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TEZĂ DOCTORAT

Cercetări privind procesul de ardere la motoarele diesel navale în scopul diminuării emisiilor poluante

Research on the combustion process of naval diesel engines in order to reduce pollutant emissions

REZUMAT în limba engleză

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SUMMARY OF THE DOCTORAL THESIS

PHD THESIS SUMMARY

This paper, structured in 6 chapters, contains 241 pages and includes 134 figures, 8 tables and 219 bibliographic references.

Keywords: **internal combustion naval engines, 1997 Marpol 73/78 Amending Protocol, NOx emissions, combustion mechanisms, combustion process modeling, pollutant emissions formation, fuel injection process, fuel injection advance, nitrogen oxide emission reduction.**

CHAPTER I: OBJECTIVES AND PURPOSE OF THE WORK

The modeling of process in internal combustion engines has been a permanent concern of specialists in the field. The complexity of the phenomena and the strong interdependence between them make the approach particularly difficult.

The aim of the paper is to develop a simulation program to estimate the emissions of nitrogen oxides produced by compression ignition engines. The idea is based on an analysis of the methods of implementing the provisions of Annex 6 to the 1997 Protocol on the amendment of MARPOL 73/78, concerning the limitation of polluting emissions from ships, in this case nitrogen oxides. The application of the provisions of the Protocol required the signatory countries, from 2010, to put in place measures to reduce nitrogen oxide emissions, especially in coastal areas, the most affected being, in the first instance, port technical vessels. In this context, the development of a simulation program is a very useful tool. Emission estimation and theoretical verification, in the first phase, of the solutions applicable to in-service engines could greatly reduce research and production costs, given that there are a variety of engines on board ships and measurements in operation are very difficult.

The engine chosen for the investigations is D110 (naval landed T650 engine), frequently encountered in the port-river maritime field (as Diesel-generator), an area for which a stricter application of the regulations of the Protocol is foreseen.

CHAPTER II: MODELING OF COMBUSTION AND POLLUTING EMISSIONS IN DIESEL ENGINES

The combustion process is by far the most important and complex process that takes place in engines, its importance being derived from the supply of energy flow used in the engine, respectively is the source of all pollutant emissions, engine efficiency is directly influenced by it. The mechanisms of combustion are particularly complex and are not fully known even today, the most difficult problem being the mechanisms of mixture formation and the chemistry of the combustion process.

For modeling the phenomena that characterize combustion and pollutant emissions in naval engines are representative:

▪ **0-dimensional thermodynamic models** – the oldest used outside the ideal cycles, but can not be considered as models because the applicability is very low, being used for teaching purposes or for rough estimates necessary for general calibration calculations. It is grouped into:

➪ *Monozonal models* (the simplest) – addresses the problem of combustion only thermodynamically, considering as a control volume the entire combustion chamber; are useful for studying the combustion process and are based on the characteristic of heat release, when known;

Plurizonal models – eliminates the basic hypothesis of monozonal (unrealistic), which considers the motor fluid homogeneous. The jet of liquid fuel, introduced in the combustion chamber, is sprayed in drops, wich spread inside and form area of various concentrations, wich have different evolutions and determine specific features of the combustion process. The characterstics of the mixture formation process lead to the conclusion that the interior of the combustion chamber can be structured as a multizonal domain.

▪ **Phenomenological models** – have the highest applicability in practice, being superior performance. The limitation is given by the connection with the practical conditions for determining certain relations used and the phenomenological approach, wich simplifies the complex system of conditions improsed by the purely theoretical approach, implicitly the systems of equations that need to be solved. Unlike thermodynamics, they introduce the influence of organized and turbulent movements on the combustion process in the cylinder. We have two categories of phenomenological models: 1. developed from multizonal models in jet layers; 2. developed from multizon models with drop packs.

▪ **Multidimensional models** – mainly take into account the characteristics of the flow in the cylinder, wich strongly influences the development of processes in diesel engines; treats thermogasodynamically the phenomena in the cylinder, introducing the influences of the flow in the cylinder on the thermal and physico-chemical processes that take place during combustion. Most are based on the general equation of the jet, wich takes into account all aspects: fragmentation, collisions, coalescence and turbulence, taking into account the thermo-physico-chemical inhomogeneity in the cylinder. It expresses the variations of the flow field, of temperature, composition, pressure and turbulence, analyzing, in the case of biphasic flows, processes in conditions out of equilibrium.

CHAPTER III: EXPERIMENTAL MEASUREMENTS

The experimental determinations were performed on the test stand in the Multifunctional Laboratory at the Nautical Base of the Maritime University of Constanța, equipped with a Diesel engine with omega combustion chamber and ecological injection pump that equips T650 universal tractors and which are also used in the field naval.

The aims pursued are subsequent: a. to determine the evolution of the level of pollutant emissions, especially those of nitrogen oxides, depending on the operating regime and the influence of constructive and regulatory factors on them; b. creating a database for calibrating a program that can model the combustion process and pollutant emissions; c. establishing methods to enable engines in operation in the maritime area to harmonize with the requirements of the 1997 Protocol on amending MARPOL 73/78, on the polluting emissions.

Fig. 1 Engine test stand

The test stand shall be drawn up in accordance with the recommendations of ISO/TC 70/SC8 of 21 september 1995, the 1997 Protocol on amending MARPOL 73/78 and Directive 97/68/EC of 16 december 1997. The stand is intended for testing engines with a power output below 100 HP, speed below 5000 rpm and maximum torque at shaft below 300 Nm.

The experimental determinations were performed according to the 1997 Protocol, and the preparations concerned the engine and the measuring equipment. The choice of the test cycle was aimed at studying the pollutant emissions, mainly $NO_{X,}$ and the influences of various factors, functional and regulatory, on them. The aim was to accumulate various experimental data, designed to highlight the influence of the most important parameters on the mechanism of formation of pollutant emissions, to calibrate a calculation program capable of modeling engine performance and formation of pollutant emissions (NO_X mainly). They were interesed in measures to reduce emissions (NO_X, being the only limited in the naval field, since 2000). The **cycle in 8 points type C1** was chosen. The data was collected with the acquisition system, the text files were converted to EXCEL databases, after wich they were graphically processed in EXCEL and Matlab. The data on the cylinder pressure and the injection system are the most complex, being read directly without mediation, with a frequency of 10 measurements per degree of rotation of the crankshaft, the others being mediated by the acquisition system during 60 s. The first data set (Table 1) was obtained for the normally equipped engine, for the C1 nozzle cycle at 8 points with an intermediate speed of 1440 rpm, determined according to the recommendations of the Protocol and the actual injection advance of 5.4 °RAC.

NOX [g/h]	CO [g/h]	tHC [g/h]	citit [%]	citit [min-1]	8 Mode C1	NOX*wfi [g/h]	CO*wfi [g/h]	tHC*wfi [g/h]	P*wfi [kW]
380.3798	165.8068	12.50749	100	2400	0.15	57.05698	24.87102	1.876123	7.7445
278.3957	71.75288	13.25397	75	2400	0.15	41.75935	10.76293	1.988095	5.793
231.982	72.67026	13.84014	50	2401	0.15	34.79729	10.90054	2.076021	3.885
124.0397	178.715	22.95071	10	2400	0.1	12.40397	17.8715	2.295071	0.525
87.94658	131.9192	16.81904	100	1440	0.1	6.055186	2.982347	0.769022	0.0195
274.745	19.52551	6.818545	75	1439	0.1	33.70205	11.87159	0.827603	3.53
242.9459	27.47304	9.204387	50	1439	0.1	27.4745	1.952551	0.681854	2.609
40.36791	19.88231	5.12681	0	812	0.15	24.29459	2.747304	0.920439	1.783
					SUM=	237.5439	83.95977	11.43423	25.889
						NOX [g/kWh]	CO [g/kWh]	tHC [g/kWh]	
						9.175477	3.243067	0.441664	
			European Limits			9.2	6.5	1.3	
			1997 Protocol			9.8			

Subsumed to the reduction of nitrogen oxide emissions, the easiest way to do this is to change the injection advance. In order to choose an optimal advance to the injection, the advance characteristics at different speeds and maximum load are raised; the most favorable area is obtained for 3 and 5 °RAC real injection advance adjusted to the nominal speed, with optimum in the area 4-5 °RAC. The pump injection feed was adjusted so that an actual injection feed rate of 4 °RAC was obtained at rated speed and the noxious test was repeated (results in Table 2).

NOX [g/h]	CO [g/h]	tHC [g/h]	citit [%]	citit [min-1]	8 Mode C1	NOX*wfi [g/h]	CO*wfi [g/h]	tHC*wfi [g/h]	P*wfi [kW]
385.6414	186.9515	10.70036	100	2400	0.15	54.6424	27.90323	1.602522	7.557
276.1286	89.31642	12.32377	75	2400	0.15	38.96805	13.3339	1.845137	5.658
203.9593	104.3723	16.94219	50	2401	0.15	28.74279	15.58189	2.535635	3.7635
98.16899	155.4063	27.53781	10	2400	0.1	9.173228	15.46942	2.746364	0.516
297.5112	115.7389	6.677281	100	1440	0.1	28.05851	11.51846	0.666838	3.452
227.6865	31.45395	7.210631	75	1439	0.1	21.36202	3.130896	0.719823	2.606
188.7585	31.10178	8.959457	50	1439	0.1	17.65085	3.095992	0.894022	1.726
30.08974	14.80048	3.28709	0	812	0.15	4.198821	2.210057	0.491648	0
					SUM=	202.7967	92.24385	11.50199	25.279
						NOX [g/kWh]	CO [g/kWh]	tHC [g/kWh]	
						8.022496	3.649103	0.455011	
			European Limits I			9.2	6.5	1.3	
			1997 Protocol			9.8			

The analysis confirmed the choice made, although it can be seen that reducing the advance below 4 °RAC could lead to a decrease in nitrogen emissions; this is not preferable because the engine performance decreases, the power decreases and the consumptions increase, respectively the emissions of unburned hydrocarbons increase.

CHAPTER IV: THE MODEL DEVELOPED AND ITS FRAMEWORK IN CURRENT TRENDS

The developed model is of multidimensional type, solving systems of specific equations: a. compressible turbulent flow; b. chemical reactions (combustion of fuel , respectively noxious emissions); c. the flow and evaporation of jets of liquid particles. The formulation is spatially two-dimensional and allows the plane and axially symmetrical approach of the combustion chamber geometry (facilitates the consideration of the swirl motion around the axis of symmetry, increasing the spatial resolution and partially implementing the tird geometric dimension). The numerical solving scheme is ahead of time, using a finite volume discretization and a partial default algorithm. The dicretization network is adjustable and consists of generalized quadrilaterals whose corners are specified by time-dependent coordinates relative to the lower position of the piston, wich allows the Eulerian or Lagrangean problem to be approached as needed.

Circumscribed to the theoretical foundation, the calculation plane is xOy, the *x*-axis – parallel to the piston radius, *y* – with the axis of symmetry of the cylinder, the equations being reported:

➢ **fluid phase**: the partial density of species *k* is denoted by ρ_k, and the density of the fluid phase, excluding the mass of the fuel droplets, is given by the relation:

$$\rho = \sum_k \rho_k \,. \tag{1}$$

The continuity equation for species *k* is given by the relation:

$$\frac{\partial \rho}{\partial t} + \frac{1}{R} \underset{\sim}{\nabla} \bullet \left(R \rho_k \underset{\sim}{u} \right) = \frac{1}{R} \underset{\sim}{\nabla} \bullet \left[R \rho D \underset{\sim}{\nabla} \left(\frac{\rho_k}{\rho} \right) \right] + \dot{\rho}_k^C + \dot{\rho}_S \, \delta_{k1} , \tag{2}$$

If we add the equation after *k* (the number of species in the existing gas phase) we obtain the *continuity equation*:

$$\frac{\partial \rho}{\partial t} + \frac{1}{R} \underset{\sim}{\nabla} \bullet \left(R \rho \underset{\sim}{u} \right) = \dot{\rho}_S \,. \tag{3}$$

The *equation of moment for the fluid mixture* can be written:

$$\frac{\partial}{\partial t} \left(\rho \underset{\sim}{u} \right) + \frac{1}{R} \underset{\sim}{\nabla} \bullet \left(R \rho \underset{\sim}{u} \underset{\sim}{u} \right) = - \underset{\sim}{\nabla} p + \frac{1}{R} \underset{\sim}{\nabla} \bullet \left(R \underset{\approx}{\sigma} \right) - \frac{\left(\sigma_0 - \rho w^2 \right)}{R} \underset{\sim}{\nabla} R + \underset{\sim}{F} + \rho \underset{\sim}{G} , \tag{4}$$

The *tensor of the viscous tensions* is given by the relation:

$$\underset{\approx}{\sigma} = \mu \left[(\underset{\sim}{\nabla} \underset{\sim}{u}) + (\underset{\sim}{\nabla} \underset{\sim}{u})^T \right] + \left(\frac{\lambda}{R} \right) \underset{\sim}{\nabla} \bullet (R \underset{\sim}{u}) \underset{\approx}{I} , \tag{5}$$

$$\sigma_0 = \left(\frac{2\mu}{R} \right) \underset{\sim}{u} \bullet \underset{\sim}{\nabla} R + \left(\frac{\lambda}{R} \right) \underset{\sim}{\nabla} \bullet (R \underset{\sim}{u}) , \tag{6}$$

The *equation of the angular* momentum, wich allows the determination of the swirl speed *w*:

$$\frac{\partial}{\partial t}(R\rho w)+\frac{1}{R}\underset{\sim}{\nabla}\bullet\left(R^2\rho w\underset{\sim}{u}\right)=\frac{1}{R}\underset{\sim}{\nabla}\bullet\left(R\underset{\sim}{\tau}\right)+N\text{ , (7)}$$

Where the swirl tension $\underset{\sim}{\tau}$ is given by the relation:

$$\underset{\sim}{\tau}=\mu R^2\underset{\sim}{\nabla}\left(\frac{w}{R}\right), \quad (8)$$

The angular momentum equation only makes sense if there is swirl motion and the problem has axial symmetry.

Internal energy equation:

$$\begin{aligned}\frac{\partial}{\partial t}(\rho I)+\frac{1}{R}\underset{\sim}{\nabla}\bullet\left(R\rho I\underset{\sim}{u}\right)&=-\frac{p}{R}\underset{\sim}{\nabla}\bullet(R\underset{\sim}{u})+\underset{\approx}{\sigma}:\underset{\sim}{\nabla}\underset{\sim}{u}+\tau\bullet\underset{\sim}{\nabla}\left(\frac{w}{R}\right)\\&+\frac{(\sigma_0)}{R}\underset{\sim}{u}\underset{\sim}{\nabla}R-\frac{1}{R}\underset{\sim}{\nabla}\bullet(R\underset{\sim}{J})+\dot{Q}_C+\dot{Q}_S\end{aligned}, \quad (9)$$

The *heat flux* is give by the relation:

$$\underset{\sim}{J}=-K\underset{\sim}{\nabla}T-\rho D\sum_K h_k\underset{\sim}{\nabla}\left(\frac{\rho_k}{\rho}\right), \quad (10)$$

The *chemical reactions* that take place inside the cylinder have been treated as:

$$\sum_k a_{kr}X_k\leftrightarrow\sum_k b_{kr}X_k\ , \quad (11)$$

➢ **particle jet**, whose general equation can be written as a function of *f*, the droplet distribution function, in the form:

$$\frac{\partial f}{\partial t}+\nabla_x\left(fu_p\right)+\nabla_{u_p}\left(fF_p\right)+\frac{\partial}{\partial r_p}\left(fR_p\right)+\frac{\partial}{\partial T_p}\left(f\dot{T}_p\right)=\dot{Q}\,. \quad (12)$$

For a particle k, the simplified impulse conservation equation is:

$$m_k\frac{d}{dt}u_{pk}=m_k g-\frac{m_k}{\rho_k}\nabla p+D_k(u_g)(u_g-u_{pk}); \quad (13)$$

$$u_{pk}=\frac{dx_{pk}}{dt}, \quad (14)$$

Temporal differentiation is based on the ICE (Implicit Continuous-fluid Eulerian) algorithm. Time differentials are expressed as follows:

$$\frac{\partial M}{\partial t}=\frac{M^{n+1}-M^n}{\Delta t}. \quad (15)$$

Time steps are required to advance some sizes; each cycle is performed in three time subphases or phases. The approach is directly related to spatial discretization which is based on the ALE (Alternate Lagrangian Eulerian Method). The three phases are as follows: 1. Phase A – explicit Lagrangian calculus; 2. Phase B - Lagrangian calculus in essence; 3. Phase C – is an Eulerian calculation, a resonance after the Lagrangian calculation, wich allows the estimation of the convective component of the flow equations.

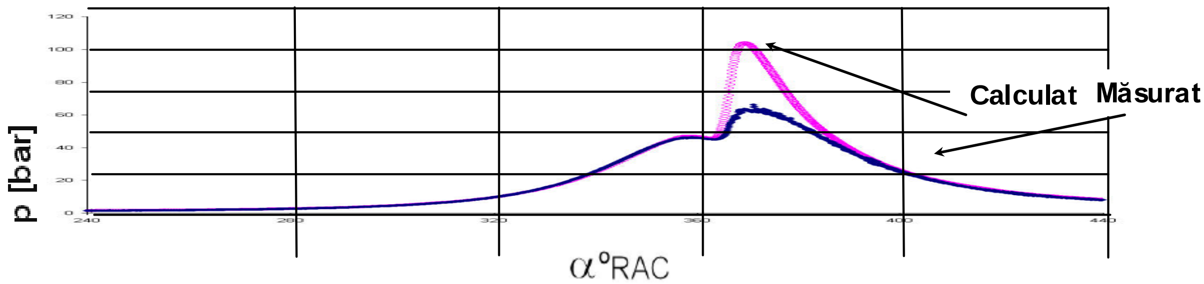
CHAPTER V: NUMERICAL SIMULATIONS AND INTERPRETATION OF RESULTS

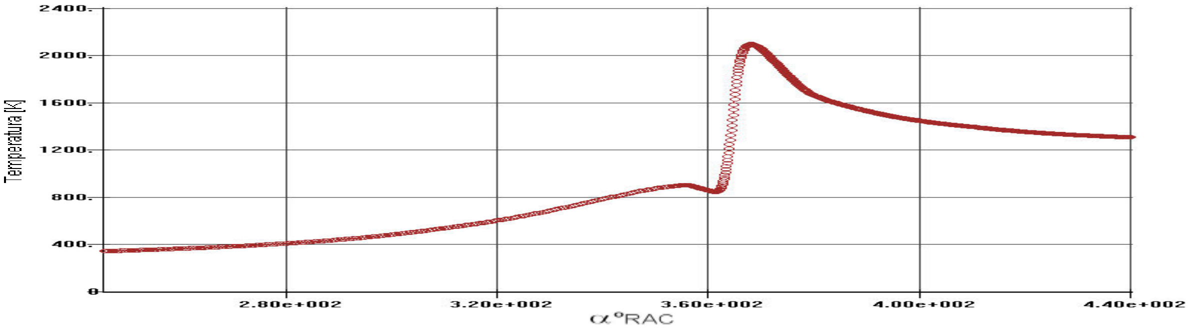
The model developed in the paper was used for numerical simulations subsequent to the analyzed engine T650. Four applications have been developed for T650 engine, as follows:

■ **Application 1** – is based on the preliminary calculation of the T650 engine;

The calculation was performed at maximum speed and power: speed of 2400 rpm; effective power 48 KW; specific consumption 238,077 g/kWh; effective injection advance $\beta = 5.5$ °RAC. The program calculates an engine cycle strating at 120 °RAC before the inner neutral (PMI). At this point it was hypothesized that the turbulence in the cylinder uniformized the velocity gradients, so that the velocity is approximately uniform in the cylinder. The program works with 12 specii, and the fuel was considered saturated pure hydrocarbon $C_{10}H_{20}$. Reactions were divided into: 4 slow kinetic reactions (fuel oxidation and the extended Zel'dovich mechanism for the formation of nitrogen oxides) and equilibrium reactions 8 (element dissociation reactions). The injection advance is 5.5 °RAC, and the injection duration is 17 degrees. The value for the average Sauter is $DSM = 1 \cdot 10^{-3}\ cm$. The discretization network consists of 624 nodes and 583 cells. The size of the discretization network can be changed using a set of parameters.

Results: The figure shows the comparative values of pressure and temperature





The maximum pressure per cycle influenced the temperature, wich caused a sharp increase in the concentration of nitrogen oxides.

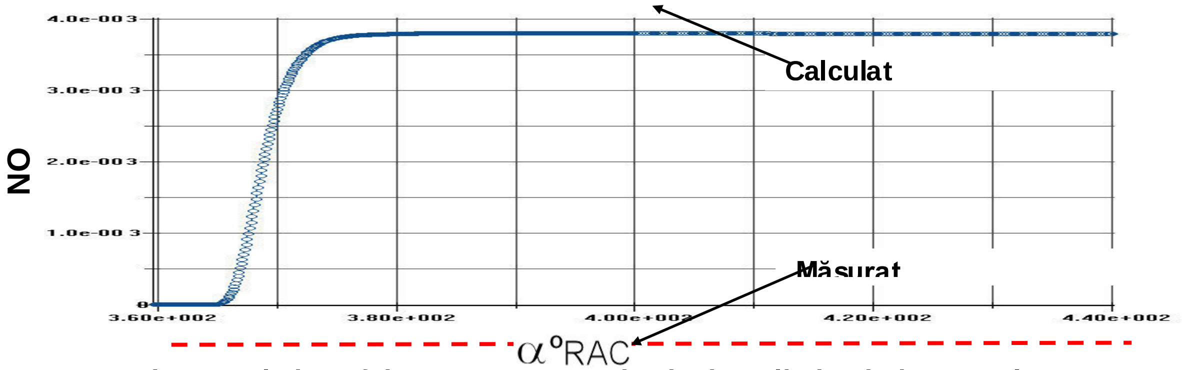


Fig. 4 Evolution of the NOx concentration in the cylinder during a cycle.

The results have only a qualitative character, the values not being in accordance with the measured ones, but the evolution of the phenomena inside the cylinder is captured with sufficient accuracy: the vortices created by the fuel jet; the effect of the combustion chamber on the piston head; the presence of flame at the periphery of the fuel jet in the tire; NOx formation in areas with high temperatures and poor mixtures at the periphery of the jet; the presence of vortices at the outlet of the rich mixtures from the combustion chamber in the piston head; the presence of areas on the wall with low temperatures and areas without mixing where, depending on the flow regime, unused air can remain. After improving the performance of the calculation program (improving the discretization network; reducing the pressure gradient; reducing or eliminating errors caused by the numerical methods used and discrete representation of real numbers), it was found that increasing the fineness of the discretization network does not significantly reduce the pressure gradient and requires the application of better numerical schemes. The PSG (pressure gradient scaling) method was used to increase the convergence rate. The equation of the moment is rewritten:

$$\frac{\partial}{\partial t}\left(\rho \underset{\sim}{u}\right)+\frac{1}{R}\nabla\bullet\left(R\rho \underset{\sim}{u}\,\underset{\sim}{u}\right)=-\frac{1}{\underset{\sim}{a^2}}\nabla p+\frac{1}{R}\nabla\bullet\left(R\,\underset{\approx}{\sigma}\right)-\frac{\left(\sigma_0-\rho w^2\right)}{R}\underset{\sim}{\nabla} R+\underset{\sim}{F}+\rho \underset{\sim}{G} \qquad (16)$$

■ **Application 2** – model calibration calculation; because the data obtained by simulations for NOx emissions were not in accordance with the experimental data, improvements of numerical algorithms were made and models subsumed to other mechamisms of NOx emissions formation werw implemented.

The calculation program has been modified and the calculation subroutines presented above have been added. The application has the same general imput data as the first one and uses the same discretization network, but the calculation is made at two modes on the external characterisctic: a. of maximum speed; b. maximum torque.

For a better interpretation of the obtained data, a calculation model was developed using the specialized program Wave 5.0 of Ricardo.

Results: Average working time - 42 hours and 43 minutes in each case, for a discretization network with 624 calculation cells and 1230 packets of drops. The modification of the program led to: 1. reduction of the pressure per cycle; 2. significant reduction in the number of calls for the negative mass correction routine; 3. the numerical diffusion did not significantly affect the result compared to the case using the donor-acceptor scheme; 4. the results obtained had errors. It can be observed that: a. the deveopled model has a better sensitivity, due to the gas-dynamic approach; b. the calculation of NOx emissions using the Zeldovich equilibrium mechanism, as performed in Wave 5.0, is inappropriate for the case under consideration, and the model developed is more efficient and predictive, because it kinetically approaches the reactions of the Zeldovich mechanism and has temperatures calculated from the flow equations, so closer to the real ones.

Two more mechanisms for the formation of NOx emissions are known from the specialized literature: 1. flame formation, proposed by Fenimore; 2. formation of organic compounds containing nitrogen from the fuel. The activation conditions of the two mechanisms were: temperature above 540^0C and the coefficient of excess air α less than 0.7. From the analysis of the variation of NOx concentration for maximum speed and maximum torque, with and without additional modules, the conclusion is that the calculation effort is not justified, as a variation of 1% is insignificant and, in addition, acting to increase NOx concentration alters and more the result.

■ **The 3rd application** – also validated the model in terms of Nox emissions.

The data used and the results obtained are presented in the tables below. The variation of the NOx concentration on the load characterisctics at two reference speeds was represented for the calculated and measured values.

Table 3

Speed	Specific consumption	Torque	Real advance	NOx-measured	NOx-calculated	Error
[rpm]	[g/kWh]	[Nm]	[deg]	[ppm]	[ppm]	[%]
2400	239.725	205.4	5.4	1108	1215	-9.657
2400	245.1579	153.6	5.1	787	836	-6.226
2401	264.0927	103	6.1	639	586	8.2942
2398	635.0476	50.9	11.1	330	296	10.303
1440	223.966	1.6	7.6	1631	1743	6.867
1440	225.4504	234.2	6.2	1272	1348	-5.975
1439	234.8289	173.1	9.6	1088	1024	5.8824
808	534.154	118.3	12.8	323	274	15.17

Table 4

Read	n	wfi	Clc.GEXHW	Clc.GEXHW_calc	Power
task	read	8 Mode	NOX*wfi	NOX_calc*wfi	Power*wfi
[%]	[rpm]		[g/h]	[g/h]	[kW]
100	2400	**0.15**	**57.05698**	**61.56697**	**7.7445**
75	2400	**0.15**	**41.75935**	**43.35929**	**5.793**
50	2401	**0.15**	**34.79729**	**31.91113**	**3.885**
10	2399	**0.1**	**12.40397**	**10.12599**	**0.525**
0	812	**0.15**	**6.055186**	**5.63938**	**0.0195**
100	1440	**0.1**	**33.70205**	**35.71575**	**3.53**
75	1439	**0.1**	**27.4745**	**28.85834**	**2.609**
50	1439	**0.1**	**24.29459**	**20.60910**	**1.783**
		SUM=	**237.5439**	**237.78595**	**25.889**
			NOX [g/kWh]	**NOX_calc [g/kWh]**	
			9.175476	**9.18483**	
European Limits			**9.2**		
1997 Protocol			**9.8**		

The calculated result is very close to the measured one, in technical terms, in fact, the two values are equal. From the analysis of the variation curves of the NOx emission concentration on the load characteristics necessary for the calculation of the noxious cycle, it resulted that: 1. the emission modulus is strongly influenced by the thermal regime of the engine; 2. the emission formation mechanism is much more complex and is strongly influenced by the combustion mechanisms, which cannot be satisfactorily represented by the 11 chemical reactions used, especially in the case of low temperatures. The conclusion is that a well-calibrated data calculation program can provide good results, which fall within an average margin of error of 10% for cases similar to the one studies and even lower under favorable conditions.

▪ **The 4th application** – aimed at verifying the optimal range for the injection advance, so as to minimize NOx emissions.

Among the possibilities to reduce NOx emissions, the change in the injection advance is the most important, because: a. NOx emissions are strongly influenced by the size of the injection advance; b. injection advance modification is most convenient for in-service engines, because it is easy to execute.

The input data used and the results obtained are in table 5.

Table 5

Speed	Real advance	NOx	NOx-calc	Error	Cons_cb	Power
[rpm]	[deg]	[ppm]	[ppm]	[%]	[g/kWh]	[kW]
2400	0.2	752	776	-3.191	258.2074	47.64
	1.3	782	858	-9.718	256.1854	48.34
	2.2	850	929	-9.294	250.0907	49.59
	3.7	964	1056	-9.543	244.2857	50.4
	4.2	1008	1101	-9.226	242.8684	50.62
	5.4	1112	1215	-9.262	241.5013	50.89
	6.5	1211	1327	-9.578	239.3538	51.38
1440	2.6	1156	1232	-6.574	231.1098	33.88
	3.5	1221	1321	-8.190	232.5629	34.18
	4.6	1344	1455	-8.259	228.125	34.56
	5.6	1498	1573	-5.007	226.1494	34.8
	5.9	1557	1612	-3.532	225.7233	34.91
	7.3	1689	1784	-5.625	224.5634	34.93
	8.3	1792	1904	-6.250	224.7998	34.96

The error between the measured and calculated values is between 3 and 12 %, most being in the range of 8-10%, being relatively large, but satisfactory, taking into account the complexity of the phenomen and the relatively low capabilities of the model. At partial loads the errors can be higher. The program can be used to study the influence of injection advance on NOx emissions.It can be seen that the results are better for the regimes at which the calibration was performed (speeds of 2400 and 1440 rpm).

CHAPTER VI: FINAL CONCLUSIONS, DIRECTIONS OF FURTHER DEVELOPMENT AND PERSONAL CONTRIBUTIONS

The objective of the thesis was achieved, which was subsumed under the elaboration of a calculation program meant to allow the estimation and, implicitly, the limitation of pollutant emissions produced by naval Diesel engines, in particular NOx emissions regulated by Annex 6 of the 1997 Protocol, effective in 2010.

Briefly, the conclusions are: 1. The program meets the purpose for wich it was created; 2. the results obtained are consistent with the experimental data, especially for the calibration and high load regimes; 3. it is sufficiently predictive to be used in the analysis of concrete cases; 4. well calibrated, can be used to analyze the combustion process in order to improve technical and economic performance and reduce NOx emissions of diesel engines (mainly by optimizing the injection advance, the shape of the combustion chamber, organized and turbulent movement in the cylinder); 5. the results depend quite a lot on the constants of the models and, for this reason, they need to be carefully analyzed and interpreted with discerment; 6. program performance is limited by the models used; 7. the program is an open, working one, and for reason it is difficult to use as advanced knowledge of programming and motor theory is required.