11th Brazilian Congress of Thermal Sciences and Engineering – ENCIT 2006
Brazilian Society Of Mechanical Sciences and Engineering – ABCM, Curitiba, Brazil, Dec. 5-8, 2006

Paper CIT06-0841

EXERGOECONOMIC COMPARISON BETWEEN SPARK-IGNITION ENGINE AND COMPRESSION-IGNITION ENGINE

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Abstract. *The present work establishes a comparison between spark-ignition engine and compression-ignition engine in power rating from 50 to 500 kW. A numeric simulation in EES platform was implemented to calculate the thermodynamic and exergoeconomic parameters, considering the complete combustion with air-theoretical. The exergoeconomic analysis was based on the fuel-product methodology, which uses complementary equations necessary to calculate the exergetic cost of each flow. The expected results were very significant, because, despite of being a theoretical simulation it has used monetary values currently practiced by the market for the equipment and fuel prices.*

Keywords. *exergoeconomic, thermoeconomic, exergy, internal combustion engine.*

1. Introduction

The theoretical approach of thermal systems – most of the time – are always based on conservation laws of mass and energy, and the determination of the efficiency coefficient is given by the relation between the work produced and the spent energy. This methodology is well known as Energetic Analysis or Thermodynamics First Law, based on the analysis of the quantitative effect, without considering the qualitative effect, the energy or temperature levels in which the process occur. Internal combustion engines are also submitted to this analysis.

Internal combustion engines are equipment largely used by industrial, automotive and transportation sectors as well as power generator devices. This multiple application gives the equipment a significant commercial importance, since invariably it is responsible for a large amount of the investment where this machine is inserted.

The thermal efficiency of internal combustion engines, according to the Otto Cycle, is basically function of the compress ratio and this efficiency increases according to the increase of this relation; since the engine – according to the Diesel Cycle – shows the same performance, the comparison between them is inevitable, and in fact, the diesel engine operates with a compress ratio higher than the spark-ignition engine and - this way – shows a higher thermal profit. The reason for this fact is that in the spark-ignition engine, the compression is air-fueled and the detonation (or auto-ignition) becomes a restricting factor for Otto Cycle engines. The comparison established in the present work is an exergoeconomic one.

The exergoeconomic (or thermoeconomic) methodology was created in order to foment a thermal and economical analysis for thermal systems. These analysis aims to ratify the technical and economical viability of equipments and industrial plants, which have always produced reliable results. This new approach, also based on conservation concepts, enlarge the energetic analysis including the use of the Second Law and concepts of irreversibility and exergy.

2. Methodology

The exergoeconomic analysis have as a meaning objective, among others, determine the exergetics and monetary costs of all system components; allowing the knowledge and the comprehension of the forming process of these costs; promoting the optimization not only of the specific variables of each system component, but of the whole system.

This detailed analysis was obtained with the contribution of the Thermodynamics Second Law in conjunction with exergetic analysis, in which, according to Tsatsaronis (1993), it would permit a better measurement to evaluate the magnitude of lost energy in relation to the amount of supplied energy under the form of energetic resource; it would also permit a better measurement of quality or loss from a thermodynamic point of view, thus becoming a good variable to define the reasonable efficiency for the energetic system.

2.1. Exergoeconomic analysis

Tsatsaronis *et al.* (1994), had synthesized this analysis based on the *Fuel – Product* concept, proposing a systemic approach in each component, with the purpose to generate auxiliary equations needed to calculate the exergetic costs of each flow. He had called this methodology of Exergoeconomic Analysis, which comprehends the following steps:

- Make an exergetic balance in each system component;
- Make an economic analysis of the subsystems;
- Obtain the costs balance of exergy flows in each component; and
- Calculate the parameters that would permit the analysis of the processes associated to each component.

The economic rating of the thermodynamic flows that perform one cycle, will be set up for the operational conditions later defined, always focusing the utilization of the available exergy from burning process of octane (C_8H_{18}) and dodecane ($C_{12}H_{26}$), respectively known as gasoline and diesel oil. The exergoeconomic method combines the exergetic and economic analysis, and was applied to Otto and Diesel Standard Cycles to reveal which one is thermoeconomically more efficient.

2.2. Exergetic Cost

The exergy balance gives the destruction value of exergy in each component of the system and this destruction is equal to the difference between incoming and out-coming exergy from the volume control. This happens because, in a real process, there will be always destruction and loss resulting in a bigger exergy in the entrance of the process, in relation to the product exergy. The exergetic cost of a product is therefore composed by the resources exergy, the external loss exergy and the irreversibility. As well shown in Figure (1), the exergetic cost of the product will always be bigger than the resource.

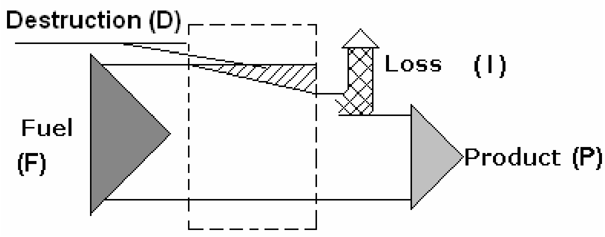

Figure 1. Resource (F), Product (P), Destruction (D) and Loss (I).

The exergetic balance in each component can be expressed as:

$$B_P = B_F - I - B_D \tag{1}$$

Starting from the concept of exergetic or reasonable efficiency, that is the relation between the product exergetic variation and the resource exergetic variation needed to the process, we can define the unitary exergetic cost as being the inverse relation, such as:

$$k = \frac{Exergy\ of\ \mathrm{Re}source}{Exergy\ of\ \mathrm{Pr}oduct} = \frac{B_{resource}}{B_{product}} = 1 + \frac{Loss(I) + Destruction(D)}{\mathrm{Pr}oduct(P)} = \frac{\mathrm{B}_i^*}{\mathrm{B}_i} \tag{2}$$

where ***k*** is the unitary exergetic cost , B_i^* is the exergy expressed in (kW) necessary to conceive the desired product, and B_i is the minimum quantity of exergy expressed in (kW) corresponding to the ideal process for the conception of the product. Obviously, ***k*** will be greater than or equal to the unity, and in an externally reversible process this factor will become equal to one.

Valero *et al.* (1986) have formulated an endowment proceeding of exergetic costs, based only in thermodynamics precepts, such as:

- The exergetic cost of a flow (B^*), resource (F*), or product (P^*) is the real quantity of exergy needed to produce it;
- A detailed analysis of the global nature of the process and of the function of each subsystem in progressive formation of the final products, is the only requirement needed to solve the endowment problem of exergetic costs;
- The exergetic costs in the entrance of an equipment or component of the system should be rated with the flow that outcome from it.

Based on these postulates, a collection of proposition has been created and the systematic application on the equipments will permit us value the exergetic costs of the flows. These propositions will be set up in a general way, and afterwards will be applied in the systems to be considered.

- Proposition 1 – The exergetic cost is a conservative property

$$\sum_{entrance} B_i^* - \sum_{exit} B_j^* = 0 \tag{4}$$

- Proposition 2 – for a system or control of volume with more than one energetic resource, the exit unitary exergetic costs must be equal to the entrance ones (resource rules)

For a general system example as shown on Figure (2), we have:

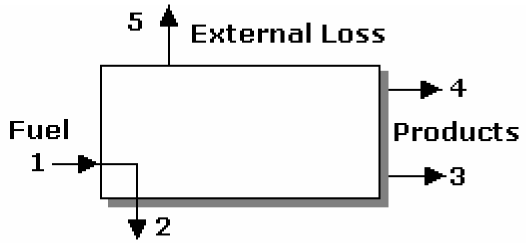

Figure 2. General System Example (Torres, 1999)

$$\frac{B_1^*}{B_1} = \frac{B_2^*}{B_2} \tag{5}$$

- Proposition 3 – if a system has a product formed by various flows, the exergetic cost will be the same for each one of them (product rule). In the Figure (2) example we have:

$$\frac{B_3^*}{B_3} = \frac{B_4^*}{B_4} \tag{6}$$

- Proposition 4 – in the absence of value of an external loss flow, we shall admit a null exergetic cost. In this example we have:

$$\frac{B_5^*}{B_5} = 0 \tag{7}$$

- Proposition 5 – in the absence of external value, the exergetic cost of the entrance flows in the system is equal to its exergy. In this example we have:

$$B_1^* = B_1 \tag{8}$$

3. ENERGETIC AND EXERGOECONOMIC FORMULATION OF ENGINES

3.1. Energetic Analysis of Internal Combustion Engines

The energetic analysis of internal combustion engines involve chemical reaction, which include hydro-carbonated fuel combustion, and since we are mentioning a power generator device, the thermodynamic analysis of reagent systems are basically an extension of Thermodynamic First and Second Laws. Schematically, a surface of control involving our objective of study is shown on Figure (3), characterizing the air and fuel entrance flows, and the exit of exhaustion gases, heat and – mainly – work.

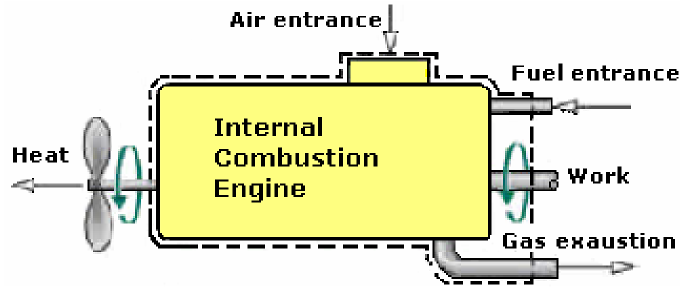

Figure 3. Surface of control involving an internal combustion engine

During the combustion process, the mass of each element remains the same. This way, describing a chemical reaction basically implies on mass conservation of each composing element of the fuel and the oxidant. This way, we should consider the octane combustion with quantity of air theoretical, which is the minimum amount of air needed to supply enough oxygen for a complete combustion of all carbon and hydrogen, as follows:

$$C_8H_{18} + a\,(O_2 + 3{,}76\,N) \longrightarrow b\,CO_2 + c\,H_2O + d\,N_2 \qquad (9)$$

Where *a, b, c* and *d* represent the number of moles of oxygen, dioxide of carbon, water and nitrogen respectively. In the first member we consider that 3,76 moles of nitrogen join each mole of oxygen. Applying the principle of mass conservation to carbon, hydrogen, oxygen and nitrogen respectively, we can obtain a system of four equations and four unknown members. Solving this system, the balanced chemical equation becomes as follows:

$$C_8H_{18} + 12{,}5(O_2 + 3{,}76\,N) \longrightarrow 8\,CO_2 + 9\,H_2O + 47\,N_2 \qquad (9.a)$$

In the same way, for the dodecane combustion we should have:

$$C_{12}H_{26} + 18{,}5(O_2 + 3{,}76N) \longrightarrow 12\,CO_2 + 13\,H_2O + 69{,}56\,N_2 \qquad (10)$$

The energetic balance for reagent systems in permanent regime can be evaluated admitting the hypothesis that air-fuel combination and combustion products can be considered ideal gases, besides ignoring the effects of kinetic and potential energy, thus:

$$\frac{\dot{Q}_{vc}}{\dot{n}_{vc}} - \frac{\dot{W}_{vc}}{\dot{n}_{vc}} = \sum_P n_s(\bar{h}^{o}_{f} + \Delta\bar{h})_s - \sum_R n_e(\bar{h}^{o}_{f} + \Delta\bar{h})_e \qquad (11)$$

With these conceptions, the balance of mass and energy for a volume of control with various entrances and exits becomes evident by the Eq. (11), where $\dot{Q}_{vc}$ and $\dot{W}_{vc}$ are respectively the heat flow and the work flow (power) getting out of the volume of control; $\dot{n}_{vc}$ represents the molar fuel flow; the coefficient $\boldsymbol{n_s}$ e $\boldsymbol{n_e}$ correspond to the respective coefficient of Eq. (9.a) and Eq. (10) of the respective reaction, supplying the reagent moles and the products per mole of fuel; $\bar{h}^{0}_{f}$ is the enthalpy of formation of products and reagent and $\Delta\bar{h}$ symbolizes the difference of molar enthalpy of reagents and products in the entrance and exit of the volume of control.

The fuel consumption denoted by $\dot{m}_f$ is well defined as the relation between the fuel masses outflow and its time unit. The mostly used parameter to relate the fuel flow to the power *(Pot)* is the fuel specific consumption (*sfc*) given by Eq. (12).

$$sfc = \frac{\dot{m}_f}{Pot} \qquad (12)$$

For spark-ignition combustion engines, the fuel specific consumption according to Heyhood (1988) assumes values of 75 µg/J = 270 g/kWh. For compression-ignition engines this parameter assumes values of 55 µg/J = 200 g/kWh.

3.2. Exergetic Analysis of Internal Combustion Engines

Exergy can be defined as the greater theoretical job possible to be obtained in relation to the state of reference to a temperature T_o and pressure P_o. The system exergy can be obtained by:

$$\mathrm{B} = (\mathrm{E} - \mathrm{U_o}) + \mathrm{P_o}(V - V_o) - \mathrm{T_o}(S - S_o) \tag{13}$$

Where *E* (the sum of internal energy, potential and kinetic), *V* and *S* denote the energy, the volume and the system entropy – respectively, while U_o, V_o, and S_o are values of the same properties if the system were in the state of reference. There are several ways to calculate the exergy with a system of equation that must be specifically chosen for each case. We have, for example, the exergy due to the heat exchange that is defined by:

$$\mathrm{B_{heat}} = \left(1 - \frac{\mathrm{T_o}}{\mathrm{T}}\right).\dot{Q}_{vc} \tag{14}$$

The chemical exergy is defined according to the type of process and chemical reactions of each process. In this case, the chemical exergy of the fuel in liquid form has been defined according to Szargut *et al.* (1988) by the product of coefficient γ and the lower heat power, as follows:

$$\mathrm{B_{ch}} = \gamma.\mathrm{PCI} \tag{15}$$

For hydro-carbonated liquid fuel, the coefficient γ can be determined by Eq. (16):

$$\gamma = 1{,}0506 + 0{,}0144.\frac{\mathrm{H}}{\mathrm{C}} \tag{16}$$

where H/C is the atomic relation between its elements.

Moran and Shapiro (2002), suggested for the calculation of chemical exergy of combustion gases of a hydro-carbonate type of C_mH_n an equation given in terms of Gibbs Function of the respective substance products of combustion, as follows:

$$\mathrm{B_{ch}} = \left[\bar{g}_{fuel} + \left(m + \frac{n}{4}\right).\bar{g}_{O_2} - m.\bar{g}_{CO_2} - \frac{n}{2}.\bar{g}_{H_2O(gás)}\right](T_O, P_O) + \bar{R}.T_O.\ln\left[\frac{(\mathrm{y}_{O_2})^{m+n/4}}{(\mathrm{y}_{CO_2})^{m}.(\mathrm{y}_{H_2O})^{n/2}}\right] \tag{17}$$

where $\bar{g}_{fuel}$, $\bar{g}_{O_2}$, $\bar{g}_{CO_2}$ and $\bar{g}_{H_2O}$ are Gibbs functions for fuel, oxygen, dioxide of carbon and water steam, respectively. In the same way y_{O_2}, y_{CO_2}, y_{H_2O} are molar fractions of oxygen, dioxide of carbon and water steam, respectively.

Defined the equations to determine the exergy of each flow, we started from Eq. (4) and taking Figure (3) as volume of control we can establish the exergetic balance for the internal combustion engine and apply particular equations to the case in study, obtained this way for the exergetic cost (B*) the following equation:

$$\mathrm{B^*_{fuel}} + \mathrm{B^*_{air}} - \mathrm{B^*_{power}} - \mathrm{B^*_{heat}} - \mathrm{B^*_{gases}} = 0 \tag{18}$$

With the support of the complementary equations, previously defined by the propositions of cost endowment, we can establish a system of equations in order to define the exergetic cost for each flow. From the proposition 3 (products rule) we can assert that the unitary cost of heat is equal to the unitary cost of power, or:

$$\frac{\mathrm{B^*_{heat}}}{\mathrm{B_{heat}}} = \frac{\mathrm{B^*_{power}}}{\mathrm{B_{power}}} \tag{19}$$

From proposition 4 we can attribute a null exergetic cost for the gas exhaustion flow and for the air flow:

$$\frac{B^*_{gases}}{B_{gases}} = \frac{B^*_{air}}{B_{air}} = 0 \tag{20}$$

From proposition 5 we can attribute exergetic cost equal to exergy itself of the fuel

$$B^*_{fuel} = B_{fuel} \tag{21}$$

3.3. Analysis of Monetary Parameters

The methodology to value the monetary costs is an application of a cost balance to a subsystem or equipment as shown on Fig. (4).

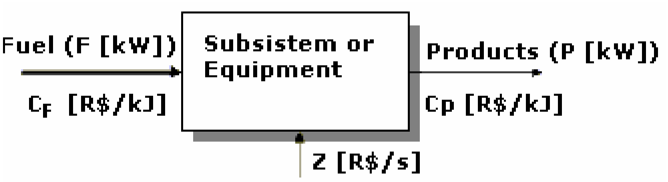

Figure 4. Balance of Monetary Costs

The balance shown on Fig. (4) can be mathematically represented as follows:

$$C_P.B_P = C_F.B_F + Z \tag{22}$$

Where C_F and C_p represent the costs in monetary unit per exergy unit for the resource and the product respectively; in the same way B_f and B_p represent the amount of exergy for the resource and the product, and Z is the invested capital. In the particular case of a plant in operation and already paid, we can take Z as a null value, although that’s not the case, because the capital invested in each internal combustion engine is an important economic parameter for comparison. To determine Z, we shall consider:

$$Z_{(i)} = \frac{3600 * \left(\frac{A}{P}\right)}{t_{op}} * F_i \tag{23}$$

Where t_{op} represents the useful time life (in seconds); F_i represents the investment for each equipment or subsystem; (A/P) represents the capital recovering factor and will be calculated by Eq. (24), considering I the interest rate (varying from zero to 1); and N represents the reimbursement period (in years)

$$\left(\frac{A}{P}\right) = \frac{I * (1+I)^N}{(1+I)^N - 1} \tag{24}$$

4. RESULTS AND DISCUSSION

These results were obtained from the numeric simulation in EES platform, attempting to the thermodynamic modeling established on the third section, where the parameters of entrance were established for power rating from 50 to 500 kW. Other parameters of entrance were obtained from literature as shown on Table (1); the monetary values of the equipment (diesel and gasoline engines) and the fuel were researched and obtained from values currently practiced by market.

Table 1. Entrance data for numeric simulation

Entrance Parameter	**Gasoline engine**	**Diesel engine**
Power [kW]	50 to 500	50 to 500
Air Temperature [K]	298	298
Fuel Temperature [K]	298	298
Exhaustion Gas Temperature [K]	1800	1800
Fuel specific consumption [kg/kWs]	0.000075	0.000055
Initial Investment [R\$]	45,000.00	60,000.00
Fuel Price [R\$/m^3]	2,500.00	1,750.00
Lower Heat Power [kJ/kg]	44425	44109
Useful Life Time - (10 years) [s]	315360000	315360000

Table 2 summarizes the thermodynamic properties at various points at the power operating conditions. The exergy analysis of the system and its components are carried out using fuel-product relationships and Eqs. (13) – (17).

Table 2 – Thermodynamic properties (exergy and exergetic cost) at octane and dodecane fuel.

Power [kW]	$B_{inC8H18}$ [kW]	$B_{inC12H26}$ [kW]	$B_{outC8H18}$ [kW]	$B_{outC12H26}$ [kW]	$B_{heatC8H18}$ [kW]	$B_{heatC12H26}$ [kW]	$B^*_{heatC8H18}$ [kW]	$B^*_{heatC12H26}$ [kW]
50	178,8	130	2841	1930	0,6242	0,2695	2,204	0,697
100	266,6	193,9	4237	2878	1,294	0,5125	3,405	0,9886
150	399,9	290,8	6355	4317	2,911	1,153	7,613	2,219
200	533,2	387,8	8473	5757	5,175	2,05	13,45	3,934
250	666,5	484,7	10592	7196	8,086	3,203	20,88	6,132
300	799,8	581,6	12710	8635	11,64	4,613	29,88	8,807
350	933,1	678,6	14828	10074	15,85	6,278	40,42	11,96
400	1066	775,5	16947	11513	20,7	8,2	52,47	15,58
450	1200	872,5	19065	12952	26,2	10,38	66	19,67
500	1333	969,4	21183	14391	32,35	12,81	80,99	24,22

Table 3 shows results obtained from the exergoeconomic analysis of the system for octane and dodecane fuel at engine correspondent. The exergoeconomic analysis of the system and its components are carried out using fuel-product relationships and Eqs. (18) – (24).

Table 3 – Results of exergoeconomic analysis for operating conditions.

Power [kW]	$B^*_{powerC8H18}$ [kW]	$B^*_{powerC12H26}$ [kW]	$C_{Fuel-C8H18}$ [R\$/kJ]x10^6	$C_{Fuel-C12H26}$ [R\$/kJ]x10^6	$C_{powerC8H18}$ [R\$/kJ]	$C_{powerC12H26}$ [R\$/kJ]	$Pr_{powerC8H18}$ [R\$/s]	$Pr_{powerC12H26}$ [R\$/s]
50	176,6	129,3	52,45	37,02	0,00843	0,01109	0,4215	0,5543
100	263,2	192,9	52,45	37,02	0,004261	0,005567	0,843	1,109
150	392,3	288,6	52,45	37,02	0,002887	0,003735	1,264	1,663
200	519,7	383,8	52,45	37,02	0,0022	0,002819	1,686	2,217
250	645,6	478,6	52,45	37,02	0,001788	0,00227	2,107	2,771
300	769,9	572,8	52,45	37,02	0,001514	0,001903	2,529	3,326
350	892,7	666,6	52,45	37,02	0,001317	0,001642	2,95	3,88
400	1014	759,9	52,45	37,02	0,00117	0,001445	3,372	4,434
450	1134	852,8	52,45	37,02	0,001056	0,001293	3,793	4,989
500	1252	945,2	52,45	37,02	0,000964	0,001171	4,215	5,543

The entrance exergy in the volume of control of each fuel is obtained from Eq. (15). The fuel mass flow will always be a function of power and specific consumption as well defined by Eq. (12). Figure (5) shows that the exergy of each fuel is a linear function of the power.

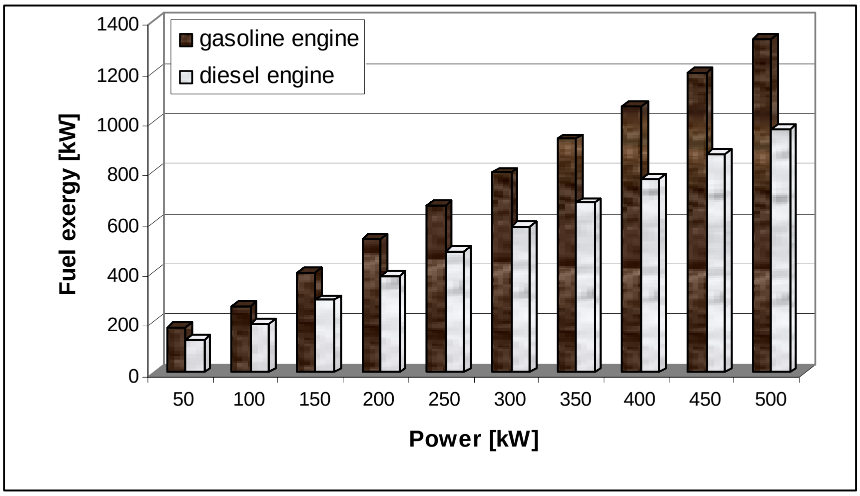

Figure 5. Fuel Exergy in function of the Power

The exergy produced by the heat demonstrated in Eq. (14) associated to the solution demonstrated in Eq. (11) gives us the amount of exergy produced by heat. Fig. (6) demonstrates the variation of heat exergy in function of the power.

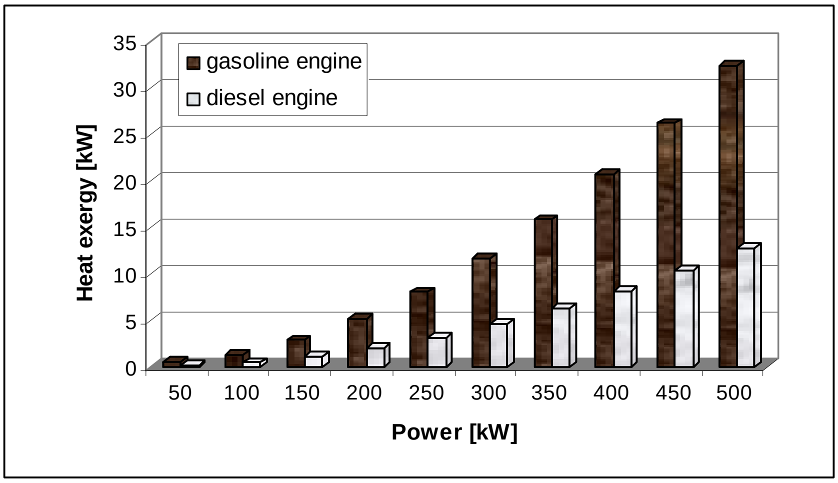

Figure 6. Heat Exergy in function of the Power.

The exit exergy of exhaustion gases in function of the power is well demonstrated in Fig. (7).

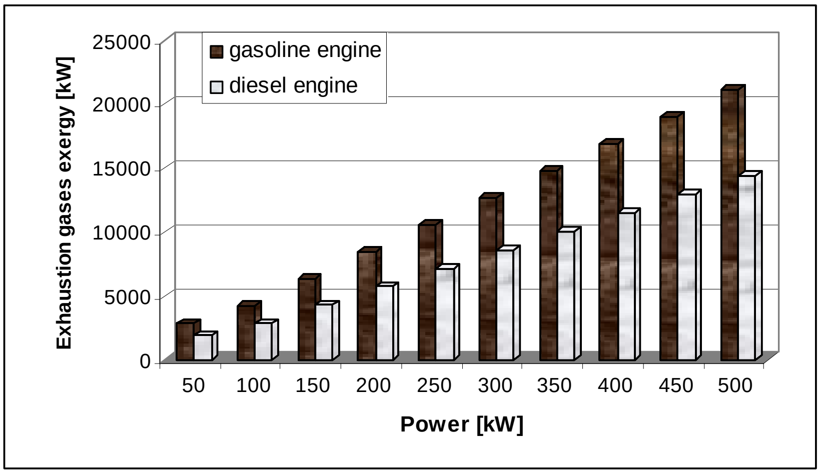

Figure 7. Gas Exhaustion Exergy in function of the Power.

The exergoeconomic parameters were compared, based on the exergetic costs (B_i^*) of the fuel as shown in Fig. (8).

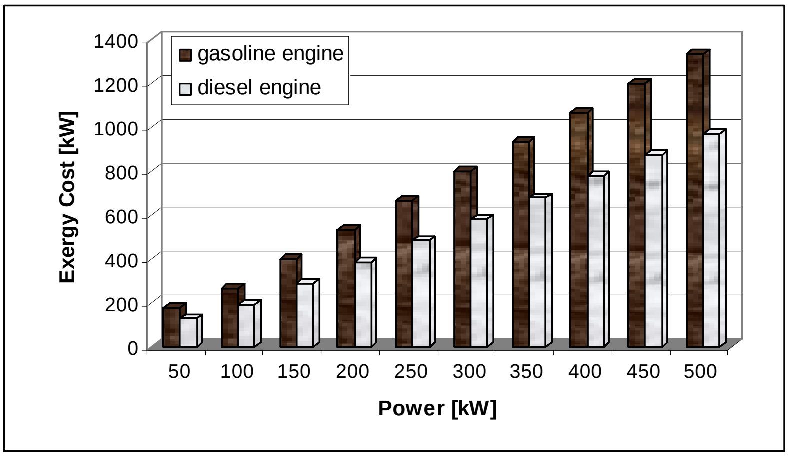

Figure 8. Exergetic Cost of Fuel in function of the Power.

The monetary cost per energy is obtained from the cost balance shown in Fig. (4), once the resource is equal to the fuel and the product is equal to the power, as already defined by Eq. (22). Fig. (9) demonstrates that diesel engines are more expensive than gasoline ones, although this cost tends to decrease with the raise of the power.

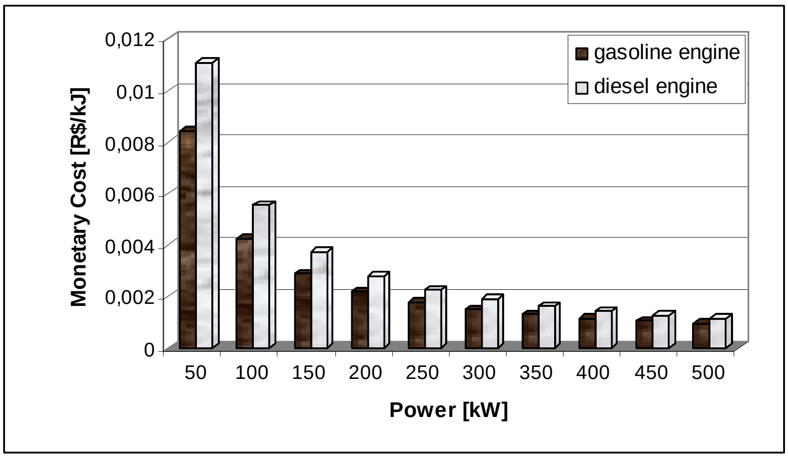

Figure 9. Monetary Cost of the Power.

The monetary cost per time unit is obtained dividing the energy monetary cost by the power itself. Fig. (10) demonstrates that diesel engines revealed more expensive performances.

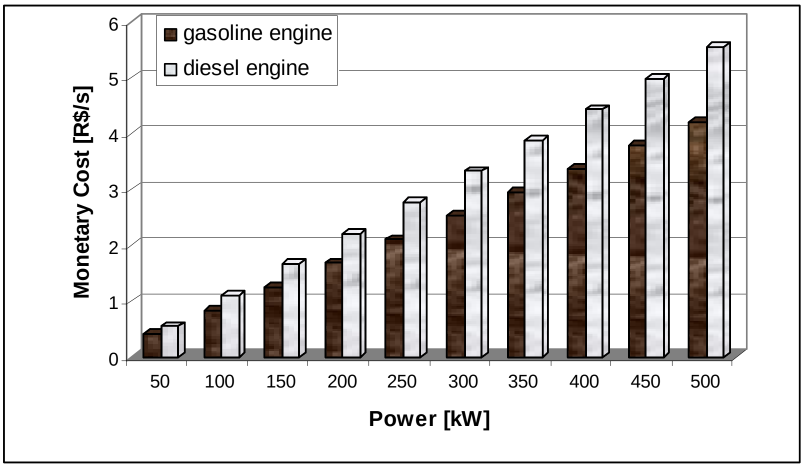

Figure 10. Power Monetary Costs related to Time Unit.

5. CONCLUSION

The methodology applied in the present work, which has connected the systemic approach of the concept of *"Fuel – Product"* of Tsatsaronis (1994) to the rules of cost endowment proposed by Valero *et al.* (1986), has provided the viability of identification of greater costs – both exergetic and monetary – of the flows going through the control surface of internal combustion engines supplied by gasoline and diesel oil.

The simulation has demonstrated that, although diesel engines have revealed a higher thermal performance than gasoline engines, the flows of exergetic and monetary costs of the power in diesel engines are more expensive than in gasoline ones; that is due to the octane exergy be greater and initial capital of investment – in gasoline engines – be lower than in diesel engines.

6. REFERENCES

Heywood, J.B., 1988, "Internal Combustion Engine Fundamentals", Ed. McGraw-Hill Book Company, São Paulo, Brazil, 930p.

Kotas, T.J., 1995, "The Exergy Method of Thermal Plant Analysis", Ed. Krieger Publishing Company, Malabar, Florida, U.S.A, 328 p.

Moran, M.J. and Shapiro, H.N., 2002, "Princípios de Termodinâmica para Engenheiros", 4^a. Ed., LTC Editora, Rio de Janeiro, Brasil, 681p.

Szargut, J., Morris, D.R., Steward, F.R., 1988, "Exergy Analysis of Thermal, Chemical, and Metallurgical Processes", Hemisphere Publishing Corporation, New York, U.S.A., 330p.

Torres, E. A., 1999, "Avaliação Exergética e Termoeconômica de um Sistema de Cogeração de um Pólo Petroquímico". FEM/UNICAMP, Tese Doutorado, Campinas, SP, Brasil.

Tsatsaronis, G., (quest editor) 1994, "Invited Papers on Exergoeconomic, Energy Special Issue", V. 19, nº 3.

Tsatsaronis, G., (quest editor) 1993, "Thermoeconomic Analysis and Optimization of Energy Systems", Evergy Combust, V. 19, pp 227-257.

Valero, A., Losano, M. A., Muñoz, "General Theory of Exergy Saving". ASME, WAM, 1986, AES, V. 2-3, pp. 1-21.