and diagrams. For the tedious job of processing a partly difficult to handle manuscript and for useful assistance in editing Mr. B. v. Mitzlaff deserves the author's special thanks.

Göttingen, September 1988 Klaus v. Mitzlaff

Content

1. Scope of this publication

It is the aim of this book to provide a source for the basic understanding, the planning and the execution of issues and ventures in relation to biogas engines. The scope therefore needs to comprise a range of information from the theory of internal combustion engines to the actual way. of modification and to a guide on the parameters influencing a useful and economic operation of biogas engines.

The readers of this book are likely to come from various fields with a non-uniform background of specific experience and knowledge of the matter. On the basis of experience gained in a number of biogas programs and activities within the last ten years the publishers (GATE) and the author came to the understanding that a certain minimum of technical knowledge on the reader's part shall be taken for granted. The very basics, e.g. of the way an internal combustion engine functions or of workshop technology, are therefore not elaborately explained. The book is mainly addressed to readers with some technical background and those who are eager to further-embark on biogas engine matters. Some will find it useful as a handbook and reminder while others may realize that there are more parameters to be considered for a successful implementation than just buying an engine.

Parts of the contents, especially the chapters on the essential theory of internal combustion engines and on the operation of an engine together with a driven machine, are naturally not only biogas-engine-specific. It was however felt that in many cases people are only coming into contact with engine and machine operation issues in connection with a possible use of biogas for mechanical/ electric power. Sufficient knowledge or expertise on engines and their operation can therefore not always be assumed.

Furthermore there is an additional quality in using an engine fuelled by biogas. Here the whole fuel generation and supply side becomes an integral part of the system. There is a direct interdependence between the management and operation of the biogas plant, its size and gas storage facilities and the size and operations of the engine cum driven machine. It was therefore considered essential to elaborate on the system character of an issue comprising the generation of fuel energy, its transformation into mechanical energy and the consumption of the energy in a useful and economic way. Biogasfuelled engines easily turn out to be less practical and economic than other alternatives or solutions if the system aspect is undervalued.

Two chapters, one on the utilization of the engine's waste heat and one on the use of biogas in vehicle engines, have been added. The use of waste heat plays an essential part in making an energy system economic which utilizes only about 30 % of the fuel energy but has the potential of exploiting a total of about 80 % of the biogas energy if a useful purpose for the heat energy can be found.

Utilization of biogas in a processed form, i.e. almost pure methane CH_4, is becoming more and more important in vehicle applications. While an effort in plant investment and process energy is necessary, a specific fuel situation may well provide economic incentives to use biogas in tractos, lorries and smaller vehicles. Institutions in Brazil are presently running elaborate research and development programs on this issue.

The type and size of engine considered in this book were limited by two factors. One is the conception that the modification of the engines should be possible with "local" means and expertise, i.e. without sophisticated laboratory-type methods. The other factor is that the basic engines used for modification should be standard engine types from larger series for reasons of availability and the. access to spares and service. The idea of self-handling of engines and modification also limits the engine's size. From the experience of a larger number of biogas projects a power range of about 50 kW was found to be a good compromise. While the theory is valid for the larger engines also, they often incorporate more sophisticated technology such as turbocharging.

This publication cannot and does not attempt to meet the claim of a recipe book for all possible cases. It rather wants to explain and make understandable the various design, economic and other influential parameters and their function in biogas engine issues.

The given examples therefore provide proposals on how to use the given information in a specific situation in order to arrive at a meaningful solution. Proposing standard solutions or final answers does not appear to be appropriate in dealing with an energy system with too many variables which are situation-specific and not always primarily technical ones. A change of only one variable can easily result in a totally different solution.

Positively speaking there is sufficient room and incentive for the reader's own engineering which he will hopefully enjoy after having worked his way through the following chapters. There is after all a better chance of planning, implementing and running a successful project with a broader understanding of the issues concerned.

2. Review of existing literature

Literature with relevance to the topic of this book comes from different fields. One naturally is the standard literature on internal combustion engines which is elaborate to an extent that it would go far beyond the framework of this book to give a complete list[1]. The first Otto engines at the turn of the century were gas-fuelled engines. They are well covered in the standard literature on engines.

Another field is the literature on biogas, dealing mainly with issues concerning the biofermentation and the various plant designs for different biomaterials, plant sizes, etc. The greater portion of the literature was written within the last ten to fifteen years while the awareness of the role and potential of biogas as an energy gradually increased. In a standard sourcebook for renewable energies for developing countries from 1976 biogas did not yet receive any attention [1][2].

Others, however, quoted biogas but mainly as an alternative energy for household use [2]. From the mid-seventies onwards a large number of papers in conferences and journals signalized the growing importance of biogas, not only for small-scale use in households but as a product of municipal and industrial waste treatment with anaerobic fermentation. To name only a few there is L. Sasse's standard book on biogas plants for rural applications [3], BORDA's Biogas Handbook [4] and more recent publications like Oekotop's "Biogas" on the more practical and implementation issues in developing countries [5] and the GTZ's "Production and Utilization of Biogas in Rural Areas of Industrialized and Developing Countries" [6].

The importance given to biogas in the developing countries themselves is documented in numerous publications and seminar proceedings like "Energy for Development in Eastern and Southern Africa" [7] and many others especially from India and China where the small-scale biogas technology development had gained momentum one generation before it became an international development issue.

With the increase in biogas production towards larger quantities the technical utilization like the transformation into mechanical energy became an issue to be researched on. While larger engines specifically designed for gas were on the market, smaller engines modified from standard Otto or diesel engines were seen to fill the gap for small to medium and decentralized applications. Indian [8] and Chinese [9] publications mainly dealt with the modification of small stationary diesel engines for dual fuel operation. Others went on to modify medium-sized diesel engines including their governors [10], or researched the performance paramters of dual fuel biogas engines in more detail [1 1].

Biogas as a fuel for vehicles has been an issue since the 1950's. While in Europe the use in tractors seems to be the issue [12, 13], in Brazil the aim is to substitute petrol and diesel fuel in the automotive sector using purified and compressed biogas or natural gas [14].

Much useful material and information have been contributed in recent years by publications of manufacturers of gas engines and modified engines or suppliers of equipment and modification kits for standard Otto and diesel engines. Some of their publications are named in the Literature Reference List.

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, cm³) 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)

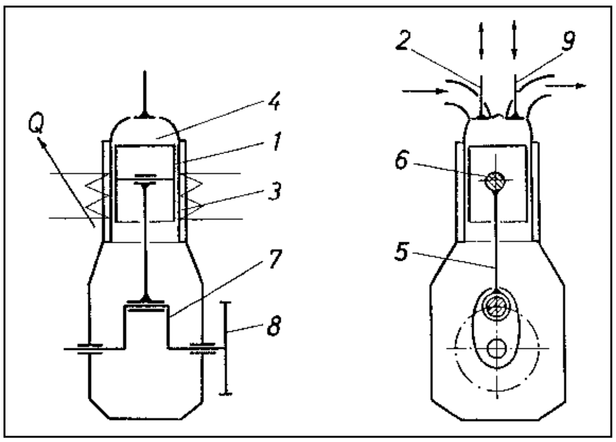


Fig.31:Principal scheme of a 4-stroke engine.1 piston, 2 inlet valve, 3 cylinder, 4 combustion chamber, 5 connection rod, 6 gudgeon pin, 7 crankshaft, 8 flywheel, Q head rejected (cooling).

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)

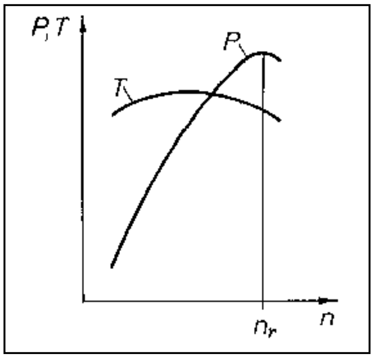


Fig. 3.2: Engine power output P and torque T as a function of engine speed n: nr marks the rated speed.

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,ê

$$e = \frac{V_{tot}}{V_c} = \frac{V_{d,c} + V_c}{V_c} = 1 + \frac{V_{d,c}}{V_c} \quad \text{(Equ. 3.4)}$$

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

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

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
ê = 17, p_s = 0.9 bar, n= 1.3
$p_c = p_s \cdot ê^n$
$p_c = 0.9 \cdot 17^{1.3}$ = 35.8 bar

- For a standard Otto engine with ê = 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_c[1] (without ignition)

$T_c = T_s \cdot ê^{n-1}$ (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.

Example:
- Diesel engine, ê = 21, T_s = 330 K, n = 1.3

$T_c = T_s \cdot e^{n-1}$
$T_c = 330 \text{ K} \cdot 21^{0.3}$ = 823 K (= 550 °C)

- Otto engine, ê = 8.5, T_s = 330 K, n = 1.35

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

3.1.9 Necessary Compression Chamber Volume, V_c

$$V_c = \frac{V_h}{(p_c / p_s)^{1/n} - 1} = \frac{V_h}{(T_c / T_s)^{1/(n-1)} - 1} = \frac{V_h}{e - 1} \quad \text{(Equ 3.8)}$$

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.

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

whereby the fuel energy flow/consumptionis given as

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

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):

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

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

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.

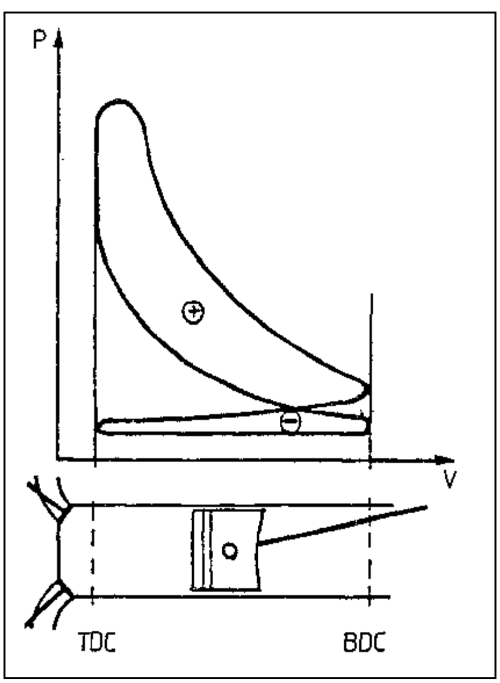


Fig. 3.3: Pressure, volume (p, v) diagram of a 4-stroke engine cycle

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:

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

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" λ:

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

so that

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

The best combustion performance will always occur at values near $\lambda = 1$. Mixtures at values below $\lambda = 0.5$ rich or above $\lambda = 1.5$ 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 (λ= 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 $\lambda = 0.9$ 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.

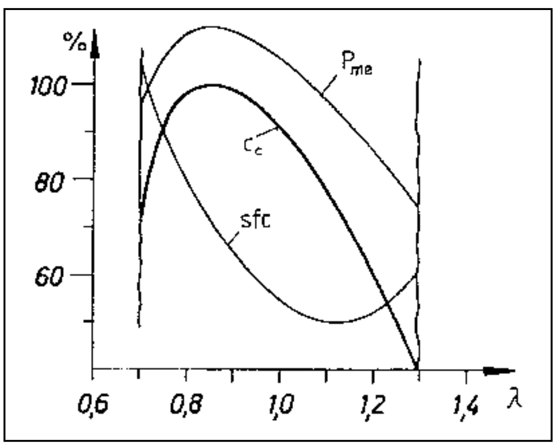


Fig. 3.4: Mean effective pressure Mme, combustion velocity c_c and specific fuel consumption sfc as a function of excess air ratio λ

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 λ (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.

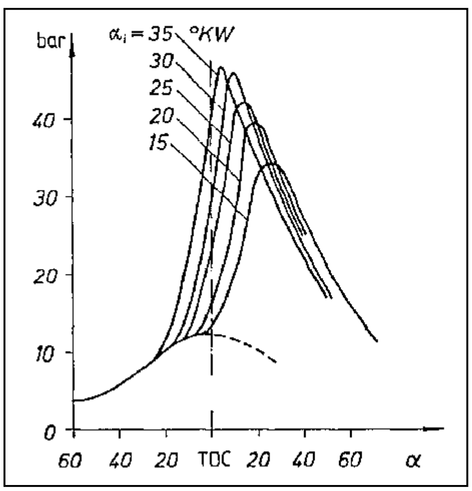


Fig. 3.5: Course of pressure as a function of the crank angle α (point) of ignition α_i

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).

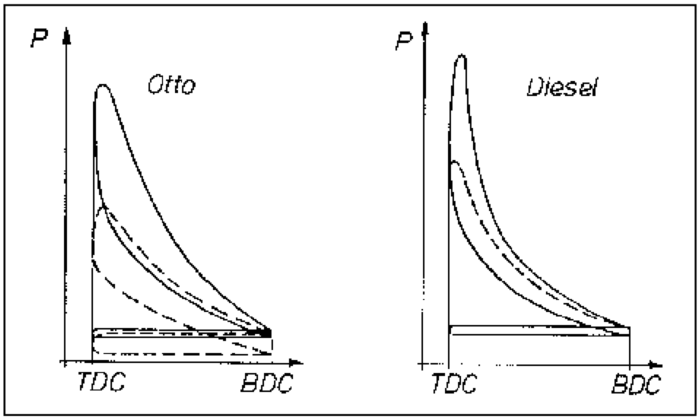


Fig 3.6: Partial load (- - -) and full load (____)p, v-diagrams for Otto and diesel engine

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

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

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.

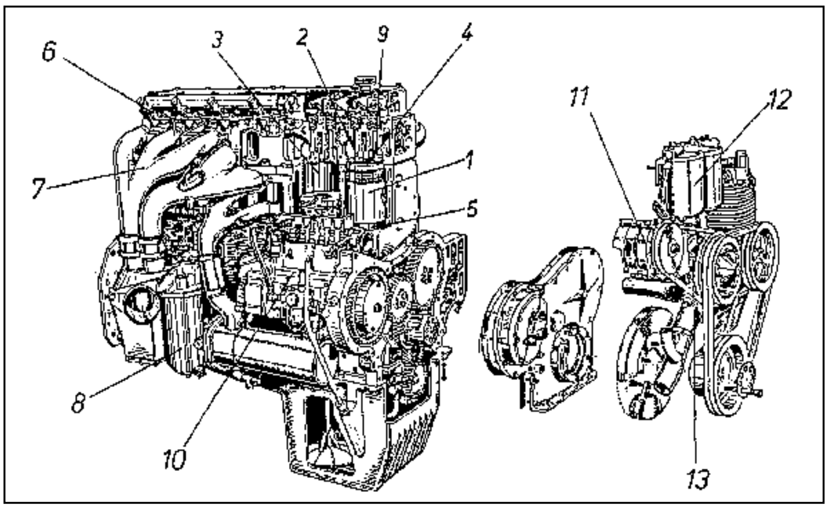


Fig. 3.7: 6-cylinder diesel engine, partly opened (MAN).
1 piston, 2 inlet valve, 3 cylinder, 4 combustion chamber, 5 connection rod, 6 injector nozzle, 7 suction mainfold, 8 oil filter, 9 outlet valve, 10 injector pump, 11 alternator,12 fuel filter, 13 cooling water pump.

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.

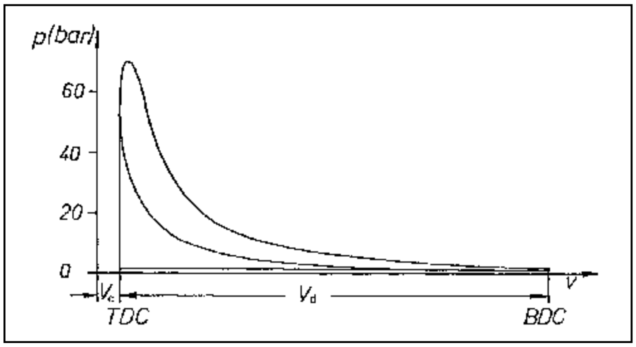


Fig. 3.8: Simplified p, v-diagram of a diesel process

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).

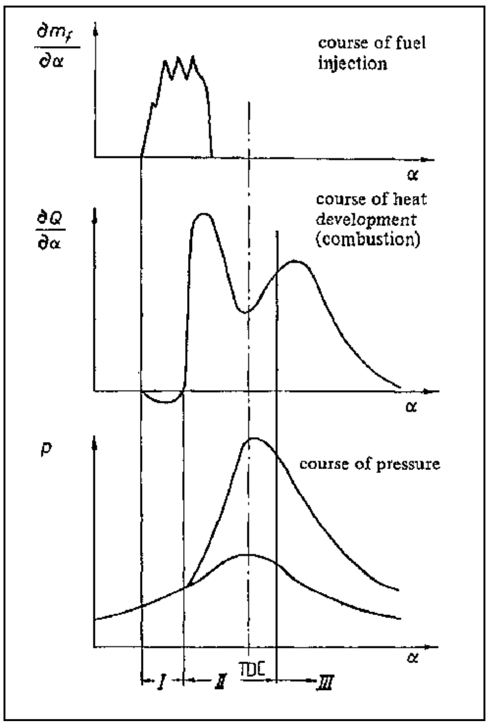


Fig. 3.9: Courses of fuel injection, combustion and pressure as a function of the crank angle α

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.

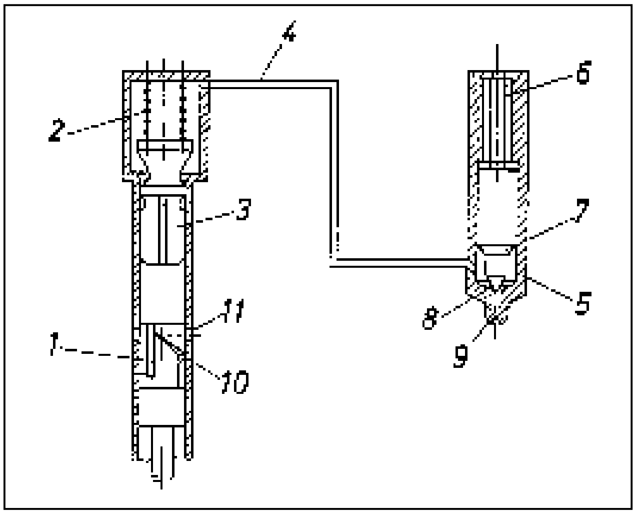


Fig.3.10: Principal scheme of injection system (numberscorespond to text).

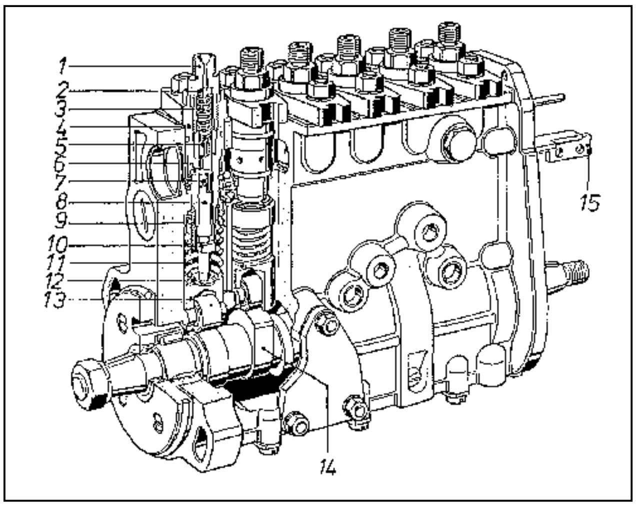


Fig. 3.11: Diesel injector pump (Bosch).
1 valve holder, 2 filling piece, 3 valve spring, 4 pump cylinder, 5 valve, 6 suction and control bore, 7 oblique control edge, 8 plunger, 9 control bush, 10 plunger lug, 11 piston sprig, 12 sprig holder, 13 roller shaft, 14 cam, 15 control rack.

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 $P_{m,e}$ 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. η_{tot} = 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).

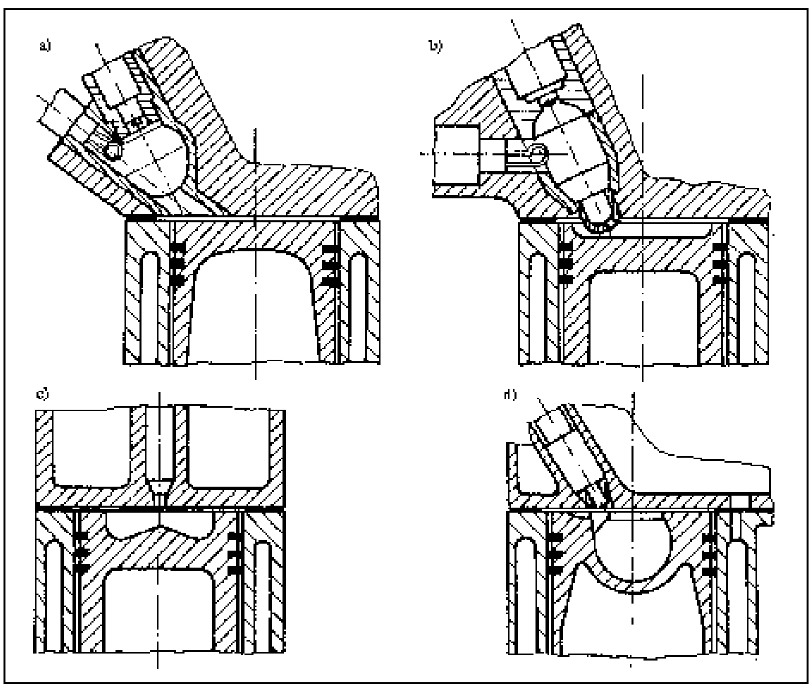


Fig. 3.12: Different combustion chamber forms and methods of fuel injection. a) swirl chamber, b) antechamber, c) direct injection, d) MAN method

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.

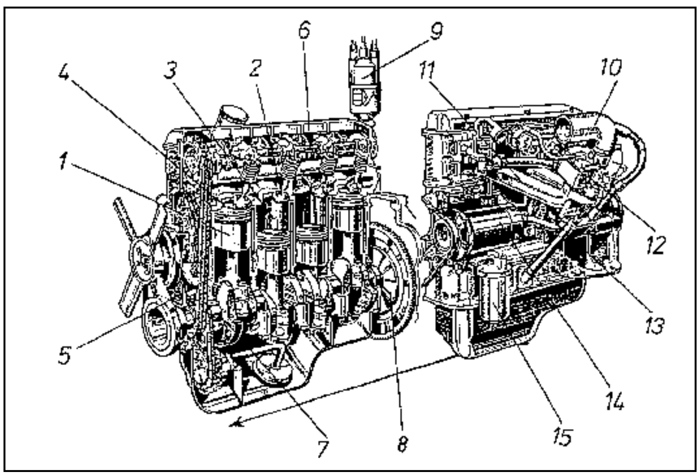


Fig. 3.13: Otto engine, partly opened (BMW).
1 piston, 2 inlet valve, 3 cylinder, 4 combustion chamber, 5 connection, 6 overhead camshaft, 7 crankshaft bearing, 8 flywheel, 9 distributor, 10 suction from air filter, 11 suction mainfold, 12 carburetor, 13 starter motor, 14 generator, 15 oil filter.

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.

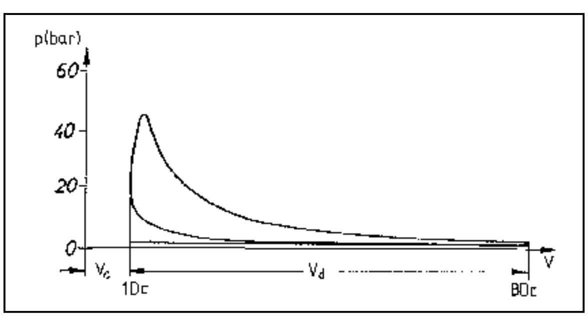


Fig. 3.14: Simplified p, v-digram of an Otto process

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 $\lambda = 1$ are required whereby at
- $\lambda = 0.9$ 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).
- $\lambda = 1.1 \ldots 1.15$ the fuel economy is at a maximum; the CO content in the exhaust gas is almost zero. Nitrogen oxide NO_X increases however (toxic).
- $\lambda = 1.3$ 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: ê = 7 . . . 8.5
- for superpetrol: ê = 8.5 . . . 9.5
- for gas (CH4, LPG): ê = 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 $\eta = 0.25 \ldots 0.32$ 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. liquid 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.

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)

$$\frac{\rho_2}{\rho_1} = \frac{T_1}{T_2} \quad \text{(Equ. 4.4)}$$

- constant pressure process (p = constant)

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

- constant temperature process (T = constant)

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

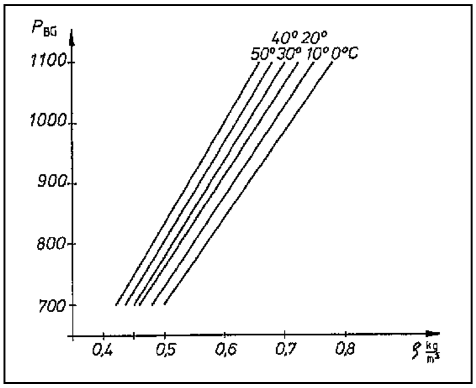


Fig. 4.1: Density ρ of CH_4 as a function of biogas pressure and temperature

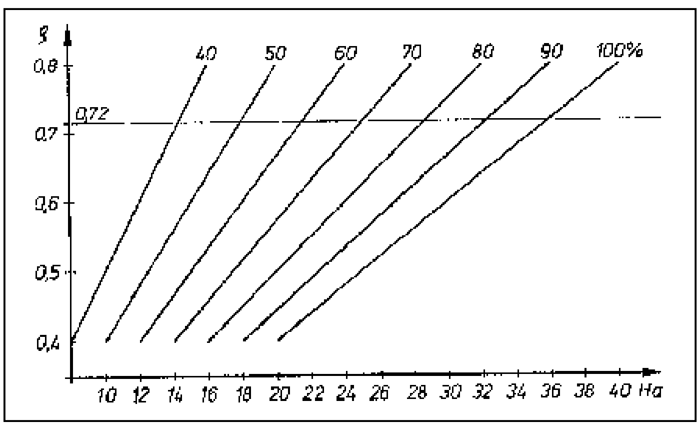


Fig. 4.2: Calorific value of biogas as a function of the density and volume %-age of its CH_4 content (ρ=0.72 is the density at a standard condition)

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
$P_t = 950 + 20 = 970 \text{ mbar} \mathrel{\hat{=}} 0.97 \cdot 10^5 \ P_a$

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)

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

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

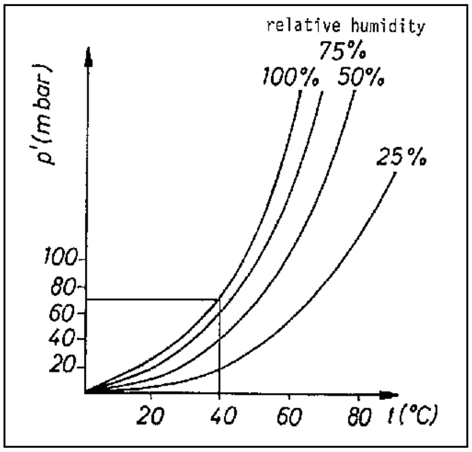


Fig. 4.3: Partial pressure of water vapor in a mixture with biogas as a function of a biogas temperature and relative humidity

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/m^3 n transform this figure by multiplying by 3600 to obtain it in kJ/m^3 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/m^3 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 m^3 n/h with the calorific value of the biogas in kJ/m^3 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/m^3 using the procedure explained in Chapter 4.2.

Step 5:

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

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

Example:
Manufacturer's engine specification:

- power rating P = 20 kW
- fuel consumption at rated power fc = 10 m^3 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

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

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

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

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

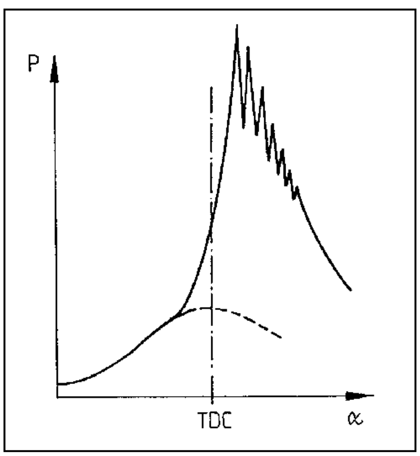


Fig.4.4: "Knocking" in a p, alfa-diagram of an engine

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.

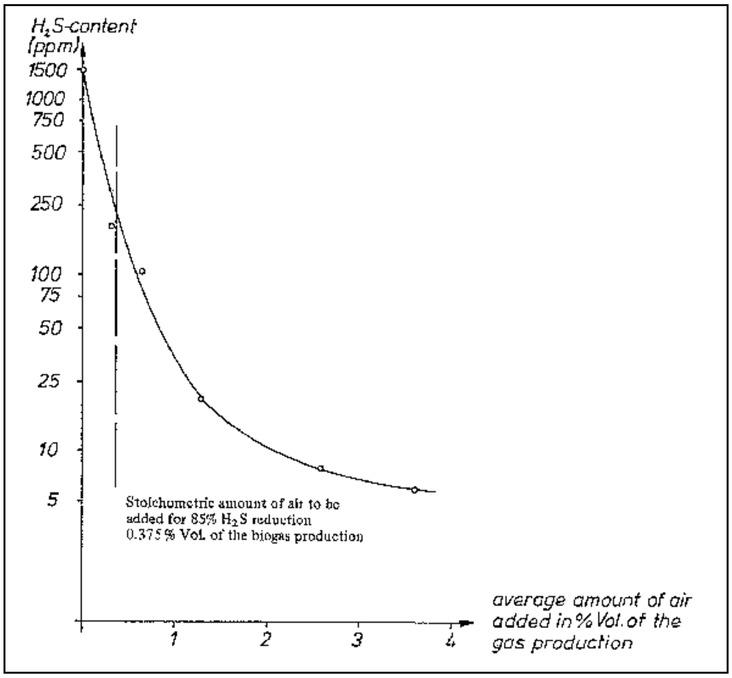
H_2S can be removed by filtering with earth or with iron oxide (e.g. filings) whereby the filters need to be regenerated or the material exchanged periodically [24]. Recent experiments in a large biogas plant in Ferkessedougou, Ivory Coast [25], have revealed that by purging a small amount of air into 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.



H_2S-content (ppm)
1500
1000
750
500
250
100
75
50
25
10
5
Stoichometric amount of air to be added for 85% H_2S reduction 0.375 % Vol. of the biogas production
average amount of air added in % Vol. of the gas production
1
2
3
4

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.

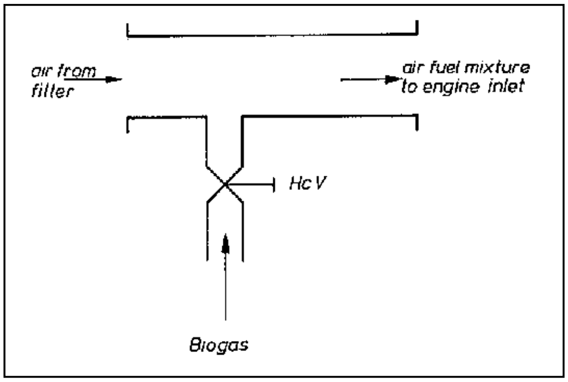


Fig. 5.1: T-joint mixer

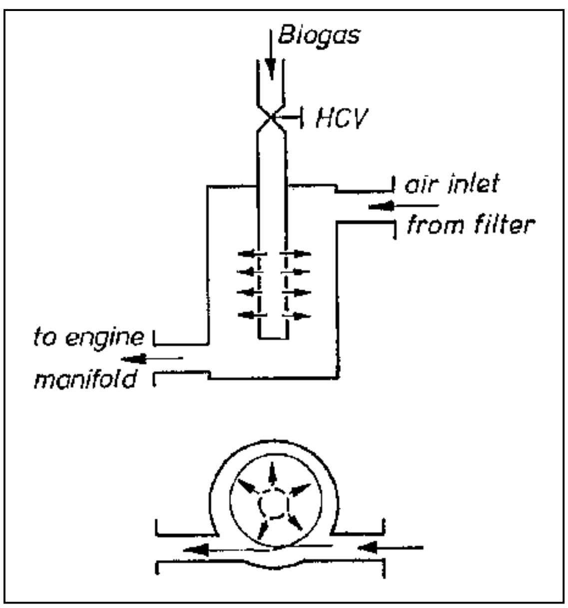


Fig.5.2: Simple mixing chamber with hand - controlled valve (HCV)

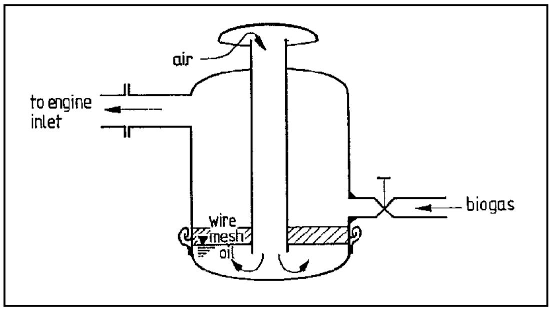


Fig. 5.3: Air filter modified into mixing chamber

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 $\lambda = 1.5$ 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:

$$\text{sfc}_{\text{cal}} = 3.3 \frac{\text{kWh}}{\text{kWh}}$$

- proportion of biogas in total fuel: 80%

Step 1:

Find the total volumetric fuel demand (consumption)

$$\text{fc}_{\text{vol}} = \frac{\text{sfc}_{\text{cal}} \cdot (3600\ \text{s/h}) \cdot \text{P}}{\text{H}_{\text{u,vol}}} = \frac{3.0 \cdot 10\text{kj/s} \cdot 3600\text{s/h}}{20000\text{kj/m}^3}$$

$$\text{fc}_{\text{vol}} = 5.4 \frac{\text{m}^3}{\text{h}} = 0.0015 \frac{\text{m}^3}{\text{s}}$$

Step 2:

Consider proportion of biogas, i.e. 80%

$$\text{fc}_{\text{vol,bg}} = 0.8\text{fc}_{\text{vol}} = 0.0012 \frac{\text{m}^3}{\text{s}} = 4.32 \frac{\text{m}^3}{\text{h}}$$

The volumetric fuel demand in this case is 4.32 m³/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 m³/s, c = flow velocity in m/s, Across-sectional area in m².

From an energy balance for a tube flow² at two different cross-sectional areas (1 and 2) the velocity can be calculated:

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

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

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 $\Delta p = 0.02$ 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: $\eta_{vol} = 0.85$
- 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):

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

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

Step 2:

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

$$Ai = \frac{d^2 \cdot \pi}{4} \quad \text{(Equ. 5.5)}$$

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

Step 3:

Intake velocity, c_i:

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

Step 4:

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

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

Step 5:

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

$$fc_d = 0.8 \cdot fc = 0.8 \cdot 0.0056 = 0.0045 \text{ m}^3/\text{s}$$

Step 6:

Cross-sectional area Ag and diameter dg of nozzle:

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

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

The gas nozzle diameter of $d_g = 17 \text{ mm}$ 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).

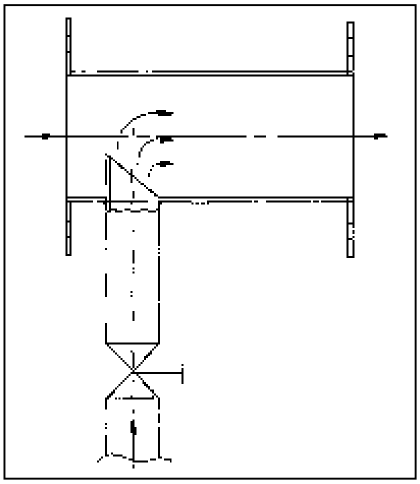


Fig. 5.4: T-joint mixer with oblique, protrunding gas inlet

- 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).

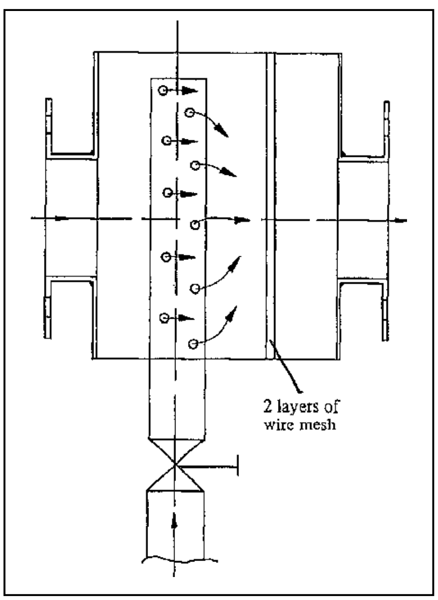


Fig. 5.5: Mixing chamber with gas distribution pipe and wire mesh for intensive mixing

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.

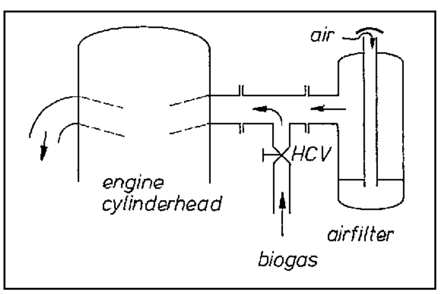


Fig. 5.6: T-joint mixer installed between air filter and engine inlet

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).

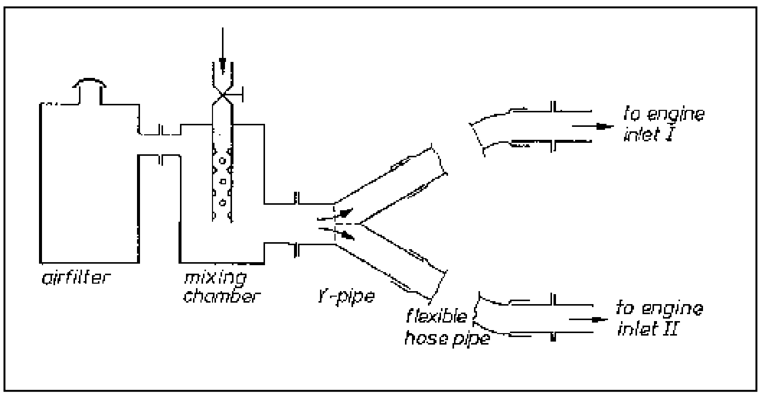


Fig.5.7: Mixing chamber connected to a V-type engine (engine)

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.

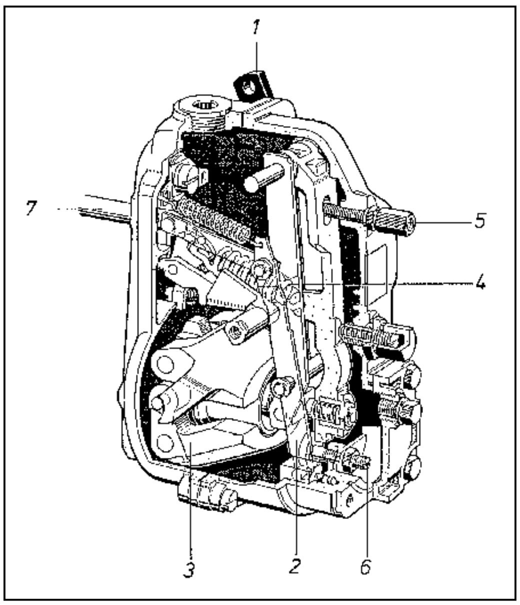


Fig. 5.8: Governor for diesel fuel injector pump (Bosch).
1 control lever, 2 governor lever, 3 centrifugal weights, 4 governor main spring, 5 idling adjustment, 6 full load adjustment, 7 control rack to injector pump

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.

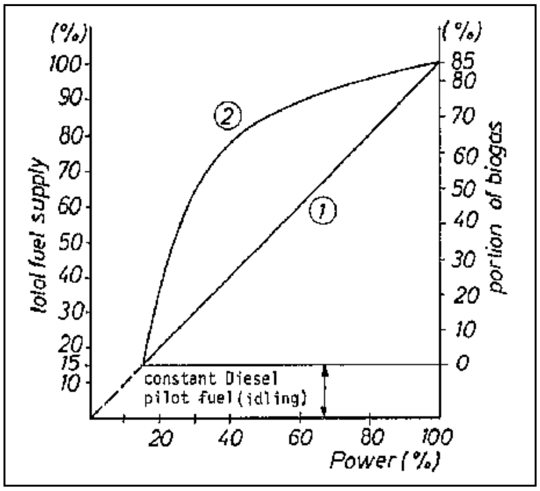


Fig. 5.9: Fuel supply vs. power output Diesel pilot fuel constant, biogas controlled (manualy or automaticaly) according to power demand.
1 total fuel, 2 portion of biogas in total fuel (simplifying assumption: sfc=constant).

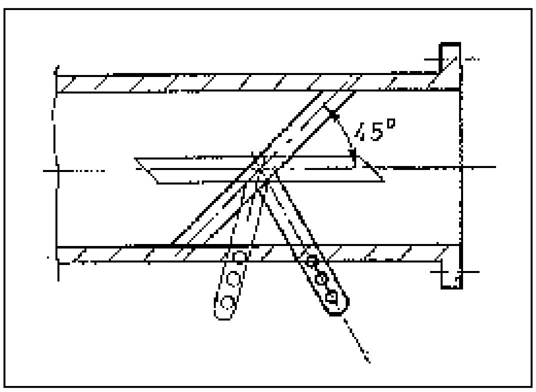


Fig.5.10: Butterfly gas control valve with elliptic butterfly for small angular movement (45º)

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:

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

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.

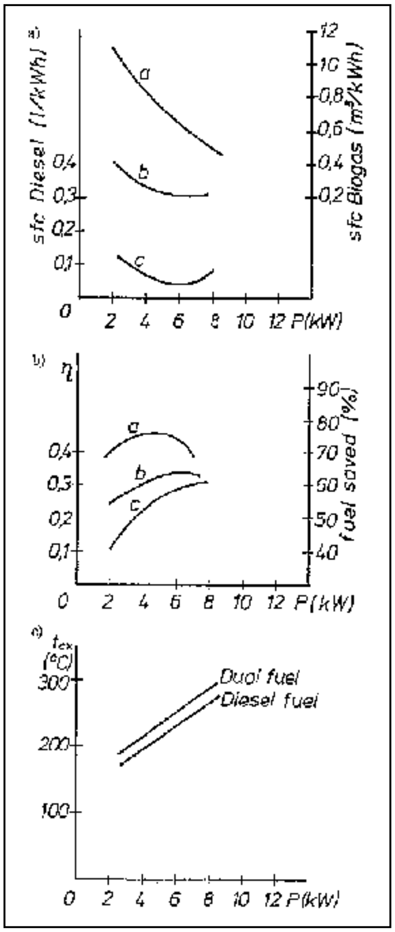


Fig. 5.12: Performance charts of a 10 kW single cylinder gas engine with biogas at n=1500 min^{-1} (from[11]).
a) a sfc biogas in dual fuel mode , b sfc diesel in diesel fuel mode, c sfc diesel in dual fuel mode
b) a diesel fuel saved, efficiency dual fuel mode
c) exaust gas temperature at silencer outlet

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 $\varepsilon = 10 \ldots 12$,
- 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 $\varepsilon = 12$ 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 $\varepsilon = 10.5 \ldots 11.5$ 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 V_c (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

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

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:

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

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

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

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).

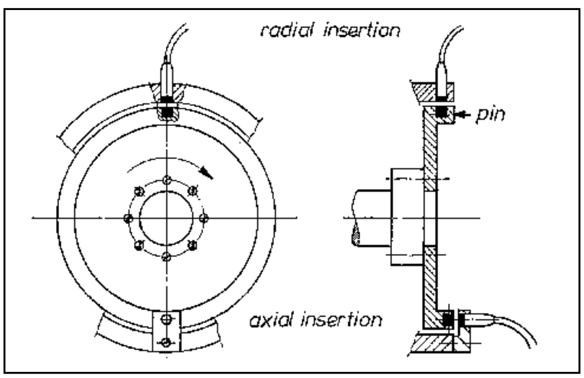


Fig. 5.14: Fixing the pick-up in the casing and the flywheel, two different versions; upper half: radial insertion, lower half: axial insertion.

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.

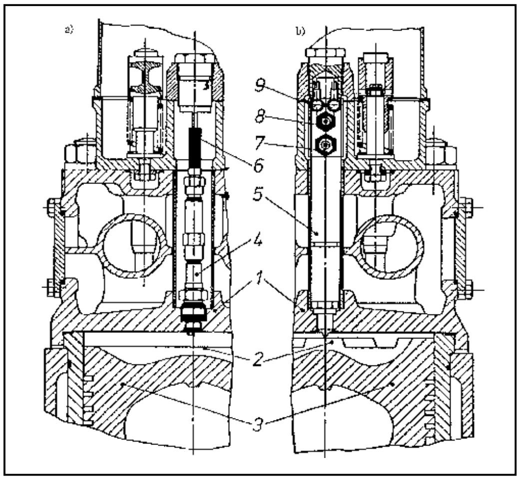


Fig. 5.16: Cylinder head modified with spark plug on increased combustion chamber volume (a) vs. original diesel version with injector (b)
1 cylinder head, 2 combustion chamber, 3 piston, 4 spark plug, 5 injector nozzle, 6 ignition cable conection, 7 fuel supply from main injector-pump, 9 cooling oil 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.

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.

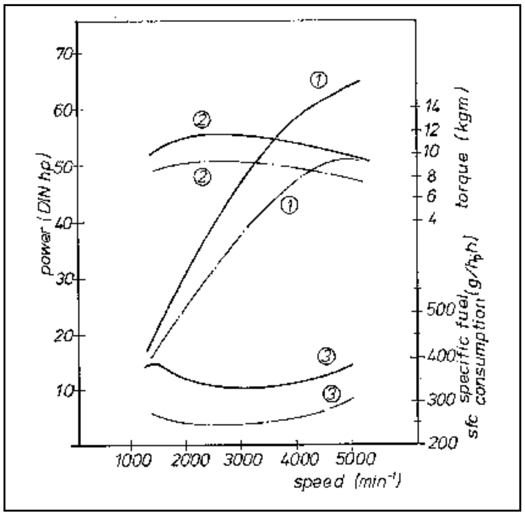


Fig. 6.1: Performance diagram of an Otto engine using liquid fuel (——) and methane(—·—) alternatively (Rodagas).
1 power, 2 torque, 3 specific fuel consumption.

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.

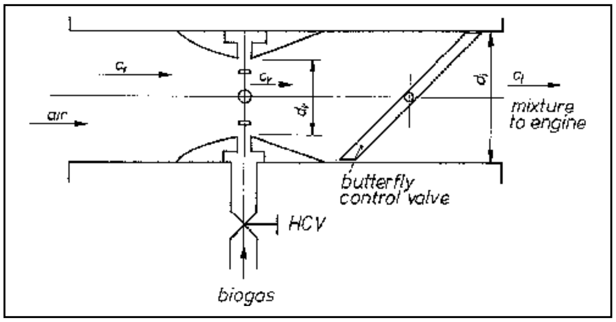


Fig. 6.2: Venturi mixer with gas supply through several bores.
c_1 velocity at mixer inlet, c_v velocity at venturi contraction, d_i diameter of mixer/engine inlet, d_v diameter of venturi contraction, c_i velocity of mixture at engine inlet.

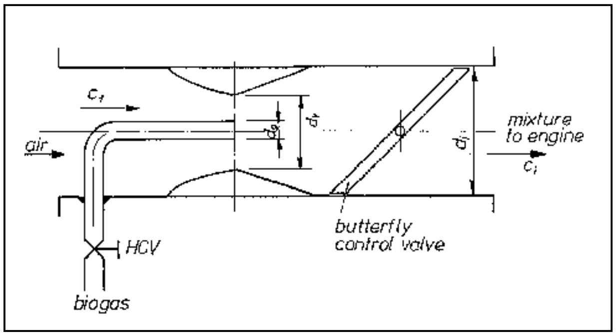


Fig. 6.3: Venturi mixer with a single gas inlet nozzle.
d_g diameter of gas inlet nozzle, other symbols as in Fig. 6.2

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 m³/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

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

Its diameter d_v is found accordingly:

$$\eta = \frac{\text{mech. power} (+\text{heat flow})}{\text{fuel energy consumption}} = \frac{P_m (+P_h)}{E_f} \quad \text{(Equ. 6.2)}$$

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

$= 144 m^3/d \cdot \frac{1}{0.8 m^3/(m^3_{plant} \cdot d)}$ (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

$$\frac{r_2}{r_1} = \frac{v_1}{v_2} = \frac{T_1}{T_2}$$

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.

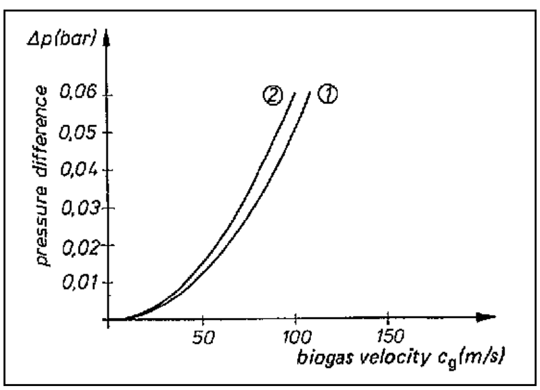


Fig. 6.4: Biogas flow through a nozzle cg as a function of the active pressure difference /\p.
1 gas density p=1.2 kg/m³.

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

$$c_g = \sqrt{\frac{2 \cdot \Delta p}{\rho} + c_{bg}^2} \quad \text{(Equ. 6.4)}$$

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

$$Ai = \frac{d^2 \cdot \pi}{4}$$

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

$$\gamma = \frac{c_p}{c_v}$$

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

$$c_v = \frac{\dot{V}_i}{A_v}$$

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.

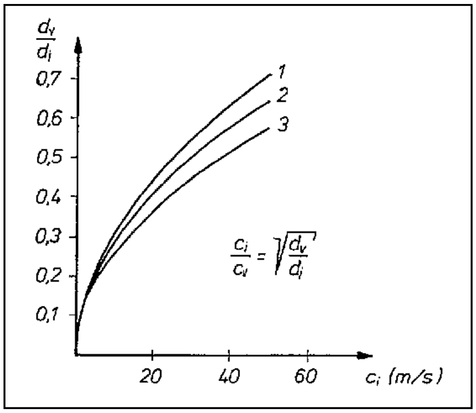


Fig. 6.5: Venturi diameter ratio d_v/d_i as a function of intake velocity c_i and the required velocity at the venturi bottleneck c_v
1 c_v = 100 m/s, 2 c_v = 120 m/s, 3 c_v = 150 m/s

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.

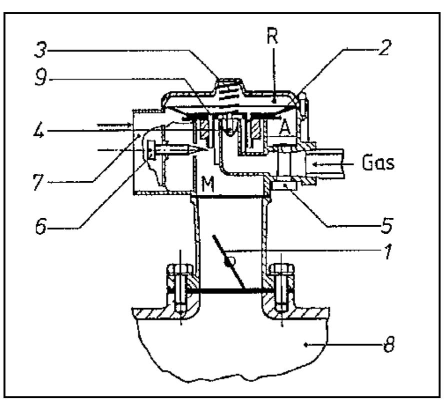


Fig. 6.6: Cross-sectional view of gas mixing valve.
1 butterfly throttle valve, 2 diaphragm, 3 spring, 4 gas valve cone, 5 mixture adjustment valve, 6 air bypass adjustment, 7 air inlet, 8 engine inlet, 9 bore for suction pressure, connects M and R, A space of air inlet before mixing zone, M space of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

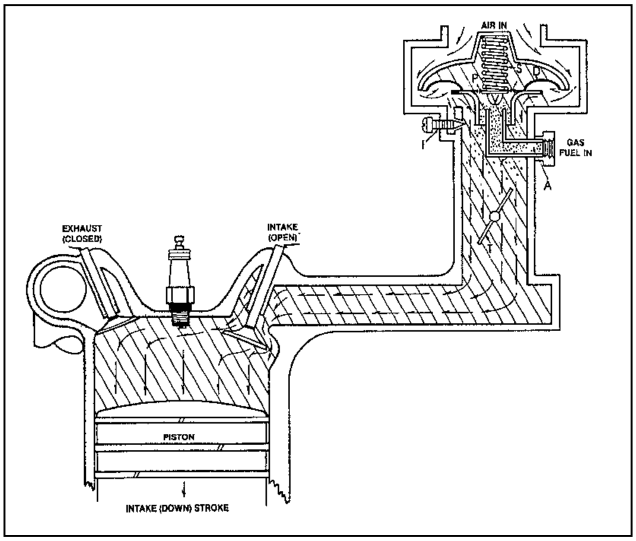


Fig. 6.7: Gas mixing valve in operation, schematic (Impco). S metering spring, D diaphragm, P vacuum transfer passage, V gas metering valve, I idle air bypass adjustment, A power mixture adjustment, T throttle valve.

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. lower 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

$$H_2 = H_1 \cdot (\frac{n_2}{n_1}) = 74(0.0875)^2 = 53.5\text{m} \quad \text{(Equ. 6.5)}$$

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.

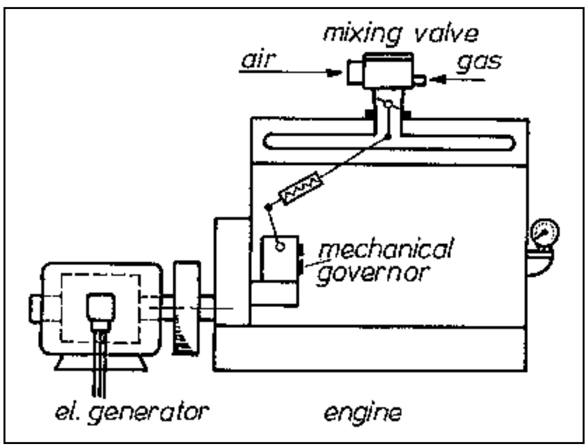


Fig. 6.8: Mechanical speed governor directly acting on the butterfly throttle in the mixing valve.

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

7. Planning a biogas engine system

7.1 The biogas engine as a module integrated into an energy system

The supply of mechanical or electric power from biogas is only feasible using a biogas engine. The installation of a biogas engine however requires an appropriate planning of the fuel production and also the consumption/operation procedures. This is a crucial exercise which can usually be-avoided when the power is purchased from an electric grid.

As an engine in general does not supply energy, but rather transforms one form of energy, here biochemical, into another form, mechanical energy, its operation requires a source of energy on one side and s consumer of the energy on the other. The coordination of the energy source (biogas production plant), the transformer (engine) and the consumer (driven machine) is therefore of utmost importance for a technically and economically satisfactory performance of the whole system. The following parameters have an influence on the system's performance:

a) Technical Parameters

- Biogas production in the biogas plant under consideration of the plant's size, inputs and operation as well as the reliability of the gas supply system.
- Power demand of the driven equipment with regard to its anticipated fluctuation or the anticipated point of continuous operation.
- Demand of low and medium temperature heat from engine's waste heat (cogeneration).
- Daily schedule of operation with regard to biogas consumption, plant size and necessary gas storage capacity.
- Speed or speed range of the driven machine and the engine.
- Mode of control, manual or automatic.
- Local availability of engine service, spare parts, technical expertise and sufficiently competent operating personnel.
- Anticipated development of energy supply and demand in the future.

b) Economic Parameters

- Price of biogas plant cum ancillaries.
- Price of engine cum modification.
- Price of driven machine and energy distribution system (electrical wiring, water system, etc.) unless already existing.
- Operational cost of biogas system, i.e plant, engine and driven machine.
- Cost of the system's service and maintenance.
- Capital costs (interest rates, pay back periods, etc.).
- Expected revenue from provision of selling energy or services, including the use of the engine's waste heat.
- Savings by the omission of cost for other fuels or forms of energy.
- Anticipated development of economic parameters (inflation, laws, regulations, fuel taxes, etc.).

c) Alternative Possibilities of Power Supply

- Electric motors under consideration of availability, reliability and price of electricity from another (e.g. public) supplier.
- Small hydropower in favorable areas for direct drive of machines or generation of electricity.
- Wind power in favorable areas under consideration of the schedule of power demand and the wind regime.

- Diesel, petrol, alcohol or LPG as engine fuels under consideration of availability, price and given infrastructure for a reliable supply.

To summarize, a biogas engine is only one module in a system and can only perform to satisfaction when all other components are well integrated. Furthermore the economic and boundary conditions, realistically assessed, have to be more favorable than for alternative solutions. Last but not least the actual situation sur place, the availability of technical equipment and expertise or other constraints can significantly influence the choice of the system and the planning process as a whole.

7.2 Economic and Operational Considerations

There are different basic situations out of which the use of biogas for the generation of mechanical or electric energy may be considered.

a) Biogas availability or potential

- A biogas plant already exists and the gas yield is larger than what is already consumed in other equipment or the yield could be increased.
- Organic matter is available and otherwise wasted; the boundary conditions allow for anaerobic digestion.
- Environmental laws enforce anaerobic treatment of organic waste from municipalities, food industries, distilleries, etc.

b) Demand for mechanical power

- Other fuels are practically not available.
- Other sources of energy or fuels are more expensive or their
- supply is unreliable.
- Having a fuel at one's own disposal is of specific advantage.

c) Possible revenue through selling mechanical power, electric power or related services to other customers (e.g. the public electricity supply company).

In all cases it is essential to combine the modes of the generation of the fuel and its consumption. While the biogas is produced in a continuous mode, the demand for power, hence fuel, is often discontinuous. Biogas, unlike liquid fuels, can be stored in larger quantities either in a compressed form requiring special efforts or in large, low pressure storage tanks. However, both ways are costly. This provides an incentive to avoid extensive storage through a well balanced production and consumption of biogas.

One way of equalizing the demand profile (Fig. 7.1) is the continuous operation of the engine, hence continuous fuel consumption. Instead of operating a powerful machine and engine for a short period per day the same service can often be obtained by a smaller system operating for a longer period. A similar effect is reached by the operation of different equipment in a sequence rather than at one time, e.g. water is pumped overnight while grains are milled during the day. The smaller system not only requires lower investment itself, but it also requires smaller or no gas storage capacities. The planning of the operational schedule of the equipment has a considerable effect on the economics and feasibility of biogas engine projects.

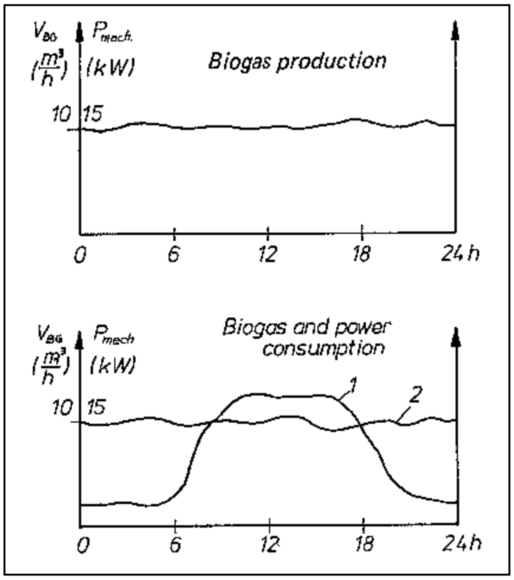


Fig. 7.1: Fuel and power production vs. consumption/demand profile. 1 typical example for operation of machines during the day and little lighting at night, 2 demand balanced and adapted to biogas production.

In cases where biogas is used for electricity generation, the mode of operation, i.e. in an isolated grid or in parallel to an existing larger grid (e.g. public utility), further influences the power demand situation and the choice of the gen-set's power class.
The principal different solutions are discussed further below.

7.2.1 The Specific Situation of Electricity Generation in Grid Parallel Operation

Above all, the economic viability of supplying electricity or mechanical energy to a place which has access to electricity needs to be thoroughly assessed. The mere demand for mechanical power could easily be satisfied by an electric motor which is usually less than half as expensive as an engine and needs far less efforts regarding operation, service and maintenance. The economic justification of the investment for the installation of a biogas-driven gen-set in this situation can only be based on high costs for the purchased electric energy or from severe operational problems through poor reliability of the public electricity supply.
A high degree of utilization of the biogas energy, i.e. power (approx. 30 %) and waste heat (approx. 5070), is often required to achieve the necessary return. Other justifications than economic ones tend to lose actuality, especially when the simple return to another supply system can make life easier.

The aspect of convenience of receiving power from a grid instead of operating, servicing and maintaining a gas engine cum biogas plant should not be underestimated. Even with smaller problems in the biogas engine systems it appears to be a quick and easy solution to revert to drawing power from the grid instead of trying to tackle the system’s problems. The operation of a biogas engine always requires more competent and committed personnel who could be dispensed with when power is purchased from outside. The availability of competent manpower can be crucial for the success of a biogas engine project.

Needless to say, the reliability of one's own biogas engine system is vital, especially when the agreement with the utility stipulates penalty-like conditions for drawing electricity from the public grid.

Operation of a gen-set in grid parallel operation requires specific technical equipment, such as a synchronization unit, safety switch gear for power failures from either side and a sensitive speed control system to secure operation at the grid-synchronous frequency (speed). The extra equipment involves corresponding investment. The connection of a gen-set to an outside grid can only be done in cooperation with the owner or administration of this grid.

While the technical problems can thus be solved, the operation turns out to be more sensitive. The conditions for receiving electricity from the grid are usually different from the ones for supplying electricity to the grid. Public utilities sometimes pay a low price for electricity they buy from small producers while they charge a high price when the same client needs to draw electricity from-the public grid.

As long as the customer's own electricity production remains lower than his demand, he remains a net consumer, substituting his demand as far as the biogas production and the power class of the gen-set allow. The price for the remaining electricity still purchased from the grid may well be the standard consumer price. If the utility does not agree to grid parallel operation, one can decide to make some of one's own power consumers detachable with a changeover switch and satisfy their power demand directly from the biogas gen-set in a separate isolated grid. In case of problems with the biogas gen-set this "sub-grid" can be switched back to the main grid. The economiy of this operation is based on the reduction of power costs by one's own substitution system.

Wherever one's own power production is constantly higher than one's own demand, the economy of the system is based on saving the previous power cost for one's own consumption together with the revenue from the power supplied to the grid. As a net supplier, however, one sometimes has to face specifically high power purchase prices in case one's own system is out of service. Some agreements with public utilities therefore include a certain allowable amount of purchase from the grid per month or year to cover service periods and unforeseen failures. For any purchase above the stipulated amount a penalty price may be charged by the utility.

Similar considerations count in cases where the daily biogas production and the power demand are equal. While during low demand periods power is supplied to the public grid, in peak demand periods power is purchased from it. If favorable conditions can be negotiated with the public utility, the biogas gen-set can be designed for continuous operation in accordance with the continuous biogas production rate.

7.2.2 Biogas Production Exceeds Demand for Mechanical/Electric Energy

(see Fig. 7.2)

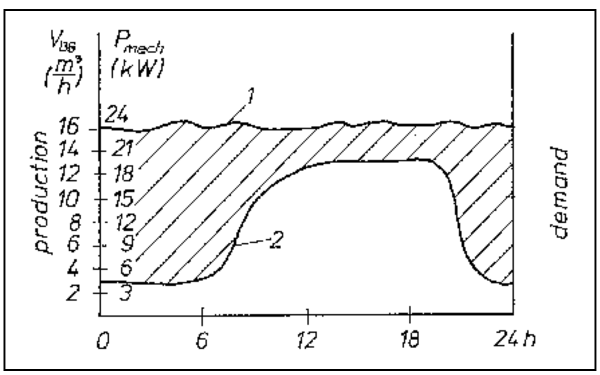


Fig. 7.2: Daily profiles: 1 biogas/potential power production' 2 own power demand. Surplus of biogas for a) other direct utilization or b) extra power production and supply to outside parallel grid. //// excess biogas production.

7.2.2.1 Isolated Operation

Other potential energy users should be sought or further developed such as heating, cooking, lighting, baking, roasting, drying, etc. Their operational schedule needs to consider the engine's schedule aiming at a balanced biogas demand profile, thereby matching the production profile as far as possible. The choice of the engine's power class will be dependent on the power required by the driven equipment with the aspect of using smaller, less power-consuming equipment and engine but extending operation time.

7.2.2.2 Grid Parallel Operation

Excess electricity produced but not utilized directly can be supplied to the (public) network, receiving revenue or saving other fuels in the parallel operating engine/generator sets. As the operation is continuous, the choice of the engine's/generator's power class depends on the available biogas production rate. The savings and earnings from the excess electricity produced from biogas have to provide an economic incentive to invest in a larger biogas plant, engine and generator than actually needed to satisfy one's own demand. Another alternative is to simply reduce the power output of the gen-set and follow the demand profile, i.e. operate similarly to the isolated mode. At very low demand however the gen-set will operate with a low efficiency too.

7.2.3 Biogas/Power Demand Exceeds Production (see Fig. 7.3)

7.2.3.1 Isolated Operation

Further to the exploitation of all possibilities to raise biogas production the power demand which cannot be satisfied by biogas will have to be satisfied through other fuels such as diesel, petrol, LPG or alcohol. Here the dueal fuel diesel gas engine offers a specific advantage as it can operate not only at fluctuating rates of biogas but also at a comparatively high efficiency in part load operation. This makes the diesel gas engine an ideal choice for uneven power demand profiles in cases of insufficient biogas supply. The power class of the engine to be chosen depends on the demand of the largest single consumer or the sum of the consumers operating simultaneously.

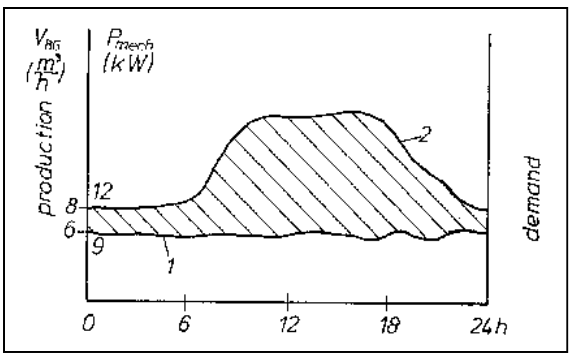


Fig 7.3: Daily profiles: 1 biogas and potential power production, 2 power demand, \\\\shortage in biogas/// power production. Power shortage to be compensated by other fuels/energies in isolated operation or by purchase of electric power from grid in parallel operation.

Building a storage for unutilized biogas from low demand hours for supplementation in high demand hours is one solution and will find its economic justification in relation to the cost and availability of the supplementary fuel saved by the storage. Last but not least the power demand on the biogas engine system may be lowered by using other means to satisfy it or to refrain from its satisfaction partly.

7.2.3.2 Grid Parallel Operation

In cases where the electric supply from another grid already exists, the biogasdriven gen-set only supplements part of the demand. The project as such remains a net consumer. The power class of the engine and generator is chosen in accordance with the biogas production rate (1 m³/h - 1.5 kW mech).

The gen-set should be operated continuously to avoid storage.

7.2.4 Power Demand Partly Higher, Partly Lower than Biogas Fuel Production

7.2.4.1 Isolated Operation

As long as the biogas produced during the low demand hours can satisfy the additional requirements in the high demand period intermediate storage is a possible solution. Wherever the excess power demand cannot be satisfied by stored biogas, additional fuel is required with the diesel gas engine offering a good solution.

Any remaining biogas can serve other useful purposes.

The engine's power class is chosen in accordance with the power required from the largest consumer or the sum of the requirements of equipment necessarily operated simultaneously.

7.2.4.2 Grid Parallel Operation

If the biogas-driven gen-set is operated in combination with diesel-driven gen-sets within a larger isolated network under a common administration the savings are directly felt in the reduction of the diesel fuel consumption of the other gen-sets. The biogas-driven gen-set's power class is chosen in accordance with the biogas production rate and is operated continuously.

In combination with a public utility the choice of operation and power class of the engine is largely a function of the contract concerning the tariffs for supply to the grid and drawing from the grid (see Chapter 7.2.1). A detachable cub-network for isolated operation of selected equipment may be an alternative as then the project remains a net consumer of electricity.

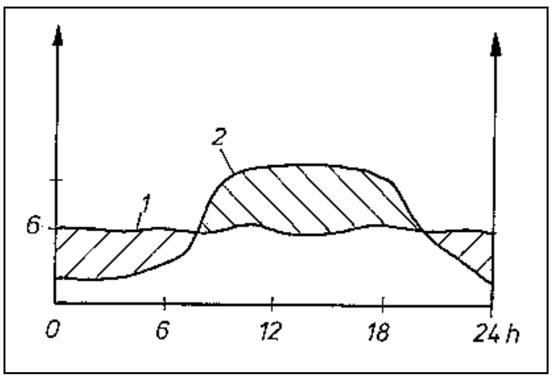


Fig 24: Daily profiles: 1 biogas and potential power production, 2 power demand; option for storage of excess biogas for periods of biogas shortage. /// excess biogas production \\\biogas shortage

7.2.5 Investment and Operational Cost

Investment for the biogas engine system will differ from case to case, depending on what is actually required for completion of the system:

- biogas plant, gas storage,
- biogas piping and instrumentation,
- engine cum modification,
- driven machine cum transmission,
- civil works, i.e. foundations, sheds, fences, etc.,
- wiring, piping, switchgear.

Often the biogas plant already exists or is being built as a biological treatment plant for wastes, residues or other. It is therefore not part of the investment for the engine system. In other cases an engine cum driven machine is already there while a plant, its infrastructure and engine modification are needed.

The operational costs involve the manpower, service and maintenance of the system as mentioned earlier. Again, if for instance the operation of the plant is done and paid for under a different aspect, e.g. waste treatment, the "biogas fuel price" is lowered as it only needs to consider the efforts for gas preparation, e.g. piping, storage, measuring,etc. Further influence on the fuel price comes from the production rate of the biogas plant.

The establishment of a biogas fuel price (per m³ or per kWh) is useful where a biogas engine competes against differently fueled engines or electric power Whatever the actual situation, biogas will never be a fuel absolutely "free of charge."

7.2.6 Two Critical Remarks

The evaluation of the economic parameters is subject to the individual situation in the country and region concerned. The economic analysis of the many different cases would not only be tedious but, being a subject of its own, would go beyond the framework of this publication. Even though the issues are mentioned here, some projects may require a deeper economic analysis. The use of more specialized literature on the economics of renewable energy systems [18] and of the planning, design and operation of the biogas production plants [3, 4, 5, 6] is therefore recommended.

After careful consideration of the planning parameters the solution to refrain from a biogas engine venture and to obtain the services expected from the biogas system in an alternative way may appear reasonable. The "zero" solution should not prematurely or categorically be excluded in the planning process. The more reasons for doubt about the feasibility of such a project, the greater is the possibility of eventual failure. The waste of effort and economic resources involved is a pity, all the more so when these resources are scarce. Another aspect is that the biogas technology is still new in some areas and is not approved of by everyone. A failure of a biogas engine project would only discourage further projects which might have become successful in their specific situation.

7.3 Adaptation of plant, engine and driven machine

7.3.1 Dimensioning of Biogas Plant and Gas Storage

One of the determining factors for the dimensioning of the biogas plant is the biogas production needed to satisfy the fuel demand for the production of mechanical/ electric power per day. The combining figure is the biogas consumption of an engine per unit of mechanical power produced, i.e. the specific fuel consumption. It ranges from 0.5 . . . 0.8 m³/kWh and is largely dependent on gas quality, temperature, pressure as well as the engine's own efficiency and point of operation. (For determination of the actual calorific value of the biogas see Chapter 4.2. For guidelines for the design of a biogas plant see Appendix V.)

If the anticipated mode of operation of the engine cum driven machine is continuous the biogas plant must be designed to continuously produce the amount of biogas demanded by the engine at the required power output. The daily consumption of the engine is established by

$$d_g = \sqrt{\frac{4 \cdot A_g}{\pi}} = \sqrt{\frac{4 \cdot 0.000225}{\pi}} = 0.017 \text{ m} = 17 \text{ mm} \quad \text{(Equ. 7.1)}$$

The production rate of the biogas plant may need to be bigger than the calculated value for the engine if other gas consumers are operated at the same time (cooking, heating, lighting).

In the case of non-continuous operation of the engine, e.g. only several hours per day at different loads, the plant still needs to produce the required amount of biogas needed each day but at a lower production rate per hour than consumed by the engine. A storage gas holder can be filled while the engine remains idle. It is emptied while the engine is in operation and consumes more than the plant produces. The actual volume of the gas holder is a function of the plant production rate, engine consumption as well as the frequency and duration of the engine operation periods. The following example shall demonstrate the interdependence of the above-mentioned parameters:

- -Anticipated machine power demand (= engine operational power output): P = 10kW
- -specific fuel consumption: sfc = 0.6 m³/kWh i.e. consumption per hour: fc = 6 m³/h
- -specific gas production rate:

sgp = 0.8 · m³/m³ plant · day

operational daily schedules, alternative:

a) continuously,
b) 8 hours once a day,
c) 4 hours twice a day with 8-hour standstill between each operational period.

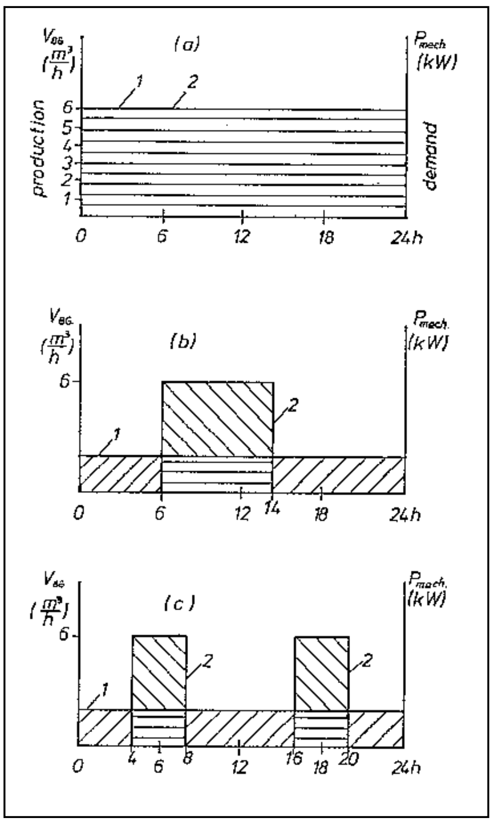


Fig. 7.5: Daily production and demand profiles for the example. - biogas production directly consumed. /// excess biogas production for storage, \\\biogas drawn from store to cover for actual shortage.

Solutions:

a) The plant needs to produce a daily volume rate Vbg of

Vbg = 24 h/d · 0.6 m³/kWh ·10 kW = 144 m³/d (see Equ.7.1)

144 m³/d of biogas. Its size, i.e. digester volume Vd, can be established by

Vd=Vbg·(1/sgp) (Equ. 7.2)

$$= 144\text{m}^3/\text{d} \cdot \frac{1}{0.8\text{m}^3/(\text{m}^3_{\text{plant}} \cdot \text{d})}$$

The plant size is 180 m³; extra gas storage is theoretically not necessary.

b) Plant production rate per day

Vbg = 8 · 0.6 · 10 = 48 m³/d

or per hour

Vbg = 48/24 = 2 m³/h

Plant size (digester volume)

Vd = 48· 1/0.8 = 60 m³

Gas storage capacity

The gas storage capacity needs to consider the rate of production as well as the rate and the period of gas consumption. In this example gas is needed at a rate of 6 m³/h for an operational period of eight hours. The gas volume consumed per period is 6 8 =48 m³.

The production of gas was found to be V'bg = 2 m³/h which results in a volume produced of 8 2 = 16 m³ during the operational period.

The gas storage volume Vs only has to cater for the difference between the volume consumed and produced during the operational period to (in h):

Vs = (fc · to) - (Vbg · to) (Equ. 7.3)

Vs=to (fc-Vbg)

In this specific case the storage volume is

Vs =8h(6m³/h-2m³/h)=32m³.

c) Plant production rate per day

Vbg=8 · 0.6 · 10= 48m³/d = 2m³/h

Plant size (digester volume)

Vd = 48 · (1/0.8) = 60 m³

Gas storage capacity

In this case the operational time of eight hours per day has been split into two periods of four hours each. The gas storage volume

Vs = 4 h (6 m³/h-2 m³/h)= 16 m³

is only half as large as in the previous case where the machine was operated in one long period instead of two shorter ones. The digester size is not affected.

In the above example it was assumed that the standstill periods between the operational periods were equally long so that sufficient time for refilling was available. If frequency and duration of operational and standstill periods are unequally distributed the gas store will have to be suitably larger. A balance calculation with the production rate and time will be useful to ensure that the gas store is always full enough for the next operational period.

For reasons of fluctuations in the gas production and the fuel consumption a certain storage volume should however always exist.

Likewise storage tanks should always be oversized by about 10%.

Existing storage capacity within the digester (depending on type) reduces the required storage volume accordingly.

The examples show that there is an incentive to consider the effect on the gas storage volumes when planning the daily operational schedules of engine and driven machine. On the other hand it will not be very advantageous for the engine to be operated in short stop-and-go periods only as the phases of warming up and cooling down (condensation) expose an engine to more wear and tear than normal operation. A compromise has to be found between the lower investment for a smaller gas storage and the risk (cost) of a possibly shorter life span of the engine. Two periods of operation per day may serve as an orientation value whilst the actual economic situation or other boundary conditions may provide good reasons to decide differently.

7.3.2 Choice of Engine

An engine is mainly specified by its type and by its maximum (rated) power at its maximum speed (e.g. "diesel engine, 30 kW at 2000 1/min or rpm"). What this means is that it may well be operated at lower speeds and power output but not above the maximum data given. An operation at lower power and speed than the maximum will often be found more economic in terms of fuel consumption and engine life. When considering the purchase of an engine one should not confuse the maximum or rated performance as given in the technical specification of an engine with the optimum performance in economic terms. The engine's performance curves, i.e. power, torque and specific fuel consumption vs. speed, are much more useful in determining the point of operation and selecting an engine that will meet the driven machine's rquirements while it operates at a high efficiency.

The determination of the main operational parameters of an engine, i.e. range of power and speed, is largely a function of the requirements of the driven machine. The choice of engine type, however, follows the availability, the market situation (price) for fuel, spares and service and some other operational parameters like the required type of control, fuel availability, etc. The following elaborations shall explain the relevance of these parameters in more detail.

For a better distinction between the different power terms the following definitions shall be used:

Peng,r rated (maximum specified) engine power,
Peng,a actual operating engine power,
Pmach power required by driven machine,
Pgen power required by electric generator,
Pel electric power produced by electric generator.

7.3.2.1 Engine Speed

Every machine has a certain but limited speed range within which it can be operated. Within this range lies a point or narrow range of optimum operation where the specific fuel consumption is relatively low. The longer the engine is operated, the more relevant are the savings in fuel (cost) when the engine operates in or near its optimum performance.

Fig. 7.6 shows that the specific fuel consumption has a minimum value at about 80 . . . 90% of the maximum (rated) speed nr. The maximum obtainable power at this speed, i.e. 80% of the rated speed mark is again about 80% of the rated power. For reasons of fuel economy and engine life the operational speed should therefore be selected within the optimum range, e.g. 70 . . . 90% in the above example. If the speed of the driven machine is equal or near the optimum speed of the engine, direct shaft drive is possible, otherwise a V-belt transmission or gear can be used to adapt the speeds of the two machines as required.

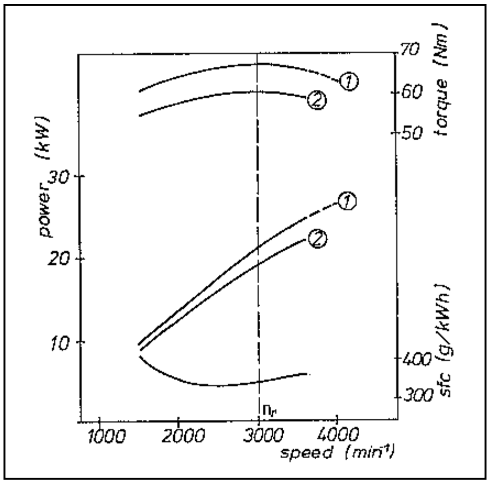


Fig. 7.6: Typical engine performance curves showing the power, torque and specific fuel consumption as a function of the speed. 1 maximum shortterm performance, 2 allowable performance for continuous operation.

Some driven machines (pumps, generators) are available in speed versions of 1500 1/min and 3 000 1/min (or 1800/3 600 for 60 Hz).

The high speed versions require a high speed engine for direct coupling. For a similar power range high speed engines are smaller, hence cheaper to buy (Otto), but have a lower efficiency in biogas operation and a lower life expectancy.

7.3.2.2 Engine Power

When looking at the power output in selecting an engine one needs to consider the future main regime of operation:

- continuous, i.e. periods each longer than about one hour, or
- non-continuous, i.e. shorter periods.

For shorter periods the engine may be operated at its maximum power obtainable at the selected speed, i.e. about 80% of the maximum rated speed following the speed/fuel argument above. Subsequently the power required by the driven machine Pmach should not exceed 80% of the engine's rated power if specified at maximum speed:

Pmach = 0.8 · Peng,r (Equ. 7.4)

For continuous operation, which is the more usual mode, the power output needs to be lower than the maximum rated. Engine manufacturers themselves often quote two different types of power, maximum power and continuous power. For a given (or selected) speed the continuous power is usually between 10% and 20% lower than the maximum power (see Fig. 7.6) as the specific fuel consumption, which is not constant over the whole power range, has its lowest value at 80 . . . 90% of maximum power. The power demanded by the machine shall therefore equal 80 . . . 90% of the engine's maximum power at the selected speed. In other words, in continuous operation the power selected for optimum fuel economy is now reduced by two issues. One reduction is caused by selection of the optimum speed (see Equ. 7.4) and another one by operating at a lower power output than possible to improve the fuel consumption even further:

Pmach = 0.8 · 0.8 · Peng,r (Equ. 7.5)

The engine selected for a given power demand from a machine will hence have a higher maximum power output:

Peng,r = 1/(0.8 · 0.8) · Pmach = 1.56 · Pmach (Equ. 7.6)
i.e. more than 50% greater than the power at which it will later have to operate.

The type of engine, i.e. diesel or petrol, chosen for modification has a further influence on the power rating of the selected engine.

Diesel engines do not significantly lose power when operated in dual fuel mode. They therefore only need to follow the selection criteria explained above.

Diesel engines modified into Otto engines or modified petrol engines are subject to a decrease of about 20 % of their former performance after modification to a biogas engine because of a decrease in volumetric efficiency. In other words, the choice of the power class of an Otto engine needs to consider the

- -lower output in continuous operation for reasons of speed and fuel economy as explained earlier, and the
- lower power output as a result of modification, i.e. reduction of volumetric efficiency.

The power rating of the still unmodified Otto engine in relation to all mentioned criteria is

Peng,r = 1/(0.8 · 0.8 · 0.8) · Pmach=1.9 ·Pmach (Equ 7.7)

i.e. almost two times the actual power demand in operation with biogas.

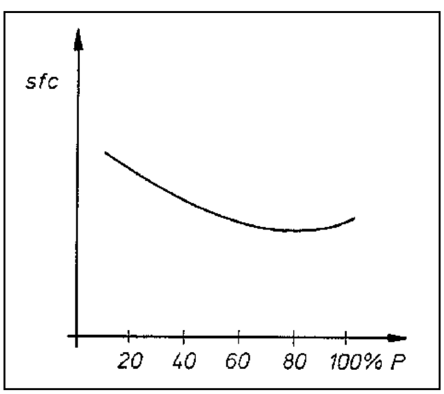


Fig. 7 7: Specific fuel consumption, sfc, as a function of power output at constant speed (schematic)

In case an Otto engine is expected to operate at a much lower speed than 80% of what was specified for its original power output (e.g. 1500 1/min instead of 4000 1/min), the expected power output decreases even further, almost by the same rate as the speed (see Fig. 6.1). This may explain why commercially available Otto gas engines produce only about 10 kW per lifer displaced volume at a low speed (2 000 1/min) while a standard vehicle petrol engine produces about 30 kW per lifer at higher speeds (5 000 1/min).

The above analysis while useful for the understanding of the influential factors for the engine selection shall however be understood as a guideline rather than an instruction to be followed too strictly. Some engines are operated within a range of speeds, not one speed only. Others are only rarely operated so that the fuel economy is a secondary aspect. When calculating the power rating for an engine to be purchased one will not often find the exactly required engine but choose a smaller or larger one. Otto engines however should not be oversized more than necessary to prevent operation at partial load with lower efficiency. Dual fuel engines do not lose much efficiency in partial load.

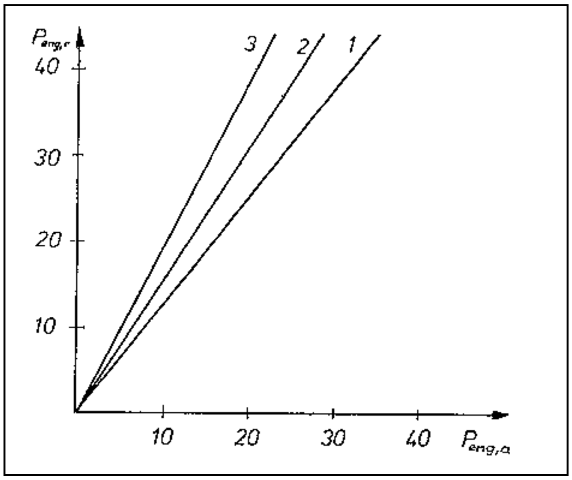


Fig. 7.8: Relation of rated power of engine (before modification), Peng,r and its actual power output Peng a at optimal economic conditions with biogas. Operational speed = 0.8 X max. speed. 1 diesel gas (dual fuel) engine operating short periods only, 2 as 1 but operating continuously, 3 Otto biogas engine, continuous operation.

The power considerations above have normally been considered by manufacturers of commercially available biogas engines. They can therefore be ordered specifying the actual power demand/speed of the driven machine. "Oversizing" by 10 . . . 20% is necessary when these engines are originally designed for LPG or natural gas but not specifically for biogas.

7.3.2.3 Engine Availability and Price

The above-mentioned selection criteria may be affected by considerations of the engine's price, its own availability and the availability of spares and service when necessary. A larger engine which may run more slowly and at a lower fuel consumption rate may be more expensive, also in terms of service and maintenance. A realistic anticipation of running costs (lubricant, service, manpower) and the actual operational periods is therefore necessary.

In other cases a certain engine may already be available and the question of purchasing another one does not arise at all.

7.3.2.4 Engine Control

The anticipated mode of control, i.e. whether automatic or manual, may be decisive for the engine type. Diesel gas engines can be automatically controlled using their governor while Otto engines usually need additional equipment for that purpose.

7.3.2.5 Fuel Consumption

The fuel consumption is mainly dependent on the demand of mechanical power from the driven equipment or the demand of electric power from the grid or connected consumers. The type of the engine, the modification and the individual engine efficiency however also play their role in the actual fuel demand. The nomogram in Fig. 7.9 gives a random relation between biogas production and mechanical/electric power obtainable for diesel gas and Otto gas engines. As some simplifying assumptions had to be made, the nomogram is to be seen as a planning instrument rather than for the final calculations in designing the system.

7.3.2.6 Fuel Availability

In cases where the supply of fuel is not assured an alternative or auxiliary fuel would be required. Diesel dual fuel engines provide an option to use diesel fuel at any time and at any rate. On the other hand they require a supply of diesel fuel together with biogas. Otto engines are independent of liquid fuel supply. They may use LPG in case of biogas scarcity or run on alcohol or petrol again if the carburetor has been retained in its original function.

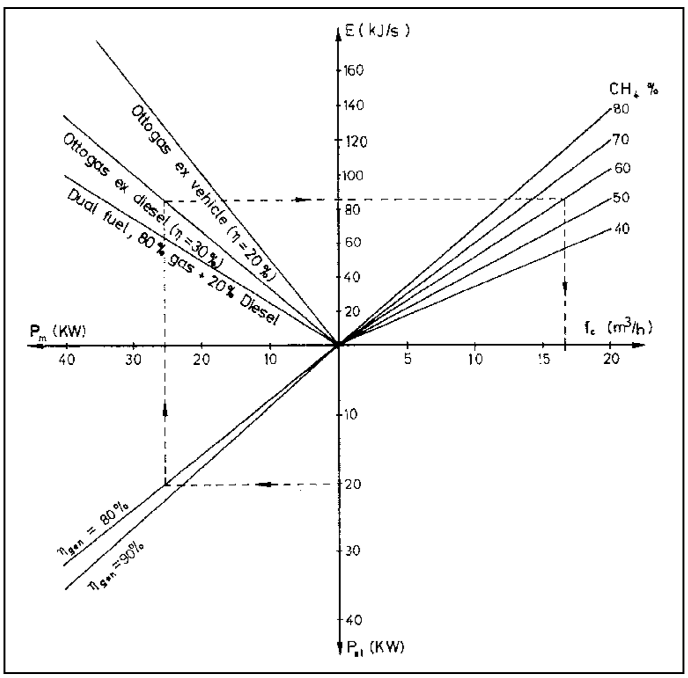


Fig. 7.9: Nomogram for the relation of fuel consumption/fuel demand fc, biogas quality CH_4%, fuel energy flow E, type of engine used, mechanical power output Pm, electric generator efficiency ngen, and electric power output Pel. Basic gas data assumed: temperature 25 °C, pressure 960 mbar, ref. humidity 100%.

Example for the use of the nomograph:

Given data:	el. power required	Pel= 20 kW	
	generator efficiency	ηgen = 80%	
	engine chosen	Ottogas ex-diesel	
	biogas available	CH_4% = 60	
Result:		fuel consumption	fc = 16.7 m³/h
	specific overall fuel consumption	sfc =16.7/20 = 0.835 m³/Kwh	

7.3.2.7 Expected Engine

It is common knowledge that diesel gas engines or Otto engines on the basis of diesel engines are more appropriate for longer service than ex-vehicle Otto engines. Their higher price, however, requires justification by long and frequent periods of operation respectively. In general slow running engines last longer than fast ones but are larger and more expensive.

7.3.3 Choice and Operation of Driven Machine

The kind of driven machine chosen is clearly a function of the required service. For the final determination of the machine's type and size, however, there are a few more considerations to be made with respect to the consequences for the engine and even the biogas supply side. It is therefore of advantage for the economy of the whole system if possible alternatives for the future service and operational schedule can be anticipated (see also Chapter 7.2).

A good example is the filling of a water storage tank which requires a certain amount of water daily. The energy for the daily job of water lifting shall be 400 kW and remains constant irrespective of the type of pump, engine and operational schedule requirements. Likewise the size or daily gas production rate of the biogas plant is not effected under the simplifying assumption of a uniform efficiency of engine and pump. The interdependence of pumping schedule, gas storage and size of pump and engine, however, shows a significant difference in results (see table below).

Interdependence of operational schedule, biogas storage and power of engine and driven machine (pump)

Operational schedule selected (frequency	**Mechanical (pump) power required (kW)**	**Gas storage (kWh/m³)[1]**	**Engine rated power (kW)[2]**
X h/d)			
1 X 4	100	333/56	133
2 X 2	100	167/28	133
1 X 12	33	200/33	44
2X 6	33	100/17	44
1 X 24	17	0[3]	23

[1] For biogas with 60% CH4 at standard conditions.
[2] Assuming Pmach = 0.75 Peng.
[3] 0-storage is merely theoretical; a minimum storage of 1-fur operation should be provided.

The cheapest solution in terms of investment is obviously a small machine set, no or only little biogas storage and a continuous service. It is recommendable as long as continuous supervision, service and maintenance are assured. Under further consideration of the effect of continuous service on the engine's life span, necessary overhauls, the fact that an engine cum machine may already exist and other external factors, one might however have to select another schedule as an appropriate compromise.

7.3.4 Choice of Transmission

The transmission not only serves to connect the shafts of the engine and the driven machine, but it also provides for a possibility of an alteration of speeds and speed ratios.

Common engine speed ranges are:

- n =1 300 . . . 3 000 min^{-1} (rpm) for diesel engines
- n = 1500 . . . 5 000 min^{-1} (rpm) for petrol engines

whereby each engine should be operated at its optimal speed range as explained in Chapter 7.3.2.

Machine speeds can also have different ranges but are often designed to match with standard speeds of electric motors in order to facilitate a direct connection via elastic coupling or shaft. Standard speeds for electric motors (AC) and direct driven machines are:

- $n = 1\,500\ min^{-1}$ or $3\,000\ min^{-1}$ for a frequency of 50 Hz
- $n = 1800\ min^{-1}$ or $3\,600\ min^{-1}$ for a frequency of 60 Hz.

These speeds may well coincide with the optimum speed range of an engine so that direct coupling or shaft drive is possible. Direct coupling, however, requires matching flanges of engine and machine housings for direct mounting of a rubber-damped coupling at the crankshaft. Otherwise an external coupling with rubber elements or a propeller shaft is required. All direct drives cause the directions of rotation of engine and machine to be opposite. They offer the better solution for the drive of equipment that requires a high degree of speed (frequency) stability, e.g. electric generators.

Should the direction of rotation not meet the above conditions or should the speeds of machine and engine not coincide well enough, a transmission with V-belts or flat belts and pulleys is recommended.
The transmission ratio is determined by the ratio of diameters of the pulleys Deng/Dmach:

$n_{eng}/n_{mach} = D_{mach}/D_{eng}$ (Equ. 7.8)

Flat belts are still used in places where V-belts are scarce. Their advantage is that they can be cut to size from a long piece and joined together with a clamp which also allows repair. More slip and power loss through friction as well as the fact that they tend to run off the pulleys when not properly aligned is however disadvantageous.

While the direct transmission by shaft or rubber-damped coupling is almost free of power losses, slip and friction consume a certain amount of the power transmitted from the engine. For V-belts the power loss ranges from 3 . . . 8%, for flat belts from 10 . . . 20%. A transmission efficiency ñT can be defined as

$$\eta_T = \frac{P_{mach}}{P_{eng,a}} = 0.8 \cdots 0.97 \quad \text{(Equ. 7.9)}$$

so that finally the actual power demand from the engine in case of belt transmission is

$$P_{eng,a} = \frac{1}{\eta_T} \cdot P_{mach}$$

i.e. larger than the power demand at the machine shaft. For direct coupling without losses assume $\eta_T = 1$

The transmission of power by belts imposes a radial load on the bearings of both engine and machine. While most driven machines and stationary engines are designed to also operate with V-belts (see specifications), most vehicle engines are designed to transmit their power by an axial connection to their gearbox. The radial load may therefore be harmful to the engine's crankshaft bearing.

In cases of doubt a separate axial shaft for the pulley with its own bearings for holding the radial load will resolve the problem (see Fig. 7.10, d).

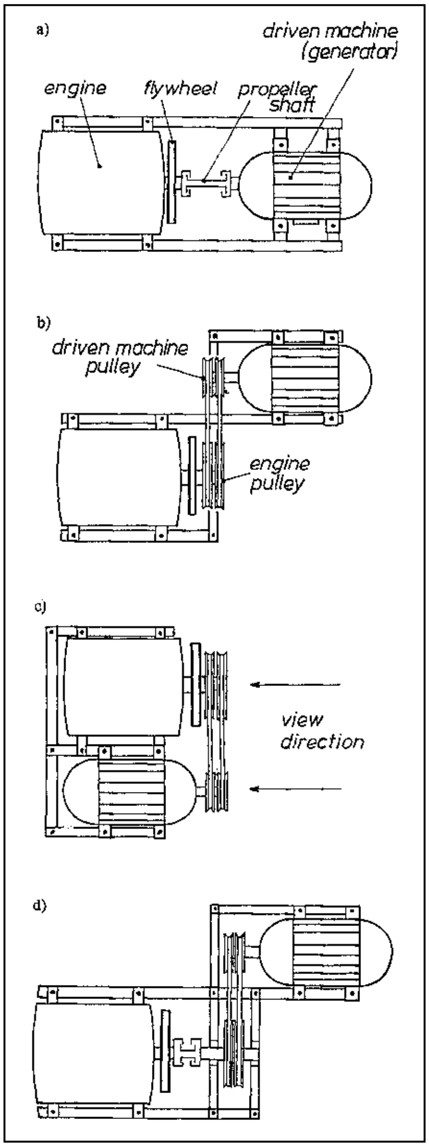


Fig 7.10: Alternative positioning of engine and driven machine depending on direction of rotation and type of transmission. Direction of rotation is given as viewed facing the shaft ends, example c).

a) Propeller shaft, transmission ratio 1: 1; engine rotation: anticlockwise, machine rotation: clock- wise.
b) V-belt and pulleys, transmission ratio variable with pulley diameters; engine rotation: anticlock- wise, machine rotation: clockwise.
c) as in b); engine rotation: anticlockwise, machine rotation: anticlockwise.
d) as in b) but extra propeller shaft and pulley bearings to hold radial load; for engines with shaft bearing not designed for radial drive (vehicle engines).

Belt drive offers an additional advantage for cases where the engine has difficulties to start up while already pulling the machine under load. The belt can be loosened to allow the engine to first gain speed. It is then gradually tightened (on the unloaded side!) with a tensioner until the machine has also gained its speed. With very frequent start-ups in this way the wear and tear of the belts will however naturally increase.

An alteration of the direction of rotation between engine and machine can be effected by placing the engine and the machine either beside each other or in a row. Other alternatives for transmission are gears, either open or in a casing (gearbox). They are however much more expensive, require lubrication and may only be economically justified for continuous service in terms of years and for larger machines.

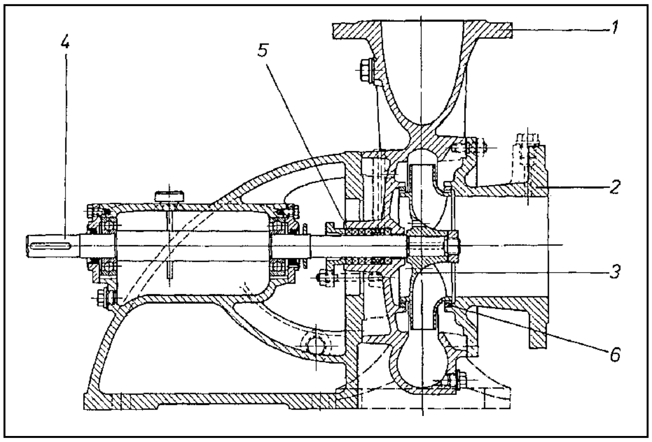


Fig. 7.11: Cross-sectional view of single-stage centrifugal pump (KSB).

1 discharge (pressure) flange with diffusor, 2 inlet (suction) flange, 3 impeller, 4 drive shaft, 5 stuffing box, 6 impeller/casing seal.

The other transmission elements are standard components, easy to manufacture (pulleys) or to be obtained even from unserviceable vehicles (propeller shafts). Both pulleys and shafts require an adapting flange or hub to be connected to the shaft (flywheel) of the engine and of the machine respectively. These flanges require precision in manufacture for reasons of rotation balance. An unbalanced shaft or pulley brings about destruction of the shaft bearings prematurely.

7.4 Engine and machine, two common examples

7.4.1 Engine and Water Pump

Water pumping, whether for municipal, industrial or agricultural purposes, cares for a substantial demand of mechanical energy. The most common type of pump is the centrifugal pump built in single-stage versions up to about 100 m waterhead or in multistage versions for higher heads.

A pump transforms mechanical energy into hydraulic energy and has, like other energytransforming machines, its specific performance characteristics. An example is shown in Fig. 7.12

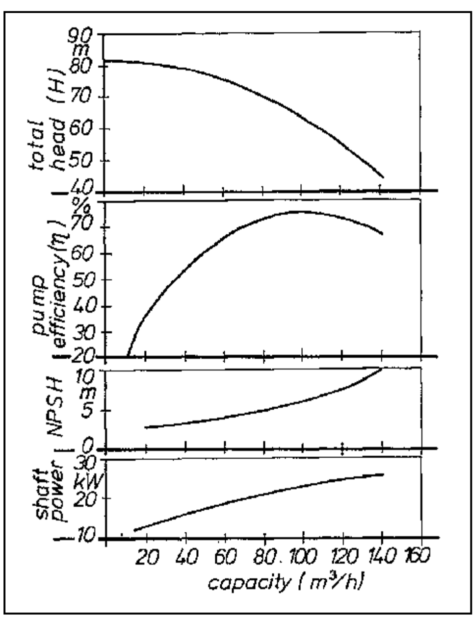


Fig. 7.12: Characteristic curves of a radial centrifugal pump at constant speed (KSB)

The charts of Fig. 7.12 demonstrate the essential pump parameters and their interdependence:

- -The head (sometimes given in pressure rise /\p) increases when the capacity Q (or volume flow rate) decreases.
- -The power demand increases with the capacity Q even though the head decreases. The influence of the increasing capacity is stronger.
- -The efficiency has its maximum at the "design point" of the pump, i.e. at the values of capacity, head and speed chosen to provide the basic data for the design of the impeller and the volute casing.

The power demand of a pump is established by the following equation:

$$P = Q \cdot H \cdot g \cdot \rho \cdot \frac{1}{\eta_P} \cdot \frac{1}{1000} (kW)$$ (Equ. 7.10)

with: Q = capacity in m³/s, H = total head in m, g = gravity constant (9.81 m/s²), p = density (water: 1 000 kg/m³), ñp = pump efficiency (0.5 . . . 0.75).

A centrifugal pump's design data (Q, H, P) are either specified at one selected speed n (on the nameplate) or given in a performance diagram similar to the one given in Fig. 7.12 supplied with the pump.

While centrifugal pumps are designed to match with standard electric motor speeds (see Chapter 7.3) they may well be operated at other shaft speeds, preferably below the design speed. When operated at a lower speed than specified, the values of capacity, head and power demand change as follows (indexed 1 at specified speed, 2 at actual speed):

$$Q_2 = Q_1 \cdot \frac{n_2}{n_1}$$ (Equ. 7.11)

$$H_2 = H_1 \cdot (\frac{n_2}{n_1})^2$$ (Equ. 7.12)

$$P_2 = P_1 \cdot (\frac{n_2}{n_1})^3$$ (Equ. 7.13)

In cases where the pump is specified by its pressure rise /\p rather than by its head H, use the transformation

$$\Delta p = \rho \cdot g \cdot H(in \cdot N / m^2)$$ (Equ. 7.14)

Some pump manufacturers supply diagrams indicating the pump's performance at different speeds as in Fig. 7.13.

As can be seen from Fig. 7.13 a change in speed results in a new characteristic curve. Speed variation provides a practical way of control for head and capacity. This mode of control is far more energy-economic than throttling the flow with a valve, as the pump and hence the engine would consume extra energy to overcome the flow resistance produced by the throttle valve. As engines can vary their speed, engine-driven pumps should be speed-controlled.

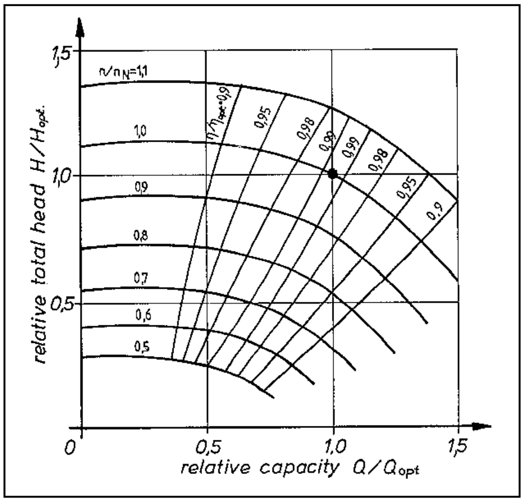


Fig. 7.13: Performance chart of a speed-controlled centrifugal pump (KSB)

Centrifugal pumps should never be throttled on their suction (inlet) side because of cavitation which wild gradually damage the impeller. However, as a valve on the pressure side is usually necessary for facilitating the start-up of the engine it can also be used for capacity control. The valve is to be kept closed when the engine is started and after one or two minutes gradually but fully opened. Priming of centrifugal pumps is necessary on the suction side if the pump does not suck water by itself (self-priming pump). Dry running of pumps is to be avoided.

The pump's performance chart and the other equations given will be useful for the specification of the engine in terms of power and speed. The engine should be chosen with the aim to match the operational point (or range) of the pump with the most fueleconomic point (or range) of the engine. The example below shall demonstrate the procedure.

Example:

Given situation: Water is to be supplied to a cattle farm with a daily consumption of 1 500 m³. The farm's buffer tank is located 40 m above the level of a river from which water shall be pumped. The flow resistance in the piping is estimated at an equivalent of 10 m; the total head for the pump is therefore 40 + 10 = 50 m. The pump available shall have the characteristics shown in Fig. 7.12; the speed specified is n = 3 000 1/min .

Step 1:
Transform the capacity into units matching the pump's diagrams and the formula for power:

$Q = 1\,500\ m^3/d: 24\ h/d = 62.5\ m^3/h$
$Q = 62.5\ m^3/h: 3\,600\ s/h = 0.0174\ m^3/s$

Step 2:
Establish the actual power demand using Equ. 7.10 and the efficiency from the pump chart at Q = 62.5 m³/s:

$$P = 0.0174 \cdot 50 \cdot 9.81 \cdot \frac{1}{0.67} \cdot \frac{1}{1000}$$

P = 12.74 kW

From the performance diagrams at Q = 62.5 m³/h the pump produces a head of H = 74 m and requires a power of P = 19 kW. If the pump was operated at its designed shaft speed of n = 3 000

min^{-1} water would be jetted into the tank at high speed, unnecessarily consuming extra power. The power difference between the diagram value and the one calculated would be wasted, hence extra fuel for the engine. A reduction of speed will solve the problem.

Step 3:
Adapt given pump to given situation using Equ. 7.13

$$n_2 = n_1 \cdot (\frac{P_2}{P_1})^{1/3} = 3000 \cdot (\frac{12.74}{19.0})^{1/3} = 2626\,\text{min}^{-1}$$

n2/n1= 2626/3000 = 0.875

The new head at Q = 62.5 m³/s, using Equ. 7.12

$$H_2 = H_1 \cdot (\frac{n_2}{n_1}) = 74(0.0875)^2 = 53.5\text{m}$$

H=53.5 m is sufficient for the given case of H = 50 m.

Step 4:
Specification of the engine (still unmodified) Data required by the machine (pump)

- operational speed n_{mach} = 2 626 1/min
- operational continuous power P= 12.74 kW

a) Otto engine (ex-petrol)

rated engine power P_{eng} = 1.95· P_{mach} =24.8 kW

rated engine speed n_{eng} = 1/0.8 · n_{mach} =3 283 1/min

A petrol engine to be purchased for modification should have a rated power of about 25 . . . 30 kW at a speed of about 3 300 . . . 4 000 1/min. Suitable engines are found in a variety of vehicles with a cubic capacity of about 1.5 . . . 2.0 lifers. An Otto (ex-vehicle) engine would however be less recommended in a case of continuous operation. With estimated overhaul periods of about 3000 . . . 4 000 hours it needs a total overhaul after every six months.

b) diesel engine for dual fuel operation

rated engine power $P_{eng,r}$ = 1.56 · P_{mach} =19.9 kW

rated engine speed n_{eng} =1/ 0.8 · n_{mach} =3238 1/min

The speed appears to be fairly high for a diesel engine, especially the stationary type. For a transmission using V-belts it is recommended to use an engine with a lower speed, preferably between 1 500 . . . 1 800 1/min. The rated maximum power should be 20 . . . 25 kW. Suitable diesel engines would be types with two or three cylinders, stationary, with a capacity of 2.0 . . . 2.5 lifers.

c) ready-made Otto gas engine (possibly exdiesel)

The specifications given by commercial suppliers already consider the reductions explained earlier for engines to be modified. Such engines can be ordered giving the specified machine data. A little reserve in power and speed, however, may be useful in case the machine requires a higher performance.

Step 5:
Establishing the biogas fuel consumption fc per day

a) Otto engine (self-modified) with
sfc = 0.6 . . . 0.8 m^3/kWh
fc = sfc · P · operation time = 0.6 . . . 0.8·12.74 · 24 = 183 . . . 245 m^3/d

b) diesel dual fuel engine with
sfc = 0.5. . . 0.7 m^3/kWh
fc = 0.5 . . . 0.7 · 12.74 · 24 = 153 . . .214 m^3/d

c) ready-made Otto gas engine with
sfc = 0.5 . . . 0.7 m^3/kWh
fc = 153 . . . 214 m^3/d

Step 6:
Dimension of biogas plant (digester) volume V_{dig}

The specific production rate (spr) of a biogas plant depends on factors like input material, retention time, temperature, etc. as explained in the relevant literature [4, 5]. Practicable values range from 0.3 . . . 1.0 m^3 biogas/m^3 digester volume and day, a range which shows the necessity for a fairly realistic establishment of the spr value. Assuming a value of spr = 0.8 for this example the digester volume is for
a) Otto engines (self-modified)

Vp = fc/spr = 183...245/0.8 = 229 . . .306 m^3

b) diesel dual fuel and ready-made gas Otto engines

Vp = 153...214/0.8 = 179...268 m^3

In order to compensate for fluctuations a 10% oversizing of the biogas plant is recommended. A small gas storage, possibly integrated into the; digester anyhow, of 5. . .10% of the daily production is furthermore useful. Considerations of future increases in water demand have to be made before final planning.

The pump chosen here can easily cater for about twice the capacity (see its per. formance diagram) but would need a larger engine. A slight oversizing of the engine is useful as during operation the piping may gradually become clogged by deposits. Some extra power helps to rinse or unblock the piping or overcome the resistance put up by the deposits.

As an alternative to the given example the following version is possible:

- pumping time 12 hrs/day,
- pump capacity 125 m^3/h,
- engine power rating about 40 kW,
- farm water storage tank volume min: 750 m^3,
- biogas storage tank volume min: 120m^3.

The advantage of a shorter daily operation period (manpower) and a larger interval between the overhauls is likely to be out-weighed by the extra investment for a larger engine and storage tanks for water and biogas.

7.4.2 Biogas Engine and Electric Generator ("Biogas Gen-Set")

The generation of electrical energy represents another suitable option for the utilization of the energy potential of biogas. Electric generators, which can be driven by a choice of turbines and engines, are available in a large variety of sizes and types from various manufacturers. They are usually designed according to standards and enjoy a generally good reputation in terms of reliability, easy maintenance and a relatively low price as the smaller and medium sizes are manufactured in larger series.

Electric alternating current (AC) generators are designed in two different types:

- synchronous,
- asynchronous.

The synchronous type requires a direct current (DC) exitation, either from an external source (e.g. battery) or from an integrated exitation system, the latter of which are known as brushless, self-exitating generators. The frequency of the AC current produced is a function of the rotor (shaft) speed and the number of pole pairs in the stator:

$F = n \cdot Pp \cdot 1/60$ (Equ. 7.15)

with: F = frequency in Hz or s^{-1}, n = speed in min^{-1}, Pp = number of pole pairs.

Example:

$n = 1\,800\ min^{-1}$
$Pp = 2$
$F = ?$

$F = 1\,800 min^{-1} \cdot 2 \cdot 1/60 = 60\ Hz$

The frequency of AC produced from a synchronous generator can be only as stable as the engine speed control system allows. For consumers which require extreme frequency stability the engine needs an adequately sensitive control system. For consumers like electric motors for water pumps or grain mills which can tolerate within certain limits operation with fluctuating frequency, hence speed, a synchronous generator and an engine with a standard control system are well suited.

Asynchronous generators guarantee frequency stability by means of their specific way of exitation. This is achieved by appropriately dimensioned capacitors in isolated operation or taken from the grid frequency in grid parallel operation. An asynchronous electric machine works as a generator as long as its rotor speed is slightly higher than the exact speed for synchronous operation (see Equ. 7.14), the "synchronous speed". It will however work as a motor and consume electricity when operating at a speed lower than the synchronous speed.

This specific feature principally allows the use of standard asynchronous motors as generators. In isolated grid operation, however, a well calibrated exitation system is to be connected, while for grid parallel operation no alterations are required.

Synchronous motors on the other hand require some modification with respect to their exitation systems when used as generators.
Competent expertise is necessary in this case.

The direction of rotation of the generator rotors is usually optional; the connections to the grid will have to be made in accordance with its actual direction of rotation. In case the connection and the direction of rotation do not match the following alterations should be made:

- for 2-phase:
exchange the connections
plus for minus,
minus for plus,
earth remains unchanged;

- for 3-phase:
exchange any two out of the three connections, e.g.
U for V,
V for U,
W,N and earth remain unchanged.

In a case where the gen-set is the only supplier of electricity in an isolated grid a wrong connection results in a wrong direction of rotation of the connected electric motors with possible damage to the driven equipment. In grid parallel operation the phase sequence in three-phase grids must first be established (with a three-phase sequence indicator) before the generator is connected accordingly. Wrong phase connection can damage the generator-severely.

The connection data differ from one country to another. The most commonly used systems are the following two:

- 50 Hz, 220 . . .230 V, 2-phase
- 50 Hz, 380 . . 400 V, 3-phase
- 60Hz, 110V, 2-phase
- 60 Hz, 254V, 2-phase
- 60 Hz, 440V, 3-phase

For the specification of a generator the following data are required:

- Electrical connection data (as above): phase, voltage, frequency;
- Speed: The generator speed should be selected with a view to direct transmission, i.e. propeller shaft or rubber-damped coupling. For diesel gas engines or Otto engines modified from diesel engines a speed of n = 1 500/1 800 min^{-1} (for 50/60 Hz) is optimal. For Otto engines modified from petrol (ex-vehicle) engines n = 3 000/3 600 min^{-1} may be a viable option also, especially as they often show poor performance at speeds lower than n = 2 000 min^{-1}. Life span however is shorter at higher speeds.
- Power: The electric power to be produced must be established from the requirements of the anticipated electric consumers operating simultaneously (check operational schedule). Voltage U, current I and the cos £ value (for electric motors) are either known or can be measured from existing consumers. The electric power demand Pel

Of each piece of equipment can be calculated as follows:

phase apparent power (kVA)

2-phase $Pel = U \cdot I$
3-phase $Pel=U \cdot I \cdot \sqrt{3}$

phase active power (kW)

2-phase $Pel = U \cdot I \cdot \cos\varphi$
3-phase $Pel = U \cdot I \sqrt{3} \cdot \cos\varphi$ (Equ. 7.16)

Resistors like heating, lighting, etc. have a cos φ value of 1, i.e. the active power drawn from the grid equals the apparent power. Electric motors with a cos φ value of 0.75 . . . 0.9 draw active power which is less than the apparent power resulting from measurements with simple A/V meters. The actual cos φ is therefore required to specify the actual (active) power drawn from the grid or generator. Generator manufacturers, however, specify the generator's power output in kVA as the future type of consumption is unknown to them.

Modern generators can bear a short overload of about 2.5 times the specified current. The overload occurs during start-up of electric equipment, especially motors to overcome the break-away torque. To limit the overload for the generator, three-phase electric motors should have star/delta switches. Overdimensioning of gen-sets for start-up peak loads should not be necessary, especially as also the engine can usually produce more power for a short period.

For the selection of an adequate engine the generator's demand in mechanical power Pgen has to be established. The generator's efficiency ηg which considers losses like heat, bearing friction and the power consumption of its own cooling fan is defined as

$$\eta_g = \frac{P_{el}}{P_{gen}} \leq 1 \text{ (Equ. 7.17)}$$

The generator efficiency is specified by the manufacturer and usually ranges at ηg = 0.82 . . .0.92. In case of belt transmission, the transmission efficiency needs to be considered also in a way that the total power demand of engine cum transmission required from the engine Peng, a is:

Peng,a = 1/ηg · 1/ηT · Pel (Equ. 7.18)

Before foal specification of the generator and engine according to the power required by one's own equipment, the operational schedule or the power demand profile respectively has to be sufficiently studied as it also determines the power or power range of the gen-set.

The following example shall serve as a demonstration of the layout of a biogas-driven generator set.

Given situation:

-biogas production (potential)	Vbg = 180 m^3/d = 7.5 m^3/h
-mean specific biogas consumption of engine (estimated)	sfc=0.65 m^3/kWh
-efficiency of generator	ηg=0.9
-transmission	direct, no losses
-voltage (country standard)	U=220/380 V
-frequency (country standard)	f=50 Hz
-daily electric power demand:	
- from 0 to 7hrs	Pel= 2 kW
- from 7 to 17hrs	Pel=12 kW
- from 17 to 24 hrs	Pel= 2 kW

Solution:

Step 1: Establish amount of biogas needed daily for the generation of the required electric power. Electric energy demand per day,
Eel:

Eel =(14h· 2kW)+(10h ·12kW) = 148 kWh/d

Biogas demand for the gen-set per day, Vbg:

$Vbg = Eel \cdot 1/\eta g \cdot sfc$

$Vbg = 148$ kWh/d · 1/0.9 · 0.65 m³/kWh = 107 m³/d

The biogas production of 180 m³/d is more than sufficient for the generation of the power demand, but on a daily basis only.

Step 2:
Establish the mechanical/electric power directly available from the continuous biogas production rate and complete the daily supply/demand profile:

$$P_{el} = V_{bg} \cdot \frac{1}{sfc} \cdot \eta_G = 7.5m^3/h \cdot \frac{1}{0.65m^3/kWh} \cdot 0.9 = 10.4kW$$

-from 0 to 7 furs: excess biogas available
-from 7 to 17 furs: power demand is higher than related biogas production
-from 17 to 24 furs: excess biogas available.

The excess biogas produced during the low demand period provides the possibility for storage to be supplied to the gen-set during the high demand period. Furthermore other consumers like lighting, heating, baking, cooking can utilize the excess gas during that time.

Step 3:
Establish necessary biogas storage.
Biogas demand per hour

$Vbg = Pel \cdot 1/\eta g \cdot sfc$

a) high demand period (7 to 17 furs)

Vbg = 12 kWel· 0.9 · 0.65 m³/kWh = 8.7 m³/h

b) low demand period (0 to 7 hrs and 17 to 24 hrs)

Vbg = 2 kWel · 1/0.9 · 0.65 m³/kWh = 1.44 m³/h

The gas storage capacity Vs needed for the high demand period (see Equ. 7.3):

Vs = 10 h (8.7 m³/h - 7.5 m³/h) = 12 m³

Note that a certain gas quantity is usually stored in the biogas digester itself. A certain volume is however necessary for the normal fluctuations in biogas production in most plants and possible fluctuations in power demand.

In the given situation it would be worthwhile to reconsider the operational schedule with the aim to better adapt biogas production and biogas consumption to each other:

- -Lower the demand for electric power while the operational period is extended (e.g. less water pumped per hour during a longer operation time). The gas storage could be avoided.
- -Raise biogas production to Vbg = 8.7 m³/h to avoid the gas storage. At the same time secure a use for the excess gas produced during the low demand period.
- If no other gas use is available build a smaller biogas plant for a lower biogas production. The daily amount of biogas required to be sufficient for the generation of 148 kWh was 107 m³/d or 4.5 m³/h. A larger gas store is then necessary with Vs = 10 h

(8.7 - 4.5) = 42 m³. For future extension - one may consider about a 25% increase in these figures.

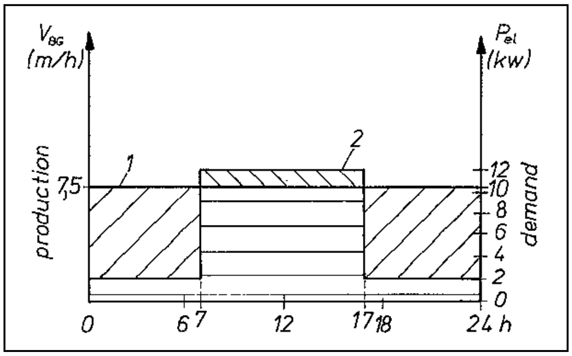


Fig. 7.14: Daily profiles: 1 biogas and potential power production, 2 electric power demand (refers to example)

Step 4:
Specify the generator:
Voltage: 380 V
Frequency: 50 Hz
Phase: 3-phase (unless all existing electric equipment is 2-phase)
Speed: 1500 1/min
Type:

- asynchronous if net parallel operation is anticipated
- synchronous (brushless, self-exitating) if isolated grid operation is anticipated

Power:

- present maximum demand 12 kW
- provision for future extension 25%
- total electric power 15 kW

Current: (necessary for specification of switchgear, cables. connections, etc.)

for 3-phase (cos φ assumed = 0.85)

$$I = \frac{P_{el}}{U \cdot \sqrt{3} \cdot \cos\varphi} = \frac{15\text{kW}}{380 \cdot \sqrt{3} \cdot 0.85} = 26.8\text{A}$$

Step 5:
Select engine

- mechanical power demand from machine, i.e. generator:

$$P_{mech} = \frac{1}{\eta_G} \cdot P_{el} = \frac{1}{0.9} \cdot 15\text{kW} = 16.7\text{kW}$$

The operational power of the engine is about 17 kW while the value of the rated or maximum power of the engine (Peng,r) to be selected for modification is higher (see Chapter 7.3.2).

a) Diesel engine to be modified for dual fuel

$P_{eng,r} = 1.56 \cdot P_{mach} = 1.56 \cdot 16.7\ kW = 25.1\ kW$

b) Otto engine (ex-petrol) to be modified for biogas

$P_{eng,r} = 1.95\ P_{mach} = 1.95 \cdot 16.7\ kW = 32.6\ kW$

- pilot fuel demand:

The diesel engine requires about 20% of its rated diesel fuel consumption for pilot ignition, i.e.

$fc = 16.7\ kW \cdot 0.31/kWh \cdot 0.2 = 1\ l/h$

- speed, transmission:

The generator speed is suitable for direct transmission by rubber-damped coupling or propeller shaft:

$n_{eng} = n_{mach} = 1\ 500\ min^{-1}$

The direction of rotation is usually optional for generators and only related to the mode of cable connection.

The given demand profile allows operation of the engine at a good efficiency for 10 hours a day. During the low demand period the engine can only be operated at almost idling conditions. Its use in this case is hardly economic as the cost for service, maintenance, operating personnel and the depreciation process are dependent on the actual operation period irrespective of the actual power produced.

Some alternatives would be worth considering:

- -Sell electric power to other consumers or public grid at a rate which allows the full utilization of the excess biogas.
- Buy the electric power needed during the low demand period and only switch over to self-generation during the high demand period.
- -Consider renouncing the use of electricity during the low demand period, i.e. operate the engine in high demand period only.
- -Use an additional smaller biogas-driven generator set (3 . . . 4 kW) for the low demand period; this however requires extra investment.

In the last three cases find a useful purpose for the excess gas produced during the low demand period or consider a smaller biogas plant cum appropriate gas storage (see step 3).

8. Utilization of the engine's "Waste" heat

(Cogeneration)

8.1 Theoretical aspects

The degree of utilization of the energy content of engine fuels for power production alone is fairly low, i.e. between 25% and 35% only. Through cogeneration of power and heat the total utilization degree can be improved to about 85%. This provides an incentive to try and use both forms of energy simultaneously whenever possible.

Not only should the waste heat of an engine be utilized whenever power production is the initial issue. Especially in cases where biogas is considered for low temperature heat generation (about 100 °C) an engine should be introduced. The thermodynamic validity of mechanical power is much higher than that of low temperature heat.
Any fuel suitable for utilization in engines has the potential to generate power.

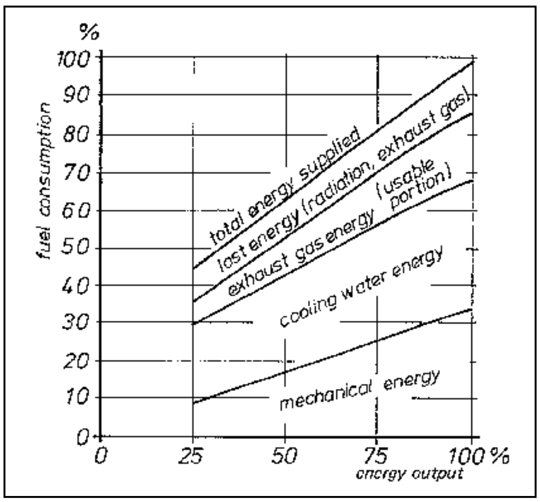


Fig. 8.1: Distribition of fuel energy in an engine (schematic)

When low temperature heat is required, the degree of utilization (efficiency) of the combustion Process in a boiler is about 85%, hence similar to that of a cogeneration unit:

low temperature	Using the relations for establishment of the	
heat demand:	Q = 20 kW fuel consumption and considering power and	
efficiency of boiler:	$\eta_b = 0.085$	heat either separately or as a sum for cogeneration:
mechanical power demand:		
	P = 10 kW	
efficiency of engine:	$\eta_{eng} = 0.3$	fc =1/η *(P + Q) x 1/Hu x 3600
efficiency of cogeneration:	$\eta_c = 0.85$	(see Equ. 4.11)
	separate generation of fuel and power	cogeneration
biogas needed for engine	fc,e = 6 m³/h	-
biogas needed for boiler	fc,b = 4.2 m³/h	-
total biogas needed	fc,tot = 10.2 m³/h	fc,tot = 6.4 m³/h

However the latter can transform one third of the fuel energy into power, a chance that is missed when using a boiler alone.

A simple example shall demonstrate this advantage of a cogeneration unit:

In the case of an optimal matching of heat and power demand in cogeneration the mechanical power of 10 kW is achieved with an additional fuel demand of only 6.4 - 4.2 = 2.2 m^3/h. The same power requires 6 m^3/h when being produced separately. In other words the efficiency of power production in cogeneration is increased from 30% in separate production to more than 80% in cogeneration. It is understood that demand and supply rarely match so perfectly. But as long as satisfying one type of demand, either power or heat, includes the free benefit of at least partially satisfying the other, it is well worth being considered.

As, however, the demand profiles for power and heat have to be somewhat parallel, continuous operation of the whole system appears to be the most favorable condition for cogeneration in general.

8.2 Technical aspects

The potential of the engine's heat energy cannot be utilized fully for two reasons:

- The exhaust gas must leave the heat ex- changer at temperatures above 180 °C. Lower temperatures would allow condensation of fuel impurities such as H_2S which are corrosive with humidity.
- A certain part of the heat is emitted from the engine housing itself to the surroundings (can be useful to heat the machine room if required).

The diagram in Fig. 8.1 helps to establish the actual quantities of the heat obtainable from the cooling water or air of the engine or the exhaust gas. The respective percentage is multiplied with the engine's total fuel energy consumption as calculated earlier in Chapter 4. As a rough estimation the following relation can also help to establish the total fuel energy input Ef in kJ/s:

Ef = 3 . . .4 x engine operating power (in kW) (Equ. 8.1)

Out of the total energy input the following portions can be utilized as heat (values differ with engine type, size, efficiency, etc. by ± 10%):

- cooling water directly:
  35% at temperatures up to 80 °C,
- cooling air from fan:
  35% at temperatures up to 50 °C,
- exhaust gas:
  15% at temperatures up to 200 °C, so that the actual amount of heat obtainable becomes:

Q = proportion x Ef (Equ. 8.2)

whereby: Q = heat flow/transfer in kJ/s.

The temperatures to which the cold flow is actually heated depend on its flow rate. Smaller amounts of water flowing through an exhaust gas heat exchanger will be heated to higher temperatures than a larger water flow. This means that besides the temperatures of the two different media it is also their individual flow rates which determine the amount of heat transferred. The heat increase or decrease of a medium between inlet (1) and outlet (2) of a heat exchanger is given by

$Q = m \cdot cp \cdot \Delta t$ (in kJ/s) (Equ. 8.3)

whereby: Q = heat decrease/increase (in kJ/s), m = mass flow rate of medium (in kg/s), cp = specific heat of medium (kJ/kg· K)2, $\Delta t = t_2 - t_1$, temperature difference of the medium between inlet and outlet of heat exchanger; positive value indicated heat absorption, negative value heat emission.

The heat exchanger surface A, i.e. the area of the material through which the heat is exchanged from the hot to the cold medium, is established using the heat flow to be utilized, e.g. the actual proportion of the total energy Q as found under Equ. 8.2:

$$A = \frac{Q}{k \cdot \Delta t_m} \quad \text{(Equ. 8.4)}$$

The heat transfer coefficient k depends on the types of media flowing on each side of the separation wall, the flow characteristics, and the wall material itself. Assuming that the wall material is metal (steel, brass, copper, aluminum) and the surfaces clean, the following mean k values can be used considering however that they can differ by + 50%:

- liquid - metal wall - liquid: $k = 2500 \frac{kJ}{m^2 \cdot h \cdot K}$
- liquid-metal-gas: k = 150
- gas-metal-gas: k = 80

The liquid is usually water; the gas can be air or exhaust gas.

The active mean temperature difference Δtm varies with the type of heat exchanger but can be established within acceptable tolerances by (see Fig. 8.2):

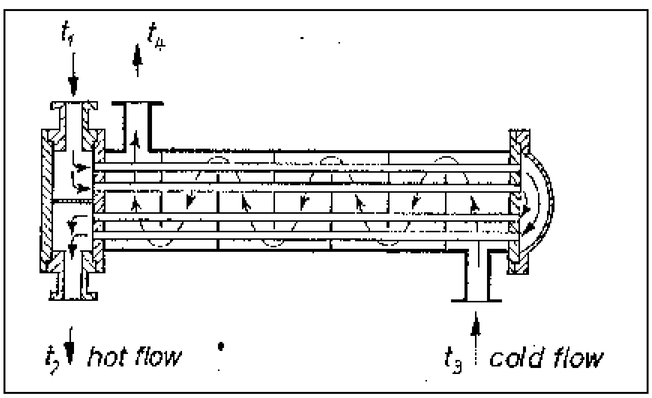


Fig. 8.2: Principial scheme of heat exchanger (mixed flow type)

$\Delta tm = 0.5/(t_1 + t_2) - (t_3 - t_4)/$ (Equ. 8.5)

Counterflow exchanges as in Fig. 8.2) have a higher Δt value, parallel flow a lower value and mixed flow/crossflow types (e.g. vehicle radiators) have roughly the value calculated in Equ. 8.5.

When using the cooling air from an aircooled engine or air emerging from a standard engine radiator, these airflows shall in no way be subjected to resistance in the following heat exchanger nor directly connected to equipment, e.g. dryer. The original blowers are normally too weak to overcome any extra resistance. The cooling of the engine can therefore become insufficient and the engine overheats.

For water heating from exhaust gas a heat exchanger design with two concentric tubes is recommended and is easy to manufacture. For water heating from cooling water a heat exchanger in the form of a coil within a larger vessel or tank is suitable. The coil or other heat exchanger shall not be too long, too narrow or otherwise impose too much resistance to the cooling water flow as the normal water pump is not designed for much extra flow resistance. Here too the cooling of the engine may become insufficient. Larger heat exchangers should be insulated from outside as they emit heat to the surroundings which is lost for the initial purpose.

The operation of the heat exchanger, especially when using the engine's cooling air or water, needs a safety control to ensure that the cooling of the engine is always sufficient and the temperature of the cooling water returning to the engine is at around 50 °C with only minor fluctuations. The amount of heat conducted away from the engine has to match with the amount automatically produced in the engine according to its actual load. If more heat is produced than consumed, the control can be achieved using thermostats and a separate safety bypass cooler. If more heat is consumed from the cooling water than produced, the engine will gradually operate at too low temperatures which increases wear.

Taking too much heat from the exhaust gas reduces the final outlet temperatures and may lead to corrosion in the exhaust gas heat exchanger. The operation of the heat exchangers or heat users must therefore always consider that the heat production in the engine is linked directly to the mechanical power output, i.e. the driven machine's operation. Under the control aspect continuous operation of the whole system is therefore the best precondition for waste heat utilization.

The following example shall demonstrate the layout of a heat exchanger:

Given conditions:

- engine mechanical power: P = 15 kW
- engine efficiency: η_{eng} = 0.32
- cooling water temperatures:
- from engine: 80 °C, back to engine: 50°C
- exhaust gas temperatures:
- from engine: 350 °C,
- to surroundings: 180°C
- cold water temperature: tw = 20 °C

Problem: Supply of as much hot water at 60 °C as possible at constant rate.

Solution: Step 1:
Establish total fuel energy input:

$$E_{tot} = \frac{1}{\eta_{eng}} \cdot P = 47\mathrm{kW} = 47\mathrm{kJ/s} \text{ (see Equ. 3.9)}$$

Step 2:

Establish amount of heat to be obtained (see Equ. 8.2):

a) from cooling water
Qcw = 0.35 · 47 = 16.5 kJ/s

b) from exhaust gas
Qex = 0.15 · 47 = 7.1 kJ/s

Step 3:

Establish heat exchanger surface size (see Equ. 8.4):

a) cooling water heat exchanger

$$A_{CW} = \frac{Q_{CW}}{k \cdot \Delta t_m} = \frac{16.5 \cdot 3600^3}{2500 \cdot 25} = 0.95 m^2$$

with

$$- k = 2500 \cdot \frac{kJ}{m^2 \cdot h \cdot K}$$

- Δtm = 0.5/(80 + 50) - (20 + 60)/ = 25 K[4]

b) exhaust gas heat exchanger

$$A_{ex} = \frac{Q_{ex}}{k \cdot \Delta t_m} = \frac{7.1 \cdot 3600}{150 \cdot 225} = 0.76 m^2$$

with

$$- - k = 150 \cdot \frac{kJ}{m^2 \cdot h \cdot K}$$

- Δtm = 0.5/(350 + 180) - (20 + 60) /= 225 K

The exchanger areas can be materialized with a number of tubes in parallel to prevent excessively long exchangers.

Step 4:

Establish how much hot water is available (see Equ. 8.3):

a) from cooling water exchanger

$$m_{CW} = \frac{Q_{CW}}{c_p \cdot \Delta t} = \frac{16.5 \cdot 3600}{4.2 \cdot 40} = 354 kg$$

b) from exhaust gas

$$m_{ex} = \frac{Q_{ex}}{c_p \cdot \Delta t} = \frac{7.1 \cdot 3600}{4.2 \cdot 40} = 152 kg$$

with

c_p = 4.2 kj/(kg · k)

Δt= 60-20=40K
for the water circuit.

The two heat exchangers produce a total of 506 kg/in (or 1/h) of hot water. The parallel arrangement, i.e. cold water flows to both heat exchangers, allows for more flexibility in the water use:

- When water of higher temperature is wanted, the exhaust gas unit can supply it at a lower water flow rate.
- When less water is used the exhaust gas unit can be reduced to 0 l/h; only the final exhaust gas outlet temperature rises.
- When the warm water demand is further reduced, but the engine continues operation at the same load, part of the heated water from the engine's cooling water exchanger (not the cooling water itself) can be purged off to maintain engine cooling unless the cooling water cycle can be switched over to a bypass cycle with a standard engine radiator cum fan.

Ready-made cogeneration units for supply of heat and electricity are on the market in various sizes and versions. Heat exchangers can also be found in a large variety; some are even supplied for ready mounting to certain engine types. For both see Chapter 10.

9. Biogas for vehicles

The utilization of biogas in vehicles requires a method of compact storage to facilitate the independent movement of the vehicle for a reasonable time. Larger quantities of biogas can only be stored at small volumes under high pressure, e.g. 200 . . .300 bar, or purified as methane in a liquid form at cryogenic conditions, i.e. -161 °C and ambient pressure. The processing, storage and handling of compressed or liquified biogas demand special and costly efforts.

Compression is done in reciprocating gas compressors after filtering of H_2S. At a medium pressure of about l5 bar the CO_2 content can be "washed out" with water to reduce the final storage volume. Intermediate cooling and removal of the humidity in molecular sieve filters are essential as the storage containers should not be subjected to corrosion from inside. The storage cylinders, similar to oxygen cylinders known from gas welding units, can be used on the vehicle as "energy tank" and in larger numbers as refilling store.

One cylinder of 50 l volume can store at a pressure of 200 bar approximately

- 15 m³ unpurified biogas (CH_4 = 65% Vol) with an energy equivalent of 98 kWh or 101 diesel fuel, or
- 13 m³ purified biogas (CH_4 = 95% Vol) with an energy equivalent of 125 kWh or 12.51 diesel fuel.

The storage volume thus required on the vehicle is still five times more than is required for diesel fuel. Purification of biogas to CH_4 increases the storage efficiency by 25 . . . 30% but involves an extra gas washing column in the process.

Purified biogas, i.e. methane, has different combustion features than biogas because of the lack of the CO_2 content. It combusts faster and at higher temperatures; this requires different adjustments of ignition timing. Dual fuel methane engines are prone to increased problems with injector nozzle overheating and have to operate on higher portions of diesel fuel (about 40%) to effect sufficient cooling of the jets.

Liquification of biogas requires drying and purification to almost 100% CH_4 in one process and an additional cryogenic process to cool the CH_4 down to -161 °C where it condenses into its liquid form.

Storage is optimal at these conditions as the volume reduction is remarkable, i.e. 0.6 m³n with an energy content of 6 kWh condense to one lifer of liquid with an energy equivalent of 0.61 diesel fuel. The required tank volume is only 1.7 times the volume needed for diesel fuel.

This advantage is opposed by a more sophisticated multistage process, the handling of the liquid in specially designed cryo-tanks with vacuum insulation and the fact that for longer storage it has to be kept at its required low temperature in order to prevent evaporation. This requires additional energy and equipment. The practicability of such systems is still being researched with commuter bus traffic in Sao Paulo, Brazil. Data on the economic viability are not yet available.

The use of biogas as a fuel for tractors on farms has been elaborately researched. The processing of the gas does not only require about 10% of the energy content of the gas, mainly for compression, but also involves considerable investment. The tractor itself needs to carry four gas cylinders at least for a reasonable movement radius. A 40-kW tractor can then operate for about six to seven hours at mixed/medium load. The modification of the tractor has to include a three-stage pressure reduction system as the fuel gas is fed to the mixer at low pressure, i.e. about 50 mbar.

Modification into an Otto gas engine includes the risk of non-availability of the tractor at biogas shortage. It therefore needs LPG as spare fuel or another diesel tractor standby. Dual fuel tractor engines, on the other hand, are difficult to control, especially because of their frequent speed and load changes during operation in the field.

Biogas for road service has become an issue in Brazil lately. It must however be seen in connection with the specific situation in this country. The main issue is to utilize the large natural gas resources for substitution of diesel fuel which is scarce. Purified biogas is therefore integrated into a larger "methane program", for which the government may decide to give specific economic preferences because of energy-political reasons. The biogas will furthermore be obtained and processed in larger units, e.g. municipal sewage plants and sugar factories which reduces the cost per m^3 considerably.

With the current (political) price of fuel in industrialized countries the equivalent price for "vehicle biogas" is about two or three times higher than for diesel fuel. It is therefore presently not economic though technically feasible to use biogas in vehicles on a larger scale. The infrastructure for processing and filling however must also be developed accordingly.

10. Overview of Commercially Available Systems

10.1 Engines

Some manufacturers offer engines for the use of biogas, either diesel gas (dual fuel) or Otto types. Some manufacturers are listed below with specifications of their engines (as far as they were made available to the author) as well as some general comments.

In cases where more detailed specifications were given they will be reproduced in the appendix. Citation of manufacturers' names and specifications has only informative character and should not be regarded as advertising for any of the manufacturers.

(Data as per January 1987)

10.1.1 Smaller Engines up to 15 kW

10.1.1.1 Hirloskar Diesel Gas Engines, India

The Indian diesel engine manufacturers offer some of their standard diesel engines as diesel gas engines also.

- Basic engine type: 4-stroke diesel engine with direct injection.
- Type of modification: Addition of simple hand-controlled mixing chamber. Mounted directly to the inlet manifold. As no other modification is undertaken, the engine remains a fully functioning diesel engine with the option to utilize biogas for up to about 80% of its fuel requirements.
- Type of control: The control mechanism for diesel fuel operation is fully maintained. The biogas is manually controlled with a hand-operated valve at the biogas inlet to the mixing chamber.

Rough control of power and speed is achieved by setting the required speed at the governor lever and then opening the biogas valve to admit the allowable amount of biogas to the mixing chamber. Fine control, if not done manually by an operator, can be taken over by variations of the diesel fuel amount through the governor/ injector.

The amount of biogas admitted must however be less than the maximum possible 80% (see Chapter 5.1.4).

According to information of the company the diesel fuel control is equipped with a minimum fuel adjustment to ensure that sufficient pilot fuel is always supplied. Problems with injector nozzle overheating are said to be unknown.

Engine specification of type TV I G

(Information on other types is available from the manufacturer.)

Bore (mm)	87.5
Stroke (mm)	110
Max. speed (rpm)	2000
Min. idle speed (rpm)	750
Min. operating speed (rpm)	1200
Cooling system	water-cooled

B.H.P. (at 2000 rpm)	
B.S. 649: 1958 continuous	8.7
S.F.C. at full load when fully on diesel(g/bhp-h)	176.00
S.F.C. at full load of ignition diesel when on-dual fuel (g/bhp-h)	30
Biogas requirement for dual fuel (m^3/bhp-h) (15 cubic feet/bhp-h)	0.425

Biogas specification: The data are based on biogas with

- methane content: 60% Vol
- pressure: 100 mbar (± 50 mbar)
- H_2S content: no data but a value of 0.2% Vol should not be exceeded.

Engine/machine units readily available:

- engines cum electric generator, various types
- engines cum water pumps, various types.

Comments: Kirloskar diesel engines, some of which are based on models of other international manufacturers under license, are widespread in India where they apparently enjoy a goodreputation and a considerable share in the market.

Manufacturer's address:
Kirloskar Oil Engines Ltd.
Luxmanrao Kirloskar Rd.
Khadki, Pune 411003
India

10.1.1.2 Shanghai Bioenergy Engineering Co., China

The Chinese company offers a biogas Otto engine obviously modified from a standard single-cylinder diesel engine.

- Basic engine type: 4-stroke, single-cylinder Otto engine.
- Type of modification: Basic diesel singlecylinder engine modified and equipped with gas mixer (venturi), spark ignition, alternator, etc. (see Chapter 5.3).
- Type of control: no direct information available but obviously mechanical governor acting on venturi throttle valve as engine is designed to work with a generator at constant speed.

Engine specifications (as far as given in manufacturer's leaflet)

- -model: S 195 DZ
- -type: 4-stroke, horizontal, watercooled

- number of cylinder(s):	1
- displaced volume(l):	0.82
- speed (1/min):	2000
- power (kW): approx	6

Biogas specifications:

- methane content min.: 70% Vol
- hydrogen content max.: 5% Vol

Engine/machine units readily available:
- engine cum electric generator with Pel = 5 kW

U=220/380V, F=50 Hz
Comments: no further information available.

Manufacturer's address:
Shanghai Bioenergy Engineering Co.
P.O. Box
Shanghai
People's Republic of China

10.1.1.3 Montgomery/Yanmar, Brazil

The Brazilian manufacturer offers smaller Otto engines for generators, pumps, etc. They are built for various fuels such as petrol, alcohol, kerosene and biogas.

- Basic engine type: 4-stroke, single-cylinder Otto engine with standing valves
- Type of modification: The housing of the original carburetor for petrol/alcohol is used to form a venturi type of mixing chamber.

Biogas is introduced through a nozzle pointing into the core of the venturi bottleneck. Timing of valves, ignition and the compression ratio remain unchanged from the basic petrol version and this version is therefore not optimal for biogas operation.

-Type of control: mechanical speed governor with possibility for manual setting, acts on carburetor throttle valve.

Engine specifications:

models:	R-137	R-320	R-480
type:	4-strocke, vertical,	air-cooled	(for all models)
number of cylinder(s):	1	1	1
displaced volume (1):	0.14	0.32	0.48
speed range:	n=1800.3600 1/min	(for all models)	
power (kW at n = 3 600 1/min):			
(approximately)	1.5	4.0	6.5

Biogas specifications:

methane content:	60 . . .70% Vol
pressure:	120 . . . 180 mbar

Engine/machine units readily available:

- Engine types R-137 and R-320 with electric generators 0.9 kW and 2.5 kW respectively
- engine types R-137 and R-320 with centrifugal pumps
- engine types R-320 and R-480 with self-priming pumps

Comments: the company has so far only sold about 300 units for biogas, mainly in Brazil. According to the company's own statement the experience is not always positive and some further developments would be required. The engine as such is of an outdated design (standing valves, excentric compression chamber), the compression ratio (about 7: 1) very low for biogas. The reliability of the magnetic ignition system could be improved. The engines, as they are designed presently, do not appear to be recommendable for continuous service but might be useful for occasional operation.

Manufacturer's address:

Cia. Yanmar
Av. Dr. Gastao Vidigal, 2001
Cx. Postal 542
Sao Paulo Bazil

In principle other small direct injection diesel engines can be used with simple modification (mixing chamber) for biogas dual fuel operation even though other manufacturers do not specifically offer biogas versions. The same is true for small Otto engines as long as they can operate on unleaded petrol. They require either a modification of their carburetor, or an adapted venturi/mixing valve. Their performance with biogas will however be significantly (up to 40%) lower than with petrol.

10.1.2 Larger Engines

10.1.2.1 C.A.S./Henkelhause-Deutz, Federal Republic of Germany

The German company specializes in the commercial modification of standard "Deutz" diesel engines into Otto gas engines for natural gas and biogases. Their air-cooled engine series range from 15 kW to 144 kW. The water-cooled series from 122 kW to 500 kW are not included here; specifcations may be obtained from the company.

- Basic engine type: 4-stroke Otto gas engine in "Deutz" module design, air-cooled. Type of modification:
- Engine block cum cylinders, crank- and camshaft, cooling and lubrication system are retained from the diesel engine version.
- Low compression (e = 11.5) pistons, cylinder heads with provision for spark plugs are mounted in exchange for the original parts of the diesel version.
- Ignition system with distributor connected via angular gear to the drive for the obsolete injector pump, ignition coil and 24-V alternator cum batteries are added.
- A suction-pressure-controlled air/gas mixing valve with butterfly throttle for the control is connected to the original air inlet manifold.
- A gas inlet control system with filter, shut-off solenoid valve, constant pressure regulation valve and suction gas pressure manometer is added.
- A separate large lubrication oil tank for extended oil exchange intervals is added.

Type of control: The company offers two types of control according to the future operation of the engine:

- mechanical control, using the original governor for the diesel injector pump. The motion of the control rack is passed on to the butterfly throttle of the gas mixing valve. The precision ("droop") is about 5 . . . 8% of the speed set at the governor lever. The mechanical control is sufficient for less sensitive isolated grids and for the drive of pumps or other machines;
- electronic control, using a magnetic pick-up to sense the engine speed from the flywheel ring gear. The speed pulse signal is transmitted to the electronic control box where the actual and the desired speeds are compared. The correcting pulse is given to the actuator which via a linkage moves the throttle in the gas mixing valve. The precision ("droop") of this unit is less than 1%, hence considerably better than the mechanical unit. The control system is made by Barber Colman, USA.

Safety devices like cut-out at overspeed, low oil pressure, low gas pressure and at too high a temperature are part of the system.

Engine and biogas specifications of naturally aspirated 4-cylinder engine (F 4 L 912): (Information on other types is available from the manufacturer.)

Rated power for continuous operation (10% overload, DIN 6271, 27 °C, 100 m above sea level,

60% ref. Humidity)	28 kW
Speed	1500 rpm
No. of cylinders	4
Cylinder arrangement in line	
Bore/stroke	100/120 mm
Swept volume	3.77 l
Mean effective pressure at rated power	5.90 bar
Mean piston speed 6.00 m/s	
Fuel consumption at full load:	
- natural gas	8.8 Nm³/h
- biogas	12.6 Nm³/h
Gas pressure	13-20 mbar
Max. Permissible mm H2S 0.15%	
Lube oil consumption 60 g/h	
Lube oil capacity	10.1 l
Direction of engine rotation anticlockwise	
Heat quantities:	
- radiation heat of engine 4.0kW	
- heat quantity to be dissipated in cooling air	31.1 kW
- heat quantity to be dissipated in exhaust gas	23.5 kW
Exhaust quantity (referred to 20 °C)	160 m³/h
Exhaust temperature	460 °C
Air requirement/hour for	
- cooling air	1810 m³/h
- combustion air (20 °C) 133 m³/h	
Quantity of used air (70 °C) 2234 m³/h	
Engine dimensions:	
- length	813 mm
- width	661 mm
- height	803 mm
Weight	300 kg
Noise (measured at 1-m distance)	91 dB(A)
Power when using	
- refuse dump gas	26 kW
- lean gas	24 kW

Engine machine units readily available:

- engines cum electric generator in accordance with customer's planned operation and specification.

Comments Henkelhausen/Deutz gas Otto engines enjoy a good reputation for reliability and a high service factor. Over two hundred machines are in the field, the majority for electricity generation, hence in continuous service. Biogases from waste water treatment plants, waste disposal fields and producer gas are used in about 60% of the engines; natural gas is used in about 40%.

Manufacturer's address:
G.A.S., Henkelhausen/Deutz
Hafenstrasse 5 1
D-4150 Krefeld 12
Federal Republic of Germany

10.1.2.2 Deutz MWM, Federal Republic of Germany

After the recent merger of the two large diesel and gas engine manufacturers Deutz and MWM the group offers a large range of gas engines in diesel gas (dual fuel) and gas Otto versions, from about 20 kW to 3 000 kW. The gas engines are based on standard stationary diesel engines. Specifications for the larger series with a power of more than 100 kW are not included here but can be obtained from the manufacturers.

- Basic engine type: 4-stroke Otto gas engine, vertical in line and V-type, watercooled.
- Type of modification: basically as described in Chapter 10.1.2.1 above.
- Type of control: electronic control, similar to the system described for 10.1.2.1. Manufacturer of the control system is Bosch, Federal Republic of Germany.

Engine specifications:

models:G 227-3 G 227-4 G 227-6 G 232 V6 G 232 V8
types:4-stroke Otto gas engines, water-cooled (for all models)

	vertical in line			V-type	
number of cylinders:	3	4	6	6	8
displaced volume (1):	2.83	3.77	5.65	8.8	11.8
compression ratio:	11.6:1 (for all models)				
speed(1/min):	1500/1800 (for all models)				
power (Pel in kW):	18/21	24/28	36/43	65/77	87/103

Biogas specifications:

- methane content min.: 65% Vol
- H_2S content max.: 0.1%Vol

Engine/machine units readily available:

- engine cum electric generator and heat cogeneration according to specifications of the customer
- engine cum pump or blower (for aeration ponds in waste water treatment plants)

Comments: The company has sold several thousand gas engines for natural gases and biogases worldwide. Their engines have a good reputation for reliability and a high service factor. The majority of the driven machines are electric generators, furthermore blowers, pumps and heat pumps, all in continuous service. The greater part of the group's activities lies however in the larger power range.

Manufacturer's address:
MWM Motorenwerke Mannheim
Carl-Benz-Str.
D-6800 Mannheim
Federal Republic of Germany

KHD Klöckner Humboldt Deutz A.G.
P.O. Box 800 509
D-5000 Köln-Deutz
Federal Republic of Germany

10.1.2.3 Volkswagen (VW) do Brasil, Brazil

The Brazilian VW affiliate has developed Otto engines based on standard vehicle engines for operation with alcohol as a consequence of the Brazilian national alcohol program. Due to the increased compression ratio (e = 12) these engines are suitable for biogas operation. About 50 engines are running in vehicles with purified and compressed biogas; a few others produce electricity in stationary application.

- Basic engine type: 4-stroke Oto gas engines, vertical in line, V-type and opposed arrangement ("boxer") type.
- Type of modification: increase of compression ratio to about 12: 1 with smaller variations according to the engine type. Exchange of (alcohol) carburetor for venturi gas mixer or addition of a simple venturi onto the existing carburetor with possibility of switching back to liquid fuel.

Type of control:

- in vehicles: butterfly throttle is connected to driver's pedal as common in vehicles;
- mechanical: a separate mechanical governor (Franker) is driven by the normal V-belt and acts on the throttle valve of the gas mixer/carburetor. Precision is said to be ± 5% of set speed.

Engine specification:

models:	318-3F	1600 (boxer)	1600 (in line)
types: 4-stroke Otto gas engine	V-type	boxer type	in-line type
number of cylinders:	8	4	4
displaced volume (1):	5.2	1.6	1.6
compression ratio:	9.5: 1	10:1	10: 1
speed, max. (1/min):	4 000	4 000	4 000
power, max., short term (kW):	100	30	40

Biogas specification: purified biogas or natural gas with methane content between 90% and 100% and a calorific value between 32 400 and 36 000 kJ/Nm³; H2S content about zero because of intensive filtering.

Engine/machine units readily available:

- small pick-up vehicle ("Saveiro" type) with 1.6-1 engine,
- light truck (Type 6 - 140) with 5.2-1 engine,
- combi with 1.6-1 boxer engine.

Comments: The basic engines are taken from standard Otto engine series and are of proven quality. The versions for methane/ purified biogas have only recently been developed but have performed satisfactorily so far. As the engines are mainly projected for use in vehicles the speed and power output are relatively high while engine life will be around 3 000 hours. For continuous service and longer engine life speed and power will have to be reduced to 50% which the company decided to do in one larger stationary application. Engine versions for direct use of untreated biogas are so far not offered by the company but one might consider using them on an individual basis. The power will of course be further lowered according to the actual calorific value of the untreated biogas.

Manufacturer's address:
Volkswagen do Brasil S.A.
09700 Sao Bernardo do Campo
Sao Paulo
Brazil

10.1.2.4 Ford Motor Company, Federal Republic of Germany

The German Ford affiliate offers Otto gas engines based on their standard vehicle engines.

- Basic engine type: 4-stroke Otto gas engines, vertical in-line type, water-cooled
- Type of modification: increase of compression ratio to 11: 1 for natural gases and biogas. (For LPG compression ratio remains at 8: 1 as for petrol versions.) Exchange of carburetor against gas mixing valve (Impco).

Type of control:

- mechanical: a separately mounted governor acting onto the butterfly throttle of the mixing valve;
- electronic: as described earlier under 10.1.2.1.

Engine specifications:

models:	2274 HC	Dovergas S.1.4	Dovergas S.1.6
type: 4-stroke vertical in-line Otto gas engine (for all models)			
number of cylinders:	4	4	6
displaced volume (1):	1.6	4.15	6.22
compression ratio (natural gas/LPG):	11:1/8:1 (for all models)		
speed(min^{-1}):	1500/3000	1500	1500
power (Pel in kW):	12/24	33	50

(Bio)gas specification:

- standard LPG, natural gas (CH_4 > 90%)
- for biogas no data available.

Engine/machine units readily available: The manufacturer supplies the engine alone while other engineering companies use the Ford engines to offer engines cum electric generators and heat cogeneration.

Comments: The basic engines are taken from standard mass production series and are of proven quality' also for stationary purposes. The majority of the engines delivered so far work on LPG and natural gas; for the use of biogas the power data given will have to be reduced by about 30%.

Manufacturer's address:
Ford Werke A.G.
Edsel Ford-Strasse
D-5000 Cologne 71
Federal Republic of Germany

In other countries refer to the local Ford representative.

10.1.2.5 Peugeot, France

The French automobile manufacturers offer gas versions of their standard vehicle engines.

- Basic engine type: 4-stroke Otto gas engine, vertical in-line and V-type, water-cooled.
- Type of modification: increase of compression ratio, exchange of carburetor against gas mixing valve.
- Type of control: electronic, similar to system described under 10.1.2.1.

Engine specifications:

models:	2 E 1A	X N 1 P	ZN 175
type:	4-stroke	Otto in line	4-stroke Otto V-type
number of cylinders:	4	4	6
displaced volume (1):	1.1	2.0	2.8
compression ratio:	9.6:1	8.8:1	9.5:1
speed (1/min):	3000 3000	3000	
power (Pel in kW):	20	32	48

(Bio)gas specifications:

- standard LPG and natural gas
- biogas, methane content min.: 60% Vol H_2S content max.: 0.5% Vol

Engine/machine units readily available:

- engine cum electric generator with cogeneration using the engine's waste heat.

Comments: The basic engines are taken from standard mass production series and are of proven quality. The relatively low compression ratios, while suitable for LPG, will result in a higher fuel consumption for methane gases compared to higher compressed engines. Power data given may have to be reduced by about 30% for operation with biogas.

Supplier's address (in the Federal Republic of Germany): Peugeot Motoren GmbH
Bonner Ring 17
D-5042 Erftstadt-Lechenich
Federal Republic of-Germany

In other countries refer to the local Peugeot representative.

10.2 Engine modification kits, other accessories

10.2.1 Impco Gas/Air Mixing Valves, USA

The US company has long been offering carburetion kits for the modification of Otto petrol engines into Otto gas engines. Such kits include suction pressure-controlled mixing valves with butterfly throttles, constant pressure reduction valves, adapters for manifolds and other ancillaries. Elaborate equipment for alternative dual fuel (gas/ petrol, not diesel) is available but is rather geared to vehicle use.

Originally designed for the use of LPG in vehicles, the equipment also functions for natural gas with a calorific value of not less than 37 000 kJ/m³. For gases with lower calorific values, such as biogas, special valve types can be supplied.

The gas mixing valves are offered in a variety for engines from about 10 kW up to about 500 kW. The normal gas pressure at inlet to the mixer should range from 20 . . . 50 mbar which in some cases is too high for Gobar gas-type biogas plants.

Comments: The standard Impco mixer types for high calorific value gases may produce a mixture with biogas which is too lean (λ 1.3) for satisfactory performance even at fully opened gas adjustment throttle. A compensation by throttling the airflow externally before the mixer may result in a better excess air ratio value but lowers the performance and efficiency of the Otto engine because of extra reduced filling. It therefore appears recommendable to only utilize the specifically designed digester gas ("DG") types for biogases with a methane content of 60 . . . 90%. A self-modification of the valve cone in the mixing zone of the valve is not recommended.

Manufacturer's address:
Impco Carburetion Inc.
16916 Gridley Place
Cerritos, CA. 90701
USA

10.2.2 Rodagas,Brazil

The Brazilian company has developed modification kits for vehicle and stationary engines, both for Otto gas and diesel gas (dual fuel) engines. The equipment is based on kits for LPG but was also further adapted to the use of purified biogas and natural gas. The mixers are based on the venturi principle which allows adaptation to the actual calorific value of the gas more easily than in mixing ,valves as explained earlier. Other equipment like constant pressure reduction valves and ancillaries for the modification of diesel engines (dual fuel) are also available. The company participates in the Brazilian research program for the extended use of methane gases in small and large vehicles.

The various kits offered include modifi- cation kits for existing carburetors, mixers to be mounted onto carburetors, mixers with butterfly throttles in exchange for carburetors and all necessary control and safety accessories.

Comments: Even though the equipment is mainly designed for the use of compressed methane in vehicles the mixers are well suited for direct use of biogas in stationary engines also. An existing gas adjustment throttle and the possibility of exchanging or modifying the separate venturi ring (widen holes) offer the possibility to adapt the mixers to low calorific biogases.

Manufacturer's address:
Rodagás
Rua Campante 713/721
(04224 Ipiranga)
Sao Paulo
Brazil

10.2.3 Kromschröder Gas Handling Accessories, Federal Republic of Germany

The company supplies a variety of equipment for handling of different types of gases. In the field of biogas as a fuel for engines the following selection from their program is particularly useful:

Filters, gas governors, ball valves, safety valves, butterfly valves, magnetic relief valves, pressure switches, flow meters, pressure gauges, fittings and others.

The equipment meets DIN and other international standards. Most of the equipment is resistant to H_2S corrosion, but a gas specification should be sent with enquiries.

Comments: Many suppliers of biogas engines use Kromschröder equipment for their gas preparation and control and have expressed their satisfaction with the quality of the products.

Manufacturer's address:
Kromschröder A.G.
P.O. Box 2809
D-4500 Osnabrück
Federal Republic of Germany

10.3 Other equipment

10.3.1 Barber Colman Electronic Control System, USA

The company offers complete electronic control systems with speed pick-up, electronic control box and actuator. Speed precision is very high, with tolerances as low as 0%. The control system is suitable for grid parallel operation of engine-generator sets.
Comments: The system is widely used in cases where electricity generation with a high degree of frequency stability is required. Under difficult conditions in terms of service and spare part availability, however, a mechanical governor should be preferred if the operation allows slight engine speed fluctuations.

Manufacturer's address:
Barber Colman Company
Precision Dynamics Division
1354, Clifford Ave.
P.O.Box 2940
Loves Park, IL.61132-2940
USA

10.3.2 Fiat Totem, Italy

The company supplies compact cogeneratior units in a standard module version for 15 kW electric power and a heat supply of about 30 kW. The unit is equipped with automatic control and suitable for isolated and grid parallel operation. The cost of the system (about US$ 1 500) is only justified when heat and power are fully utilized. The engine i based on a standard Fiat (127) Otto vehicle type engine. The company has supplied unit for natural gas and biogas. The system can be supplied through the local Fiat representative.

10.3.3 Communa Metall, Federal Republic of Germany

The company supplies cogeneration units in standard module versions in a range from 6 . . . 65 kW electric power combined with 15 . . . 90 kW of heat. The units are equipped with automatic control for isolated and grid parallel operation. The engines are Ford standard types, modified for natural gas and biogas. A high degree of utilization of both power and heat is necessary for economic operation.

Manufacturer's address:
Communa Metall Uhlandstr. 17
D-4900 Herford I Federal
Republic of Germany

10.3.4 Sauer und Sohn, Federal Republic of Germany

The company supplies heat pump and cogeneration units driven by gas engines, based on Ford standard engines as described earlier. It also offers ready-made heat exchangers for waste heat utilization of engines, some of which are designed for straight application to the Ford engine model series 2270, 2700, 2710, 2720.

Manufacturer's address:
Sauer und Sohn
P.O. Box 1240
D-6110 Dieburg
Federal Republic of Germany

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Appendix I

Symbols and abreviation		
Symbol	**Name**	**Unit**
A	Ampere (unit for current)	
A	area	m², mm²
Ag	cross-sectional area of gas supply pipe	m², mm²
Av	cross-sectional area of venturi contractor ("bottleneck")	m², mm²
α	angle, crank angle	°
α_i	crank angle at ignition	° before TDC
AC	alternating current	
bar	pressure unit	
BDC	bottom dead center, piston's lowest position	
BG, bg	biogas	
c	velocity	m/s
c_i	intake velocity	m/s
c_v	velocity at venturi bottleneck	m/s
c_g	velocity through gas nozzle	m/s
°C	degree Celsius (temperature unit)	
c_p	specific heat at constant pressure	kJ/(kg K)
c_v	specific heat at constant volume	kJ/(kg K)
D, d	diameter	m, mm
D_{eng}	diameter of engine pulley	m, mm
D_{mach}	diameter of machine pulley	m, mm
d	day (time unit)	
d;	inlet diameter	mm
dg	gas nozzle diameter	mm
dv	venturi contraction diameter	mm
Ap	pressure difference	mbar
ΔV	volume to be added	cm³
DC	direct current	
E	energy	J, kJ, kWh
E	energy flow	flow J/s, kJ/s, kW
E_f	fuel energy	J, kJ, kWh
E_f	fuel energy flow	J/s, kJ/s, kW
η	efficiency	
η_b	boiler efficiency	
η_c	cogeneration efficiency	
η_{eng}	engine efficiency	
η_g	generator efficiency	
η_{mach}	driven machine's efficiency	
η_{mech}	mechanical efficiency	
η_p	pump efficiency	
η_t	transmission efficiency	
f	frequency	l/s, Hz
f_c	fuel consumption	1/h, m³/s, m³/h
f_{cd}	Diesel fuel consumption	l/h
g	gram (unit for mass)	
g	gravity constant	9.81 m/s2
γ	isentropic exponent (= cp/cv)	
h	hour (time unit)	

H	height, water head	m
HCV	hand-controlled valve	
I	current	A
i	number of cylinders	
J	Joule (energy unit)	
k	heat transfer coefficient	kJ/(m² · h · K)
K	Kelvin (temperature unit)	
kg	kilogram (unit for mass)	
kJ	kilojoule (energy unit)	
kW	kilowatt (power unit, energy flow unit)	
l	liter (volume unit)	
λ	excess air ratio in air/fuel mixture	
m, mm	meter, millimeter (length unit)	
m	mass	kg
m	mass flow	kg/s, kg/in
m_f	mass flow of fuel	kg/s, kg/in
m³ n	cubic meter at standard conditions	
mbar	milibar (pressure unit)	
min	minute (time unit)	
n	polytropic exponent	
n	shaft (rotational) speed	min^{-1}
n_r	rated shaft speed	min^{-1}
n_{eng}	speed of engine	min^{-1}
n_{mach}	speed of driven machine	min^{-1}
P	power	kW
P_{el}	electric power	kW, kVA
P_{gen}	power of generator	kW
P_{mach}	power demand of driven machine	kW
P_{mech}	mechanical power	kW
P_p	number of pole pairs	
p	pressure	bar, Pa, mm WH
P_a	ambient pressure	bar, mbar
P_c	pressure after compression	bar
p_p	biogas plant pressure	bar; mbar
p_s	section pressure	bar
P_a	Pascal (pressure unit)	
ppm	parts per million (volume unit)	
Δp	pressure difference	bar
Q	heat	kJ
Q	heat flow, heat transferred	kW, kJ/s
Q	pump capacity (volume flow rate)	m³/s, m³/h
R	specific gas constant	kJ/(kg K)
r	"rated" (design conditions)	
ρ	density	kg/m³, kg/1
ρ_w	density of water	1000 kg/m³
s	second (time unit)	
sfc	specific fuel consumption m³/kWh,l/kWh,kWh/kWh	
sgp	specific gas production	m³ga/(d* Vp)
t	Celsius temperature	°C
Δ t	temperature difference	°C, K
Δtm	mean temperature difference	°C, K
T	absolute temperature	- K

T_c	absolute temperature after compression	K
T_s	suction temperature	K
TDC	top dead center, highest position of piston	
t_o	time of operation	h
U	voltage	V
V	Volt (unit for "tension", voltage)	
V	volume	m³, 1, cm³
V_c	compression volume	1 cm³
V_{dc}	displaced volume of one cylinder	1 cm³
V_{de}	displaced volume of engine	1, cm³
V_p	volume of biogas plant	m³
V_{prev}	previous volume of compression chamber	cm³
V_{new}	new volume of compression chamber	cm³
V_s	storage volume for biogas	m³
V_{tot}	total volume of cylinder	l, cm³
V	volume flow rate	m³/s, m³/h, 1/h
V_{bg}	biogas volume flow rate	m³/s, m³/h
V_w	water volume flow rate	m³/s, m³/h
ΔV	volume to be added	cm³
WH	water head	m

Appendix II

Tables, Conversion Factors

Properties of various fuels

fuel	density	calorific value (kJ/kg)	ignitability	ignition temperature in air (°C)	stoichiometric air/fuel ratio (kg/kg)	methane no.
			(Vol % gas in air)			
methane	0.72	50000	5.0...15.0	650	17.2	100
	kg/m³n					
LPG	0.54 kg/l	46 000	2.0 . . .9.0	400	15.5	30
propane	2.02	46 300	2.0. . .9.5	470	15.6	35
	kg/m³ n					
butane	2.70	45 600	1.5 . . .8.5	365	15.6	10
	kg/m³ n					
petrol	0.75 kg/1	43 000	0.6 . . .8.0	220	14.8	-
diesel	0.85 kg/1	42500	0.6 . . .8.5	220	14.5	-
natural	0.83	57500	5.0 . . . 17.0	600	17.0	80
gas	kg/m³ n					
biogas[1]	1.2	18000	5.0. . . 15.0	650	10.2	130
(60% CH_4)	kg/m³ n					

[1]) H_2S content should be at 0.15 Vol % (1500 ppm), but never more than 0.5 Vol % (5000 ppm).

Other useful correlations

- Calorific value of biogas by methane content
100% CH_4: Hu = 36 000 kJ/m³ n = 10 kWh/m³ n
each 10% of CH_4 content in biogas: Hu = 3600 kJ/m³ n = 1 kWh/m³ n

Example:
65% CH_4: Hu = 23 400 kJ/m³n = 6.5 kWh/m³n

- Energy equivalents of biogas

1 kWh biogas =0.1 l diesel fuel =0.11 l petrol
1 m³ n biogas =0.6 l diesel fuel =0.67 l petrol
1 m³ n biogas = 1.5 kWh mechanical energy= 1.3 kWh electrical energy

- Pilot fuel requirement for diesel gas engine (at 20% of consumption in diesel fuel mode): 0.06 1/kWh
- Change (decrease) of engine performance with ambient conditions
- location altitude approx. 1% each 100 m above sea level
- pressure approx. 1% each 10 mbar below design conditions
- temperature approx. 1% each 5 °C above 20 °C
- rel. humidity approx. 2% each 10% above 65%
- Change of ambient pressure with location altitude approx. 10 mbar each 100 m

Metric conversion table:

Energy				
	kcal	**kWh**	**kJ**	**kNm**
kcal	1	$1.163*10^{-3}$	4.187	4.187
kWh	860	1	3600	3600
kJ	0.239	0.278	1	1
kNm	0.239	0.278	1	1

Pressure				
	PA	**bar**	**m WG**	**N/m²**
PA	1	10^{-5}	10_{-4}	1
bar	10^5	1	10	10^5
mWG	10^4	0.1		10^4
N/m²	1	10_{-5}	10^{-4}	1

Power				
	kcal/h	**kW**	**kJ/h**	**HP***
kcal/h	1	$1.163*10^{-3}$	4.187	$1.6*10^{-3}$
kW	860	1	$3.6*10^3$	1.36
kJ/h	0.239	$0.27*10^{-3}$	1	$0.38*10^{-3}$
HP*	633	$2.65*10^3$	0.736	1
* 1 HP = 745.70 W, HP metric = 735.49875 W				

Factor	Unit	Equivalents
Length:	**m**	**1 m= 0.001 km**
	= 100 cm = 1000 mm	
Volume:	m³	1 m³= 1000 l
Time:	s	1 min= 60 s,
	1 h = 3600 s,	
	1 d= 24 h,	
	1a=365 d	
Temperature: K	0 ºC = 273 K	
	T [K] = 273 [K] + °C	

Conversion of SI units into British/American units

Combined measures can be converted by inserting the appropriate conversion factors into the original expression, e.g.:

BTU/h ft^2 F = 1055/(3600*0.0929*5/9) = 5 678 W/m² K

SI (metric) to Brit./Amer.		Brit./Amer. to SI (metric)	
Length			
1 cm	= 0.3937 in (inch)	1in	= 2.5400 cm
1 m	= 3.2808 ft	1ft	=12in= 0.3048 m
	= 1.0936 yards	1 yard	= 3ft= 0.9144 m
1 km	= 0.6214 mile (statute)	1 mile (statute)	= 1.60934 km
Area			
1 cm²	= 0.1550 sq in	1 sq in	= 6.4516 cm²
1 m²	= 10.7639 sq ft		= 0.000645 m²
	= 1.1960 sq yards	1 sq ft	= 0.0929 m²
1 ha	= 2.471 acres	1 sq yard	= 9 sq ft = 0.836 m²
	= 10000 m²	1 sq mile	= 2.590 km²
		1acre	= 0.4047 ha
Volume			
1 cm³	= 0.06102 cu in	1 cu in	= 16.3870 cm³
1 dm³	= 61.024 cu in		= 0.01639 dm³
1 l	= 0.03531 cu ft	1 cu ft	= 28.317 dm³
	= 61.026 cu in	1 cu yard	= 0.7646 m³
	= 0.21998 gal (Brit.)	1 gal (Brit.)	= 4.546 l

	= 0.26428 gal (Am.)	1 gal (Am.)	= 3.785 l
1 m³	= 35.315 cu ft	1 quarter (Brit.)	= 64 gal
	= 1.308 cu yards	.	= 290.9 l
	= 6.299 Petr. barrels	1 Petr. barrel	= 0.15876 m³
			=42gal
1 quart (Am.)	= 2 pints		
			= 0.946 dm³
1 bushel (Am.)	= 35.2421		
1 bushel (Brit.)	= 36.37 1 = 8 gal		
1 Nm³	= 37.97 cu ft	1 cu ft	= 0.02635 Nm³
	(60°F, 30 in moist)	(60°F, 30 in moist)	
1 Nm³	= 37.22 cu ft	1 cu ft	= 0.02687 Nm³
	(60°F, 30 in dry)	(60°F, 30 in dry)	
Weight, mass, density			
1 g	= 0.03527 oz (av)*	1 grain	= 0.0648 g
	= 15.432 grain	1 oz (av)*	= 28.35 g
1 kg	= 2.2046 lb (av)*	1 lb (av)*	= 16 oz
	= 0.0787 quarter (Brit.)		0.4536 kg
			= 7 000 grains
1 t	= 0.984 long tons	1 quarter (Brit.)	= 28 lb = 12.701 kg
	= 1.102 short tons	1 long ton (Brit.)	= 1016 kg
1 kg/m³	= 0.06243 lb/cu ft	1 short ton (Am.)	= 2000 lb = 907.2 kg
		1 lb/cu ft	= 16.0185 kg/m³
1 g/kg	= 7.0 grain/lb	1 grain/lb	= 0.1426 g/kg
I g/m³	= 0.437 grain/cu ft	1 grain/cu ft	= 2.2884 g/m³
1 g/m³	= 2.855 ton/sq mile	1 ton/sq mile	= 0.3503 g/m³
1 m³/hm³	= 0.0547 cbf/sq ft	1 cfm/sq ft	= 18.3 m³/hm³

* Avoirdupois (av), the generally accepted series of weight units based on a pound of 16 ounces and an ounce of 16 drams, as opposed to the troy system based on a pound of 12 ounces and an ounce of 20 pennyweights or 480 grains.

Velocity and flow			
1 m/s	= 196.85 ft/min	1 ft/min	= 0.508 cm/s
1 km/in	= 0.6214 mph	I mph	= 1.60934 km/in
1 Kn	= 1.852 km/in	1 km/h	= 0.54 Kn
	= 0.514m/s		= 0.278m/s
1 m³/h	= 4.403 gal/min (Am.)	1 gal/min (Am.)	= 0.227 m³/h
	= 3.666 gal/min (Brit.)	1 gal/min (Brit.)	= 0.273 m³/h
1 m³/h	= 0.5886 cu ft/min	1 cu ft/min	= 28 317 1/min
		= 1.700 m³/h	
1 kg/in	= 0.0367 lb/min	1 lb/min	= 27.216 kg/in
Power			
1 W (Watt)	= 3.412 BTU/h	1 BTU/h	= 0.2931 W
1 kW	= 3412 BTU/h	1 HP	= 0.7457 kW
Enthalpy and entropy			
1 kJ/m³	= 0.02684 BTU/ft3	1 BTU/ft3	= 37.26 kJ/m³
1 kJ/kg	= 0.43021 BTU/lb	1 BTU/lb	= 2.3244 kJ/kg
1 kJ/K	=0.5266 BTU/F	1 BTU/F	= 1.899 kJ/K
Pressure and force			
1 N (Newton)	= 0.2248 lb (f)	1 lb (force)	= 4.448 N
1 N/m²	= 0.0209 lb/ft2	1 lb/in (psi)	= 6895 N/in2
(Pascal)			= 68.95 mbar

			= 703.1 mm H_2O
1 bar	= 14.504 psi	1 lb/ft2	= 47.88 N/m²
	= 29.530 in Hg		= 0.4788 mbar
	= 0.987 atm		= 0.0470 mm H_2O
1 mbar	= 0.0145 psi	1 in H_2O	= 249.08 N/m²
	= 0.0295 in Hg		= 2.4908 mbar
	= 0.4019 in H_2O		= 25.4 mm H_2O
	= 2.089 lb/ft2	1 in Hg	= 33.864 mbar
1 mm H_2O	= 0.0394 in H_2O	1 ft H_2O	= 29.89 mbar
1 atm	= 14.696 psi	1 atm	= 1.013 bar
1 mm H_2O/m	= 1.1993 in H_2O/100 ft	1 ft H_2O/100 ft	= 98.10 N/m²*m
1 N/m² m	= 0.1223 in H_2O/100 ft	1 in H_2O/100 ft	= 8.176 N/m²*m
1 mbar/m	= 0.442 psi/100 ft	1 psi/100 ft	= 2.262 mbar/m
Energy			
1 J (Joule)	= 0.948 10-3 BTU	1 BTU	= 1.055 kJ
1 kJ	= 0.948 BTU	1 ft lb (force)	= 1.356 J
1 kWh	= 3414.5 BTU	1 HPh	= 2685 kJ
1 MWh	= 34.1297 therms	1 therm	= 0.1055 GJ
		(100 000 BTU)	(29.288 kWh)
Specific heat			
1 kJ/kgK	= 0.2388 BTU/lb F	1 BTU/lb F	= 4.187 kJ/kgK
1 kJ/m³K	= 0.0149 BTU/ft3 F	1 BTU/ft3 F	= 67.070 kJ/m³K
Heat			
1 kJ/m².	= 0.0881 BTU/ft2	1 BTU/ft2	= 11.357 kJ/m²
1 W/m²	= 0.3170 BTU/h ft2	1 BTU/h ft2	= 3.155 W/m²
1 W/m²K	= 0.1761 BTU/h ft2 F	1 BTU/f ft2 F	= 5.678 W/m²K
1 W/mK	= 0.578 BTU/h ft F	1 BTU/h ft F	= 1.7296 W/mK
	= 6.9348 BTU in/h ft F	1 BTU in/in ft2 F	= 0.1442 W/mK
1 m²K/W	= 5.6786 h ft2 F/BTU	1 h ft2 F/BTU	= 0.1761 m²K/W
1 mK/W	= 1.7296 h ft F/BTU	1 h ft F/BTU	= 0.5782 mK/W
	= 0.1442 BTU in/in ft2 F	1 h ft f/BTU*in	= 6.934 mK/W
Refrigeration			
1 kW	= 0.2843 tons of refrigeration	1 ton of refrigeration	= 3.517 kW
Heating			
1 kW	= 0.1019 HP (boiler)	1 HP (boiler)	= 9.809 kW
			= (33 475 BTU)
1 kW	= 14.22 EDR (steam)	1 EDR (equivalent	
	= 22.74 EDR (water)	direct radiation)	= 70.34 W
	= 3412 BTU/h water	steam	= 43.97 W

Appendix III

Comparative Summary of Engine Features

Feature	Diesel	Otto	
1. Design data			
- compression ratio e	15 . . . 21	6 . . . 9.5 petrol	
			6 .. .12 alcohol
- pressure after compression			
without ignition	35 . . . 60 bar	15 . . . 20 bar	
- temperature after compres-			
sion without ignition	600. . . 900 °C	400 . . .600 °C	
-excess air ratio λ	1.3 . . . 4.0	0.7 . . .1.2	
-efficiency	0.3 . . . 0.4	0.2. . . 0.35	
-specific fuel consumption	230 . . . 350 g/kWh	300 . . .400 g/kWh	
-volumetric efficiency	0.7 . . . 0.9	0.3 . . . 0.9 (low values for	
			partially closed throttle)
-exhaust gas temperature	400 . . . 600 °C	500 . . . 900 °C	
- speed ratio			
- stationary	1,300 . . . 2,500	1,300 . . . 2,500 (gas)	
	- vehicle	1,300 . . .5,000	1,300 . . . 7,000
	- ignition type	self-ignition by injection of fuel into hot compressed air shortly before piston reaches TDC	spark ignition by spark plug
	2. Control principle	variation of amount of fuel injected by the injector pump. Airflow is not controlled, i.e. full compression is always achieved. The variation of amount of fuel is done by the centrifugal mechanism of the governor with the aim to maintain the speed chosen and set by the control lever position.	variation of admission of ready air/fuel mixture by throttle valve between mixing device (carburetor, venturi, mixing valve) and engine inlet. Throttling reduces actual suction pressure of engme, hence absolute compression and efficiency.
	- manual	by setting the governor control lever to the required speed which remains constant within small limits, irrespective of the actual power demand. Speed changes can be achieved by setting the lever to a different position.	by setting the lever of the butterfly valve (throttle) in the carburetori load/speed variations require an appropriate regulation of the throttle.
	- automatic	using the same mechanism as above.	- using a separately mounted mechanical governor to operate the throttle.
			- using an electronic speed sensor with control unit and actuator to operate the throttle.

Features of Biogas Engines
(only where different from unmodified engines)

Feature	Gas diesel	Gas Otto
1. Design data		
-compression ratio e	15 . . .18	10. . . 12
-excess air ratio \	1.3 . . . 4.0	0.9 . . . 1.3
-specific fuel consumption	0.55 . . . 0.75 m³/kWh	0.65 . . .1.0 m³/kWh
	(+ pilot fuel)	
-exhaust gas temperature	500 . . . 700 °C	500 . . . 900 °C
- ignition type	self-ignition of pilot fuel inject- ed into a hot compressed mix ture of air and gas which is ignited by the pilot fuel sub sequently.	as in other Otto engines
2. Control principle	A small amount of diesel fuel is injected to facilitate ignition. Variation of the amount of fuel gas supplied to the mixing de vice is used for variation of power output. The airflow is not con trolled to maintain a high pres sure and ignition temperature. mode	as in other Otto engines
- manual	The governor/injector system is fixed at supplying the pilot fuel amount only. The gas valve at the mixing chamber is set to achieve the required speed/power output.	as in other Otto engines. Carburetor is replaced by venturi mixer or gas mixing valves.
- automatic	Using the same mechanism as above. The gas valve is however operated by a governor or an actuator of an electronic control system.	as in other Otto engines.
Overview of Mode of Operation, Control and Mixing Device		
Mode of operation	**Type of control**	**Type of mixing device**
speed: constant load: constant (e.g. pump with constant head and capacity; electric generator with constant load and frequency)	- manual	
	Otto: fixed setting of gas and air or throttle in the case of a venturi	Otto: - simple mixing chamber with manually operated control valve for air and for gas, or
		- venturi-type mixer with manu diesel:
	fixed setting of gas	- simple mixing chamber with manually operated control valve
		- automatic not necessary as long as load remains constant
speed: constant load: varying (e.g. electric generator with constant frequency and varying electricity demand; pump with varying capacity and head)	- manual	
	Otto: adjustment of gas/air valves or throttle (venturi) whenever load changes. Not recommended with frequent load changes.	Otto:
		- venturi mixer with manually operated throttle (Simple mixing chamber with two valves is impracticable for readjustments.)
		- mixing valve with manually operated butterfly throttle diesel:
	diesel:	
	adjustment of gas valve whenever load changes. Without adjustment load variations are compensated by variations in diesel fuel supply automatic ally. Substitution of diesel fuel by biogas is however reduced.	- simple mixing chamber with manually operated gas valve
	- automatic	
	Otto:	Otto:
	speed governor or electronic control system operating the butterfly throttle of mixing device diesel: fixed setting of pilot fuel injection. Electronic control or governor operating the gas valve of the	- venturi mixer, or

	mixing chamber.	
		- mixing valve with butterfly valve operated by control system
		diesel:
		- simple mixing chamber with gas valve operated by control system
speed: varying load: varying (e.g. drive of different machinery)	- manual	
	Otto: adjustment of throttle valve in accordance with required load/speed	Otto: - venturi mixer, or - gas mixing valve with manually operated butterfly
	diesel:	diesel:
	adjustment of gas valve in accordance with required load/speed	- simple mixing chamber with manually operated gas valve
	- automatic	
	Otto:	Otto:
	mechanical governor or electronic control system with practicable mode of set point adjustment	- venturi mixer, or - gas mixing valve with butterfly valve operated by control system
	diesel:	diesel:
	fixed setting of pilot fuel injection. Electronic control system or (separate) speed governor with	- simple mixing chamber with gas valve operated by control system practicable mode of set point adjustment.

Appendix IV

Design Drawings of a Venturi Mixer for Self-Manufacture (Example)

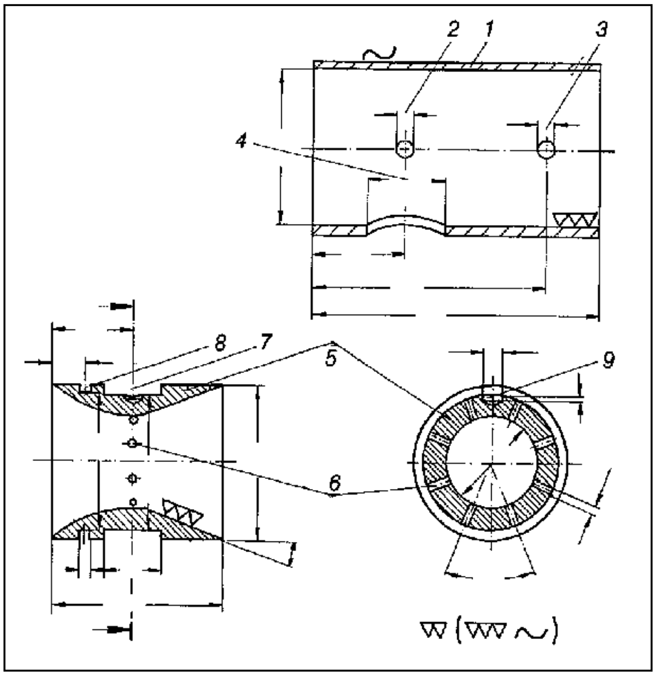


Fig. 1: Venturi mixer parts 1 Mixer body (tube), 2 Bore for venturi ring holder bolt, 3 Bore for butterfly valve shaft, 4 Bore for gas supply pipe connection (brazed, welded), 5 Venturi ring, 6 Calibrated bores for gas inlet, 7 Gas supply ring channel, 8 Groove for seal ring (O-ring), 9 Bore for venturi ring holder bolt; all dimensions according to calculations and engine inlet size (refer to Chapter 6).

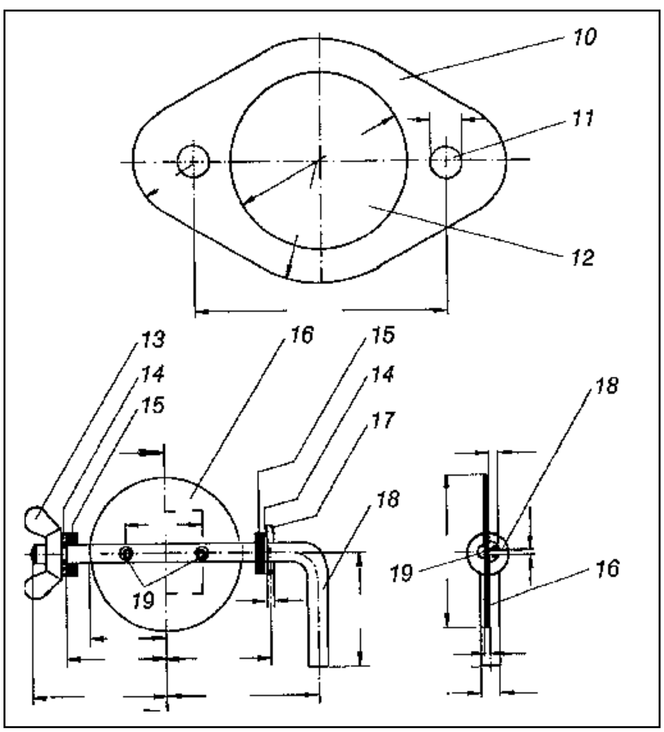


Fig. 2: Venturi mixer parts, continued. 10 Connection flange, 11 Bore for connection bolts to engine and air filter, 12 Bore for connection of mixer body (brazed, welded), 13 Wing nut for fixing the butterfly valve shaft, 14 Washer, 15 Rubber/plastic seal ring, 16 Butterfly valve, 17 Washer fixed to butterfly valve shaft, 18 Butterfly valve shaft cum control lever, 19 Small bolts for fixing butterfly to shaft.

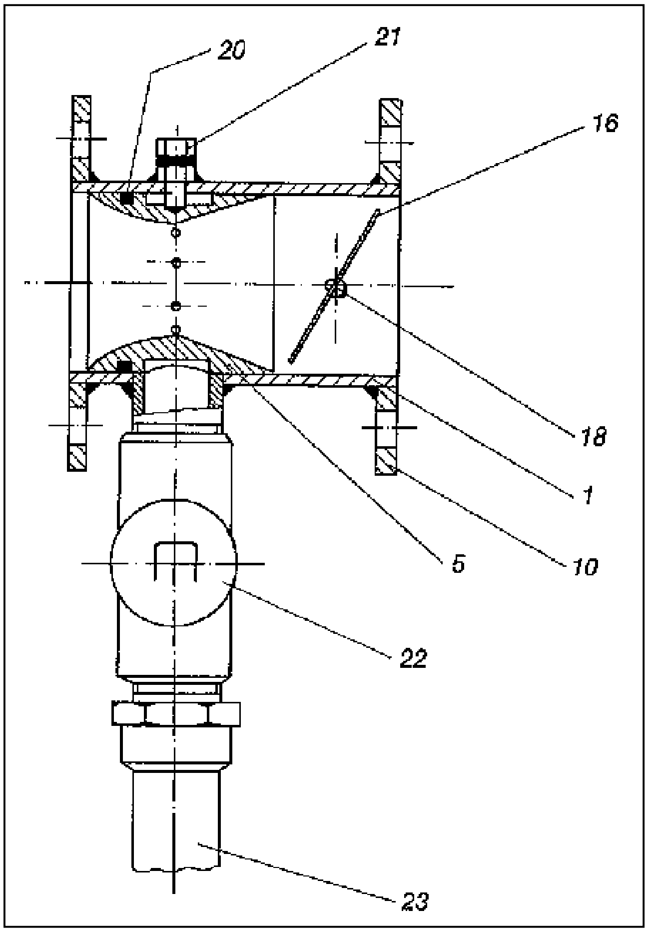


Fig. 3: Venturi mixer assembly
20 Venturi seal ring (0-ring), 21 Venturi holder bolt and nut, 22 Gas inlet valve, 23 Gas supply pipe

Appendix V

Planning Scheme for the Lay Out of a Biogas Plant (1

Step 1: establish gas requirement
Guide values for gas consumption

Cooking: 0.25 m³ (8 cu ft) per person per day
Lighting: 0.12-0.15 m³ (4-5 cu ft) per hour per lamp
Driving engines: 0.75 m³ (17 cu ft) per kW per hour

Gas requirement per unit x No. of units = Total gas requirement

Example: 0.25 m³ per person and per day X _4_ persons =__1.0__m³

Gas requirement for cooking: 0.25 m³ per person per day X _________persons =__________m³
Gas requirement for lighting: 0.15 m³ per lamp per hour X ___________lamps =___________m³
Gas consumption of engines: 0.75 m³ per kW per hour X ________operating hours = ______m³

Gas requirements for other processes: _________X__________=__________m³
- Refrigeration _________X__________=__________m³
- Drying plant _________X__________=__________m³
- Production __________m³
Total gas requirement per day

Is this gas requirement likely to satisfy needs in 5 years?

Z Additional gas requirement _________X__________=__________m³

Total gas requirement _____________m³

Step 2: establish gas production
Gas generation - guide values

Type	Manure (moist) per day	Gas per kg per day	Gas yield per animal
1 head of cattle	10 kg	361(1.3 cu ft)	3601(13 cu ft)
1 water buffalo	15 kg	361(1.3 cu ft)	5401(19.5 cu ft)
1 pig (approx. 50 kg)	2.25 kg	781(2.8 cu ft)	1801(6.3 cu ft)
1 chicken (approx. 2 kg)	0.18 kg	62 1(2.2 cu ft)	11.21 (0.4 cu ft)
Adult human excrete	0.4 kg	701(2.5 cu ft)	281(1 cu ft)

Gas yields refer to material with its natural moisture content.

For the final design of a biogas plant the use of specific literature e.g. 131, [4], [5], [6] is recommended, likewise the consultation of a biogas expert if available.

Actual production
Fertilizer production Gas production
Number X volume per unit = volume per day Number X volume per unit = gas per day

Example: 2 buffalo X 15 kg/day = 30 kg/day 2 x 0.540 m³/day = 1. 08 m³/day

Buffalo	X 15 kg/day	=	kg/day X 0.540 m³/day =	m³/day
Cows	X 10 kg/day	=	kg/day X 0.360 m³/day =	m³/day
Calves	X 5 kg/day	=	kg/day X 0.200 m³/day =	m³/day
Pigs (50 kg)	X 2 kg/day	=	kg/day X 0.180 m³/day =	m³/day
Horses	X 10 kg/day	=	kg|day X 0.350 m³/day =	m³/day
Sheep	X 2 kg/day	=	kg/day X 0.100 m³/day =	m³/day
Chickens	X 0.18 kg/day	=	kg/day X 0.011 m³/day =	m³/day
Toilets	X 0.4 kg/day	=	kg/day X 0.030 m³/day =	m³/day
Green material	kg/day X 0.200 m³/day =	m³/day		

Gas and manure production per day kg/day m³/day
Does this correspond with livestock
in 5 years?
Increased
level X kg/day = kg/day m³/day = m³/day

Gas and manure production
potential kg/day m³/day:

Step 3: Comparison between gas volume needed and gas generation potential

Does potential gas production match requirements?	If so, the chosen size of plant is correct, and the next step can begin.
Is production greater than required?	It may be a good idea to build this plant nonetheless and to ask a neighbor if he alsorequires biogas; if not, build a smaller plant.
Is consumption higher than potential gas production? Check the following possible measures:	- Can consumption be lowered (calculate gas requirements again)? - Can more organic material be acquired as fuel (calculate gas production again)? - Can a plant be constructed jointly with a neighbor?

Step 4: Calculating influencing factors on the biogas plant

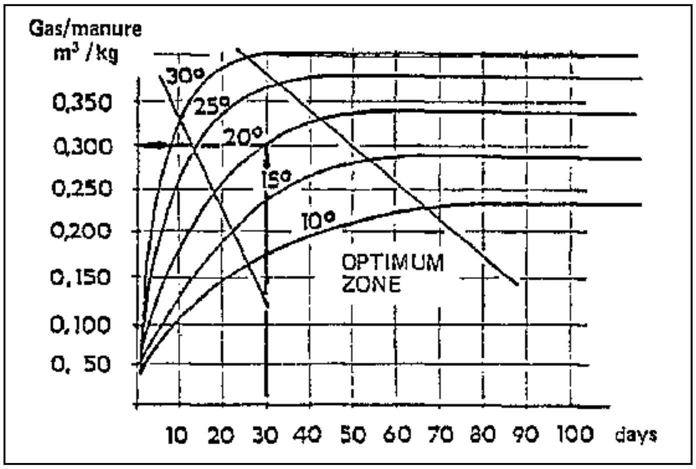
Temperature - fermentation period in digester

The fermentation time is an important factor in determining the size of the biogas plant and epends on the temperature in the digester. The fermentation period is defined as the time taken for material to flow through the plant from input to output. The following guide values apply to the regions stated:

30 - 40 days	Hot, tropical plains climate: e. g. Sudan, Cameroon, Sri Lanka, Indonesia, Venezuela,Central America
40 - 60 days	Hot regions which cool down only slightly in winter: e. g. India, Thailand, Philippines, Kenya, Ethiopia60 - 90 days More temperate climate with distinct drop in temperature during winter: e. g. China,Korea, Turkey

The table below shows the relationship between material fermentation time, temperature and gas output.

In regions with a distinct winter season or severe differences between daytime and nighttime, temperatures (mountainous regions) assume a temperature 5 °C lower for calculation purposes.



Digester pit temperature ºC
Fermentation period days

Quantity of material added

The material must be added in the form of a free-flowing liquid, or else blockages will occur, However, if it is diluted too much, gas production will be reduced.

Generally speaking, the solid material must be mixed with at least the same volume of water.

An accurate calculation depends on the analysis of the material and should be based on the list shown below.

Typical mixing ratios

Cow dung, fresh: water	1:0.5
Cow dung, superficially dry: water	1:1
Horse and sheep's dung: water	1:1
Green refuse: water	1:0.5 to 1:2

Quantity of material added per day:

Type of manure/material	Quantity (kg = 1)	Water (kg = 1)	Liters
Cow dung	+	=	
Pig dung	+	=	
Other animal faeces	+	=	
Human excrete from toilets	+	=	
Agricultural refuse	+	=	

Volume added per day liters

note: in case of concrete stable floor the collected urine is sufficient for dilation, no water needs to be added

Step 5: Establishing the dimensions of the biogas plant

Establishing the dimensions of the biogas plant
The volume of the digester pit is determined from the volume of material added per day multiplied by the

fermentation time.
Volume of material added per day X fermentation time =

	kg/day X	days =
	(1000 l = 1 m^3)	
Gas volume from plant per day:	m^3/day	
Size of gas holder = approx. 1/2 of daily gas production =	m^3	

About

With the steady increase in demand for the useful exploitation of renewable energy resources the transformation of biogas into shaft - or electrical power appears as one of the sensible options for biogas utilization.

This book wants to provide a source of information not only for the various technical aspects of modification of internal combustion engines, both Diesel- and Gasoline (Otto-)engines, to operate on biogas-fuel but also for planning and economic operation of these engines in a system comprising of the fuel generating biogas plant and the power consuming driven equipment.

The reader, who is assumed to have basic technical interest and understanding, will furthermore find information on the use of the engine's waste heat and a commented list of manufacturers of biogas engines and available equipment for the self-modification of engines.

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