The Internal EGR in the Z engine

All today's passenger car diesel engines have external EGR in order to reduce NO_x emissions. In those engines, the EGR gas is taken from the exhaust manifold before the turbo charger and it is directed through the EGR cooler into the intake manifold. This means that the intake manifold pressure cannot exceed the exhaust manifold pressure to make the EGR flow possible. During the exhaust cycle, the piston pushes all the exhaust gases out from the cylinder against the exhaust manifold pressure. Because the intake pressure is lower than the exhaust pressure, the positive transport work received by the piston during intake is same or lower than the exhaust gas transport work. This means that the turbo charger doesn't rise the process efficiency of the engine.

There are also some other disadvantages of the external EGR. The carbon particles can gather in the EGR cooler, in the intake manifold and on the intake valves. Also, there is the risk of the corrosion in those parts.

In the case of internal EGR in the Z engine, a portion of exhaust gas, for example 15-30%, remains in the cylinder and the residual gas is then mixed with the incoming fresh air charge. The amount of residual gas can be controlled by phasing the camshaft. A 2-cylinder Z engine, equal to a 4-cylinder diesel engine, is an ideal case to use an impulse turbo charger as the exhaust pressure pulses follow after 180° each others. In the case of the impulse turbo, the pressure difference over the turbine after the "blow down" is almost negligible. This means that, the piston doesn't have to work against relatively high exhaust manifold pressure while rising to the top dead center during exhaust cycle. Lower piston work during exhaust cycle makes better efficiency possible.

The modern turbo chargers can reach pressures of 4 bar (absolute pressure) and even higher. It is not common to have turbo pressures over 2 bar in passenger car diesel engines. In the Z engine, it is useful to use as high turbo pressure as possible. At full load, the turbo pressure can be 4 bar in the Z engine. The high turbo pressure reduces the size and the power consumption of the piston compressor. This increases the efficiency of the Z engine for its part.

At part load, the turbo charger pressure drops until the atmosphere pressure, thus reducing the mass flow over the engine. This means that it is possible to keep the air/fuel equivalence ratio low even at part load without throttling the intake manifold in the Z engine. An unwanted waste of the energy to compression can be avoided and the low air/fuel equivalence ratio means higher temperature rise during the combustion. This rises the part load efficiency of the Z engine compared to the diesel engine.

Because the turbocharger doesn't increase the process efficiency of common turbocharged diesel engine, an easy way to study the internal EGR of the Z engine is to compare the compression with a naturally aspirated diesel engine. To achieve comparable results the comparison point has to be exactly the same (temperature, pressure and the gas composition). Let's choose 10 bar (absolute) for the comparison point pressure and for the gas exchange pressure of the Z engine. Exhaust gas air/fuel equivalence ratio is set to 1,5. The exhaust gas temperature before scavenging is assumed be 900 K and the average pressure from BDC to start of scavenging is assumed to be 1,3 bar (absolute). The intake temperature after the 2nd intercooler is altered to match the temperature of the diesel engine at the comparison point. Two cases, 3 bar and 4 bar (2 bar and 3 bar overpressure), are calculated. The results are in

the table. The specific works are calculated for the mass of the comparison point (not to match the intake mass flow). The specific piston works $q_{p,exh}$ and $q_{p,sca}$ include the work against the atmospheric pressure.

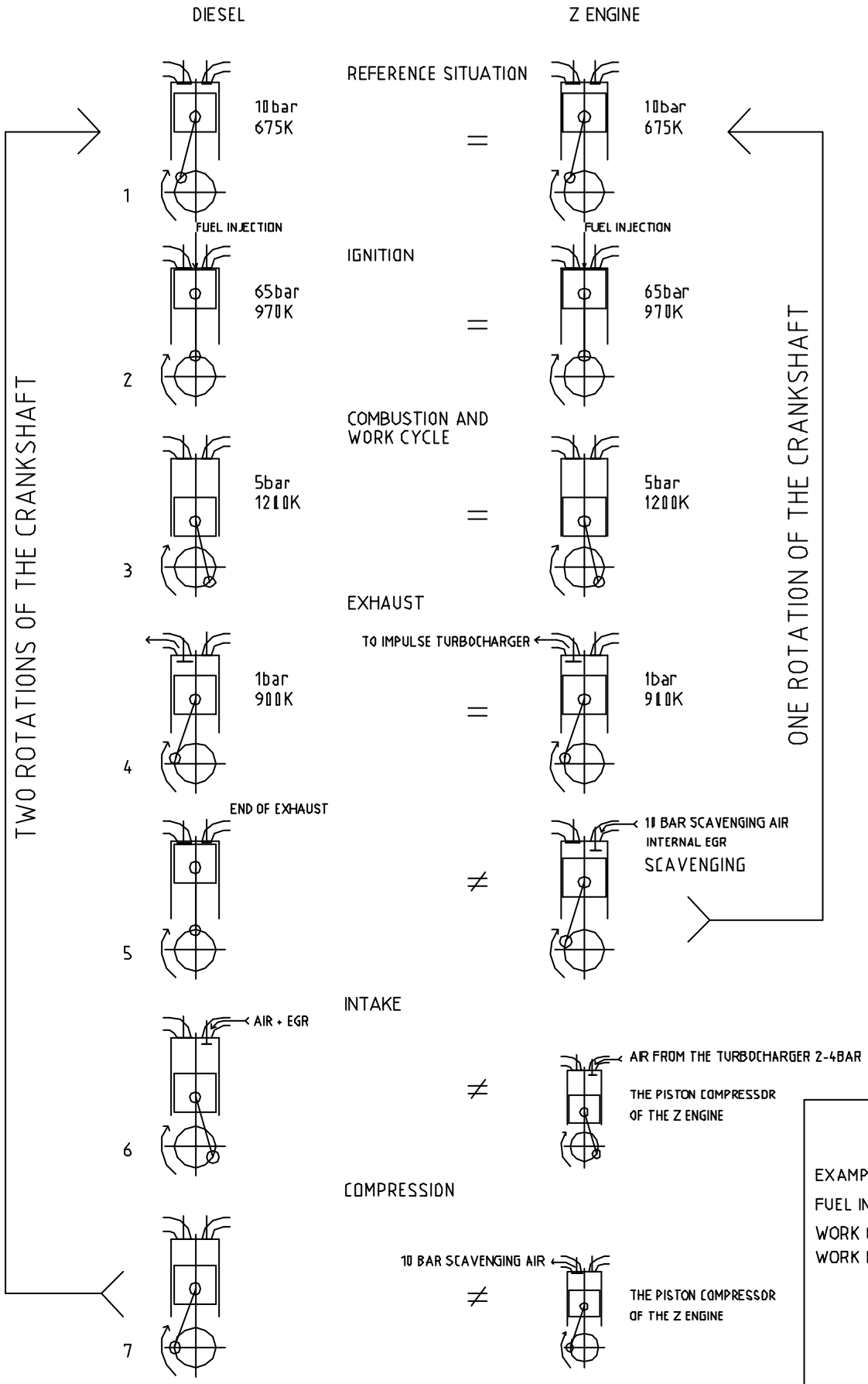
Turbo pressure, (bar, absolute)	3,0	4,0
Temperature after IC2 (K)	362,1	362,1
Piston compressor specific compression work into gas, q_{3-4} (kJ/kg)	69,2	50,4
Piston comp specific transport work q_{tr2} (kJ/kg)	95,0	87,5
Piston compressor specific positive transport work (kJ/kg)	-67,4	-67,4
Specific piston work BDC - SS, $q_{p,exh}$ (kJ/kg)	54,8	54,8
Specific piston work during SS-ES, $q_{p,sca}$ (kJ/kg)	59,9	59,9
Total specific work to achieve the comparison point (kJ/kg)	**211,5**	**185,2**
Air/fuel equivalence ratio at the comparison point (-)	7,05	7,05
Calculated EGR rate (m_{EGR}/m_{tot}) (-)	0,22	0,22
Temperature at the comparison point (K)	675,74	675,74

For the diesel engine, the intake temperature is assumed to be 300 K. The temperature rise before compression (in the intake manifold and the cylinder) is assumed to be 50 K. The calculated specific compression work to the comparison point is 234,5 kJ/kg. The specific work includes the work against the atmospheric pressure.

As seen from above, the Z engine reaches the point 10 bar in the work cylinder with less work than the diesel engine when EGR is needed. This means that the Z engine has a chance for higher efficiency than the diesel engine, because from this point both engines can do the rest of the compression with the same work and the combustion and the work cycle can be carried out in similar way. The turbo charger doesn't change this situation, because there is an external EGR in the diesel engines and the turbo charger doesn't rise the efficiency of the engine.

There is a comparison chart of the Z engine and naturally aspirated diesel engine on the next page.

COMPARISION BETWEEN THE CYCLES OF DIESEL AND Z ENGINE



EXAMPLE: 1g AIR

FUEL INJECTION = 2010J

WORK OUT = 810J DIESEL

WORK OUT = 823–851J Z ENGINE

PHASE 5-7:

DIESEL = 235J

Z ENGINE = 185–212J