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Application of the Experimental Designs on the Modeling of the Combustion's Parameters

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Abstract: Experimental modeling is based on the statistical methods of the experimental designs; its fundamental role is to analyze the experimental data processing. This is effective for the study of process which contains several independent variables. In this cask, parameters vary simultaneously according to the experimental design, contrary to the Newtonian method. The factors studied can have two, three or several levels according to the conditions of experiments and the measuring accuracy. Usually, these factors have various measuring units; this is why the experimental design is made up on the basis of the coded values. Applications were carried out on an injection pump of the diesel engine having used plunger pistons. This pump is regarded as very important body in the diesel engine and plays a capital part in its operation. The wear of this plunger pistons which is abrasive nature, has a very harmful impact on the techniques and economic indices of the engine. The aim of this study is to evaluate how the wear of the elements of the injection pump on combustion. Experiments were carried out, by simultaneously varying parameters considered to be influential, to lead to the modeling of the provided pressure and the feeding coefficient.

Key words: Experimental modeling, regression, plunger-pistons, wear, supply coefficient

INTRODUCTION

In 1919, Ronald Aylmer Fischer showed a big concern about the statistical methods of the experimental designs (Scheffler, 1986). The aim of theses methods is the analysis of the experimental data processing.

The problem is to know the relationship between the phenomenon and the considered variables. As an answer, an experience is necessary. During which, various values will be given to the planned variables in order to know their influence on the phenomenon.

Once the mathematical model obtained (regression equation), we carried out statistical analysis of the results. The purpose of this analysis is to check the significance of the coefficients of regression and the adequacy of the model.

The wear of the pistons of the injection pump is of abrasive nature. It appears in the form of parallel furrows beginning at the head's extremity, reaching the helicoids slope of the piston. On the level of the helicoids slope wear is of erosive/abrasive nature characterized by curvilinear furrows which converge towards the median of the zone worn (Hebbar *et al*., 2000; Hebbar, 2001).

The wear of these pistons influences negatively on the pressure and the feeding coefficient, begetting a very ominous effect thus on the technical and economic indications of the Diesel engine. The analysis of the influence of regulating parameters of an injection pumps on the pressure, can be worked out with the help of the design of experiments (Goupy, 1999; Novik, 1981; Bandemer and Bellmann, 1976; Nalimov and Tschernova, 1965; Kafarov, 1974), based on the statistical methods.

The pressure and the feeding coefficient are considered by many authors as the main and essential criteria for the determination of the qualitative state of an injection pump (Hebbar *et al*., 2000; Hebbar, 2001).

In order to appreciate the technical state of the plunger pistons, we have used the hydraulic sealing such a criteria is precise and afford commodity. The sealing is represented in percentage with respect to a new piston (Hebbar, 2001).

In this study, we carried out experiments planned on an injection pump having pistons of different sealing (different degrees of wear). These experiments make it possible to know the influence of this wear on the diesel engine characteristics, namely: the presses, the feeding coefficient and finally to optimize the parameters of adjustment to ensure an adequate operation of the engine.

EXPERIMENTAL DESIGNS

The experimenter is confronted by the choice of an experimental design, allowing him to lead to the result sought without requiring a high number of tests

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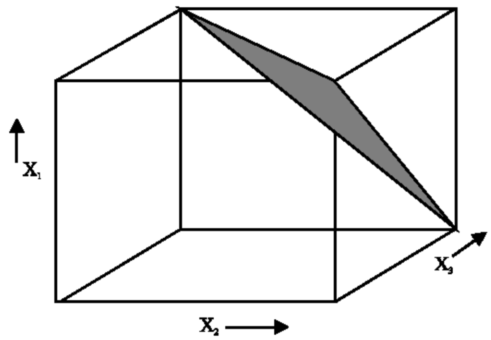

Fig. 1: Representation of the plan of the type 2^3

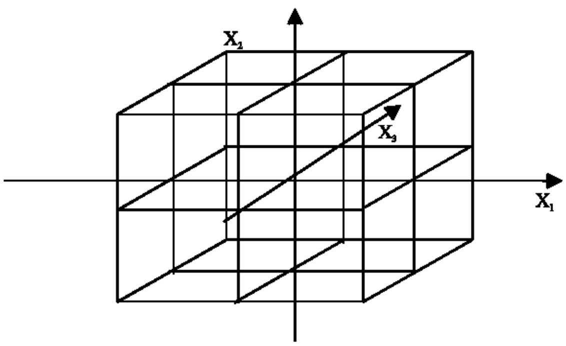

Fig. 2: Experimental design of type 3^3

(Goupy, 1999). In practice, we use a linear equation in priority, according to the need we move to the highest degree (Novik, 1981).

Several experimental designs can be considered according to the choice of the experimenter and the degree of accuracy of the experiment. As an example, we can quote:

Experimental design of first order

Experimental design of the 2^k type: In an experimental design of the 2^k type, each factor has two levels with values +1 and -1. If k = 3 the points of experiments are on the tops of a cube. It is noted that the region of the optimum is situated on shadowed area (Fig. 1).

Experimental design simplified of type 2^{k-p}: For the study of four, five or several parameters, the experimental designs of the 2^k type require a significant number of experiments. Due to the interactions of weak influences, there is possibility of reducing the number of experiments. A plan of the type 2^7 requires 128 experiments, it could be replaced by a plan simplified of type 2^{7-4} for example to become a plan of the type 2^3 requiring eight experiments only with one not very reduced precision.

Experimental design of second order: One generally uses a plan of second degree when the mathematical model resulting from a plan of first order is not adequate, in spite of the repetitions of the doubtful experiments.

The plans of second order require a considerable time; it is for that that only the most influential parameters will be maintained.

Experimental design of the 3^k Type: In an experiment of the type 3^3 for example, each parameter includes/understands three levels (Fig. 2), with values -1, 0 and +1 and in the event of need, it can have up to five levels -α, -1, 0, +1, +α.

Experimental design of the type $2^k3^{k'}$: In uses a plan of this type when one expects quadratic effects of some parameters only and not of the others. Same procedure carried out on the plan 2^3, a plan of the type 3^5 requires 243 experiments and it can also have the form $(2^3.3^2)$ to only carry out 72 experiments. One can still replace the plan 2^3 by 2^{3-1}, for finally carrying out 36 experiments only instead of 72.

The mathematical model: When the statistical methods are used, the mathematical model is appeared as a polynomial, which is a truncation of the Taylor series in which develops the unknown function (Vivier, 2002; Nalimov and Tschernova, 1965).

$$Y(X_1, X_2, \ldots, X_k) = \beta_o + \sum_{i=1}^{k} \beta_i X_i + \sum_{i,j=1}^{k} \beta_{ij} X_i X_j + \sum_{i=1}^{k} \beta_{ii} X_i^2 + \ldots \tag{1}$$

With:

$$\beta_i = \frac{\partial Y}{\partial X_i}; \beta_{ij} = \frac{\partial^2 Y}{\partial X_i \partial X_j}; \beta_{ii} = \frac{\partial^2 Y}{\partial X_i^2}; \ldots$$

According to the need for precision, one can develop the Taylor series with higher powers $X_3 \ldots, X_k$.

Generally, in practice one uses the equation in linear form in priority and according to the need one passes to the higher degree.

Boxes and Wilson (Boxes and Wilson, 1951) proved that by satisfying the matrix of the data input with the three conditions:

$$\sum_{i=1}^{N} X_{0i} X_{ij}^2 = 0; \quad u \neq j$$

$$\sum_{i=1}^{N} X_{ij}^2 X_{iu}^2 = 0; \quad j \neq 0$$

One must check, if the mathematical model satisfied the mathematical criteria (adequacy). In if necessary, one must remake the experiments according to a model of higher power (2nd order, 3rd…).

TRIBOLOGICAL APPLICATIONS

MODELING OF THE MAXIMAL PRESSURE OF AN INJECTION PUMP

Material: Experiences have taken place on a Bench of pumps test (1) BELCAN, Type DS 867 N°S 750, while using an injection pump (2) Bosch PES 4A 75C 410/3 Rses 1183, equipped of worn-out pistons and valves; on its rack, it is placed a graduated stick (6), permitting the positioning of the rack, the used injectors (3) are Bosch mark and calibrating to 175 bars, the number turns of the camshaft is indicated on a speed meter (4) of the Bench and in short a manometer of pressure (5) indicating to every test the maximal pressure and the chamber pressure (7).

Experimentation: In this experimentation, one records the maximal pressure to the level of the hydraulic tubing at the time of the instantaneous closing of the nozzle needle (Fig. 3). These measures are taken to every time under the influence of five parameters to know:

- The speed of camshaft rotation (X_1) [rpm]: [60; 100]
- The positions of the rack (X_2) [mm]: [22; 34]
- The hydraulic tightness of pistons (X_3) [%]: [18; 48]
- The hydraulic tightness of valves (X_4)[%]:[09;16;23]
- The height of the tappet (X_5) [mm]: [33; 33.5; 34]

For the above-mentioned, chosen levels of parameters, we will be in need for a plan of type $2^3.3^2$ experiences (72 experiments). This last requires a considerable time; to reduce the length of the experimentation, a reduced simple rotator plan of type $2^{3-1}.3^2$ = 36 experiences responds to the same requirements.

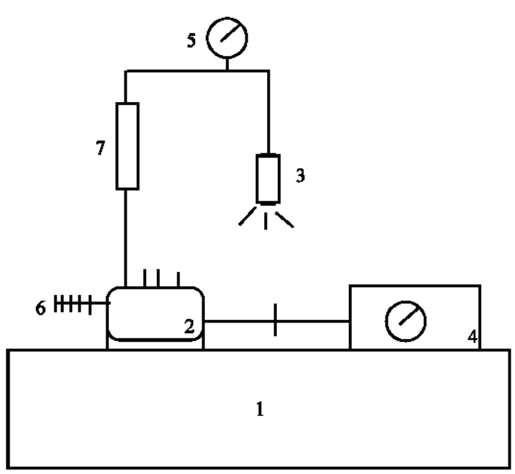

Fig. 3: Bench of test

RESULTS AND INTERPRETATIONS

The count of regression coefficients (Bandemer and Bellmann, 1976; Nalimov and Tschernova, 1965; Kafarov, 1974) permits to get the following model:

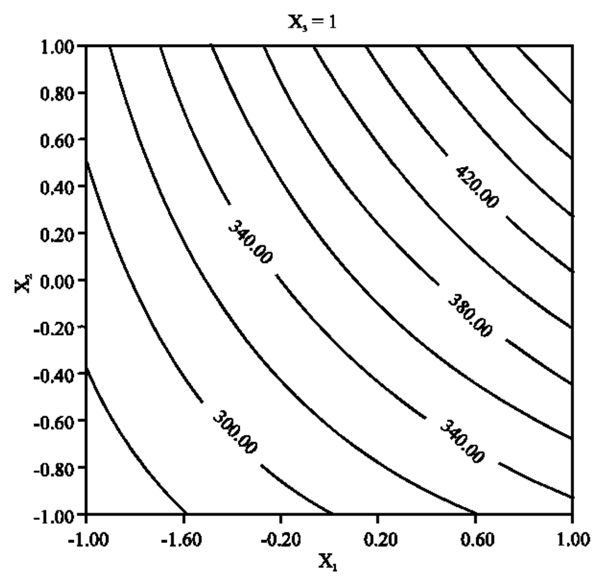

Fig. 4a: Effect of the hydraulic tightness and the speed of rotation on the pressure

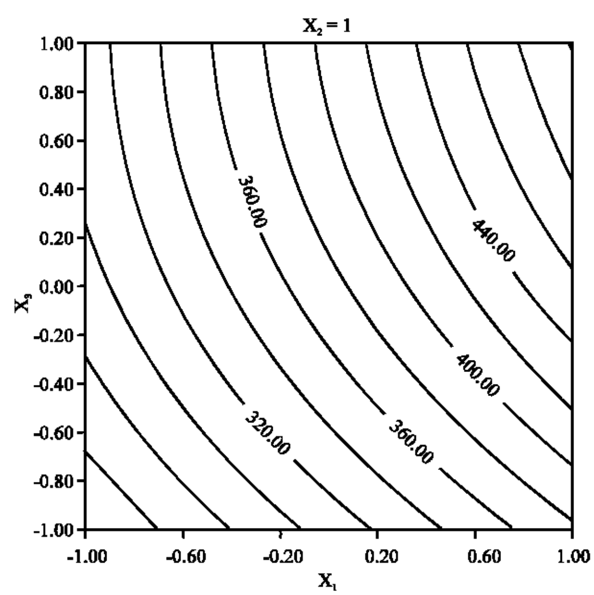

Fig. 4b: Influence of the rack and the speed of rotation on the pressure

Table 1: Matrix of the plan $2^2.3^1$

N°	X_1	X_2	X_3	X_1X_2	X_1X_3	X_2X_3	$(X_3^2\text{-}2/3)$	Y_1	Y_2	Y_3	$\bar{Y}$
1	-	-	-	+	+	+	1/3	180.0	200	186	188.66
2	+	-	-	-	-	+	1/3	205.0	228	212	215.00
3	-	+	-	-	+	-	1/3	255.0	235	230	240.00
4	+	+	-	+	-	-	1/3	395.0	366	395	385.33
5	-	-	+	+	-	-	1/3	280.0	260	255	265.00
6	+	-	+	-	+	-	1/3	330.0	344	340	338.00
7	-	+	+	-	-	+	1/3	310.5	295	310	305.16
8	+	+	+	+	+	+	1/3	525.0	500	505	510.00
9	-	-	0	+	0	0	-2/3	263.0	260	242	255.00
10	+	-	0	-	0	0	-2/3	293.0	270	272	278.33
11	-	+	0	-	0	0	-2/3	288.5	306	300	298.16
12	+	+	0	+	0	0	-2/3	430.0	432	458	440.00

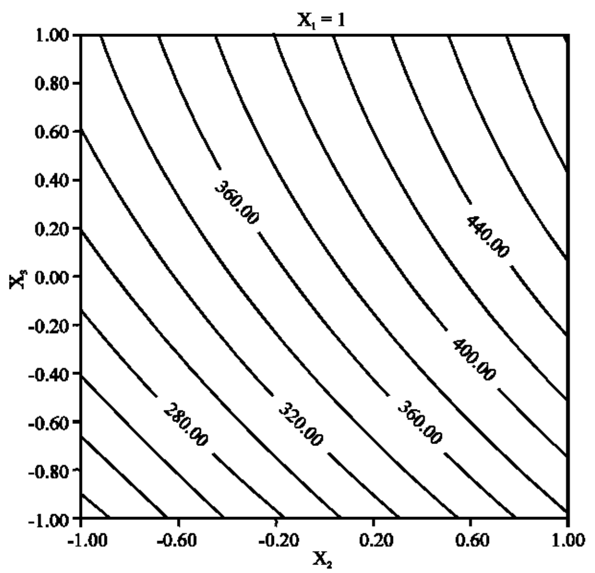

Fig. 4c: Effect of the rack and the hydraulic tightness on the pressure

$$Y(Xi, bi) = 319.97 + 51.76X_1 + 49.47X_2 + 27.97X_3 + 30.45X_4 + 11.63X_5 \quad (2)$$

This first model provides us with the value $F_{exp} = 4.337$ that is superior to F_{th}. The model is therefore inadequate.

One proceed again white using a model of a second degree and considering just only the most influential parameters on the phenomenon, to know:

- The speed of the camshaft (X_1), that has for values limits [60; 100]
- The tightness of pistons (X_2), of which values limits [18; 48]
- The positions of the rack (X_3), that varies himself in the domain [22; 28; 34]

The influence of the tightness of valves and the height of the tappet has been disregarded, for they are just secondary parameters without primordial importance. As for as the above mentioned hypothesis, the plan of experiments takes then the type $2^2.3^1$ that only requires 12 experiments (Table 1).

The mathematical model will take the following shape:

$$Y(Xi,bi) = 321.09+51.306X_1+53.26X_2+48.646X_3 +30.82X_1X_2+13.27X_1X_3-16.75X_3^2 \quad (3)$$

The value of $F_{exp} = 2.60$, F_{exp} is lower to F_{th}. The model is very adequate for the description of the phenomenon. The Fig. 4a-c show the degree of influence of the different parameter on the pressure.

For a maximal position (X_3 = 34 mm) of the rack (full output), the influence of the speed of the camshaft appear more than the influence of the tightness of pistons on the pressure. The rise of these two variables increases the pressure in a no linear way (Fig. 4a).

The Fig. 4b, shows that for pistons of a tightness of 48%, the influence of the speed of the camshaft appears than the output on the pressure. Here, as well, the pressure rises in a non-linear way as far as the growth of these two parameters is concerned.

When one maintains the speed constant rotation which is equal to 100 rpm, the pressure of the fuel grows linearly for a variation of the tightness of pistons of 18 to 30.6% and positions of the rack of 22 to 27.22 mm. Beyond these values, it takes a no-linear aspect until to reach a maximal value of 501.64 bars, for the maximal values of these two variables (Fig. 4c).

MODELING OF THE FEEDING COEFFICIENT OF AN INJECTION PUMP WITH WORN OUT PISTONS

Materials: As show in Fig. 3, the experimentation requires an injection pump equipped by pistons with different sealing and an anti-pressure chamber.

Experimentation: The experimentation is made on a bench test (Fig. 3). According to a planning of experiments of

Table 2: The influencing parameters

Parameters		Units	-α	-1	0	+1	+α
Rotational speed of the camshaft	(X1)	rpm	50	60.0	80	100.0	110
Rack Positions	(X2)	mm	22	23.5	28	32.5	34
Hydraulic sealing of plunger-pistons	(X3)	%	10	18.0	33	48.0	55
Injection pressure	(X4)	bar	75	100.0	150	200.0	225
Chamber pressure	(X5)	bar	0	20.0	40	60.0	80

second order and of type two requiring 27 experiments under the influence of parameters whose values are reported in Table 2.

RESULTS AND INTERPRETATIONS

Computation of the regression coefficients:

$$\overline{Y} = \frac{\sum_1^N X_0 Y}{\sum_1^N X_0^2} = 78.31;\ \beta_{ii} = \frac{\sum_1^N X_i^* Y}{\sum_1^N X_i^{*2}};$$

$$\beta_0 = \overline{Y} - \frac{1}{C^*}\sum_1^k \beta_{ii} = 88.75$$

$\beta_{11} = -2.068$; $\beta_{22} = -6.025$; $\beta_{33} = -7.539$;
$\beta_{44} = -0.478$; $\beta_{55} = -2.037$; $\beta_1 = -0.637$;
$\beta_2 = 7.234$; $\beta_3 = 12.231$; $\beta_4 = -0.629$;
$\beta_5 = -2.689$; $\beta_{12} = 2.32$; $\beta_{13} = 0.275$;
$\beta_{14} = -2.245$; $\beta_{15} = 4.996$; $\beta_{23} = 2.549$;
$\beta_{24} = -1.027$; $\beta_{25} = 1.261$; $\beta_{34} = -3.886$;
$\beta_{35} = 3.637$; $\beta_{45} = 1.026$; $|\Delta\beta_i| = 2.509$

By factorial analysis, we can obtain the following mathematical model:

$$\begin{aligned} Y(X_i,\beta_i) = {} & 88.75 + 7.234X_2 + 12.231X_3 - 2.689X_5 \\ & + 4.996X_1X_5 + 2.549X_2X_3 - 3.886X_3X_4 \\ & + 3.637X_3X_5 - 6.025X_2^2 - 7.539X_3^2 \end{aligned} \quad (3)$$

This is an adequate model since the Test of Fischer $F_{exp} = 1.85$ is less than F_{th}.

The graphs of Fig. 5a-c show the influence of the different parameters on the feeding coefficient.

For an average value of the rotational speed, the injection pressure and of the pressure of the chamber respectively such as $X_1 = 80$ rpm, $X_4 = 150$ bars, $X_5 = 40$ bars, the mathematical model will have the Form:

$$\begin{aligned} Y(X_i, \beta_i) = {} & 88.75 + 7.234X_2 + 12.231X + \\ & 2.549X_2X_3 - 6.025X_2^2 - 7.539X_3^2 \end{aligned} \quad (4)$$

The feeding coefficient increases nonlinearly and rapidly under the influence of the increase of the flow rate and the degree of sealing of the plunger-pistons (Fig. 5a).

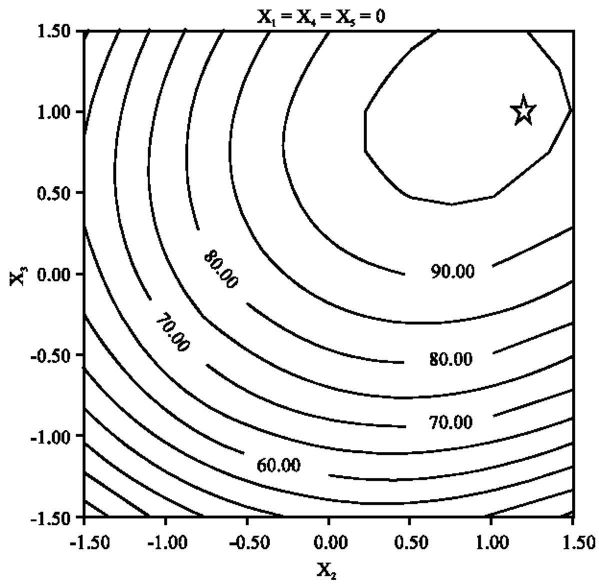

Fig. 5a: Effect of the flow rate and hydraulic sealing on the feeding coefficient

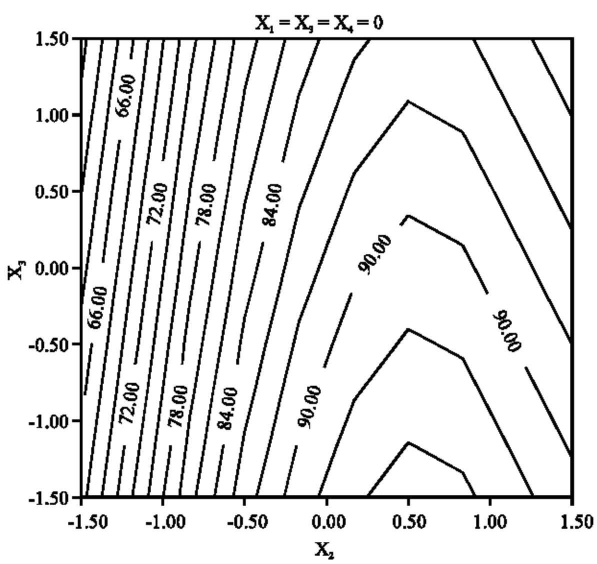

Fig. 5b: Effect of the hydraulic sealing and anti-pressure on the feeding coefficient

We can read the optimum value of the feeding coefficient (*) directly from the graph. Its coordinates are:

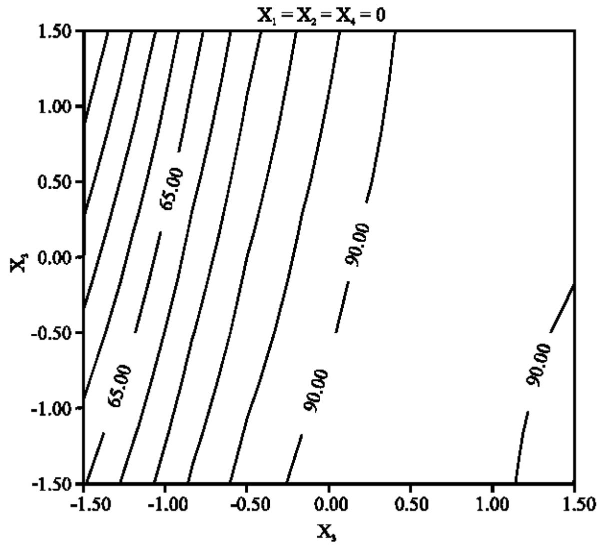

Fig. 5c: Effect of the flow rate and anti-pressure on the feeding coefficient

$X_2 = 0.8\times4.5+28 = 31.6$ mm and $X_3 = 48\%$, by substitution Y will have a value of 97.41% which is considered as the optimal point.

If we maintain the parameters (rotational speed, rack positions, injection pressure) constant for average values respectively $X_1 = 80$ rpm, $X_2 = 28$ mm and $X_4 = 150$ bars (Fig. 5b), we obtain a model of the form:

$$Y(X_i, \beta_i) = 88.75 + 12.231X_3 - 2.689X_5 + 3.637X_3X_5 - 7.539X_3^2 \quad (5)$$

For a decreasing variation of the anti-pressure from 80 to 0 bars and an increase in the hydraulic sealing up to 39%, the feeding coefficient reaches 90%. This value can also be obtained if we set the injection pressure to 36 bars with pistons having a 39% sealing.

For average values and constants $X_1 = 80$ rpm of the camshaft $X_3 = 33\%$ pistons sealing and $X_4 = 150$ bars injection pressure (Fig. 5c), the model will take the form:

$$Y(X_i, \beta_i) = 88.75 + 7.234\,X_2 - 2.689X_5 - 6.025X_2^2 \quad (6)$$

With a creasing variation of the anti-pressure of 80 bars, the feeding coefficient increases linearly and rapidly when the position of the rack moves from 22 to 24.4 mm; then increase slowly in nonlinear manner until the position of the rack of 30.25 mm.

The feeding coefficient begins to decrease slowly and linearly when the flow rate varies from the position of 30.25 to the position of 34 mm with an increase in the pressure of the chamber.

CONCLUSIONS

The wear of elements of high precision of the injection pump is especially accelerated in the African countries (sandy environment). It is essentially owing to foulness contaminating the fuel, in large part at the time of its transport or its inadequate storage.

This wear, of abrasive nature, has an ominous impact on the functioning of the injecting pump. This produces a falling of pressure, which is exhausted toward, injector, begetting a bad combustion.

We can deduct from the above analysis, that an extremum feeding coefficient of 97.41% leads to a suitable operating conditions of a diesel motor, can be obtained by the pistons of an injection pump having a sealing of only 48% with a rotational speed of camshaft of 80 rpm with a rack position at 31.6 mm and an injection pressure of 150 bars and an anti-pressure of 40 bars.

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