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**PRODUCTION AND FEEDING OF ETHANOL VAPOR FOR
AN INTERNAL COMBUSTION ENGINE OPERATING WITH
PRE-VAPORIZED FUEL**

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School of São Paulo University (EESC-USP), as part
of the requirements to earn the M.Sc title in Mechan-
ical Engineering.

Concentration area: Mechanical Engineering
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Symbol List

A	Area. [m ²]
c_p	Specific Heat at constant pressure. [J/g.K]
c_v	Specific Heat at constant volume. [J/g.K]
D	Diameter. [m]
e	Rugosity. [m]
f	Friction factor, dimensionless.
g	Gravity acceleration. 9,81m/s ² or 32,18 ft/s ²
ΔH	Head loss. [meters of water column]
h	Convection coefficient [W/m ² .K], or enthalpy. [J/g]
h_{lv}	Vaporization latent heat. [J/g]
K	Concentrated head loss. [s ² /m]
k	Thermal conductivity. [W/m.K]
L	Length. [m]
M	Mach number. Ratio between speed and sound speed.
MM	Molar mass. [g/mol]
Nu	Nusselt number. $\frac{hD}{k}$.
P	Power. [W]
IHP	Inferior Heat Power. [J/g]
SHP	Superior Heat Power. [J/g]
p	Pressure [Pa], [kPa] ou [kgf/cm ²]
q	Heat. [J]
q''	Thermal flux, $\frac{\partial^2 q}{\partial t \partial A}$. [W/m ²]
Q	Mass flow. [kg/s]
R	Perfect gas constant, 278J/(kg.K).
Re	Reynolds number. $\frac{\rho v D}{\mu}$.
r	Radius. [m]
T	Temperature. [K] or [°C]
U	Global heat transfer coefficient. [W/m ² K]
v	Speed. [m/s]
W	Work. [J]
Y	Dimensionless proportional concentration.
$ \phi $	Throttle valve opening. [%]

Greek characters

γ	Politropic air constant. Ratio between constant pressure and constant volume specific heats.
ϵ	Dimendionless rugosity. [$\frac{\epsilon}{D}$]
η	Yield, dimensionless.
Λ	Mass air-fuel ratio.
λ	Mass air-fuel ratio divided by stoichiometric ratio $\frac{\Lambda}{\Lambda_{esteq}}$ 8,33 for hydrated ethanol
μ	Viscosity. [N.s/m ²]
ρ	Density. [kg/m ³]
τ	Torque [Nm]
r	Compression ratio, dimensionless.
Ω	Thermal resistance, $\frac{\Delta T}{q''}$. [K/(m ² W)]

Conversion between SI and Imperial Units

1 in =	25.4 mm
1 ft =	304.8 mm
1 lb =	0.45359 Kg
1 lbf =	4.448 N
1 Kg =	2.20462 lb
1 yd = 36 in =	0.914 m
1 mi = 1760yd =	1609.344 m
F Fahrenheit degrees =	$(C + 32) \times 1.8$ where C are Celsius degrees
1 US gal	$231 \text{ in}^3 = 3.785 \text{ l}$
1 Nm =	0.7375 ft.lb
$1 \text{ kgf/cm}^2 = 10^5 \text{ Pa} =$	14.223 psi
1 psi =	7.0274 kPa
$1 \text{ ft}^2 =$	$9.2093 \cdot 10^{-2} \text{ m}^2$
$1 \text{ ft}^3 =$	$2.8316 \cdot 10^{-2} \text{ m}^3$
1 lb.ft =	1.356 Nm
1 HP =	745.7 Watt
1 BTU =	1054,8 J
R =	$2.4859 \times 10^{-5} \frac{\text{BTU}}{\text{lb Rankine}} = 0,287 \frac{\text{J}}{\text{Kg K}}$

Resumo

ALVES, F.J. (2007) *Produção e fornecimento de vapor de etanol para motor de combustão interna operando com combustível pré-vaporizado*. Dissertação (Mestrado). Escola de Engenharia de São Carlos, Universidade de São Paulo, São Carlos-SP.

O Motor a Álcool Pré-vaporizado tem potencial para ser uma alternativa mais eficiente e menos poluente aos motores a álcool convencionais. Nele, o combustível é vaporizado com calor rejeitado pelo próprio motor e admitido na fase gasosa, aproveitando-se das vantagens dos motores com combustíveis nessa fase sem alguns dos seus inconvenientes. O projeto foi aperfeiçoado buscando viabilidade técnica e econômica para sua instalação em veículos automotores. Água do sistema de arrefecimento cede calor para a ebulição do combustível. As novas tecnologias para injeção de combustíveis gasosos contribuem para esse objetivo, bem como o desenvolvimento de um sistema sustentável e auto-ajustável de geração de vapor de etanol que usa a água do sistema de arrefecimento. Conseguiu-se maior eficiência em quase todos os regimes de funcionamento estudados, bem como meios de reduzir as principais emissões automotivas indesejáveis.

Palavras-chave: Combustíveis gasosos, energia renovável, injeção de combustível.

Abstract

ALVES, F.J. (2007) *Ethanol vapor production and feeding for an internal combustion engine operating with pre-vaporized fuel*. Thesis (M. Sc.). São Carlos Engineering School, São Paulo University, São Carlos-SP, Brazil.

Pre-vaporized ethanol engine (PVEE) has potential to be more efficient and less pollutant than conventional ethanol-powered engines. In it, fuel is vaporized with heat rejected by engine itself and intook in gaseous form, taking advantage of this kind of fuel but without some of its inconveniences. The PVEE project was polished looking for economical and technical liability to future use in automotive vehicles. New gaseous fuel injection technologies contribute to this goal, together the development of a sustainable and self-adjustable ethanol vapor generating system who uses water from engine's cooling system. Better efficiency was achieved in almost all investigated regimes, as well as were found ways to reduce the main undesirable automotive emissions.

Keywords: Gaseous fuels, renewable energy, fuel injection.

Resumé

ALVES, F.J. (2007) *Production et fournissement de vapeur d'éthanol pour moteur à combustion interne alimenté par du carburant pré-vaporisé*. Dissertation (Master). Escola de Engenharia de São Carlos, Universidade de São Paulo, São Carlos-SP (École d'Ingénierie de São Carlos, Université de São Paulo, São Carlos-SP, Brésil).

Le moteur à alcool pré-vaporisé est potentiellement une alternative plus efficace et moins polluante que les moteurs à alcool conventionnels. Le carburant y est vaporisé en grâce à la chaleur émise par le moteur même puis admis sous forme gazeuse, profitant des avantages des moteurs à carburant sous cette phase éliminant, de surcroît, certains inconvénients. Le projet fut amélioré visant sa viabilité technique et économique pour être installé sur des véhicules automoteurs. L'eau du système de refroidissement cède de sa chaleur pour que le carburant entre en ébullition. Les nouvelles technologies d'injection de carburant gazeux permettent que ce but soit viable, ainsi que le développement d'un système viable et soi-réglable de production de vapeur d'éthanol qui utilise l'eau du système de refroidissement. Furent atteintes une efficacité supérieure pour l'ensemble des régimes de fonctionnement étudiés, ainsi qu'une réduction des principales émissions automobiles indésirables.

Mots-clé: Carburants gazeux, énergies renouvelables, injection de carburant.

Resumen

ALVES, F.J. (2007) *Producción y suministro de vapor de etanol para un motor de combustión interna que opera con combustible pré-evaporado*. Tesis de Maestría. Escuela de Ingeniería de Sao Carlos, Universidad de Sao Paulo, Sao Carlos-SP, Brasil.

El Motor a Alcohol Pré-evaporado (MAPE) tiene potencial para ser una alternativa más eficiente y menos contaminante a los motores a alcohol convencionales. En él, el combustible es evaporado utilizándose calor despreciado por el próprio motor y admitido en fase gaseosa, aprovechandose las ventajas de los motores que operan con combustibles en esta forma, pero sin algunos de sus inconvenientes. El proyecto ha sido refinado buscándose su viabilidad técnica y económica, con el objetivo de lograr su instalación en vehículos automotores. El agua del sistema de enfriamiento provee calor para la vaporización del combustible. Nuevas tecnologías para inyección de combustibles gaseosos contribuyen para lograr ese objetivo, además el desarrollo de un sistema sustentable y auto ajustable de generación de vapor de etanol que usa agua del sistema de enfriamiento. Se logró más eficiencia en casi todos los regímenes de funcionamiento estudiados, además como maneras de disminuir las principales emisiones automotivas indeseables.

Palabras-clave: Combustibles gaseosos, energía renovable, inyección de combustible.

Chapter 1

Introduction

After difficult years for Proálcool¹, which occurred between the end of 1980's and almost all the 1990's, consumption of this renewable fuel has increased on Brazil and is considered again an energetic alternative for Internal Combustion Engines (ICE) which operate in Otto cycle. Other countries also question the possibility to incentive using it in their fleets, pure or blended to gasoline [1]. According to Anfavea (Brazilian National Association of Automotive Vehicle Makers) [2], trade of new ethanol-powered vehicles had been multiplied by 40 between 1998 ("bottom of pit" for Proálcool, when we came to the sell of kits for engine conversion to gasoline feeding), and 2002. The main factors which contributed to elevation of its prestige are:

- Need to fit to greenhouse gas and other pollutants reduction in emissions, like Kyoto protocol. Use of ethanol, even partially, decreases responsibility of its participants in elevation of carbon dioxide concentration on the atmosphere, because being ethanol a renewable energy source the tailpipe emitted gas is re-absorbed during sugar cane growth. Worry on greenhouse effect has become more intense on the later years, with announcement of grave climate changes on our planet and melting of polar icecaps that would lead to sea level raise and flood of seashore cities.
- Political unstabilities on the main oil-production regions, which allied to the strong increase in its demand have elevated the price of this supply. By November 2007 oil barrel price had break the US\$ 100 barrier, forcing upwards fossil fuel prices throughout the world. Despite seasonal oscillations due to its production cycle (annual harvest), ethanol's price is kept in advantage in relation to gasoline's value in several regions of Brazil. An additional increase in fossil fuels price can turn this advantage more scandalous.
- Investments on logistics and infra-structure to enhance distribution and transport of carbureting ethanol, both for internal market and for export [3]. Ampliations of harbours and

¹The Brazilian pro-ethanol program, which stimulated development in production and use of this fuel in automobiles after the first oil crisis of 1970's

hydroways, dedicated modal terminals and even ethanol ducts are in the plans of state-owned and private companies for the next years.

- In February 2007 IPCC (Intergovernmental Panel on Climate Change) released its Fourth Assessment Report [4], publishing results of their studies which give trustability to the hypothesis of the climate changes are due to human interferences, which have increased concentration of some gases on the atmosphere, discarding the possibility to attribute it to natural causes. That document had wide world broadcast and has been too much discussed in the media worldwide; leading to many people, companies and government administrations to think about these issues and to look for alternatives. Ethanol fuel is one among them.
- Trade agreements made with countries interested in buying Brazilian ethanol stimulate compromise by part of producers in keep supplying of it in a stable and regular form. Supply crisis, as occurred in Brazil in 1989 and 1990 would dirt our image, as also would take difficult celebration of new trade contracts involving this commodity. Ethanol plant owners have a dilemma on Brazil, which lies on direct the correct proportion of cane harvested to ethanol and sugar production according to market demand and international prices for sugar, once ethanol is not an international commodity yet. Elevation in international sugar prices is a strong temptation on sugar/ethanol plant owners to produce less fuel.
- On Brazil, release of flex-fuel vehicles² in 2003 has allowed customer to decide which fuel use without engine alterations or other technical complications. Despite being not the more efficient choice compared to the dedicated ethanol engine, such a engine setup gives psychological comfort to the Brazilian driver, who is traumatized with recent lacks in ethanol supply and now has the possibility to use gasoline if ethanol prices are prohibitive in some months of the year or in some non-ethanol producing regions.
- New technologies which take easier cold starting, propelled by electronic fuel injection development. There are people thinking in cold start systems totally independent of more volatile fuels like gasoline or propanone, focusing on safety given by the absence of new fuel reservoirs and reduction in development costs brought by the need to lose many prototypes in crash tests need to homologate vehicle with these new reservoirs.

²Fuel injection systems of these vehicles detect ethanol-gasoline proportion by Lambda sensor signal, adapting fuel amount and ignition timing on the first crankcase revolutions if tank was fed. More than half of new Brazilian automobiles are flex fuel currently.

- Oil is not renewable, which means some day we will not have this source of energy and chemical supplies. Estimates preview 40 to 50 years of duration for world reserves, since current patterns of population number, income and consumption continue growing at the same ratios. [5]. So, the worry appears on obtain other source of energy for Otto ICE's which allow operate it in conditions near or even better than those seen on gasoline.
- Carvalho [6] appoints to a scenery in which ethanol production on Brazil leads to nearly 150 times the number of employments by the same amount of energy produced, compared to gasoline.

Despite the spread use of ethanol blended to gasoline, in pre-fixed (part of pre-2003 Brazilian automobiles and maybe other countries' ones in the future) or random (flex fuel) amount; an ICE dedicated to operate exclusively with ethanol allows to intake it in vaporized form. Being a pure substance, in opposition of gasoline which is made by dozens of hydrocarbons with different boiling points, its boiling temperature depends only on pressure.

The fact of be a pure substance, without light or heavy fractions, also gives to ethanol advantages of less evaporative losses, also taken as air pollution nowadays, and the absence of dirt deposits in the combustion chamber and in the carburettor/fuel injectors. Absence of aromatic compounds and other hydrocarbons commonly found in gasoline also are good in the way to lead to less emissions of health hazardous substances.

An Otto ICE supplied with pre-vaporized ethanol can obtain advantages to its similar fed by liquid ethanol; as lower pollutant emissions and better thermodynamic efficiency. There is also advantages in storing fuel in liquid state, boil it in proper time and just so intake it in vapor form. Such advantages will be discussed in the next chapter.

Other renewable fuel which appears as an alternative for Otto ICE's is 2,5 dimethylfuran (2,5-DMF). Román-Leshkov et al. [7] describe a chemical route to produce it from isomerization of any kind of biomass, what can turn feasible its production in industrial scale for the next years, using vegetable raw materials. There are advantages and disadvantages of this fuel related to ethanol. 2,5-DMF has combustion heat power and vaporization latent heat similar to those found in gasoline, and by being capable to be synthesised from any kind of biomass may come to present good productivity in relation to the agricultural area used for its production; but on the other hand it boils at higher temperatures than those observed for ethanol, for given pressures. 2,5-DMF is not water soluble, so it does not absorb moisture from atmosphere³. The shape of its molecule (a four-carbon-one-oxygen ring, with methyl group in two oxygen-sided carbons), leads to the suppose of its *octan number*, or resistance to self ignition, be high enough

³Pure ethanol can suck air moisture until reach 7% of mixture weight in water, turning itself into hydrated ethanol. Dehydrating it before blending to gasoline is an relatively expensive process

to allow its use in engines whose compression ratio and ignition timing is set up to gasoline, or maybe in more aggressive values. On the other hand, 2,5-DMF is toxic and would cause more environmental and human health damage if it leaks, accidentally inhaled or drunk⁴. Ethanol used as fuel has addition of denaturants to prevent its use as beverage. These denaturants must have higher volatility (higher vapor pressure) than ethanol to boil before it and do not be accumulated in ethanol vapor production circuits.

It must be clearly pointed that adaptations for gaseous fuels only would be possible in flex fuel vehicles if different fuels be stored in separated tanks. Gasoline in any part of ethanol vapor production system is strictly forbidden.

⁴Ethanol is 40% of cachaça, one of the most popular alcoholic beverages of Brazil, the remainder is compound almostly by water; so accidental contact with ethanol would cause the same health injuries of those caused by distilled alcoholic beverages

Chapter 2

Bibliographic Revision

2.1 Concern on air pollution

On the past, when internal combustion engines has appeared, researches was directed almost exclusively to reach maximum power. After oil shocks of 1970's, worry was moved to fuel economy. Nowadays the main goal is holding emissions of pollutants into acceptable levels which does not threat life quality and sustainability, specially in great metropolis with many automotive vehicles circulating.

On Brazil, Ibama (*Instituto Brasileiro do Meio Ambiente e dos Recursos Naturais Renováveis*, or Brazilian Institute for Environment and Renewable Natural Resources) had created, in 1986, Proconve (*Programa para Controle de Emissões Veiculares*, or Program for Vehicle Emissions Control), aiming to loose impact caused by automotive emissions in Brazilian cities. Significant results were reached since its implantation, in 1986, as can be seen in table 2.1.

Current and future for emissions are in table 2.2. One can note there are specific restrictions for GNG (Compressed Natural Gas) propelled vehicles, because the importance this fuel has achieved in the fleet of the most important Brazilian regions, like São Paulo and Rio de Janeiro States.

Another class which had its importance strongly increased was the motorcycles one. According to Abraciclo (*Associação Brasileira dos Fabricantes de Motocicletas, Ciclomotores, Motonetas, Bicicletas e Similares*; or the Brazilian Association of Manufacturers of Motorcycles, Bicycles and Similar Vehicles) [9], production of this kind of vehicle has jumped from 83,458 units in 1993 to 1,057,333 in 2004, an increase of 1167%, which means they have elevated too much their importance on Brazilian fleet share. Use in them of richer mixtures and carburetors instead electronic injection in majority of models sold; allied to absence of catalytic converters and the huge increase in number of these vehicles are factors that worry so much Proconve's authorities. To contour this problem Ibama created Promot (*Programa de Controle da Poluição do Ar por Motociclos, Ciclomotores e Similares*, or Program for Air Pollution Control from

Table 2.1: Mean emissions in new Brazilian light vehicles, since 1980.

Year/model	Fuel	CO (g/km)	THC (g/km)	NO _x (g/km)	Aldehydes (g/km)	Evaporative emissions (g/test)
Pre-1980	Gasoline	54	4.7	1.2	0.050	ND [†]
1980-83	E22 [‡]	33	3.0	1.4	0.050	ND
	Ethanol	18	1.6	1.0	0.160	ND
1984-85	E22	28.0	2.4	1.0	0.050	23.0
	Ethanol	16.9	1.6	1.6	0.180	10.0
1986-87	E22	22.0	2.0	1.9	0.040	23.0
	Ethanol	16.0	1.6	1.8	0.110	10.0
1988	E22	18.5	1.7	1.8	0.040	23.0
	Ethanol	13.3	1.7	1.4	0.110	10.0
1989	E22	15.2(46%) [◊]	1.6(33%)	1.6(0%)	0.040(20%)	23.0(0%)
	Ethanol	12.8(24%)	1.6(0%)	1.1(8%)	0.110(39%)	10.0(0%)
1990	E22	13.3(53%)	1.4(42%)	1.4(13%)	0.040(20%)	2.7(88%)
	Ethanol	10.8(36%)	1.3(19%)	1.2(0%)	0.110(39%)	1.8(82%)
1991	E22	11.5(59%)	1.3(46%)	1.3(19%)	0.040(20%)	2.7(88%)
	Ethanol	8.4(50%)	1.1(31%)	1.0(17%)	0.110(39%)	1.8(82%)
1992	E22	6.2(78%)	0.6(75%)	0.6(63%)	0.013(74%)	2.0(91%)
	Ethanol	3.6(79%)	0.6(63%)	0.5(58%)	0.035(81%)	0.9(91%)
1993	E22	6.3(77%)	0.6(75%)	0.8(50%)	0.022(56%)	1.7(03%)
	Ethanol	4.2(75%)	0.7(56%)	0.6(50%)	0.040(78%)	1.1(89%)
1994	E22	6.0(79%)	0.6(75%)	0.7(56%)	0.036(28%)	1.6(93%)
	Ethanol	4.6(73%)	0.7(56%)	0.7(42%)	0.042(77%)	0.9(91%)
1995	E22	4.7(83%)	0.6(75%)	0.6(62%)	0.025(50%)	1.6(93%)
	Ethanol	4.6(73%)	0.7(56%)	0.7(42%)	0.042(77%)	0.9(91%)
1996	E22	3.8(86%)	0.4(83%)	0.5(69%)	0.019(62%)	1.2(95%)
	Ethanol	3.9(77%)	0.6(63%)	0.7(42%)	0.040(78%)	0.8(92%)
1997	E22	1.2(96%)	0.2(92%)	0.3(81%)	0.007(86%)	1.0(96%)
	Ethanol	0.9(95%)	0.3(84%)	0.3(75%)	0.012(93%)	1.1(89%)
1998	E22	0.8(97%)	0.1(96%)	0.2(88%)	0.004(92%)	0.8(97%)
	Ethanol	0.7(96%)	0.2(88%)	0.2(83%)	0.014(92%)	1.3(87%)
1999	E22	0.7(98%)	0.1(96%)	0.2(88%)	0.004(92%)	0.8(97%)
	Ethanol	0.6(96%)	0.2(88%)	0.2(83%)	0.013(93%)	1.6(84%)
2000	E22	0.73(97%)	0.13(95%)	0.21(87%)	0.004(92%)	0.73(97%)
	Ethanol	0.63(96%)	0.18(89%)	0.21(83%)	0.014(92%)	1.35(87%)
2001	E22	0.48(98%)	0.11(95%)	0.14(91%)	0.004(92%)	0.68(97%)
	Ethanol	0.66(96%)	0.15(56%)	0.08(93%)	0.017(91%)	1.31(87%)
2002	E22	0.43(98%)	0.11(95%)	0.12(95%)	0.004(92%)	0.61(97%)
	Ethanol	0.74(96%)	0.16(90%)	0.08(93%)	0.017(77%)	ND
2003	E22	0.4(98%)	0.11(95%)	0.12(93%)	0.004(92%)	0.75(97%)
	Ethanol	0.77(95%)	0.16(90%)	0.09(93%)	0.019(89%)	ND
	Flex-E22	0.5(98%)	0.05(98%)	0.04(98%)	0.004(92%)	ND
	Flex-Ethanol	0.51(88%)	0.15(90%)	0.14(93%)	0.020(89%)	ND

Continued

Conclusion

2004	E22	0.35(99%)	0.11(95%)	0.09(94%)	0.004(92%)	0.69(97%)
	Ethanol	0.82(95%)	0.17(89%)	0.08(94%)	0.016(91%)	ND
	Flex-E22	0.39(99%)	0.08(97%)	0.05(97%)	0.003(94%)	ND
	Flex-Ethanol	0.46(97%)	0.14(91%)	0.14(91%)	0.014(92%)	ND

† ND=Não Disponível=Not Available

‡ E22=22% Ethanol - 78% gasoline blend, sold in Brazilian fuel stations instead pure gasoline.

◇ Emission reduction compared to pre-Proconvian era.

Source: Ibama [8].

Table 2.2: Emission limit of pollutants for light non-commercial vehicles

Pollutants: Maximum limit	until dec 31 2006	since jan 01 2005†	from jan 01 2009
Carbon monoxide (CO, in g/Km)	2.00	2.00	2.00
Hydrocarbons (HC, in g/Km)	0.30	0.30‡	0.30‡
Hydrocarbons not methane (NMHC, em g/Km)	NE◇	0.16	0.05
Nitrogen oxides (NO _x , in g/Km)	0.60	0.25§ or 0.60¶	0.12§ or 0.25¶
Particulate material (MP, in g/Km)	0.05	0.05	0.05
Aldehydes (CHO, in g/Km)	0.03	0.03	0.02
Evaporative emissions (g/test)	2.00	2.00	2.00
Crankcase emissions	null	null	null

† Requirement for 40% of vehicles sold in 2005, 70% in 2006 e 100% in 2007.

‡ Just for CNG vehicles.

§ Gasoline or Ethanol powered vehicles.

¶ Diesel powered vehicles.

◇ Not required.

Source: Ibama [8].

Motorcycles and Similar Vehicles) in 2002, aiming establish limits for their emissions. Current limits and those stipulated for new motorcycles from 2009 to ahead are described on table 2.3.

Aiming to fit to Proconve's requirements automobiles are equipped with catalytic converters (simply known as catalysers), which are devices which agilize some chemical recations which would occur naturally but taking more time, and turn pollutants into less dangerous substances. Calalyzers, therefore, have some inconveniences:

- They work well after some warming up time. During this period the majority of travels are done by drivers and air-fuel mixture is generally rich for aim engine working with its part cold yet. This implies on lower conversion efficiency exactly when its capacities would be more desirable. Alternatives to contour it are oxygen-retainer molecular sieves which hold excedent oxygen when mixture is lean; or even thermal insulated conversors which keep themselves hot enough until the next use of vehicle, generally on the next day. Brandt *et al.* [12] developed models for study three-way catalysts transient behaviour during its warm up and taking into account oxygen storing phenomenon made possible

Table 2.3: Promot's established limits for motorcycle emissions

	Motorcycles	Similar	Idle engine
after jan 01 2003	CO: 6.0g/km HC+NO _x : 3.0g/km	CO: 13.0g/km HC: 3.00g/km NO _x : 0.3g/km	6.0% CO vol. up to 250cc 4.5% CO vol. above 250cc
Manufactured or resset-up after 01/01/2005	CO: 5.5g/km HC: 1.2g/km 1.0 g/km \geq 150cc NO _x : 0.3g/km	CO: 7.0g/km HC: 1.5g/km NO _x : 0.4g/km	idem
From 01/01/2009	Up to 150cc CO: 2.0g/km HC: 0.8g/km NO _x : 0.15g/m	Above 150cc CO: 2.0g/km HC: 0.3g/km NO _x : 0.15g/km	idem

Source: Ibama's resolutions number 297, (feb 26 2002) and number 342, (25 sep 2003) [10, 11].

by cerium oxide, which corrects slight variatons in air-fuel-ratio during engine operation.

- Catalytic converter can be deactivated during its lifecycle by thermal and/or chemical reasons. Chemical contamination is caused mainly by sulphur and phosporus found on gasoline. Thermal deactivation can happen due to unregular working of engine, which can lead to feeding it with air streams higher than it supports. Such a device is not substituted when it breaks or melts because its high price (nearly the equivalent to US\$ 700 actually, on Brazil).
- The small channels in which exhaust gases pass through require great head losses for their passage. According to Martins [13] this kind of head loss in a 2.0 liter engine can reach 24%.

There is expectation to satisfact Proconve's requirements with a pre-vaporized ethanol engine (PVEE), operating without catalyst converter. On PVEE there are other way to control Brazilian law-regulated emissions: use of leaner mixtures. Three-way catalysts currently used need an air-fuel ratio very close to stoichiometric to convert successfully both CO, HC and NO's.

On United States, EPA (Environmental Protection Agency) recommends use of oxygenated fuels, it means, with one or more oxygen atoms in its molecules, blended to gasoline to mitigate CO emissions. Main additive are MTBE (Methil-terc-butyhl ether), ethanol and methanol. A higher content of oxygen is required on the winter (minimum 2,7% in weight), when the effect of CO's emissions are more grave. Such an advise is based on United-statesian law on air pollution, the Clean Air Act [14, 15].

Poulopoulos *et al.* [16] and Pau [17] studied emissions influence of blending ethanol on gaso-

line. It was observed slight decrease on CO and HC emissions, remembering higher latent heat of ethanol delays its evaporation compared to gasoline; diminishing its effect of oxygen presence on emissions and leading also to non-burnt ethanol and aldehyde emissions.

The main cause of CO emissions is the incomplete fuel burn, and it increases quickly with enrichment of air-fuel mixture, as related by Hochgreb [18]. Oxygenated fuels like ethanol and methanol take atom meeting easier, taking oxidation easier.

2.2 Development of hydrated ethanol-powered engine

The Internal Combustion Engines Laboratory of EESC-USP contributes to the development of ethanol powered engine, pre-vaporized or not, since Proálcool's early years, in late 1970's. One important Brazilian military Engineering college, ITA (*Instituto Tecnológico de Aeronáutica*, or Aeronautics Technological Institute), contributed to this task too in its first years.

Pioneer studies of Celere [19] and Venanzi [20] already have shown that use of vaporized ethanol has potential to turn this kind of engine more efficient and less pollutant. Bergman [21] adapted an originally designed Diesel engine to intake ethanol vapor together Diesel from original fuel injection. Notable torque increase were observed at low speeds, like emissions decrease compared to original similar Diesel engines. Celere and Venanzi used electrical resistances to boil fuel, while on Bergman and D'Ávila's [22, 23] researches the worry in design an auto-sufficient energy supply for vaporize fuel was present; through making use of heat rejected by the engine and present on exhaust gases or water from the cooling system.

Main feasible advantages of PVEE are referred to specific consumption and pollutant emissions. Better tolerance of gaseous fuels to operation on leaner mixtures in comparison to liquid fuels, allied to the oxygen atom present on its molecule, can lead to low CO and HC emissions, despite the lower energy content by unit of mass compared to heterogeneous and non-oxygenated fuels. On several countries there is interest in add to gasoline some additive like ethanol, methanol or MTBE, aiming their advantages (which are more difficult to be observed in ancient vehicles). Ethanol has the advantage to be non-toxic as the other ones are, requiring lesser care procedures in its handling.

Evaporative emissions are fuel losses caused before it can be burn, still in the fuel tank, and can reach 50 grams per day for an automobile parked in a hot place. Temperature oscillation through the day make not hermetically closed tanks to breathe, increasing mass transport to the environment by renewing of air inside the tank. Light fractions of fossil fuels like gasoline contribute to this phenomenon too [24]. Current electronic fuel injection systems have sealed tanks and vapor storing devices made of activated coal. In them, fuel vapors are released to intake

manifold and consequently combustion.

Nitrogen Oxides (NO_x) also can be controlled according to air-fuel ratio, which has effect on maximum flame temperature, closely linked to these kind of emissions [18]. There are other mechanisms not temperature-related which form NO_x too, but with lower share on the total emissions of this substance [25].

Gaseous fuels, like LPG (Liquified Petroleum Gas) and CNG also offer the chance to control flame temperature through the quantity of fuel intook (and consequently the quantity of heat released), but have the inconveniences of requiring heavy, volumous and requiring of special safety devices tanks for its storage.

Influence of air-fuel ratio on gasoline-powered emissions was also investigated by Harrington e Shishu [26] apud [18]. One can note leaner mixtures take easier fuel and oxidant molecule meeting, lowering CO and HC down to a given point, beyond which there is no more stable flame spreading, leading to misfires and consequent emissions of partially burnt hydrocarbons. Slightly lean mixtures, on $\lambda = 1,1$ range, are used to reach maximum flame temperatures, which is good for thermal efficiency but bad for human health because high NO emissions. Leaner mixtures, possible with gaseous fuels, can lead to further lower temperatures and give a bit of each advantage.

Ethanol has lower affinity to lubricating oil than hydrocarbon-based fossil fuels, like gasoline and Diesel. This leads to less lubricant contamination, extending its lifecycle. Injection vaporized ethanol fuel-oil contact is further low, because there is no fuel bubbles contacting the oil film inside the cylinder walls.

Crevice on combustion chamber and on piston ring packs are responsible for a great part of solid deposits of unburnt fuel, accordingly to Hochgreb [18]. Such a phenomenon occurs due to flame extinction on the narrow channels in which flame propagation is difficult, because their higher local area-volume relation suitable for higher heat transfer rates. Gasoline-powered engines tend to present more solid deposits due to the heavier fractions of its composition, while on gaseous engines density (and consequently quantity) of it is lower, reducing fuel losses and keeping the combustion chamber clean. Ethanol engines have cleaner combustion chamber too due to absence of heavy compounds which are difficult to vaporize. According to Ferguson [27], unburnt fuel deposits formed initially on gasoline ICEs increase contact area between fuel and combustion chamber surface due to its porosity, turning more intense flame extinction and unburnt fuel emissions.

In carburetted engines, or even on those indirectly fuel-injected use of gaseous fuel the use of gaseous fuels can bring advantages in transient regimes (acceleration). Transport of fuel mixed to air is faster than those made by aerodynamic drag through intake manifold and valve walls.

Second mode of fuel transport is too much slower, which delays arrival of it to the cylinders. Hohscho *et al.* [28] studied the behavior of an 1.3 liter displacement engine under transient accelerating conditions. Tight increase on throttle valve opening have lead to torque and speed floatings, having a slight drop after beginning of sudden acceleration. The responsible for that is the delay of fuel supply caused by its lower speed related to the air flowing through intake manifolds. Hohscho *et al.* also have realized tests on the same engine using electrical heaters on the lower part of intake manifold, where higher amounts of liquid fuel were seen. Lower torque floating and absence of speed drop were achieved.

One solution for this problem in conventional carburetted engines is to use a slightly rich air-fuel ratio for compensate leaning effect of transient accelerations, having the inconveniences of rise consumption and pollution natural of operating ICEs in these conditions, as is verified by Ferguson [27], Sher [24], Cooper [29] and by da Silva [30] in the case of the ethanol-powered engine.

The motorcycle Honda CB 500, assembled in Brazil until mid-2004, has a partial fuel vaporization system through pipes which use water of its cooling system passing on its carburettor bowls. Such a device gives to that vehicle a better answer to acceleration (either starting from stand or from intermediate speeds) requirements in comparaton to its direct concurrents in the same range of power and displacement, according to the investigated by Brazilian motorcycle specialized magazine *Duas Rodas* (Two Wheels) [31]. One can note even on motorcycle engines, which are used to have short intake manifolds, any advantage which aims to turn fuel flow faster is welcome. Engines equipped with indirect eletronic fuel injection, even multipoint, can achieve such advantages using gaseous fuels.

A positive consequence of a faster fuel burn is a slight increase of thermal efficiency. In an ideal Otto cycle, the heat addition process (air-fuel mixture burning) occurs at constant volume. In the real cycle, combustion time is finite, with piston moviment during it. Observing figure 2.1 one can note a slower combustion with former beginning increases pressure inside cylinder during compression phase, which increases compression work. Realizing the same amount of work during expansion phase decreases net work, according to the simulations made by Ferguson [27]. On that figure, compression stage of an Otto cycle include four different possibilities: the superior line, continuous, normal condition mixture compression; dashed line represents a faster combustion with spark ignition delay; dash-and-point line defines an ideal combustion whose times tend to zero and so at constant volume. Dotted inferior line represents a constant volume compression with in-cylinder refrigeration due to the higher latent heat (ethanol has higher latent heat than gasoline) of vaporization of the fuel. On the three last alternatives there are reduction of compression work, leading to better thermal efficiencies. The simulaton done

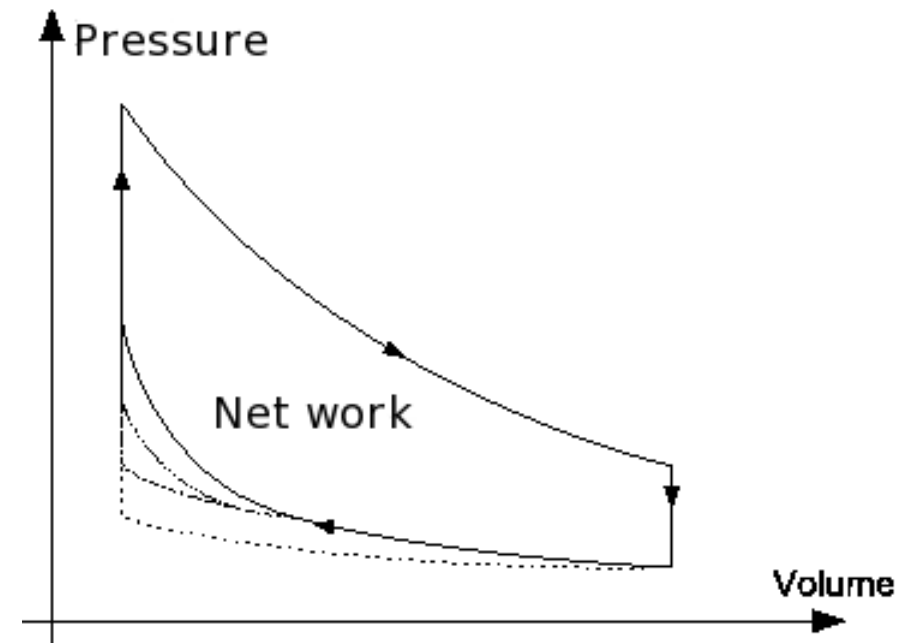


Figure 2.1: Different Otto cycles with different combustion times and cylinder refrigeration during compression phase.

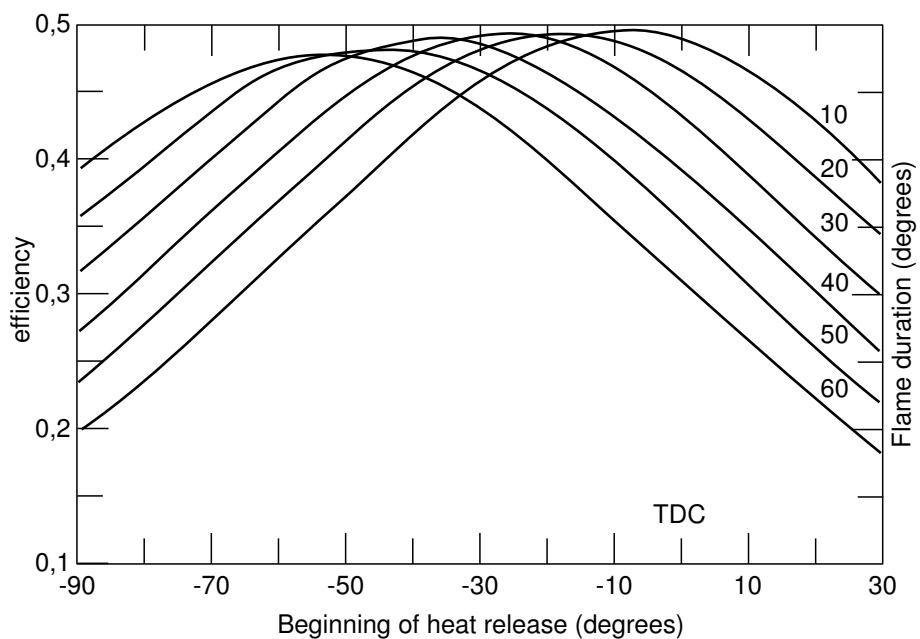


Figure 2.2: Numeric simulation of thermodynamic efficiency as a function of spark ignition timing and of combustion time.

Source: Ferguson [27].

by Ferguson [27] relating thermal efficiency, combustion time and ignition delay, whose result is on the figure 2.2, shows the two main condition for maximum efficiency being achieved: end of combustion exactly when the piston reaches the Top Dead Center and combustion time as short as possible. Combustion chambers with more than one spark gives agility to this process, as its adequate localization for allow flame fronts run trough combustion chamber in the lower time interval as possible. The hemispherical combustion chamber with one spark on its center is one of the most favorable setups. Souza's study [32] about combustion chamber formats throughout the whole history of Otto ICEs indicates this alternative and completes it saying the use of pre-vaporized ethanol could dispensate the use of inside or outside chamber devices which aim to mix and atomize liquid fuel, allowing to concentrate efforts on production of combustion chambers which avoid knocking, heat losses and undesirable flame extinction. On conventional engines, performance is a bit sacrificed in favor of higher durability, mainly thorough some ignition delay which decreases maximum pressure during the cycle.

On the other hand, an advantage of liquid fuels, specially alcohols like ethanol and methanol, is the ability to retire heat from the environment during its vaporization, due to their higher vaporization latent heats. Heat needed to their evaporation is taken from air inside intake manifold and even inside cylinder, increasing its density (increasing volumetric efficiency) and decreasing the mechanical work required during compression phase (figure 2.1). Such a phenomenon and its influence on engine efficiency were studied by Feitosa [33] on his Thesis about cold starting on ethanol engines. Sato *et al.* [34] studied the influence of ethanol and methanol injection in ATREX (Air Turbo Ram-jet Engine Expander-cycle) jet engines, aiming to cool some regions. These fuels are not suitable for jet propulsion engines due to their lower heat combustion capacities in relation to those derived from fossil fuels like kerosene or even hydrogen. Their advantage consists on injecting some alcohol in low quantites in relation to the main fuel (nearly 3%), to cool the intaken air, increasing the efficiency of the Brayton thermodynamical cycle in which it operates. Such a method is cheaper and more feasible than cool air through heat exchangers, which require maintenance and form ice croasts on their surfaces.

Souza [32], however, claims attention to the undesired heat extraction of combustion chamber walls, which decreases their temperatures, increases flame extinction and its soon undesirable cooling. On Brazil ethanol use as a fuel is economically advantageous in relation to gasoline. The last has higher inferior heat power (IHP) than ethanol by mass unit: 40019J/g against 26838J/g, according to [37]. IHP is used for energetic power comparison because the exhaust gases have water vapor because they let the engine at temperatures higher than those of dew point for the pressure they are at, being impossible to them to supply the latent heat of water condensation. Gasoline sold on Brazil and ethanol densities are, respectively, 746kg/m³ e 735kg/m³, accord-

Table 2.4: Latent heat and heat power for come fuels

	Methanol (CH ₄ O)	Ethanol (C ₂ H ₆ O)	Heptane (C ₇ H ₁₆)	2,2,4-trimethylpentane (C ₈ H ₁₈)	Methane (CH ₄)
h_{lv} (J/g)	1180	1020	317.8	297.5	-
SHP (J/g)	21100	27710	44444	44350	56375

Source: Gutheil [35] apud Feitosa [33] and Wikipedia [36].

ing to Petrobras (*Petróleo Brasileiro S.A.* or Brazilian Petroleum Inc.) [38]. It is important to include density on economic viability calculations beacuse fuels are sold by volume units, and efficiency/energy ones are done using mass units. On the present case, one liter of ethanol has more mass than one liter of gasoline, having influence on the amount of energy available.

For defining the price limit above which ethanol is not advantageous, it is still needed the theoretical thermal efficiency of an Otto ICE according to its compression ratio and air fuel mixture used.

$$\eta = 1 - \frac{1}{r^{(\gamma-1)}} \quad (2.1)$$

Typical values for compression ratio (r) for passenger automobiles can be taken as 9:1 for gasoline and 12:1 for hydrated ethanol. For stoichiometric mixtures, the constant volume/constant pressure specific ratio (γ) values 1,34 for air-ethanol and 1,355 for air-iso-octane mistures [37]. For each configuration, theoretical Otto cycle efficiency (η) is:

$$\eta_{textrmethanol} = 1 - \frac{1}{12^{(0.34)}} = 0.5734 \quad (2.2)$$

$$\eta_{gasoline} = 1 - \frac{1}{9^{(0.355)}} = 0.5416 \quad (2.3)$$

With these data (IHP, density and theoretical Otto efficiency) the limit for price ratio in volumetric measures is:

$$\frac{26838}{40019} \times \frac{0.5734}{0.5416} \times \frac{735}{746} = 0.6995 \quad (2.4)$$

It is, while ethanol price of one liter is lower than 69,95% the price of one liter of gasoline, it is cheaper to run with it, since similar vehicles are adapted to each fuel. The 70% mark is commonly accepted by Brazilian engineers.

Higher compression ratios are possible with ethanol because it is a more resistant to knock fuel, in comparation to gasoline. Such a phenomenon is caused by compression of unburnt gases inside combustion chamber, leading them to temperatures high enough to cause spontaneous ignition on them before the flame front arrival, with undesirable consequences for Otto engine working. Together knocking higher pressure peaks which damage piston ans cylinder head walls, with their shock waves audible on its characteristic metallic sound. Shorter and more ramified molecules like ethanol's one are a feature of fuels resistant to spontaneous ignition

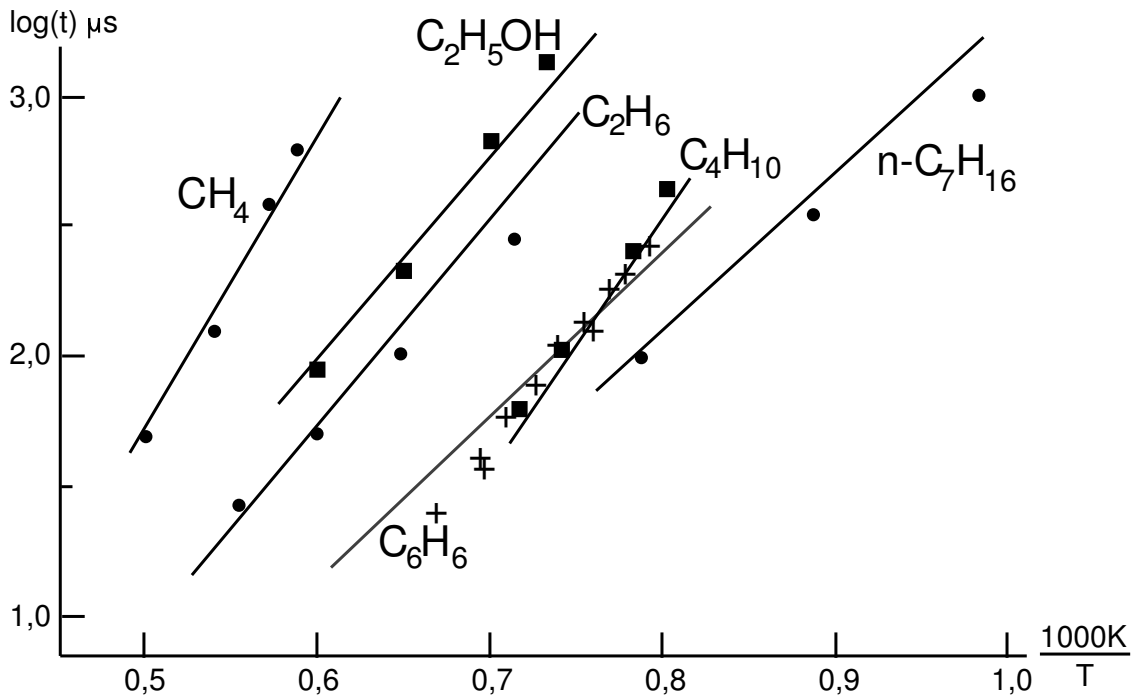


Figure 2.3: Ignition delay for some fuels.
Source: Warnatz *et al.* [25].

[25].

Shock waves created on detonation reach supersonic speeds (above 340m/s), while flame speed on Otto ICEs are of the order of dozens of meter per second in normal conditions. According to Souza [32], even using gaseous fuels flame speed does not achieve critical values (near sound speed), having no risk of detonation regimes. Higher flame speed of air-ethanol mixture contributes to its arrival at the most far places in relation to the spark, of a combustion chamber, before self-ignition conditions happen. Warnatz *et al.* [25] describe mechanisms, dependent or not of self-ignition at high (range of 1200K) and low (800-900K range) temperatures. Burning fuels at there is a phenomenon called *ignition delay* which during some time interval keeps some chemical reactions which consume few chemical radicals and release little heat. After a determined instant the concentration of these radicals is high enough to start a faster reaction, leading to an explosion/detonation. Spontaneous ignition delay time also depends on mixture temperature, as is shown at figure 2.3. For a given temperature, it can be noted ethanol resists to spontaneous ignition for more time than n-heptane, and that the later resists to the same time for self-ignition at lower temperatures than ethanol does in the same conditions.

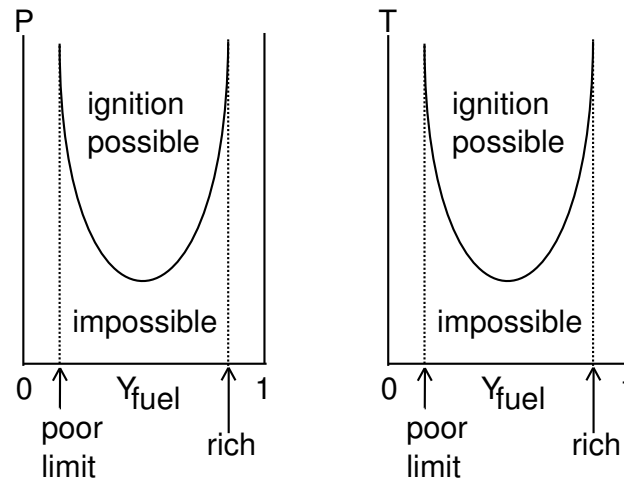


Figure 2.4: Influence of fresh mixture pressure and temperature on flammability limits, as a function of adimensional concentration.

Source: Kanury [40].

2.3 Physical advantages on burning gaseous fuels

Tolerance of gaseous fuels for operating in a wider air-fuel range maybe can allow an Otto engine to work with its throttle valve totally open in some regimes, where working conditions would be controlled only through ratio of fuel supply, like on Diesel ICEs, but keeping the spark ignition. Without the need to control the amount of intaken air through throttle better thermal and volumetric efficiencies could be achieved, once there would be one less obstacle to air passage. This goal was pursuit in earlier works involving PVEE [19, 21, 22, 23]. According to Strahle [39], previously homogenized flammable mixtures are capable to keep a stable flame for λ between 0,3 and 2,0 in a coarse estimative, which can be different for each fuel. Flammability limits depend also on pressure and tempeature in which mixture is at ignition moment, being aided by increase in any of these quantities, as appointed by Kanury [40]. However, one should attempt to the fact of burning flammable mixtures in the range of $\lambda = 1, 1$ leads to higher flame temperatures, reflecting on high nitrogen oxides emissions; while burning them in so poor limits flame stability is affected, causing high emissions of nonburnt fuels or even lower flame speeds.

More homogeneous mixtures burn in a more fast and complete way. Intaking liquid fuel not totally vaporized into the chamber, it needs more time to consume it completely because it is needed penetrate onto the drops, supply heat to vaporize it and just at this stage burn the fuel contained on them. According to Warnatz *et al.* [25], these phenomena cause ignition delays, which retard the start of burning process in each drop. Relative air speed impressed to these drops by intaken air aims to their evaporation, because helps mass and heat convection; reduc-

ing its diameter and the inconvenience of bigger drops quickly.

Experiences realized under microgravity environments at the International Space Station (ISS) by Kazakov *et al.* [41] allied to computational simulations also show smaller diameter drops burn more quickly than the bigger ones, because there is the need for heating the fuel drops and diffuse their vapor into the air with a more adequate area-volume in the case of smaller drops. On this kind of experiment it was still noted the influence of air humidity on these mass and heat transfer, thanks to ethanol solubility on water. According to Feitosa [33] and Kazakov *et al.* [41], sensible and latent heat transfer, after water vapor arrival at dew point, supplies heat which helps in ethanol evaporation in this phase. But in some conditions where flammable mixture starts before total vaporization of fuel drops, water concentration on drop borders can be high enough to turn difficult to burn remaining ethanol on it.

2.4 External surface boiling (pool boiling)

Phase change heat transfer obey rules different to those observed in heat transfer without phase change. Temperature of the surface supplying heat to the boiling liquid is slightly above to saturation temperature for that liquid at a given pressure, giving an important parameter called wall superheat. Other properties like liquid surface tension, latent heat, viscosity, solid-liquid contact angle and surface finish treatment determine the conditions in which phase-changing heat transfer occurs [42, 43, 44].

Figure 2.5 shows boiling regimes in an external smooth surface (Ni-Cr wire), accordingly to Nukiyama's experiments [45] apud Incropera [42]. A=boiling start; B=absence of alone bubbles; P=maximum heat transfer coefficient ($h = \frac{\partial q''}{\partial \Delta T}$); C=maximum heat flux; D=minimum flux, Leidenfrost' point.

In water ebullition, as superheat gets higher and higher, heat flux begins as purely convective. After achieving some wall temperature separated bubbles appear. Its separation of the wall shakes liquid in its surroundings, improving heat transfer coefficient. Above the maximum heat flux allowed (P point), heat transfer regime passes those of stable film which involves the wall surface permanently, difficulting thermal exchange. If superheat is controlled instead heat flux, there is a heat flow fall between points P and D, marking transition to the stable film regime. Increasing further heat flux or superheat at stable film regime some time fusion temperature of material will be achieved, breaking the wire in the case of Nukiyama's experiment [45] apud Incropera [42].

There are surface treatments suitable to enhance phase change heat transfer conditions, like

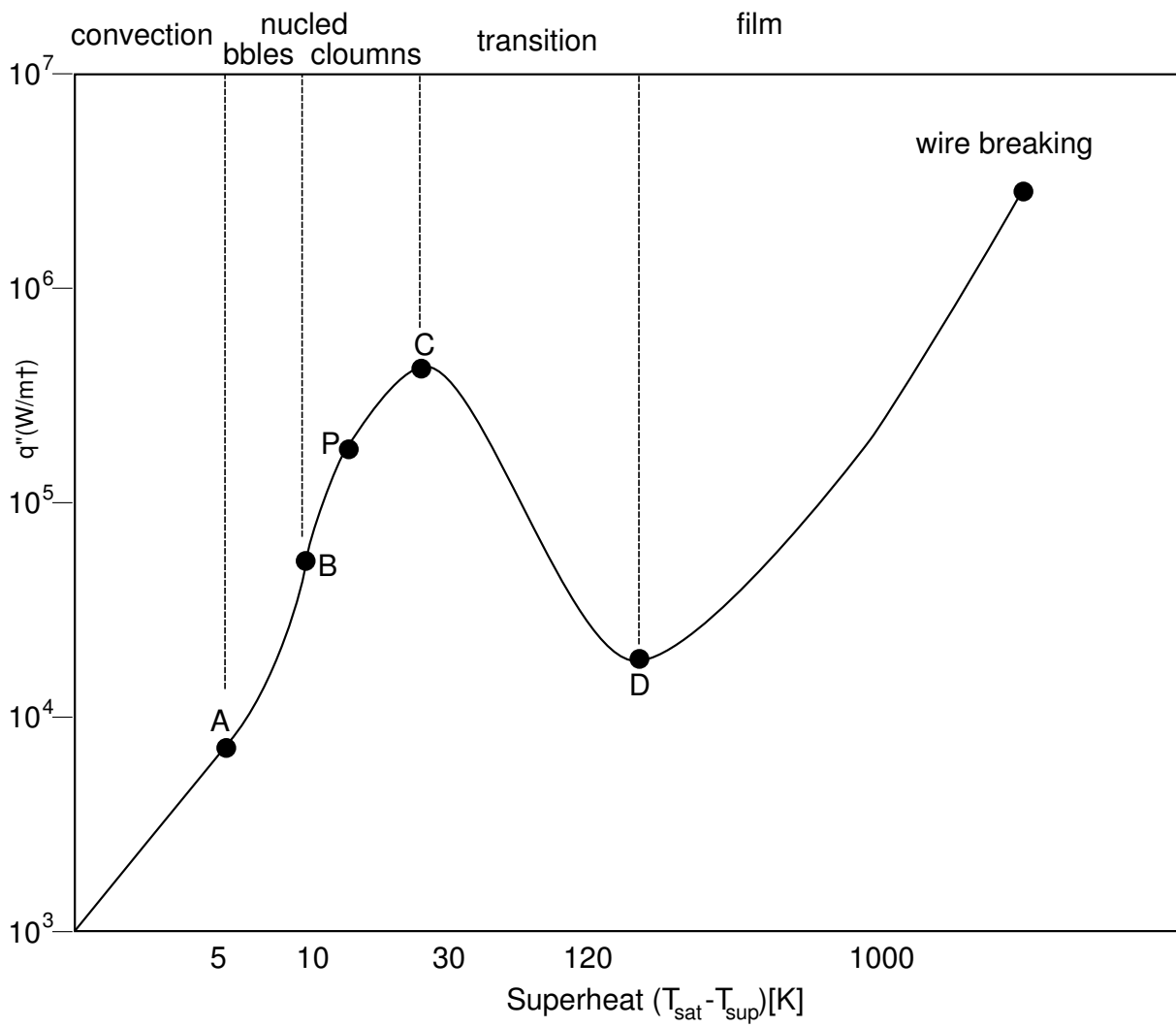


Figure 2.5: Water boiling at 1 atm in a Ni-Cr external surface.
Source: Nukiyama [45] apud Incropera [42].

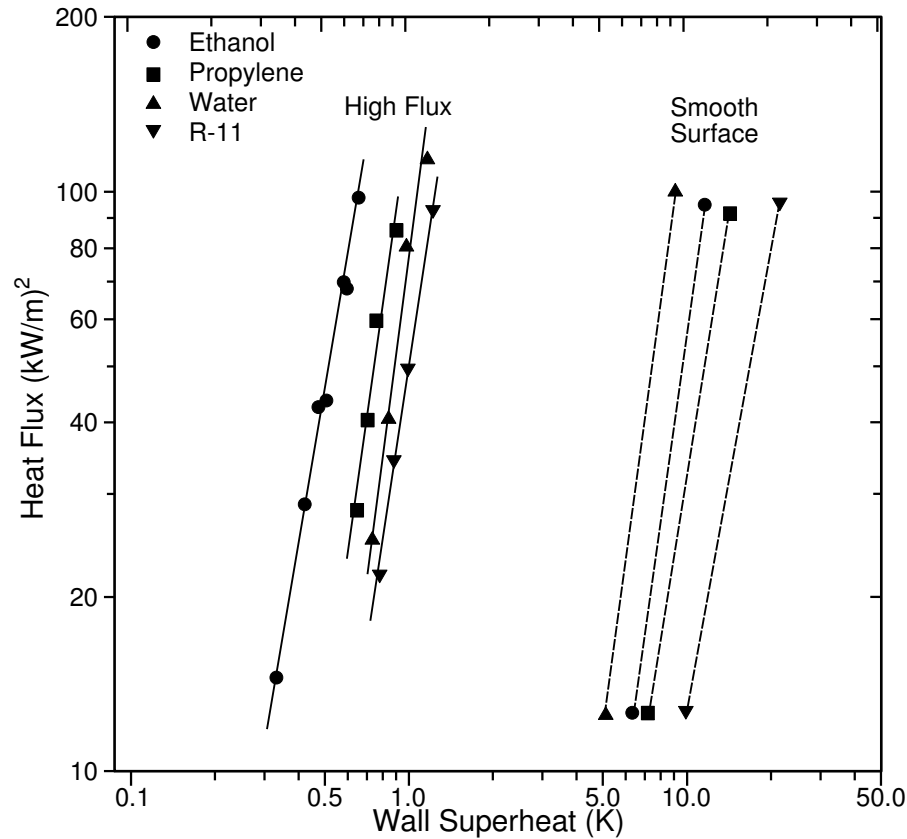


Figure 2.6: Transmitted heat on boiling several fluids, as a function of wall superheat.
Source: Thome [44].

heat transfer coefficient and minimum wall superheat required. Sandblast, oxidation, microfins, sinterizations and micromachining are used. Research on special surfaces however privileges some fluids used in refrigeration systems, some hydrocarbons, water, ammonia, liquid oxygen and nitrogen. There are few studies on ethanol boiling, probably because there are (still) few processes where vapor of ethanol is needed. Thome's studies [44] illustrate the influence of latent heat flux caused by parameters like wall material, surface treatment, wall superheat and fluid used. Development of several available surface treatments is also discussed, as well as superheat need for start boiling in each case.

Basically, ebullition surfaces can be changed through macroscopic ways (fins) or microscopic (abrasion, sinterization, oxidation). On microscopically extended surfaces there are enhancement on bubble formation due to heat transfer to small quantities of liquid through relatively higher areas. After initiated, some bubble has more ease to develop itself until be released from the wall surface. For ethanol ebullition in some surfaces, one of existing relations is on figure 2.6. Thome also determined minimum values for wall superheat for transition from merely convective heat transfer to starting ebullition in the specific case of ethanol. In smooth metallic

surfaces superheat needed can vary in the range of 6-23K. “High Flux” surfaces (trademark for some boiling surface treatment) assures boiling with a just 6 Kelvin superheat.

Chapter 3

Materials and methods

3.1 Engine

Engine: Volkswagen AP, Otto cycle, four in-line cylinders, water cooled with forced circulation, manufacturing year 2001.

Bore/stroke: 67.1mm/70.6mm

Displacement: 999cm³

Original compression ratio: 14.1:1[46]

Original fuel feeding: Hydrated ethanol. Indirect *multipoint* injection with Bosch electronic management. Changes done on fuel injection system are described on section 3.3.

During the test performed, the engine ran without air filter nor catalytic converter. Differences on atmospheric pressure, air humidity and others due to adaptations done may lead to different measures of consumption, torque and emissions. So, it is not correct to compare data of the present study with those presented on the owner's manual of vehicles equipped with the engine used [46].

3.2 Steam production

There are three alternatives to supply heat needed for boiling liquid fuel for feed a PVEE: water from cooling system, exhaust gases and lubricating oil. On D'Ávila's work [23] exhaust gases were used, taking advantage of the high temperatures of them to enhance heat transfer. However, were found obstacles in problems like wide range temperature and mass flow change, which caused unstabilities on ethanol vapor supply to the engine. Depending on the heat exchange setup, other possible inconvenient is the sudden vaporization of huge quantities of fuel in case of it contacts a surface which is at temperatures near those of exhaust gases.

On the present study the option for using water from cooling system was made because it operates in a closer temperature range and these temperatures are lower too, turning its deal safer.

Closer temperature difference, oscillating near predetermined values, offers less heat to boil fuel, which is in part compensated by the higher mass flow and heat capacity of water in relation to the air. It is difficult to measure the flow of water of an engine's cooling system, and there are no data from water pump operating conditions available from its manufacturer (Dana Corporation).

To design the heat exchanger, however, it was assumed near 1/3 of energy content on fuel is wasted on the cooling system. It is also assumed a first approach of $\Delta T = 10^\circ\text{C}$ for temperature difference through water cooler, based on thermocouple measured data with both engine and cooler fan working. Inferior Heat Power of ethanol values nearly 27700J/g. Water flow on cooling system, (\dot{q}), is given by:

$$\dot{q} = \dot{Q}_{\text{water}} c_p \Delta T_{\text{cooler}} \cong \frac{\dot{Q}_{\text{fuel}} \times IHP}{3} = 45690W \quad (3.1)$$

$$\dot{Q}_{\text{water}} \cong \frac{\dot{q}}{c_p \Delta T_{\text{cooler}}} = 970g/s \quad (3.2)$$

Where \dot{q} is the available rate of heat rejection and c_p is the sensible heat of warming fuel. Heat exchange area should be enough to supply the amount of energy needed to raise fuel temperature (sensible heat) and to its boiling (latent heat). On the tests performed by D'Ávila on the same engine [23] the maximum observed fuel consumption rate was 5.2g/s. Starting from a room temperature of 10°C until its boiling at 82°C (corresponding to the 120kPa pressure), heat power needed is

$$\dot{Q} [c_p (T_{eb} - T_{\text{inlet}}) + h_{lv}] \cong 5700W \quad (3.3)$$

Where h_{lv} represents latent heat of vaporization of fuel. Inside some limits that does not compromise engine working nor liquid phase of cooling water, its temperature can be controlled using a proper thermostatic switch¹. For calculations effects, it was assumed water lets engine at 96°C and ethanol was boiling as 82°C , generating a 14°C temperature difference. Water temperature should vary a little during its passage through boiler in relation to the temperature drop suffered on the cooler, once heat used for boil fuel is too much lower than those rejected by the cooler (nearly 1/8). Such temperature floating does not damage fuel boiling, as it will be seen later.

Several models were evaluated for the setam generator. The main criteria were: Possibility to put it into the hood of an 1.0 liter displacement engine²; safety; ability to serve quickly to sudden changes on fuel demand; ease of manufacturing and maintenance and; avoid liquid hammer shocks inside fuel pipeline. Measures were taken in a Volkswagen Gol's hood (the car equipped

¹Known on Brazil as *cebola*, which can be translated as "big onion".

²The most sold kind of automobile on Brazil

with the engine used) in order to assure the boiler size is suitable for use in passenger cars in future driveability tests.

The heat exchanger was built in a shell-and-tube model, with water flowing through internal side of tubes and ethanol on the external side of them. Boiling fuel inside the tubes would be safer and would avoid fuel skaking caused by longitudinal and centrifugal forces the vehicle would be subjected. However it was opted by its flow outside the tubes because the ease of sandblasting the tubes' external surface, needed to enhance heat transfer ability in phase change conditions. Such an equipment works on vertical position.

According to existent relationships for ethanol ebullition presented on Chapter 2, minimum superheat temperature which allows ethanol ebullition in smooth metallic surfaces can vary from 6°C up to 23°C . In surfaces with "High Flux" treatment ebullition it is guaranteed to happen at 6°C superheat [44]. On the same study it is observed that sandblasting a smooth chromium surface, a relatively simple and cheap process, may rise phase-change heat transfer coefficient nearly 25%, and double it using oxidation followed by sandblasting. One may note on figure 2.6 that there are few measurement points for heat flux as a function of wall superheat, which forces anybody to adopt conservative estimates when designing these kind of equipment.

Ebullition surface of steam generator used on the present work was polished with a "160" sandpaper, oxidized during oxi-acetylene welding, followed by sandblast cleanup of excessive oxide layer. Such a treatment would be more difficult to perform in the internal side of the tubes, so the boiling on their external side was chosen.

Available space for install the boiler under the hood imposes a relatively big distance between steam exit and its intake into the engine's throat. In order to avoid risk of condensation a jacketed tube was manufactured, that is, a tube protected by a wider one which transports part of water from cooling system, keeping its walls at temperatures above those riskful to ethanol vapor condensation. This tube is still protected by a nearly 15mm thick layer of polyurethane foam. Part of cooling water destined to this task is also used for keeping Venturi (throat) warm.

3.2.1 Thermal exchange area calculations

To determine a suitable extension for boiling fuel, it is needed to find thermal transfer coefficients in both sides, as well as those related to conduction on the tubes. Having no phase change it is also needed to know mass flow ratios and sensible specific heats of involved fluids, in order to determine temperature change as each one advance inside heat exchanger. On the case of the ethanol boiler there is phase change only on ethanol side, while on water side there is just a little temperature change which will be neglected by now, because the amount of heat

rejected by it for boiling fuel is small in comparison to the heat rejected on the cooler. Cooling system water also suffer other kinds of variation during engine operation, the important is to have a conservative estimate of the minimum temperature difference and say if these minimum is enough to lead boiling surafces to the desired temperatures which allow phase change.

Tube beam was built using copper, with 48 3/16" outside diameter tubes. They are 235mm long and have 1/8" as internal diameter. Thermal exchange area at external side is, therefore:

$$0.00254 \times \frac{3}{16} \times 48 \times 2\pi \times 0.235 = 0.3375m^2 \quad (3.4)$$

Where the 0.00254 factor converts inches to meters. On the internal side, where cooling water flows:

$$0.00254 \times \frac{1}{8} \times 48 \times 2\pi \times 0.235 = 0.2250m^2 \quad (3.5)$$

Area of transversal section where flows water which suplies heat, for the 1/8" diameter 48-tube beam, values:

$$A = 48\pi r^2 = 48\pi (1.6 \times 10^{-3}m)^2 = 3.86 \times 10^{-4}m^2 \quad (3.6)$$

For the internal side, subject to almost constant temperature heat conduction, convection coefficient is calculated by the subsequent Reynolds(Re) and Nusselt(Nu) relations:

$$v = \frac{Q}{A\rho} = \frac{0.97}{3.86 \times 10^{-4} \times 0,9615} = 2,65m/s \quad (3.7)$$

$$Re = \frac{\rho v D}{\mu} = \frac{961.5 \times 2.65 \times 3.175 \times 10^{-3}}{2.89 \times 10^{-4}} = 27992 \quad (3.8)$$

$$Nu = \frac{hD}{k} = 0.023 Re^{0.8} Pr^{0.3} = 0.023 \times 27992^{0.8} \times 1.80^{0.3} = 99.07 \quad (3.9)$$

$$h = \frac{Nu.k}{D} = \frac{99.07 \times 0.679}{3.1785 \times 10^{-3}} = 21185W/m^2K \quad (3.10)$$

Where v means fluid speed, A represents thermal exchange area. ρ , density; k , thermal conductivity; D , diameter and water properties are used at 370K (96.85°C).

Thermal resistance (Ω_{Cu}) for heat conduction on copper tube walls values:

$$\Omega_{Cu} = \frac{\ln(r_2/r_1)}{2\pi k_{Cu}L} = \frac{\ln(0.0047625/0.0031785)}{6.2832 \times 397 \times 0.235} = 6.917 \times 10^{-4}m^2K/W \quad (3.11)$$

It remains to determine thermal exchange coefficient on external side of the tubes, where there is ebullition. For a heat exchanger, global heat transmission coefficient is given by the following relation:

$$U = \frac{1}{\frac{D_i}{D_e} \frac{1}{h_{int}} + \Omega_{Cu} + \frac{1}{h_{ext}}} \quad (3.12)$$

For the present heat exchanger, the engine's maximum power regime requires 5700W as thermal load. Being its boiling area 0.3376m^2 , maximum heat flux required is:

$$q'' = \frac{5700\text{W}}{0.3376\text{m}^2} = 16886 \frac{\text{W}}{\text{m}^2} \quad (3.13)$$

With an estimated temperature difference between fluids of 14K, minimum thermal exchange coefficient should be:

$$U_{min} = \frac{q''}{\Delta T} = 1206 \frac{\text{W}}{\text{m}^2\text{K}} \quad (3.14)$$

To reach this value h_{ext} should have a minimum value, calculated under the following conditions:

$$\begin{aligned} 1206 &= \frac{1}{\frac{D_i}{D_e} \frac{1}{21185} + 1.441 \times 10^{-5} + \frac{1}{h_{ext}}} \\ 1206(4.5879 \times 10^{-5} + \frac{1}{h_{ext}}) &= 1 \\ 5.5221 \times 10^{-2} + \frac{1206}{h_{ext}} &= 1 \\ \frac{1206}{h_{ext}} &= 0.944779 \\ h_{ext,min} &= \frac{1206}{0.944779} = 1276.5 \frac{\text{W}}{\text{m}^2\text{K}} \end{aligned} \quad (3.15)$$

One can note minimum heat exchange coefficient required for external side is near the global one, and both are much lower if compared to the other ones (internal side and copper walls); so these later have few importance. If the external surface has the heat transfer coefficient enough high, temperature difference of boiling ethanol and boiling surface is almost the same difference between ethanol and water. In these conditions, wall superheat can be estimated as:

$$\frac{q''}{h_{ext}} = \frac{16886\text{W}/\text{m}^2}{1276.5\text{W}/\text{m}^2\text{K}} = 13.07\text{K} \quad (3.16)$$

Which is near the difference of 14K between fluids and is enough to boil ethanol, according to the figure 2.6. Using trial and error method, convection coefficients and wall superheat can be more exactly determined, converging to values which fit on theoretical relations.

Vehicle's vibrations (if tests were not performed on stationary engine) would improve boiling because it would help bubble release, freeing warm surface to transmit heat to new liquid amounts, with higher transfer coefficients. Lee *et al.* [47] have studied influence of vibrations on phase-change thermal exchange, arriving to the determination of vibrational Reynolds and Nusselt numbers as a function of its frequency and amplitude. These parameters were not taken into account because they require vibration measurements on vehicle chassis or on the dynamometer, but it must be pointed they would be a positive influence on fuel boiling.

3.2.2 Head losses

Operation curve of water pump (relating flow and pressure difference) is not known, also not the hydraulic behaviour of engine's internal cavities. It was estimated only head losses on cooling system due to hoses and on the boiler, to evaluate impact of its installation on the original cooling system. calculations refer to the maximum flow regime estimation (around 1kg/s), allowing admission of head loss proportional to the square of mass flow.

With a Reynolds number formerly estimated as 27992, surface treatment of extruded copper tubes (dimensionless rugosity $\epsilon = \frac{e}{D} \cong 2,4 \cdot 10^{-4}$), which leads to a friction factor (f) of 0.023; head loss (ΔH) on them values, proportionally to the length L of each tube:

$$\Delta H_{\text{beam}} = f \frac{L}{D} \frac{v^2}{2} = 0.023 \frac{2.35 \times 10^{-1}}{3.2 \times 10^{-3}} \frac{2.59^2}{2} = 5.665 mca \quad (3.17)$$

At the boiler's water inlet and outlet there are sudden diameter expansions/reductions. These diameters are respectively 31.75mm(1.25"), 28mm and 108mm on the top (inlet) side and reverse order at its bottom (outlet) side. Localized head loss coefficients are 0.06 for 31.75mm–28mm diameter reduction/expansion. For the 108mm–28mm diameter sudden reduction this coefficient values nearly 0.46 and, for its reciprocal expansion, 0.86. Such results were collected from [48].

$$\Delta H_{1.25''-28} = K_{1.25''-28} \frac{v^2}{2} = 0.2 mca \quad (3.18)$$

$$\Delta H_{28-1.25''} = K_{28-1.25''} \frac{v^2}{2} = 0.2 mca \quad (3.19)$$

$$\Delta H_{108-28} = K_{108-28} \frac{v^2}{2} = 2.88 mca \quad (3.20)$$

$$\Delta H_{28-108} = K_{28-108} \frac{v^2}{2} = 1.54 mca \quad (3.21)$$

For a 28mm internal diameter connection which links two hose sections between boiler outlet and original cooler's inlet:

$$\Delta H_{\text{conexão}} = 2 K \frac{v^2}{2} = 2 \times 0.06 \times \frac{1.62^2}{2} = 0.15 mca \quad (3.22)$$

And finally the head loss caused by water admission on 1/8" tubes from the 128mm length chamber (considered an infinite reservoir for these calculation purposes). Similar head loss occurs at beam exit.

$$K = 0.5 \longrightarrow \Delta H_{\text{ent/sai}} = 0.5 \frac{2.59^2}{2} = 1.67 mca \quad (3.23)$$

Total head loss, taking out of account distributed losses alongside tubes and connections, is:

$$\Delta H = \sum_i \Delta H_i \frac{v^2}{2} = (0.06 + 0.86 + 0.46 + 0.06 + 2.05 + 2.06) \frac{1.62^2}{2} \quad (3.24)$$

$$\Delta H = 3.22mca \quad (3.25)$$

Summation of distributed head losses of tube beam to localized losses are 8.88 mca (meters column of water). For a mass flow estimated in 1kg/s these head losses require an additional pumping power estimated as:

$$P_{\text{pump}} = \Delta H \rho g Q = 8.88m \times 10^3 \frac{kg}{m^3} \times 9.81 \frac{m}{s^2} \times 10^{-3} \frac{m^3}{s} = 87W \quad (3.26)$$

proportional to the cube of mass flow.

3.3 Fuel injection

The original fuel injection system was substituted by a real-time programmable one, brand Fueltech, model RacePRO-1Fi. With it parameters like fuel injector time and spark ignition angle can be adjusted with the engine running. This way it is possible to change the moment of spark without rotate spark distributor shell and read data from stroboscopical beam at once. Spark ignition timing control was also used for allow to run the engine in conditions near to those define in its original fuel injection, allowing more fair comparisons between liquid and vaporized fuel injection.

Parts of Natural Gas conversion kit, supplied by Netgás/KGM GMS-7 also were used, in its version 5.0. Cylinder used to store gas and the pressure reductor were not purchased because they are not required in a pre-vaporized liquid fuel system.

Vaporized fuel intake was made thorough gas carburetion, in a similar way of earlier PVEE studies as those of [22, 23] and Celere [19]. Main difference relies on some technological advances on equipment used.

Brazilian GNG fleet has increased too much in recent years, due mainly to the price of this energy source which allows to save up to 50% of cost by distance run in relation to gasoline, according to Brazilian prices. Spreading of CNG conversion kits without care on consumption and environment has forced Conama (*Conselho Nacional de Meio Ambiente*, or National Environment Council) to determine emission limits since 2002 [49]. Emissions of methane, ethanal (acetaldehyde) and methanal (formaldehyde) are the most important on vehicles which operate with this fuel.

Ancient CNG kits have a valve between pressure reductor and suction device (Venturi), regulated at end of installation process just with experience of mechanical repeirer (known on Brazil by “ear adjustment” method, because uses engine noise to determine the better point of adjust). These adjusts were used to be made at nearly 3000rpm (if tachometer is present, else the engine speed would be “ear supposed” too), with shifter in neutral position.

The partially purchased CNG kit, supplied by Brazilian manufacturers KGM and Netgás, has a gate valve with step motor which allows fine adjusts in its opening (concentrated head loss) with engine running. Its 8mm stroke is divided into 250 steps. A secondary Electronic Central Unit (ECU) acquires data from original ECU, from Lambda sensor and from other ones which compose vehicle's fuel injection system. Such apparatus reduce consumption and pollutant emissions on the atmosphere.

A joystick used for on-the-fly gate valve opening adjustment in some position different of those pre-set was also supplied. This makes possible to ignore the signal of Lambda sensor and another ones which make the engine run with mixtures close to stoichiometric. This is needed to increase air-fuel ratio, looking for better thermodynamic efficiencies and lowering air choking at throttle valve. This joystick controller has two buttons which order to the valve to open or close it in one step each. There are also green and red LEDs which indicate if the air fuel mixture is lean or rich, respectively.

The acquired CNG kit has also a switch which allow to change from the original fuel injection system to it and vice versa. On the current passenger vehicles adapted to CNG there is the need to run the engine for some seconds with the original fuel and so switch to the gaseous one. Despite the easiest to burn previously mixed gaseous mixtures, specially at low temperatures, engines not originally designed to run with CNG need that.

In the PVEE case there is also the need of operating the engine for some minutes with liquid fuel until water of cooling system reach a high enough temperature suitable to produce ethanol steam at specified pressure and ratios. Experiments have shown nearly 10 minutes were enough to that, once the thermostatic switch which allows water circulation through cooler when it reaches a determined temperature.

3.3.1 Other accessories

The ethanol boiler has a float valve to control liquid fuel inlet and keep it as a correct level in order to keep the tubes immersed in fuel without overflow the boiling chamber limits. Such a fail would put liquid fuel when the ECU is programmed to run with gaseous fuel, with disastrous consequences. A polyamide (Nylon®) float, described in details on the Appendix D, with mass of 117.3g and volume estimated in 239.3cm^3 is stored into the heat exchanger's auxiliary chamber (2" tube), under the lid where fuel comes in. With all the volume of cylindrical part made of polyamide immersed on ethanol, whose density at 360K (87°C) values 728.3kg/m^3 [50], a net thrust (discounted the gross float weight) of 57.0 grams-force appears, more than the enough to keep closed this valve. Such a valve (used on carburettor bowls) has an orifice in which fuel passes through, which diameter is 2.2mm. It requires a thrust force of $(1.1^2 \times \pi \times \Delta P)$

mm² to keep it closed, where ΔP means the pressure difference between exchanger's inner and outer sides. A fuel doser Racing Parts model HPi (used for increase pressure in carburetor bowls too) was regulated to release fuel to the exchanger feeding hose at a pressure nearly 0.35kgf/cm² higher than those found inside it. This is done through a hose connected to the lower part of the heat exchanger which carries its pressure to the fuel doser. Adjusting the doser for a lower pressure difference the fuel level stabilizes lower, because a lower thrust is required to close the valve. The opposite occurs when higher pressure differences are set up on the fuel doser. This way is possible to adjust the desired level of liquid inside the fuel boiler. It was observed during ethanol boiling the bubbles rose up to 30mm above fuel level, which lead to an adjust in the range of 30–40mm below the top of ebullition chamber. In this case a 13.3 gram-force net thrust was enough for keeping the fuel in a suitable level.

Once conservative estimates were made for the heat exchange areas, the boiler was able to feed satisfactory amounts of steam in all the working regimes studies.

At heat exchanger's outlet there is a 3/8" diameter sphere valve, manually operated. This way switching fuel feeding system from liquid to gaseous can be made smoothly or suddenly. Solenoid of original CNG kit does not allow fuel rates above 2.2g/s on given conditions. Taking into account data from table 2.4, heat exchanger working at 0.8kgf/cm² above local atmospheric pressure in the range of 690mmHg; ethanol vapor density valuing 2.80kg/m³ (calculated using the interpoler polynomuim of Appendix B.1), while methane with its molecular weight valuing 16g/mol and heat combustion power as 56375J/g would have a power-volume relation of:

$$56375kJ/kg \times 0.714kg/m^3 = 40251kJ/m^3 \quad (3.27)$$

While saturated ethanol vapor at 172kPa has an amount of heat power which values:

$$27710kJ/kg \times 2.797kg/m^3 = 77505kJ/m^3 \quad (3.28)$$

It indicates the same volume of gaseous ethanol generated by the described boiler has more energy available than methane in the normal conditions of temperature and pressure (normal cubic meter). At 25°C (297K) one cubic meter of methane at 1 atm of pressure has:

$$40251 \frac{273}{297} = 36998kJ/m^3 \quad (3.29)$$

Anyway the amount of energy is greater in ethanol case. This shows the lack of capacity to allow ethanol vapor passage through piping is due to corrosion of ethanol on aluminum parts of original solenoid valve. Gas viscosities are not influenced by pressure if they are near their critical values [51], and increases together temperature. At 100°C values 1.040×10^{-4} poise for ethanol vapor and 1.331×10^{-4} for methane[52].

On the smooth opening mode, sphere valve was open with engine cool and step valve closed. As pressure inside exchanger increased, vaporized fuel production arised and liquid fuel injection lose participation through real time injection time adjustment. The programmable ECU has also a fast-adjustemet mode, which allows to increase or decrease instantaneously the whole injection map in 1% steps, as well as to advance or delay thw whole spark timing map in 1 degree steps. This way the fuel injectors get out scene slowly until fuel feeding were done just by gaseous ethanol.

With this condition reached the CNG kit was switched to gaseous fuel supply in order to allow its control by KGM/Netgás fuel injection module, resting to the Fueltech programmable ECU the task of control spark timing and supply accurate MAP measures to the other fuel managers. An injector simulator, part of CNG kit, sends signals to the original ECU lying to it, so its behaviour is like fuel injectors were still working. Fueltech programmable fuel injector have shown on its screen fuel injector opening time even when operating with gaseous fuel because it.

It was needed to keep the solenoid of CNG kit connected to the engine's electrical system, because switching from liquid to gaseous fuel has not been done if the presence of this part was not detected. Greater solenoid valves were not acted, neither using relays or resistive impedances in order to lie to it. Solid-state relays could be used, because they are acted with too low amounts of energy.

After passing through both gate valve and sphere valve, ethanol vapor follows a jacketed (protected) tube by another one in which passes water from cooling system. So, fuel is transported until the throat without risk of condensation. This throat, brass made, also has holes which receives hot water for the same goal. Between this jacketed tube and the fuel intake throat there is a 20mm transparent hose in order to assure there is no fuel condensation on the steam supply system. The opaque part of vapor transport was protectec with polyurethane expansible foam, while tha valves which need to be unmounted to much times were coated with fiberglass. The hose which carried water from the engine to the boiler were also insulated.

The same 12 Volts signal is forked to both solenoid and injector simulator. So one can think the later is turned on onky when the engine starts to run with gaseous fuel. Due to the distance between vapor generation and intake, (nearly 700mm) a signal retarder (figure 3.3.1) was installed in order to assure there is a time interval between the beginning of gaseous fuel injection work and the cut on liquid fuel supply. The retarder is adjustable to wait from 0.5s to 5.s, keeping the 12 Volts on the fuel injector simulator.

3.3.2 Adaptations on gate valve

Initially trials were made on the way of run PVEE with the original gate valve from CNG kit, which was damaged by corrosion, high temperatures and liquid fuel caused by fails on steam generator fails. Its coils had lost their varnish and short-circuited. Other valve, the same model, was used, with some adaptations. in order to protect its sensible parts from excessive heat and fuel chemical attacks. Spheric point of original valve was exchanged by a pin which moves inside a cylindrical calibrated plug, in order to pass gaseous ethanol through a narrower window than the originally designed for CNG. This was needed because no pressure reducer was used, which would require a low head loss coefficient gate valve and which would allow a quasi-linear relationship between gate area and mass flow. The method used was reducing vapor from boiler's pressure to a value near to that found at air intake through a restrictive passage. Such adaptations are described on the Appendix D.

3.3.3 Suction throat or Venturi

The Venturi supplied has shown to be not suitable for use in a PVEE. The main reasons were:

- Non-uniform fuel distribution due to its inlet happen perpendicularly to air passage. Being throttle axis not parallel to the crankshaft unequal fuel concentrations can happen in some places, and this is highly undesirable; because it can lead to different air-fuel ratios on the cylinders. The new Venturi receives ethanol vapor in tangential way, in a thoroidal chamber (with a doughnut shape), being so sucked by intook air.
- Need to keep this device warm in order to avoid vapor condensation inside it, periodic drainages and liquid shocks. Because that the Venturi was built in brass, a good heat conductor, and endowed with channels in which water from cooling system passes through, keeping these pieces warm at temperatures higher than those of ethanol condensation.
- Adequate fixation to throttle body. Original Venturi was conceived to be attached to throttle body with aid of its fixation on air filter, which is not present.
- Seal. By same reason of previous item, there are O'rings and other devices for avoid fuel leakages in the new way it was attached to the engine.
- Minimize head losses; producing a piece with smoother surface roughness, increasing minumum air passage diameter, eliminating undesirable angles and sharpen edges.

Throat's internal side was insulated with non-acetic silicone rubber, having internal channels to store it. This was necessary to minimise heat transmission to intake air, which may lower its density and cause damages to volumetric efficiency. What is alone a problem in gaseous fuels indirect injection engines.

Pressure in different Venturi sections was calculated for the maximum power regime (maximum air flow rate) based on equations which define air properties in a compressible flow through a convergent-divergent throat [53]. There was a concern in determine absolute pressure in the smallest section of the throat, where fuel suction occurs.

Air consumption with the same engine was previously measured by D'Ávila [23]. At full load, at 5000rpm, the engine requires 29.56g/s of air being fueled with vaporized ethanol at room temperature 32.5°C and pressure of 696mmHg (92793Pa). Pressure and speed of air were calculated on the 2" hose which links *plenum* to throttle body and on the smallest throat section, with 35mm diameter.

$$\rho_0 = \frac{p}{RT} = 1.092 \text{ Kg/m}^3 \quad (3.30)$$

$$\frac{Q}{A} = p_0 \sqrt{\frac{\gamma}{RT_0}} M \left[1 + 0.5(\gamma - 1)M^2 \right]^{0.5 - \frac{\gamma}{\gamma-1}} \quad (3.31)$$

$$\frac{Q}{A} = 14.585 \text{ kg/m}^2 \text{ s} \rightarrow M_1 = 0.03869 (13, 2 \text{ m/s}) \quad (3.32)$$

$$\frac{A_1}{A_2} = \frac{M_2}{M_1} \left[\frac{1 + 0.5(\gamma - 1)M_2^2}{1 + 0.5(\gamma - 1)M_1^2} \right]^{0.5 - \frac{\gamma}{\gamma-1}} \quad (3.33)$$

$$\frac{p_1}{p_2} = \left[\frac{1 + 0.5(\gamma - 1)M_2^2}{1 + 0.5(\gamma - 1)M_1^2} \right]^{0.5 - \frac{1}{\gamma-1}} \quad (3.34)$$

$$M_2 = 0.1463 (49.81 \text{ m/s}) \quad M_3 = 0.0818 (27.85 \text{ m/s}) \quad (3.35)$$

$$\frac{p_1}{p_0} = 0.9989 \quad \frac{p_2}{p_0} = 0.9852 \quad \frac{p_3}{p_0} = 0.9953 \quad (3.36)$$

Sections 0, 1, 2 e 3 refer to, respectively, the *plenum* where air is found at stagnation properties; the 2" hose already described; the 29 mm diameter section which is the smallest of the throat of original CNG kit and; the 35mm diameter section of the new Venturi built. M means Mach number, T is the temperature, R the universal gas constant and *p* means absolute pressure.

The higher air speed calculated are too low in comparison to sound speed in given conditions, which turns these flows practically incompressible. Pressure drop values are low to, in comparison to atmospheric pressure, leading to lesser pressure differences in relation to the ethanol vapor produced, easing control of fuel flow ratio through step motor gate valve. The 35mm diameter for suction was needed just to give place to the toroidal chamber in which vapor

circulates before being intook and to force alight increase in air speed at this point aiming to create a more homogeneous air-fuel mixture.

3.3.4 System response to fuel demand variations

An important concern on a fuel feeding system project for a PVEE is the fuel feeding rate according to its instantaneous requirement (*on demand* supply). It is also needed to deal quickly to fuel consumption alterations, feeding the quantity correspondent to the operation regime desired by the vehicle conductor. As fuel is not already transported in gaseous form, which is the case of LPG, CNG and some Hydrogen ICE's; it is important to produce fuel vapor at a rate which is the nearest the possible the required rate. Closing production and consumption rates great vaporized fuel accumulations or disaccumulations are avoided on he fuel supply system. The ideal stored quantity of vaporized fuel is a match between safety (the lesser quantity of saturated fuel) and capacity in support sudden demand increases.

In PVEE, fuel admission on the steam generator is controlled by a float and a needle valve, following he principle of carburettor bowls. At rate in which fuel enters into the heat exchanger, its level raises raising together float thrust, until reaching an enough value to seal the 2mm diameter hole making force to resist to fuel feeding system. As vaporized fuel is consumed and leaves the chamber, level lowers again allowing new fuel inlet for reposition.

On the heat exchange surface, ebullition rate is not exactly proportional to the temperature difference between fluid and wall. There is a minimum temperature to start ebullition, which depends on saturation temperature. Any way, as boiling fluid temperature gets close to the temperature of fluid/wall which supplies heat (in this case water from cooling system) there is a decrease of steam production, until reaching zero when temperature difference is lower than those minimum value needed to start ebullition. This way, when fuel demans decreases, readily produced steam accumulates on the heat exchanger and increases its pressure, leading to an increase on its saturation temperature; allowing it to reach higher temperatures before take its steam form. The meeting to water/wall temperature decreases vapor production rate, adequating it to the demand after a small quantity of time which depends on the vapor volume on the chamber and other heat exchanger features.

In the opposite case, when demand for fuel increases suddenly, in example when the driver requires higher power for climbings and overtakings, excessive fuel vapor exit decreases pressure of that vapor which is still in the exchanger, with imediate temperature decrease. The temperature drop is like those observed in refrigeration systems when refrigeration fluid suffers a sudden pressure reduction. On this case temperature difference between vaporizing fuel and

wall increases. While vapor production rate does not reach the desired rate, part of saturated liquid vaporizes without receive more heat from thermal exchange surface than it has already received. This happens just because vapor title increases with non-resisted expansion of a fluid. hat is *flash vapor* generation, found normally in refrigeration systems and water steam facilities. While vapor production is not adjusted, *flash vapor* anserws to the engine's requirements. This way the fuel feeding system proposed to the PVEE may adjust itself to floatings in fuel consumption rates, normal in non-stationary ICE's.

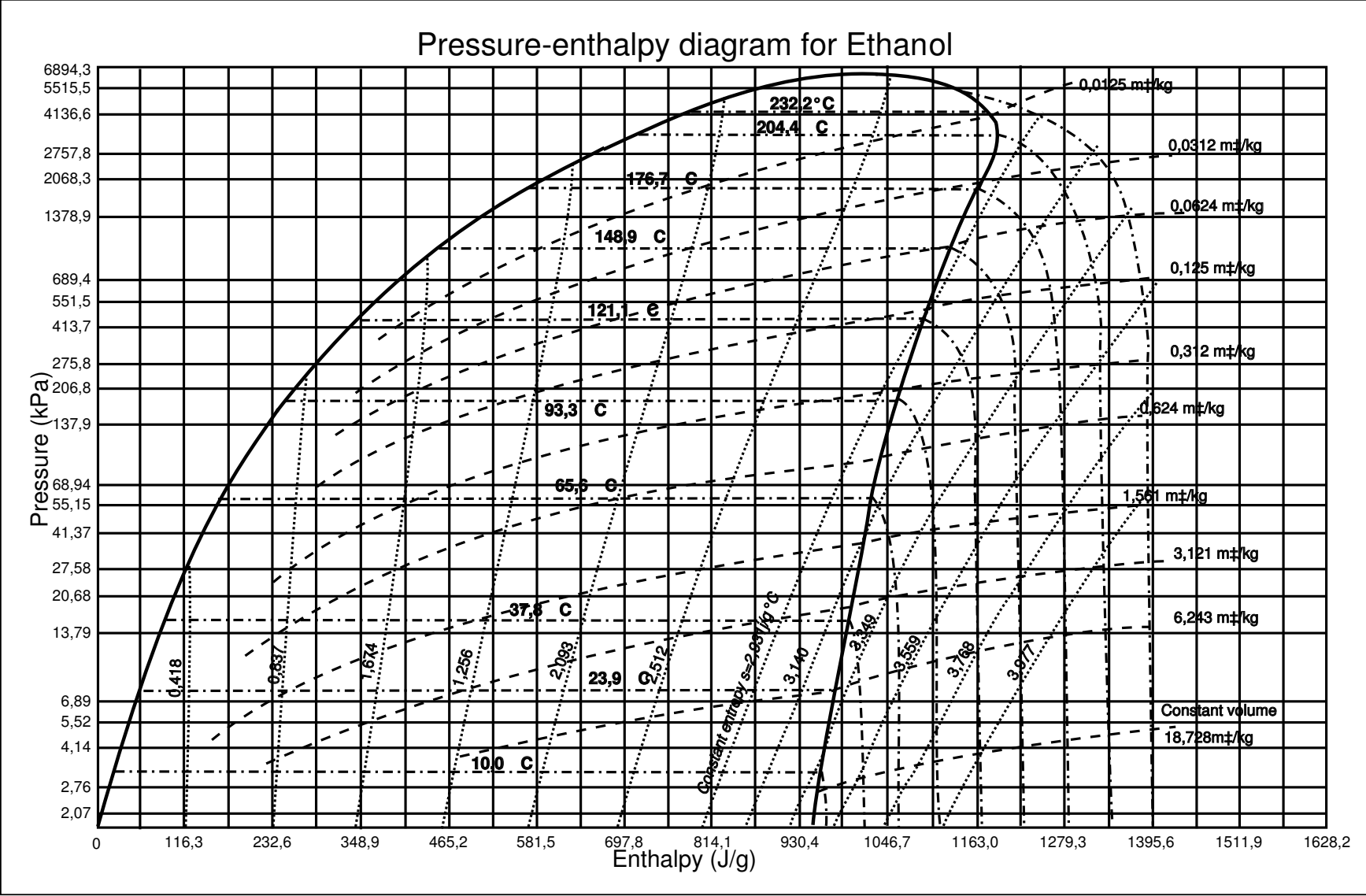


Figure 3.2: Relations between pressure, enthalpy, entropy, specific volume and saturation temperature for ethanol.

Adapted and converted to SI units from [54].

Figure 3.2 shows a pressure-enthalpy diagram for ethanol. On it, it can be seen relations among some physical and thermodynamical properties for this substance.

3.3.5 Control of vapor pressure

As cooling water temperature is not constant through operation of an ICE, the difference between it and saturating ethanol may vary too. As described earlier, consumption variations also change the pressure inside heat exchanger.

In a first approach a pressure regulator valve, used in domestic LPG ovens, with nominal flow rate of 1 kg/h, was used. This way the pressure inside heat exchanger could vary but its exit would happen at constant pressure, of just some kPa above atmospheric pressure, such a way the original step motor and gate valve of GNG engine conversion kit could be used. At this stage step motor was already protected from high temperatures and of contamination by ethanol. The other important difference in comparison to the original gate valve was the replacement of original aluminum spherical point (15 mm diameter) by another, conical, made with PTFE (Politetrafluorethylene) followed by an adequate hole, compatible to its dimensions, on gate valve's case. Regulator pressure valve also was jacketed with water from cooling system in order to avoid steam condensation and heat loss to its relatively great metallic mass.

This solution has shown inadequate because this regulator valve is not capable to feed the required fuel rate for the engine. Higher flow rate valves could be used, using the same physical principle. Experience has shown maximum mass flow rate for ethanol vapor is nearly the half of those marked on it as nominal mass flow for the original fuels it was designed, as LPG or CNG. Higher viscosity of ethanol vapor (not measured but inferred because van der Waals interactions between gaseous ethanol molecules) is maybe the main cause of this difference.

Using in the further trial a brass plug on the gate valve, calibrated hole and pin; pressure regulator was discharged and fuel was intake directly at the pressure it was produced at heat exchanger, in such a way the fuel passage was regulated by a very lower transversal area in order to cause the required pressure drop, as explained earlier. The inconvenient phenomenon observed was the influence of cooling water temperature floating, according to the cooler's fan operating cycles. Original thermostatic switch closes its electrical contact allowing electric current passage when water temperature reaches 95°C and opens its electrical contact when this temperature falls under 87°C. This thermostatic switch was replaced for another which closes its contact at 100°C and opens it at 95°C. It was noted for this case the heat exchanger pressure varied on the 0.7–1.0 kgf/cm² range. Such floating is undesirable for a stable operation of a PVEE. In permanent regime of engine speed and throttle position air fuel mixture had travelled from rich to lean and to rich again, cyclically; forbidding adequate measurements to be made.

Taking the decision of keeping water cooler's fan turned on full time its temperature had fallen under 80°C, and when the difference between it and saturating ethanol was under the 6°C limit, ethanol vapor production had ceased immediately and the PVEE went without gaseous fuel supply, confirming theory presented by Thome [44].

An almost readily available and adequate solution for experimental purposes was installing some device to control fan speed, in order to keep constant cooling water temperature.

3.4 Other experimental apparatus

Foucault current dynamometer brand Schenck, model W70.

Load cell Alfa Instrumentos, model IC331/0

Electronic fuel injection scanner Tecnomotor Rasther TM 131.

Temperature acquisition system Strawberry Tree ACPC, 8-channel analogic, using cromel-alumel thermocouples. Temperatures measured were: Suction throat; intake manifold. exhaust manifold; ethanol vapor at exchanger's exit; lubricating oil; cooling water before and after its passage through steam generator.

Water cooler brand Bongotti, unknown model. Its is well known this cooler was used in Ford Corcel 2, a 1.6 liter sedan manufactured in Brazil in the mid-1980's.

Weighing device Hobart Dayton model CB15 with 15kg load capacity and 10g resolution, chronometer Alfa Instrumentos TCC11 and optical sensors for fuel consumption measurements.

Air flow meter Bosch part number 0 280 218 053.

Manometer Class B ABNT (Brazilian Technical Norm Association) from 0 to 3 kgf/cm² for monitoring steam generator.

Manometer Emfase from 600mmHg(vacuum) up to 0.5kgf/cm² for MAP measurements.

Gas analyzer Tecnomotor TM131.

3.5 Bench setup

Equipment are disposed and communicate themselves according to the figure 3.5. Electronic devices built are described in detail in previous figures. Weighing scale had some mounted optical devices in it, connected to a chronometer (38), to determine time needed for consumption of some quantity of fuel, when opening of valve (21) fills secondary tank (18) over the weighing scale (16) for a new consumption measurement. Signal ignorer (37) sends the signal from the first passage of weighing scale hand through optical sensor to the chronometer, ignores the signal of second interruption of light when scale hand returns, transmits the third signal to the chronometer in order to stop time count and on the fourth passage resets up the chronometer for

a new measurement, when secondary tank is refilled. After first passage a 100g weigh was put over the secondary tank, such a way at third hand passage there were 100g less fuel on the tank in comparison to the first passage. This way consumption for 100g of ethanol was measured, either in liquid and gaseous way, because fuel return to the pump is in both forms directed to the secondary tank and just fuel effectively consumed was measured. Each fuel supply mode has its own return device, the original reducer for liquid and the carburettor doser return port in the case of gaseous fuel line.

Secondary ECU acquired from CNG kit cuts fuel supply when gaseous fuel is required, interrupting electric signal of fuel injectors, but keeping fuel pump running and fuel rail³ pressure. On the other hand, valve (25) is open, allowing passage of gaseous fuel. Manometer (29) indicates if there are good conditions for that, pressures near 0.7kgf/cm^2 indicate it is already possible to operate the engine with gaseous fuel. Liquid-gas switch is in this case manually made on both CNG kit and on sphere valve (31). Manometer (30) informs pressure in which heat exchanger is supplied, having inside it (exchanger) a needle valve and a float like in carburettor bowls. HP Racing Parts doser (20) acquired allows us to adjust pressure difference between heat exchanger and pump supply, which is informed to it through the 3/16" hose (28). A 0.22kgf/cm^2 pressure difference was adjusted in order to adjust an adequate level of liquid inside the exchanger. Exchanger supply was made through the original fuel pump and the original pressure regulator close, once the doser is operating at lower pressures when ethanol vapor is produced and no fuel goes to the liquid fuel rail.

Exhaust gases pass through refrigeration coils before go to gas analyzer, in order to retire any water vapor from it. This vapor could damage such an equipment. Water from cooling system passes through steam generator (22) before follow its way to the cooler (23), having part of it deviated to the line (27) in order to keep warm the Venturi and the jacketed tube (26), avoiding fuel condensation since its production site until intake manifold.

Description: 1-Air intake with flow meter; 2-Plenum; 3-Hose; 4-Hollow Venturi; 5-Throttle body; 6-Intake manifold; 7-Fuel rail; 8-Engine; 9-Exhaust manifold; 10-Condenser; 11-Gas analyzer; 12-Flywheel and Cardan axis; 13-Dynamometer and load cell; 14-Accelerator cable; 15-Dynamometer readings; 16-Weighing device; 17-Main fuel tank; 18-Secondary fuel tank; 19-Fuel pump; 20-Carburettor doser; 21-Valve of secondary tank; 22-Steam generator; 23-Water cooler; 24-Gate valve with step motor (fine adjust of gaseous fuel flow); 25-Solenoid valve of gaseous fuel system on-off; 26-Jacketed tube; 27-Warm water of Venturi and jacketed tube line); 28-Pressure information to the doser; 29-Pressure on steam generator; 30-Generator feeding pressure; 31-Sphere valve (on-off gas feeding); 32-MAP manometer; 33-ECU Netgás

³Known on Brazil as *flauta*, which means "flute".

with injector simulator; 33-ECU Fueltech; 35-Netgás Joystick; 36-Spark plugs and distributor; 37-Signal ignorer; 38-Chronometer; 39-Signal retarder; 40-Fan speed controller; 41-Pressure reducer for original fuel injection system.

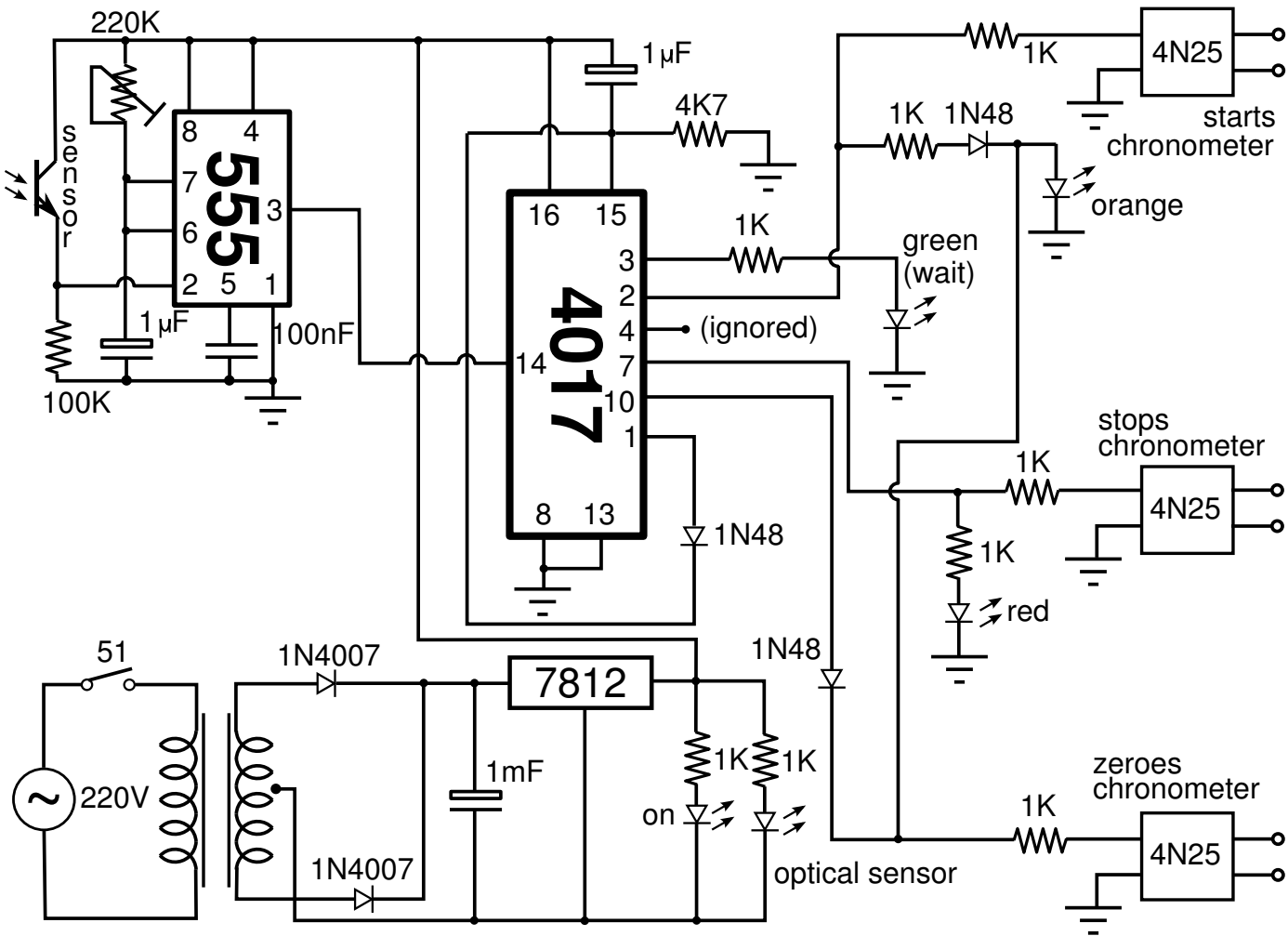


Figure 3.4: Electrical diagram for signal ignorer.

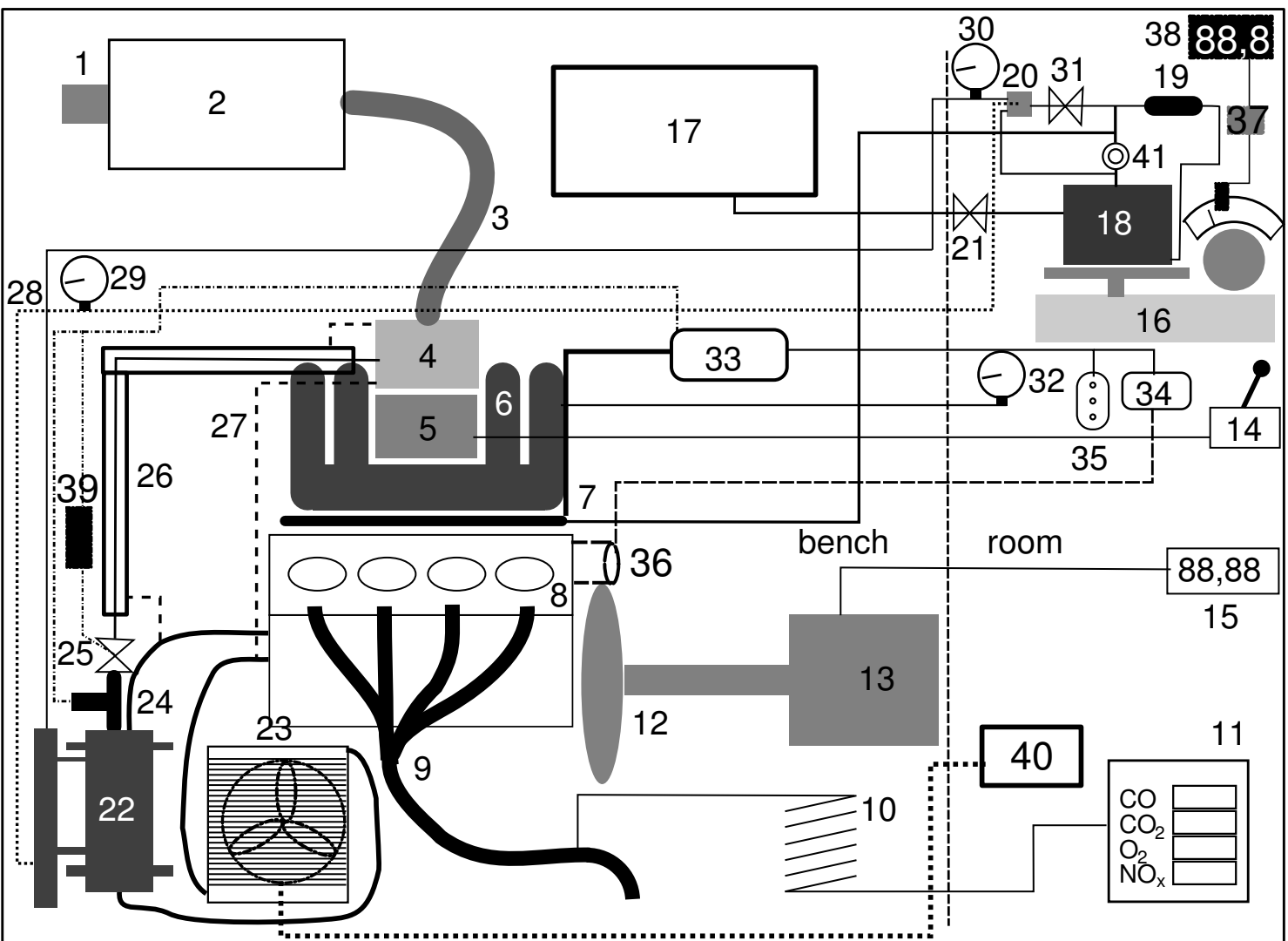


Figure 3.5: Equipment disposal.

Chapter 4

Experimental Results

4.1 Original fuel injection system

For comparison effects the engine was tested with its original electronic fuel injection system before running it on vaporized ethanol. Following Tables 4.1, 4.2 and 4.3 supply results about the engine operating under these conditions. The aim of this phase of experiments is to compare fuel consumption and emissions among both fuel injection systems for the same torque-speed combinations. Ten regimes were studied, being nine formed through combination of three throttle valve openings (25%, 50% e 100%) to three crankshaft speeds (2000. 3000 e 4000 rpm); and idle.

One could note with accelerator cable totally released the Tecnomotor fuel injection scanner presented a 4° throttle valve opening, while Fueltech's ECU was calibrated to indicate 0.00% in this condition. The same way maximum opening was detected as 82° and 100.00% by the same equipments, respectively. To work with measurements as a percentage of maximum opening, degree-based measurements from original ECU were converted to percentage through the expression

$$\left(\frac{d - 4}{78}\right) \times 100 = \text{porcentual} \quad (4.1)$$

Where d is the throttle opening measurement in degrees.

All tables were calculated based on collected data from the engine (Appendix C) and calculated in an ODF (Open Document Format) spreadsheet using the software Open Office.org version 2.0.4. The formulae used are described on section B.2.

4.2 Vaporized ethanol

For each regime studied, experiments started with vaporized ethanol supply from some look-like condition in relation to those determined on the original fuel injection system. First regime

Table 4.1: Dry Base Combustion Products, with original fuel injection system.

ω (rpm)	Total flow DBCP (mol/h)	CO (g/h)	THC (g/h)	NO _x (g/h)	CO ₂ (g/h)	O ₂ (g/h)	N ₂ (g/h)
Idle							
840	254	30	2	0	1589	76	6351
Throttle at 25% (23 degrees)							
2000	579	52	3	34	6467	128	25154
3000	1156	110	13	67	7378	255	29158
4000	1195	151	25	43	7469	310	29821
Throttle at 50% (45 degrees)							
2000	1458	102	55	85	9107	387	36116
3000	2315	194	43	136	14566	570	57598
4000	3183	339	130	187	19745	866	78942
Full throttle (82 degrees)							
2000	1460	1275	51	44	8287	149	37248
3000	2353	2214	64	54	13252	196	60069
4000	3381	2329	60	128	19784	335	85624

Table 4.2: Specific production of some gases and specific fuel consumption, with original fuel supply.

ω	Specific production (g/kWh)					Specific consumption of ethanol
(rpm)	CO	THC	NO _x	CO ₂	O ₂	
Throttle at 25% (23 degrees)						
2000	6.6	0.38	4.25	820	16.2	515.5
3000	14.3	1.68	8.73	956	33.1	585.0
4000	33.4	5.49	9.57	1657	68.7	1004.9
Throttle at 50% (45 degrees)						
2000	8.5	4.58	7.09	758	32.2	450.5
3000	10.2	2.25	7.11	763	29.9	455.9
4000	13.8	5.30	7.64	806	35.3	486.1
Full throttle (82 degrees)						
2000	103.8	4.13	3.56	675	12.2	518.6
3000	109.4	3.17	2.65	655	9.7	511.4
4000	82.7	2.13	4.54	702	11.9	509.9

Table 4.3: Other calculated parameters, referring to engine operation with original ECU.

ω (rpm)	Torque (Nm)	Power observed (kW)	η (%)	Consumption ethanol (g/s)	Λ	λ	MAP (mmHg)
Idle							
840	–	–	–	0.26	8.70	1.04	306.0
Throttle at 25% (23 degrees)							
2000	37.7	7.89	28.0	1.13	8.07	0.97	513.5
3000	24.6	7.72	24.7	1.25	8.42	1.01	383.5
4000	10.8	4.51	14.4	12.6	8.58	1.03	253.5
Throttle at 50% (45 degrees)							
2000	57.4	12.02	32.1	1.50	8.70	1.04	593.5
3000	60.8	19.10	31.7	2.42	8.62	1.04	573.5
4000	58.5	24.49	29.7	3.31	8.65	1.04	643.5
Full throttle (82 degrees)							
2000	58.6	12.28	27.9	1.77	7.62	0.92	686.0
3000	64.4	20.23	28.2	2.87	7.57	0.91	678.0
4000	67.3	28.17	28.3	3.99	7.77	0.93	676.0

tested in each case was that with same throttle valve opening and stoichiometric air-fuel ratio, concerned to find a spark angle which lead to the maximum torque possible. In some exceptional situations it was possible to influence on emissions adjusting this parameter.

On tables C.4 and C.5 each regime is defined by the following parameters: crankshaft speed, percentual throttle opening, normalized air-fuel ratio calculated by gas analyzer and spark ignition angle measured by a stroboscopic pistol.

Several throttle positions were tested, aiming to allow greater air admission rates, looking for the objective of operate the engine in a more efficient way and controlling some emissions.

It was not an usual procedure to change gate valve opening (which doses gaseous fuel). In some few situations it was really needed to change its passage area between operating regimes which would be compared to the same originally fuel supplied regime.

It was noted for the same gate valve opening, higher air consumption regimes led to higher ethanol vapor consumption rates, what is reasonable taking into account the Venturi suction feature which is proportional to the air flow rate. However, air-fuel ratio gone leaner as more air was admitted, as expected. Fuel consumption increase was relatively low.

4.3 Discussion

It was observed through the steam generator's glass windows fuel acquired an amber shade, despite the color absence on the hydrated ethanol which was supplied. Sample analyses through

Table 4.4: Mechanical and hydraulical quantities for operated regimes with vaporized ethanol injection apparatus.

ω (rpm)	$ \phi $ (%)	λ	Ignition °BTDC	τ (Nm)	P (kW)	η (%)	Pressure		Consump. ethanol (g/s)	A/F ratio calculated		H ₂ O Flow estimated (g/s)	Carbon balance (g/s)		
							MAP (mmHg)	Boiler (kPa)		Inlet			Outlet	diff. (%)	
										Λ	λ				
2000	24.25	1.00	21.4	36.77	7.70	28.2	493.6	151.6	1.10	8.16	0.98	296.9	0.53	0.48	8.89
2000	26.25	1.01	19.8	37.66	7.89	30.4	503.6	166.3	1.04	8.59	1.03	362.5	0.51	0.49	3.77
2000	41.25	1.19	17.4	49.86	10.44	34.3	626.7	166.2	1.22	10.08	1.21	270.5	0.59	0.57	3.52
2000	57.75	1.30	18.9	46.99	9.84	30.6	634.7	161.3	1.29	9.99	1.20	314.6	0.63	0.56	10.32
2000	35.50	1.36	19.5	40.17	8.41	29.9	624.1	156.3	1.13	10.65	1.28	306.1	0.55	0.51	7.74
2000	40.50	1.48	21.7	38.02	7.96	29.3	655.1	151.4	1.09	12.00	1.44	221.3	0.53	0.51	4.08
2000	53.00	1.01	13.9	58.29	12.21	30.7	673.6	151.3	1.60	7.91	0.95	324.5	0.78	0.67	13.32
2000	100.00	1.01	13.4	57.93	12.13	30.9	683.6	151.3	1.57	8.10	0.97	348.9	0.76	0.68	10.40
3000	24.50	1.01	26.9	22.24	6.99	23.5	355.4	161.1	1.19	8.16	0.98	484.2	0.58	0.52	9.31
3000	30.75	1.23	29.1	24.57	7.72	25.9	412.4	161.1	1.20	9.89	1.19	485.4	0.58	0.53	8.56
3000	31.75	1.31	24.7	25.11	7.89	26.0	438.4	164.1	1.22	10.50	1.26	371.6	0.59	0.54	8.93
3000	32.25	1.39	24.1	26.18	8.23	26.7	463.4	166.0	1.24	10.96	1.32	376.7	0.60	0.55	7.74
3000	53.00	1.01	15.6	60.80	19.10	32.8	661.8	151.8	2.34	8.77	1.05	517.4	1.13	1.11	2.13
3000	100.00	1.02	14.9	60.98	19.16	33.1	676.8	151.8	2.33	8.94	1.07	436.4	1.13	1.10	2.19
3000	100.00	1.22	14.5	49.68	15.61	30.3	676.8	141.9	2.07	10.36	1.24	504.8	1.01	0.97	3.36
3000	100.00	1.41	17.1	39.63	12.45	27.8	676.8	134.1	1.80	11.61	1.39	548.3	0.87	0.84	3.44
4000	24.50	1.00	22.3	11.84	4.96	16.8	256.0	151.7	1.19	8.77	1.05	722.1	0.58	0.56	2.13
4000	29.25	1.14	22.2	13.09	5.48	18.6	296.0	154.6	1.18	10.24	1.23	412.1	0.57	0.59	-3.01
4000	33.75	2.00	31.4	8.97	3.76	11.8	396.8	159.6	1.27	14.28	1.71	517.6	0.62	0.50	19.61
4000	32.25	1.22	19.2	12.55	5.26	14.5	346.8	156.7	1.46	9.91	1.19	506.8	0.71	0.63	10.23
2000	54.00	1.06	17.3	53.62	11.23	30.6	675.4	151.6	1.47	8.58	1.03	326.2	0.71	0.65	9.37
4000	52.00	1.37	25.8	47.17	19.76	30.7	646.0	156.6	2.59	11.24	1.35	630.2	1.25	1.18	5.82
4000	99.25	1.51	21.3	39.81	16.68	27.0	666.0	151.7	2.48	12.47	1.50	672.6	1.21	1.18	2.36
4000	53.00	1.22	19.1	52.01	21.79	30.4	646.0	166.4	2.88	10.02	1.20	702.0	1.40	1.29	7.75
4000	47.50	1.07	15.7	50.75	21.26	29.6	626.0	171.3	2.88	9.58	1.15	638.9	1.40	1.36	2.48
4000	42.50	1.01	15.0	49.32	20.66	28.5	591.0	181.1	2.91	8.41	1.01	417.2	1.41	1.33	5.63

Continued

Conclusion

ω (rpm)	$ \phi $ (%)	λ	Ignition °BTDC	τ (Nm)	P (kW)	η (%)	Pressure		Consump. ethanol (g/s)	A/F ratio calculated		H ₂ O Flow estimated (g/s)	Carbon balance (g/s)		
							MAP (mmHg)	Boiler (kPa)					Inlet	Outlet	diff. (%)
2000	60.75	1.11	8.5	45.91	9.62	28.6	675.8	146.7	1.35	9.76	1.17	366.2	0.66	0.67	-2.44
2000	100.00	1.11	11.0	46.81	9.80	28.9	685.8	146.7	1.36	9.80	1.18	368.9	0.66	0.67	-0.69
3000	55.75	1.31	23.0	49.68	15.61	31.3	665.8	141.8	2.00	10.81	1.30	488.0	0.97	0.93	4.31
3000	100.00	1.24	20.0	52.19	16.40	32.9	675.8	146.7	2.00	10.78	1.29	488.4	0.97	0.98	-1.22
4000	67.00	1.47	30.1	43.40	18.18	30.7	665.8	151.6	2.38	12.86	1.54	526.3	1.15	1.17	-1.40
2000	100.00	0.86	19.6	55.60	11.64	24.6	685.8	136.9	1.90	6.74	0.81	154.2	0.92	0.79	14.14
3000	100.00	0.95	15.8	62.05	19.49	31.7	675.4	158.4	2.47	8.39	1.01	501.5	1.20	1.19	1.03
3000*	100.00	1.00	17.8	65.46	20.56	32.3	685.4	166.3	2.56	8.41	1.01	2079.9	1.24	1.18	5.09
2000*	100.00	0.99	25.8	55.60	11.64	30.9	685.0	92.7	1.51	8.89	1.07	-	0.73	0.72	1.49

* Original regimes run with original Fueltech's ECU, for comparison effects.

$|\phi|$ Throttle valve opening.

Table 4.5: Dry base emissions related to the mechanical energy supplied, in gaseous ethanol engine run.

ω (rpm)	$ \phi $ (%)	λ	Ignition °BTDC	Dry base combustion products							Specific Production (g/kWh)					Consump.
				Flow (mol/h)	CO (g/h)	THC (g/h)	CO ₂ (g/h)	O ₂ (g/h)	NO _x (g/h)	N ₂ (g/h)	CO	THC	CO ₂	O ₂	NO _x	Ethanol
2000	24.25	1.00	21.4	975	158.3	5.6	6134.2	190.3	56.4	24712.4	20.56	0.73	796.64	24.71	7.32	512.64
2000	26.25	1.01	19.8	991	249.7	3.2	6016.6	256.8	58.6	24684.8	31.66	0.40	762.77	32.56	7.42	475.22
2000	41.25	1.19	17.4	1364	19.1	4.0	7503.9	1646.0	80.9	33989.9	1.83	0.38	718.63	157.63	7.75	420.95
2000	57.75	1.30	18.9	1455	24.5	4.5	7364.3	2207.5	86.6	35619.0	2.48	0.45	748.32	224.32	8.80	472.26
2000	35.50	1.36	19.5	1346	22.6	4.3	6632.7	2433.4	13.9	33244.4	2.69	0.51	788.32	289.22	1.66	483.69
2000	40.50	1.48	21.7	1508	29.6	4.9	6635.3	3339.4	6.0	36116.0	3.71	0.61	833.26	419.36	0.75	492.69
2000	53.00	1.01	13.9	1352	140.1	7.7	8626.9	337.5	79.6	34873.5	11.48	0.63	706.69	27.65	6.52	471.09
2000	100.00	1.01	13.4	1379	173.8	9.4	8738.8	308.9	81.4	35204.9	14.32	0.77	720.29	25.46	6.71	467.29
3000	24.50	1.01	26.9	1050	158.7	8.2	6651.2	235.1	62.0	26866.1	22.72	1.17	952.03	33.66	8.87	614.17
3000	30.75	1.23	29.1	1299	36.4	9.9	6914.1	1828.5	76.7	32637.0	4.71	1.29	895.75	236.89	9.94	557.35
3000	31.75	1.31	24.7	1455	32.6	8.7	7040.3	2560.1	32.7	35370.5	4.13	1.10	892.55	324.56	4.15	556.72
3000	32.25	1.39	24.1	1565	30.7	10.1	7230.3	3004.8	4.7	37413.8	3.73	1.22	878.97	365.28	0.57	541.10
3000	53.00	1.01	15.6	2247	169.9	12.7	14337.6	481.8	135.1	56576.3	8.90	0.66	750.66	25.23	7.07	440.17
3000	100.00	1.02	14.9	2263	133.1	12.7	14339.0	782.1	136.9	57432.2	6.95	0.66	748.53	40.83	7.15	437.45
3000	100.00	1.22	14.5	2386	60.1	8.8	12701.9	3397.4	6.1	59227.0	3.85	0.56	813.88	217.69	0.39	477.78
3000	100.00	1.41	17.1	2362	52.9	11.6	11016.4	4860.1	1.8	57680.7	4.25	0.93	884.74	390.32	0.15	520.38
4000	24.50	1.00	22.3	1130	126.6	6.5	7211.8	264.1	65.9	28688.5	25.53	1.31	1454.55	53.26	13.29	860.50
4000	29.25	1.14	22.2	1345	60.3	6.1	7696.1	1334.7	30.3	33465.3	10.99	1.11	1403.40	243.39	5.52	776.88
4000	33.75	2.00	31.4	2098	111.6	39.6	6276.7	7720.0	0.1	50253.2	29.71	10.54	1671.05	2055.30	0.02	1221.25
4000	32.25	1.22	19.2	1582	48.7	28.5	8214.3	2531.4	3.3	39843.6	9.27	5.42	1562.08	481.38	0.63	996.50
2000	54.00	1.06	17.3	1378	23.1	8.5	8486.5	701.0	82.5	34873.5	2.06	0.75	755.65	62.41	7.35	471.95
4000	52.00	1.37	25.8	3268	100.6	20.2	15383.9	6399.3	193.7	80211.8	5.09	1.02	778.65	323.90	9.80	471.14
4000	99.25	1.51	21.3	3517	98.5	21.2	15320.8	8362.4	3.2	85540.9	5.91	1.27	918.67	501.43	0.19	536.17
4000	53.00	1.22	19.1	3240	81.6	25.2	16820.3	4551.1	193.1	79659.6	3.75	1.16	772.09	208.90	8.86	475.94
4000	47.50	1.07	15.7	3007	75.8	11.3	17860.7	2501.7	178.8	76235.8	3.56	0.53	840.12	117.67	8.41	488.28
4000	42.50	1.01	15.0	2770	85.3	10.4	17425.9	443.1	15.8	67596.1	4.13	0.50	843.51	21.45	0.76	507.06

Continued

Conclusion

ω (rpm)	$ \phi $ (%)	λ	Ignition °BTDC	Dry base combustion products							Specific Production (g/kWh)					Consump.
				Flow (mol/h)	CO (g/h)	THC (g/h)	CO ₂ (g/h)	O ₂ (g/h)	NO _x (g/h)	N ₂ (g/h)	CO	THC	CO ₂	O ₂	NO _x	Ethanol
2000	60.75	1.11	8.5	1502	29.4	12.0	8792.1	956.7	86.7	36419.8	3.06	1.24	914.35	99.50	9.02	506.20
2000	100.00	1.11	11.0	1486	33.3	12.9	8698.1	1170.1	87.8	36861.6	3.40	1.31	887.25	119.35	8.96	500.29
3000	55.75	1.31	23.0	2397	53.7	21.9	12127.7	4057.3	141.9	59779.2	3.44	1.41	777.08	259.97	9.09	461.89
3000	100.00	1.24	20.0	2438	47.8	15.3	12874.7	3277.2	144.1	59641.2	2.91	0.93	785.26	199.88	8.79	440.03
4000	67.00	1.47	30.1	3475	97.3	50.3	15135.9	8205.9	51.3	84353.6	5.35	2.76	832.58	451.38	2.82	470.36
2000	100.00	0.86	19.6	1311	2562.9	2.6	6404.7	79.7	11.8	35315.3	220.11	0.23	550.05	6.85	1.01	586.89
3000	100.00	0.95	15.8	2267	1301.3	14.4	13566.5	188.6	133.3	57183.7	66.75	0.74	695.92	9.68	6.84	455.97
3000*	100.00	1.00	17.8	2360	376.6	7.2	14950.8	339.8	137.8	59447.9	18.31	0.35	727.01	16.52	6.70	448.17
2000*	100.00	0.99	25.8	1480	66.3	47.2	9311.7	440.4	88.3	37137.7	5.69	4.05	799.70	37.82	7.58	467.73

★ Original regimes run with original Fueltech's ECU, for comparison effects.

atomic absorption spectrometry had shown presence of 1ppm of copper and 20 ppb of iron, probably come from contact with these metals into the boiler.

There is a transparent plastic cord-reinforced hose between jackwted tube and Venturi, having nearly 20mm available for observation of fuel flow. During dynamometric runs it was not noted any presence of liquid in this place, this indicates the engine was supplied just with saturated or superheated ethanol vapor. Fuel level inside the boiler was kept in a safe range, producing bubbles nearly 25mm high above liquid fuel level. Once this level was nearly 40mm below the top surface of the boiling chamber, there was no risk of liquid fuel transport to the intake manifold.

It was impossible to use the pressure-enthalphy diagram for ethanol (figure 3.2) to evaluate its state because its pressure and temperatures were measured in different places (inside the boiler and after gete valve, respectively).

Some running regimes presented coarse errors on their calculated inlet-outlet carbon balances, probably due to measurement instrument misprecisions and piston ring leakages.

Appendix D has building details of steam generator and other accesories used on this study.

4.3.1 Idle engine

It was pursuit a stable idle condition, through the regime characterized by 920rpm, 0.00% throttle opening, $\lambda = 1.00$ and 6.5 degrees BTDC sparking advance; on vaporized ethanol. From here the engine running regimes will be denoted by these four parameters, put in this order and into parenthesis, in example (920/0.00/1.00/6.5) in order to refer to the idle regime of this paragraph. It was impossible, however, keep the engine in this condition for a long, due to huge oscillations in boiler inlet water temperature, whose stability is fundamental for keeping the engine in a steady-state condition. There is just an emissions record, where it is possible to observe a very higher THC emissions with vaporized fuel (1168ppm against 296ppm) and lower CO levels (0.16% against 0.42% with the original fuel injection).

4.3.2 Intermediate running regimes

For the vaporized ethanol operating regime of 2000rpm and thorottle at 23°(24.25%) the originally-fueled torque of 37.7Nm was pursuit. The studied regimes on vaporized ethanol run for comparison to the original one, from (2000/26.25/1.01/19.8) to (2000/40.50/1.48/21.7), had given better torques and efficiencies, while the first gaseous ethanol fueled regime tested (2000/24.25/1.00/21.4) supplied the same torque with pratically the same efficiency. The operating regime (2000/41.25/1.19/17.4) was those run with gaseous ethanol which presented the higher mechanical efficiency (34.3%). An efficiency raise on the range $1.1 \leq \lambda \leq 1.2$ is ex-

pected for any fuel injection system and comes with higher nitrogen oxide emissions. In this case it was possible a more intense reduction in NO_x amounts with values for λ from 1.35. Any way it was impossible to reduce THC emissions in relation to original fuel injectino system. If it be possible, leading to a more efficient flammable mixture burning, efficiency could be still better.

As regime (2000/41.25/1.19/17.4) offers a too higher torque than those used as base of comparison with the original ECU, (49.8Nm against 37.7Nm), it was made its comparison with other regime also tested with original ECU (2000/34.6/0.97/17.6), which supplies nearly the same torque (50.04Nm). Vaporized ethanol supplied regime was more efficient (34.3% against 27.1%) and has achieved good reduction on CO and THC amounts¹, letting just NO_x 's ones on the same level.

For comparison to the regime of 2000rpm and throttle at 45° (52.50%) the tests (2000/53.00/1.01/13.9), (2000/54.00/1.06/17.3), (2000/60.75/1.11/8.5) and (2000/100.00/1.11/11.0) were performed. On the first gaseous ethanol fueled regime of this set, there was an efficiency drop of neraly 4.3% and torque increase of 1.5%. Lowering of CO and THC emissions were too higher than efficiency loss, on the other hand NO_x emissions were higher. On the other regimes of this set the same tendency is observed. One can note on the third and fourth studied regimes of this row (ranging from 60.75% to 100.00% of throttle, respectively) there is a little on difference on both torque, efficiency and specific emissions; once air inlet rate is nearly the same at the end of throttle angle displacement.

For the regime of 3000rpm and throttle at 23° (24.25%) the tests from (3000/24.50/1.01/26.9) to (3000/32.25/1.39/24.1) were performed. The second one of this row (3000/30.75/1.23/29.1) had exactly the same torque of the reference regime, fueled with liquid ethanol. On it, an efficiency increase of 4.9% was observed, having a cost of increases on NO_x and THC emissions. The next regime (3000/31.75/1.31/24.7) causes a slight reduction on THC amounts and further reductions on CO and NO_x quantities, with a raise in torque of 2.2% and of 5.3% in efficiency compared to the originally fueled regime. The fourth regime, with an increase on porcentual throttle opening of 1.50% (from 30.75% to 32.25%), lead to higher values of torque and efficiency than the other ones, but emitting higher quantities of THC by amount of mechanical work delivered to the crankshaft (1.22g/kWh against 1.11g/kWh). With wider throttle positions the engine became unstable, but without increase in its fuel consumption. Due to the increase on THC emissions on this air excess range it can be presumed the flame inside combustion chamber becomes unstable for this regime as air-fuel ratio gets close to 1.4.

¹The word "amount" is out to let clear the difference to chemical meaning of "reduction" as the donation of electrons to an atom, radical or substance.

Other comparison is those made among the regimes (3000/32.25/1.39/24.1)(vaporized) and (3000/28.20/0.97/17.6)(Original ECU), which has the same torque-speed couple. With a wider throttle opening and 39 % of air excess there was an efficiency increase and reduction on specific emissions of 76.4% for CO, 80.9% for THC and 94.1% for NO_x.

For the regime characterized by 3000 rpm and throttle at 45°(52.50%) the regime fueled with vaporized ethanol (3000/53.00/1.01/15.6) presented the same torque (60.8Nm) with an increase of 3.5% in efficiency. Specific emissions of NO_x raised 15.7% (less than on other stoichiometric vaporized fuel supply regimes) while those ones derived from partially burnt fuels were reduced. On the next regime studied (3000/100.00/1.22/14.5) there was reduction of nearly 68% on CO and THC specific emissions and 93.6% for NO_x. Torque was reduced on 18.3% keeping the original efficiency. The third regime evaluated on this row (3000/100.00/1.41/17.1) presented a slight increase on THC and CO in relation to the former one, indicating deficiencies on flame propagation. There was also losses in efficiency (-7.9%) and torque (-34.8%).

The last regime of this series (3000/55.75/1.31/23.0) presented lower efficiency drops (-1.3%), the same torque decrease of the regime (3000/100.00/1.22/14.5) and higher NO_x emissions. Because the so different torque in relation to those used as basis fueled through the original ECU, both ones having 49.7Nm at 3000rpm can be compared to the regime (3000/35.9/0.97/15.9) run with the original ECU. The operating regime (3000/100.00/1.22/14.5) kept the same efficiency level with a strong decrease of specific emissions, while the regime (3000/55.75/1.31/23.0) conquered an increase of nearly 1.0% in efficiency bringing on the other hand increases of NO_x emissions 18.2% above that observed for this torque-speed couple for the original fuel injection system.

For the regime of 4000 and throttle at 23°(24.25%) the tests (4000/24.50/1.00/22.3) through (4000/32.25/1.22/19.2) were performed. The first one supplied an increase of 10.0% in torque and of 6.7% in efficiency, bringing also an increase of 38.7% on NO_x specific emissions. One can note even running with stoichiometric mixture higher efficiencies were achieved, due mainly to the more complete fuel burn and to the energy released by nitrogen oxidation. The second regime studied, (4000/29.25/1.14/22.2), obtained expressive gains in both torque and efficiency and decrease on the three pollutant emissions which are monitored. The last regime of this series (4000/32.25/1.22/19.2) had similar efficiency if compared to those fueled with the original ECU and higher THC specific emission, indicating the inflexion of the curve which related this emission to the air-fuel ratio can happen at different values for λ in each regime, once on the other ones THE emissions raised again with an air excess on the range of 40%. The former regime (4000/33.75/2.00/31.4) shows it is possible to run the PVEE up to the double amount of stoichiometric air, even at expense of fails on flame propagation which are disclosed by the

combination of low efficiency and high THC tailpipe concentrations.

For the regime of 4000 rpm and throttle at 45° (52.50%) torque was lower (on the 10% – 20% range) in all regimes evaluated, fueled with gaseous ethanol, from (4000/52.00/1.37/25.8) to (4000/42.50/1.01/15.0). Efficiency, however, was higher in all of them (up to 8.5%). In all these regimes there was decrease on CO specific emission above 73%, while THC specific emission had a drop higher than 40%. Specific emission of NO_x falls just on the regime (4000/67.00/1.47/30.1), keeping high on the others where air excess is lower. To get close of stoichiometric conditions it was needed to close the throttle valve, because the steam generator was unable to feed the engine at ratios higher than 3.0g/s of vaporized ethanol. There were performed just the tests (4000/47.50/1.07/15.0) and (4000/42.5/1.01/15.0), included on previous analysis. The last cited regime has efficiency of 28.5%, the same range of the originally fueled regime used for comparison, but with too lower specific emissions.

Regime (4000/53.00/1.22/19.1) must be compared to the regime (4000/41.1/1.00/1.27), corresponding to the same torque-speed couple on the original ECU. The regime fed with vaporized ethanol presented higher efficiency (30.4% against 29.3%) and lower specific emissions for CO and THC. Specific emission for NO_x raised 13.9%.

Ordenating operating regimes in decrescent order of efficiency, it can be noted the regimes which take better use of chemical energy contained on fuel are those of speed on 3000rpm with $|\phi|$ above 50%, occurring with values for λ in the range from 0.95 (it is unusual a rich misture operation lead to good efficiency) to 1.31. Maximum efficiency was achieved by a regime which does not obey this pattern, being an isolated case.

Comparison between regimes operated with either liquid and vaporized ethanol in similar torque-speed couples offers better efficiencies with vaporized fuel, in a general way; specially when operated at woder throttle positions. In the great part of situations the efficiency increase is due to flame stability with leaner mistures which lead to lower head losses through throttle valve. Better efficiencies were also achieved on some stoichiometric regimes operated with vaporized ethanol, it means, these tests had departed from higher efficiencies at stoichiometric with vaporized fuel before pursuing even higher efficiencies with wider throttle openings; what indicates part of merit is due to the more efficient burning of pre-vaporized and premixed fuel to the air.

In general lines efficiency raises lead to higher NO_x specific emissions. To be sure they assume low values it is needed they overcome 35% of air excess. Regimes with emissons on the range from 1900ppm to 2050ppm were achieved (the same range observed on liquid fuels) in other air excess ranges, but not in a regular form, without a stablised pattern. There are still regimes with drastic decreases of NO_x at expense of efficiency loss. Spark ignition advance adjust had shown influence on efficiency (higher for lower spark advance angles) and on NO_x emissions

(lower with delayed sparking and lower peak flame temperatures). This was noted during the adjusts made searching for the better torque regimes, it means, they were not performed tests with several spark angles with the other parameters frozen in order to evaluate this effect isolately.

Few regimes had achieved both better efficiency and decrease in the three emissions evaluated. In some of them there is efficiency reduction and a further recduction on emisions, ot at least a strong drop in NO_x ones, which is the most difficult to control and the one which had presented themore unregular behavior. The absence of the cooling effect caused by vaporization of fuel bubbles inside the chamber is one of the causes for that. There is no relationship to the air excess, exhaust manifold measured temperature and nor to the time lapse between spark and Top Dead Center.

Few regimes got a fall on specific emissions of NO_x at $\lambda < 1.35$. They are: 3000/100.00/1.22/14.5; 4000/32.25/1.22/19.2; 4000/29.25/1.14/22.2 and 4000/42.50/1.01/15.0. These are regimes of several values for efficiency, air consumption, as well as of high exhaust temperatures. Having air excess below 35%, other regimes run at 4000rpm had presented NO_x emissions on the 2000ppm range.

A well-known favorable factor to this emission drop is sparking delay. Couples of regimes with the similar crankshaft speed, air-fuel ratio and air consumption (3000/100.00/1.22/14.5 and 3000/100.00/1.24/20.0; 4000/99.25/1.51/21.3 and 4000/67.00/1.47/30.1) presented lesser NO_x emissons when the spark came in a delayed angle. The counterpart is a slight fall on both

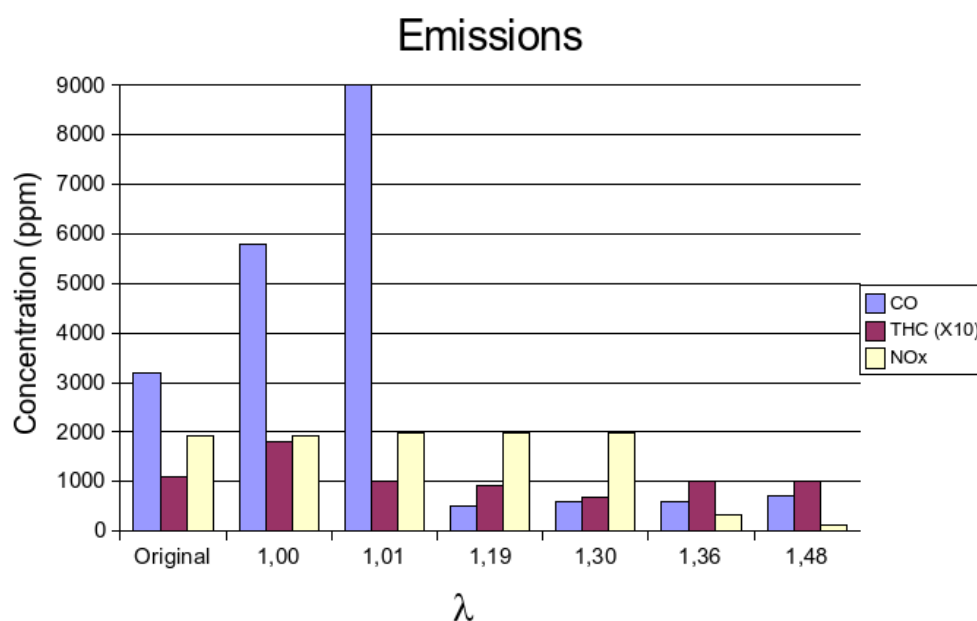


Figure 4.1: Emissions for the regimes compared to these of 2000rpm and throttle at 23%

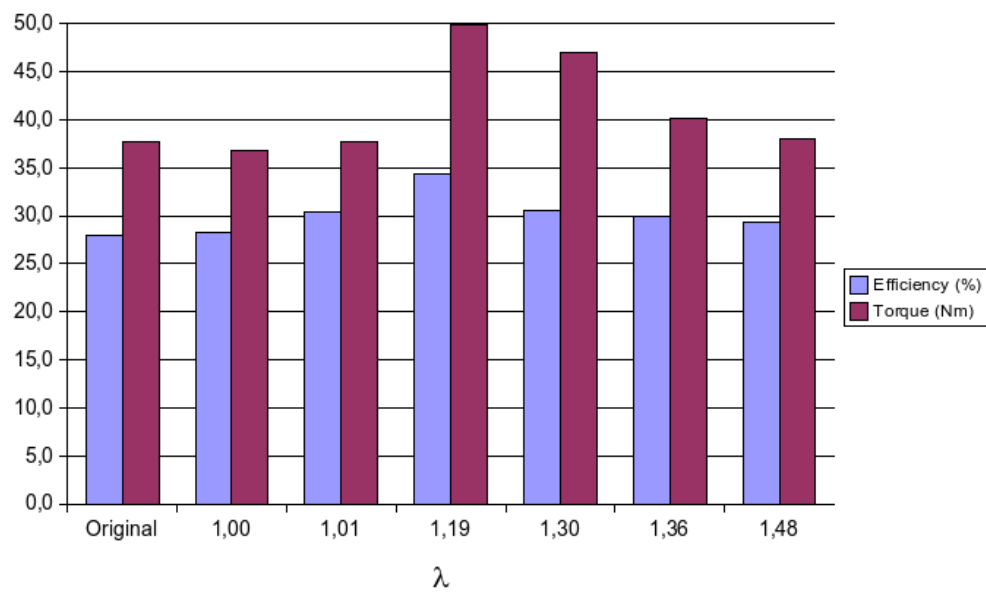


Figure 4.2: Torque and efficiency of regimes compared to these of 2000rpm and throttle at 23%

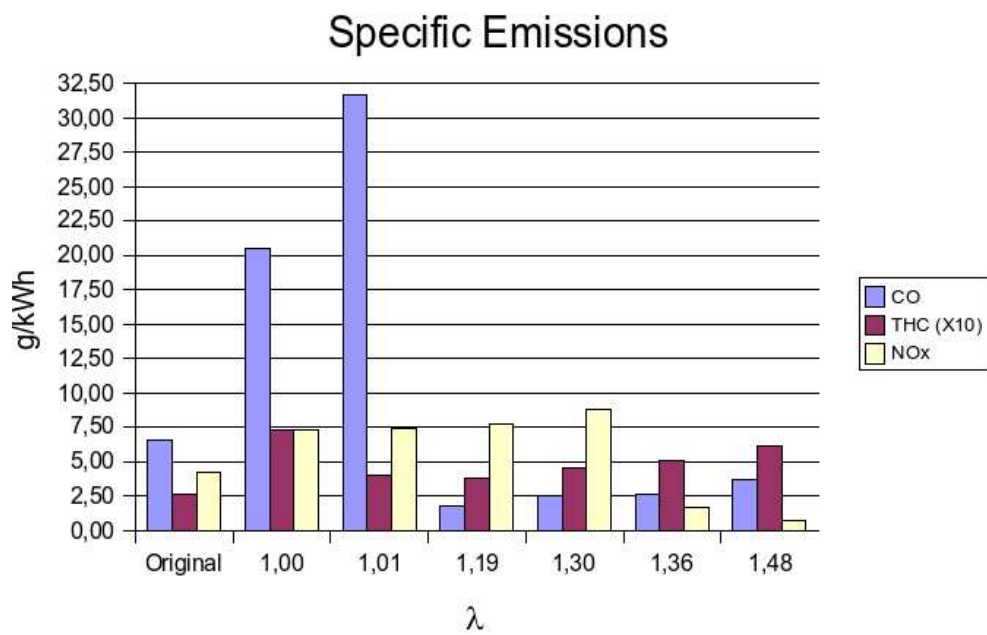


Figure 4.3: Specific emissions for regimes compares to these of 2000rpm and throttle at 23%

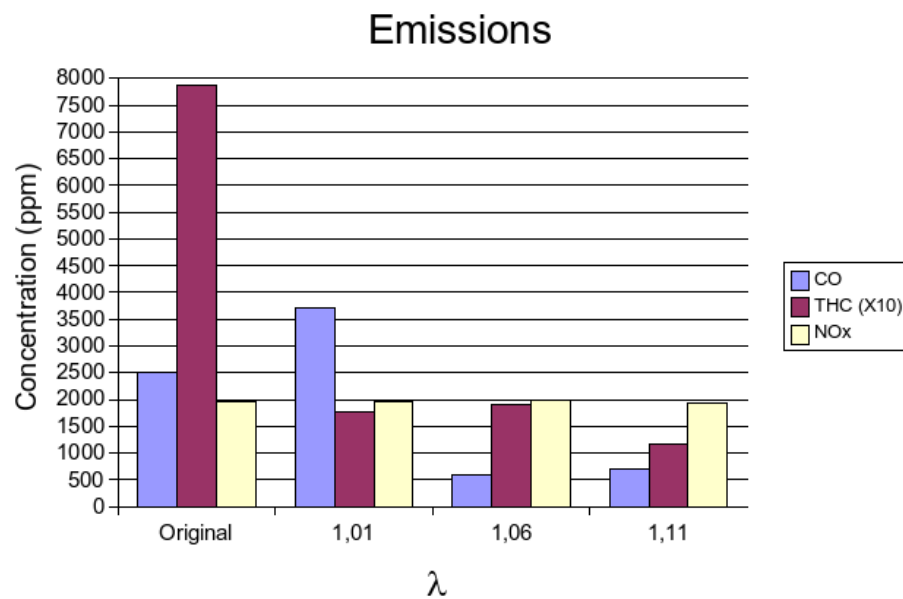


Figure 4.4: Emissions for regimes compared to these of 2000rpm and throttle at 45%

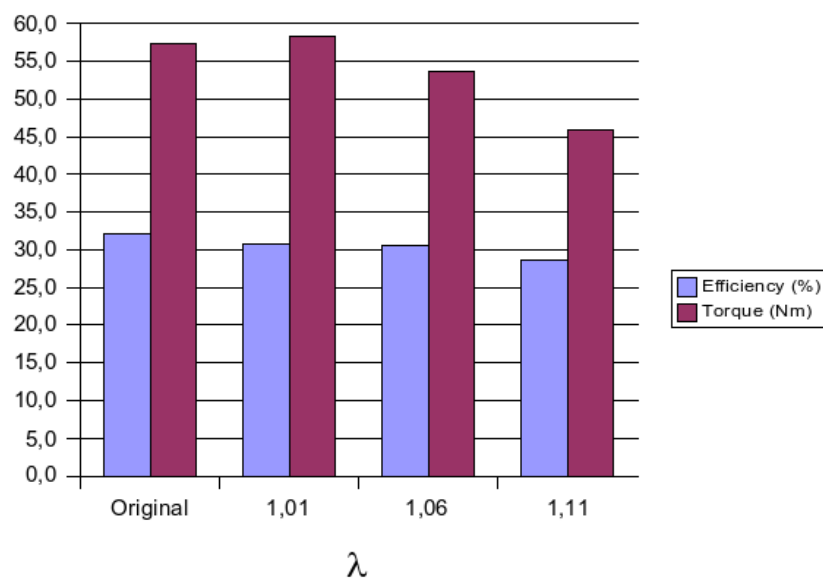


Figure 4.5: Torque and efficiency for the regimes compared to these of 2000rpm and throttle at 45%

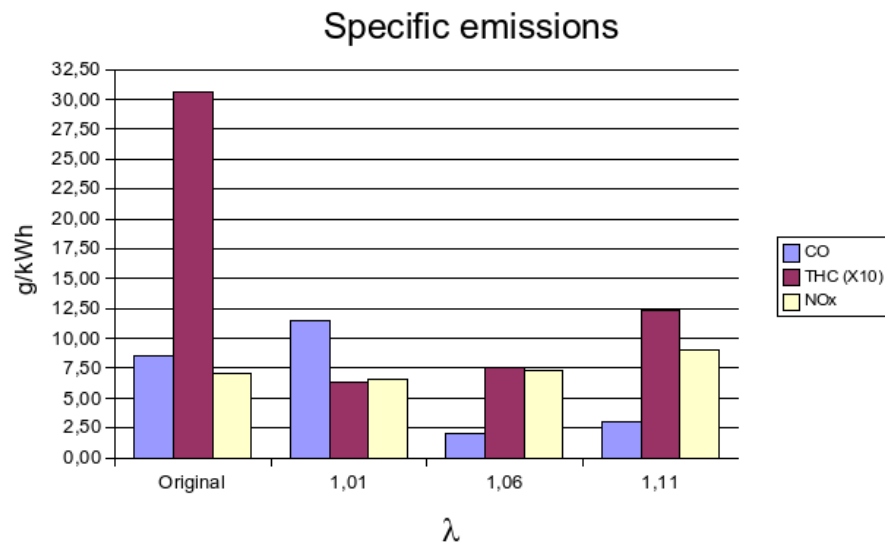


Figure 4.6: Specific emissions for the regimes compared to these of 2000rpm and throttle at 45%

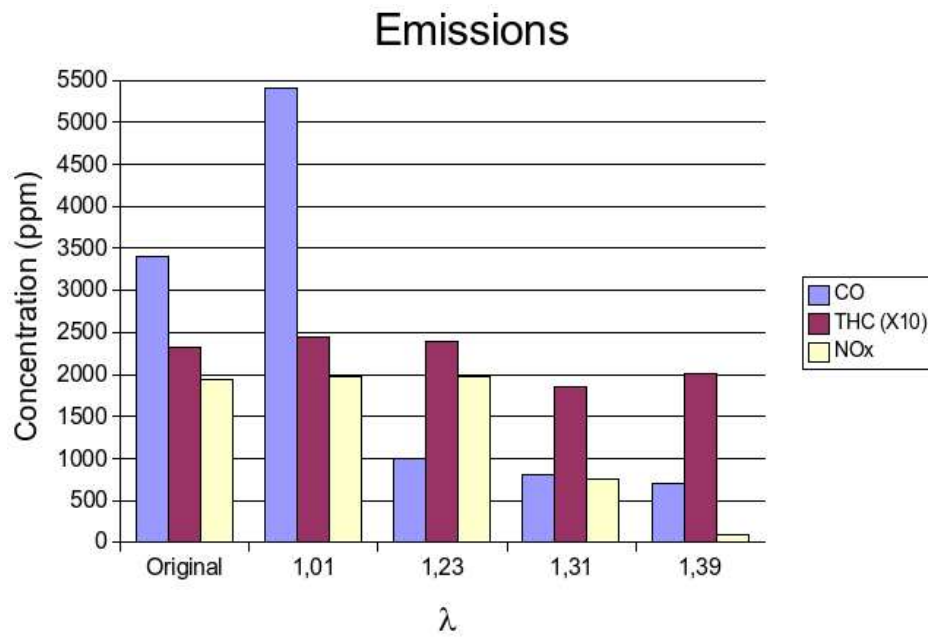


Figure 4.7: Emissions for regimes compared to these of 3000rpm and throttle at 23%

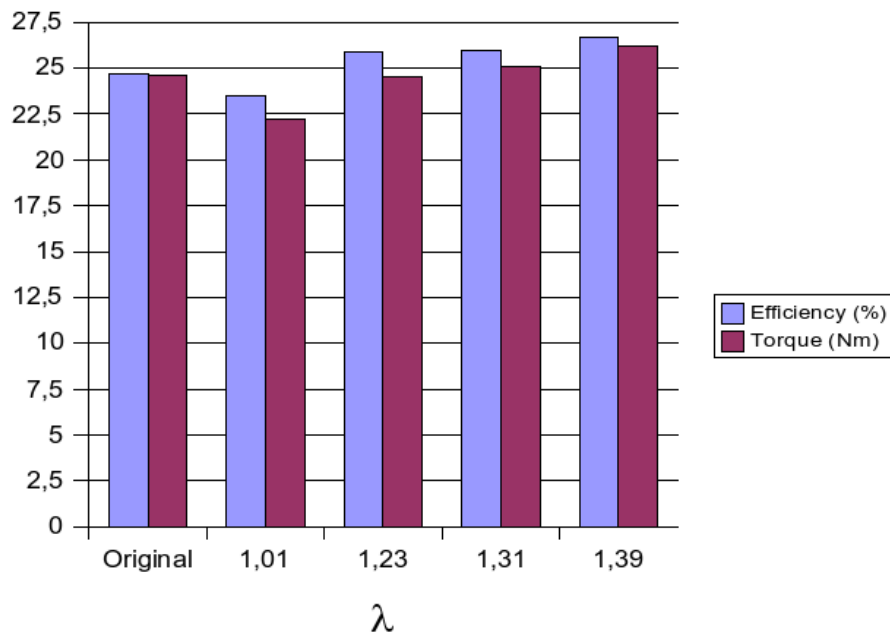


Figure 4.8: Torque and efficiency of regimes compared to these of 3000rpm and throttle at 23%

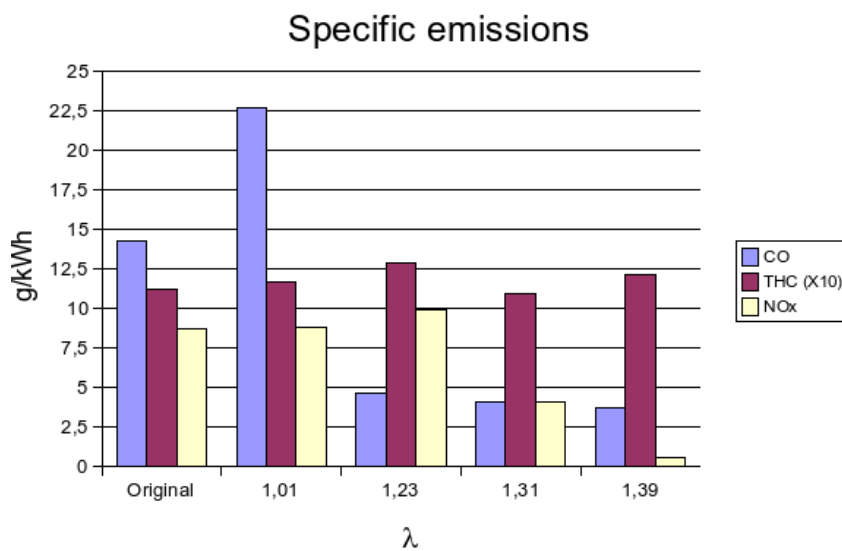


Figure 4.9: Specific emissions for the regimes compared to these of 3000rpm and throttle at 23%

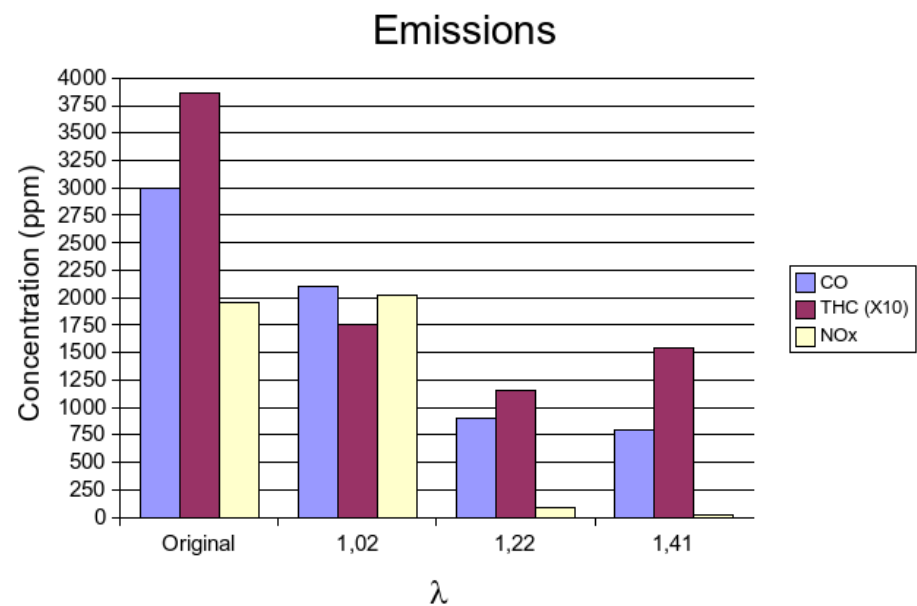


Figure 4.10: Emisissions for the regimes compared to these of 3000rpm and throttle at 45%

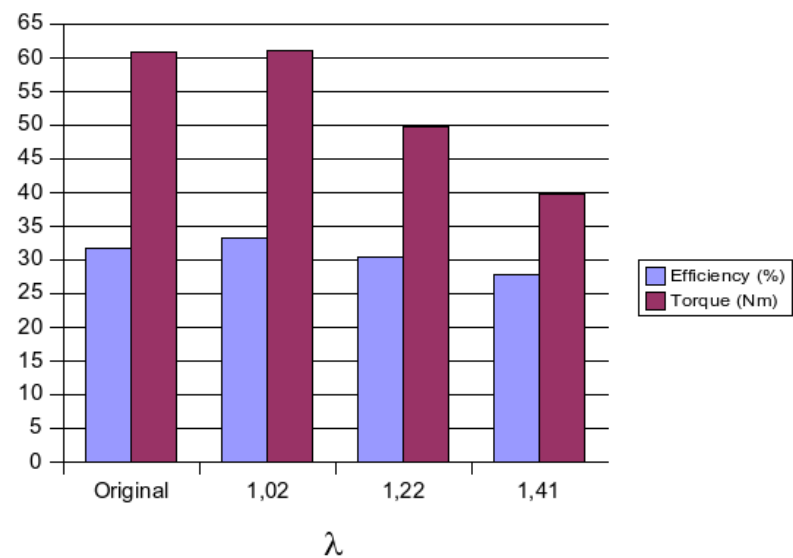


Figure 4.11: Torque and efficiency for regimes compared ot these of 3000rpm and throttle at 45%

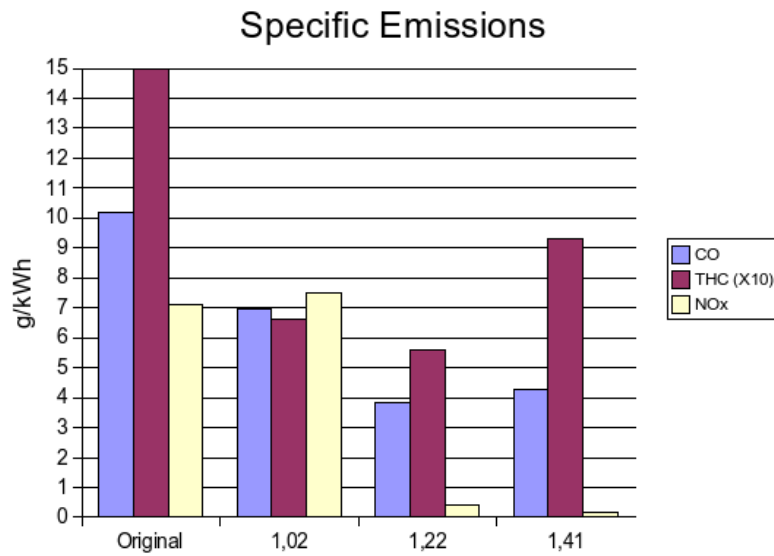


Figure 4.12: Specific emissions for the regimes compared to these of 3000rpm and throttle at 45%

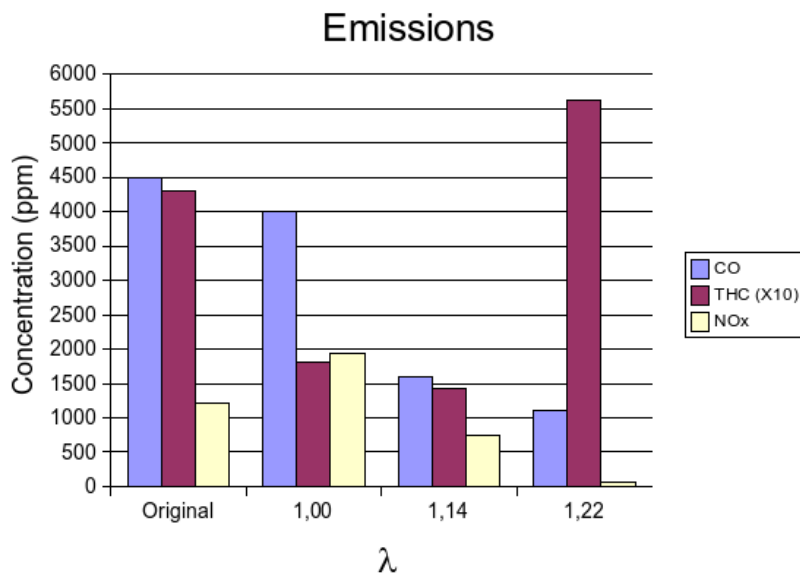


Figure 4.13: Emissions for the regimes compared to these of 4000rpm and throttle at 23%

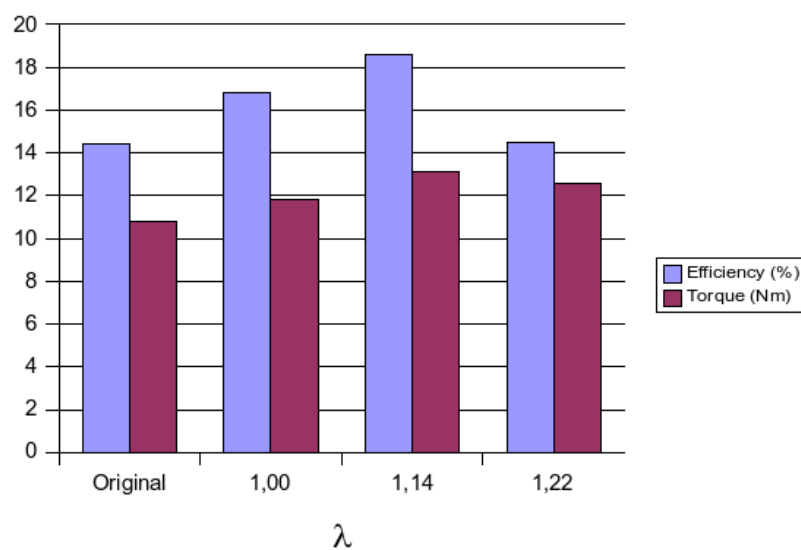


Figure 4.14: Torque and efficiency for the regimes compared to these of 4000rpm and throttle at 23%

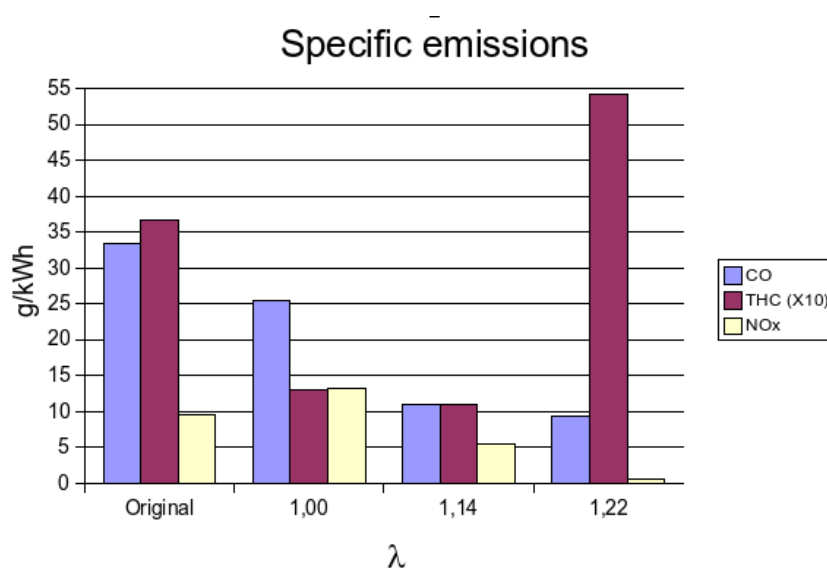


Figure 4.15: Specific emission for the regimes compared to these of 4000rpm and throttle 23%

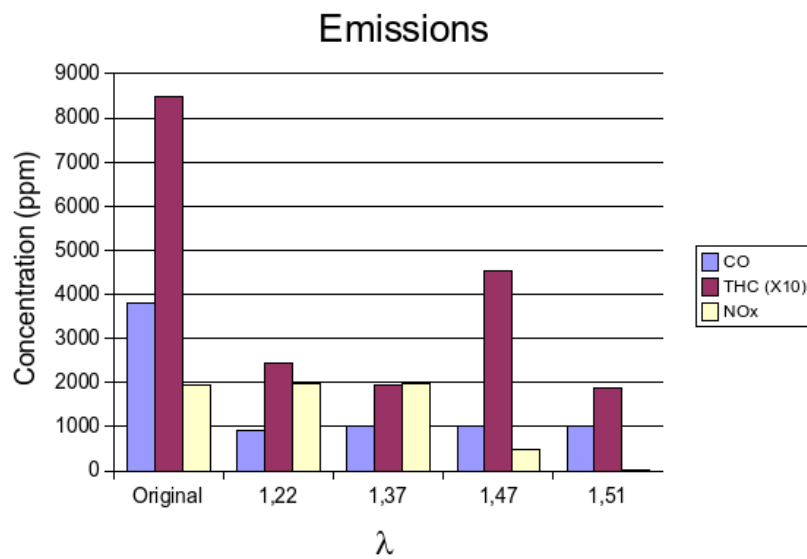


Figure 4.16: Emissions for the regimes compares to these of 4000rpm and throttle at 45%

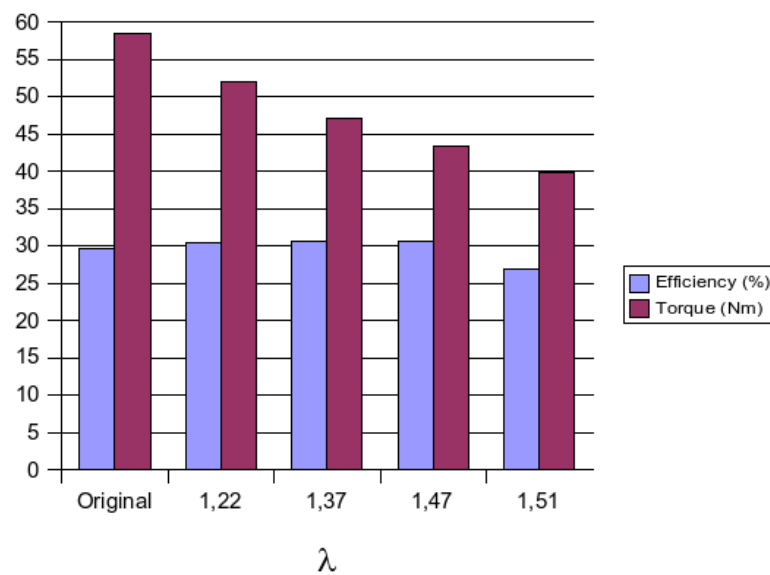


Figure 4.17: Torque and efficiency for the regimes compared to these of 4000rpm and throttle at 45%

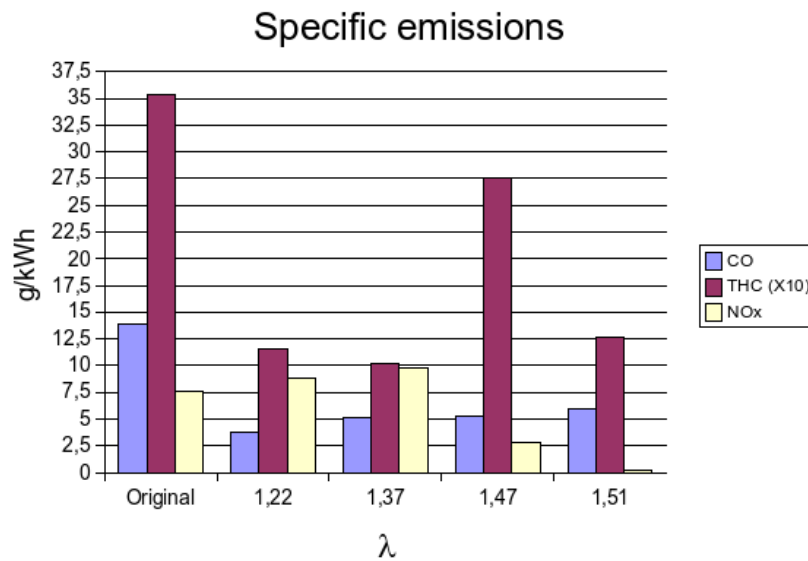


Figure 4.18: Specific emissions for the regimes compared to these of 4000rpm and throttle at 45%

efficiency and torque, as it is shown on table 4.4.

CO emission is the easiest to control, being below 0.11% in volume with just 5% of air excess. The emissions got under control below 300ppm (uncorrected values read by the gas analyzer), with few exceptions, with lean mixtures. These readings are lower than those observed for the engine running with liquid fuel and original ECU, except for throttle at 23°. Comparing regimes with liquid and vaporized fuel supply the emissions fall less easily. Specific emissions, however, arise in several cases, even on those with higher efficiency, which indicates further high efficiencies could be achieved looking for a more efficient burn of these unburnt hydrocarbons. When CO emission is low in link to high THC emission there is no advantage, because the low CO index is result of poor condition for oxidation of the fuel as a whole.

It can be supposed there is flame extinction in some places of combustion chamber in these cases. According to this hypothesis, the flame burns fuel with a good efficiency (low CO) where it passes through, and the THC comes from the places not reacted by it. The regime with $\lambda = 2.00$ presents a natural tendency of THC concentration rise with too poor mixtures and misfires, at same way some test series at the same speed presented an increase on this emission from some air excess point where it reaches its minimum value.

4.3.3 Max power operating regimes

As the original ECU is programmed to enrich air-fuel mixture in order to give maximum power from some position of throttle angular displacement, other tests were realized with full/wide open throttle and stoichiometric air-fuel mixture, as well tests on vaporized ethanol with rich

mixtures at WOT. This way more fair comparison between each kind of regimes could be made, avoiding unfair comparisons among distant air-fuel ratio regimes, which would lead to great nonsense related to efficiency and pollutant specific emissions.

Table 4.6 brings these data face to face, for 2000rpm of crankshaft speed. Even being impossible to stabilize engine operation with vaporized fuel exactly at the same ECU-programmed air-fuel ratio ($\lambda=0.92$), one can note the drop of supplied torque is low on front of economy offered when the engine operates with vaporized fuel and stoichiometric mixture. Drastic decreases also happened on specific emissions of CO and THC. Increase on NO_x emissions was expected due to the absence of the cooling effect caused by liquid ethanol inlet, on both vaporized fuel and stoichiometric liquid (less quantity of liquid). Air consumption also drops from 13.49g/s (D'Ávila's tests [23]) to 12.75g/s when vaporized fuel takes place, due to the space taken by it on intake ducts.

The same comparison was realized at 3000rpm and it is on table 4.7. It was also observed a slight decrease on air consumption due the space occupied by ethanol vapor. However, on this regime it was possible to decrease just CO emissions when comparing just stoichiometric regimes; while on rich regimes vaporized ethanol has just elevated emissions for NO_x .

As throttle valve is already totally open it can be noted there is no efficiency increase when air-fuel ratio is increased, at least on studied data. Lean mixture regimes run on vaporized ethanol, however, present lower emissions.

The last regime to be evaluated, 4000rpm and WOT, could not be its equivalent defined with vaporized ethanol. One of the factors was the limited boiler supply ability, which is not due to a lack of heat exchange area but to high lead losses between its steam exit and its arrival to the gate valve which doses its flow rate. During these regimes, consuming ethanol vapor at 3.0g/s, pressure on the boiler had not dropped, what would indicate some deficiency on its steam production capacity at requested rate. This is an indicator of the influence of head loss at the boiler's exit which was not well estimated on its project phase.

This way it was not possible to operate the engine with less than 50% of air excess at 4000rpm and WOT. This regime does not have a spark ignition advance which allows its run without knocking and with some torque. When sparking angle was delayed the engine operated in a kind of idle at 4000rpm, where it was unable to supply any torque.

Trials were made in order to stabilize the water from engine into values higher than 96°C initially previewed, in order to reach higher steam pressures which would lead to higher fuel flow ratios with the existing head losses. Up to 98°C it could be made with relative calm, above this limit water temperature became unstable and started to oscillate in an unregular way, overcoming 100°C, established limit to operate safely.

4.4 Transient warm-up

An important parameter for a PVEE is the elapsed time for warming up the water of cooling system and other parts of fuel vaporization system, in order to minimize time interval needed to the liquid-vapor switch. Warm-up time for cooling water system were evaluated, both with or without engine's thermostat² installed. This valve remains closed on the first minutes of engine operation, forbidding water to circulate through the cooler, being restricted to the engine's block. When this water reaches some determined temperature it opens allowing this water to circulate through the cooler, and on PVEE prototype, through the steam generator.

Two warm-up tests were performed, at morning, in two consecutive days, aiming to start on each of them with the whole engine at room temperature. The first test was done with thermostat installed, and the other one with this part removed. In both cases boiler pressure was monitored, finishing the test when vessel pressure has achieved 0.5kgf/cm^2 , considered a satisfactory level for supplying ethanol vapor to the engine. Temperature acquisition system was set up to register water temperatures before and after passing through the boiler, as well as lubricating oil too, taking samples of them each 5 seconds.

In both cases engine warm-up was realized under some load. The valve who links the boiler to the gate valve was open since the start of each experiment, allowing passage of small steam flow rates while boiler temperature was arising. Fueltech's programmable ECU allows a fast adjust mode on its injection mapping which raises or lowers all its points in just one command, which allows change the amount of liquid fuel injected gradually and easily. When ECU's display had shown "Ajuste rápido: Todo mapa -99%" (Quick adjust - Whole map -99%) it was known the engine was operating almost exclusively on vaporized ethanol, begin enough to switch the CNG kit to gas for start to control vaporized fuel through its gate valve. On both cases supplied torque varied in an unregular kind, but with ascending tendency, as long as engine got warmer. Figure 4.19, referring to warm-up with thermostat, shows evolution of referred temperatures through time. The engine was operated on the following conditions: $\omega=3000\text{rpm}$, $\text{DBT}=22.0^\circ\text{C}$, $\text{WBT}=19.0^\circ\text{C}$, $p_{abs}=695.8\text{mmHg}$, $\dot{m}_{air}=8.69\text{g/s}$, $|\phi|=16.00\%$. At 250s of elapsed test time some load was put into the engine through the dynamometer, at 780s boiler pressure overcame above atmospheric one, at 825s the throttle was suddenly open to $|\phi|=36.75\%$ in order to put air-fuel ratio in a reasonable value and finally at 905s of elapsed time the switch of CNG kit was done.

It can be noted water temperature at boiler inlet remains equal to room temperature until 285s, from when it starts to raise suddenly up to a stabilization tendency at 550s followed by a new

²Do not confuse it with the *thermostatic switch* (cebolão), which does other task than controlling cooler's fan.

temperature rise. Water outlet temperature has a remarkable difference to the inlet temperature until 750s, what indicates on this period fuel was receiving sensible heat until reach some temperature suitable for vaporization. Temperatures evolution of both water and lubricating oil for

Table 4.6: Performance measurements at 2000rpm and WOT with several air-fuel ratio and fuel supply possibilities.

	Liquid rich ($\lambda=0.92$)	Liquid stoichiometric	Steam rich ($\lambda=0.86$)	Steam stoichiometric
τ	58.6	55.6	55.6	57.9
η (%)	27.9	30.9	24.6	30.9
\dot{m}_{etanol}	1.77	1.51	1.90	1.57
\dot{m}_{ar}	13.49	13.45	12.79	12.75
CO (g/kWh)	104.02	5.69	220.11	14.32
THC (g/kWh)	2.76	4.05	0.23 [§]	0.77
NO _x (g/kWh)	3.56	7.58	6.85	6.71

§ Value shown by measuement device out of expected, however kept.

the warm-up test without tharmostat are illustrated on figure 4.20. Operating conditions were: $\omega=3000\text{rpm}$, $\text{DBT}=25.0^\circ\text{C}$, $\text{WBT}=22.5^\circ\text{C}$, $\dot{m}_{ar}=8.50\text{g/s}$, $|\phi|=16.00\%$. Some load was imposed to the engine at 60s of test elapsed time and supplied torque arised monothonically from 6.6Nm up to 12.5Nm when liquid-gas supply switch was done. At 707s of run cooler's fan was turned on and the switch happened at 720s. It is possoble to note temperatures of water, on both boiler inlet and outlet, evolute with linear relation to time, keeping almost constant their difference throughtout the test.

At 580s boiler presure starts to raise and reaches the 0.55kgf/cm^2 mark at 695s, faster time intervals than those observed for warm-up *with* thermostat.

In both cases lubricating oil warms up in similar form, stabilizing its temperature near 85°C . After some minutes running with hot water its temperature comes back to warm-up process up to

Table 4.7: Performance measurements at 2000rpm and WOT with several air-fuel ratio and fuel supply possibilities.

	Liquido rich ($\lambda=0.91$)	Liquid stoich.	Steam rich ($\lambda=0.95$)	Steam stoich.	Steam lean	
					$\lambda=1.2$	$\lambda=1.4$
τ	64.4	65.46	62.05	60.98	49.68	39.63
η (%)	28.2	32.3	31.07	33.1	30.3	27.8
\dot{m}_{etanol}	2.87	2.56	2.47	2.33	2.07	1.80
\dot{m}_{ar}	21.76	21.53	20.71	20.8	21.45	20.89
CO (g/kWh)	109.6	18.31	66.75	6.95	3.85	4.25
THC (g/kWh)	2.12	0.35	0.74	0.66	0.56	0.93
NO _x (g/kWh)	2.65	6.70	6.84	7.15	0.36	0.15

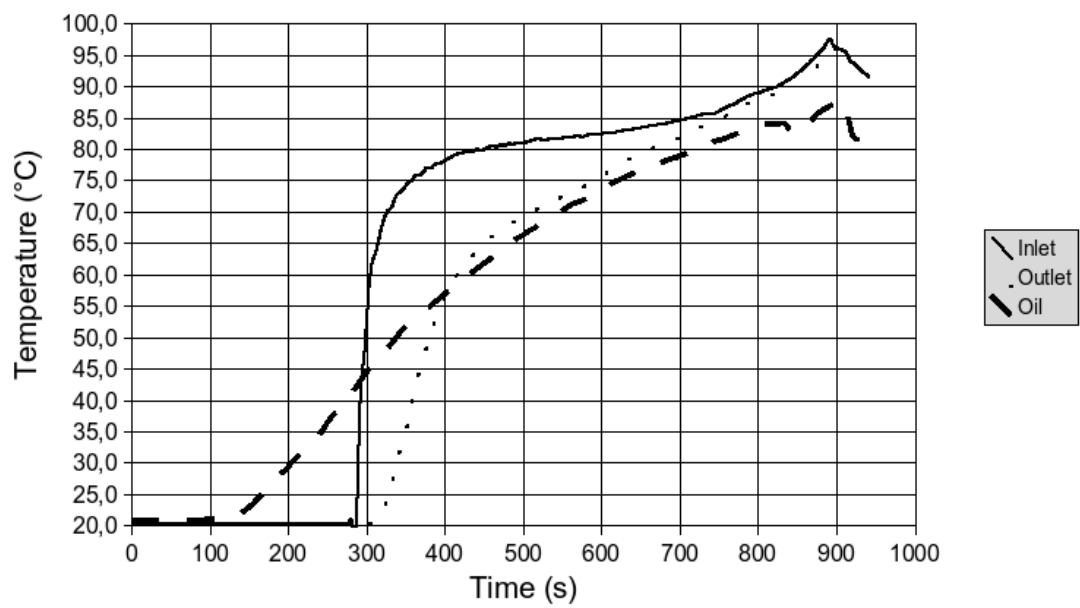


Figure 4.19: Waater and lubricating oil warm-up, with thermostat installed.

the range of 110°C, reaching 120°C peaks. Boiler pressure evolution through time on transient warming up tests is illustrated on figue 4.21. One can note without thermostat (left line) pressure reaches a reasonable level before 700s, while the same process with this device installed takes 780s to start pressure raise and nearly 920s for producing ethanol steam at satisfactory pressure.

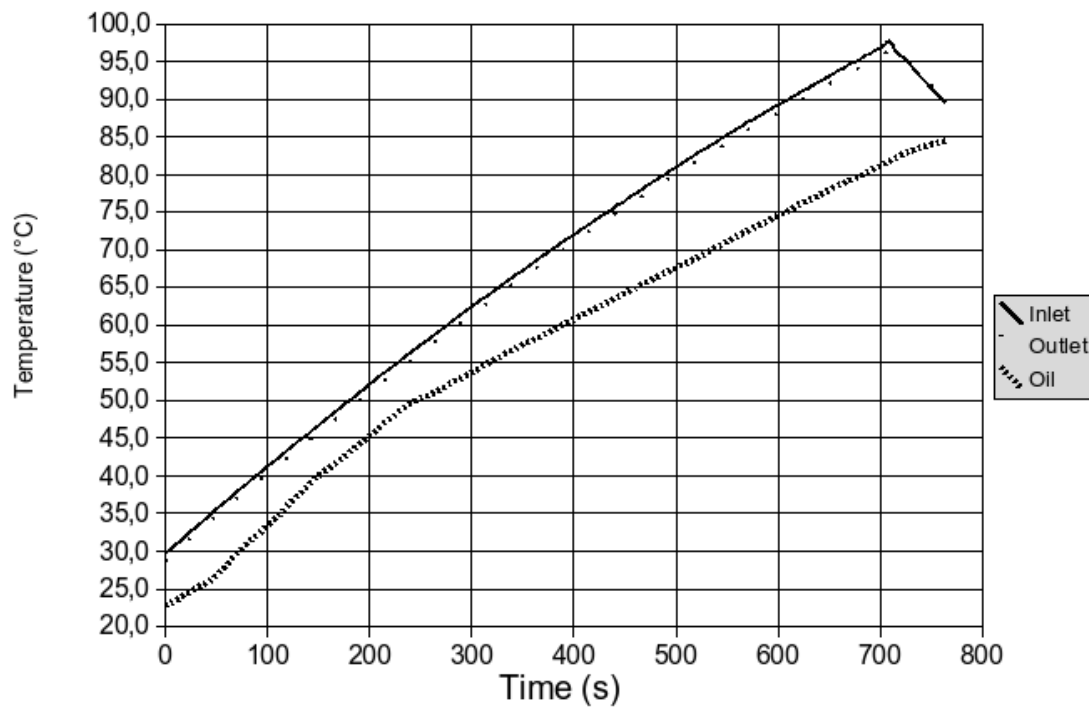


Figure 4.20: Water and lubricating oil warm-up, without thermostat.

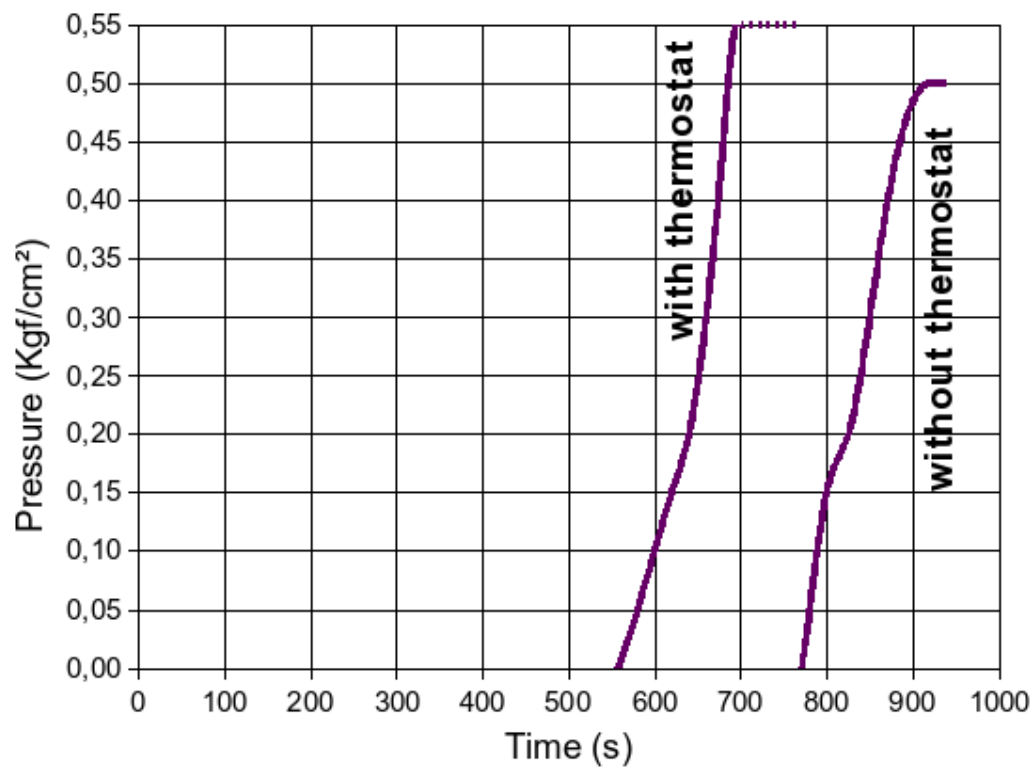


Figure 4.21: Pressure evolution of ethanol inside boiler, with and without thermostat installed.

Chapter 5

Conclusions

On majority of operating regimes studied the PVEE has shown ability to enhance efficiency, it means, produce at a given rotation speed a determined amount of torque with a lower fuel consumption. In steady-state conditions some better efficiency and/or less emissions operation points were found if compared to those equivalent on the engine equipped with the original ECU, remaining only the development of ways for run it in transient conditions with good performance and stability. Actually the success obtained in steady-state regimes allows to use it in stationary applications (where there is no natural gas availability, which is a very cheaper commodity taking into account the amount of energy both fuels deliver) ($\lambda = 2,00$) or even in hybrid vehicles (which have one internal-combustion engine and other electric one) which operate their ICE's near the maximum efficiency conditions.

A PVEE-installed vehicle, ready for real use in passenger vehicles may be put under standard emission tests, in order to investigate its abilities to meet each country legal requirements. This kind of test is not possible in a bench dynamometer. In the case of being possible to reduce the quantity of just one pollutant, for example partially oxidized fuel or nitrogen oxides, or even all them; it is possible to propose some catalytic converter which is not a three-way one and realizes just the needed tasks; or even develop a three-way catalyst which needs lesser contact areas and consequently leads to lesser head losses. It would aim to improve fuel efficiency.

The wanted torque was pursued on the vaporized fuel operating regimes with wider throttle valve openings on the majority of cases, which did not damaged the torque at higher accelerator demands. Then, intermediate power/throttle regimes up to the maximum torque for a given engine speed may be done with adjustments on the fuel injection system, which can lead to regimes with a little air excess in order to re-establish the original torque.

The fuel supply system used, indirect-suction monointer, is not favorable to high air intake ratios neither high air excess values. Fuel setam, because its too low density in comparison to liquid fuel, takes space which would be of air into the intake pipeline, blocking admission of part of air into the cylinders. In the same way the homogeneous mixture formed and pre-mixed

through air intake tubes, with air excess, is an invitation to flame irregularities. There is the side advantage of reach lower flame temperatures and lower nitrogen oxide emissions with this kind of air-fuel mixture, since fuel dilution effect balance the lack of cooling effect of liquid fuel latent heat.

Spark advance timing (and consequently combustion times) which have delivered suitable values for efficiency and emissions were not lower with vaporized ethanol for the operating regimes investigated. This way one may presume combustion was slower in these regimes, which is pre-viewable for more diluted mixtures.

Operating conditions for the PVEE with 100% of air excess ($\lambda = 2,00$) were reached, but with non-favorable results for efficiency neither CO and THC emissions. It is also impossible to run it full wide open throttle in all regimes, as it be a Diesel engine. What is possible to do by now as to work with wider throttle openings than those which would supply the same torque-speed couple for a given regime. It is not known yet anything about the studied PVEE's behavior at idle speed and with openings lower than 24%.

Chapter 6

Suggestions for future researches

In order to enhance PVEE's technical and economical feasibility, the following alternatives may be worked on:

- Study ethanol boiling varying several parameters as: surface treatments, materials, wall superheat, and steam generator geometry [44]. Such studies will allow to build more safe and compact steam generators, which may also to supply engines in several power ranges.
- Take into account the influence of vibrations caused by the engine itself on fuel ebullition, with the same subject of the previous item.
- Develop some algorithm for controlling throttle valve opening according to the desired torque in ad adequate form (*drive by wire*).
- Use devices like *Supercharger* aiming to improve volumetric efficiency, always damaged with indirect gaseous fuel injection.
- Operate a steam generator with heat available on lubricating oil, which is at higher temperatures (on the 120°C–140°C range), or with higher cooling water temperature ranges, in order to produce ethanol vapor at higher pressures. More elevated absolute pressures, between 200kPa and 300 kPa (figure 3.2) could be achieved, allowing the use of fuel injectors for gaseous fuels like CNG, LPG or hydrogen, eliminating the need of a suction device and bringing the possibilities of direction injection and stratified charge.
- Raise cooling water temperature through use of additives, which would also allow steam generation at suitable pressures at right pressures for gas fuel injectors.
- Use pressure adjusters in order to supply fuel to the engine at constant pressure, instead control it through cooling water temperature. This way pressure drop through step motor valve or thorough gaseous fuel injectors is not subordinated to flotations on water or oil

temperatures. Valves used on LPG or CNG could be used since they have their parts in contact to fuel ready to resist to ethanol vapor, as nitrilic rubber or stainless steel membranes, aluminum parts replaced by brass or stainless steel. New calibration for the pressure ranges used and for ethanol steam properties like higher viscosity must be performed too.

- Search for safe and energetically efficient ways for reduce fuel warm-up time, allowing the vehicle to use vaporized ethanol soonen and for more time.
- Study the use of vaporized ethanol on rotary piston (Wankel) engines. Disadvantages of these engines due to its combustion chamber format can be mitigated with use of gaseous fuel, like its condensation on the chamber walls and crevices or even those due to flame propagation speed. Burley [55] apud Dulger [56] have conquered lower HC emissiond running a Wankel engine using methane as fuel.
- Analyze lubricating oil deterioration in a PVEE. It is well-known ethanol –and mainly vaporized ethanol – decrease oil damage, because its lower chemical affinity and lower disposition to form solid deposits on the combustion chamber. Quantative results, however, are in lack.
- Develop combustion chamber formats which are optimized for burning vaporized ethanol.
- Study the use of catalytic converters adequate to the PVEE's new reality. Future stages of its development can lead to further lower emissions but still higher than the regulated ones, which could be converted with catalysts which cause lower pressure drops if safe pollutation levels could be achieved. It is still possible to reach a situation in which just nitrogen oxides are emitted in lower quantities, leaving to the caltalyst converters just the task of oxidizing partially burnt fuels or vice-versa; taking unnecessary the use of three-way catalyst.
- Use 2,5-dimethylfurane on the supply of an ICE in order to evaluate the possibility of use it in gaseous form too. Due to its differences to ethanol, new operating parameters would need to be discovered for make possible its use in internal combustion engines and in a steam generator. Optimum values of compression rate, air-fuel ratio and spark advance must be pursuit, as well as feasible temperature and pollutant emssion levels.
- Realize dynamical driveability tests, in racetracks or on the streets, with the PVEE installed on some vehicle instead having it on a workbench.

- The development of more compact heat exchangers and the use of lubricating oil as heat source may help the development of PVEE's for motorcycles, aiming to reduce the pollution actually caused by them.

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Appendix A

Glossary

BTDC: Before Top Dead Center. Crank angle to the cylinder axis before its highest position of its alternative movement.

Denaturant: Substance added to carburetting ethanol in order to change its taste and smell, for avoid its use as beverage.

DBT: Dry bulb temperature.

Flash setam: Instantaneously-evaporated fraction of saturated liquid when it expands. At the same specific enthalphy it corresponds to higher vapor titles for lower expansion pressures.

Higher heat Power: Heat supplied by combustion when condensation of water vapor formed on the reactionis possible. The difference between higher and lower hear powers is just the latent heat of water produced.

Lean mixture: The opposite of rich mixture.

“Liquid hammer” shock: Undesired shock of liquid which may occur either by condensed liquid in vapor pipeline or by high-pressure waves where only liquids are transported. It can damage seriously hydraulical components.

Lower heat power: Energy supplied by fuel burn, taking not into account the heat supplied by produced water vapor condensation.

MAP: Manifold Air Pressure.

Rich mixture: Mixtures which contains lower air-fuel ratios than the stoichiometric one, it means, there is more fuel than those defined on stoichiometric conditions.

Stoichiometric mixture: Mixture (on the case, air-fuel mixture) on the theoretically correct proportion in order to have no remaining reagents after the end of reaction. In practice there is some remaining reagents even on this condition.

TDC: Top Dead Center, the most elevated point a piston can achieve on its alternative movement.

TPS: Throttle Position Sensor.

Vapor title: In some vaporizing substance (or condensation) a constant pressure, it is the gaseous mass/total mass ratio.

Wall superheat: Temperature difference between transmitting heat surface to some fluid and its saturation temperature.

WBT: Wet bulb temperature. Temperature measurement made with a wet tissue around thermometer's bulb. It gives the minimum temperature a wet surface can achieve on these conditions.

WOT: Wide/total Open Throttle.

Appendix B

Formulae used

B.1 Interpoler polynomial

Ethanol vapor pressure and its density at saturated conditions were calculated even it was needed with aid of a Lagrange interpoler polynomial with equally-spaced arguments. Table B.1 brings values of absolute saturated vapor pressures and its densities, on the ranges of temperatures from -3°C up to 127°C (from 270K to 400K), with 10K steps. Equality of argument spacement (on this case the ethanol temperature) allows a variable change in the form

$$u = \frac{T - T_0}{h} \quad (\text{B.1})$$

Where T_0 is the lower table temperature and h is the non-null step between consecutive values. The interpolated value of the desired properties is then defined by

$$P(T_0 + uh) = \sum_{k=0}^n f_k \frac{(u-0)(u-1)\dots[u-(k-1)][u-(k+1)]\dots(u-n)}{(k-0)(k-1)\dots[k-(k-1)][k-(k+1)]\dots(k-n)} \quad (\text{B.2})$$

Where f_k is the tabled value for steam pressure or density and n equals 13.

Such calculations were performed on an OpenOffice spreadsheet, version 2.0.3. There is the cell for temperature value inlet, intermediate cells for calculation of parameters u (temperature-substitution variable), for the $\Lambda_k = \frac{\prod_j u-j}{\prod_j k-j}$, $j \neq k$ polynomials and for the products $f_k \Lambda_k$ used on the interpolation of both properties.

An illustration of the table used is on the figure B.1.

B.2 Calculation of emissions and other parameters referring to engine run

On this section the formulae used on the calculation of quantities presentes on tables 4.1 e 4.2 are described.

With a 4.030kg mass over load cell's weighing scale, which has an arm distance of 975mm

interpoladores.ods - OpenOffice.org Calc

File Edit View Insert Format Tools Data Window Help

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B18 f(x) Σ = =(B19-A2)/10

	A	B	C	D	E	F	G	H	I
1	Temperatura(K)	Psat (kPa)	lambda(k)	f(k)*lambda(k)	indice(k)		Vol. (m ³ /kg)	f(k)lambda(k)	
2	270	1,295	3,755E-05	0,000	0		37,6600	0,0014	
3	280	2,585	-5,533E-04	-0,001	1		19,5820	-0,0108	
4	290	4,887	3,830E-03	0,019	2		10,7180	0,0411	
5	300	8,802	-1,660E-02	-0,146	3		6,1388	-0,1019	
6	310	15,184	5,071E-02	0,770	4		3,6599	0,1856	
7	320	25,202	-1,174E-01	-2,958	5		2,2612	-0,2654	
8	330	40,403	2,191E-01	8,852	6		1,4420	0,3159	
9	340	62,776	-3,651E-01	-22,922	7		0,9462	-0,3455	
10	350	94,815	8,216E-01	77,898	8		0,6367	0,5231	
11	360	139,570	4,564E-01	63,704	9		0,4382	0,2000	
12	370	200,700	-6,086E-02	-12,214	10		0,3078	-0,0187	
13	380	282,510	9,959E-03	2,813	11		0,2201	0,0022	
14	390	389,960	-1,186E-03	-0,462	12		0,1599	-0,0002	
15	400	528,710	7,093E-05	0,038	13		0,1179	0,0000	
16									
17									
18	u	8,50							
19	Temperatura	355							
20	P interpolada	115,39							
21	Vol. específico	0,5268							
22	interpolado (vapor)								
23									

Sheet 2 / 3 Default 100% STD Sum = 8,50

Figure B.1: Electronic spreadsheet used for interpolation calculations of saturated ethanol properties.

Table B.1: Ethanol vapor pressure as a function of temperature.

Temperature (°C)	Vapor pressure (Pa)	Saturated steam dens. (kg/m ³)
-3	1295	0.027
7	2585	0.051
17	4887	0.093
27	8802	0.163
37	15184	0.273
47	25202	0.442
57	40403	0.693
67	62776	1.057
77	94815	1.571
87	139570	2.282
97	200700	3.249
107	281570	4.543
117	389960	6.254
127	528710	8.482

Source: [50]

(torque evaluated 3.929 kgfm), cell's display exhibits the number 18.8. So, the 1.0kgfm torque corresponds to $\frac{18.8}{3.929} = 4.784$ on the load cell's display, remaining the conversion to Newton-meter with its multiplication by gravity's acceleration.

$$\tau = \frac{(\text{load cell})}{4.784} \times 9.81 \quad (\text{B.3})$$

Power: using conversion factor from rpm to rad/s and from Watt to kilowatt we have

$$P = \frac{\pi \tau \omega}{30000} \quad (\text{B.4})$$

Thermodynamic efficiency: it was used the "100" factor for conversion for percentage.

$$\eta = \frac{100 \times P}{\dot{m}_{\text{ethanol}} \times \text{PCI}} \quad (\text{B.5})$$

Absolute pressure on intake manifold:

$$\text{MAP} = p_{\text{abs}} - \text{vacuum}_{\text{intk}} \quad (\text{B.6})$$

Boiler pressure: conversion factors were used between read data from gauges in kgf/cm² or mmHg to the desired units which is kPa, needed to the verifications of saturation temperature.

$$p_{\text{abs,boiler}} = \frac{p_{\text{abs}} \times 101.325}{760} + p_{\text{boiler}} \times 98.1 \quad (\text{B.7})$$

Mean ethanol consumption: the number 100 was divided by the mean times for consumption of 100g of fuel.

$$\dot{m}_{\text{ethanol}} = \frac{500}{M1 + M2 + M3 + M4 + M5} \quad (\text{B.8})$$

Where M1 ... M5 are the 5 100g consumption time measurements performed. In some very unstable regimes in which there was not able time to realize 5 measurements, the numerator 500 was replaced by 400 or 300, respectively, if just 4 or 3 measurements were possible.

the air-fuel ratio was calculated for comparison effects to the values shown by the gas analyzer. For the air-fuel ratio, we have:

$$\Lambda = \frac{\dot{m}_{air}}{\dot{m}_{ethanol}} \quad (B.9)$$

Air-fuel ratio normalized by its stoichiometric value for hydrated ethanol is defined by:

$$\lambda = \frac{\Lambda}{8.33} \quad (B.10)$$

Total combustion products flow ratio in dry base: it is needed to subtract the three water vapor molecules created on the burning of each ethanol molecule (negleting burn inefficiencies) and transform the mass flow from g/s (numerator) to mol/s according to a mean ponderated molar mass of the whole dry emissions (denominator). Because there is no exhaust gas flow measurements, it is taken as equal to the intook air (negligible *blowby*), that is the dry mass calculated on the equation's B.11 numerator. It can be observed nor nitrogen (N₂) neither Argon present on air do not join these calculations¹.

$$DBCP = \frac{\overbrace{\dot{m}_{ethanol} \times (1 - 0.07)}^{\text{anhydrous fraction}} + 0.21 \times \dot{m}_{air} \frac{MM_{O_2}}{MM_{air}} - \overbrace{(3 \dot{m}_{ethanol} \frac{MM_{water}}{MM_{ethanol}})}^{\text{condensed water}}}{\frac{\% CO_2 \times MM_{CO_2}}{100} + \frac{\% CO \times MM_{CO}}{100} + \frac{ppm CH_4 \times MM_{CH_4}}{10^6} + \frac{\% O_2 \times MM_{O_2}}{100} + \frac{ppm NO_x \times MM_{NO}}{10^6}} \quad (B.11)$$

CO production in g/h:

$$\dot{m}_{CO} = DBCP \times \frac{\% CO}{100} \times MM_{CO} \quad (B.12)$$

Hydrocarbon (THC) production, CO₂, O₂, nitrogen oxides (NO_x); respectively, in g/h:

$$\dot{m}_{THC} = DBCP \times \frac{ppm THC}{10^6} \times MM_{CH_4} \quad (B.13)$$

$$\dot{m}_{CO_2} = DBCP \times \frac{\% CO_2}{100} \times MM_{CO_2} \quad (B.14)$$

$$\dot{m}_{O_2} = DBCP \times \frac{\% O_2}{100} \times MM_{O_2} \quad (B.15)$$

Taking into consideration the emission measurment device reads NO molecules:

$$\dot{m}_{NO_x} = DBCP \times \frac{ppm NO}{10^6} \times MM_{NO} \quad (B.16)$$

¹Care must be taken for not confusing "ar" (air) with "Ar" (Argon) in the Portuguese version

N_2 which leaves the engine “pratically untouched”, neglecting the small amount of it which is transformed into some nitorgen oxide:

$$N_{2,\text{circulating}} = \frac{DBCP}{MM_{air}} \times 0.79 \times 3600 \times MM_{N2} \quad (B.17)$$

Specific emissions, it means, related to the amount of mechanical energy produced by the engine, were calculated dividing their production rate (grams/s) by the engine’s delivered power (kW), obtaining the specific production in g/kWh units for each emission.

$$\dot{m}_{i,spc} = \frac{\dot{m}_i}{P} \quad (B.18)$$

Wheree i can be any emission and \dot{m}_i represents any mass flow described on the formulae B.12 through B.16.

On tha same way, ethanol specific consumption is defined by:

$$\dot{m}_{ethanol,spc} = \frac{\dot{m}_{ethanol}}{P} \quad (B.19)$$

The estimated water flow through the cooling system was made, in order to aid the understanding of heat transfer phenomena anf those related to lead losses into the engine block’s galleries and the boiler. It is a coarse estimate, because the temperature and ethanol production rate measurements were not stable.

$$\dot{m}_{H2O,cooling} = \frac{h_{lv,ethanol} \times \dot{m}_{ethanol}}{c_{p,H2O} \times (T_{intk,cooler} - T_{exit,cooler})} \quad (B.20)$$

In order to investigate the influences on NO_x emissions the time between ignition and TDC was calculated too. This time (in milisseconds) takes into account the time for a complete crankshaft revolution (inverse of number of revolutions per second, multiplied by 1000) and the revolution fraction corresponding to the sparking advance angle.

$$t_{ign-TDCS} = \frac{60}{0.001\omega} \times \frac{^{\circ}BTDC}{360^{\circ}} = \frac{^{\circ}BTDC}{0.006\omega} \quad (B.21)$$

Appendix C

Collected data

C.1 Engine warm-up

Table C.1 contains data collected from data acquisition system, and the boiler pressure observed during transient warm-up tests (seção 4.4).

Table C.1: Transient warm-up tests.

Working time (s)	With thermostatic valve				Without thermostatic valve				Obs.:
	Inlet (°C)	Outlet (°C)	Oil (°C)	Pressure (kgf/cm ²)	Inlet (°C)	Outlet (°C)	Oil (°C)	Pressure (kgf/cm ²)	
0	20.4	20.5	20.8	0.00	29.7	28.6	22.8	0.00	Inj=2.56ms(c)
5	20.4	20.5	20.8		30.1	29.1	23.0		2.39ms(s)
10	20.5	20.5	20.8		30.7	29.8	23.3		
15	20.5	20.5	20.8		31.2	30.4	23.7		
20	20.5	20.5	20.8		31.8	31.0	24.2		
25	20.4	20.5	20.8		32.5	31.6	24.5		
30	20.4	20.5	20.8		33.0	32.1	24.9		
35	20.5	20.5	20.8		33.6	32.7	25.3		
40	20.5	20.5	20.8		34.2	33.3	25.6		
45	20.5	20.5	20.8		34.8	33.9	26.1		
50	20.4	20.5	20.8		35.4	34.5	26.7		
55	20.5	20.5	20.8		36.0	35.0	27.2		
60	20.5	20.5	20.8		36.5	35.6	27.9		cell=3.7(s)
65	20.4	20.5	20.8		37.1	36.2	28.8		
70	20.5	20.5	20.8		37.7	36.8	29.5		
75	20.4	20.5	20.8		38.3	37.3	30.2		
80	20.4	20.5	20.9		38.8	37.9	30.9		
85	20.4	20.5	20.9		39.4	38.5	31.5		
90	20.4	20.5	20.9		40.0	39.0	32.0		
95	20.5	20.5	20.9		40.5	39.6	32.6		
100	20.5	20.5	21.0		41.1	40.2	33.3		
105	20.5	20.5	21.1		41.6	40.7	33.9		
110	20.5	20.5	21.2		42.1	41.2	34.4		
115	20.5	20.5	21.3		42.7	41.8	35.1		

To be continued

Continuation

120	20.4	20.5	21.4		43.3	42.3	36.0		
125	20.5	20.5	21.5		43.8	42.9	36.6		
130	20.5	20.5	21.7		44.4	43.4	37.2		
135	20.5	20.5	21.8		44.9	44.0	37.9		
140	20.5	20.5	22.1		45.4	44.5	38.6		
145	20.5	20.5	22.4		46.0	45.0	39.3		
150	20.5	20.5	22.9		46.5	45.6	39.8		2.82ms(c)
155	20.5	20.5	23.5		47.1	46.1	40.4		c=4.6(c)
160	20.5	20.5	24.1		47.6	46.7	40.9		c=5.3(s)
165	20.5	20.5	24.8		48.2	47.2	41.3		
170	20.5	20.5	25.5		48.7	47.7	41.8		
175	20.5	20.6	26.0		49.3	48.3	42.3		
180	20.5	20.5	26.6		49.8	48.8	42.9		
185	20.5	20.5	27.3		50.3	49.4	43.5		
190	20.5	20.6	28.1		50.9	49.9	44.0		
195	20.5	20.6	28.7		51.4	50.5	44.6		2.49ms(c)
200	20.5	20.6	29.3		52.0	51.0	45.1		c=4.6(c)
205	20.5	20.6	30.1		52.5	51.5	45.6		
210	20.5	20.6	30.8		53.1	52.1	46.2		
215	20.5	20.5	31.3		53.6	52.6	46.9		
220	20.5	20.5	31.8		54.1	53.1	47.5		
225	20.5	20.5	32.4		54.6	53.6	48.0		
230	20.5	20.6	32.9		55.1	54.1	48.5		
235	20.5	20.6	33.5		55.7	54.7	49.1		
240	20.5	20.6	34.4		56.2	55.2	49.5		
245	20.5	20.6	35.4		56.7	55.7	49.9		
250	20.5	20.6	36.4		57.2	56.3	50.2		c=5.9(s)
255	20.5	20.6	37.3		57.7	56.8	50.5		
260	20.5	20.6	38.0		58.2	57.3	50.7		
265	20.5	20.6	38.6		58.7	57.8	51.0		
270	20.6	20.5	39.3		59.2	58.3	51.4		c=5.8(c)
275	20.6	20.5	40.1		59.7	58.8	51.8		
280	20.7	20.5	40.9		60.3	59.3	52.2		
285	20.6	20.5	41.7		60.8	59.8	52.5		
290	39.4	20.5	42.6		61.3	60.3	52.8		
295	47.2	20.5	43.6		61.8	60.8	53.2		
300	56.0	20.5	44.4		62.3	61.3	53.5		c=6.1(s)
305	61.9	20.6	45.2		62.8	61.8	53.9		
310	63.8	20.8	46.0		63.3	62.3	54.4		
315	66.4	21.3	46.7		63.7	62.7	54.7		
320	68.9	22.4	47.3		64.2	63.2	55.1		
325	70.3	24.4	47.9		64.7	63.8	55.5		
330	70.9	26.6	48.7		65.2	64.2	55.9		c=6.8(c)
335	72.5	29.2	49.5		65.7	64.7	56.3		
340	73.3	31.7	50.3		66.2	65.2	56.6		
345	73.8	33.8	51.1		66.7	65.7	56.9		
350	74.6	36.0	51.6		67.1	66.2	57.3		

To be continued

Continuation

355	75.2	38.9	52.1		67.6	66.6	57.6		
360	75.9	41.9	52.7		68.1	67.1	57.9		
365	76.1	44.4	53.2		68.6	67.6	58.2		
370	76.4	46.4	53.8		69.1	68.1	58.5		c=6.4(s)
375	77.3	48.6	54.4		69.5	68.6	58.9		
380	77.2	50.7	54.8		70.0	69.1	59.2		
385	77.7	52.5	55.4		70.5	69.5	59.6		
390	77.9	54.2	55.9		71.0	69.9	60.0		c=7.0(c)
395	78.1	55.8	56.4		71.4	70.4	60.3		
400	78.5	57.1	56.9		71.8	70.9	60.6		
405	78.8	58.3	57.3		72.3	71.3	61.0		
410	79.2	59.4	57.9		72.8	71.8	61.4		
415	79.4	60.4	58.3		73.2	72.3	61.6		
420	79.6	61.2	58.8		73.7	72.7	61.9		
425	79.7	62.0	59.3		74.2	73.2	62.3		
430	79.9	62.8	59.8		74.6	73.6	62.7		
435	79.9	63.4	60.3		75.0	74.1	63.0		
440	80.2	64.0	60.8		75.5	74.6	63.4		
445	80.0	64.7	61.3		76.0	75.0	63.8		
450	80.3	65.4	61.8		76.3	75.4	64.1		
455	80.6	65.9	62.2		76.9	75.9	64.5		
460	80.7	66.3	62.7		77.3	76.4	64.8		
465	80.6	66.8	63.3		77.7	76.8	65.1		
470	80.9	67.3	63.8		78.2	77.3	65.5		c=7.4(c)
475	80.7	67.7	64.3		78.6	77.7	65.8		
480	81.1	68.1	64.7		79.1	78.1	66.2		
485	81.0	68.5	65.1		79.5	78.5	66.5		
490	81.2	68.9	65.6		79.9	79.0	66.8		
495	81.2	69.2	65.9		80.4	79.4	67.2		
500	81.2	69.6	66.3		80.8	79.8	67.5		
505	81.4	70.0	66.7		81.2	80.3	67.8		
510	81.6	70.3	67.1		81.7	80.8	68.1		
515	81.8	70.6	67.5		82.1	81.2	68.4		
520	81.8	70.8	67.9		82.6	81.6	68.8		c=7.2(s)
525	81.7	71.2	68.3		83.0	82.0	69.1		
530	81.7	71.5	68.7		83.4	82.5	69.4		
535	81.8	71.8	69.1		83.8	82.8	69.8		
540	82.0	72.1	69.5		84.2	83.2	70.2		
545	81.9	72.4	69.8		84.7	83.6	70.6		
550	82.0	72.6	70.2		85.1	84.0	71.0		
555	82.1	72.8	70.6		85.5	84.5	71.4		
560	82.1	73.1	71.0		85.9	84.9	71.7		
565	82.2	73.4	71.3		86.3	85.3	72.0		
570	82.3	73.8	71.6		86.8	85.7	72.4		
575	82.2	74.1	71.8		87.1	86.1	72.7		
580	82.4	74.5	72.1		87.5	86.5	73.0	0.05	c=7.2(s)
585	82.5	74.9	72.5		87.9	86.9	73.4		

To be continued

Continuation

590	82.6	75.3	72.9		88.3	87.3	73.7		
595	82.6	75.6	73.3		88.7	87.7	74.1		
600	82.7	75.9	73.5		89.1	88.0	74.4	0.10	1.51ms(s)
605	82.8	76.4	73.9		89.4	88.5	74.7		c=6.5(s)
610	82.8	76.9	74.3		89.8	88.9	75.0		
615	82.9	77.3	74.6		90.2	89.3	75.4		
620	83.0	77.7	74.9		90.6	89.7	75.8	0.15	
625	83.1	78.0	75.2		91.0	90.0	76.1		c=6.8(s)
630	83.3	78.4	75.5		91.3	90.4	76.4		
635	83.3	78.7	75.8		91.8	90.8	76.8		
640	83.4	79.0	76.0		92.1	91.1	77.1	0.20	c=7.0(s)
645	83.5	79.3	76.3		92.5	91.5	77.5		
650	83.6	79.5	76.6		92.9	91.9	77.8		
655	83.7	79.8	76.8		93.3	92.2	78.2		
660	83.9	80.1	77.0		93.7	92.6	78.5	0.30	c=7.2(s)
665	84.0	80.4	77.2		94.0	93.0	78.8		c=7.6(c)
670	84.1	80.6	77.5		94.5	93.3	79.2		
675	84.2	80.9	77.7		94.8	93.7	79.4		
680	84.3	81.1	78.0		95.1	94.1	79.7	0.45	1.26ms(s)
685	84.4	81.3	78.3		95.6	94.5	80.0		c=6.9(s)
690	84.6	81.6	78.5		95.9	94.9	80.4		
695	84.7	81.9	78.6		96.3	95.2	80.7	0.55	c=7.0(s)
700	84.8	82.2	78.9		96.7	95.6	81.0	0.55	
705	85.0	82.5	79.1		97.1	96.0	81.3	0.55	ligou ventoinha(s)
710	85.2	82.8	79.4		97.6	96.4	81.7	0.55	c=7.8(c)
715	85.3	83.1	79.7		96.5	96.7	82.0	0.55	
720	85.5	83.4	80.0		95.7	96.3	82.4	0.55	
725	85.7	83.7	80.3		95.3	95.5	82.7	0.55	
730	85.7	84.1	80.7		94.5	94.9	83.0	0.55	
735	85.8	84.3	80.8		93.7	94.3	83.3	0.55	
740	85.8	84.6	81.0		92.9	93.5	83.5	0.55	
745	86.0	84.7	81.2		92.2	92.8	83.6	0.55	
750	86.4	85.0	81.4		91.4	92.0	83.8	0.55	1.29ms(c)
755	86.7	85.3	81.6		90.7	91.3	84.0	0.55	c=6.8(c)
760	87.0	85.7	81.8		90.1	90.6	84.2	0.55	
765	87.2	86.1	82.0		89.4	90.0	84.3	0.55	
770	87.6	86.4	82.3						
775	87.9	86.7	82.6						
780	88.3	87.0	82.8	0.05					c=4.9(c)
785	88.6	87.4	83.1						
790	88.8	87.7	83.4						
795	89.1	88.0	83.5						
800	89.2	88.2	83.6	0.15					0.26ms(c)
805	89.5	88.4	83.8						c=1.0(c)
810	89.7	88.6	83.9						
815	89.9	88.9	83.9						
820	90.0	89.0	84.0						

To be continued

Conclusion

825	90.3	89.2	84.0	0.20					$ \phi =36.75\%(c)$
830	90.7	89.4	84.1						$c=21.0(c)$
835	91.1	89.7	83.9						
840	91.4	90.1	83.1						$c=4.6(c)$
845	91.9	90.5	82.7						
850	92.4	90.9	82.6						
855	93.0	91.4	82.9						
860	93.5	91.9	84.0	0.35					
865	94.1	92.4	84.3						$c=24.9(c)$
870	94.7	92.9	84.8						
875	95.4	93.5	85.5						
880	96.1	94.1	86.0						
885	96.9	94.9	86.4	0.45					fan set on
890	97.7	95.6	86.7						$c=28.8(c)$
895	96.6	96.1	87.0						
900	96.2	95.9	87.3						
905	96.0	95.4	87.5						commutation
910	95.7	95.1	87.5						$c=12.7(c)$
915	94.4	94.7	85.3						
920	93.9	94.0	82.5	0.50					
925	93.3	93.3	81.6	0.50					
930	92.8	92.7	81.5	0.50					
935	92.2	92.2	81.6	0.50					
940	91.7	91.7	82.0	0.50					

C.2 Dynamometric tests

Before realizing the tests with the vaporized ethanol supply system the engine run with its original ECU, for comparison effects. The results of fuel consumption are on the table C.3, while the other measurements are on the table C.2.

For the vaporized ethanol runs, consumption measurements are on the tables C.4 and C.5.

Table C.2: Collected data on original ECU (except consumption).

ω (rpm)	CO (%)	THC (ppm)	NO _x (ppm)	CO ₂ (%)	O ₂ (%)	Pulse inj.(ms)	Ignitiion °BTDC	Temperatures (°C)				Consumtion air (g/s)	Pressure/vacuum (mm column)			Load cell
								Int.	Exhs.	DBT	WBT		Int.(Hg)	Exhs.(H ₂ O)	Abs.(Hg)	
Idle																
840	0.64	349	0	12.5	2.69	4.8	6.0	45.3	195.8	22.0	20.0	2.40	350	15	696.0	0.0
Throttle at 25% (23 degrees)																
2000	0.32	109	1933	25.4	0.69	7.4	18.7	92.8	312.5	21.0	19.0	9.11	180	120	693.5	21.0
3000	0.34	233	1942	14.5	0.69	5.6	18.6	51.6	259.4	21.0	19.0	10.56	310	75	693.5	13.7
4000	0.45	431	1203	14.2	0.81	4.1	22.1	51.4	290.6	21.0	19.0	10.80	440	85	693.5	6.0
Throttle at 50% (45 degrees)																
2000	0.25	786	1950	14.2	0.83	10.3	17.5	56.7	339.5	21.0	19.0	13.08	100	245	693.5	32.0
3000	0.30	387	1956	14.3	0.77	10.7	14.2	57.6	423.5	21.0	19.0	208.6	120	350	693.5	33.9
4000	0.38	849	2961	14.1	0.85	11.1	13.7	49.5	424.0	21.0	19.0	28.59	50	665	693.5	32.6
WOT (82 degrees)																
2000	3.12	724	997	12.9	0.32	11.7	15.8	50.1	293.4	22.0	20.0	13.49	10	190	696.0	32.7
3000	3.36	568	758	12.8	0.26	12.6	14.0	48.3	694.0	22.0	20.0	21.76	18	260	696.0	35.9
4000	2.46	370	1262	12.3	0.31	13.0	13.6	49.5	398.3	22.0	20.0	31.01	20	350	696.0	37.5

Table C.3: Fuel consumption with original ECU.

	2000 rpm	3000 rpm	4000 rpm
25% (23°)	87.8	79.1	80.7
	87.8	78.7	82.1
	91.4	79.9	79.9
	87.4	80.0	77.8
	88.3	80.9	76.9
Medium	88.54	79.72	79.48
50 % (45°)	66.9	42.3	30.5
	66.2	40.8	30.5
	66.0	41.7	29.7
	66.1	41.1	30.0
	67.2	40.8	30.5
Medium	66.48	41.34	30.24
Maximum (82°)	56.9	34.8	24.7
	56.0	34.8	25.2
	56.7	34.6	24.8
	56.7	35.0	25.2
	56.3	34.8	25.4
Medium	56.52	34.80	25.06
Idle (840rpm)	382.6		
	378.4		
	373.3		
	373.1		
	384.4		
Medium	378.36		

Table C.4: Pressure, air and fuel consumption measurements in several regimes, with vaporized ethanol.

ω (rpm)	$ \phi $ (%)	λ	Ignition °BTDC	Pressure				Ethanol consumption (s/100g) - measurements					Air consumption (g/s)	Load cell
				Int. [◊]	Exst. [†]	Absolute [‡]	Boiler [§]	1	2	3	4	5		
2000	24.25	1.00	21.4	202	70	695.6	0.60	89.3	90.4	91.7	92.7	91.9	8.95	20.5
2000	26.25	1.01	19.8	192	70		0.75	94.0	103.3	102.2	92.9	87.8	8.94	21.0
2000	41.25	1.19	17.4	68	130	694.7	0.75	84.4	78.6	87.9	78.2	80.4	12.31	27.8
2000	57.75	1.30	18.9	20	131	694.7	0.70	79.1	78.4	74.9	77.8	77.1	12.90	26.2
2000	35.50	1.36	19.5	70	85	694.1	0.65	92.0	87.9	87.4	86.6	88.4	12.04	22.4
2000	40.50	1.48	21.7	39	175		0.60	90.7	94.6	90.3	91.3	91.9	13.8	21.2
2000	53.00	1.01	13.9	20	185	693.6	0.60	64.2	62.4	62.5	61.8	62.1	12.63	32.5
2000	100.00	1.01	13.4	10	190		0.60	64.1	63.2	63.9	63.3	63.5	12.75	32.3
3000	24.50	1.01	26.9	338	145	693.4	0.70	83.2	84.2	83.3	83.9	84.9	9.73	12.4
3000	30.75	1.23	29.1	281	150		0.70	84.8	84.0	83.5	83.4	82.7	11.82	13.7
3000	31.75	1.31	24.7	255	15		0.73	82.6	80.8	81.7	82.4	83.0	12.81	14.0
3000	32.35	1.39	24.1	230	180		0.75	81.8	80.4	82.2	79.6	80.4	13.55	14.6
3000	53.00	1.01	15.6	35	340	696.8	0.60	43.6	43.3	43.7	41.7	41.8	20.49	33.9
3000	100.00	1.02	14.9	20	345	696.8	0.60	41.8	44.6	42.0	43.0	43.4	20.80	34.0
3000	100.00	1.22	14.5	20	345	696.8	0.50	48.2	48.5	48.5	46.0	50.2	21.45	27.7
3000	100.00	1.41	17.1	20	340		0.42	53.4	55.1	56.5	55.2	57.6	20.89	22.1
4000	24.50	1.00	22.3	440	114	696.0	0.60	85.6	81.9	88.1	84.4	81.9	10.39	6.6
4000	29.25	1.12	22.2	400	125		0.63	82.8	84.7	83.4	87.9	83.7	12.2	7.3
4000	33.75	2.00	31.4	300	160		0.68	79.5	76.1	76.3	80.8	79.7	18.20	5.0
4000	32.25	1.22	19.2	350	165		0.65	69.2	73.2	66.0	68.5	66.6	14.43	7.0
2000	54.00	1.06	17.3	20	345	695.4	0.60	64.3	63.1	67.3	71.0	73.9	12.63	29.9
4000	52.00	1.37	25.8	50	500	696.0	0.65	46.3	39.2	34.6	34.6	-	29.05	26.3
4000	99.25	1.51	21.3	30	520		0.60	39.5	41.1	41.1	40.4	39.2	30.98	22.2
4000	53.00	1.22	19.1	50	530		0.75	32.3	33.8	34.0	36.2	37.3	28.85	29.0
4000	47.50	1.08	15.7	70	510		0.80	35.0	34.7	34.5	35.1	34.1	27.61	28.3
4000	42.50	1.01	15.0	105	500		0.90	33.8	35.3	34.0	-	-	-	27.5

To be continues

Conclusion

ω (rpm)	$ \phi $ (%)	λ	Ignition °BTDC	Pressure				Ethanol consumption (s/100g) - measurements					Air consumption (g/s)	Load cell
				Int. \diamond	Exst. \dagger	Absolute \ddagger	Boiler \S	1	2	3	4	5		
2000	60.75	1.11	8.5	20	267	695.8	0.55	13.19	25.6	73.7	73.0	77.7	70.6	74.8
2000	100.00	1.11	11.0	10	255	695.8	0.55	13.35	26.1	72.0	72.7	76.7	73.1	72.5
3000	55.75	1.31	23.0	30	335	695.8	0.50	21.65	27.7	47.4	47.2	48.6	54.5	52.0
3000	100.00	1.24	20.0	20	335	695.8	0.55	21.60	29.1	49.3	51.6	49.0	50.8	48.8
4000	67.00	1.47	30.1	30	252	695.8	0.60	30.55	24.2	42.1	42.1	43.4	40.6	42.3
2000	100.00	0.86	19.6	10	275	695.8	0.45	12.79	31.0	52.1	53.1	53.1	53.4	51.7
3000	100.00	0.95	15.8	20	365	695.4	0.67	20.71	34.6	41.2	40.8	40.5	40.0	40.0
3000*	100.00	1.00	17.8	10	150	695.4	0.75	21.53	36.5	39.2	39.2	39.7	38.4	38.8
920	0.00	1.00	6.5						0.0					
2000*	100.00	0.99	25.8	10	84	695.0	-	66.6	66.0	65.7	-	-	13.45	31.0

\diamond Vacuum related to atmospheric pressure, in mmHg.

\dagger Pressure above atmospheric, in mmH₂O.

\ddagger Barometric pressure, in mmHg.

\S Pressure above atmospheric, in Kgf/mm².

* Running on liquid ethanol for fair comparison between rich and stoichiometric operating conditions.

Table C.5: Temperature and emissions measurements for several regimes, operating with vaporized ethanol.

ω (rpm)	$ \phi $ (%)	λ	Ignition °BTDC	CO (%)	THC (ppm)	CO ₂ (%)	O ₂ (%)	NO _x (ppm)	Temperatures (°C)				Boiler				
									Int.	Exhst.	DBT	WBT	Eth. steam	Oil	Venturi	In	Out
2000	24.25	1.00	21.4	0.58	180	14.3	0.61	1927	54.2	554.7	24.0	19.0	80.1	97.1	24.9	94.0	93.1
2000	26.25	1.01	19.8	0.90	100	13.8	0.81	1970	55.2	555.0	26.5	20.5	78.7	95.9	26.2	94.0	93.3
2000	41.25	1.19	17.4	0.05	92	12.5	3.77	1977	57.8	542.2			78.9	99.5	27.6	94.1	93.0
2000	57.75	1.30	18.9	0.06	96	11.5	4.74	1984		525.5	27.5	21.5	79.4	99.9	27.4	94.0	93.0
2000	35.50	1.36	19.5	0.06	100	11.2	5.65	345	54.8	532.4	207.0	20.0	78.8	97.2	25.8	94.2	93.3
2000	40.50	1.48	21.7	0.07	101	10.0	6.92	132	58.1	523.5			78.4	98.2	28.1	93.5	92.3
2000	53.00	1.01	13.9	0.37	178	14.6	0.78	1962	58.9	592.1	31.5	23.0	78.0	100.7	29.8	94.2	93.0
2000	100.00	1.01	13.4	0.45	212	14.4	0.70	1968	60.5	581.7	32.5	24.0	77.2	100.1	30.1	94.1	93.0
3000	24.50	1.01	26.9	0.54	24	14.4	0.70	1968	58.8	617.8	32.5	24.5	76.7	104.4	30.7	93.9	93.3
3000	30.75	1.23	29.1	0.10	239	12.1	4.40	1969	56.7	560.7			76.7	105.3	28.9	93.8	93.2
3000	31.75	1.31	24.7	0.08	186	11.0	5.50	750	56.1	574.0	31.5	24.0	77.0	103.8	28.6	94.1	93.3
3000	32.25	1.39	24.2	0.07	201	10.5	6.00	100	57.9	573.2			75.9	103.2	29.1	93.9	93.1
3000	53.00	1.01	15.6	0.27	176	14.5	0.67	2004	63.8	682.7	33.0	22.5	80.1	107.1	33.7	94.1	93.0
3000	100.00	1.02	14.9	0.21	175	14.4	1.08	2017	64.4	688.2			80.0	106.9	34.4	93.8	92.5
3000	100.00	1.22	14.5	0.09	115	12.1	4.45	85	63.6	664.0	33.0	22.5	78.5	105.3	33.7	91.1	90.1
3000	100.00	1.41	17.1	0.08	154	10.6	6.43	26	63.3	399.3			79.3	104.4	39.6	90.6	89.8
4000	24.50	1.00	22.3	0.40	180	14.5	0.73	1943	57.3	734.4	32.5	25.0	80.6	109.4	31.6	93.9	93.5
4000	29.25	1.14	22.2	0.16	142	13.0	3.10	750	58.8	710.5	33.5	27.0	81.2	110.6	32.6	94.2	93.5
4000	33.75	2.00	31.4	0.19	590	6.8	11.5	1	54.9	515.1	29.0	24.0	82.3	109.2	29.5	95.4	94.8
4000	32.25	1.22	19.2	0.11	563	11.8	5.0	70	59.7	752.9			81.2	120.0	34.5	94.6	93.9
2000	54.00	1.06	17.3	0.06	192	14.0	1.59	1997	66.1	252.8	35.0	27.0	82.3	105.3	34.1	94.4	93.3
4000	52.00	1.37	25.8	0.11	193	10.7	6.12	1976	65.6	541.4	39.0	27.0	84.5	116.3	38.8	97.0	96.0
4000	99.25	1.51	21.3	0.10	188	9.9	7.43	30	68.9	555.7			80.1	112.9	39.5	96.5	95.6
4000	52.00	1.22	19.1	0.09	243	11.8	4.39	1987	71.0	567.0			81.5	112.5	39.4	97.5	9.65
4000	48.50	1.07	15.7	0.09	117	13.5	2.60	1982	70.0	611.2			81.9	119.4	41.2	99.0	97.9
4000	42.50	1.01	15.0	0.11	117	14.3	0.50	190	70.4	622.5	40.0	27.0	82.3	120.0	42.2	99.0	7.3

To be continued

Conclusion

ω (rpm)	$ \phi $ (%)	λ	Ignition °BTDC	CO (%)	THC (ppm)	CO ₂ (%)	O ₂ (%)	NO _x (ppm)	Temperatures (°C)							Boiler	
									Int.	Exhst.	DBT	WBT	Eth. steam	Oil	Venturi	In	Out
2000	60.75	1.11	8.5	0.07	249	13.3	1.99	1924	61.6		22.0	19.0	81.6	101.2	33.0	93.9	93.0
2000	100.00	1.11	11.0	0.08	271	13.3	2.46	1969	61.9		22.0	19.0	81.7	100.5	32.2	93.4	92.5
3000	55.75	1.31	23.0	0.08	286	11.5	5.29	1973	61.4		22.0	19.0	82.0	107.5	31.7	92.9	91.9
3000	100.00	1.24	20.0	0.07	196	12.0	4.20	1970	61.7		22.0	19.0	82.2	107.8	31.9	94.0	93.0
4000	67.00	1.47	30.1	0.10	452	9.9	7.38	492	63.8		22.0	19.0	84.3	115.2	31.8	96.4	95.3
2000	100.00	0.86	19.6	6.98	63	11.1	0.19	300	57.6		22.0	19.0	80.8	96.9	28.8	96.9	93.9
3000	100.00	0.95	15.8	2.05	198	13.6	0.26	1960	62.9		32.0	22.5	83.4	109.9	33.5	98.1	96.9
3000	100.00	1.00	17.8	0.57	95	14.4	0.45	1946	59.8		32.0	22.5	84.1	107.4	29.9	90.6	90.3
920	0.00	1.00	6.5	0.16	1168	14.2	1.16	0	58.2		32.0	22.5					
3000*	100.00	1.00	17.8	0.57	95	14.4	0.45	1946	59.8		32.0	22.5	84.1	107.4	29.9	90.6	90.3
2000*	100.00	0.99	25.8	0.16	997	14.3	0.93	1989	54.0	549.2	32.0	21.0	-	80.0	34.3	-	-

★ Running with liquid ethanol for a fair comparison between rich and stoichiometric regimes.

Table C.6: Extra tests performed with original ECU, for torque-speed regimes comparison.

ω (rpm)	$ \phi $ (%)	λ	Ignition °BTDC	Pressure			Ethanol consumption (s/100g) - measurements					Air consump- tion (g/s)	Load cell	Injection time (ms)
				Int.	Exhst.	Absolute	1	2	3	4	5			
2000	34.6	0.97	17.6	75	109	695.2	70.3	69.8	69	71.1	72.1	12.42	27.9	9.7
3000	28.2	0.97	17.6	75	109	695.2	75.3	73.2	72.1	72.1	70.9	12.20	14.7	5.9
3000	35.9	0.97	15.9	120	127		48.4	48.3	48.4	48.4	48.4	17.7	27.8	8.6
4000	25.6	0.99	21.0	420	105	695.2	76.0	72.8	76.2	75.8	75.1	11.64	7.0	4.4
4000	41.1	1.00	12.7	120	294		33.2	33.5	33.8	-	-	25.56	29.0	

ω (rpm)	$ \phi $ (%)	λ	Ignition °BTDC	CO (%)	THC (ppm)	CO ₂ (%)	O ₂ (%)	NO _x (ppm)	Temperatures (°C)					
									int.	Exhst.	DBT	WBT	Oil	Venturi
2000	34.6	0.97	17.6	0.27	1435	14.4	0.75	1977	51.3	596.3	37.0	24.0	71.8	33.6
3000	28.2	0.97	17.6	0.35	1237	14.5	0.58	2010	55.8	682.9	37.0	24.0	94.0	38.8
3000	35.9	0.97	0.97	15.9	0.35	1195	14.6	0.60	2023	60.6	696.3		103.8	40.2
4000	25.6	0.99	21.0	0.44	899	14.4	0.67	1201	58.0	733.7			105.0	37.5
4000	41.1	1.00	12.7	0.36	519	14.5	0.71	2037	61.5	778.5			113.5	37.2

Units and references for pressure measurements are the same of table C.4.

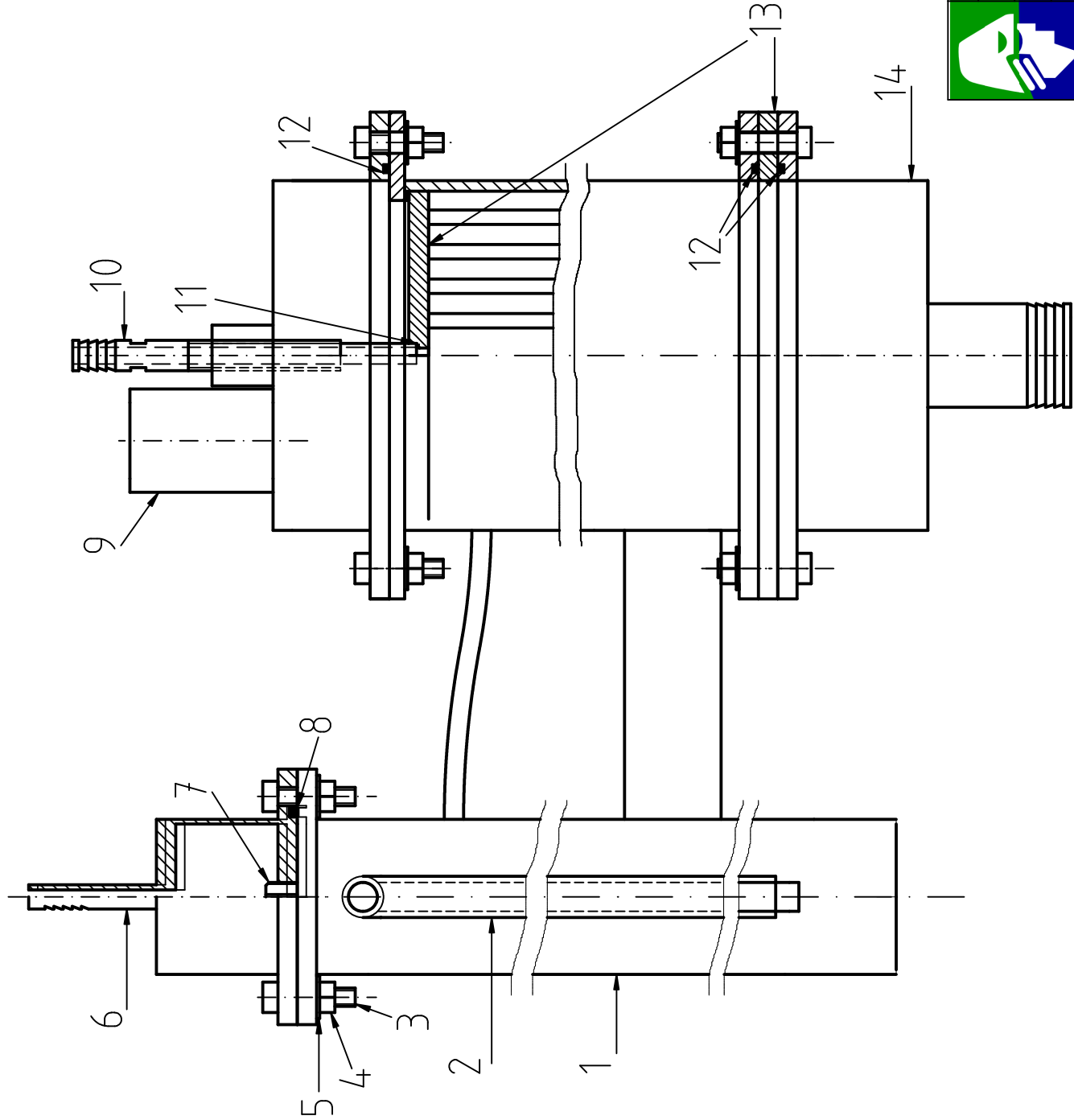
Appendix D

Technical drawings

Ahead, it is presented technical drawings which were used for allow operation of PVEE. Pieces are presented as they *were* manufactured instead as they *should be* produced. Such differences are due to unpredictable obstacles, neccessary *on the fly* changes in the project, handmade production mode and materials availability.

The mechanical assemblies presented are: Steam generator; generator's bowl float; suction device or Venturi; jacketed steam transport tube and adaptations on the valve with step motor.

No.	Qt.	Component	Material
1	1	Case	st. steel
2	1	Level inspector	glass
3	32	Paraf. Allen M6x1"	carbon steel
4	32	Nut M6	carbon steel
5	32	Arruela M6	carbon steel
6	1	Fuel inlet	st. steel
7	1	Needle vv. ø1,5mm	brass
8	1	O'ring 2137	rubber
9	1	Upper water inlet	st. steel
10	1	Steam exit	st. steel
11	1	O'ring 2008	rubber
12	3	O'ring 2158	rubber
13	1	Tube beam	copper
14	1	Lower water exit	st. steel
15	2	O'ring 2035	rubber
16	2	Window ø59x4mm	glass
17	2	Window nut	st. steel
18	1	Hose conn. 8mm	brass
19	1	Tee conn. 5/16"	carbon steel



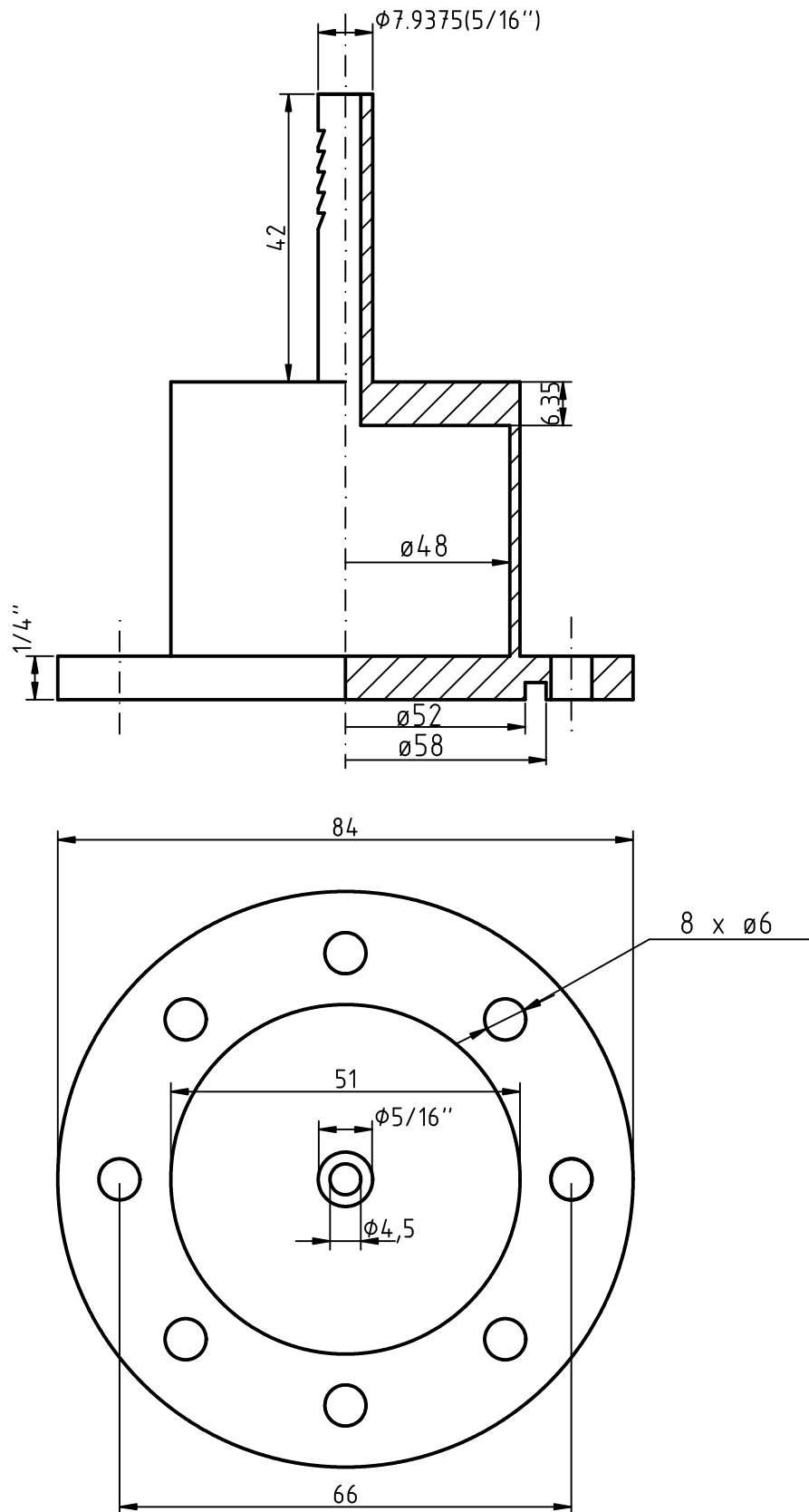
EESC-USP PVEE Project

Steam generator (group)

Author: Francisco José Alves

Date: sep 24, 2006 Sheet: 1/9 Scale: 1:2

No.	Qt.	Component	Material
6	1	Fuel inlet	st. steel



Main unit: milimeter



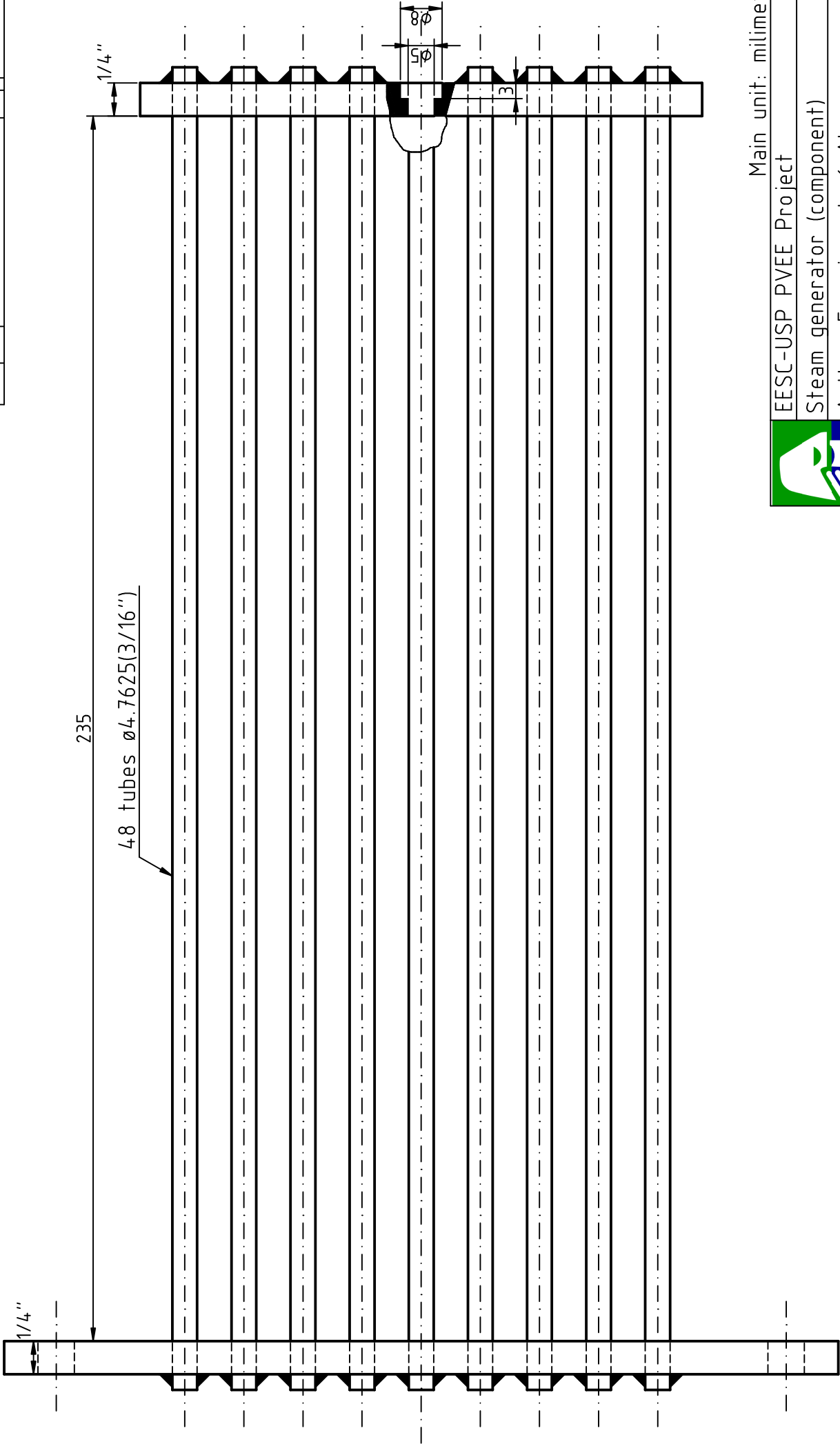
EESC-USP PVEE Project

Steam generator (component)

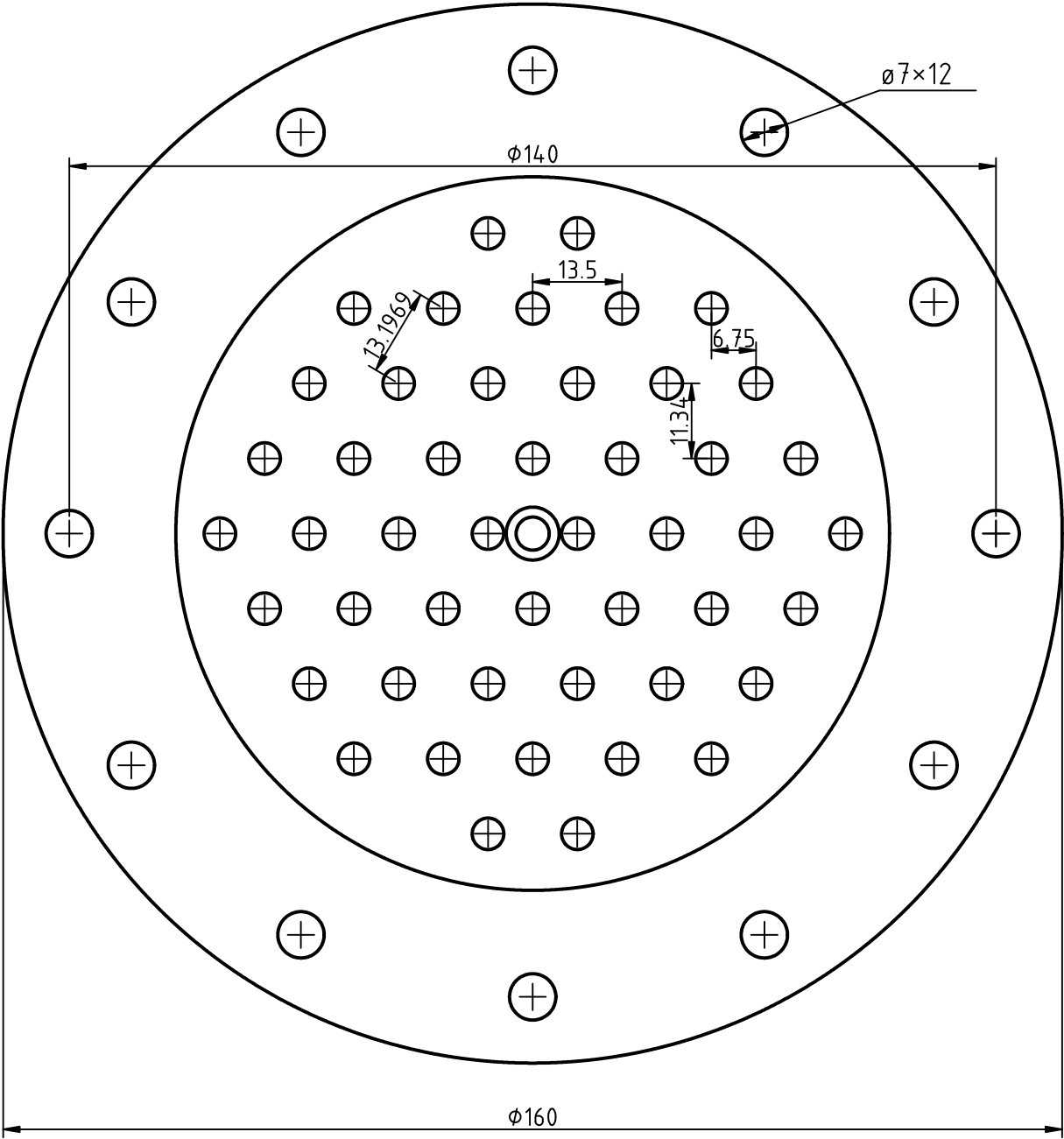
Author: Francisco José Alves

Date: jan 20, 2006 Sheet: 3/9 Scale: 1:1

No.	Qt.	Component	Material
13	1	Tube beam	copper



No.	Qt.	Component	Material
13	1	Tube beam	copper

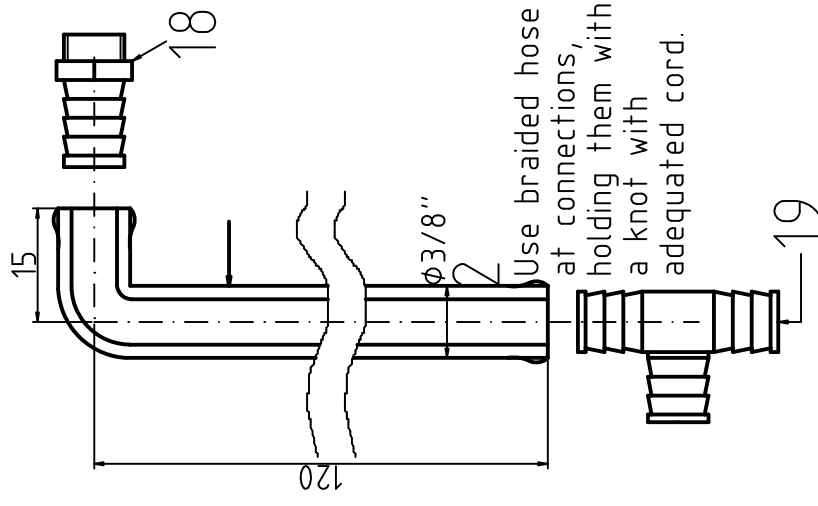
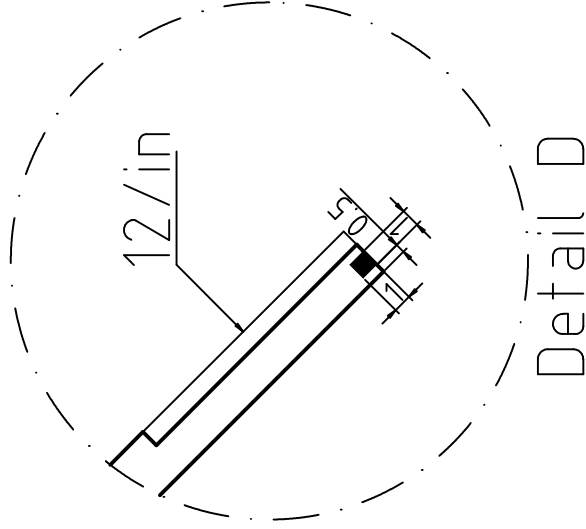
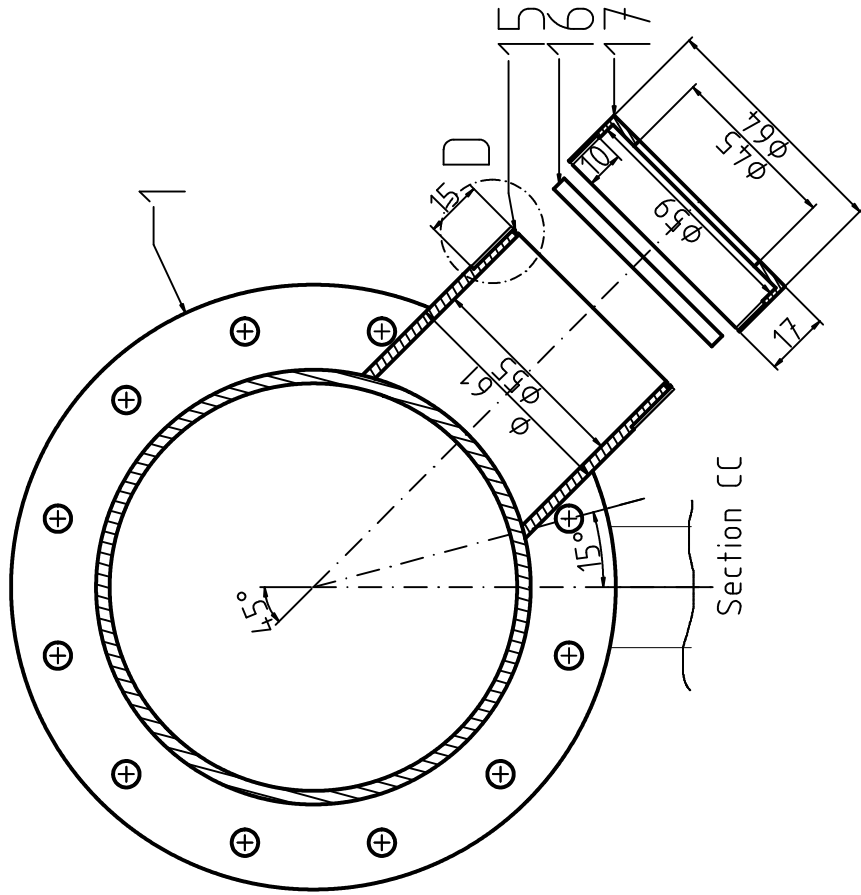


Main unit: milimeter



EESC-USP PVEE Project		
Steam generator (component)		
Author: Francisco José Alves		
Date: jan 20, 2006	Sheet: 8/9	Scale: 1:1

No.	Qt.	Component	Material
1	1	Case	st. steel
2	1	Level viewer	glass
15	2	O'ring 2035	rubber
16	2	Disc $\varnothing 59 \times 4 \text{ mm}$	glass
17	2	Window nut	st. steel
18	1	Hose conn. 8mm	brass
19	1	Tee 5/16"	carbon steel



Use braided hose
at connections,
holding them with
a knot with
adequated cord.

Main unit: milimeter



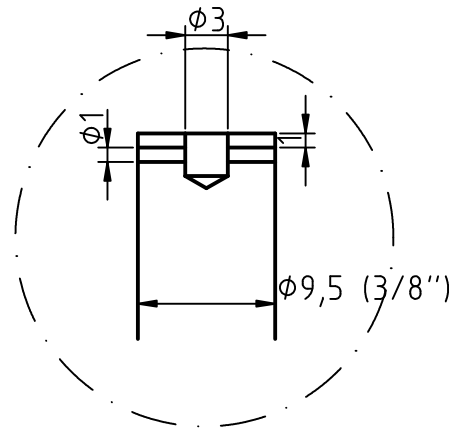
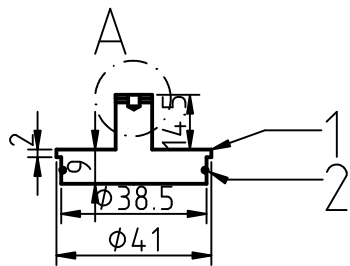
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Steam generator (components)

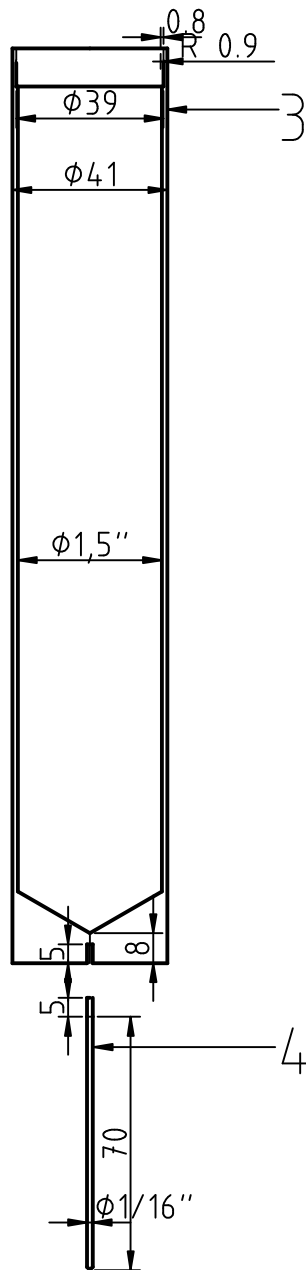
Author: Francisco José Alves

Date: sep 24, 2006 Sheet: 9/9 Scale 1:2

No.	Qt.	Component	Material
1	1	Lid	nylon
2	1	O'ring 2028	Rubber
3	1	Body	nylon
4	1	Guide	st. steel



Detail A

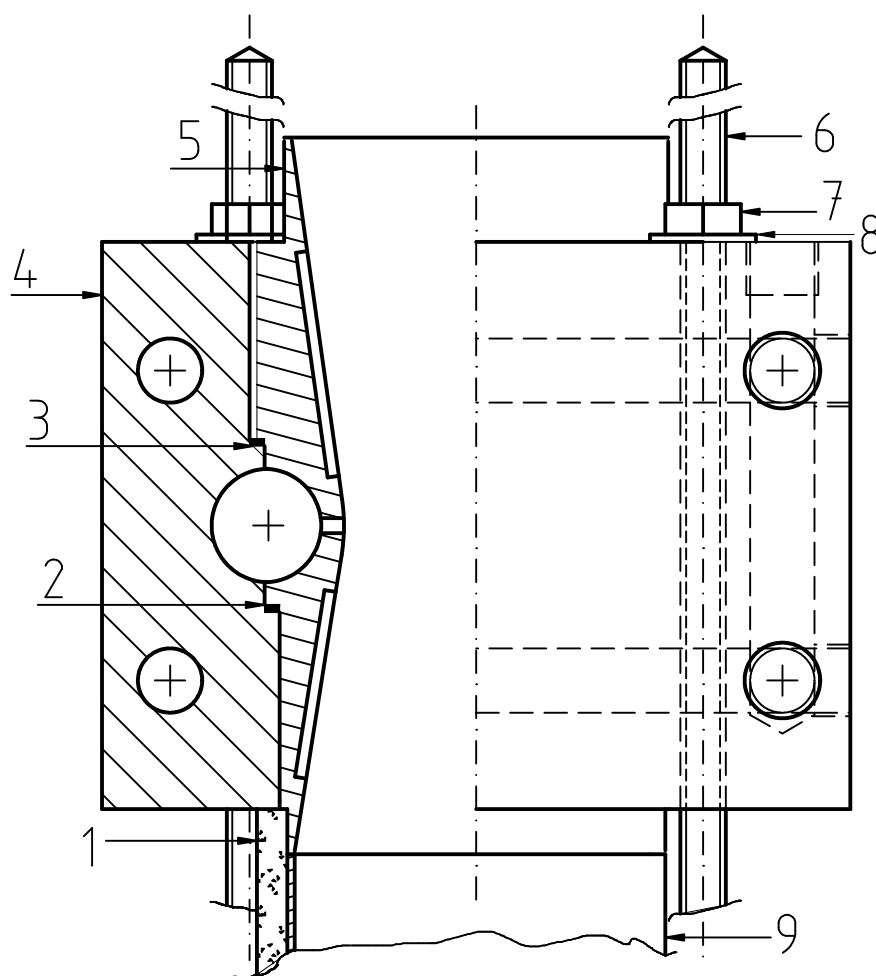


Main unit: milimeter



EESC-USP PVEE Project		
Steam generator float (group)		
Author: Francisco José Alves		
Date:sep 26, 2006	Sheet:1/1	Scale: 1:2

No.	Qt.	Component	Material
1	1	Hose 2"X30mm	rubber
2	1	O'ring 2137	rubber
3	1	O'ring 2139	rubber
4	1	Throat block	brass
5	1	Throat	brass
6	4	Threaded bar M6	carbon steel
7	8	Nut M6	carbon steel
8	4	Arruela M6	carbon steel
9	1	Throttle body	?



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Venturi (group)

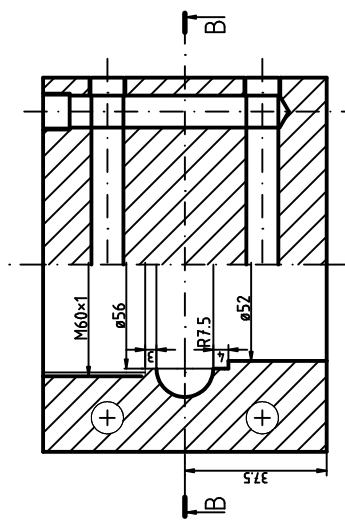
Author: Francisco José Alves

Date: jan 19, 2006

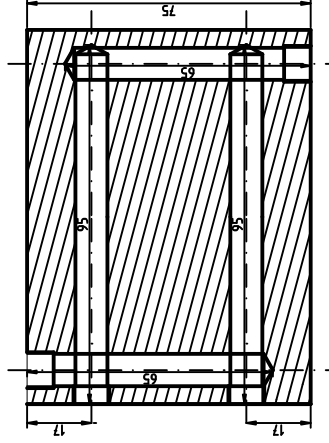
Sheet:1/3

Scale: 1:1

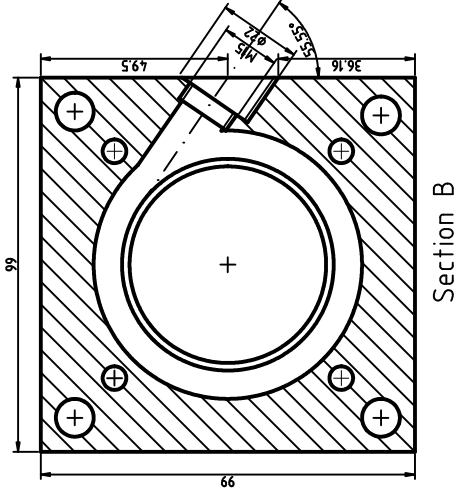
No.	Qt.	Component	Material
1	1	Throat block	brass



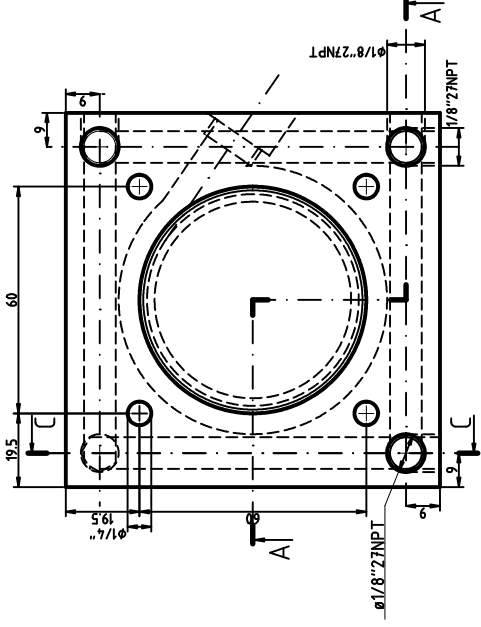
Section A



Section C



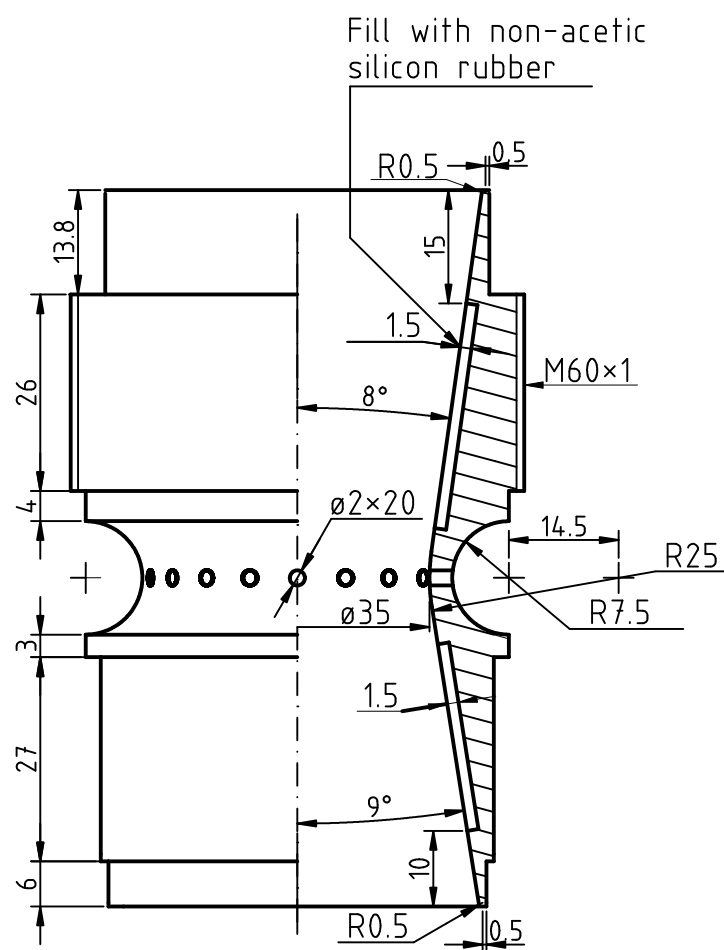
Section B



EESC-USP PVEE Project	Main unit: millimeter
Venturi (Component)	
Author: Francisco José Alves	
Date: jan 19, 2006	Sheet: 2/3 Scale: 1:2



No.	Qt.	Component	Material
1	1	Throat	brass



Main unit: milimeter



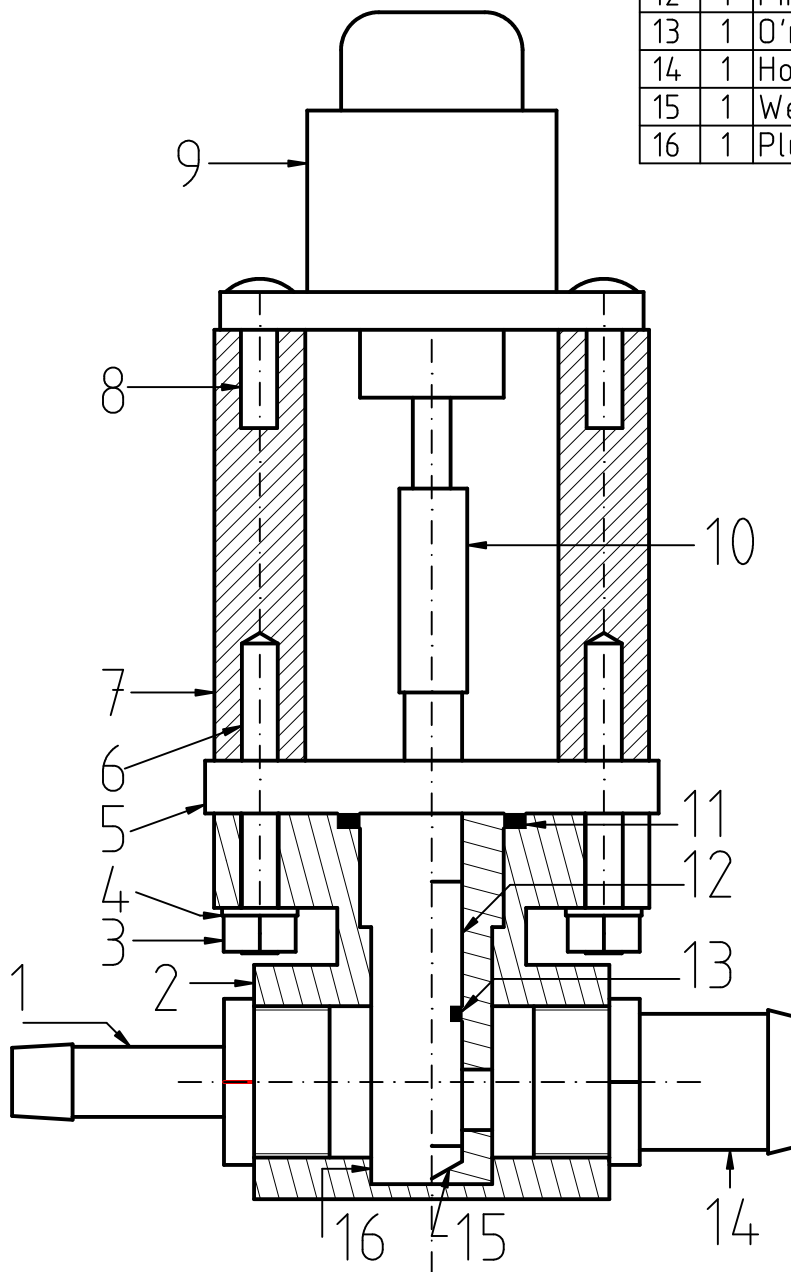
EESC-USP PVEE Project

Venturi (Component)

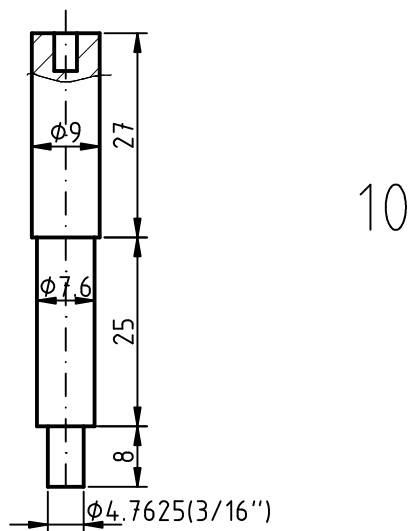
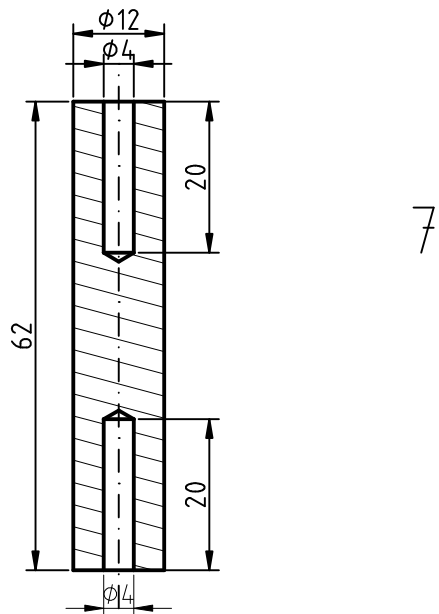
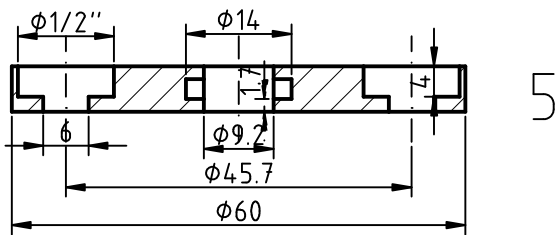
Author: Francisco José Alves

Date: jan 19, 2006 | Sheet: 3/3 | Scale: 1:1

No.	Qt.	Component	Material
1	1	Hose conn. 3/8"	brass
2	1	Case	aluminum
3	2	Nut 3/16"	carbon steel
4	2	Arruela 3/16"	carbon steel
5	1	Separator disc	aluminum
6	2	Thr bar 3/16"x41	carbon steel
7	2	Side bar	nylon
8	2	Bolt 3/16"x20mm	carbon steel
9	1	Motor de passo	several
10	1	Central bar	PTFE
11	1	O'ring 2116	rubber
12	1	Pin	brass
13	1	O'ring 2008	rubber
14	1	Hose conn. 15mm	brass
15	1	Wedge	PTFE
16	1	Plug	brass



No.	Qt.	Component	Material
5	1	Separator disc	aluminum
7	2	Side bar	nylon
10	1	Central bar	PTFE

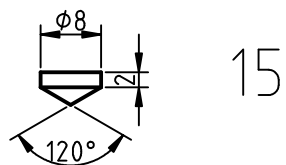
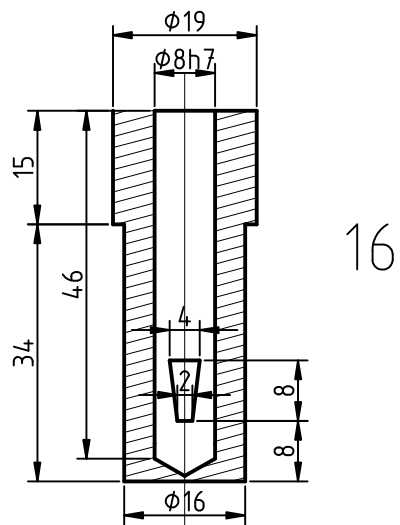
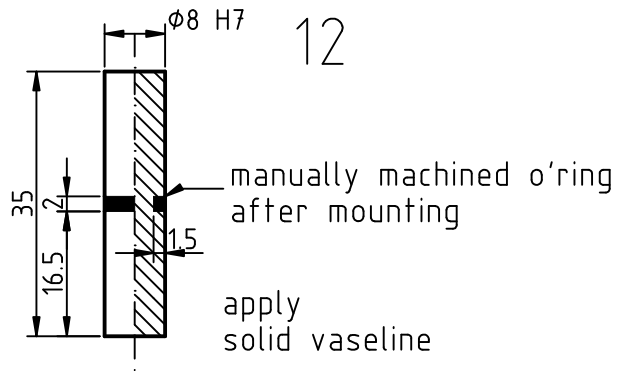


Main unit: milimeter



EESC-USP PVEE Project		
Step motor adaptations (components)		
Author: Francisco José Alves		
Date:dec 09, 2006	Sheet: 2/3	Scale: 1:1

No.	Qt.	Component	Material
12	1	Pin	brass
13	1	O'ring 2008	rubber
15	1	Wedge	PTFE
16	1	Plug	brass



Main unit: milimeter

