GUIDE-VANES UPSTREAM THE IMPELLER OF CENTRIFUGAL COMPRESSOR.

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ABSTRACT

Pre-swirl is obtained by using adjustable inlet guide vanes, located upstream the turbomachinery's impeller, in order to realize an angular deviation of the absolute velocity of the fluid at the leading edge of the impeller. This process is used mainly for regulating both the flow-rate and the power of the fans and blowers and it allows to keep an acceptable value for the turbomachinery's efficiency. In order to reduce the "turbo-lag" of the automotive engine, we have studied the possibility of using pre-swirl to increase the rotational speed of the turbocharger at low engine speed, and thus, to improve the "transient response" of the propulsion unit (motor and turbocharger) in the case of a sudden acceleration. In this paper are presented five different adjustable inlet guide vanes, one of which has been patented. This article shows the analysis and comparisons of their respective performances, in association with a centrifugal compressor.

INTRODUCTION

Traditionally, the performances of a turbocharged automotive engine are characterized by its torque curve vs rotational speed. At a transient speed during a sudden acceleration, as the surpercharge pressure is not available instantaneously, the torque variation corresponds to the graph below.

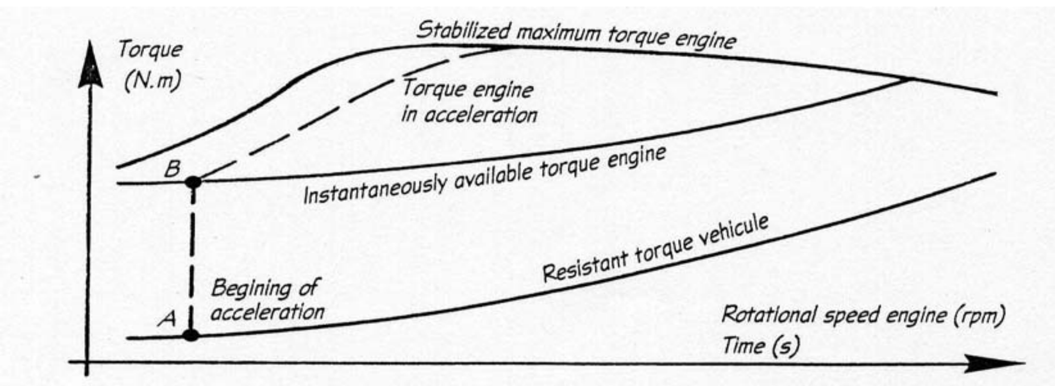


FIG 1 Torque's variation of a turbocharged engine during an acceleration.

The torque at slow running (point A) is located on the curve of resistant torque vehicle (rolling and aerodynamic losses). If the accelerator is pressed on, the torque engine immediately reaches point B, on the curve of the available torque engine in acceleration. This curve can't be superimposed on the curve of stabilized maximum torque engine, except at really high rotational speed engine. The torque at point B corresponds to the supercharge pressure available when the system is running at point A, as the turbocharger has not accelerated yet, and as the passage from point A to point B is instantaneous. The gap between the two torque graphs is due to the time of increasing rate of the turbocharger. This response delay can be reduced in three ways:

- by improving inlet and outlet pipes.
- by decreasing the inertia of the rotor of the turbocharger (using ceramic impeller)
- by reducing the gap, on the rotational turbo speed, between slow running and maximal running.

Pre-swirl addresses the third possibility by reducing the compressor's torque (see Euler's equation) and at constant energy received by the turbine, which consequently increases the turbocharger's rotational speed.

ROLE AND GOAL OF PRE-SWIRL.

Pre-swirl is obtained by using adjustable inlet guide vanes, located upstream the turbomachinery's impeller (fig 2), in order to realize an angular deviation of the absolute velocity of the fluid at the leading edge of the impeller. The principle of such a tuning is issued from the Euler's equation:

$$P_{ower} = q_m \left[U_2 V_2 \cos \alpha_2 - U_1 V_1 \cos \alpha_1 \right]$$

According to the value of the deviation created by the inlet guide vanes, the velocities triangle at the leading edge of the impeller will be as presented on figure 3.

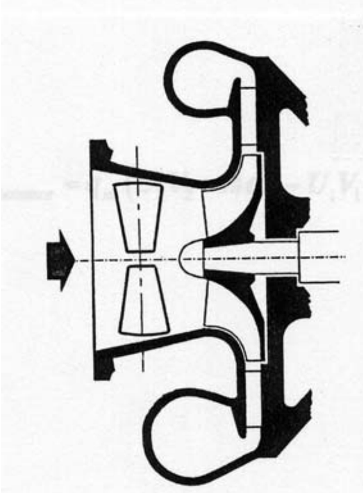


FIG 2 Centrifugal compressor with pre-swirl.

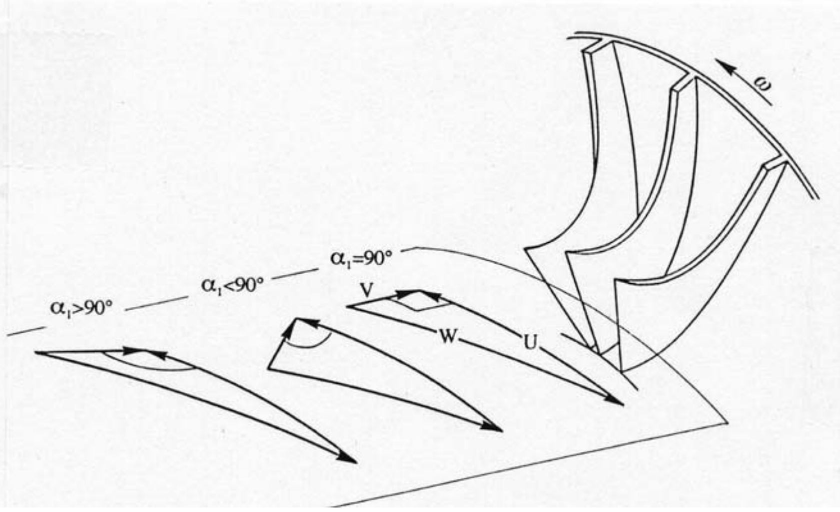


FIG 3 Velocities triangle evolution with the pre-swirl angle

We can see immediately that if $\alpha_1 < 90°$, the shaft power used by the compressor is decreased, and then, the rotational speed of the turbocharger is increased.

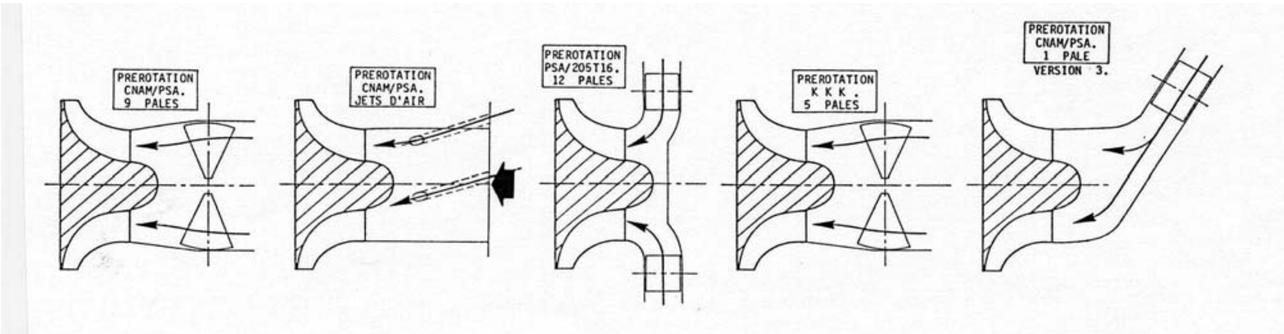
An appropriate adjustment of the pre-swirl's vanes angle allows to increase the rotational turbo speed, at slow running for instance, and thus favours the reduction of the turbo-lag during a sudden acceleration.

EXPERIMENTAL STUDY.

The experimental study have concentrated on five different pre-rotators:

1) two axial pre-rotators, one of which includes 5 blades, and the other one 9 blades.
2) two radial pre-rotators, one of which includes 12 blades, and the other one, original, [1], with 1 blade only.
3) One static pre-rotator with peripheral jets.

These five pre-swirl systems are represented on the figure 4 below.



(a) (b) (c) (d) (e)

FIG 4 The five tested pre-rotators.

Three among the five pre-rotators are also shown on the photographs of the figure 5.

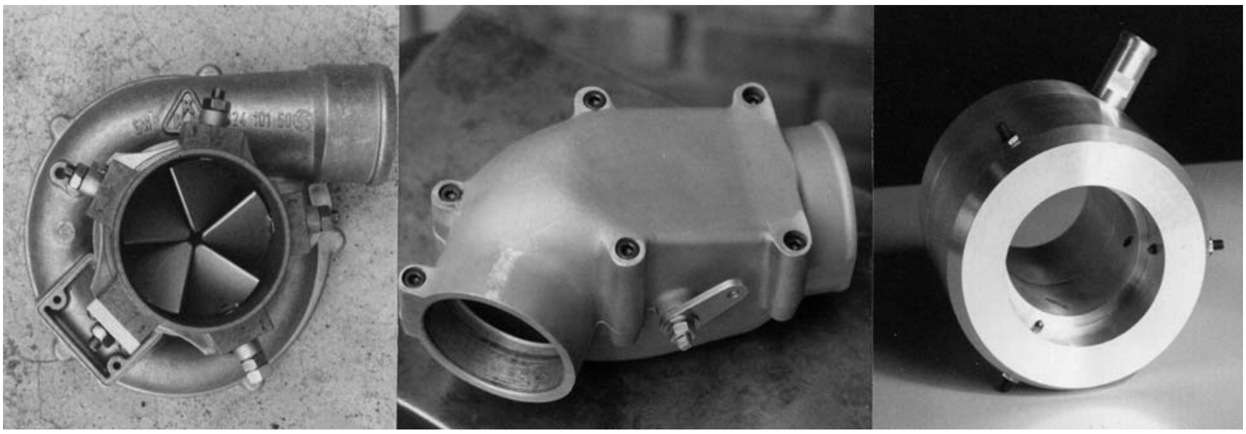


FIG 5 axial 5 blades (d) radial 1 blade (e) static pre-rotator (b)

All the experimentations were led by associating each one of the five pre-rotators with a centrifugal compressor driven by its coupled turbine KKK K16 model.
The test bench used for these experiments is the one of CNAM laboratory of turbomachinery at Saint-Cyr-l'Ecole. Its diagram and photograph are shown on figure 6.

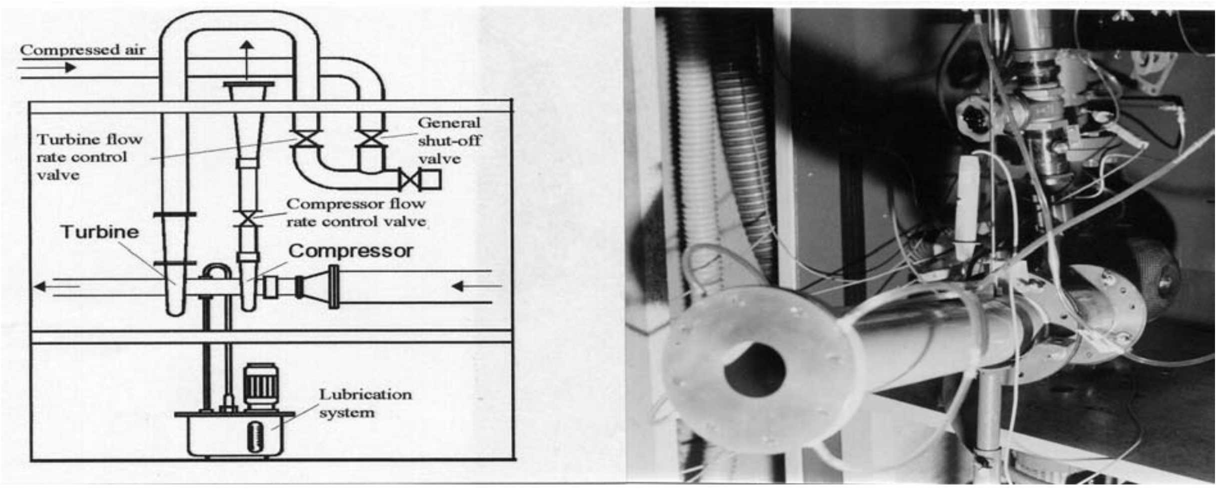


FIG 6 Turbochargers test bench

UNFOLDING OF THE EXPERIMENTS.

Many experimental periods have been realized. In this paper, we present only characteristic results about the pre-rotator + compressor entity. We had previously determined by other experimentations, for each different system, the angular value of the pre-swirl's blades direction for which we could get some identical characteristic curves. For instance, figure 7 allows to compare the systems to each other for a "positive" angle of the pre-rotator's blades..

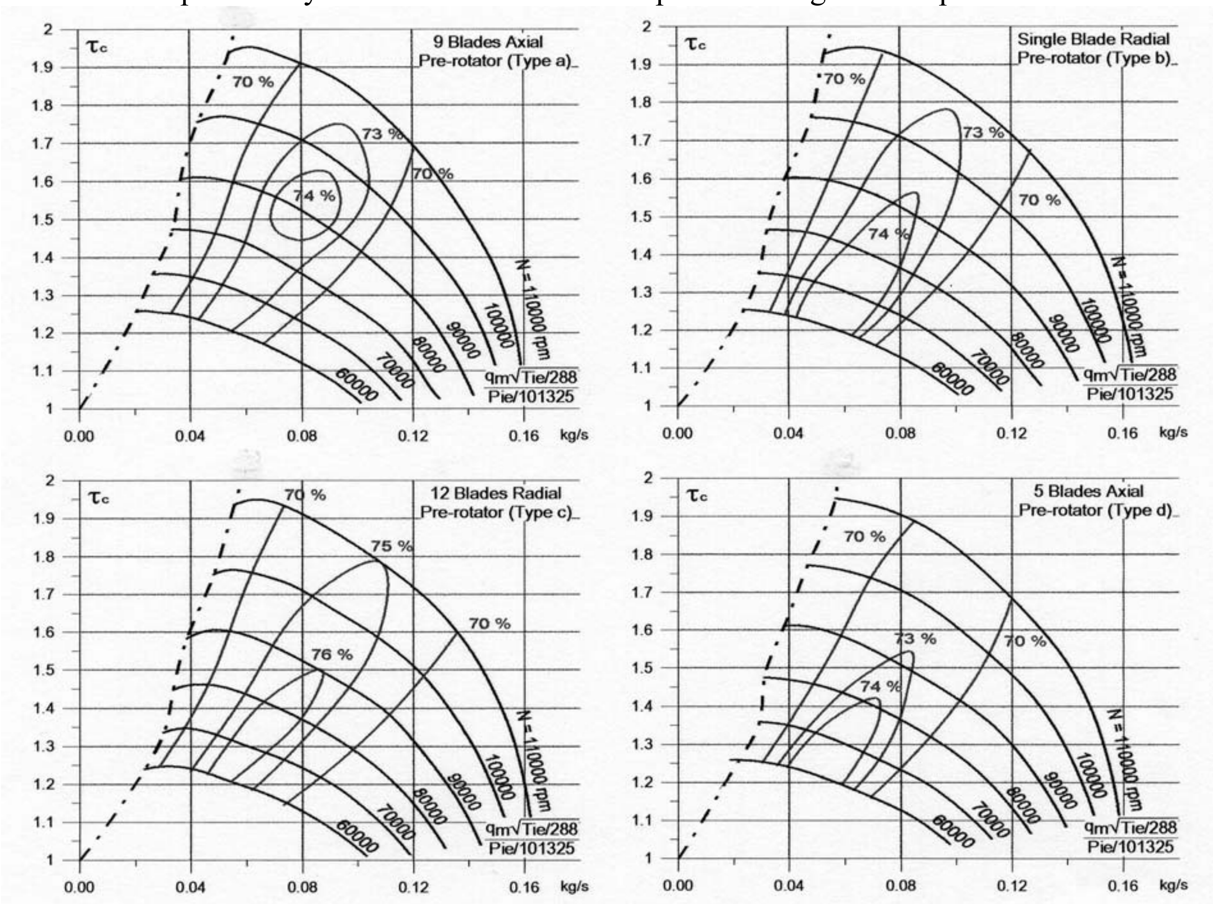


FIG 7 Characteristic curves compressor with 12, 9, 5 and 1 blades pre-rotators.

Characteristic curves obtained with a medium adjustment of inlet guide vanes (not the same angle for each system), positions which give a quasi similar pressure ratio and mass-flow rate.

Comparison and analysis of these characteristic curves show that:

1) The 12 blades radial pre-rotator is the more efficient (η_{max} =76%). In return, this system is too expensive and dimensionally very important, in regard to other models.
2) The 3 other systems give quasi-identical performances (η_{max} =74%). The single blade pre-rotator must obviously be selected. It is simpler and can be manufactured at low cost.
3) On the other hand, the turbocharger's implantation under the bonnet of a car engine often includes a 90° bend, just before the compressor; this system may be a judicious solution. This original pre-rotator has been patented by Cnam and Peugeot SA.

ENGINE TEST BENCH EXPERIMENTS.

Now, we present some experimental results obtained with the 12 blades radial pre-rotator operating with a Peugeot turbocharged petrol engine, 2 litres capacity. The KKK k16 model and the 12 blades radial pre-rotator are shown on the figure 8.

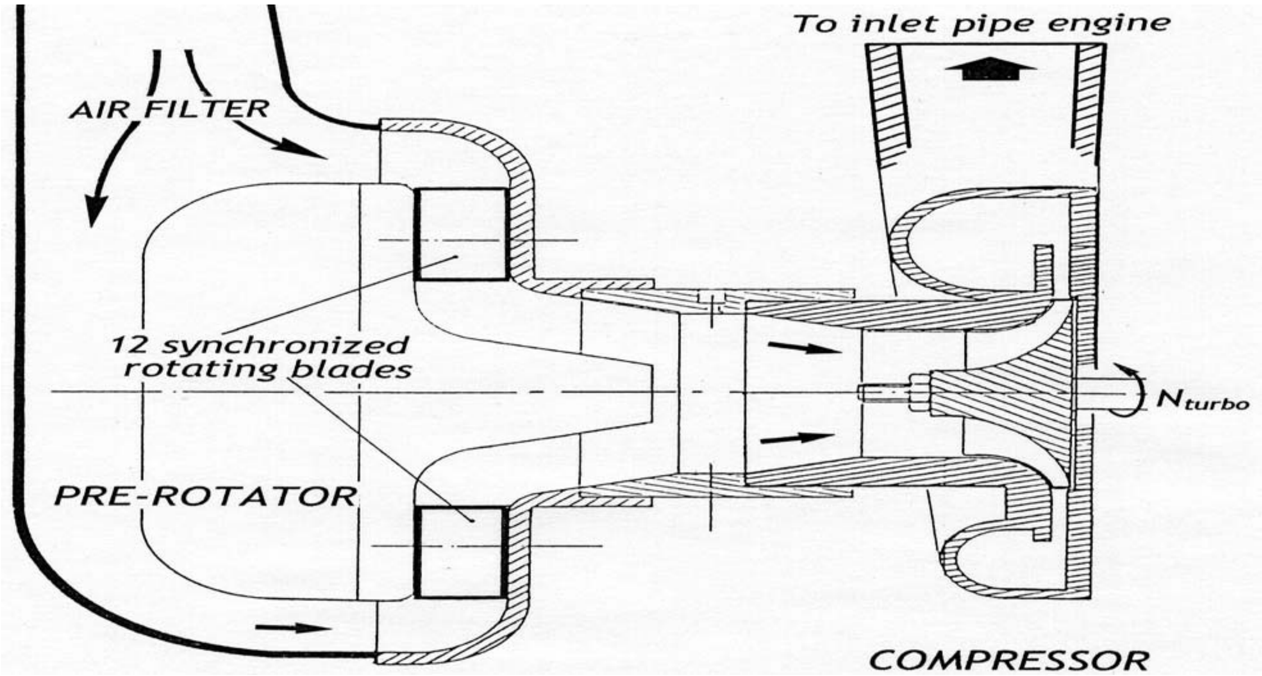


FIG 8 PSA Pre-rotator and KKK k16 compressor.

The methodology of experiments stands as follows:
At constant engine rotational speed and engine load (the torque), we test nine different inclinations of the inlet guide vanes . Theses experiments are made for five different values of the torque and for six values of the engine rotational speed.

Beyond the engine parameters (rotational speed, torque, specific consumption, richness and different pollutants), many other data concerning the turbocharger and his inlet guide vanes are registred (pressures and temperatures upstream and downstream inlet guide vanes and compressor, rotational speed of turbocharger…).

The mass flow rate of the compressor is:

$$q_{m_{air}} = \frac{q_{m_{petrol}} \times 14,624}{R_{ichness}} \quad (kg/s) \qquad \text{with richness} = \frac{\left(q_{air} / q_{petrol} \right)_{stoechiometric}}{\left(q_{air} / q_{petrol} \right)_{effective}}$$

The compression pressure ratio and the isentropic efficiency of the inlet guide vanes + compressor entity are computed from the different measurements of pressures and temperatures.
The figure 9, relative to the 3500 rpm engine speed, shows the increase of turbo rotational speed for the 5 values of the torque (30,9 to 215,8 N.m) according to the positions of the inlet guide vanes's blades. The iso-torques are also iso air mass flows.

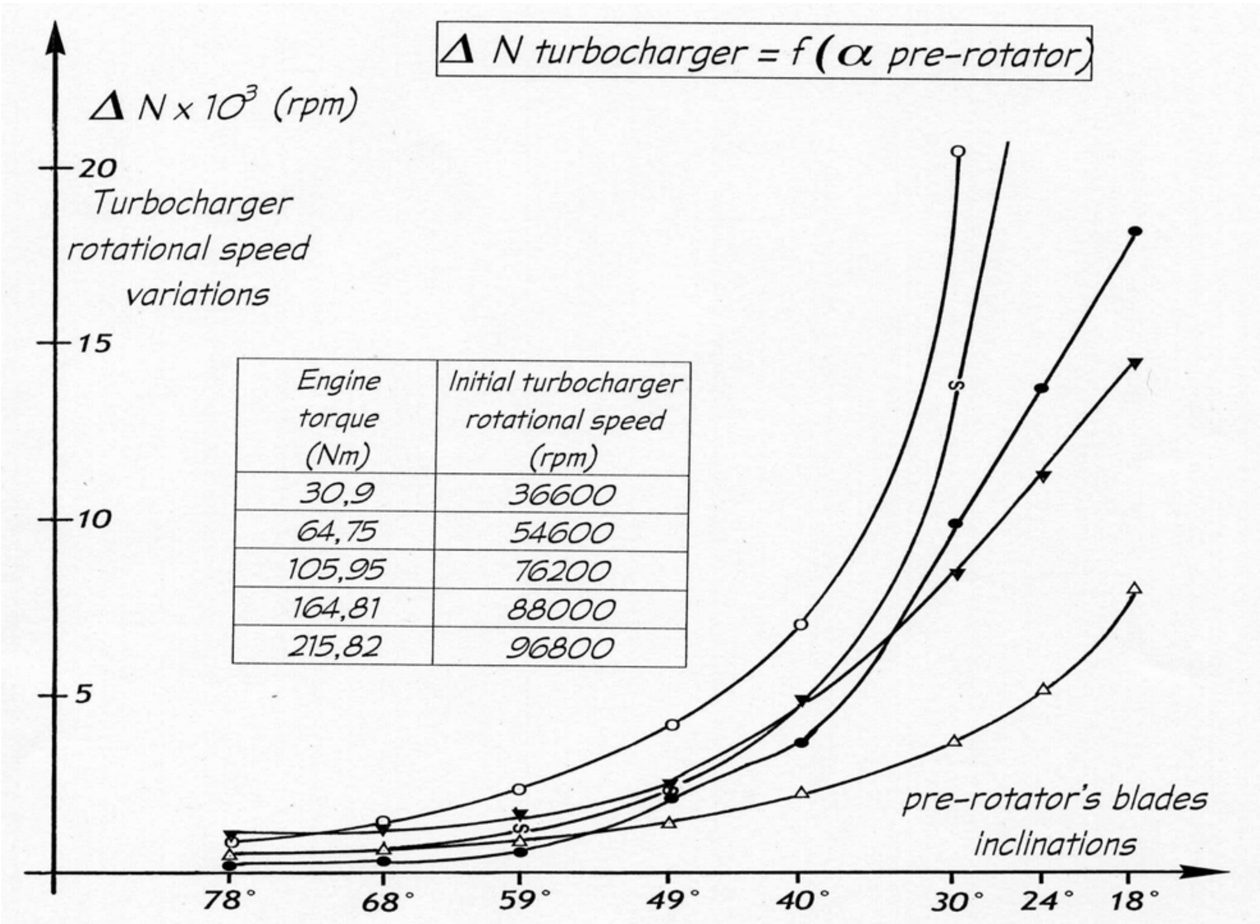


FIG 9 Turbocharger's rotational speed increase due to pre-gyration.

The extensive examination of these results, points out that isentropic efficiency and compression pressure ratio are almost constant up to the value of 40°. Beyond this value, efficiency and pressure ratio become quickly very bad, owing to the pressure loss due to the inlet guide vanes's closure. Figure 10 shows the inlet impeller velocities triangle variation according to the inlet guide vanes's blades angle.

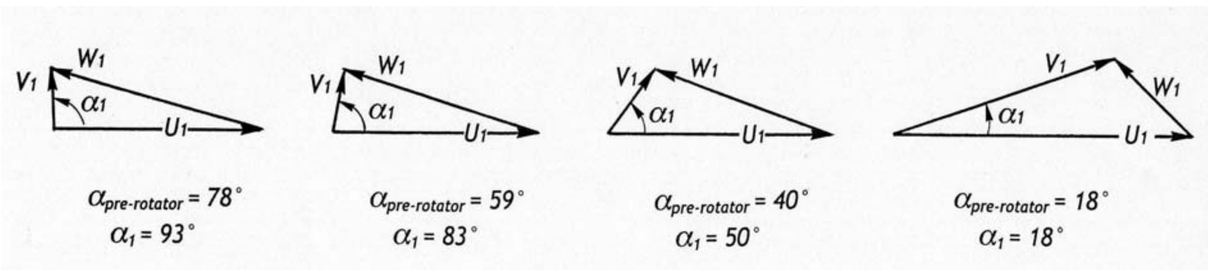


FIG 10 Variations of the velocities triangle at the inlet of the impeller with pre-swirl

We find that the $U_1 V_1 \cos \alpha_1$ value changes notably with the angle imposed by the pre-rotator. For instance, in this test, the power $q_m U_1 V_1 \cos \alpha_1$ fluctuates from 0 to 2917 watt.

CALCULATION OF THE TURBOCHARGER'S ACCELERATION TIME.

The acceleration time of a turbocharger, other wise called "turbo-lag", is the required time for going from a stabilized partial load to a stabilized full load when the engine is suddenly accelerated. This "turbo-lag" is equal to:

$$\Delta t_{acceleration} = I_{rotor} \int_{\omega_{init}}^{\omega_{max}} \frac{\omega \, d\omega}{P_{shaft}}$$

With:

Δt = acceleration time	(s)
I_{rotor} = inertia of the turbocharger's rotor	($kg.m^2$)
ω_{init} = initial rotational speed of the turbocharger	(rd/s)
ω_{max} = maximal and final rotational speed of the turbocharger	(rd/s)
P_{shaft} = available power shaft for his own acceleration	(W)

For our turbocharged engine, it is easy [1, 2 and 3] to determinate the following table:

torque (N.m)	without pre-swirl		with pre-swirl			turbo-lag decrease (%)
	ω (rd/s)	t (s)	α (°)	ω (rd/s)	t (s)	
30,9	3833	1,77	24°	4367	1,68	5%
64,75	5718	1,46	30°	6650	1,31	10,3%
105,95	7980	1,10	30°	9006	0,94	14,5%
164,81	9215	0,91	30°	10692	0,67	26,4%
215,82	10137	0,76	40°	10870	0,64	15,8%

FIG 11 Turbo-lag decrease due to pre-swirl.

The turbo-lag decrease due to pre-swirl remains unfortunately too small at low partial load and low rotational speed engine. It becomes significant at 40% partial load, where the turbo-lag decreases by approximately 15% .

CONCLUSIONS.

1) Most inlet guide vanes reduce the turbo-lag, and increases rotational speed of the turbocharger, even at low partial load.
2) Their effect becomes significant at engine partial loads above approximately 40% partial load engine.

3) The static pre-swirl has not given satisfaction. Nevertheless, it would be interesting that experiments on this model still go on.
4) However this long-lasting research leads to the development of an original pre-swirl system, with a simple technology, ratified by a patent deposit [4].
5) This study only aimed to quantify the increase turbo-lag obtained by using adjustable inlet guide vanes upstream the compressor. On no account it has been worked out to increase power. Increasing the air mass flow delivered to the engine whatever the rotational speed is no problem, and won't be obtained by regulating it with the help of a pre-swirl system.

ACKNOWLEDGEMENTS.

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