A Technical Analysis of Heat Phenomena of the Cyclical Synchronization Power and Geometrical Parameters of the Turbocharging System of a Diesel Engine

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터보과급 디젤엔진의 싸이클 동력동기화 및 형상변수에 대한 열현상의 기술적 분석

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요 약

엔진 배기가스의 동력과 유량이 배기행정의 직전 단계에서 관찰되었다. 배기가스 양을 적당히 조정함으로써 터보 과급의 입구 압력을 증가 시킬수 있었으며 엔진의 흡기, 소기 및 배기과정에서 가스 질량과 엔진의 동력, 그리고 터보과급 효과도 감소하였다. 터보 과급장치를 기하학적으로 적절화시킴으로써 싸이클의 동기화 및 동력의 효율이 고려된 열교환 과정의 효율 기준도 제기되었으며 디젤 엔진의 연소싸이클을 재수정하는 과정과 터빈의 동역학적 특성도 제시되었다.

SYMBOLS AND ABBREBIATIONS

G_{ij} : mass flow rate of the gas relative to specific exhaust phases [m^3/s]

H_{ij} : gas flow net power relative to specific exhaust phases [m^3/s]

R : gas constant number.

T_{tij} : the gas temperature front of the turbine [°K]

η_{ij} : internal efficiency of the turbine

i : index of the current crank shaft angle

H_{ij} : gas flow net power relative to specific exhaust phases [m^3/s]

η_{ij} : internal efficiency of turbocharger turbine during specific exhaust phases.

$(\mu F_T)^2_{ij}$: admission capacity

T^*_{gij} : total temperature [°K]

p^*_g : pressure of the gas at the front of the turbocharger turbine [m^2/s]

c_p : gas specific heat capacity

K_g : gas isentropic number

p_2, ρ_2 : pressure, density at the turbocharger turbine outlet $[m^2/s]$

ζ_1 : coefficient of nozzle losses

ζ_2 : coefficient in the wheel blades

ζ_k : coefficient in the turbocharger turbine chamber

1. INTRODUCTION

In conditions of variable, partial load, diesels with medium pressure turbocharging systems are most preferred. Most engines for fishing ships and other commercial vehicles are equipped with impulse turbocharging systems.

Air deficiency problems become more pronounced when the pulse gain is reduced, and when turbocharger efficiency decreases under partial load. Diesels with impulse turbocharging systems are characterized by great losses of power from their turbochargers. There is a conflict between the cyclical operation of the pistons and continuous flow of the turbocharger.

Changing of phases are used for decreasing fluctuation of the gas pressure amplitude, but usually they do not give desirable results, because decreasing of the amplitude is accompanied by losses of efficiency and power in turbocharging systems.

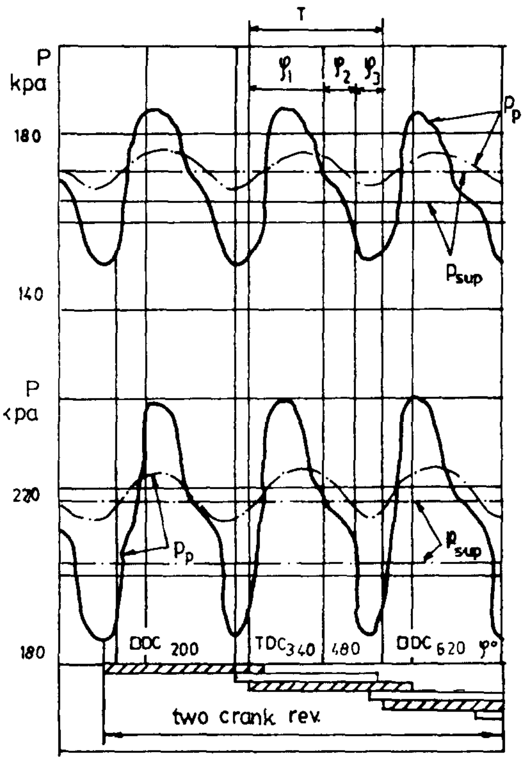
2. THEORETICAL ANALYSIS

More efficient use of gas flow power could be achieved by redistribution of gas flow energy in exhaust stroke phases of diesel piston engines. Redistribution is accompanied by decreasing of the exhaust gas pressure fluctuation. During a single exhaust stroke from a cylinder, mass quantity and power of the gas flow is changed. There is a surplus of mass quantity and power of the gas flow in the beginning phase of the exhaust stroke. This phase is known as the preliminary and free part of the exhaust process. A surplus build up of gas volume is evident in front of the nozzle area and there is increasing pressure in front of the turbine, which entails exhaust back pressure and a decrease of turbocharger efficiency, therefore much of this energy is not used.

There is a shortage of mass quantity and power from the gas flow at the forced exhaust and scavenging phases. A deficiency of mass quantity and gas power entails a reduction of turbine power and consequent decreasing of turbocharger efficiency.

A lack of conformity of the nozzle area to the gas flow quantity during the gas exhaust phase from every cylinder decreases efficiency of the turbocharging system and the diesel engine as a whole. There is a possibility to improve utilization of the exhaust gas power for turbocharging by dividing and redistributing flow power phases ϕ_1, ϕ_2, ϕ_3 of gas exhaust from cylinders. Turbine's power, N_{ts} during one cycle of the exhaust is augmented by three power phases. The first power phase, N_{t1} at the preliminary and free part of the exhaust period : the second power phase of the free exhaust flow : the third power phase, N_{t3}, at the scavenging period and induction stage. The dividing and redistributing power characteristics of the gas flow are not defined by exact boundaries.

The diagrams in Fig. 1 illustrate the pressure of the gases and the phases ϕ_1, ϕ_2, ϕ_3 in two exhaust manifolds near the exhaust valves of a turbocharged six - cylinder engine. In the beginning phase through the opening exhaust valve, the mass flow rate is increased rapidly from minimum to maximum values. In the period when mass flow rate is at its highest, the turbine nozzle area is required to be at its highest too. In the period when mass flow rate becomes minimal, the turbine nozzle area is



exaust valve opened
inlet valve opened

Fig. 1 Pressure in the exhaust manifold of six cylinder diesel under two Loads. Psup - air in inLet pipe ; Pp - pressure in exhaust pipe ; ϕ_1 - pre Liminary and free part exhaust ; ϕ_2 - second haLf free exhaust and compalsory exhaust, ϕ_3 - scavenging and filling cylinder.(—) without synchronization ; (---) with sinchronization

required to be minimal too. The co - ordination of the turbine nozzle area with the mass flow rate during each exhaust phase from every cylinder requires cyclical synchronization. During each cycle, the adjustment must be made for every cylinder. For two - stroke engines during one revolution, for four - stroke motors during two revolutions. Number of the cycles are equal to the number of cylinders of the engine. For example in a six - cylinder turbocharged diesel during two revolutions the accommodation is made six times. One possibility to increase power of the turbocharger is reached by redistributing and recombining the total power of every exhaust phase to the turbine in every cycle period T.

$$N_{TS}=N_{T1}+N_{T2}+N_{T3} = \sum_{\phi_1} G_{ij}\mu_{ij}\eta_{ij} + \sum_{\phi_2} G_{ij}\mu_{ij}\eta_{ij} + \sum_{\phi_3} G_{ij}\mu_{ij}\eta_{ij} \qquad (1)$$

Synchronization gives a possibility simultaneously to increase the efficiency and power of the turbine during the preliminary phase and free part of the exhaust ϕ_1 and in the scavenging and induction phases ϕ_3. During preliminary and getting out sub - phase of the exhaust period, power of the turbine is increased by enlarging the turbine nozzle and by increasing a passage permitting increased mass flow rate through the turbocharger turbine. During scavenging and induction phases, turbine power is increased by reducing the nozzle area, which increases gas flow net power.

The gas flow net power

$$H_{ij} = \frac{G_{ij}^2}{2(\mu F_T)_{ij}^2 \rho_2} = c_p T^*_{gij}\left[1-\left(\frac{p_2}{p^*_g}\right)^{\frac{K_g-1}{K_g}}\right] \qquad (2)$$

In Fig. 2(part a), the change in nozzle area is shown during one cycle T of the exhaust phase from the cylinder. The turbine nozzle area is changed by a staff drive. This synchronization method is simpler and may be effected by two disks on a shaft, connecting with the crankshaft by a gear transmission (part b) shows the changing pressure at the front of the turbine during one cycle T. The pointer 1 shows the pressure front of the turbine without synchronization of the nozzle assembly area. The pointer 2 shows the pressure at the front of the turbine with synchronization of the nozzle area. In ordinary turbocharger systems a turbine nozzle during all phases of the exhaust

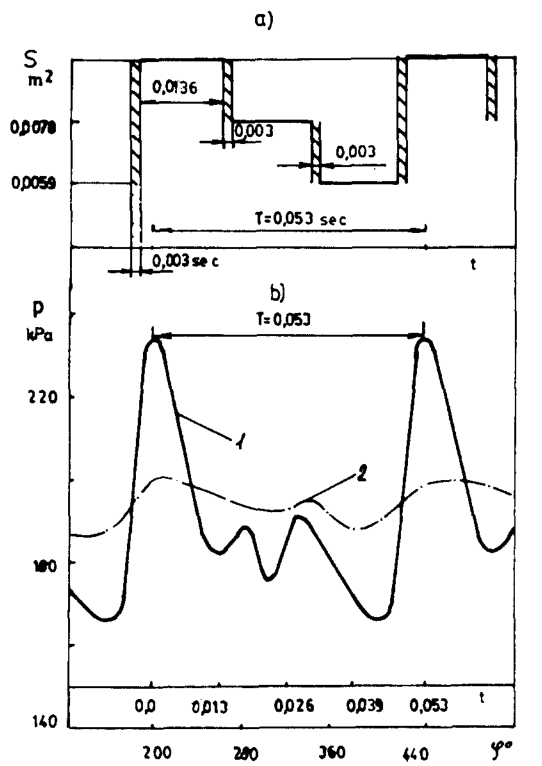


Fig. 2 The parameters in turbine
a) change turbine nozzle area
b) change pressure before turbine ; 1 - without synchronization ; 2 - with syncronization

remains constant. In scavenging and induction phases for example operating processes of the turbine are characterized by the diversion of the flow direction to the turbine blade wheel leading edges. It is known low pressure in the front of the turbine forms a small absolute velocity c_1, from the nozzle assembly. The direction of this absolute velocity remains constant, consequently, the relative velocity w, of the gases entering the rotary blade wheel can be changed not only in magnitude but in direction β_1, too due to the constant peripheral velocity u_1 of the rotary wheel. The non coordinated changes of direction $(\beta_1' - \beta_1)$ velocity w_1 forms impact losses on the inlet side and brakes the rotary blade wheel, as shown in Fig. 3. shows

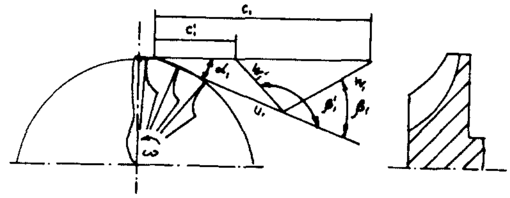


Fig. 3 Changing feow conditions of the fLad wheel of radial turbine, without synchronization

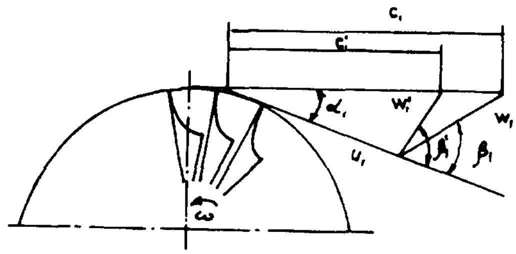


Fig. 4 Changing flow conditions of the bLade whel of radiae flow turbine, with synchronization

the flow conditions of the blade wheel of a radial flow turbine was changed by when the gas pressure during the exhaust cycle cause absolute velocity flow from turbine nozzle c_1 ; relative velocity w_1 ; peripheral speed u_1 ; inlet angle of absolute velocity α_1 ; inlet angle of relative velocity β_1.

The labels with $'$ show geometrical flow parameters when pressure before the nozzle assembly correspondd with scavenging and induction conditions. The labels without($'$) show geometrical flow parameters when pressure before nozzle assembly corresponds with preliminary and free part of the exhaust phases from the cylinder.

The turbine internal efficiency

$$\eta_{ij} = 1 - \zeta_1 \frac{c_1^2}{2} - \zeta_2 \frac{w_2^2}{2} - \frac{c_2^2}{2} - \zeta_k \qquad (3)$$

In Fig. 4 shows changing of flow conditions of the blade wheel of a radial flow turbine with cyclical synchronization are given.

3. DISCUSSION

If during preliminary and free part sub-phases of the exhaust phase, the nozzle assembly causes more than standard, then absolute gas velocity c_1 will be decreased. If during scavenging and induction phases of the cylinder, the nozzle is smaller than standard, then absolute velocity c_1 will be increased. As shown in Fig. 5 and 6, the fluctuation of absolute velocity c_1 and relative velocity x_1 are decreased. The impact losses on the reverse side and braking effect are decreased too, because the fluctuation of the relative velocity directions is decreased.

A technical embodiment of the cyclical synchronization principle is possible with the help of a closing device connecting with the crankshaft. It is fitted in a gas passage channel near the turbine nozzle assembly. A connection with the diesel crankshaft may be achieved by mechanical or electro-mechanical transmission; the partial closing of the turbine nozzle assembly is coordinated with scavenge and induction phases. A necessary difference between pressure in the fresh air intake and the cylinder in this period is achieved by increasing the compressor pressure. The cyclical synchronization parameters of a turbocharger system achieve performance improvement of medium and high power diesels used in conditions of variable loading.

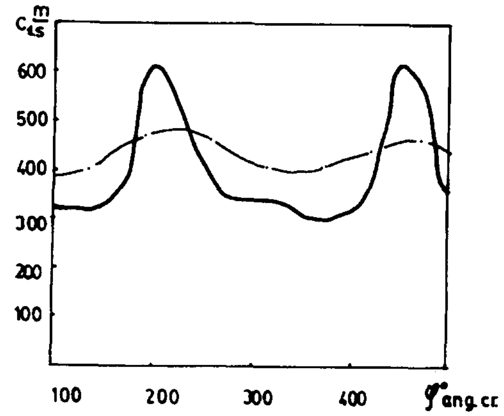


Fig. 5 Changing absolute flow velocity from turbine nozzle c_1 (—) without synchronization : (- · -) with synchronization

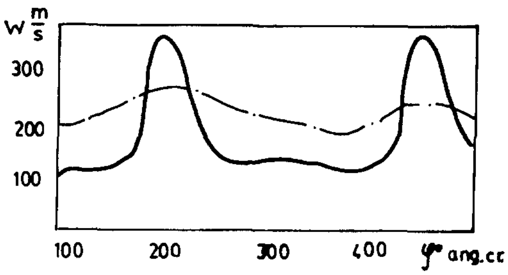


Fig. 6 Changing relative velocity w₁ (—) without synchronization : (_ . _) with synchronization

4. CONCLUSIONS

1. These new principles of heat management decrease power losses of exhaust gas in tur-

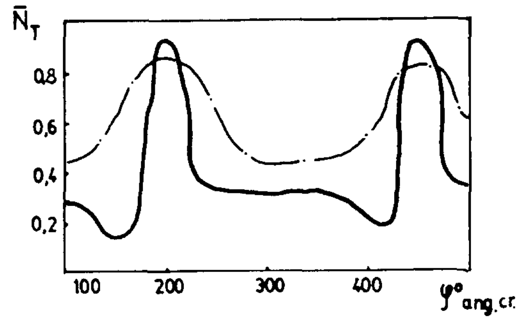


Fig. 7 Relative power of turbine (—) without synchronization : (- · -) with synchronization

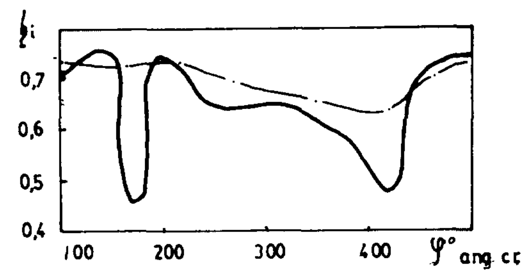


Fig. 8 Efficiency of the turbine (—) without synchronization (· ·) with synchronization

bocharging systems by redistribution of the heat parameters in the exhaust phases.

2. The synchronization of the parameters gives a possibility to decrease fluctuation of the specific gas flow net power during the corresponding exhaust phase which is used in the turbine.

3. The changing of the turbine admission capacity in the exhaust phases decreases fluctuation relative velocity w_1, and outlet velocity c_2 from the turbine and decreases accordingly losses occurring in the wheel as well as outlet velocity.

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