A CONTRIBUTION TO THE DEFINITION OF THE INFLUENCE OF TURBOCHARGERS ON THE DYNAMIC CHARACTERISTICS OF THE 12 CN 15/18 DIESEL ENGINE

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Abstract
The paper presents the results of laboratory testing of the diesel engine 12 CN 15/18 applied in fast caterpillar vehicles. A comparison between different combinations of turbines and compressors is given by text, table and figures. Special attention was paid to the transient regimes, the acceleration of engine. Different accelerations under criteria of load and speed were analyzed. The paper ends with recommendations for choosing turbochargers.

Key words: diesel engines, turbo charging, turbocharger, engine testing, transient regimes

Introduction
All theoretical analyses and measurements show that, under the transient regimes of engine work, turbochargers are the most inertial component. Therefore, it is interesting to see the influence of turbines and compressors on the transient regimes in various combinations. In that sense, a number of measurements, static and dynamic, were conducted on the V46 12 CN 15/18 diesel engine. Various combinations of turbines and compressors were applied. Here will be presented three combinations called:
1. **A** – “big compressor – big turbine”
2. **B** – “big compressor – small turbine”
3. **C** – “small compressor – small turbine”

The “small” compressor comparatively had a 6% smaller inducer diameter and a 7.7% greater blade height. The “small” turbine had a 13.2% smaller nominal throat, a 2.2% bigger exducer diameter and a 7.7% greater blade height. The division of the compressors into “small” and “big” was made on the base of their maps, but turbines were divided on the base of their corporal size and flow sections. Turbochargers in every combination gave almost identical stationary characteristics of the engine at the maximum of power.

Analysis
Comparing the stationary flow characteristics of both applied turbines, Fig. 1, it could be seen that the small turbine, even with a smaller throat, gives 2% greater mass flows for the same values of the pressure (p_3) and the temperature (t_3) of exhaust gases. The small turbine is also better in the field of efficiency. The figure shows shells of the best achieved results of different turbine speeds. Therefore the diagram also shows the speeds in rpm at which the best results were achieved. Continuous lines present data for the big turbine.
The analyses of the results presented in Table 1, show that on the nominal regime the turbine expansion ratio was in the range from 2.4 to 2.7. The turbocharger speed (n_{TK}) was in the range from 68,000 to 73,000 rpm. On the turbine’s maps it could be seen that for the best performances it is necessary to reach 80,000 rpm. Because turbines work at lower speeds, real efficiency is much lower. The impulsive character of the exhaust gases makes the situation even worse. Bearing in mind the conditions that existed while maps were defined and that they are only provisional, it could be said that both turbines are smaller than optimal for the regime of maximum power under stationary conditions. Because the economy is not the only criteria for such kind of engines, it is not possible to say that the chosen turbines are a bad choice for the engine.

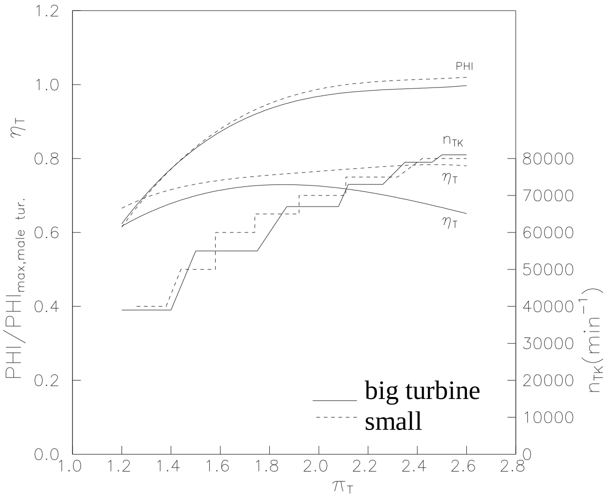

Fig. 1 Basic characteristics of the applied turbines

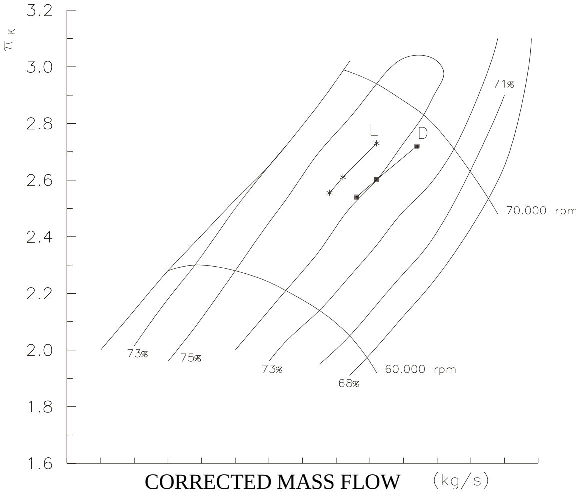

Fig. 2 Compressors at higher speeds at engine full power – Combination A (L-left, D-right)

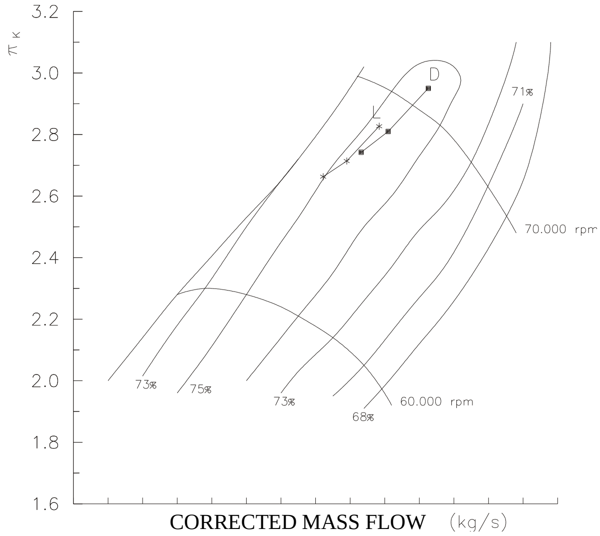

Fig. 3 Compressors at higher speeds at engine full power – Combination B (L-left, D-right)

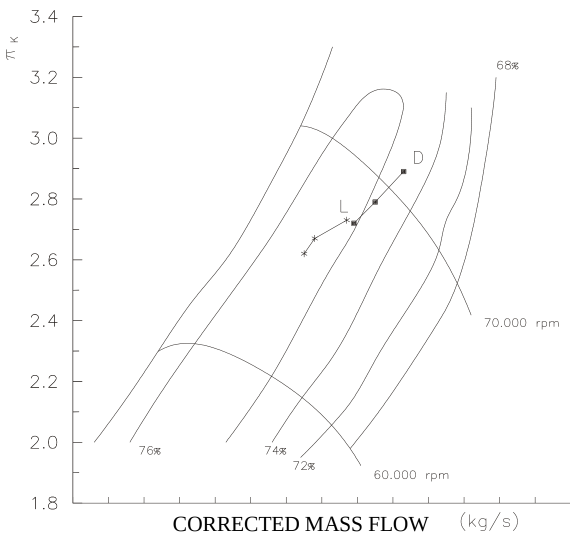

Fig. 4 Compressors at higher speeds at engine full power – Combination C (L-left, D-right)

Figures 2, 3 and 4 show the positions of compressors' working points for the engine tested at a full power in the range of higher speeds. Because the only changes on the engine were changes of the turbochargers' components, maps could be a good illustration for the influence of turbochargers on engine air supplies.
For more detailed analysis of the influences of the turbochargers' components on the engine behavior it is more convenient to use numerical values.
They are shown in Table 1.

Table 1

Combination	Size	Revolution per minute					
		$n = n_{nom}$		$n = 0.9\ n_{nom}$		$n = 0.85\ n_{nom}$	
		L	R	L	R	L	R
A	$\mathbf{m_l}$	1	1.09	0.945	1.012	0.919	0.972
	$\mathbf{n_{TC}}$	69100	68300	66960	66600	66000	65271
	$\boldsymbol{\pi_K}$	2.728	2.73	2.602	2.61	2.54	2.55
	$\boldsymbol{\pi_T}$	2.409	2.527	2.265	2.342	2.19	2.25
	$\mathbf{Me/M_{e(Pe,nom)}}$	1		1.092		1.151	
	$\mathbf{g_e}$ **(g/kWh)**	231		222		220	
B	$\mathbf{m_l}$	0.988	1.082	0.926	1.005	0.880	
	$\mathbf{n_{TC}}$	71250	72192	69031	69478	67786	
	$\boldsymbol{\pi_K}$	2.827	2.952	2.714	2.813	2.663	
	$\boldsymbol{\pi_T}$	2.7	2.69	2.51	2.51	2.39	
	$\mathbf{Me/M_{e(Pe,nom)}}$	1.012		1.110		1.166	
	$\mathbf{g_e}$ **(g/kWh)**	230		220		214.5	
C	$\mathbf{m_l}$	0.944	1.051	0.888	0.993	0.862	0.959
	$\mathbf{n_{TC}}$	72791	72928	70807	71011	69666	69276
	$\boldsymbol{\pi_K}$	2.748	2.889	2.672	2.798	2.629	2.732
	$\boldsymbol{\pi_T}$	2.643	2.655	2.469	2.628	2.401	2.424
	$\mathbf{Me/M_{e(Pe,nom)}}$	1.009		1.100		1.161	
	$\mathbf{g_e}$ **(g/kWh)**	231		221		216	

The air mass flow through the left compressor and effective torque at the maximum power of the combination **A** are taken as reference. The values of the maximum power of the engine in all combinations are equal. The differences are in the range of measuring errors and other external influences. Fuel economy is also the same.

Significant differences appear in air flows. The differences between left (L) and right (D) compressors are the result of the engine design. The right side pistons have a longer stroke.

Comparing combinations A and B it is obvious that the use of a small turbine housing results in a significant increase of turbocharger speed, but also in a reduction of air mass flow. Pressure p_2 increased its value, but pressure p_3 grew more. Ratio p_2/p_3 felt for 8%. This shows a decrease of the total efficiency of turbochargers. Because the compressors still work with good efficiency, the reason for the degradation of total efficiency is the degradation of the turbine efficiency. The second undesired but important effect is that compressors in combination B are close to pumping limit, so they could be in danger working close to the maximum torque at higher altitudes. Thus the turbine appeared as a bottleneck of the turbo charging in combination B.

The change in combination C refers to the changing of the compressor's side. As it is mentioned before, greater blades and smaller compressor air inlet are applied here. This resulted in moving out of pumping to the safe zone, achieving higher speeds of turbo chargers, but also in further reducing of air mass flows. The pressures before turbines (p_3) dropped by 2%, but, also, the pressures after the compressors dropped. Combination C has a more suitable compressor, but in this case the turbines suffer more than in combination B, so C could not be recommended for nominal power regime.

Similar analyse could be conducted for other regimes from the table. It is visible that lowering engine speed results in better specific fuel consumption at full power and in improving turbo charging system characteristics in all three combinations. This signifies that there surely exist better turbochargers for the engine working stationary at nominal power. From the analysed combinations, combination B gives the best results.

Very impressive is the enormous growth of brake torque while lowering the engine speed. The engine torque and power characteristics look as those of spark ignited engines. Such combinations could be recipes for engine elasticity improving.

It is interesting to see the influence of the mentioned combinations of turbochargers on the engine behaviour in transient conditions. Here will be presented the results of testing conducted at the test rig shown in Fig. 5.

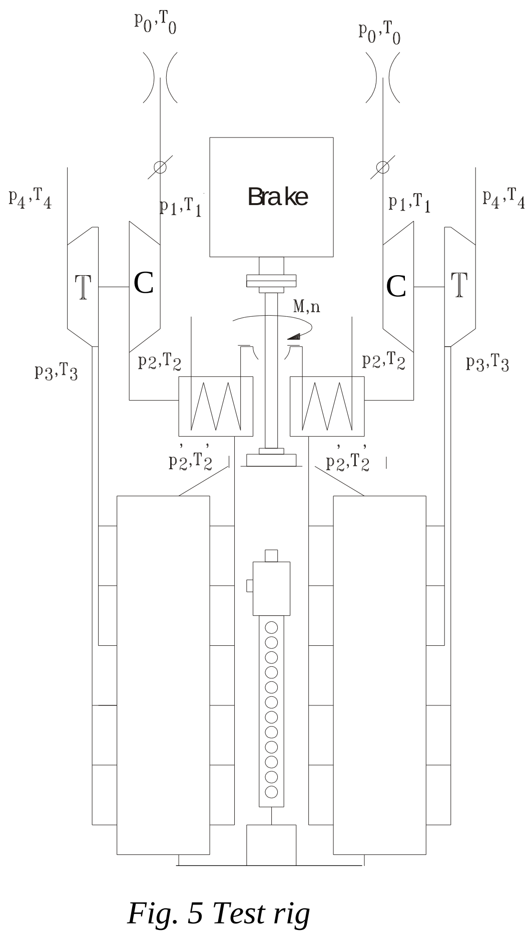

Fig. 5 Test rig

Several cases under regime M = *const* are considered:

- Engine acceleration under low values of resistance torque and various speeds;
- Engine acceleration in lower speed area under various resistance torques;
- Engine acceleration in higher speed area under various resistance torques.

Figure 6 shows the results of measuring engine speed in time for all three combinations at the regime $\boldsymbol{M} = 0.45 \cdot \boldsymbol{M}_{e(P_{e,nom})} = \boldsymbol{const}$. It is obvious that the times for reaching demanded speed are almost the same in combinations A and C, but combination B gives worse results. Absolute values could be neglected because of very small differences and possible mistakes in defining starting points, but the difference in gradients are obvious. Expected improving transient regimes in Combination B did not take place although smaller turbine housing was applied. The engine load during the acceleration was too low. A bigger inertia of turbine in Combination B compared with Combination A resulted in a lower acceleration gradient. The results were improved by the introduction of a smaller compressor in Combination C. This compressor has a bigger efficiency coefficient under lower flows, so better results were reached although the inertia of turbocharger was bigger than in Combination A. Turbines worked at partial loads so a possible better efficiency of the "small" compared to the "big" turbine did not take place.

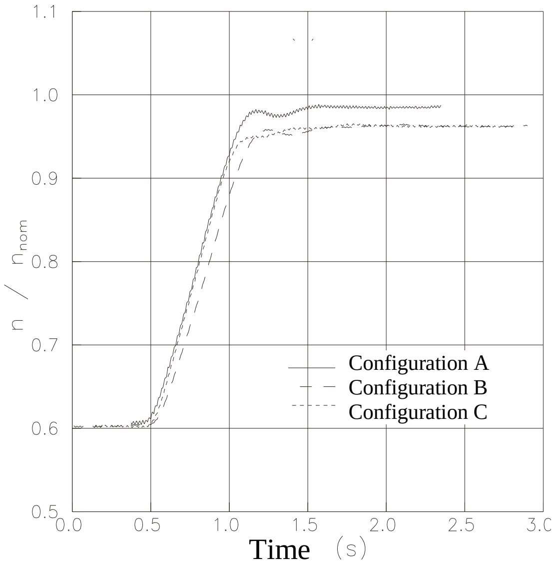

Fig. 6 The influence of turbochargers on the acceleration of the 12 CN 15/18 engine under low loads and M=const

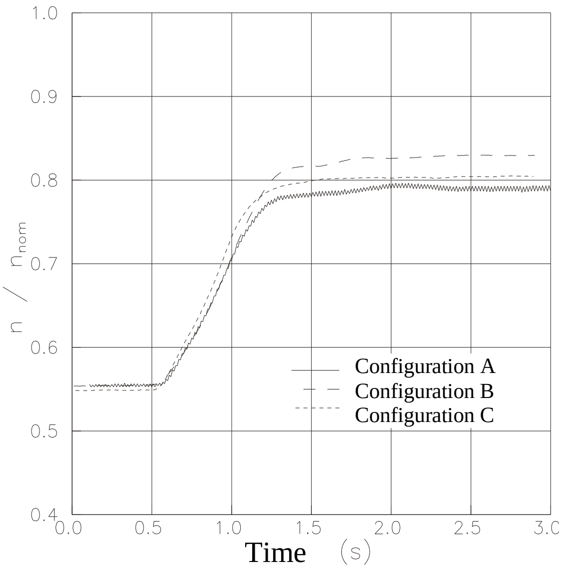

Fig. 7 The influence of turbochargers on the acceleration of the 12 CN 15/18 engine under high loads and M=const at lower speeds

The change of load changes the engine behavior in all three combinations. In the cases of engine acceleration from n = 0.55n_{nom} to n = 0.8n_{nom} under $\boldsymbol{M} = 0.75 \cdot \boldsymbol{M}_{e(P_{e,nom})} = \boldsymbol{const}$ results shown in Fig. 7 were achieved. Hydraulic brakes give no possibility to get identical results in various measurements especially when the measurements are not successive. The other parameters also have their influences on the results, so the end engine speeds at different experiments are not equal. But, from the part of curves it is possible to make a correct analysis. Combination C gave the best results, although the benefit is measured in the tenths of a second. In Fig. 7 different gradients compared with Fig. 6 are visible. While the gradients in Fig. 6 are mostly linear, in Fig. 7 after a short linearity a crook appeared leading to higher accelerations. A delay of approximately 0.25s shows on turbochargers' influence. This phenomenon is visible also in the engine Configuration C working on high torque regimes $\boldsymbol{M} = 0.8 \cdot \boldsymbol{M}_{e(P_{e,nom})} = \boldsymbol{const}$ at upper engine speeds, Fig. 8. The "small" compressor during this transient regime worked on higher pressures and efficiencies. These resulted in a bigger air mass in cylinders in the first phase of transition so more favorable conditions for the combustion of additional fuel existed. The compressor worked efficiently and the realized increasing of turbine power was utilized for the turbocharger rotor acceleration. This is the reason that the curve of engine speed does not show any crook and nonlinearity.

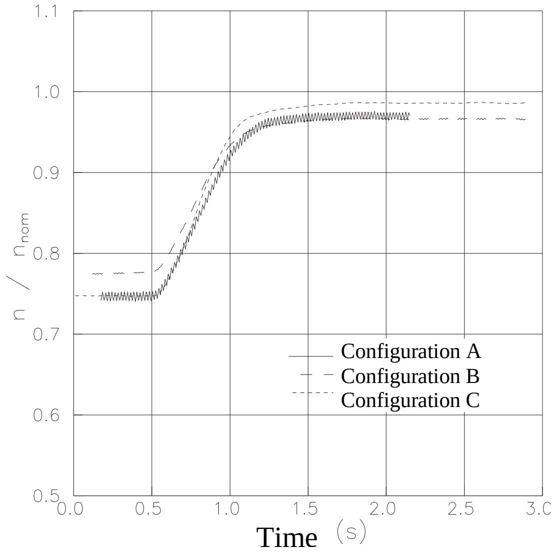

Fig. 8 The influence of turbochargers on the acceleration of the 12 CN 15/18 engine under high loads and M=const at higher speeds

Configuration A gave good results at the very beginning of the transition, but in the next phase had lost the race. A better efficiency of compressors in C exceeded a bigger inertia of the turbocharger rotor.

Conclusion

Comparing the results achieved by stationary and dynamic testing of turbochargers' influence on the engine performances it could be concluded that it is possible to reach a better dynamic response by using a compressor with a smaller wheel than an optimum for stationary characteristics. The economy is not essentially reduced. The use of smaller turbine housing gives no advantages in dynamic response. This conclusion is valid only in cases when the turbocharger is adequately chosen for the engine when the main criterion was achieving the best results at maximum power. In other cases such as using by pass valve it is necessary to make a special analysis.

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A Contribution to the Definition of the Influence of Turbochargers on the Dynamic Characteristics of the 12 CN 15/18 Diesel Engine

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