UNIVERSIDAD TECNICA FEDERICO SANTA MARIA
VALPARAISO - CHILE

INTERNAL COMBUSTION ENGINE ADAPTATION TO THE USE OF BIOGAS IN PUMPS AND GENERATION SETS

COURSE AND WORKSHOP PREPARED FOR
OLADE IN JAMAICA, SEPTEMBER 1986.

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OBJECTIVES

The aim of this course is to train 15 Jamaican Technicians in the field of internal combustion engine adaptations to the use of biogas as alternative fuel. Mainly generating sets and pumps up to 5KW power output will be seen. Therefore 9 sets will be analyzed, in order to decide the necessary modifications, their design and practical construction as well as the in field adjustments.

A theoretical preparation in classroom according to the following contents will be held by the author. Other applications performed out by chilean research groups will also be shown.

1. TYPES OF MOTORS AND GOVERNORS.

There exist more than one criterion to define types of motors, but two main types are here considered: Compression self-ignited (Dieseloil) and external or spark ignited (gasoline). The first one is also known as quality-regulated (more or less Dieseloil for the same air) and the second one as quantity-regulated (more or less mixture of almost the same air/fuel relation) (Figure: 1.1.)

In the spark-ignition engine, fuel metering to provide the required quality of mixture is normally achieved by measuring the air flow in some way (by venturi, or by sensing engine speed, air temperature and pressure), and matching the fuel flow accordingly. The quantity of mixture entering the cylinders is controlled independently by means of a variable throttle plate. In the compression-ignition engine, the fuel must be injected directly into the cylinder in order to achieve ignition at compression temperatures and pressures, consequently it is customary to dispense with the throttle plate and control both mixture quality and quantity by means of the amount of fuel injected.

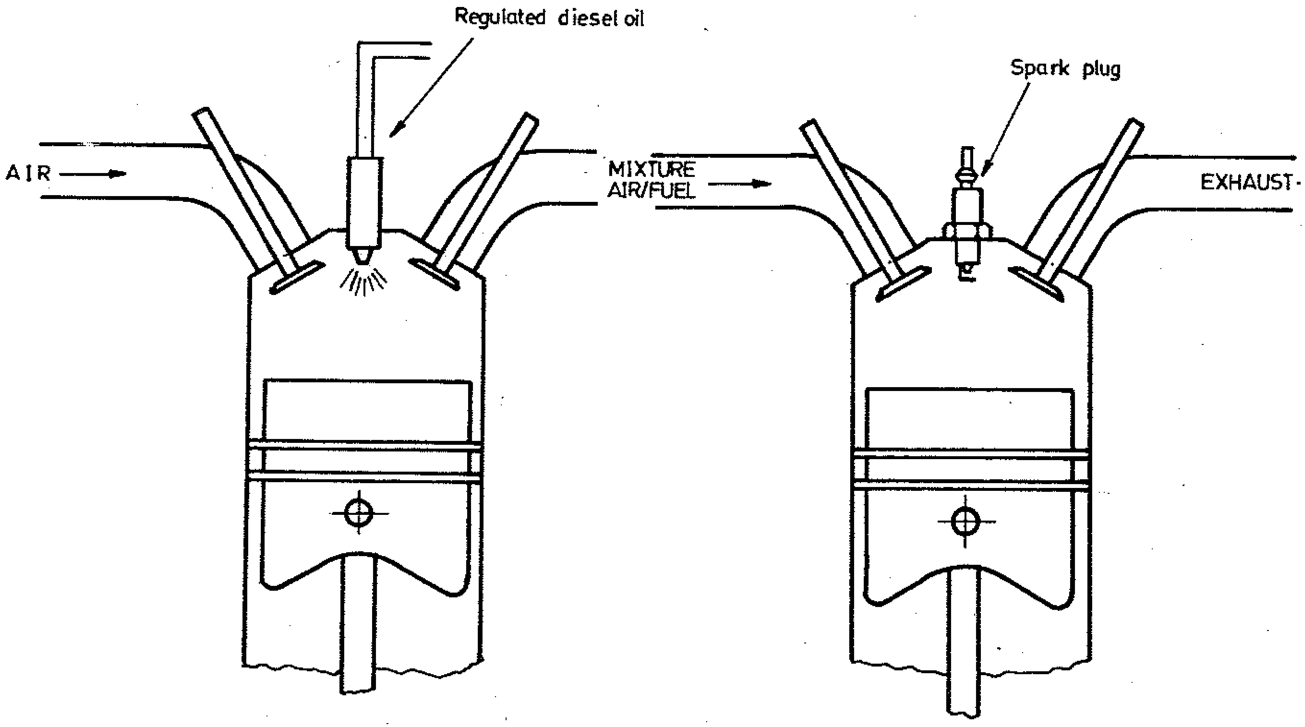


Fig. 1.1: a) Compression ignited and, b) Spark ignited engine.(Ref: 1)

Mixture preparation is achieved easily by upstream mixing in the case of gaseous fuels, and by carburetion with the volatile gasolines, although some multi-cylinder spark-ignition engines achieve improved mixture distribution by means of continuous injection into the manifold, or by timed injection either into the inlet ports or direct in the cylinders themselves. In compression-ignition engines, timed cylinder injection is essential.

The air/fuel relation (R-A/F) has values near the stoichiometric relation ($\lambda=1$) for the spark ignited engine, but varies greatly for the compression self-ignited one, with a minimum around 30 to 40 percent ($\lambda=1,3$/ $\lambda=1,4$) air excess. This is done to avoid soot formation because of slow combustion in the final phase. So the possibility of supplying more energy per volume of combustion chamber is limited due to the fuel characteristic (Figure: 1.2.)

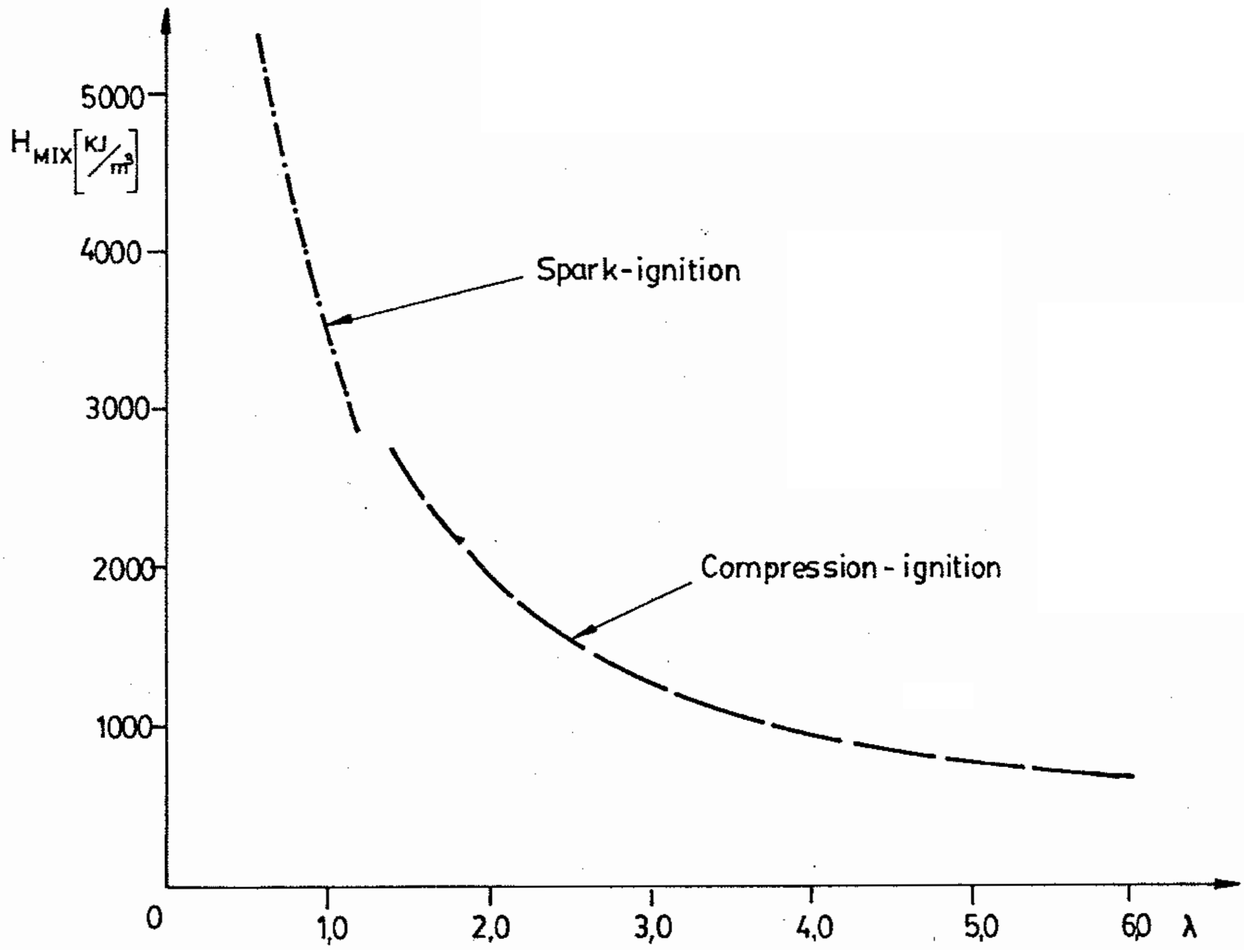


Fig. 1.2: Energy content per volume. (Ref: 1).

The Energy Content per Volume of Mixture, H_{mix}[kJ/m^3] for a Gasoline Engine is:

$$H_{mix} = \frac{m_f \cdot Hi}{V_{mix}} \quad \text{, where:}$$

$$V_{mix} = \frac{m_{mix}}{\rho_{mix}} = \frac{1}{\rho_{mix}} \quad (m_{Ai} + m_f) = \frac{m_f}{\rho_{mix}} \left(\frac{m_{Ai}}{m_f} + 1\right)$$

but: $\frac{m_{Ai}}{m_f} = \lambda \cdot A_{ST}$, so:

$$\boxed{H_{mix} = \frac{H_i \cdot \rho_{mix}}{\lambda \cdot A_{ST}+1} \quad \left[\frac{kJ}{m^3} \right]}$$ Gasoline

m_f = Fuel mass in mixture [kg]

H_i = Fuel heating value [kJ/kg]

V_{mix} = Mixture air/fuel volume which contains m_f[m^3]

ρ_{mix} = Mixture density [kg/m^3]

m_{Ai} = Air mass in mixture [kg]

A_{ST} = Stoichiometric air [Kg_{Air}/kg_{Fuel}] (see 2.3.1.)

For a Diesel: $H_{mix} = \frac{m_f \cdot H_i}{V_{Ai}}$

$$\boxed{H_{mix} = \frac{H_i \cdot \rho_{Ai}}{A_{ST} \cdot \lambda} \quad \left[\frac{kJ}{m^3} \right]}$$ Diesel

1.1. 2 AND 4 STROKES.

Motors can be classified in: two-stroke anf four-stroke engines (Figures: 1.1.1. to 1.1.4.). It is important to identify the lubrication system of a motor to be used with biogas, since the oil/fuel mixture used in some two-stroke engines to lubricate the important moving parts, would lead to cero lubrication if straight replacement is applied.

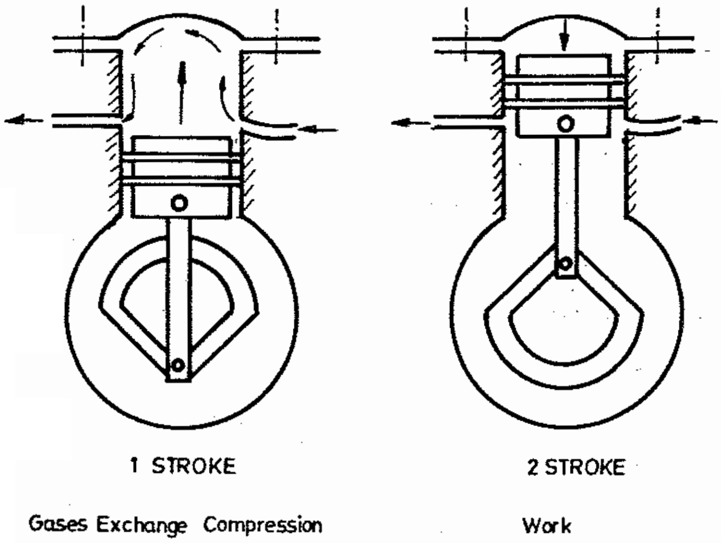


Fig.1.1.1: Two - Stroke engine.(Ref: 11)

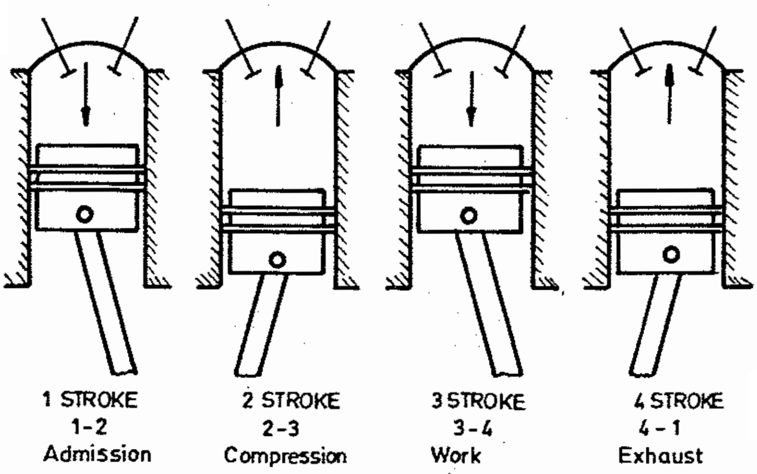


Fig.1.1.2: Four - Stroke engine.(Ref: 11)

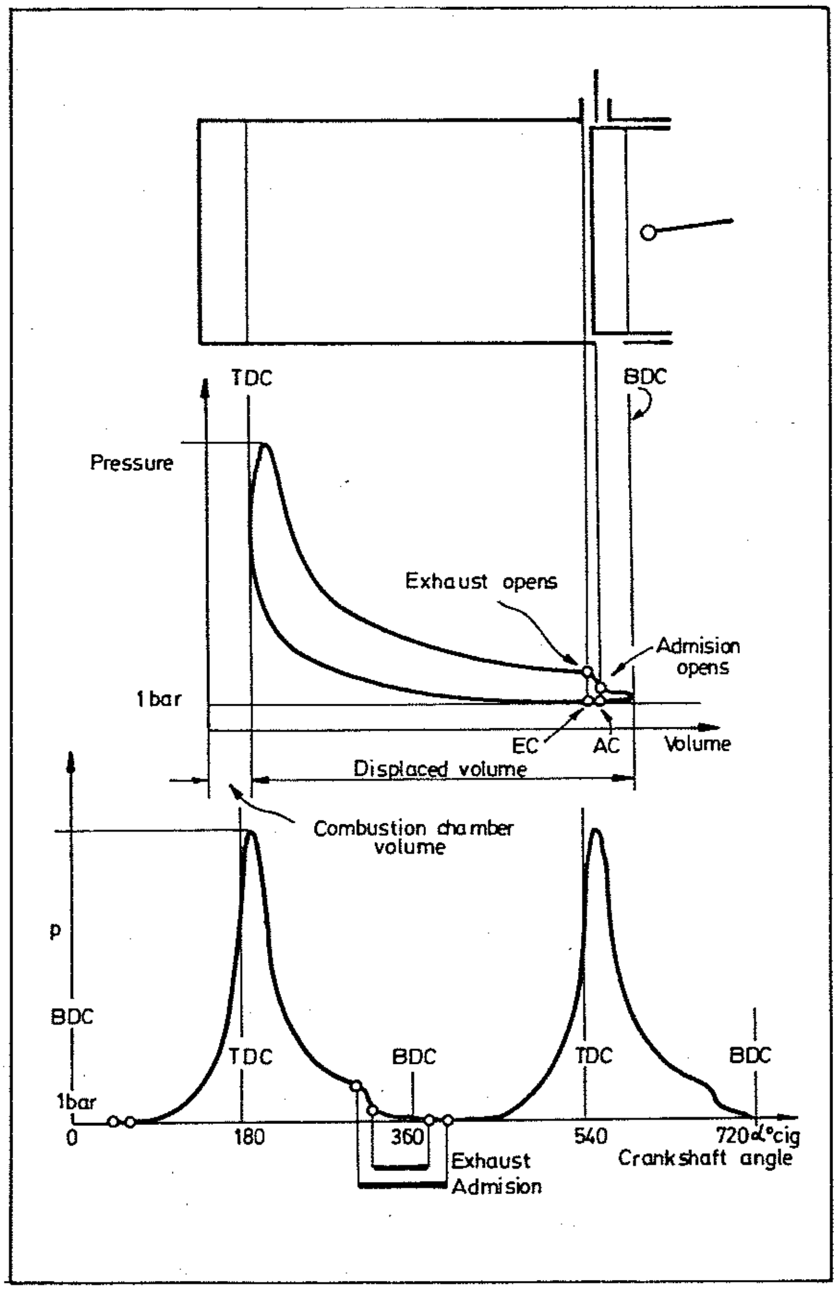


Fig.1.1.3: Two - Stroke motor.(Ref: 1)

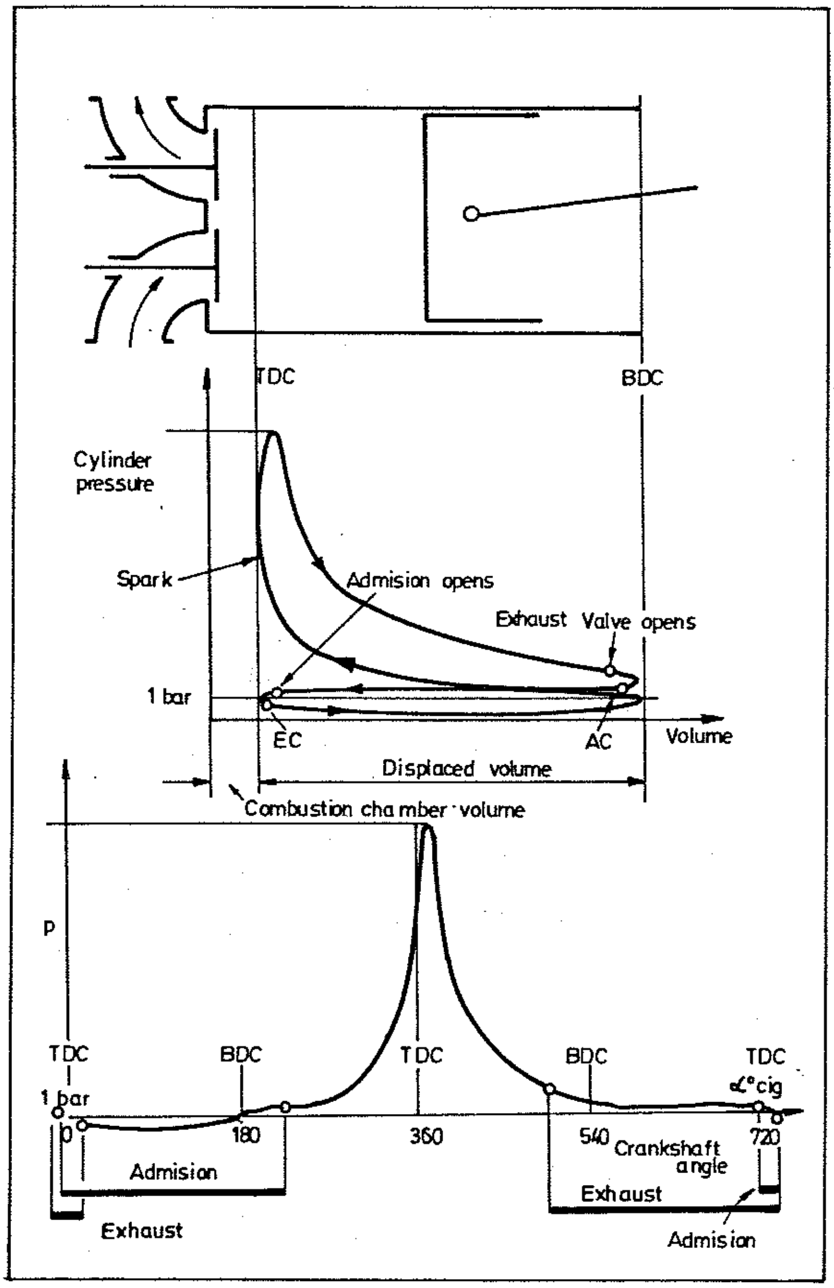


Fig.1.1.4: Four - Stroke motor. (Ref: 1)

1.2. GOVERNORS

Of interest here are governors able to control fixed constant speeds (normally about a maximum 5% drop). They operate either on a signal or on a mechanically transmitted force from counterweights which actuates at increasing speed (as setted) to limit the fuel supply.

When more power is demanded, the speed reduces and so does the force of the counterweights permitting more fuel supply. If more power than the maximum allowable for the engine is asked, the engine will stall.

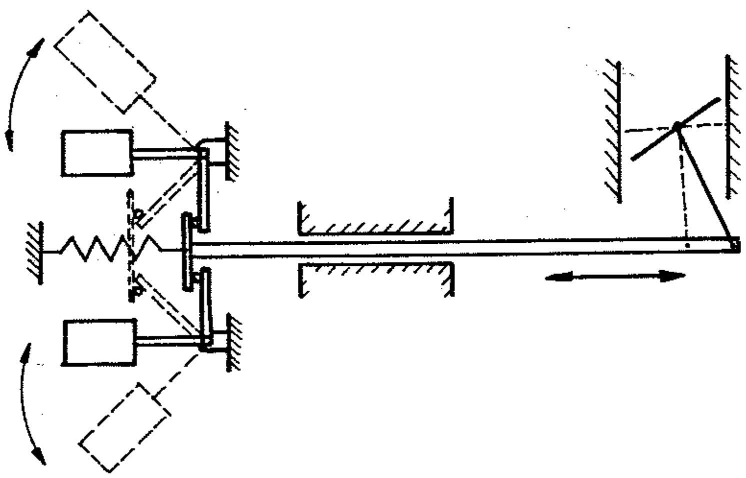


Fig.1.2.1: Governor Scheme.(Ref: 1)

2. FUEL PROPERTIES.

The traditional crude oil derivates such as gasoline and Dieseloil are liquids which have to be gasified before combustion can take place. Even if their higher density is convenient for transport (vehicles) applications, a gasification device (as the carburetor for gasoline engines) is needed. The gaseous fuels, as methane (LPG) are therefore known as noble fuels. This also because of the reduction in contaminants in the exhaust gases, due to a cleaner combustion (see Fig. 2.1.).

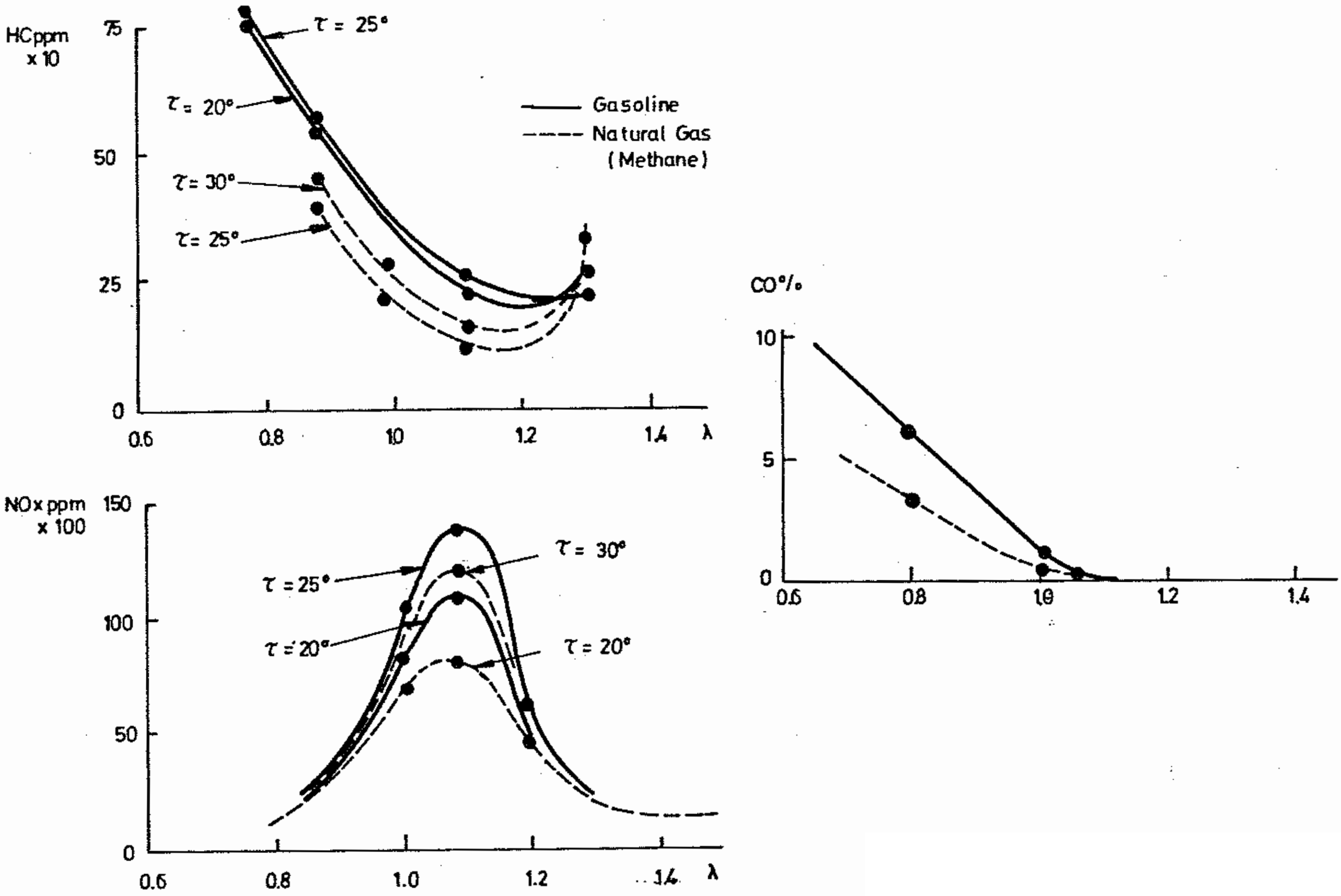


Fig. 2.1: Pollutants in Spark-ignited Engine (Ref: 2)

Some characteristics of the fuels of interest are shown in Table: T.2.1.

TABLE: T.2.1. FUEL PROPERTIES.

PROPERTY	METHANE (CH_4)	60% CH_4 40% CO_2	GASOLINE	DIESELOIL
H/C Atomic Ratio	4		1,83	1,63
Self Ignit. Temp. °C			245	240
Boiling Point °C	-161,5	-128	Curve	Curve 195-450
Critical Temp. °C	- 82,6	--	Mixture	Mixture
Critical Pressure MPa	4,62	--	Mixture	Mixture
Inf.Heat Valve MJ/Kg	50,05	17,5	42,50	40,60
Net MJ/Kg Stoic Mix	2,75		2,74	2,73
Octane Rating			81-98	--
Cetane Rating	See Fig.2.1.1		8-14	45-55
Density (Normal) kG/m^3	0,72	1,2	0,76	815-855
Stoichiometric Air [kg/kg]	17,2	6,1	14,7	14,5
Sulfur Content % M.	--		<0,1	0,2

The boiling levels of petroleum and alternative fuels are shown in figure 2.2. and the distillation curves for representative petroleum fuel blends in Fig. 2.2.

Biogas density depends on its composition as can be seen in Figure 2.4.

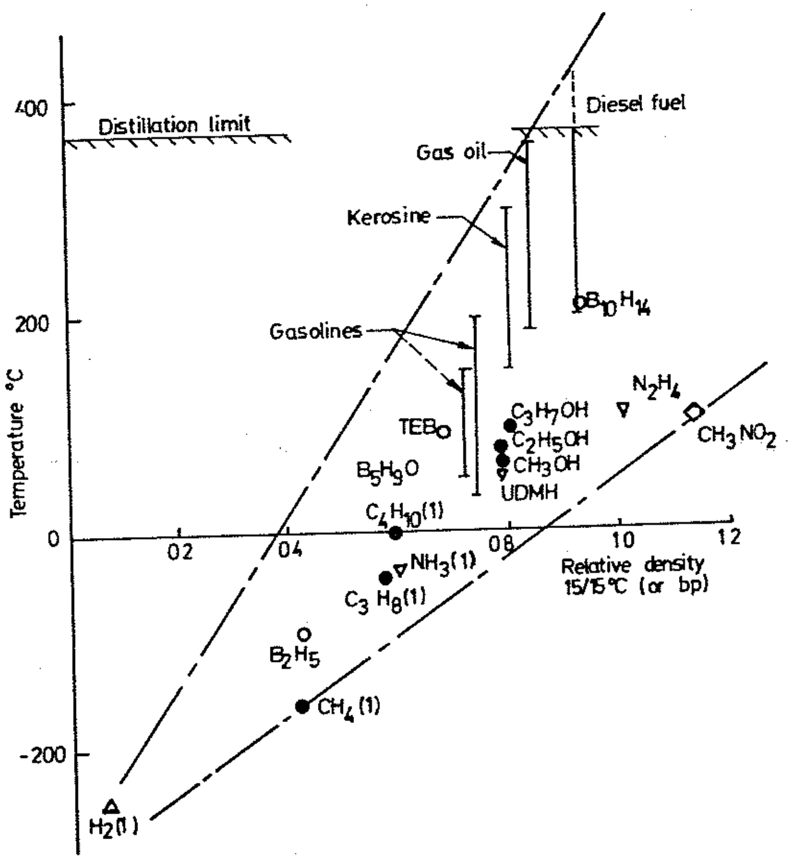


Fig.2.2: Boiling levels of petroleum and alternative liquid fuels.(Ref: 3)

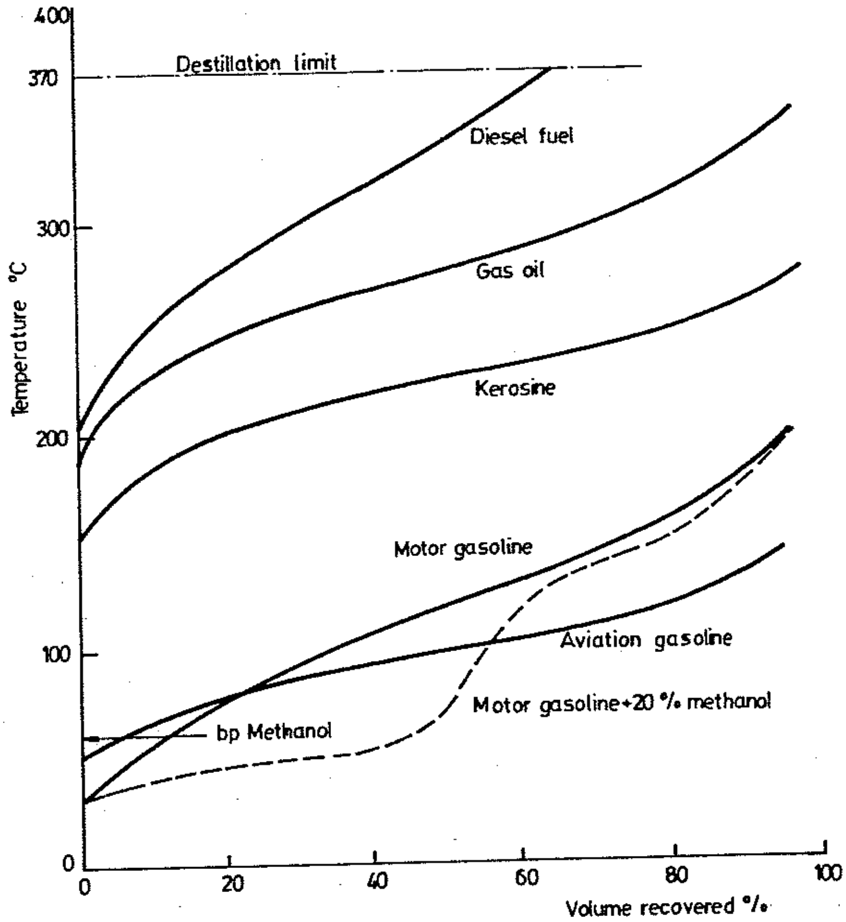


Fig.2.3: Distillation curves for representative petroleum fuel blends.(Ref: 3)

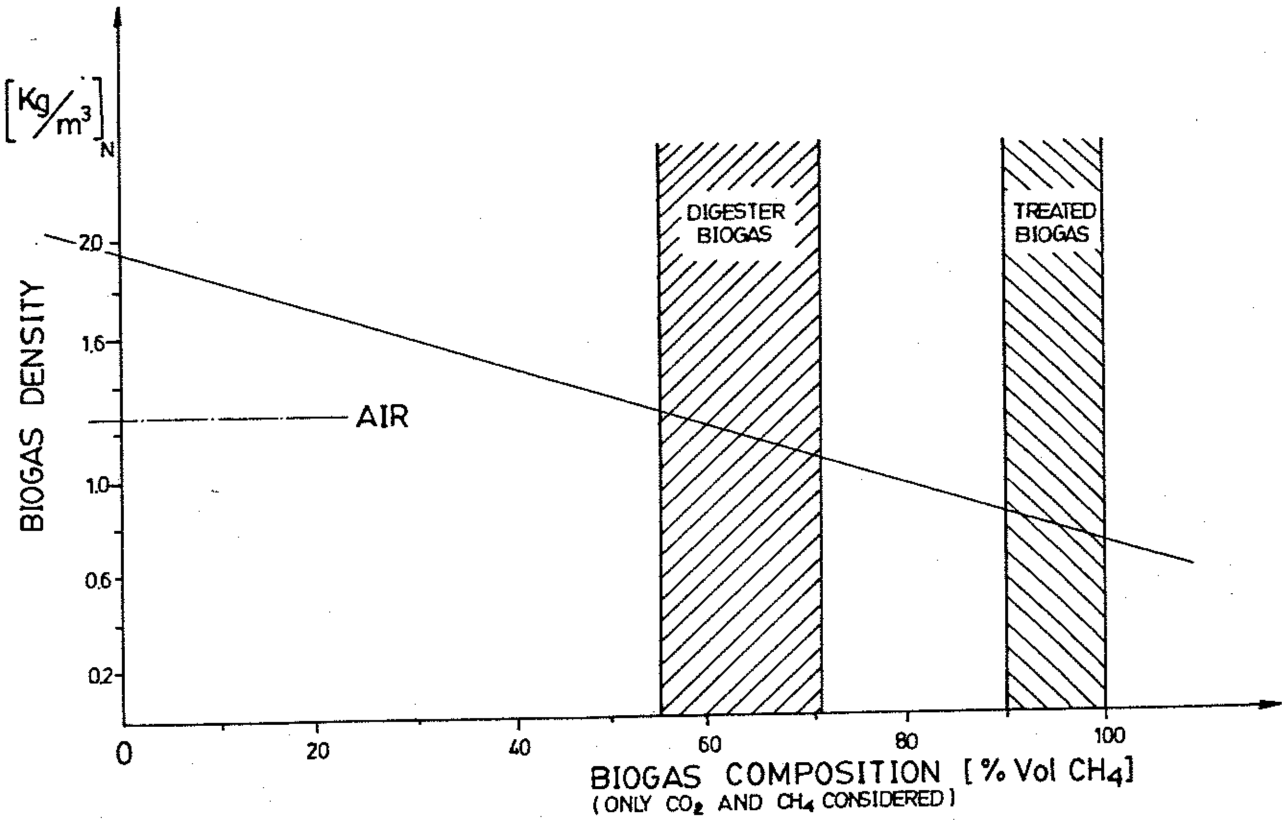


Fig.2.4: Density of Biogas.(Ref: 4)

2.1. OCTANE AND CETANE RATINGS.

The octane rating of a fuel is a measure of its self ignition speed. The higher the octane rating the slower the fuel self ignites. In a spark-ignition engine, a high octane rating is desirable because it allows a fuel to resist detonation and preignition (the spark knock, pinging and run-on associated with many of today's low octane unleaded gasolines). A Diesel engine, on the other hand, runs best with a fuel that ignites easily and burns quickly. A good Diesel fuel is one that has a high cetane rating (the opposite of a high octane rating).

When biogas is used, the cetane number is a function of the substitution rate and the methane content of the biogas (Fig. 2.1.1.). Low cetane numbers at over 50% of substitution will affect the combustion development and the pattern of the pressure-time curve. Specially, the ignition delay may become too high for a smooth Diesel engine operation.

It is common practice to express the amount of gas as a percent of its energy content in relation to the total energy input. In other words, it may be understood as the fraction replaced by gas, of the total fuel required.

$$\boxed{\psi = \frac{mf\ BG \cdot HBG}{mf\ BG \cdot HBG + mfD \cdot HD}\ 100\ [\%]}$$

ψ	=	Substitution rate [%]
mf BG	=	Biogas mass [kg]
HBG	=	Biogas heating value [kJ/kg]
mfD	=	Dieseloil mass [kg]
HD	=	Heating value of dieseloil [kJ/kg]

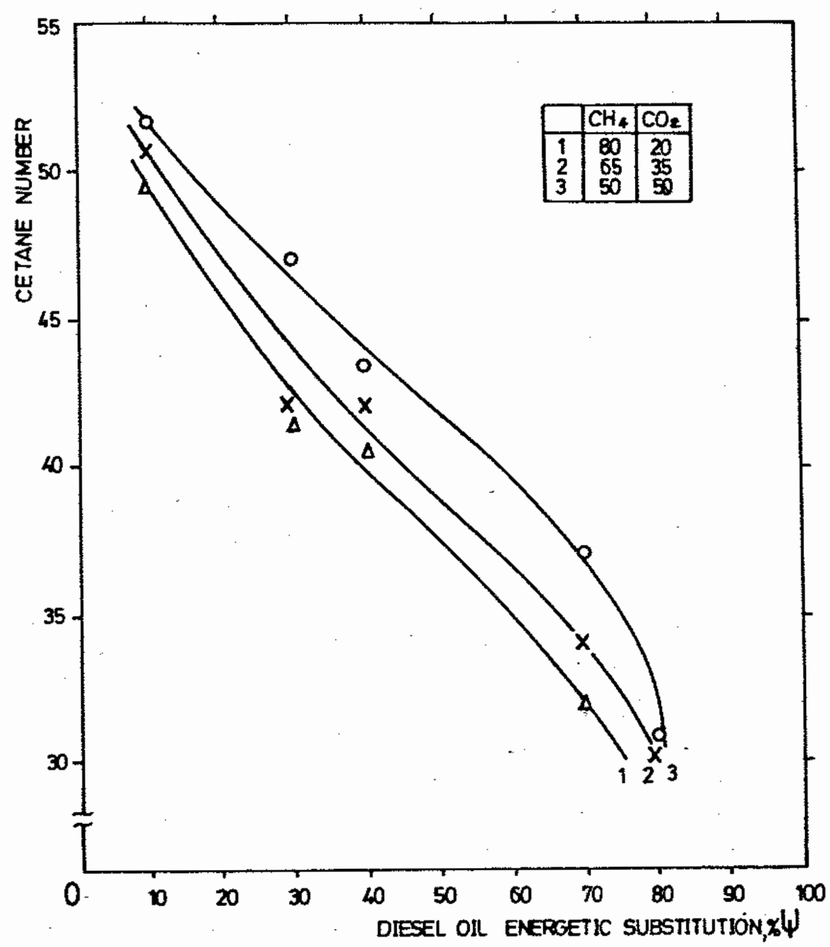


Fig.2.1.1: Cetane rating of mixtures Diesel and Biogas as function of Biogas composition and substitution rate (energetic).(Ref: 5)

2.2. ENERGETIC CONTENT AND AIR REQUIREMENTS.

The heating values of the pure fuels (Table 2.1.) are known. The dependence of different mixtures Dieseloil/Biogas is shown in figure 2.2.1. Using Diesel engines the Air/fuel ratio varies widely and so does the energetic content of the mixtures (Figure 2.2.2.). Also the air requirements will depend on the substitution rate as well as on the biogas quality. For stoichiometric conditions this dependency is shown in Figure 2.2.3. As fuel, it is usually considered the diesel fuel mass plus the total biogas mass. For pure methane it can be seen that more air is required; but the limitation for diesel fuel (λ = 1,3 to 1,5) probably may be overcome, getting closer to stoichiometric combustion, since the soot problem is less severe.

2.3. AIR REQUIREMENT EXERCISES.

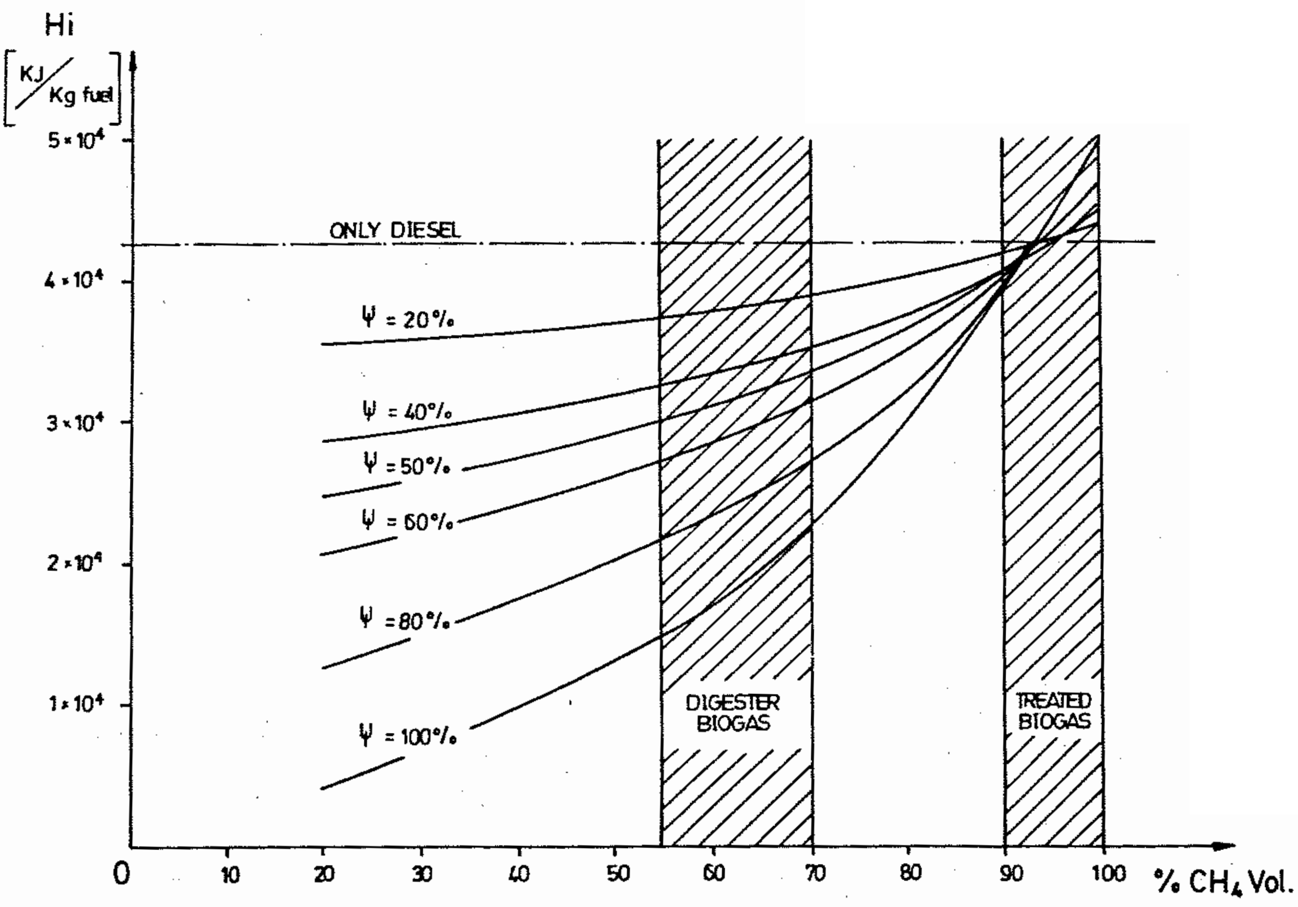


Fig. 2.2.1: Heating value of mixtures Biogas and Diesel oil (Ref: 4)

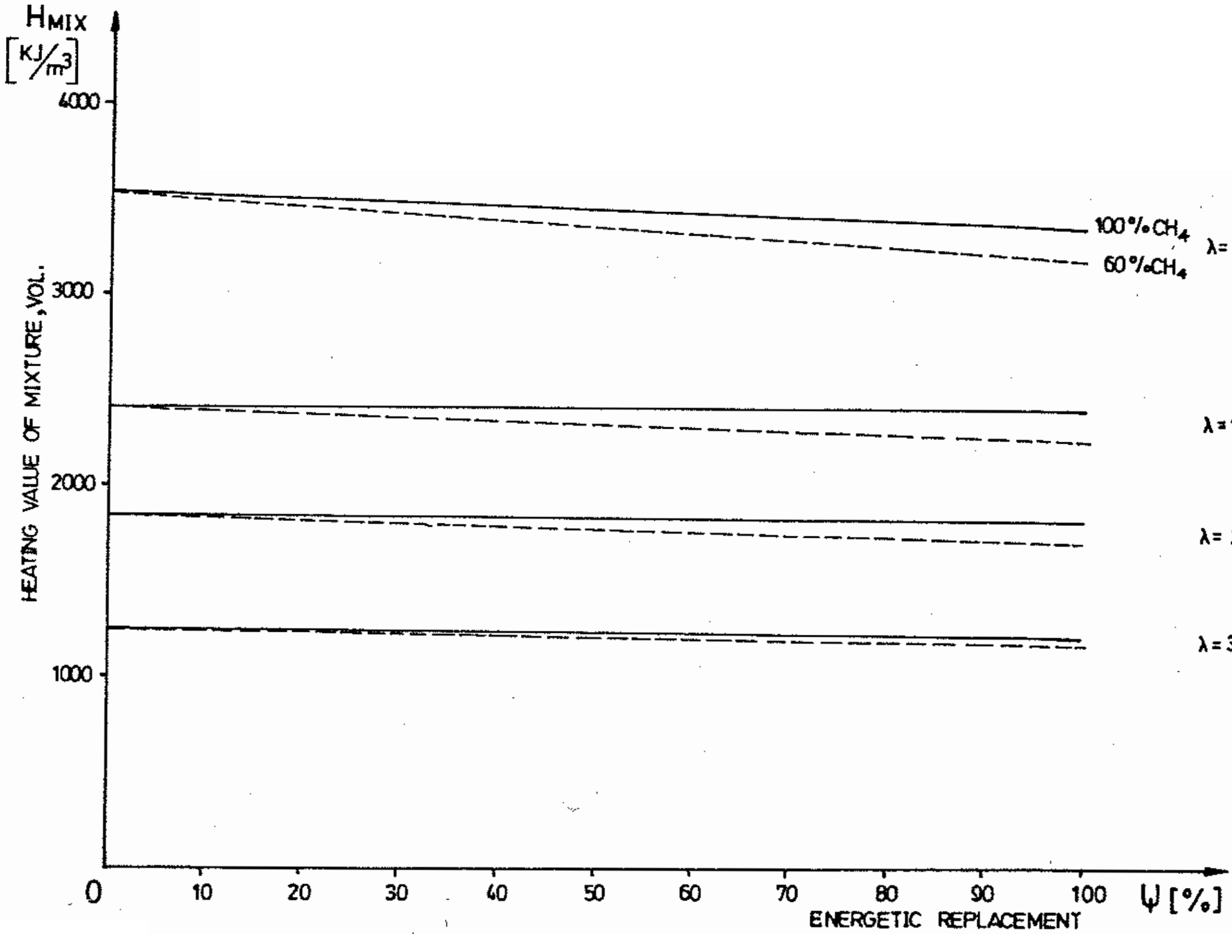


Fig.2.2.2: Energetic content of fuel mixtures Biogas-Diesel oil. (Ref: 4)

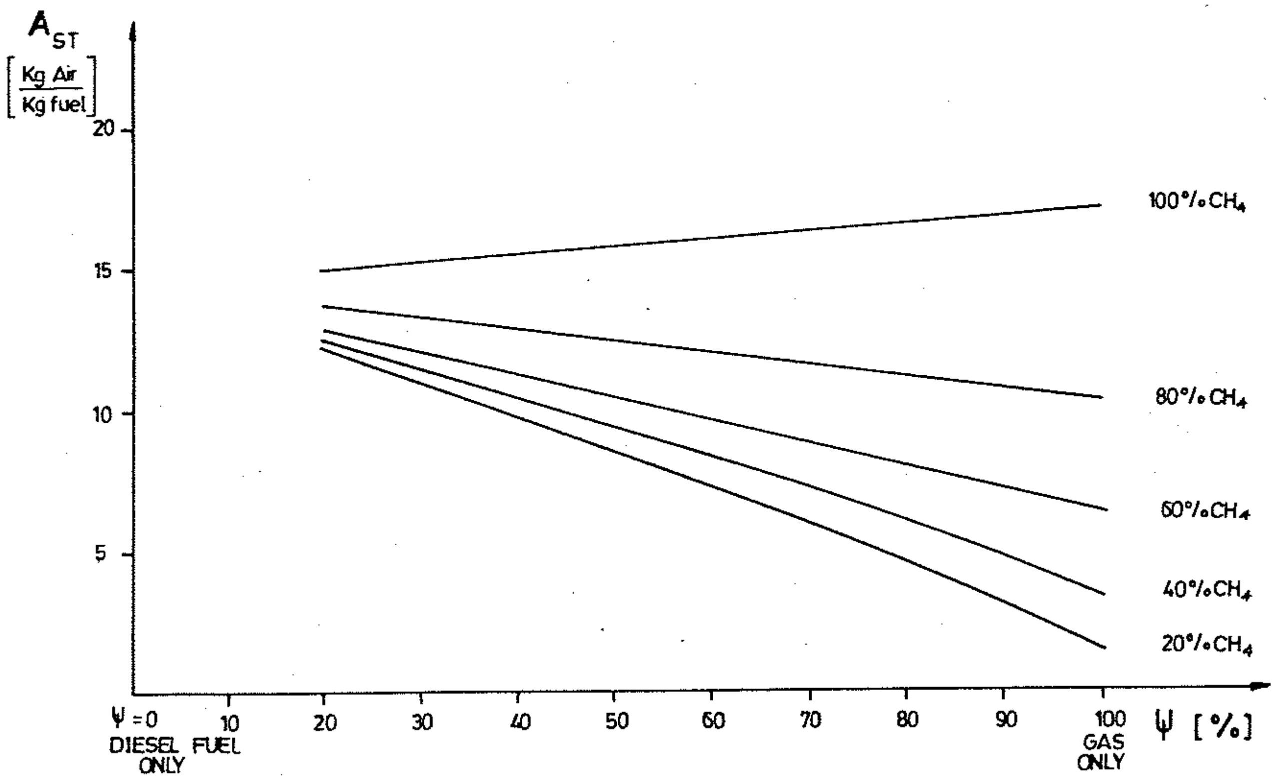


Fig.2.2.3: Air requirements for Biogas and Diesel oil.
(Ref: 4)

2.3.1. STOICHIOMETRIC AIR.

Find a formula to determine the stoichiometric air, A_{ST} for a fuel as function of the elemental composition in mass %: c, h, o, s.

$$C : \quad 1 \text{ MOL } C + 1 \text{ MOL } O_2 \rightarrow 1 \text{ MOL } CO_2 \quad (\text{if } \lambda < 1 \rightarrow co)$$

$$\Downarrow \text{Mass}$$

$$12{,}01 \text{ kg } C + 32\text{kg } O_2 \rightarrow 44{,}01 \text{ kg } CO_2$$

$$\text{PER } 1\text{kg } C + 2{,}664 \text{ kg } O_2 \rightarrow 3{,}664 \text{ kg } CO_2$$

$$\underline{H} : \quad H_2 + 1/2\ O_2 \rightarrow 1H_2O$$

$$1\text{kg } H_2 + 7{,}937 \text{ kg}O_2 \rightarrow 8{,}937 \text{ kg } H_2O$$

:

Oxygen Requirement in KgO_2/kg fuel, summarizing

$$\text{kg}O_2/\text{kgFUEL} = 2{,}664 \cdot c + 7{,}937 \cdot h + 0{,}998 \cdot s - o$$

Air has only $21\%_v O_2 \rightarrow 23{,}2\%m.\ O_2$

$$\boxed{A_{ST} = \frac{1}{0{,}232}\left[2{,}664 c + 7{,}937 \cdot h + 0{,}998 s - o\right] \frac{\text{kg AIR}}{\text{kg FUEL}}}$$

2.3.2. DUAL FUELING-DIAMETERS.

If 25%m of isooctane (C_8H_{18}) is burned as a mixture of 75%m of biogas (60%v. CH_4 - 40%v CO_2), determine:

a) c = %m of carbon in fuel (only for combustion)!

b) h = %m of hydrogen in fuel mixture.

c) $\dot{m}_A$ = Air requirement (kg A/kg F) for $\lambda = 1{,}2$

d) V_G/V_A = Volume flux relation biogas/air.

e) d_G/d_A = Diameter relation of gas and air restriction in mixer.

$\underline{C_8H_{18}}$: 1 MOL C_8H_{18} = 8 Mol C + 9 Mol H_2

$$\Downarrow \text{mass}$$

114,144 kgC_8H_{18} = 96 kg C + 18,144 kg H_2

$$c, C_8H_{18} = \frac{m_c}{m} = \frac{96}{114{,}144} = 0{,}841 \text{ kg C/kg } C_8H_{18}$$

$$h_2, C_8H_{18} = \frac{mH_2}{m} = \frac{18{,}144}{114{,}144} = 0{,}159 \text{ kg } H_2\text{/kg } C_8H_{18}$$

BIOGAS : 1 Mol Biogas = 0,6 Mol CH_4 + 0,4 Mol CO_2

$$= 0{,}6 \text{ Mol C} + 1{,}2 \text{ Mol } H_2 + \underbrace{0{,}4 \text{ Mol } CO_2}_{\text{INERT}}$$

$$\Downarrow \text{mass}$$

26 kg BIOGAS = 7,2 kg C + 1,2 kg H_2 + 17,6 kg CO_2

$$c, \text{BIOGAS} = \frac{7{,}2}{26} = 0{,}277 \text{ kg C/kg BG} \quad h, \text{BG} = \frac{1{,}2}{26} = 0{,}046 \text{ kg}H_2\text{/kgBG}$$

FUEL MIXTURE :

$$c = c, _{C_8H_{18}} \cdot \%m\, C_8, H_{18} + c, _{BG} \cdot \%m \text{ BG}$$

a) $c = 0{,}841 \cdot 0{,}25 + 0{,}277 \cdot 0{,}75 = 0{,}413$ kgC/kgMIXT

b) $h = 0{,}159 \cdot 0{,}25 + 0{,}046 \cdot 0{,}75 = 0{,}074$ kgH_2/kgMIXT

c) $A_{ST} = \frac{1}{0{,}232}[2{,}664c + 7{,}937h + 0{,}998s - o] = 7{,}274 \text{ (kgAIR/kgMIXT)}_{ST}$

$mA/m_F = \lambda \cdot A_ST = 1{,}2 \cdot 7{,}274 = 8{,}73$ kg AIR/kg MIXT

Each kg fuel mixture contains 0,75 of biogas if total relation $\dot{m}A/\dot{m}MIXT$ = 8,73 kg Air/kg Mixt (Assuming air by-pass for isooctane). Then the relation $\dot{m}A/\dot{m}_{BG} = \dot{m}A/\dot{m}$ Mixt $\cdot$ 0,75

$$\dot{m}_A/mBG = 6{,}59 = kg_{AIR}/kg_{BG}$$

For volume relation biogas/air we need the densities:

Air	1,28 Kg/m^3
BG	1,20 Kg/m^3

d) $\dot{V}_G/\dot{V}_A = \dot{m}_{BG}/\dot{m}_A \cdot \varrho_{BG}/\varrho_{Air} = \frac{1}{6,55} : \frac{1,2}{1,28} = 0,143$

e) $\dot{V}_G = \alpha_G \cdot A_G \cdot c_G \qquad V_A = \alpha_A \, A_A \cdot c_A$

Assuming: $\alpha_G = \alpha_A$ and $c_G = c_A$ m/s

$$\frac{\dot{V}_G}{\dot{V}_A} = 0,143 = \frac{\alpha_G \cdot A_G \cdot c_G}{\alpha_A \, A_A \, c_A} = \frac{A_G}{A_A} = \frac{\pi d_A^2/4}{\pi d_G^2/4} = \left\{\frac{d_A}{d_G}\right\}^2$$

$\frac{d_G}{d_A} = \sqrt{0,143} = 0,378$ (compare with simplified mixer calculations 5.4.).

3. GENERATING SETS AND PUMPS..

The special characteristic of these applications is the relative constant speed, maintained by a governor which actuates the main butterfly of the carburetor on a gasoline motor or the injection rack on a diesel oil engine (Figure 3.1.).

3.1. OUTPUT POWER.

Generating sets have electrical parameters (voltage, intensity, cos φ), which allow to determine power output. Through different combinations of resistances the output may be regulated. The power consumed by a pump may be found knowing the mass flow, pressure difference and fluid characteristics.It can be regulated by a valve.

3.2. FUEL SUPPLY.

The necessary fuel energy input for the desired power output depends on the thermal efficiency of the engine and its combustion process. As a rule of thumb, for gasoline engines:

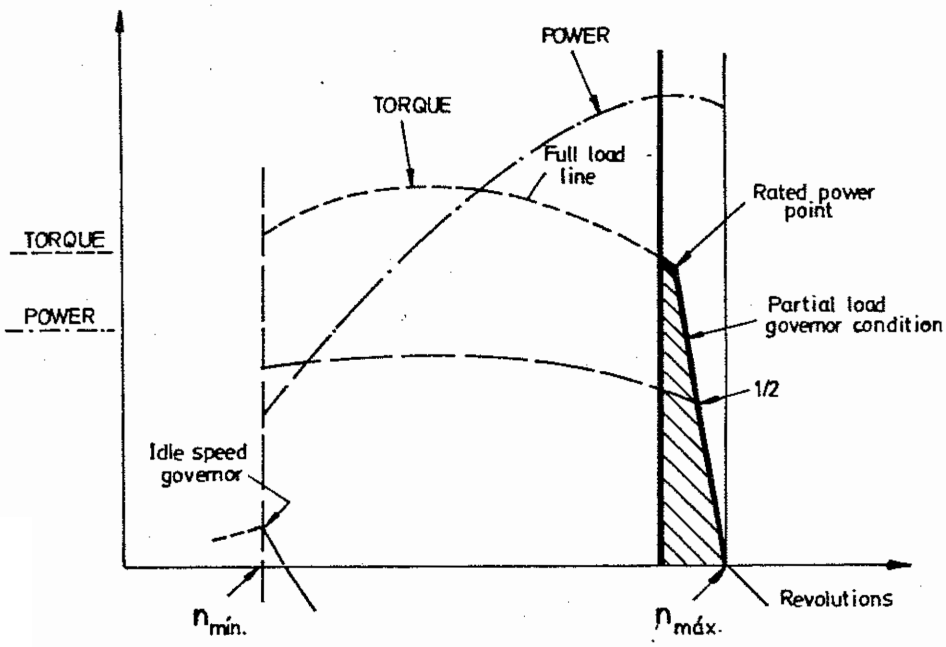


Fig.3.1: Torque and Power outputs.(Ref: 1)

4 times the power output (η =25%) and for Diesel engines: 3 times (η =33%) (η = overall efficiency).

Figure 3.2.1. shows the theoretical thermal efficiency of the internal combustion engine as a function of λ and CR. The upper curve is the theoretical air cycle where heat is added at constant volume, or where combustion is instantaneous.

We see that for good economy we should operate at high CR and lean mixtures. The lower shaded area shows where we operate with current spark ignition engines. As the throttle is partially closed we operate at still lower effective compression ratios and are less efficient. This is due to air pumping losses which increase with throttle closure.

The upper part of the figure shows diesels operate at high compression ratios and with no throttling losses. The major difference in fuel economy between diesel and spark ignited engines occurs at part throttle conditions, where the unthrottled diesel does not have these high pumping losses. At wide open throttle conditions the overall efficiencies of both engines become more alike.

Because of the generally higher efficiency our research group decided to study the latter engines

The thermal efficiency of an engine is defined as the ratio of the mechanical work output divided by the heating value of the fuel used. The thermal efficiency increases as the compression ratio (CR) increases. λ is the air/fuel ratio by weight. λ = 1,0 is the ratio where the oxygen (AIR) which has been provided is exactly the amount necessary to convert every carbon atom and every hydrogen atom of the fuel to CO_2 and H_2O in the combustion process. This ratio is called stoichiometric.

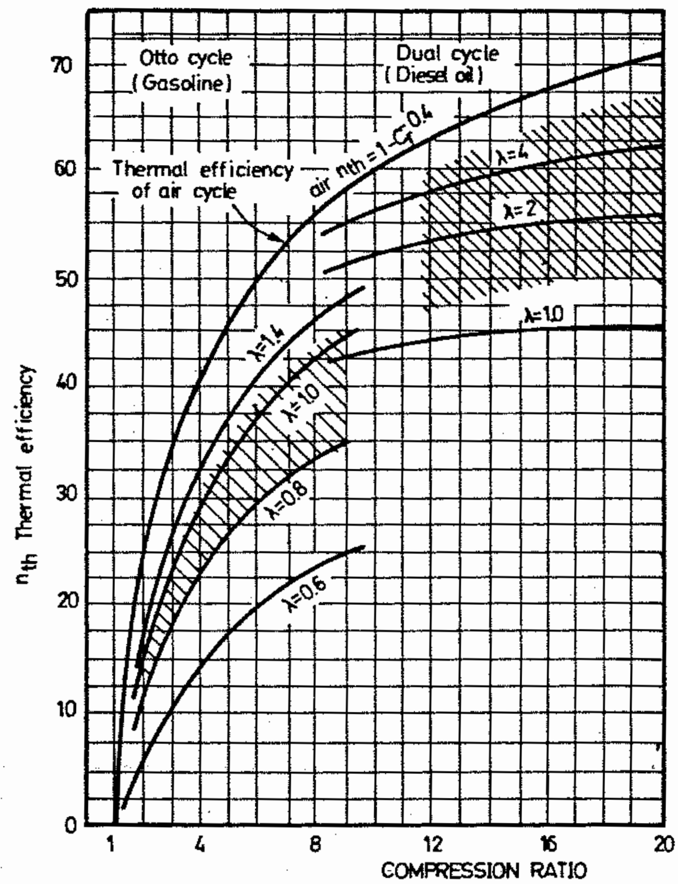


Fig.3.2.1: Thermal Efficiency vs Compression Ratio and Air/Fuel Ratio.(Ref: 2)

3.3. DUAL-FUEL OR STRAIGHT REPLACEMENT.

In a Dieseloil motor the low cetane number of methane does not permit a compression self-ignition. So, it is necessary to seek for an external ignition (spark plug, hot surface, glow plug) or to ignite the mixture with a torch or by self-ignition of an auxiliary high cetane number fuel. This means a dual-fuel system where the Dieseloil supply is maintained, but reduced to a minimum. Simultaneously seek the gaseous fuel is introduced premixed with the air (Fig. 3.3.1.)

On the other hand, the high octane number of biogas permits, in gasoline engines, the combustion with a higher compression ratio (which means, better efficiency, but also internal modifications, and therefore additional COSTS), Fig. 3.3.2.

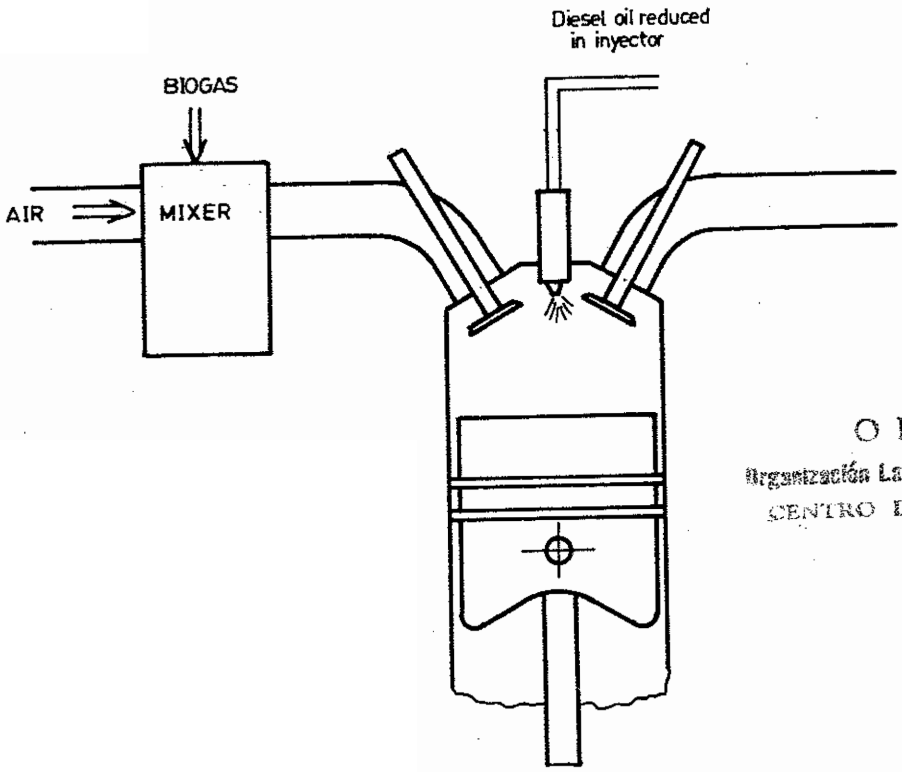


Fig.3.3.1: Dual-Fueling in Diesel engine. (Ref: 1)

The knock limit and ignition limits of methane make possible a much better choice of engine operating point. Note how the lean fueling limit is independent of CR for a gaseous fuel such as methane. Since the fuel is a gas, mixing is better. The lean boundary for gasoline which seemed to vary with compression ratio is not a fundamental limit of the fuel but, rather, is a result of poorer mixing of gasoline at low CR.

From the engine point of view, natural gas provides an opportunity to operate at high CR, with high efficiency, and with low pollution levels.

The fine air fuel distribution one can achieve in the cylinders eliminates the need for much of the paraphernalia that has encrusted current engines. Gaseous fueled engines permit the elimination of chokes, fuel enrichment devices at large throttle angle, accelerator pumps, and catalytic converters. The use of a gaseous fuel reduces oil dilution and prolongs engine life.

Methane is not a poor substitite for gasoline. Methane is a superb fuel.

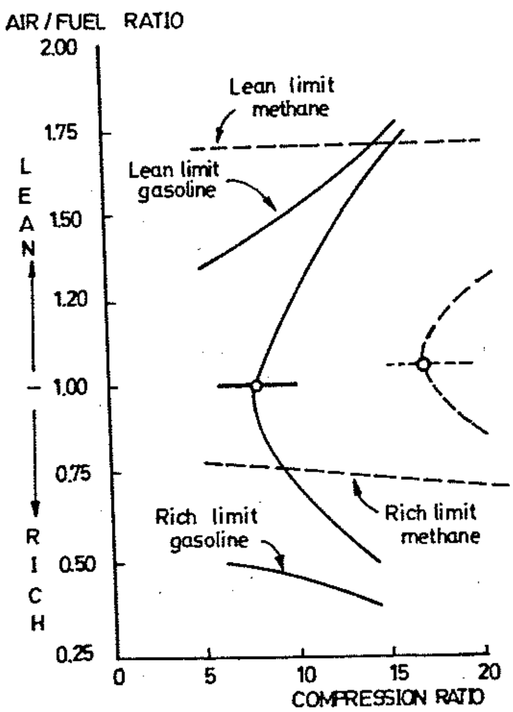


Fig.3.3.2: Ignition and Combustion Limits.(Ref: 2)

In gasoline engines straight replacement is recommended, but in some cases users prefer to maintain the original installation (carburetor) for the emergency case of failure. The biogas supply. The motor can be operated at will: with gasoline or biogas (Fig.: 3.3.3.).

If a two-stroke gasoline motor is lubricated by a mixture of lubrication oil added to the gasoline it is possible to supply biogas but not to replace totally. The gasoline, since otherwise the lubrication could not take place. So the following arrangement is proposed in figure 3.3.4.

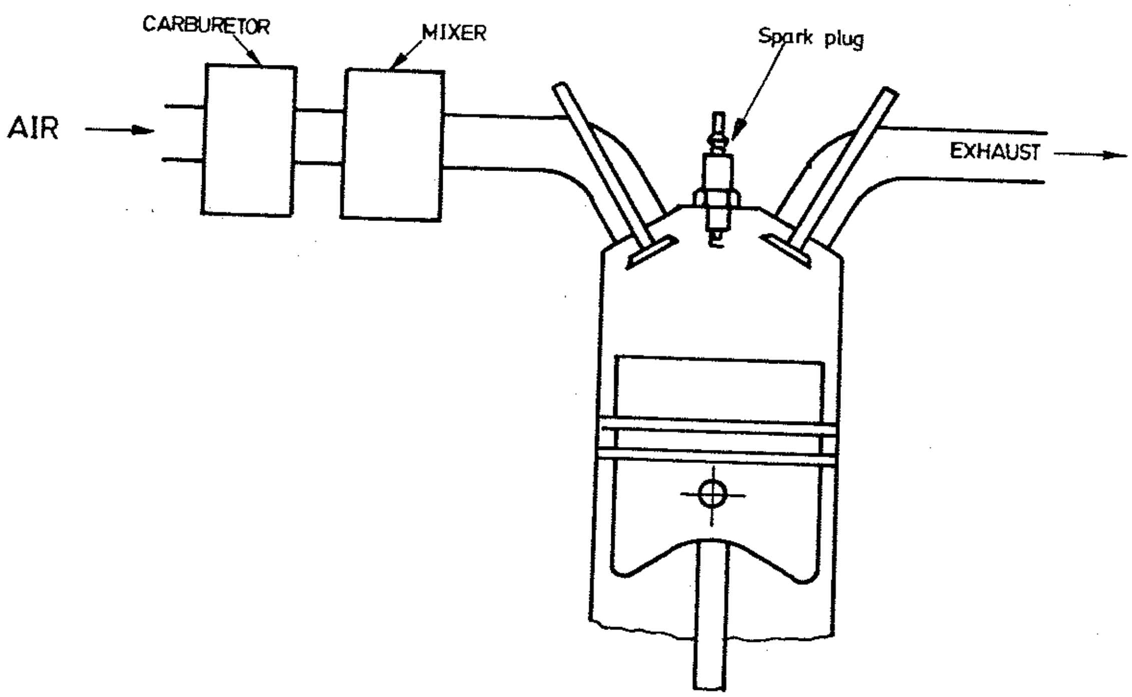


Fig.3.3.3: Spark ignited engine.(Ref: 1)

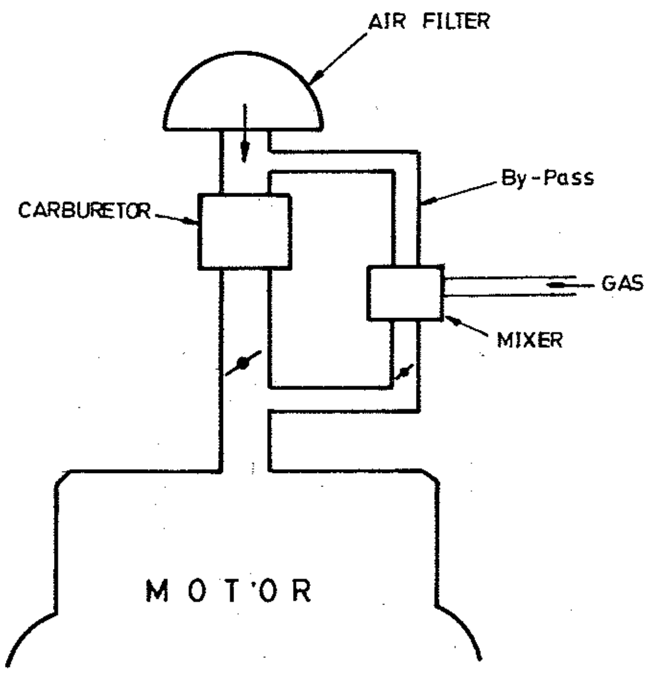


Fig.3.3.4:By-Pass proposed for two stroke engine (When mixtures lubrication Oil/Gasoline is used).(Ref: 1)

4. BIOGAS QUALITY AND PURITY.

The methane content of biogas varies between 50% and 70% (volumetric) when produced and therefore the heating value varies as can be seen in Fig.: 2.2.1., the higher the methane content the more interesting it is, specially for vehicle applications where we have to transport our fuel.

There are some processes to wash out (Scrubb) or separate carbon dioxide, but we also can use biogas almost as it comes as long as we take care to eliminate:

- Particulates
- Sulfhydric acid (H_2S)
- Excess water.

Also we should provide constant and sufficient pressure matching design conditions.

4.1. BIOGAS TREATMENT.

In the CRC publication "Fuel Gas Production from Biomass", Edward Ashare presents a very complete and organized survey
The different processes can be classified as:

a) Chemical solvents

Amines (Monoethanolamines, diethanolamines, triethanolamines, diglycolamine, diisopropilamine, and mixtures with or triethylene glycol).

Potassium Carbonate (Different processes, lycensed as: Benfield, Catacarb, Vetrocoke, Lurgy and others).

Other chemical solvents (Sulfinol, Alkazid, etc.)

b) Physical solvents

One of the most economical and widely used is scrubbing with water at an intermediate pressure (7 to 34 bar).

Others are (Fluor solvent, Purisol, Rectisol, Selexol).

c) Adsorption on solids

Many commercial adsorbents suit the needs, among them: activated Carbon, alumina and some molecular sieves.

d) Membrane separation

Still not very popular, but with a promising future because of its compactness and easy operation.

e) Distillation under cryogenic conditions.

When combined with liqueafaction becomes very interesting, but it is still too expensive.

After any of these processes the gas cames out with 95% to 100% of methane, ready for pipeline transportation, pressurizing up to 200 bar or even for cryogenic liquefaction (-160 C and 2 bar), whichever procedure suits the particular use of the gas.

For the use of biogas in stationary Diesel engines the gas treatment is not so necessary, because of the following reasons:

- The stoichiometric air to biogas volumetric ratio usually being around 6:1, implies that the increase of inerts in the mixture due to the $CO2$ is less than 1/12, when compared to pure methane firing.

- The fact that around 15% Diesel fuel is injected for ignition and that values of $\lambda > 1$ are used, further minimizes the effect of the presence of $CO2$ in the Biogas.

- If low H_2S content in Biogas is found, say less than 0,2% (as H_2S, v), this one is significantly smaller than the tollerated sulfur content in Diesel fuels, wich according to DIN and ASTM standards ranges up to 0,7% (as S, m) for heavy fuel oils.

However, for vehicular use, where compression of the fuel gas is necessary, the ellimination of a 40% of inert CO_2 is convenient.

The cost of compression, or cooling to cryogenic liquefaction levels, economically and practically justifies a previous gas treatment process.

The amounts of H_2S are simultaneously elliminated by most of the processes at no additional cost, with the extra gain in the deodoryzing of the fuel gas.

4.2. PARTICULATES

Only in sanitary landfills this problem will need a special treatment. In biodigesters of low production rates almost cero particulates reach the gas ducts.

4.3. HYDROGEN SULFIDE

There are some recomendations for the elimination of H_2S It was published one filter as can be seen in Fig. 4.3.1.

Reactions in solid adsortion:

$$Fe_2O_3 + 3H_2O \longrightarrow 2Fe(OH)_3$$

$$2Fe(OH)_3 + 3H_2S \longrightarrow 2FeS + S + 6H_2O$$

Regeneration: Exposition to air

$$4\ FeS + 3O_2 \longrightarrow 2Fe_2O_3 + 4S$$

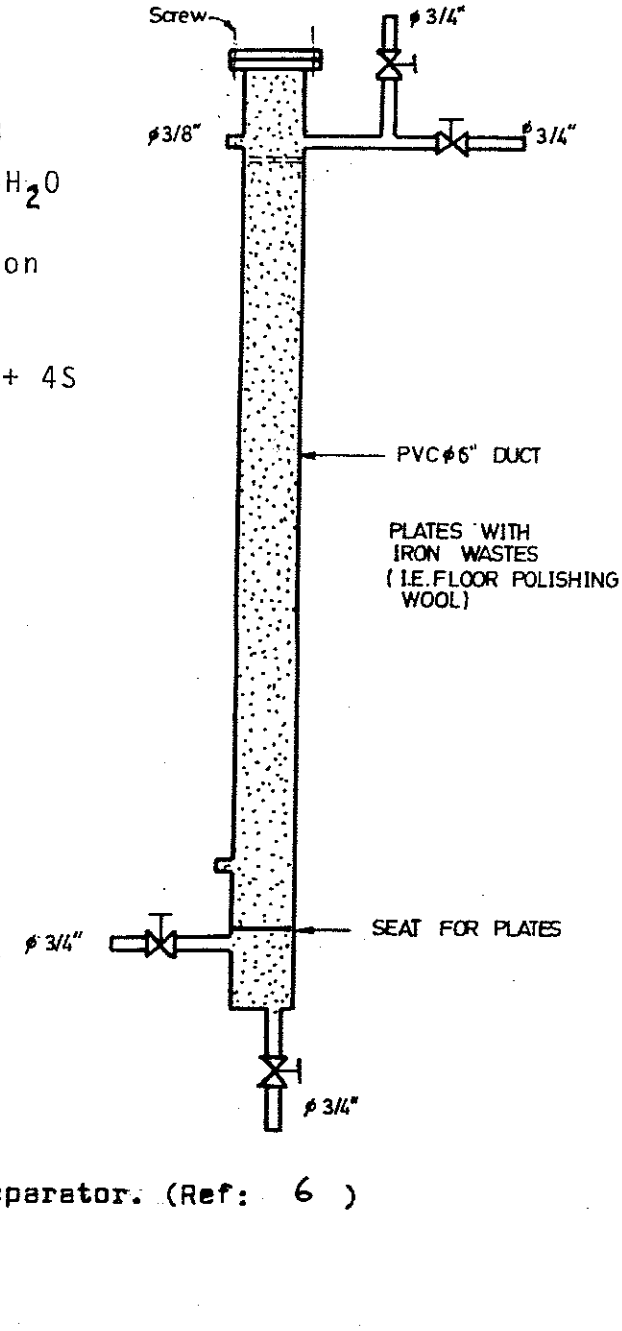


Fig. 4.3.1: H_2S, Separator. (Ref: 6)

Hydrogen sulfide (H_2S) may be present in biogas in concentrations from 1.500 to 5000 ppm $\hat{=}$ 0,15-0,5% vol 2,1-7g/m^3. Corrosion is due to H_2S and its subproduct SO_2, specially in non-ferrous metals. The literature informs regarding motorlife reduction up to 15% of the usual one. Also acidification of lubrication oil, even special lubricants, reduces oil changes intervalls up to 1/5 of the normal one.

The exhaust track suffers the severest attack of corrosion (SO_2,H_2SO_4).

If acceptable operation, maintenance and investment costs are considered, a dry-desulfuration method is to apply. "Gate" proposes an adsortion filter. The adsortion mass contains iron in particular compositions. There exist some industrial products for that purpose, but also some typical earths contain sufficient iron in the required compositions.

Once adequately prepared this material is homogenously distributed on trays (20-30 cm) and introduced in the separator (Fig. 4.3.2.). The gas flow is a upstream flow. Once the mass is sulfitted it can be regenerated through exposition to air (Fig. 4.3.3.).

Using two or more separators a continuos biogas supply with reduced H_2S can be obtained.The used mass can be eliminated by returning it to the earth (anaerobic).

Separator design. For a given maximum velocity the gas volume flow determine the cross surface. The volume of the separator itself and therefore the cleaning-mass will give us the operation time until the next regeneration or until the exchange of the cleaning-mass. The calculation procedure is given in the following pages:

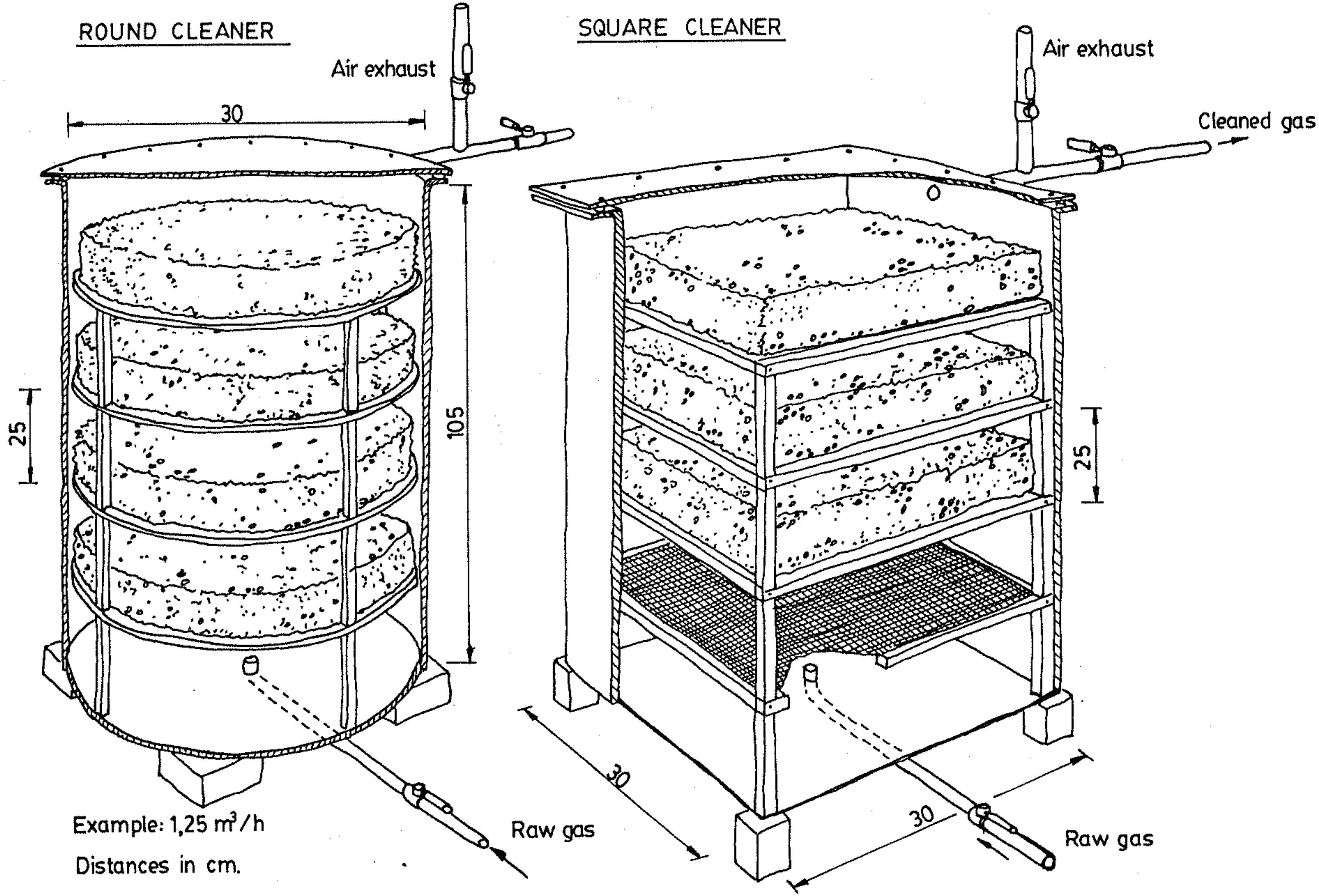


Fig.4.3.2: Round and square H_2S-Cleaner.(Ref: 7)

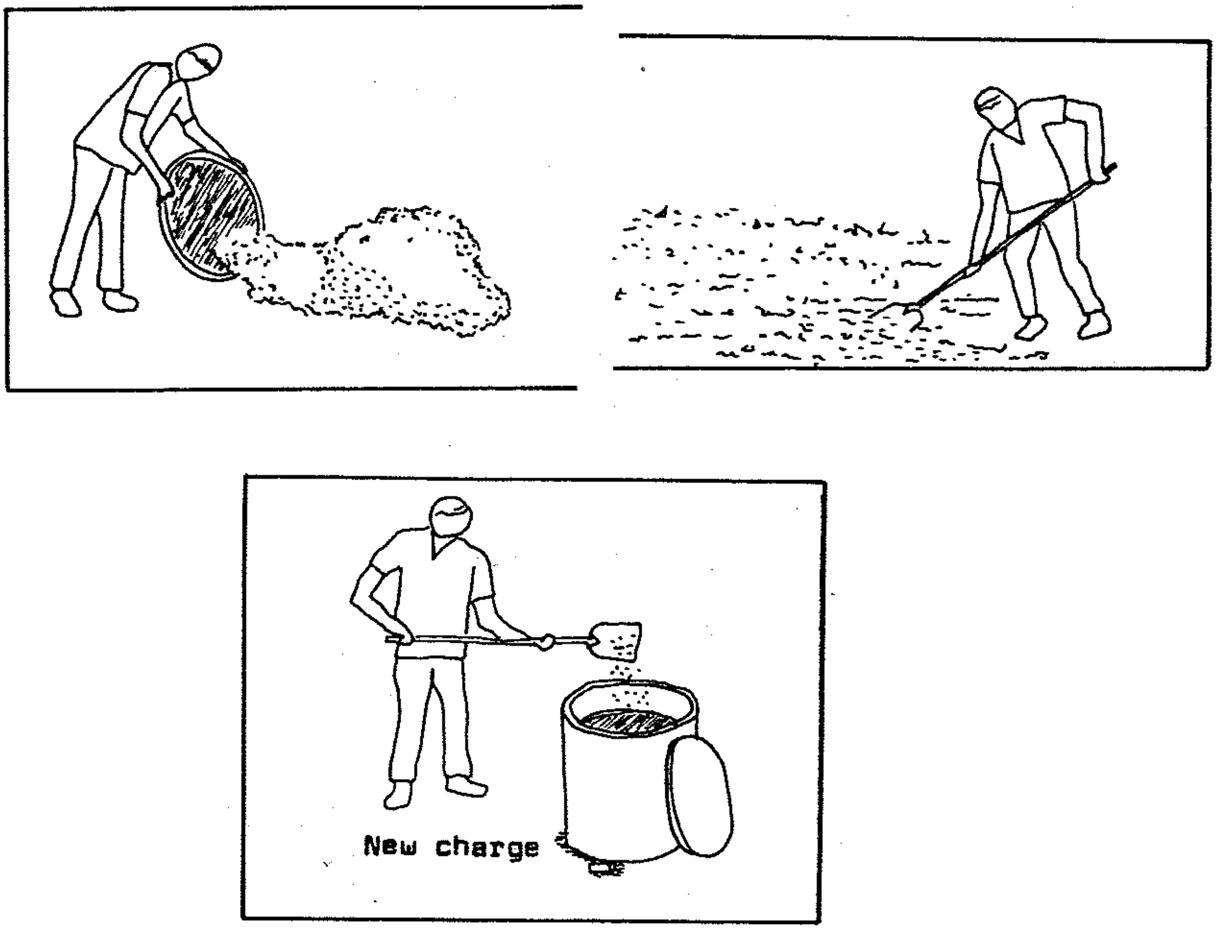


Fig.4.3.3: Sulfitted adsortion mass regeneration.
(Ref: 7)

BIOGAS SULFHYDRIC ACID ADSORPTION CLEANER
CALCULATION - PROCEDURE

1. CROSS SURFACE

$$\frac{m^3 Gas/h\ GASFLOW \times 1.000 \times 1.000}{3.600} = \_\_\_\_ \ cm^3\ Gas/s\ GASFLOW$$

[THE MAXIMUM GAS VELOCITY IS 0,5 cm/s]

$$CROSS\ SURFACE = \frac{cm^3 GAS/s,\ FLOW}{0{,}5\ cm/s,\ VELOCITY} = \_\_\_\_ \ cm^2\ CROSS\ SURFACE$$

IF CUADRATIC, $WIDE = \sqrt{CROSS\ SURFACE} = \_\_\_\_$ cm

IF CYLINDRICAL, $RADIUS = \sqrt{\frac{CROSS\ SURFACE}{\pi}} = \frac{CR.SU.}{3{,}14} = \_\_\_$ cm

2. HEIGHT

☐ m^3 Biogas/d GASFLOW

☐ g H_2S/m^3 Biogas (measured) or Assuming: 3 g H_2S/m^3

☐ days of operation time until next mass exchange or regeneration

____ m^3Gas/d x ___ gH_2S/m^3Gas x ___ days op.ti

= ____ gH_2S per operation time

Val es which can be assumed (but depending of type of mass) for captured H_2S in cleaning-mass:

a) without regeneration : app. 15g H_2S/kg mass
b) with regeneration : app. 150g H_2S/hg mass

$$\frac{\_\_\_\_ \ g\ H_2S/op.ti}{\_\_\_\_ \ g\ H_2S/Kg\ mass} = \_\_\_\_$$ kg cleaning mass/operation time until next mass exchange or regeneration

Density of cleaning-mass:

____ kg/l (measured) or assumed: 0,8 kg/l

$$\frac{\_\_\_\_ \ kg\ cleaning\text{-}mass/op.ti}{\_\_\_\_ \ kg/l\ density} = \_\_\_\_$$ l volume of cl.-mass/op.ti.

[Add for head, bottom and interfloor spaces: 2 5%]

VOLUME OF CLEANER:

_ _ _ l volume of cleaning-mass/operation time

\+ _ _ _ l (+25%)

_ _ _ _ l volume of cleaner x 1.000 _ _ _ _ m^3

HEIGHT:

$$\frac{\text{Cleaner volume}}{\text{Cross surface}} = \frac{\_\_\_\ m^3}{\_\_\_\ m^2} = \_\_\_\_\ \text{m HEIGHT}$$

3. NUMBERS OF TRAYS.

$$\frac{\text{HEIGHT [m]}}{\text{0,25 [m] FLOORHEIGHT}} = \frac{\_\_\_\ \text{[m]}}{\text{0,25 [m]}} = \_\_\_\_\ \text{FLOORS}$$

[MINIMUM: 3-4 FLOORS]

4.4. EXCESS WATER AND HUMIDITY.

The excess of water can be reduced by cooling the gases in a recipient. Cool ground water can be used as the coolant media. (Fig. : 4.4.1. and 4.4.2.)

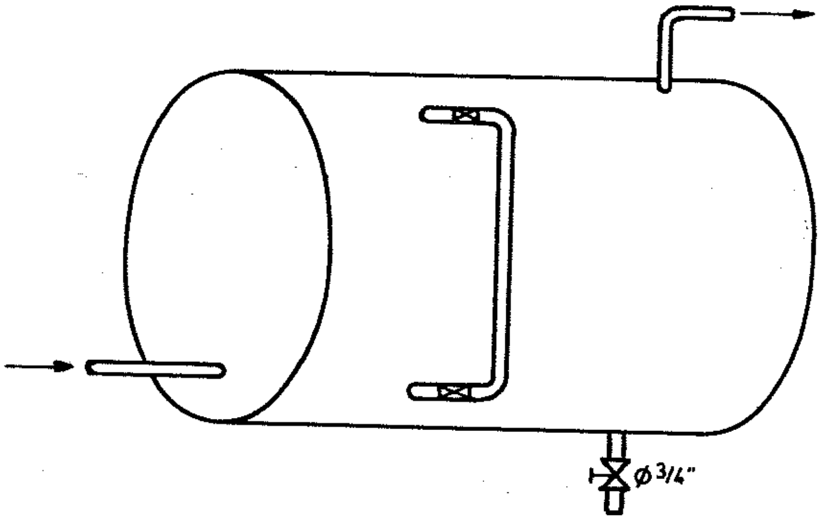


Fig.4.4.1: Condenser. (Ref: 6)

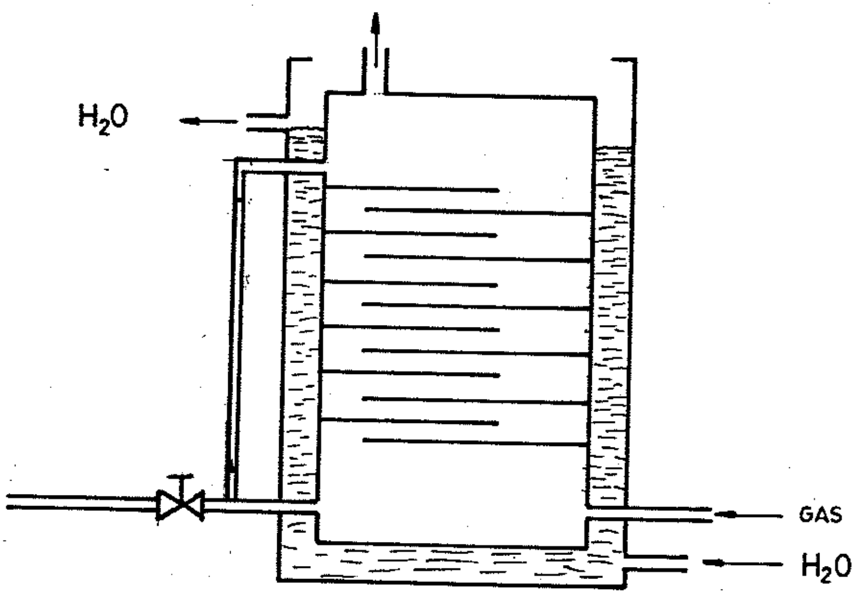


Fig.4.4.2: Water separator. (Ref: 15)

4.5. CONSTANT PRESURE

The traditional propane (LPG) kits may serve (Fig. 4.5.1.) including the key-element: pressure reducer air-fuel controller which is exactly matched to the mixer used, when biogas at high pressure is available. Some device must regulate the pressure, when only the digester is producing. It may be a compressor or a gasometer, for example a plastic bag (Fig. 4.5.2.)

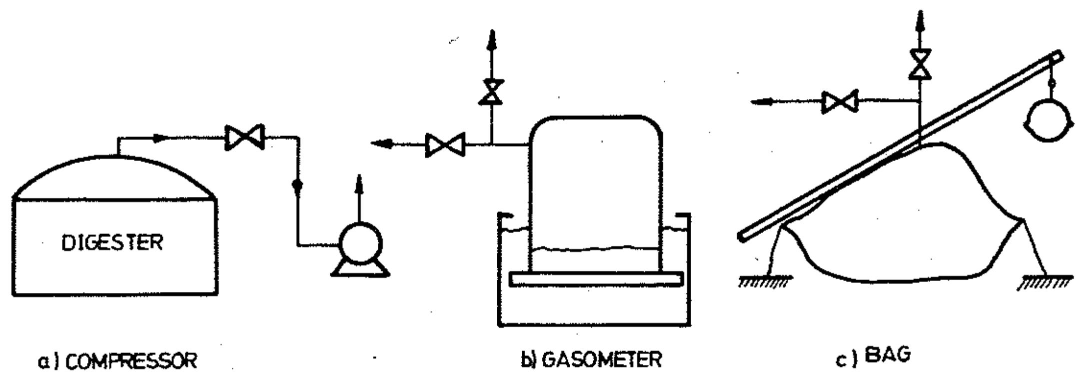
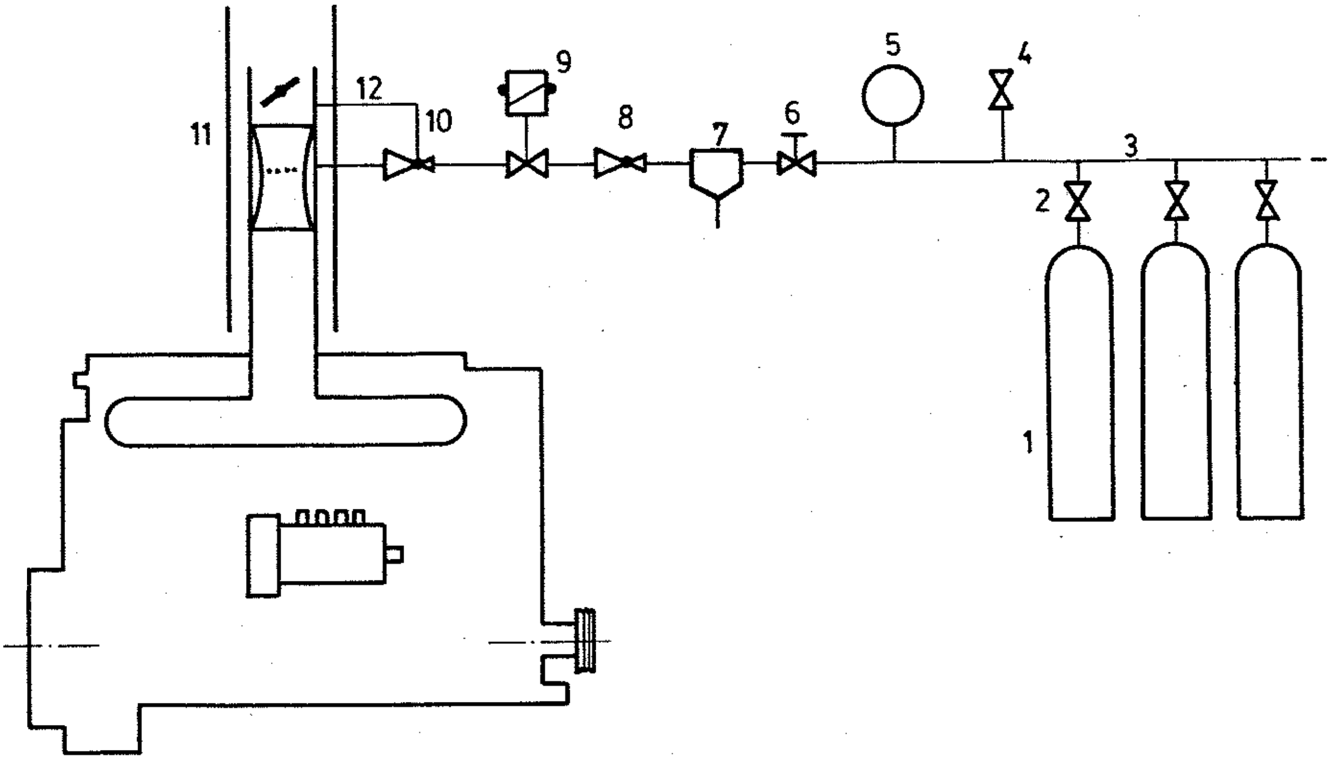


Fig. 4.5.2: Pressure regulation devices. (Ref: 1)



1. High pressure st orage bottles
2. Valve
3. Manifold
4. Charging valve
5. Manometer
6. Manual shut off
7. Strainer and moisture separator
8. First stage regulator
9. Magnetic valve
10. Second stage regulator (sub-atm)
11. Mixer
12. Pressure reference line

Fig. 4.5.1. TIPICAL VEHICLE INSTALLATION. [Ref4]

5. DUAL-FUELING IN GASOLINE TWO-STROKE MOTOR (LUBRICATED BY MIXTURE OIL/GASOLINE).

Assuming that lubrication is provided by oil/fuel mixture, a minimum oil/fuel mixture must be left enriching oil/fuel relation properly. If 20% of gasoline is supplied the oil quantity per gasoline should be multiplied by 5. A gasoline limiting device is needed, for example a reduced main metering jet diameter.

5.1. BIOGAS SUPPLY

Now, the remaining necessary energy supply through biogas can be approximately determined, assuming similar efficiency and supposing similar air/fuel ratios. The air relation between the gasoline/carburetor and the one through the by-bass (and the biogas/mixer) should be kept as 80 to 20. This can be achieved by a proper linkage between the main butterflies and also the original governor system may be used.

The final adjustments required must be performed in the field. Gasoline mass flow as well as output power and co-content in exhaust gas will make corrections necessary (carburetion and timing).

5.2. BY-BASS DESIGN

Depending upon space and materials aivailability the duct diameter should be similar to that of the air supply to the carburetor. Sufficient space for the p oper installation of the mixer and the main butterfly must be provided. The butterfly must be located in a way to permit the suggested linkage in between the desired ratio and to the governor.

5.3. MIXER DESIGN OR SELECTION

A Venturi will produce a depression in the throat when air flows through it. The higher the flow, the greater the depression. In this way the suction generated in the throat can serve to mix into the air the biogas. The relation throat diameter to duct diameter depends on the engine parameters. Rounded finishing for lower losses are recommended.

A simple mixer with different idling devices is shown in figure 5.3.1. for illustration.

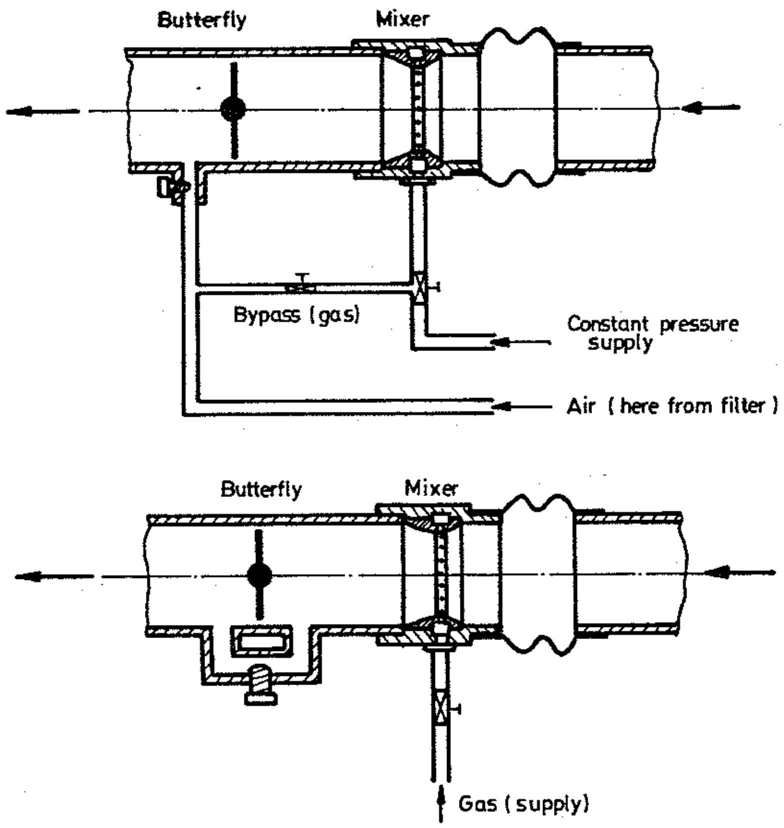


Fig.5.3.1: Different idling methods.(Ref: 8)

The proper blow-by recovery connections of another simple mixer is indicated in Figure 5.3.2.

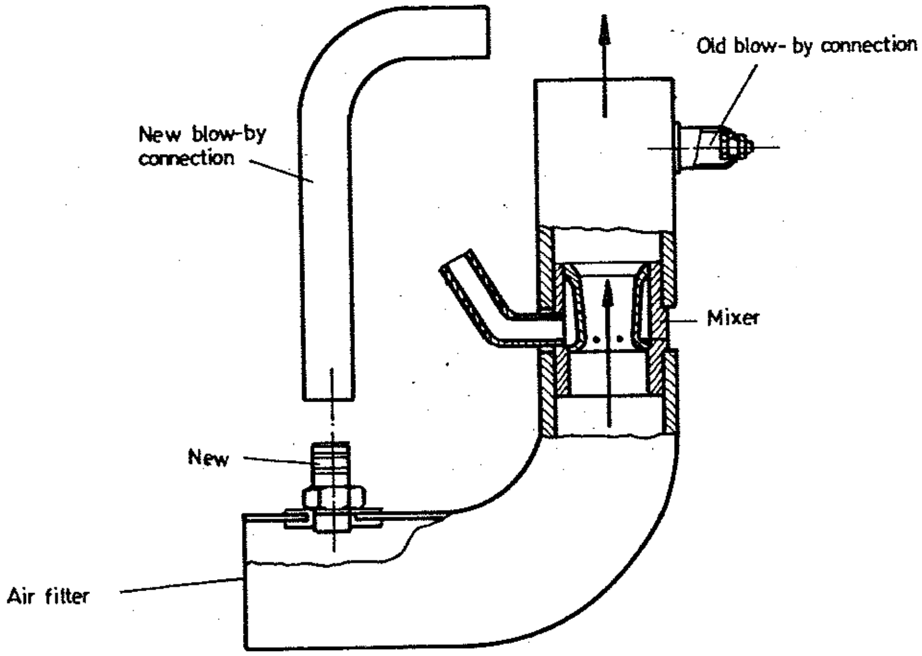


Fig.5.3.2: Schematic drawing of a simple mixer and proper blow connection.(Ref: 8)

There are also some designs with gas supply after the main butterfly of a carburetor with an additional gas flow regulation (Fig.: 5.3.3.)

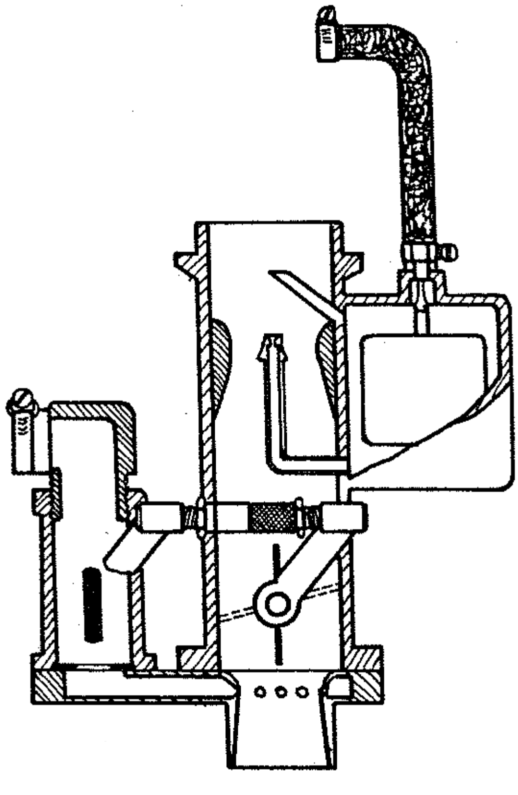


Fig.5.3.3: Scheme of a special system,after main butterfly (Necam,Century) with gasflow regulation. (Ref: 8)

Specially adapted systems are offered like that one in Figure 5.3.4. (Landi den Hartog).

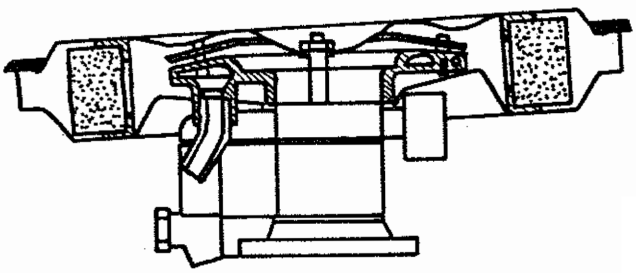


Fig.5.3.4: Special mixer (Landi den Hartog).
(Ref: 9)

Not recommended is the installation like Figure 5.3.5. is, because there is great a big explosive mixture volume in the filter.

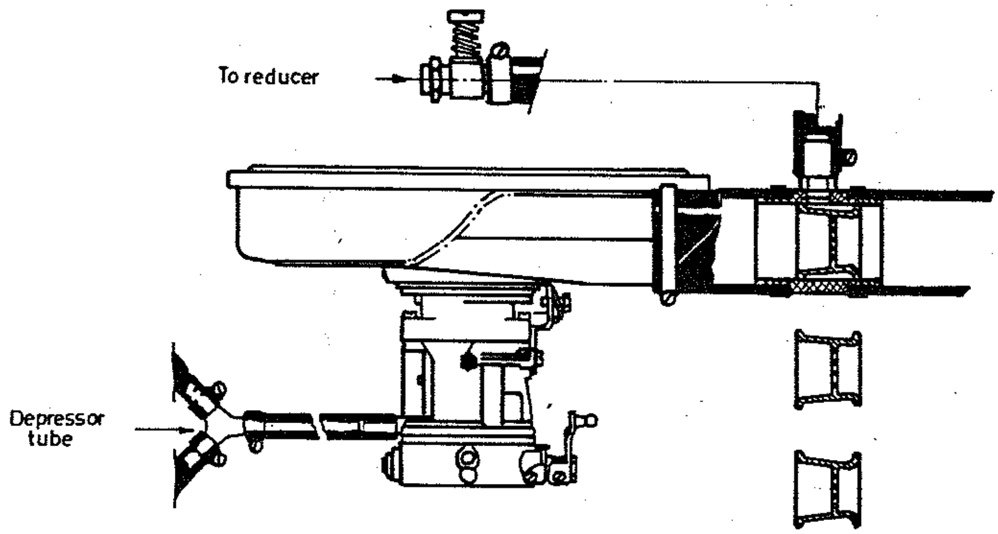


Fig.5.3.5: Not recommended mixer. (Ref: 10)

The mixer should therefore be installed between the filter and the carburetor (Fig.: 5.3.6.).

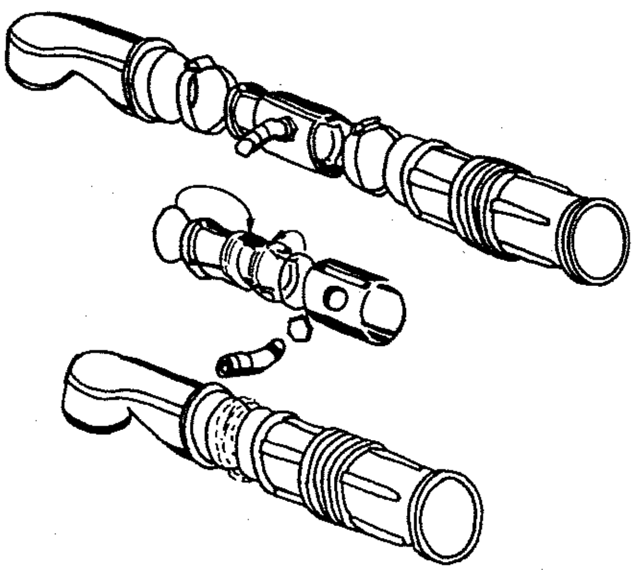


Fig.5.3.6: Mixer. (Ref: 8)

The simpliest mixer, for descendent one-throat carburetors, is displayed in Figure 5.3.7. with an inclination of 10°.

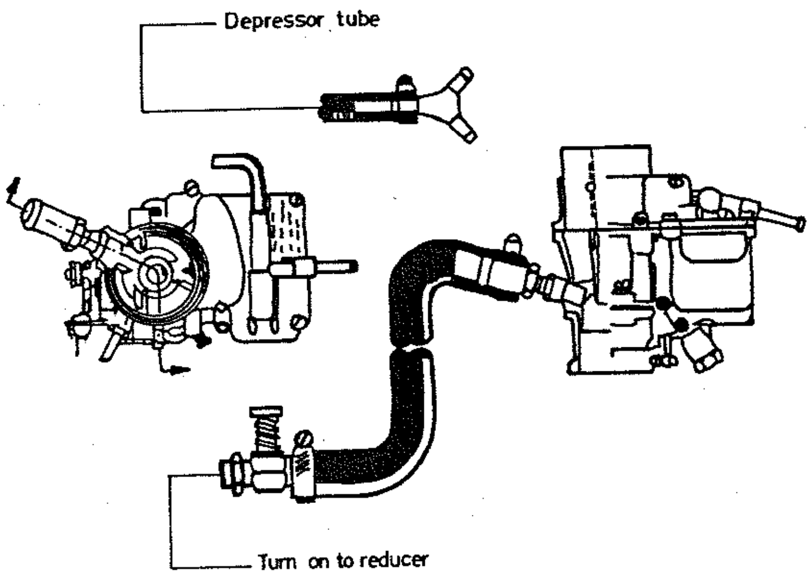


Fig.5.3.7: Simple mixer
(Ref: 10)

A straight gas-mixer is seen in figure 5.3.8.; it works together with a pressure reductor/regulator

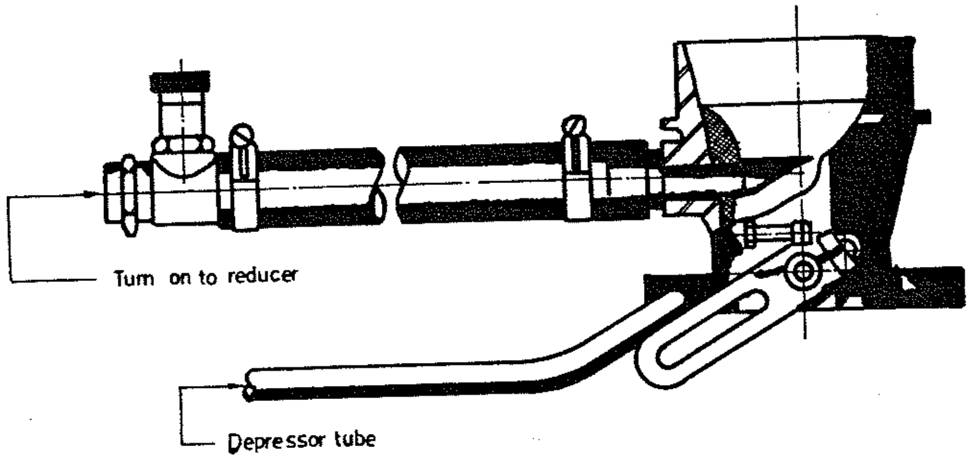


Fig.5.3.8: Gas - Mixer. (Ref: 10)

If there is enough space and material to allow boring, the original carburetor venturi may be used and modified. (Figure: 5.3.9)

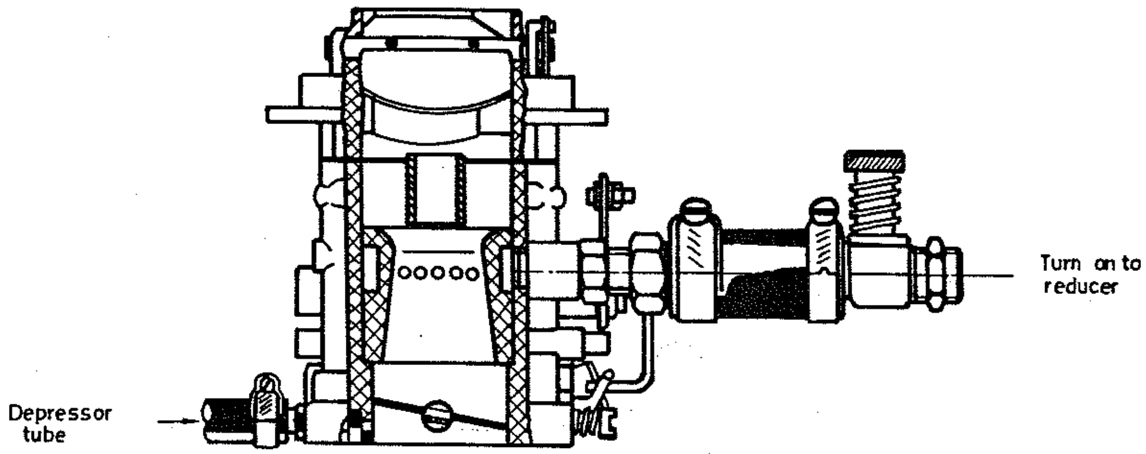


Fig.5.3.9: Venturi modified.(Ref: 10)

5.4. SIMPLIFIED MIXER CALCULATION

Continuity: $\dot{V} = \frac{\pi d_A^2}{4} c_M$

$\dot{V}$ = Air volume flux (volume per time)

d_A = Air ventury diameter (see Fig.5.4.1)

c_M = Mean velocity (practical values: 1 CYL = 75 m/2, 2 ÷ 3 CYL = 100 m/s 4 and more CYLS = 120 m/s)

Motor Suction: $\dot{V} = V_D \cdot n \cdot i$ $\qquad V_D = Z \cdot V_d \quad V_d = \frac{\pi D^2}{4} \cdot s$

V_D = Displaced volume (total)

n = Rotacional velocity $\qquad \dot{V} = \frac{\pi D^2}{4} \cdot s \cdot n \cdot i \cdot Z$

i = Working cycles per rotation (4T=1/2, 2T=1)

Z = Number of cylinders

V_d = Displaced volume per cylinder

D = Cylinder diameter

s = Stroke

$$\boxed{d_A = D \sqrt{\frac{s \cdot n \cdot i \cdot Z}{c_M}}}$$, Venturi diameter.

Gas/air relation:

$$\dot{m}_A = \alpha_A \cdot A_A \cdot c_M \cdot \rho_A \quad \text{(A=air/G=gas)}$$

$$\dot{m}_G = \alpha_G \cdot A_G \cdot c_G \cdot \rho_G$$

$\dot{m}$ = Mass flux (mass per time)

α = Flux coeficient ($\alpha_A \approx 0{,}8 / \alpha_G \approx 0{,}8$)

A = Cross section

c = Mean velocity

ρ = Density

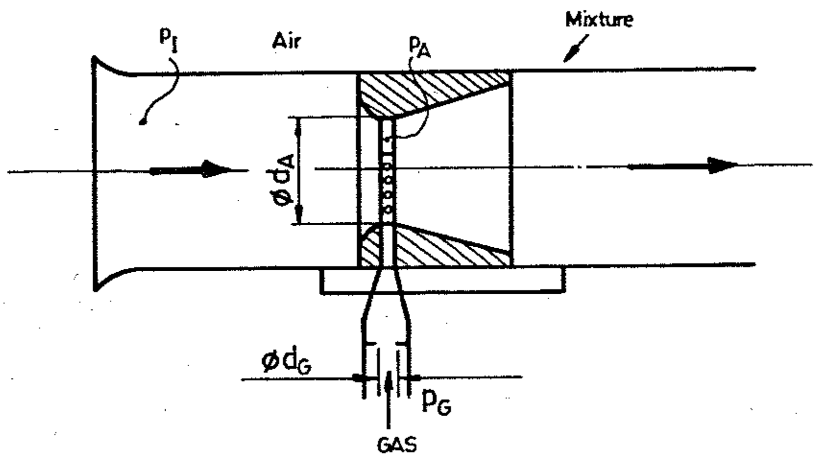


Fig.5.4.1: Mixer.(Ref: 11)

<u>Velocities</u>: $c_M = \sqrt{\frac{2}{\varrho_A}(p_I - P_A)}$. $c_G = \sqrt{\frac{2}{\varrho_G}(p_G - P_A)}$

$\dot{m}_A = \alpha_A \cdot A_A \sqrt{2\varrho_A (p_I - P_A)}$ $\dot{m}_G = \alpha_G \cdot A_G \sqrt{2\varrho_G (p_G - P_A)}$

<u>Air/fuel ratio</u>: $\dot{m}_A = \lambda \cdot A_{ST} \cdot \dot{m}_G$

λ = Air/fuel relation to stochiometric

A_{ST} = Stochiometric air (mass air/mass fuel)

Considering: $A_G = \pi d_G 2/4$ and $A_A = \pi d_A^2/4$

$$\boxed{d_G = d_A \sqrt{\frac{\alpha_A}{\alpha_G}} \cdot \sqrt{\frac{1}{\lambda \cdot A_{ST}}} \cdot \sqrt{\frac{\varrho_A}{\varrho_G}} \cdot \sqrt{\frac{p_I - P_A}{p_G - p_A}}},$$

Gas restriction diameter (if circular)

(For gasoline: $d_G \approx d_A$ (0,050 ... 0,053))

<u>EXAMPLE</u> : Motor with D= 0,10[m] s = 0,10 [m]
n = 5.000 [rpm] i = 1 (2 stroke) Z = 2
Let : c_M = 80 m/s

Venturi Diam. $d_A = 0{,}05 \sqrt{\frac{0{,}05 \cdot 5.000/60 \cdot 1 \cdot 2}{80}} = 0{,}0125\text{m} = 12{,}5$ mm

Gas Rest. Diam: Let: $\alpha_A = \alpha_G$ $\lambda = 0{,}95$

$A_{ST} \approx 6{,}5 \frac{K_g A}{K_g G}$ (60%CH_4) $\varrho_A \approx 1{,}28 \frac{K_g}{m^3}$ $\varrho_G \approx 1{,}2$ Kg/m^3(60% CH_4)

$(\Delta_p)_A = (\Delta_p)_G$

$$\boxed{d_G = d_A \sqrt{\frac{1}{1}} \cdot \sqrt{\frac{1}{0{,}95 \cdot 6{,}5}} \cdot \sqrt{\frac{1{,}28}{1{,}2}} \sqrt{\frac{(A_p)_A}{(A_p)_G}}}$$

$d_G = d_A \cdot 0{,}416$ (compare with dual-fueling diameters 2.3.2.)

5.5. INTAKE SYSTEM MODIFICATIONS

The only modifications recommended for the intake system are those that help keep temperatures down. The heat riser plumbing (if it exists) that feeds hot air from around the exhaust manifold into the air cleaner can be eliminated. The heat riser plumbing is necessary with gasoline to prevent stumbling and hesitation during the warm-up period and to control exhaust emissions. Because methane is already a vapor when it reaches the carburetor, the extra heat isn't needed.

6. 4 STROKE GASOLINE ENGINE. (Or 2 stroke with fuel-independent lubrication).

The foregoing limitation with respect to cero lubrication does not exist in this case. Therefore straight replacement can take place, but the possibility is also given to keep the original gasoline carburetor for emergency use. Proper shut-off valves must be installed in this case.

For mixer design and gas supply the same systems shown before are valid, but now we have the full air-flow going through the mixer. Adjustments have to be performed in the field.

6.1. PIPING

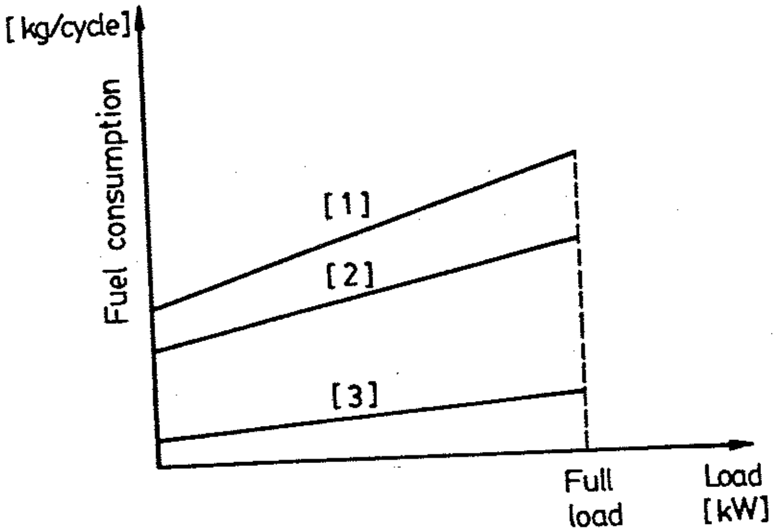
If high pressure is used, a pressure reducer ahead of mixer is needed.

If only very low pressure is available, as that generated by the digester (100-1000 mm H_2O), a constant pressure device (gasometer, bag) must be installed and sufficient wide ducts (for example: 25 to 50 mm Ø) and short distances (to reduce flow losses) should be considered.

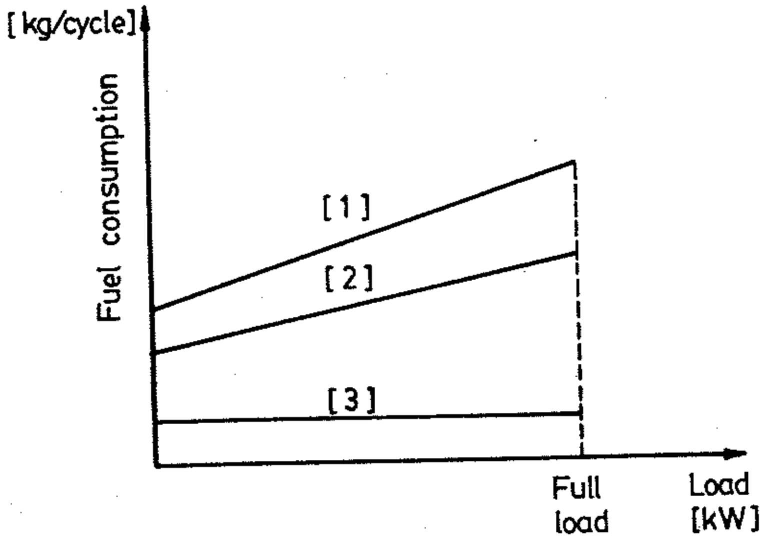
7. MODIFICATIONS FOR STATIONARY DIESELS

This chapter briefly describes the necessary modifications and devices needed to convert a standard diesel engine to dual fuel. The goal is to minimize the modifications of the engine itself and to use it like it comes from the factory. Two possibilities exist, when it comes to dual fuelling in the search of optimal diesel fuel substitution. One, consists in a constant diesel fuel injection, varying only the gas supply to meet the power needs of the engine. This, has the eventual dissadvantage, of overheating the injectors, and produces a poor atomization of the liquid fuel, when its contribution to the total mixture gets low. the second possibility is to maintain a constant diesel fuel/biogas ratio throughout the whole operation range. This permits the handling of the above described condition according to each particular motor.

Both replacement possibilities are shown in Figure 7.1.



Constant diesel fuel / biogas ratio.



Constant diesel fuel supply.

Curve [1] Diesel fuel at normal operation
Curve [2] Biogas supply at dual fueling
Curve [3] Diesel fuel supply at dual fueling

FIG. 7.1.: DUAL - FUELING GRAPHS [Ref. 12]

7.1. INTAKE SYSTEM

The only addition to the system is the air/gas mixer. According to gas feed pressure, two types may be designed: the positive pressure mixer, which consists in a direct connection to the manifold, and the second type consisting of a venturi tube which generates a depression which is function of the air flow. This one is neccessary if biogas is supplied by a subatmospheric regulator. A special case is constant speed and constant load operation (pump with constant height or generator with constant demand). Here a manual operated regulation valve may be enough to fix the energetic replacement ratio. Safety devices must not be forgotten.

7.2. GAS SUPPLY

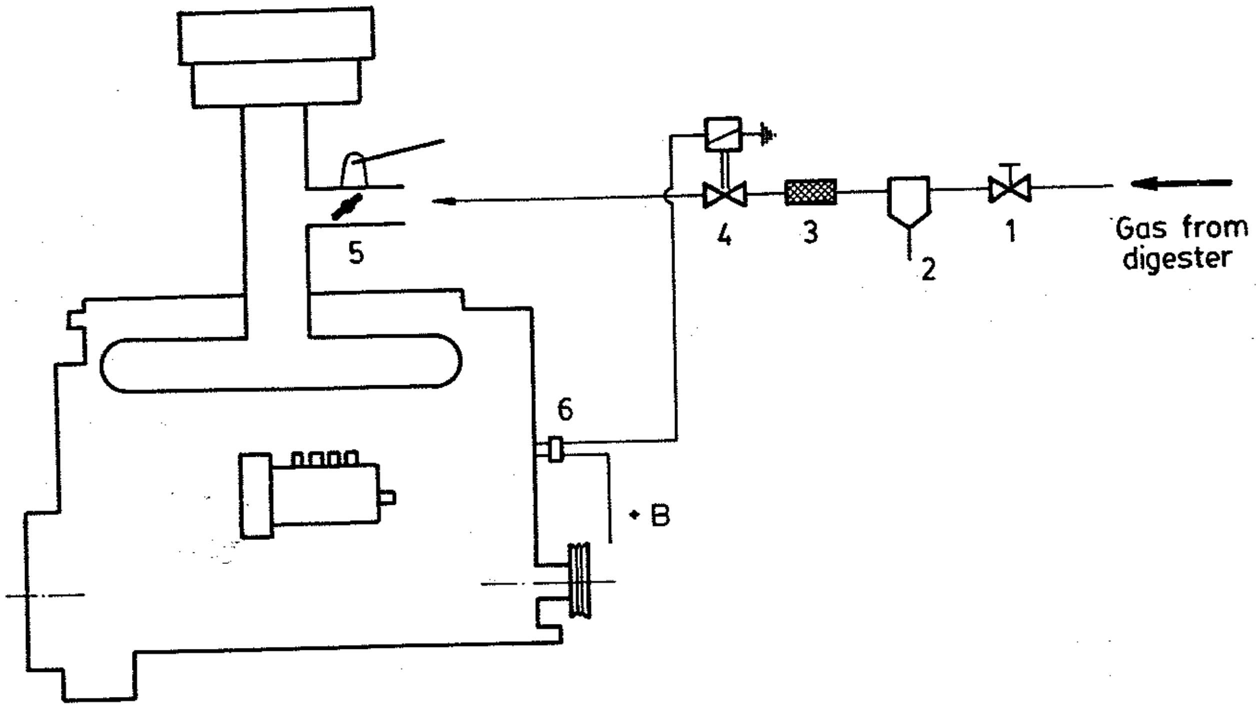
In most cases, stationary engines will be located near the gas source, therefore the system will be simplified to a few accesories. Among them a magnetic valve wired to a safety device is reccommended. Piping size is very important for a low pressure source and must be carefully calculated for minimal pressure drop.

If a sub-atmospheric regulator is used, combined with a venturi type mixer, the system will have a safety feature: gas will not flow when the engine stops or idles. A magnetic valve is not necessary for this purpose, however a second system for safety in a Diesel engine may be appropriated. (Figure 7.2.1.

7.3. INJECTION SYSTEM

Stationary engines in generator or pumping applications have a governor which regulates the injection for constant speed. Therefore no major modifications are to be introduced in this system, because, when biogas is supplied the speed tends to increase, which causes the governor to reduce the diesel fuel injection. The main task will be only to regulate the replacement ratio to the maximum permitted for that particular engine.

Anyhow, a diesel fuel overdose is convenient for starting



1 Cutoff valve
2 Moisture separator
3 Filter
4 Magnetic valve
5 Regulation valve
6 Safety device

FIG. 7.2.1. TYPICAL LOW-PRESSURE INSTALLATION [Ref. 4]

7.4. GOVERNORS

There are basically two alternatives for stationary engines powering independent generators or other machines. If the diesel oil consumption is limited to a fixed quantity, a new governor will be needed only for the gas valve to regulate according to the load demand.

For constant gas/diesel ratio the best choice (and the most expensive) is to use a strong governor, such as centrifugal-servohydraulic or alternatively some electronic type. These types of governos can be used succesfully to drive the fuel rack plus the gas valve and are not restricted to butterfly valves.

Another possibility using an inexpensive type governor, valid for whichever substitution ratio used, consists of keeping the original one in its original function. A second governor is needed to control only the gas. However, proper performance depends of a careful matching between them, to keep the desired gas/diesel fuel ratio.

7.5. EXAMPLE

One of our experiences was the adaptation of a diesel truck

Development of local technology for the conversion of automotive type Diesel engines was performed out. Specifica-lly, garbage collection trucks. The ultimate aim is to reduce atmospheric contamination and reduce consumption of the classical Diesel fuel.

The gas supply system was designed to deliver it into the inlet manifold and mix with the input air in the right amount, according to speed and load. This mixer is sub-atmospheric, for greater safety: gas will not flow if the engine stops.

The mixer posses two double butterfly type valves, mounted 90° one from the other (Fig. 7.5.1).

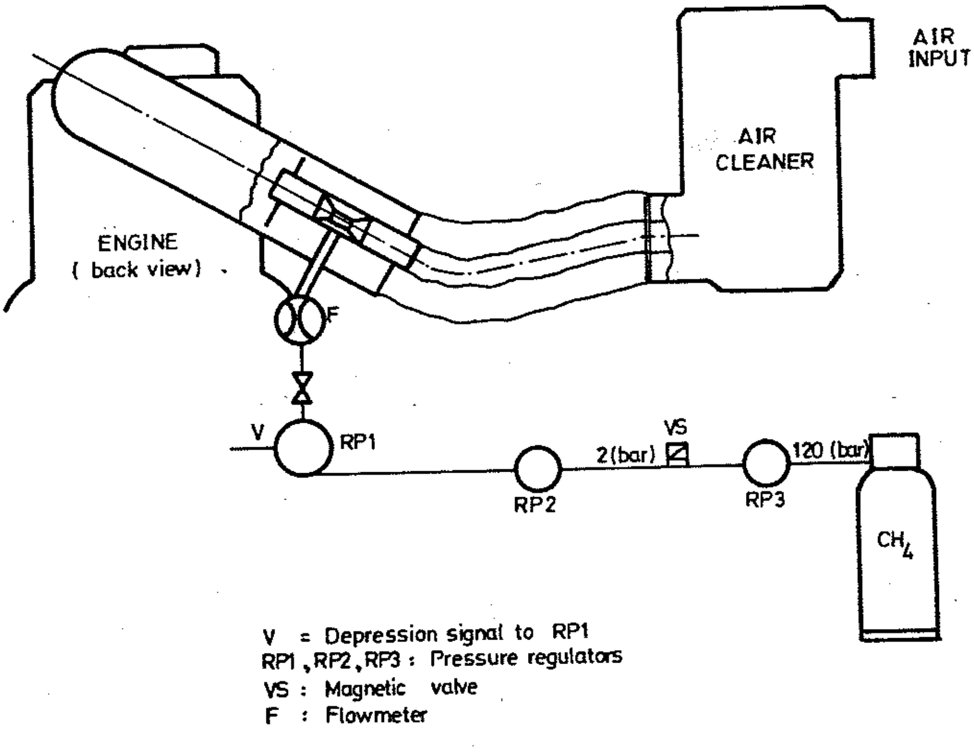


FIG.: 7.5.1 : GAS SUPPLY SYSTEM [Ref. 4]

Air flow is divided. Part of it passes through the inside venturi and the rest through the annulus cavity between the venturi and the duct. Each fraction of the total flow is controlled by each butterfly (Fig. 7.5.2.)

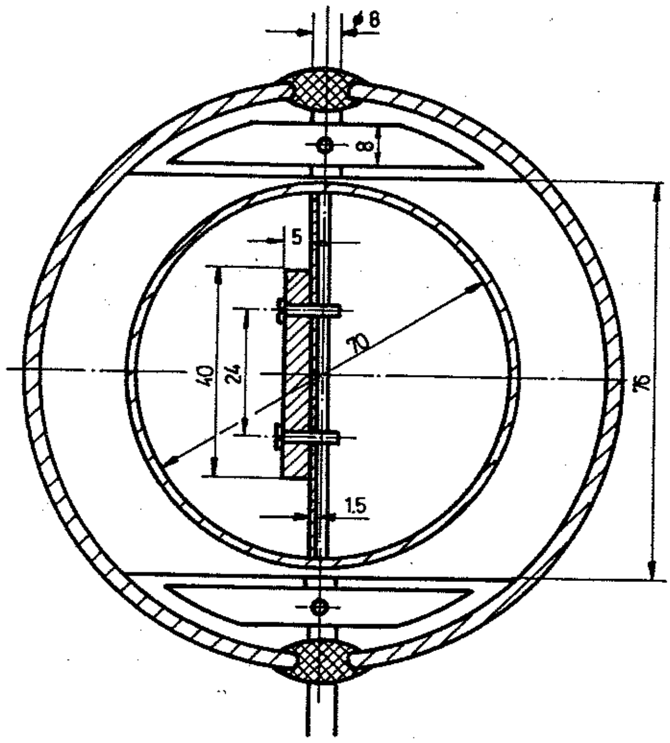


FIG.: 7.5.2 : DUAL BUTTERFLY VALVE [Ref. 12]

Engine operation in vehicular applications is considerably more complex than stationary power generators.

Apart of normal travelling, all other operating conditions must be taken into account: idling, deceleration, downgrade running and overspeed. In such cases, the RQ type governor (and most governors) cuts off Diesel injection completely.

To prevent explosion hazard and gas wastage, it is necessary to ensure gas ignition by allowing a small amount to be injected, or otherwise cut off gas supply when not required.

To meet these requirement, plus the contant gas/Diesel ratio concept, it was necessary to begin with two minor modifications to the injection equipment:

- A system to prevent zero injection (Fig. 7.5.3)
- A system to limit rack travel (Fig. 7.5.4)

The venturi admit gas flow in proportion to engine speed and, for a given speed, the double butterfly system alters gas flow according to engine load (throttle position).

Additionally, an electronic control circuit was designed and built for extra safety: it energizes the solenoid gas valve only when the engine runs within normal operating range. Its input signal is the AC voltage of the alternator. If frequency falls to idling speed or bellow, the gas valve shuts off. Also, if the frequency rises above maximum (overspeed condition) the valve shuts off again. Schematic diagram of this installation is also shown in Fig. 7.5.3.

Fig. 7.5.5. shows full load power consumption curves with Diesel only and dual fuel. It can be observed that maximum power obtained in both conditions are practically the same and the average energetic replacement is 85% over a wide range.

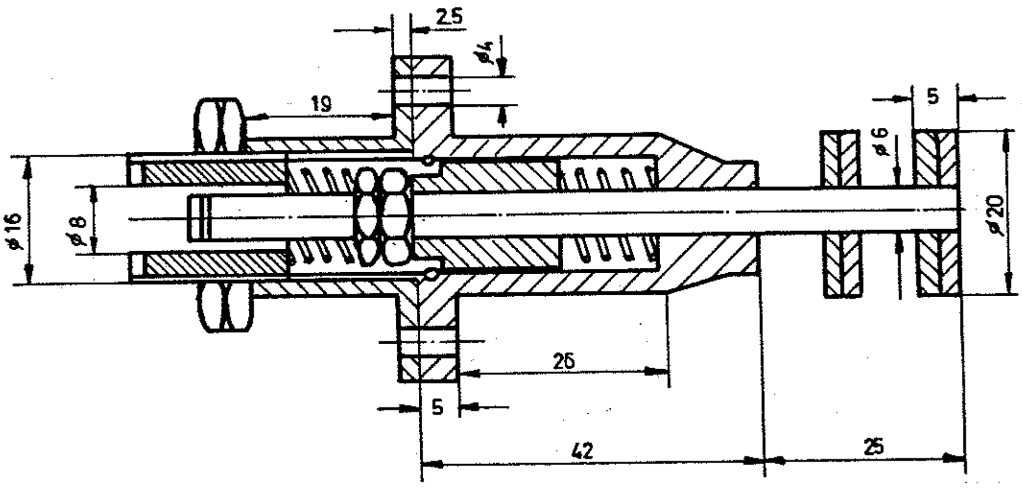


FIG.: 7.5.3.: RACK LIMITER (BOSCH) [Ref. 12]

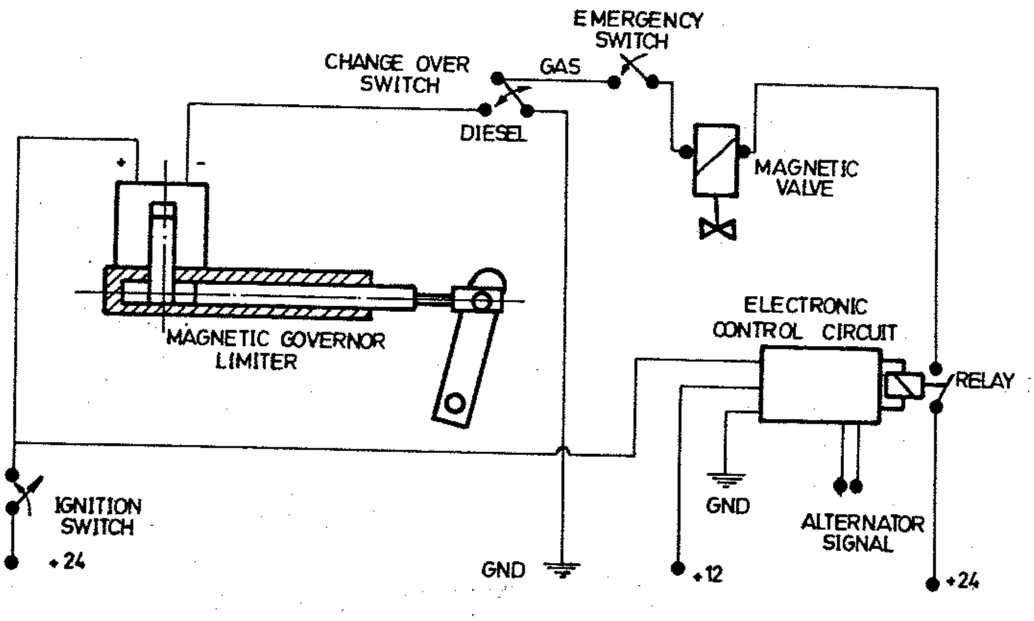


FIG.: 7.5.4.: GOVERNOR LIMITER AND ELECTRICAL CIRCUIT [Ref. 12]

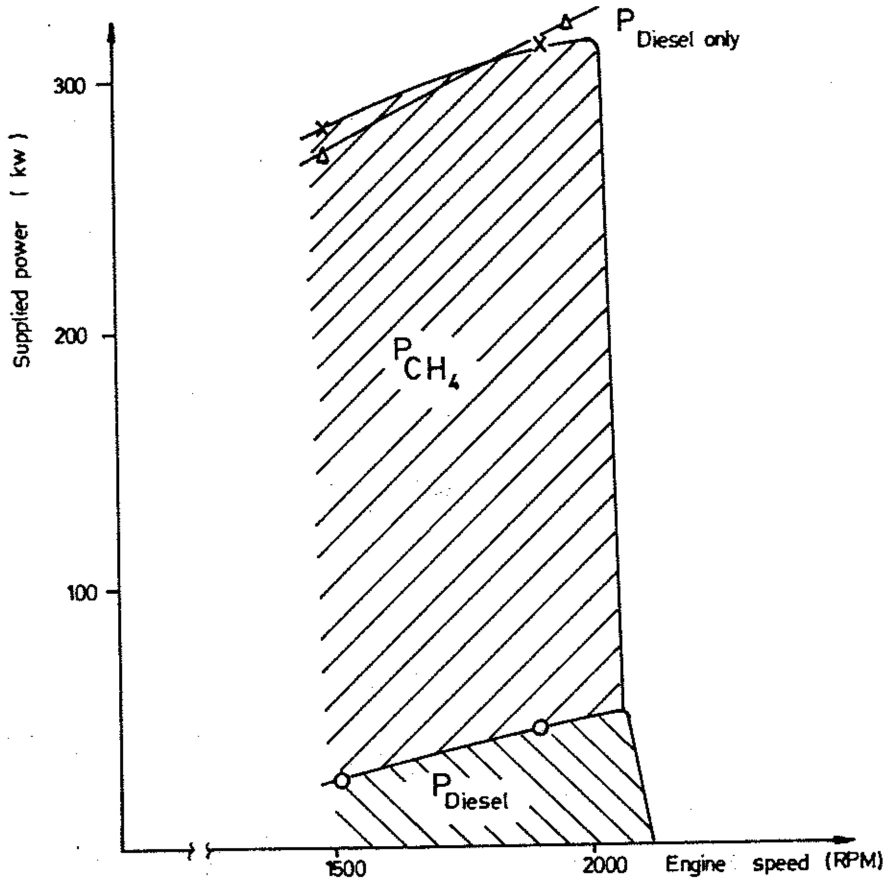


Fig.7.5.5: Power supplied at full load.(Ref: 15)

Also, exhaust emissions were evaluated by means of the "Bosch Smoke Number". Results in Fig. 7.5.6.

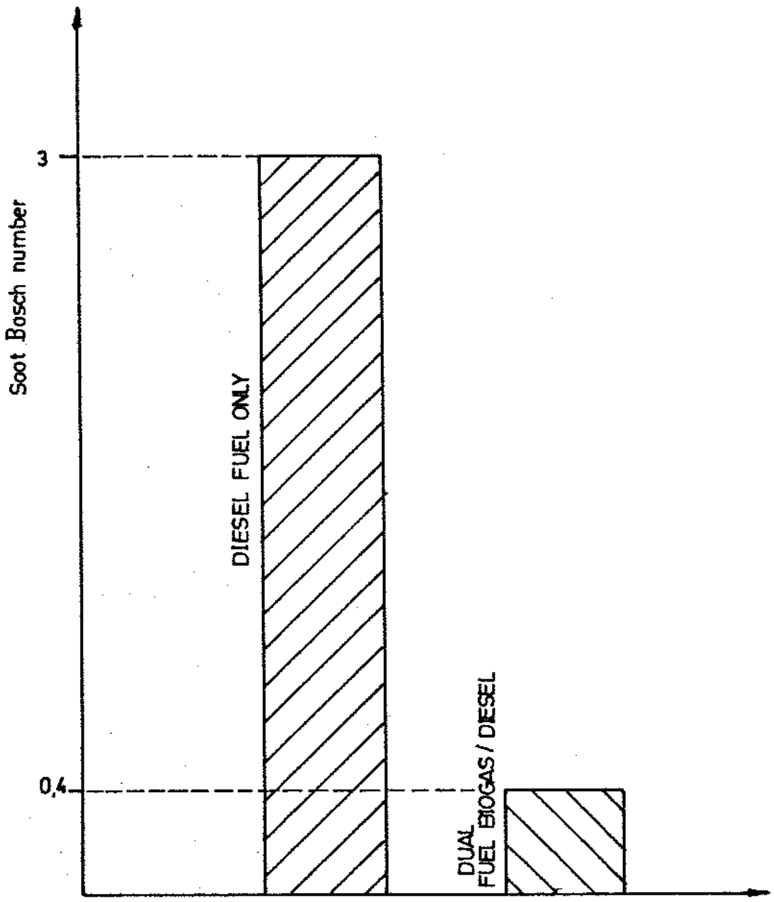
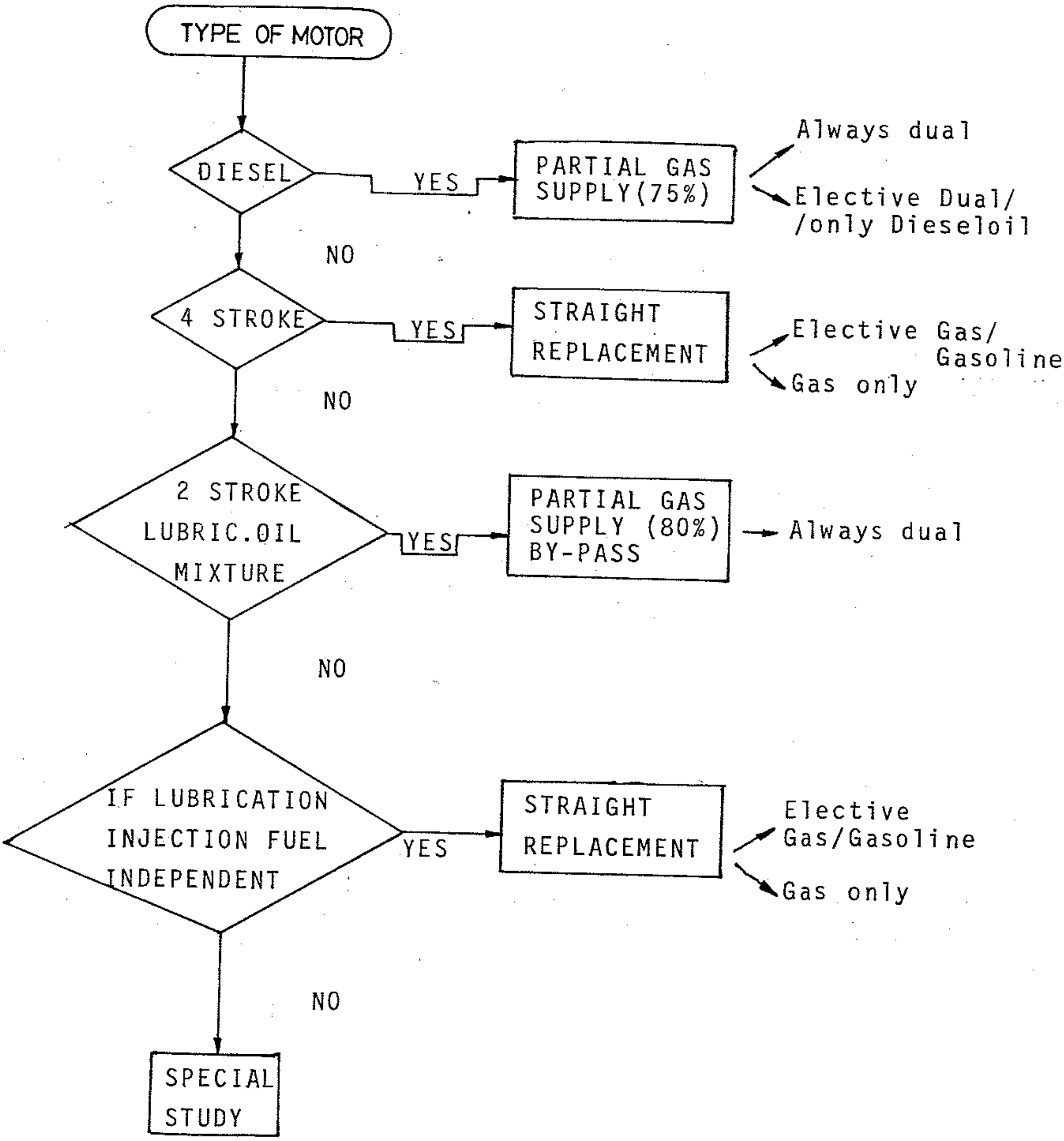


Fig.7.5.6: Soot emssion. (Ref. 15)

7.6. ADAPTATION TYPE DECISION PROCEDURE



8. SAFETY

A Biogas conversion is not something a novice mechanic should attempt without supervision or guidance. Biogas is flammable. One mistake while doing a conversion, resulting from ignorance, carelessness, or whatever, can transform an innocent-looking system into a potentially lethal bomb.

Attention has to be paid to the fact that the CH_4 individually has a density of 0,72 kg/m3, so that under segregation tends to raise. This is important when it comes to design of the ventilation in closed areas.

Explosive mixtures are formed with air, depending on the CO_2 content. (See Table T.8.1.)

Table T.8.1: Flammability ranges of Gas/Air mixtures[%v]

Hydrogen	4,1 - 75
Propane	2,1 - 9,4
Butane	1,86- 8,41
Methane	5 - 15
Biogas (55% CO_2)	11 - 33
(45% CO_2)	9 - 27
(35% CO_2)	7,7 - 23
(25% CO_2)	5,9 - 17,6

In Diesel dual fueling conditions, for safety and gas economy, the systems must meet the following conditions:

- Gas input to the engine must be prevented if no diesel fuel is being injected. Otherwise unburned gas will pass to the exhaust pipe with danger of explosion in the silencer, apart of the gas waste.

- Gas flow must be prevented when the engine is not moving. Otherwise, gas may come throug the air filter out to the atmosphere, with the danger of explosion in the engine compartment.

In other words, gas should be delivered only when the engine is operating within the normal working zone.

8.1. SAFETY DEVICES

Apart from the classical engine protections (oil pressure and water temperature), it is recommended to prevent gas from being released to the atmosphere. A sub-atmospheric regulator will provide this feature. Otherwise, another device, such as an electro-magnetic valve can be used. For alternators synchronized to the main supply line, it is convenient to wire the valve in such a way as to shut off gas supply if the generator becomes motorized or disconnected from the line. Also overspeed cero diesel oil injection must be avoided.

Some special features have been developed, called "sniffers", which sence the presence of gas.

To avoid back-fire in the intake system a proper backfire prevention mesh installation is recommended.

9. ADJUSTMENTS

Misadjustments of the carburetor or ignition timing are usually the result of sloppiness or "trial-and-error" tuning techniques. There's no substitute for proper tune-up equipment. Tuning by ear can't begin to match the accuracy of a timing light, CO analyzer, or dynamometer. Misadjusted timing and air/fuel mixtures can cause a variety of problems.

In generating sets and pumps the dynamometer can be replaced by an adecuate measuring device for the output power.

9.1. IDLE SPEED.

The idle speed on a straight conversion or a dual-fuel conversion should be set so that the engine idles at approximately the same rpm as before. Refer to the vehicle's service manual or a tune-up specification guide for the recommended rpm setting. For most engines, 600 to 800 rpm is the recommended setting.

To adjust the idle speed, connect a tachometer to the ignition system and start the engine. Turn the adjusting screw on the mixer until the desired rpm is achieved. Once set, the idle speed requires no further adjustment unless ignition timing changes are made.

9.2. IDLE MIXTURE

The idle mixture controls the air/fuel ratio when the engine is idling and at low speeds. The idle mixture adjustment is changed by turning a screw on the mixer (Fig. 9.2.1.).

To set the idle mixture adjustment, make sure the engine is at normal operating temperature. Next, use the lean drop method to find the smoothest idle. Turn the idle mixture screw

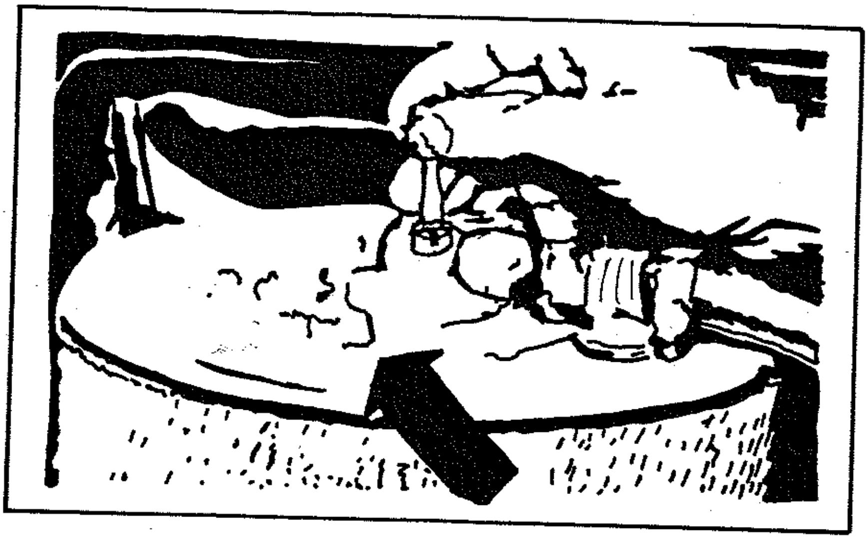


Fig.9.2.1: Adjusting the idle mixture on a dual fuel mixer. Shown is an Impco unit. (Ref: 13)

in until the idle speed begins to drop. Then back it out slowly until the smoothest idle is achieved. Readjust the idle speed as required to maintain the recommended setting.

If a CO exhaust analyzer is used to check the adjustment, it should about 1 percent carbon monoxide with the engine idling. A leaner mixture (lower Co reading) can cause a loss of power and possibly misfiring. A richer mixture (higher CO reading) will not get as good fuel economy and it creates more pollutants.

9.3. POWER MIXTURE

The maximum combustion temperature does not occur at the stoichiometric mixture but, due to dissociation and specific heat variation, it occurs at a slightly rich mixture. The maximum power is thus obtained at a slightly rich mixture.

The power mixture screw is used to adjust the air/fuel ratio for full throttle operation. Light load mixtures are controlled by the designed system and are generally not adjustable.

The power adjustment must be set with the motor operating under full load. This is best done on a dynamometer using a CO exhaust analyzer. The power adjustment depends on the type of motor and how it will be used. A heavy-duty truck operating with propane under continuous full-load conditions should be adjusted no leaner than 1.5 to 2.0 percent carbon monoxide. (Fig. 9.3.1.)

The thing to keep in mind is that leaner mixtures (lower CO reading) get better fuel economy and richer mixtures (higher CO reading) generate more power. Extremely lean mixtures can result in misfiring because the added air raises the voltage requirements of the spark plugs. If the spark can't bridge the plug gap, the mixture won't be ignited.

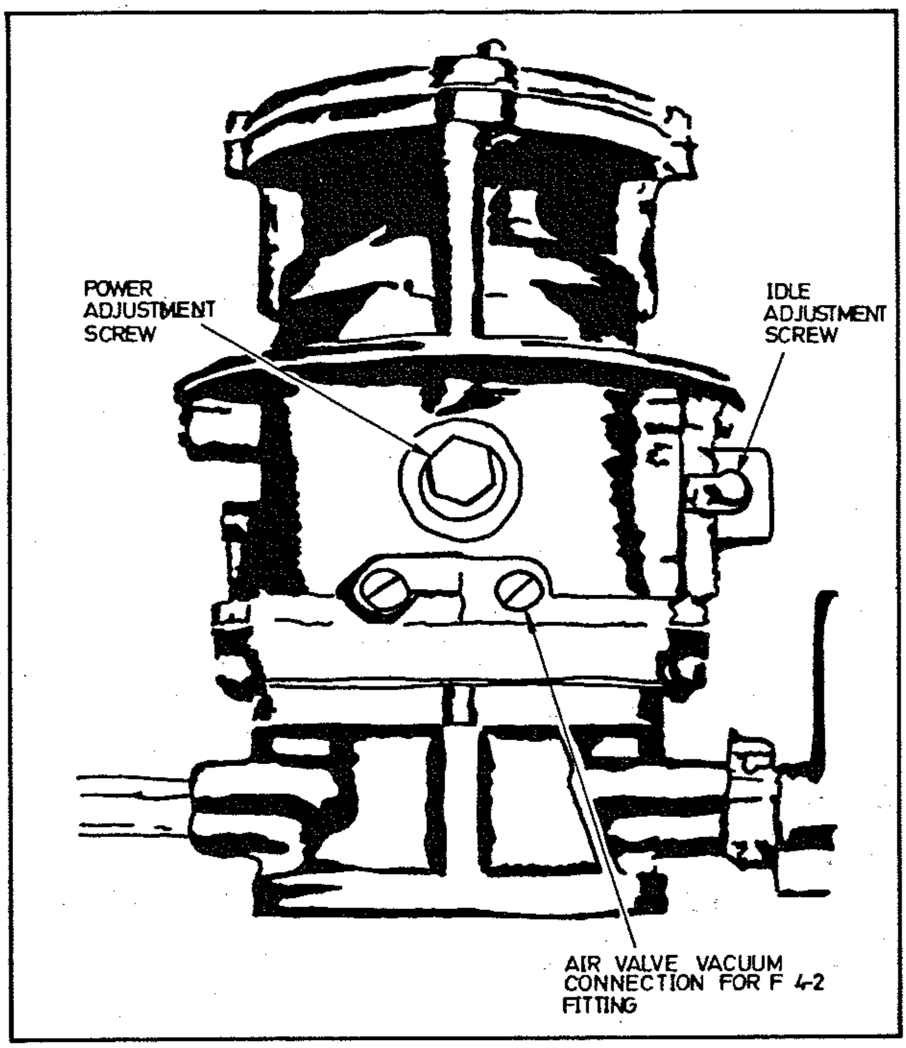


Fig.9.3.1: Idle and power adjustment points on an
Impco CA 425 propane carburetor.(Ref: 13)

Overly rich mixtures can elevate exhaust temperatures to the point where exhaust valve life is shortened. Too rich mixtures can also foul the spark plugs, waste fuel, and increase pollutants. About 3 percent CO is considered to be the maximum richness for optimum performance. Horsepower will begin to drop as the power mixture is richened beyond this point. For practical purposes, there is not much of an increase in power from 2 percent CO to 3 percent CO.

9.4 IGNITION TIMING.

Normal and abnormal combustion must be identified at first.

Weak-mixture operation results in a lower velocity of flame propagation, consequently less heat is converted to mechanical energy, and part of the additional waste heat from the exhaust products is retained in the cylinder walls causing the engine to run hotter and to be more prone to spark knock (see below). With extensive fuel weakening, the flame speed becomes so low that flame is still present when the inlet valve opens for the next admission, leading to back-firing into the inlet manifold.

When cylinder pressures and/or temperatures are particularly high, the unconsumed end gases may be stressed sufficiently to ignite spontaneously before the arrival of the main flame. This explosive form of combustion is termed spark knock, and promotes a sharp rise in pressure followed by violent vibrations of the gases, with sympathetic vibration of the engine components causing a sharp metallic noise. A further abnormal form of combustion arises with the high-compression engine when combustion deposits build up in the chamber. Under high-temperature operation, the deposits tend to glow, and to act as secondary sources of ignition. The fresh charge then ignites early, independent of the spark timing, leading to an uncontrolled over-advance

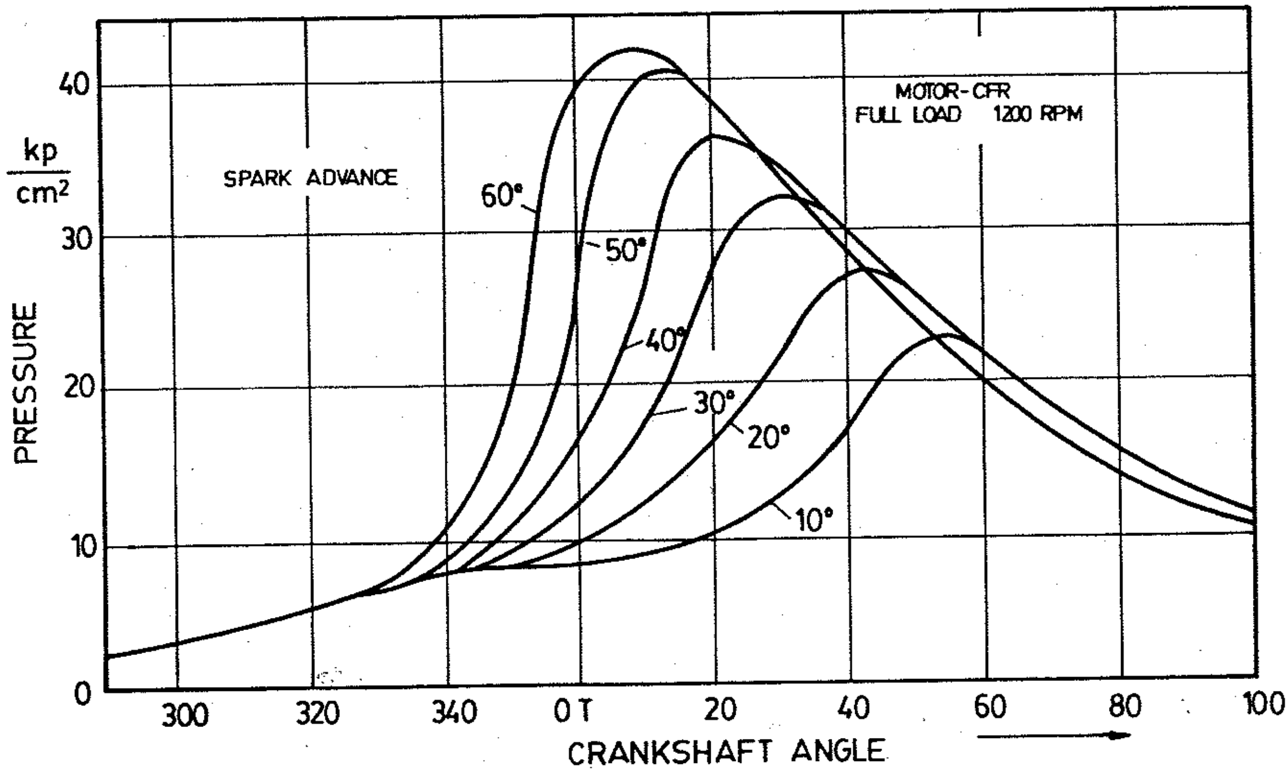


Fig.9.4.2: Influence of spark timing.(Ref: 1)

with consequent roughness and hot running. This low-pressure phenomenon is termed surface ignition, and is a function not only of the ignitability of the fuel, but also of the flame temperature of the fuel which had previously determined the temperature reached by the deposits. (Fig. 9.4.1.).

Next it is important to realize that there exists an optimum timing advance depending on fuel characteristics, rpm, combustion chamber temperature, load conditions, knock-retardant influences and others, which will give maximum power. (Fig. 9.4.2.)

Since the methane molecule is compact and very stable it is hard to ignite. Ignition temperature is about 1170° (Gasoline 570°F) Better Ignition and Timing Sustems are Needed.

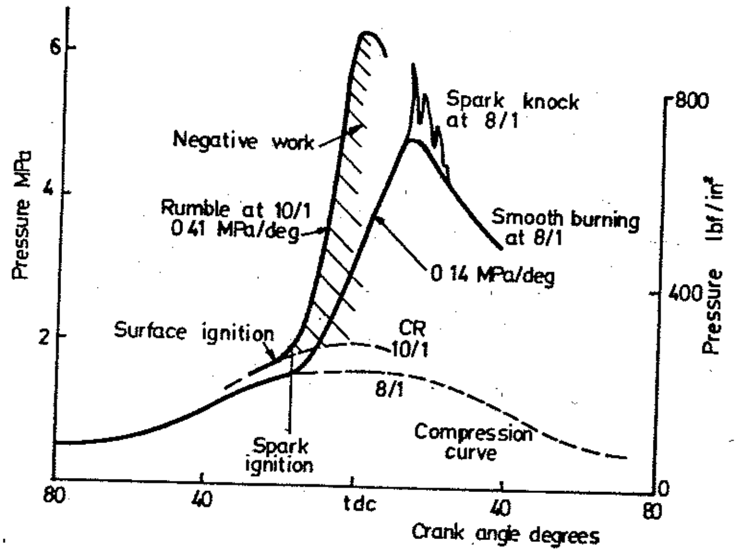


Fig.9.4.1: Pressure diagrams for normal and abnormal combustion in the spark ignition piston engine.(Ref: 3)

9.4.1. IGNITION TIMING FOR CONSTANT SPEED.

Some spark advance curve for each motor can be found as that one shown in Figure 9.4.1.1. for propane in an automotive engine.

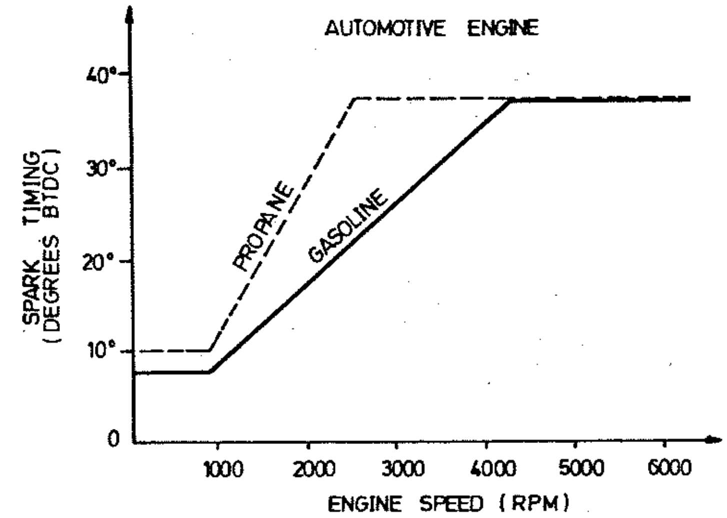


Fig.9.4.1.1: Comparison of spark advance curves for an automotive application. When converting to propane, the distributor should be recalibrated for faster spark advance, and the engine retimed for more initial advance. Total advance, however, remains about the same. (Ref: 13)

The advance curve is changed by installing heavier advance weights or lighter springs in the advance mechanism inside the distributor. Ideally, the distributor would be recalibrated on a distributor machine.

For constant speed applications the optimum for full load can be determined measuring the output power (maximum) as function of the advanced spark angle.

The effect on power due to a variation of the spark timing is shown in Figure 9.4.1,2. Note specially the relative insensitivity of power to spark timing with rich mixtures, and the extreme sensitivity at lean mixtures. Traditional operation of spark ignited engines with rich mixtures has permitted a rather imprecise control of spark timing to be quite satisfactory.

In lean operation good air/fuel ratio and timing control is necessary.

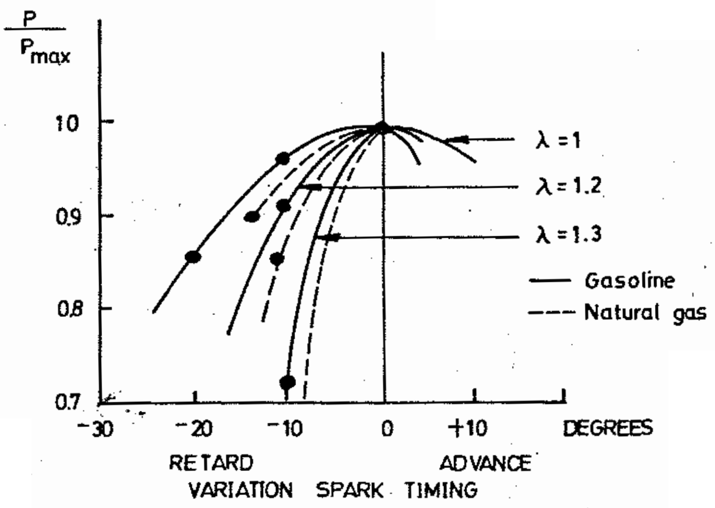


Fig.9.4.1.2: Normalized power and fuel consumption and timing sensitivity.(Ref: 2)

It will be necessary to advance the spark timing because of the reduced flame velocity (Table: T.9.4.1.1.)

Table: T.9.4.1.1. Laminar flame velocity in air [m/s]*

Methane	0,43
Propane	0,47
Butane	0,45
Hydrogen	3,5
Gasoline	0,5
Biogas* (55% CO2)	0,035
(45% CO2)	0,106
(35% CO2)	0,179
(25% CO2)	0,251

* Calculated with the Schuster equation:

$$V = V_0 \left[1 - \frac{N2\% + 1,67\ CO2\%}{100} \right]$$

V_o : Velocity of pure fuel gas.

9.4.2. RECOMMENDATIONS FOR IGNITION SYSTEMS.

Although the stock ignition system (whether conventional point ignition or electronic high energy ignition) is capable of igniting the fuel, it should be inspected closely to make sure that it is performing up to specifications. Pitted or worn contact points, a cracked distributor cap, dirty or worn plugs, a weak coil, or high resistance in the ignition cables can cause problems with any fuel.

Most experienced mechanics recommend installing "colder" spark plugs when converting to propane. For heavy-duty trucks, use two heat ranges colder. For pickup trucks, use one heat range colder. For passenger cars, use one heat range colder if the car will be subjected to a lot of highway driving or hard use. Otherwise stick with the stock plug recommendation. In dual fuel conversions, no ignition modifications are made if gasoline is to be used frequently.

9.5. POWER LOSS

At maximum engine power, the throttle is fully opened. The loss in maximum power for methane is about 10 to 15% at 2500 rpm. This power loss is caused by three factors:

1. About 8% of air is replaced by NG in the air intake. This is a higher percentage than with a liquid or more dense fuel.

2. The flame speed of NG is slower than of gasoline, hence the timing loss is larger.

3. The evaporation of a liquid fuel in the engine air intake cools the incoming air and raises its density, increasing the engine's air mass breathing capacity.

9.6. TROUBLESHOOTING

Performance problems can be caused by equipment installation errors, misadjustments, or normal wear and tear. One of the most common equipment installation errors is mounting the converter upside down so the vapor hose outlet is at the top. This causes oil to collect inside the converter and this can deteriorate the pressure diaphragms. The vapor hose outlet should be located at the bottom or underside when installed properly.

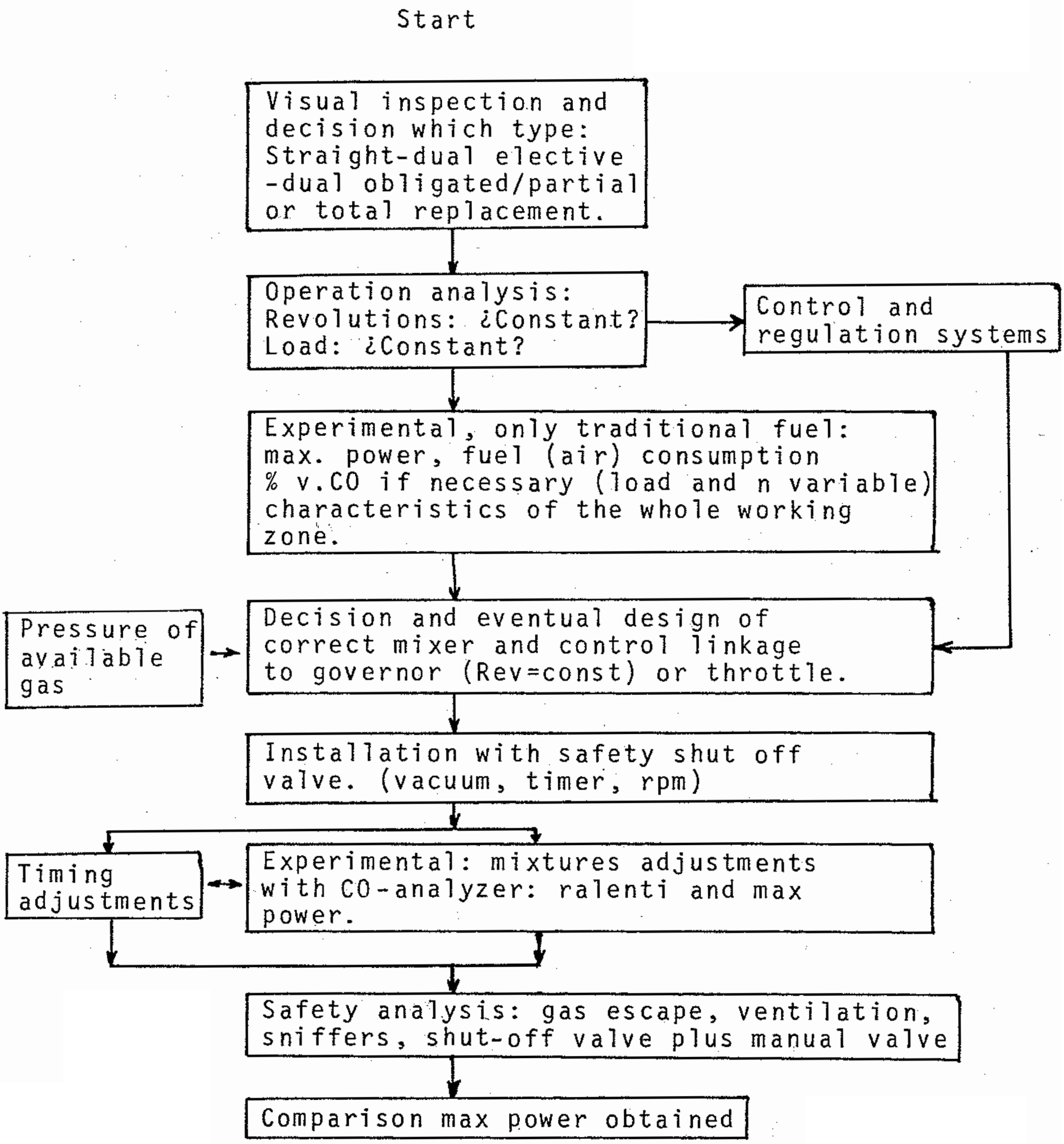
Another mistake is mounting the converter, the fuelock or the fuel line too close to the exhaust manifold or exhaust pipe.

On dual-fuel conversions, a poor seal between the gasoline carburetor adapter and fuel mixer will allow an air leak. This upsets the air/fuel mixture, and it can create idle problems and misfiring.

9.7. LOG-BOOK

It is always usefull to know the history of an engine. It will help to find out the cause of its failure or good behaviour. Also a correct maintenance according to the proposed program can be registred for this purpose an hour-meter is recommended. The following variables should be recorded for each operation: Date, hour, running time, fuels used, operator, load, oil level, water level, observations. Also the maintenance intervals for: oil change, plugs-revision/change, air filter cleaning/change, fuel filter change, contact points regulation, power output and adjustments must be included.

9.8. SUMMARY: ENGINE ADAPTATION



REFERENCES

[1] REINKE G./ Internal Combustion Engines, Lecture Drafts, 1984, Internal Information, UTFSM, Valparaîso

[2] Mc GEER/DURBIN "Methane Fuel For The Future" 1982, Plenum Press, New York (ISBN 0-306-41122-9)

[3] E.M. GOODGER "Alternative Fuels", The Macmillan Press Ltd, 1980, ISBN 0-333.

[4] The Use of Biogas in Diesel Engines, The Higher Performance an Low Contamination Advantages. /D.L. Wise, A. Sâez, G. Reinke, D. Alkalaj, J.O. Mesa/CRC Press, Title: "Biotechnology Applied to Environmental Problems" (To be published/1986.)

[5] SAEZ, A.C., REINKE, G.S., and Alkalaj, D.L., Effect of Fuel Inyection Point on Biogas Dual Fuel Natural Suction Diesel Performance, Proceedings of the III International Symposium on Anaerobic Digestion, Boston, August 14 to 19, 1983, 379.

[6] Instituto de Investigaciones Eléctricas de Cuernavaca, Morenos, México. Informe para Olade. Manual para el Promotor de la Tecnología.

[7] HELMUT MUCHE/ HARALD ZIMMERMANN "Reinigung von Biogas"/ Vieweg (Gate)/1984/ ISBN 3-528-0200-5

[8] Flussigganlagen in Krattfahrzeugen/Tuv (RWT UV), ESSEN, FRG.

[9] Landi den Hartog bulleton on "Diesel/LPG dual fuel system", Vienna, 1984.

[10] Catalogue "Rodagas", Brasil Auto Methane-Sao Paulo

[11] Otto-und Dieselmotoren./Dipl.-Ing. HEINZ GROTE/Vogel-Verlag, Wurzburg, FRG/1978/ISBN 3-8023-0052-1

[12] SAEZ, A.C., REINKE, G.S., ALKALAJ, D.L., and MESA, J.O., Biogas Application in Diesel Engines, Report to UTFSM and Municipality of Santiago, May. 1984.

[13] LARRY W. CARLEY, Propane Conversion of Cars, Trucks & RUS, Tab Books Inc. Blue Ridge Summit, PA. 17214, 1982, ISBN 0-8306-2103-2

[14] ASHARE, E., Analysis of Systems for Purification of Fuel Gas, in Fuel Gas Production from Biomass, Wise, D.L, Ed., CRC, Press, Fla., 1981.

[15] LANDFILL BIOGAS OPERATED Diesel Truck, A Chilean Experience./ A. SAEZ, G. REINKE, J.O. MESA/Conf. Gaseous Fuels For Transportation/8-10 AGO 86/ Van Couver, CANADA.