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TURBOCHARGING EFFICIENCIES -

DEFINITIONS AND GUIDELINES FOR MEASUREMENT AND CALCULATION

The International Council
on Combustion Engines

Conseil International des
Machines à Combustion

CIMAC is an international organisation, founded in 1950 by a French initiative to promote technical and scientific knowledge in the field of internal combustion engines (piston engines and gas turbines). This is achieved by the organisation of congresses and working groups.

It is supported by engine manufacturers, engine users, technical universitites, research institutes, component suppliers, fuel and lubricating oil suppliers and several other interested parties.

The National Member Associations (NMAs), National Member Groups (NMGs) and Corporate Members (CMs) as well as previous CIMAC Recommendations are listed in the back of this publication.

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FOREWORD BY THE PRESIDENT

This Recommendation is addressed to engine and turbocharger manufacturers.

The turbocharger is a fundamental component for modern large combustion engines. An important parameter to check the performance of the turbocharger is the efficiency. The turbocharging system efficiency takes into account all losses in the turbosystem except those of the turbocharger itself. Since the turbocharger efficiency is difficult to measure on an engine, a valuable alternative can be to derive it from the turbocharging efficiency.

The Recommendation is an extensive and precise theoretical approach with practical examples about the design and calculation of the turbosystems for two stroke and four stroke engines. In order to obtain a common basis of understanding, the definitions must be exact and unambiguous taking into account all relevant effects. For the daily work numerically simplified formulae can be derived.

I am convinced that this Recommendation will be widely used by engine and turbosystem designers and researchers. My sincere thanks to the members of the working group for the efforts to gather the new Recommendation.

Matti Kleimola, President
May 2007

Content

0 Introduction

The turbocharger is a crucial component of modern large combustion engines. An important parameter for checking the performance of the turbocharger is its efficiency.

Turbocharger manufacturers use an efficiency definition derived from the thermodynamics of flow machines which aims to describe the behaviour of the turbocharger under controlled conditions and without any disturbing effects.

Engine manufacturers need a more practical approach, that is an easily understood definition which does not require reference to thermodynamic tables or functions and contains information about the effectiveness of the turbocharging on the engine.

This Recommendation is addressed to engine and turbocharger manufacturers. In order to obtain a common basis of understanding the definitions must be unambiguous and exact, taking into account all relevant effects. For the daily work numerically simplified formulae can be derived from the definitions given here which allow the efficiencies for a given class of engines to be calculated in a simpler way. Some suggestions for doing this are given in the annexes.

0.1 Notation

c	[m/s]	Velocity, linear speed
c_k	[m/s]	Mean piston speed
c_p	[J/(kg K)]	Constant pressure specific heat
C_{fuel}	[-]	Correction for fuel
D	[m]	Cylinder bore
e_P	[J/kg]	Expergy
$\dot{m}$	[kg/s]	Mass flow rate
p	[bar]	Pressure
R	[J/(kg K)]	Gas constant
T	[K]	Temperature
x_c	[kg/kg]	Combustion gas mass fraction
Δh_s	[J/kg]	Isentropic enthalpy head
η	[-]	Efficiency
κ	[-]	Ratio of specific heats
ρ	[kg/m^3]	Density

Superscripts

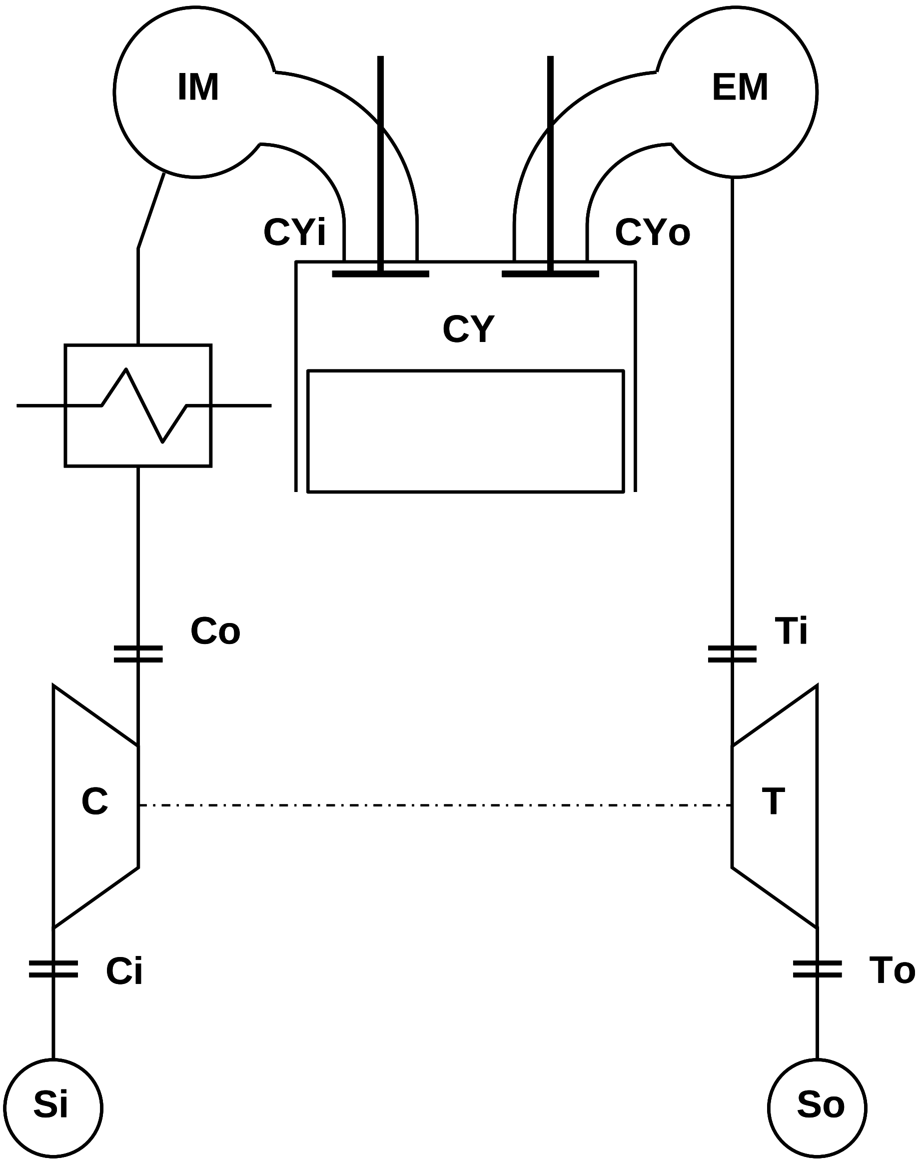
$\bar{}$ (overbar)	Mean value
*	Conventional total value
$'$	First approximation
$''$	Second approximation

Subscripts

1	generic, before compression
2	generic, after compression
3	generic, before expansion
4	generic, after expansion
A	Air
Amb	Ambient
App	Apparent
Com	Compression
eq	Equivalent
Exp	Expansion
F	At flange
G	Exhaust gas
HP	High pressure
L	Pipe
LP	Low pressure
$pulse$	Pulse system
SPS	Single pipe system
$\perp$	Normal to control area

Remark: The notation given here is reduced to the essential. Exact definitions of subscripts for the boundaries are given under the following point 0.2.
Derived definitions are described in the context.

0.2 Control positions



CY	Cylinder		
CYi	Cylinder inlet	CYo	Cylinder outlet
IM	Inlet manifold	EM	Exhaust manifold
Co	Compressor outlet	Ti	Turbine inlet
C	Compressor	T	Turbine
Ci	Compressor inlet	To	Turbine outlet
Si	System inlet	So	System outlet

1 General definitions

The general form for the definition of efficiency of a turbocharging process is always based on the comparison of two ideal processes:

- The isentropic compression of the working medium (air or mixture) from an initial state 1 to a final state 2
- The isentropic expansion of the exhaust gas from an initial state 3 to a final state 4.

The states 1, 2, 3, 4 indicate here generic states of start and end of the compression and the expansion process, respectively, and will be substituted by precise definitions in the following.

For the formulation of the efficiency it is convenient to use a special form of potential energy[1]:

$$e_{P,Com}\left(T_1,\frac{p_2}{p_1}\right)=\Delta h_{s,Com}\left(T_1,\frac{p_2}{p_1}\right) \tag{1}$$

$$e_{P;Exp}\left(T_3,\frac{p_3}{p_4}\right)=\Delta h_{s,Exp}\left(T_3,\frac{p_3}{p_4}\right) \tag{2}$$

If the flow is pulsating, the energy flow should be integrated over the cycle. In this case mean values can be used: the mean flow rates $\bar{\dot{m}}$, the mean temperatures $\bar{T}$ derived from the enthalpy conservation in the flow and the equivalent pressures p_{eq} fulfilling the conservation of the expergy with reference to a constant value, preferably the ambient pressure. More details concerning these mean values are given in Annex 1.

The general definition of efficiency for a turbocharging process is then:

$$\eta=\frac{\int\limits_{Cycle} e_{P,Com}\left(T_1,\frac{p_2}{p_1}\right)\cdot \dot{m}_1(t)\cdot dt}{\int\limits_{Cycle} e_{P,Exp}\left(T_3,\frac{p_3}{p_4}\right)\cdot \dot{m}_2(t)\cdot dt}=\frac{\bar{\dot{m}}_1\cdot e_{P,Com}\left(\bar{T}_1,\frac{p_{2eq}}{p_{1eq}}\right)}{\bar{\dot{m}}_2\cdot e_{P,Exp}\left(\bar{T}_3,\frac{p_{3eq}}{p_{4eq}}\right)} \tag{3}$$

The general definition can be applied to different system boundaries (bare turbocharger, air and gas manifolds, cylinders). In order to avoid conflict, it seems reasonable to define boundaries that coincide with the limits of responsibility (design and hardware) of the parties involved. In the following points the exact and approximate definitions for practical use are given, using Eq. (3) with different boundaries.

The various thermodynamic states can be taken as static or total (stagnation), whereby the real total state is used very seldom in engineering problems due to averaging problems. Therefore, in the following the superscript * denotes a conventional total state, defined using the static values and the mean flow velocity. For instance, for the pressure at a flange with area A:

$$p^*=p+\frac{1}{2}\rho\cdot\bar{c}_\perp^2=p+\frac{1}{2\cdot\rho}\left(\frac{\dot{m}}{A}\right)^2=p+\frac{R\cdot T}{2\cdot p}\left(\frac{\dot{m}}{A}\right)^2$$

For more details s. Annex 3

[1] The enthalpy difference in an isentropic compression or expansion process has been used by many authors [3, 5] for evaluations, but none of them has given a short name for it. Since this energy is the fraction of the exergy relevant for machines working with pressure differences, it is proposed to call it: ex-P-ergy. The expergy e_P is a state function which, multiplied by the mass flow rate, gives the isentropic shaft work potential.

1.1 Turbocharger efficiency

Exact definition:

$$\eta_{TC} = \frac{\overline{\dot{m}}_{Co} \cdot e_{P,Com}\left(\overline{T}^*_{Ci}, \frac{p^*_{Co,eq}}{p^*_{Ci,eq}}\right)}{\overline{\dot{m}}_{Ti} \cdot e_{P,Exp}\left(\overline{T}^*_{Ti}, \frac{p^*_{Ti,eq}}{p_{To,eq}}\right)} \tag{4}$$

Formula for approximate calculation:

$$\eta_{TC} = \frac{\overline{\dot{m}}_{Co} \frac{\kappa_A}{\kappa_A - 1} R_A \cdot \overline{T}^*_{Ci} \cdot \left[\left(\frac{p^*_{Co,eq}}{p^*_{Ci,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1\right]}{\overline{\dot{m}}_{Ti} \frac{\kappa_G}{\kappa_G - 1} R_G \cdot \overline{T}^*_{Ti} \cdot \left[1 - \left(\frac{p^*_{Ti,eq}}{p_{To,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]} \tag{4'}$$

Recommendations for the choice of appropriate values for κ and R for air and exhaust gas are given in Annex 2.

This turbocharger efficiency uses the physical boundaries of the turbocharger, which are the flanges or the ambient when the air inlet filter is part of the turbocharger. The detailed definitions of all items in Eq. (4) are given in the following table:

$\overline{\dot{m}}_{Co}$	Mass flow at compressor outlet flange. Small amounts of air lost through leakage or used for cooling or other purposes belong to the losses and have not to be considered in the efficiency calculation. **Exception**: When a part of the air mass flow is removed after the outlet flange (typically after the air cooler) and readmitted into the compressor (for cooling, sealing, etc.), the net compressor flow rate must be correspondingly reduced.
$\overline{\dot{m}}_{Ti}$	Mass flow at turbine inlet flange.
Ci*	Total* state at the compressor inlet boundary (inlet flange). In cases where the filter-silencer belongs to the turbocharger, the inlet state is given by the ambient pressure in the engine room and the mean temperature at the filter. This can be higher than the ambient temperature due to air conditioning, heat transfer from the machinery, forced ventilation, etc.
Co*	Total* state at the compressor outlet flange.
Ti*	Total* state at the turbine inlet flange.
To	Static state at the turbine outlet flange.

The combination “t-t, t-s” (total-to-total for the compressor and total-to-static for the turbine) has been most commonly used for turbochargers, the reason given being that the kinetic energy after the turbine is hardly usable. There is improvement potential (s. 3.1 – Turbine outlet), but the definition has been retained for historical reasons.

Actually, the kinetic energy of the gas would be lost only if the turbine were to discharge directly into the ambient. In real plants, there is at least a pipe always present after a turbine, but often other devices also can be found (heat exchanger, evaporators, boilers, catalysts, silencers, power turbines …). The total pressure after the turbocharger turbine is relevant for the dimensioning of these..

In the case of two-stage turbocharging, the total outlet state of the HP turbine coincides, ignoring the pressure losses, with the total inlet state for the LP turbine. In this case it is imperative to use the total pressure at the turbine outlet in the definition:

$$\eta_{TC,t-t,t-t}$$

for the HP-turbocharger, as otherwise the matching of the two machines would not be correct.

1.2 Turbocharging efficiency

Exact definition:

$$\eta_T = \frac{\overline{\dot{m}}_{CYi} \cdot e_{P,Com}\left(\overline{T}^*_{Si}, \dfrac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)}{\overline{\dot{m}}_{CYo} \cdot e_{P,Exp}\left(\overline{T}^*_{EM}, \dfrac{p^*_{EM,eq}}{p_{So,eq}}\right)} \tag{5}$$

Approximate definition:

$$\eta_T = \frac{\overline{\dot{m}}_{CYi} \dfrac{\kappa_A}{\kappa_A - 1} R_A \cdot \overline{T}^*_{Si}\left[\left(\dfrac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1\right]}{\overline{\dot{m}}_{CYo} \dfrac{\kappa_G}{\kappa_G - 1} R_G \cdot \overline{T}^*_{EM,eq}\left[1 - \left(\dfrac{p^*_{EM,eq}}{p^*_{So,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]} \tag{5'}$$

Recommendations for the choice of the appropriate values for κ and R for air and exhaust gas are given in Annex 2.

This turbocharging efficiency uses the physical boundaries of the turbocharging system (engine proper), which are air and exhaust gas receivers and mostly the ambient. The detailed definitions of all items in (5) are given in the following table:

$\overline{\dot{m}}_{CYi}$	Mass flow rate at cylinder inlet. It is usually equal to the mass flow rate at the outlet of all compressors. When an air wastegate or an air bypass is active, the corresponding mass flow should be subtracted from the compressor mass flow. A special case is that of condensed water separation after the air cooler. In that case the mass flow rate reduction as well as the changes in the air and gas compositions must be taken into account.
$\overline{\dot{m}}_{CYo}$	Mass flow rate at cylinder outlet. It is usually equal to the mass flow rate at the inlet of all turbines. Deviations can be due to gas wastegate, air bypass, exhaust gas recirculation (EGR), etc. It can be easily calculated as the sum of the mass flow rates at the engine inlet plus the mass flow rate of injected fuel (plus any added water). The small amount of gas flowing through the piston rings into the crankcase can usually be neglected.
Si	External inlet state at the system site. For engines aspiring from an engine room the actual conditions at the compressor inlet (state Ci^*) can be different due to air conditioning, heat radiated by machinery, forced ventilation, etc.
So	External outlet state at the system site. It is usually the ambient, except when devices under consideration are located between the system boundary and the ambient (e.g. underwater exhaust in submarines).
IM^*	Total* state in the air receiver.
EM^*	Total* state in the exhaust gas receiver. The dynamic pressure is low only on engines with constant pressure exhaust systems; in other cases a mean dynamic pressure based on the area of the exhaust gas manifold should be taken into account.

1.3 Turbocharging system efficiency

$$\eta_{TS} = \frac{\eta_T}{\eta_{TC}} \tag{6}$$

The meaning of the turbocharging system efficiency is obvious: it takes into account all losses in the turbocharging system except those of the turbocharger itself. Since the turbocharger efficiency is difficult to measure on an engine, a valuable alternative can be to derive it from the turbocharging and the system efficiencies. Details for the direct calculation of the system efficiency are given in Annex 4.

Considering the responsibilities, a further boundary could be defined between the actual engine and the rest of the power plant. This makes sense when the bare engine is delivered by the engine manufacturer with defined boundaries to the builder of the power or propulsion plant. In any case, it is the responsibility of the engine manufacturer to specify the requirements at the engine boundary, and/or take into account the consequences of non-compliance.

In general, the responsibilities of the parties cannot be strictly divided, because interactions are always possible, for instance:

- A bad velocity profile at the compressor inlet could impair compressor performance, stability and reliability.
- A too high velocity or a bad velocity profile at the compressor outlet could cause high pressure losses in the air system and even impair the air cooler performance.
- A bad velocity profile at the turbine inlet, caused by the engine manifold, could impair the turbine performance and reliability.
- A too high velocity or a bad velocity profile at the turbine outlet could cause high pressure losses in the gas pipes or connected devices.

Velocity limitations at the compressor and turbine outlets are discussed under 3.1.

1.4 Turbocharging indicated efficiencies

For a cycle analysis performed by means of simulation programs it is useful to define efficiencies based on boundaries that go deeper into the engine process.

$$\eta_{eiT} = \frac{\int\limits_{Cycle} e_{P,Com}\left(T^*_{CYi}, \frac{p^*_{CYi}}{p_{Amb}}\right)\cdot \dot{m}_{CYi}(t)\cdot dt}{\int\limits_{Cycle} e_{P,Exp}\left(T^*_{CYo}, \frac{p^*_{CYo}}{p_{Amb}}\right)\cdot \dot{m}_{CYo}(t)\cdot dt} \tag{7}$$

$$\eta_{iiT} = \frac{\int\limits_{Cycle} e_{P,Com}\left(T_{CY}, \frac{p_{CY}}{p_{Amb}}\right)\cdot \dot{m}_{CYi}(t)\cdot dt}{\int\limits_{Cycle} e_{P,Exp}\left(T_{CY}, \frac{p_{CY}}{p_{Amb}}\right)\cdot \dot{m}_{CYo}(t)\cdot dt} \tag{8}$$

The external indicated turbocharging efficiency η_{eiT} uses the boundaries closest to the engine cylinders, i.e. a boundary just before the cylinder inlet (inlet valves or ports) and just after the cylinder outlet (exhaust valves or ports) and the ambient.

The internal indicated turbocharging efficiency η_{iiT} uses as boundaries those of the enclosed volume in the cylinder.

The detailed definitions of all items in (7), (8) are given in the following table:

$\dot{m}_{CYi}(t)$	Instantaneous mass flow rate at cylinder inlet
$\dot{m}_{CYo}(t)$	Instantaneous mass flow rate at cylinder outlet
CYi^*	Total* state at cylinder inlet
CY	State in cylinder
CYo^*	Total* state at cylinder outlet

These efficiencies cannot be measured, but they can be easily calculated with the help of a gas exchange program that gives the pressure, temperature and mass flow resolved over the crank angle.

These are useful for the theoretical analysis and understanding of the turbocharging process, because they describe the effectiveness of the whole turbocharging system on the engine cylinders. They can be used also to compare different piping systems (from straight pulse pipes to constant pressure) as well as different turbocharger arrangements (single or multistage, mechanically or electrically driven superchargers or auxiliary blowers). An example of the application of these efficiencies is given in Annex 5.

In order to guarantee that results from different simulation codes are comparable it is therefore useful to formulate normalised efficiencies for this kind of evaluation.

2 Limitations

The defined efficiencies cannot always be measured in an accurate way on every engine. Therefore, it is often difficult to obtain the desired information. Some common problems and limitations are now described.

2.1 Turbocharger efficiency

The turbocharger efficiency is measured mainly on the turbocharger test rig under controlled steady-state conditions. A test rig for thermodynamic measurements is characterised by straight connecting pipes fitted at every flange, where the representative values of pressure and temperature at the turbocharger flanges are derived as mean values of many static pressure taps distributed around a circumference of the pipe mantle and many temperature sensors exploring the temperature field inside the measuring pipe.

The mass flow rate is measured with the help of normalised flow meters (usually nozzles) fitted inside a straight pipe with uniform flow.

Under these laboratory conditions the efficiency can be measured with a high accuracy, typically in the range 0.2 % to 1 %, depending on the load point.

Similar conditions cannot be found on a running engine. Similar measuring pipes could be installed only for the compressor on the air inlet side, and possibly for the turbine on the gas outlet side, but for the much more important connection on the pressure side of the compressor and turbine it is very difficult to find a suitable section with uniform flow for thermodynamic measurements. Usually, single pressure and temperature sensors are fitted on the turbocharger flanges or near to them.

The measurement at the flange is much less accurate than the test rig measurements; therefore, the derived efficiency should be given with a subscript F:

$\eta_{TC,F}$ = Turbocharger efficiency measured at the flanges

Under uniform flow conditions, the deviations of flange measurements due to the geometry of the turbocharger casings could be compensated by a calibration performed on the test rig, but the deviations due to the non uniformity of the flow caused by the engine cannot be compensated.

Much more important are the deviations due to the pulsating flow, especially in the exhaust pipe. Large pressure pulsations make reliable measurement of the turbocharger efficiency impossible. A classification of the pressure pulsations is possible on the basis of the characteristic flow velocity in the pipe, defined as:

$$c_L = c_k \left(\frac{D}{D_L} \right)^2 \qquad (9)$$

where c_k is the mean piston speed at rated engine speed, D is the engine bore, and D_L is the pipe diameter.

Three ranges can be defined:

- $c_L \leq 20$ m/s Constant pressure engines ➔ $\eta_{TC,F}$

With these engines the pressure pulsations are very moderate (except when acoustic resonances are encountered on the air side), and the flange-measured turbocharger efficiency has a typical accuracy of 1 to 3%.

- $20 < c_L \leq 35$ m/s Quasi-constant pressure engines ➔ $\eta_{TC,App,SPS}$

With these engines the pressure pulsations are already clearly visible, reaching peak-to-peak amplitudes up to 1 bar. The flange-measured turbocharger efficiency must be used with a suffix App for apparent and SPS (single pipe system). Typical deviations from the test rig efficiency can be in the range 3 to 10%.

- $c_L > 35$ m/s Pulse engines ➔ $\eta_{TC,App,Pulse}$

With these engines the pressure pulsations can be very high, up to a peak-to-peak amplitude of 4 bar. The flange-measured turbocharger efficiency must be used with a suffix App for apparent. The apparent turbocharger efficiency measured on these engines does not correlate with the test rig efficiency any more. The efficiency curves are completely different, and the deviations can range from about 10% at the engine rated power up to 100% at part load. The apparent efficiency can only be used to monitor a given configuration, but comparisons between different configurations are extremely uncertain.

2.2 Reasons for problems in unsteady flow measurements

Pressure and temperatures are usually measured as mean values with the help of high-inertia sensors which are only suitable for steady flow measurements.

The mean temperature value that is obtained can be taken as a rough approximation of the energetic mean value in the flow, since the heat transfer coefficient for a turbulent flow in a pipe is roughly proportional to the enthalpy in the gas. This is only valid as long as the amplitudes of the temperature and velocity variation are much lower than the mean value. This is usually the case at the turbine inlet, but not in the exhaust ports after the cylinder of the scavenged engine, where the sensors register mean values that are much lower than the energetic mean temperature.

With measured mean values of pressure and temperature, only an apparent efficiency can be calculated which is not representative of the turbocharger efficiency. A measure of effective turbocharger efficiency could theoretically be obtained when time-resolved pressure and temperature records are available.

Unsteady pressure measurements are not usual, but are feasible, although at a certain cost. Unsteady temperature measurements are much more difficult because they require sensors with very low inertia, which are not readily available for measurement on industrial engines.

An additional difficulty comes from the fact that pressure and temperature waves are out of phase. This is, because the pressure waves travel along a pipe at the speed of sound, while the flow and temperature waves travel only at the flow velocity.. It follows from the courses of pressure and temperature measured on a common time basis that the expergy can be integrated provided that the pressure and temperature courses are measured at the same position in a pipe. The resulting value is valid only for that position and could be slightly different for the turbine inlet flange.

It can be concluded that measurement of the turbocharger efficiency on a pulse engine requires a considerable effort as well as laboratory conditions, but is hardly viable in practice on engines for industrial applications.

2.3 Measurement of the exhaust gas temperature before the turbine

The measurement of the exhaust gas temperature before the turbine is an additional source of uncertainty for the calculation of the turbocharging efficiency.
Experience shows that on a running engine the measured temperature is almost always higher than the thermodynamic temperature of the gas flow. On 4-stroke engines the deviations are typically 20 - 30 K, and up to 50 K in extreme cases.
These deviations depend on the type, position and condition (fouling) of the thermocouple as well as on the fuel used (they are usually higher with heavy fuel oil).
The reasons for the deviations have not yet been explained.

2.4 Turbocharging efficiency

The limitations discussed under 2.1 to 2.3 also apply to the turbocharging efficiency. The situation is only better with regard to the local conditions. In most cases it is possible to find measuring positions within straight pipe sections on the air and gas system where the flow velocity is lower and has a more uniform distribution than at or near the turbocharger flanges.

For these reasons it is always advisable to measure and calculate the turbocharging efficiency and to derive from it the turbocharger efficiency with the help of measured or estimated pressure losses in the system.

2.5 Mixing problems

Another kind of problem that can arise in the measurements is that of incomplete homogeneity of the flow after a junction. In four stroke engines, in particular, there is always a cylinder pipe that discharges into the mixing pipe close to the turbine inlet. Measuring the gas temperature between the junction and the flange can lead to unpredictable results.

Even more critical is the case of an air by-pass that blows air close to the turbine. In this case, depending on the position of the temperature sensor as well as on the flow patterns, any temperature between that of the air and that of the unmixed exhaust gas can be measured. In such cases it is probably better to measure the exhaust gas temperature before the junction and to calculate the mixing temperature with an enthalpy balance. The mixing ratio is usually not measured; nevertheless, a reasonable estimation provides better results than a direct temperature measurement within the mixing zone.

3 Requirements and recommendations

3.1 Turbocharger

To calculate the efficiency the mass flow rate at the compressor outlet must be known. It is much easier to measure the mass flow rate at the compressor inlet; therefore, if the compressor volumetric efficiency has to be taken into account the turbocharger manufacturer must provide the means to evaluate the losses or deviations in mass flow.

It goes without saying that the turbocharger manufacturer must also provide the dimensions of all flanges relevant for the efficiency calculations.

In order to improve the accuracy of the turbocharger efficiency measurement on the engine it is recommended that the turbocharger manufacturer provides measuring points of pressure and temperature at the flanges on the turbocharger as well as correction factors for the measurements, when appropriate and available. Deviations due to different arrangement of the casings and of the connecting pipes can not be completely eliminated, but at least on engines with constant pressure turbocharging the uncertainty of the results can be reduced by standard measuring points.

Compressor outlet

In order to feed a mass flow into a manifold the compressor must produce the static pressure present in the manifold plus the dynamic pressure corresponding to the flow. Therefore, it is reasonable to make reference to the conventional total state at the compressor outlet. Another argument is that the dynamic pressure at the compressor outlet flange can, to some extent, be converted to static pressure; on large engines it is not unusual for the static pressure at the compressor outlet to be comparable to the manifold pressure. This means that the recovery of the dynamic pressure compensates the pressure losses through the air cooler.

All of these arguments can be sustained as long as the flow area at the compressor outlet flange assumes reasonable values. It is advisable to choose the flange area such as to give a flow velocity at rated power that does not exceed the range 75 ÷ 95 m/s, corresponding to a dynamic pressure which is about 2 ÷ 3% of the total pressure. Exceeding these values results in the velocity being too high and produces, according to the definition, a slightly higher turbocharger efficiency; in addition, the pressure losses in the pipes after the compressor are higher, resulting in a lower turbocharging efficiency.

Turbine outlet

The considerations outlined in the preceding point can also be used for the turbine outlet. The turbine can only expand a gas down to the static pressure in the following pipe, plus the dynamic pressure corresponding to the flow. In addition, the dynamic pressure can be converted to some extent into static pressure by cleverly designing the following pipes.

Since the turbine pressure ratio has almost always been given as total to static, this tradition has been retained for the turbocharger efficiency definition.

The gas outlet casing is usually the biggest part of a turbocharger. For this reason, two values are formulated for the maximum flow velocity at the turbine outlet flange at rated power:

- 120 m/s, corresponding to a dynamic pressure of about 40 mbar, for a compact system,
- 85 m/s, corresponding to a dynamic pressure of about 20 mbar, for a more efficient system.

In general, a higher gas velocity at the turbine outlet produces a lower turbocharger efficiency and, as long as the kinetic energy is not recovered, a lower turbocharging efficiency. But since the flow deceleration with low losses requires an optimal length, and this can be longer than the flow path within the turbocharger, a system with higher velocity at the turbocharger flange and an external diffuser could give a better turbocharging efficiency.

Boundary conditions

In the representation of turbocharger maps and of the resulting turbocharger efficiency it is usually assumed that the boundary conditions are invariant. This is based on elementary similarity relations and the assumption that second order effects, like friction and heat transfer, depending on local temperatures, Reynolds number and thermodynamic properties of the work medium, do not influence the results.

These effects are not always negligible:

- The oil viscosity is strongly dependent on temperature and has an influence on the mechanical efficiency of the turbochargers, especially at low loads.
- The water temperature, when cooling water is used, influences the heat losses of the turbocharger and indirectly also the oil temperature in the bearings.
- The exhaust gas temperature influences the heat flows and the material temperatures in the turbocharger. The turbine efficiency can change due to changes in the heat losses for the turbine stage but also to variations in the clearances. Depending on the turbocharger design, the exhaust gas temperature can also influence the wall temperatures in the compressor stage and consequently its efficiency. These effects are strongly size-dependent, being important for small turbochargers but much less important for large turbochargers.
- The gas properties (mainly the gas constant R und the specific heat ratio k) have an obvious influence on the calculation of the efficiency and this has to be taken into account. Additionally, the gas properties have an influence on the efficiency and on the operating point of the turbocharger. In the case of large changes in the gas properties, enhanced similarity rules should be used for the conversion of turbocharger test rig results to engine operation. In most cases the variations in the thermodynamic properties due to ambient humidity, fuel composition (for hydrocarbon based fuels) and temperature level are moderate and do not need to be considered. Only in extreme cases, such as when very different fuels are used, when humidification is extreme or there is a considerable change in temperature, should such effects be checked.
- The Reynolds number is not taken into account in the similarity rules, which consider the Mach number instead. This has a moderate effect on the turbocharger efficiency. Significant changes in the Reynolds number can only be caused by extreme variation of the compressor inlet pressure, as in the case of extreme altitude or in the high pressure stage in a 2-stage turbocharging system.

All the effects listed above usually lie within the accuracy of a turbocharger efficiency measurement, and therefore no special attention is required. Nevertheless, it is recommended that the turbocharger manufacturer perform the measurements on the turbocharger test rig under conditions that are specified and close to the conditions for operation on the engine. These are dependent on the application, as can be seen in the following table:

Engine type	Lube oil temperature	Cooling water temperature (when appropriate)	Exhaust gas temperature at turbine inlet
	[°C]	[°C]	[°C]
2-stroke (low speed)	45	80	420
4-stroke (medium speed)	65	80	520
4-stroke (high speed)	80	80	620

3.2 Engine and turbocharging system

The engine manufacturer must provide the positions and flow areas of all measuring stations relevant for efficiency calculations.

The turbocharger and turbocharging efficiencies depend mainly on the enthalpy heads derived from the pressure and temperature measurements. The mass flow ratio:

$$\frac{\bar{\dot{m}}_{Co}}{\bar{\dot{m}}_{Ti}} \quad \text{respectively} \quad \frac{\bar{\dot{m}}_{CYi}}{\bar{\dot{m}}_{CYo}}$$

is in normal cases fairly constant:

$$\frac{\overline{\dot{m}}_{CYi}}{\overline{\dot{m}}_{CYo}} = \frac{\overline{\dot{m}}_{CYi}}{\overline{\dot{m}}_{CYi} + \overline{\dot{m}}_{Fuel}} \approx 0.97 \pm 0.01$$

For best accuracy, the mass flow rates of the air from the compressor and of the fuel should be measured. These measurements can usually be performed only on the engine test rig.

The air mass flow can be measured accurately at the compressor inlet using a measuring pipe connected to the compressor suction branch and a metering device according to [1]. In some cases a less accurate mass flow measurement is obtained using a differential pressure measurement at a defined point in the system. In these cases a calibration is required.

The fuel mass flow can be measured or derived from the engine power and the specific fuel consumption provided by the engine manufacturer.

If the air flow measurement is missing, an estimate can be made based on the engine power and the specific air consumption provided by the engine manufacturer. Alternatively, the turbine can be used as a rough metering device. To this end a curve of the mass flow parameter over the expansion ratio should be provided by the turbocharger manufacturer.

All the considerations above are valid as long as the flow path from the air inlet into the system to the gas outlet is a single line. This is not true in the following cases:

- Bypass possibilities are given (e.g. connection between compressor outlet and turbine inlet bypassing the engine, exhaust gas recirculation)
- Mass is separated from the main flow (e.g. air or gas wastegate, condensed water separation)
- Mass is added to the flow path (i.e. supply of water, air or other media)

In these cases a functional schematic of the turbocharging system must be available as well as a means of measuring or calculating the additional mass flow rates.

3.3 Measurements

For an accurate efficiency assessment, state of the art measuring techniques must be used. It is beyond the scope of this recommendation to give guidelines for measuring pressures and temperatures.

Also important is the appropriate choice of the measuring positions. Whenever possible, the sensors should be placed where the flow is straight and uniform. The following should be avoided:

- Pressure measurement in highly turbulent zones (e.g. casings) or in curved pipes (if unavoidable, the pressure taps should be installed on the diameter perpendicular to the plane of curvature, where the pressure gradient is lower)
- Temperature measurements after mixing zones (e.g. a short distance after a junction in the exhaust system)
- Temperature measurements in dead zones (e.g. in the closed end of a pipe or in a corner of a plenum)
- Temperature measurements a short distance after an air cooler
- Measurements of air temperature close to an exhaust pipe

One special problem involves the measurement of the air inlet temperature for the compressor of a turbocharger mounted on an engine with filter silencer. In this case the air temperature is higher than the ambient temperature far away from the engine (the mean value is usually 8-10 K higher and highly non-uniform). This is due to the heat transferred from the engine and especially from the exhaust pipes.

4 Extensions for different turbocharging topologies

All the considerations mentioned so far apply to the simplest case with one turbocharger per engine.
For turbocharging systems with more than one turbocharger working in parallel, the extension is just a matter of multipliers, as long as the system is symmetrical.

If there are deviations from the symmetry, then an averaging problem must be solved. The averaging rules have already been given for the problem of averaging time-dependent values and can be recalled here again for different components. Average values must fulfil the following conservation criteria:

- Mass conservation for the average mass flow
- Enthalpy conservation for the average temperature
- Expergy conservation for the average pressure

These criteria should be applied to cylinders and turbochargers when there are significant deviations from the symmetry in order to arrive at representative values for the engine and the equivalent turbocharger. In a general case, the mean turbocharger efficiency from N turbochargers should be calculated as follows:

$$\overline{\eta}_{TC} = \frac{\sum_{i=1}^{N} \overline{\dot{m}}_{Co,i} \cdot e_{P,Com}\left(\overline{T}^*_{Ci,i}, \frac{p^*_{Co,eq,i}}{p^*_{Ci,eq,i}}\right)}{\sum_{i=1}^{N} \overline{\dot{m}}_{Ti,i} \cdot e_{P,Exp}\left(\overline{T}^*_{Ti,i}, \frac{p^*_{Ti,eq,i}}{p_{To,eq,i}}\right)} \tag{10}$$

If the pressures and temperatures are comparable, the formula can be simplified as follows:

$$\overline{\eta}_{TC} = \frac{\sum_{i=1}^{N} \overline{\dot{m}}_{Co,i} \cdot \eta_{TC,i}}{\sum_{i=1}^{N} \overline{\dot{m}}_{Co,i}} \tag{10'}$$

This means that the “mean” turbocharger has the mass-averaged efficiency of the individual turbochargers.

In the case of a multistage turbocharging system the problem is more complex. Two different approaches are possible:

a) The responsibility boundaries are set at the turbocharger flanges, and have to be evaluated separately from the system. The formulation (10) can then be used to define the average turbocharger efficiency.

In this case, the turbocharging system efficiency, calculated as:

$$\eta_{TS} = \frac{\eta_T}{\overline{\eta}_{TC}} \tag{6'}$$

can assume values greater than 1, because the intercooling represents an efficiency gain (typically up to 10 points) that is not dependent on the turbochargers.

b) The turbocharging group is delivered as such by the manufacturer, and so can be considered as a unit. In this case the boundaries are set in a similar way to those for a single turbocharger, and the equivalent turbocharger efficiency must be defined in another way. For the case of a two stage turbocharging system with a high pressure turbocharger (HP) and a low pressure turbocharger (LP), the equivalent turbocharger efficiency is:

$$\eta_{TC,Eq} = \frac{\overline{\dot{m}}_{Co,HP} \cdot e_{P,Com}\left(\overline{T}^*_{Ci,LP}, \frac{p^*_{Co,eq,HP}}{p^*_{Ci,eq,LP}}\right)}{\overline{\dot{m}}_{Ti,HP} \cdot e_{P,Exp}\left(\overline{T}^*_{Ti,HP}, \frac{p^*_{Ti,eq,HP}}{p_{To,eq,LP}}\right)} \tag{11}$$

If intercooling is provided, this equivalent turbocharger efficiency is usually higher than the individual efficiencies of both turbocharger stages. The turbocharging system efficiency can then still be calculated with Eq.(6) and gives the same information about system losses as in the case of single stage turbocharging.

Another special case, whose complexity goes beyond the scope of this Recommendation, is that of turbochargers with Power Take-in/Power Take-out (PTI/PTO). An obvious extension of the turbocharger efficiency definition would be to add/subtract the additional power to/from the turbine power. Two considerations are necessary:

- The denominator in Eq. (4) represents an isentropic power potential that is converted into mechanical power by the turbine with an efficiency lower than 1. For better consistency the additional power of PTI/PTO should be applied as shaft power. This would require splitting the turbocharger efficiency into component efficiencies (compressor, turbine and mechanical efficiency). It is not possible to calculate the turbocharger shaft power by means of turbocharger efficiency measurement.
- The turbocharger and the additional power are hardly comparable. The turbocharger power has a high value for the turbocharging process which considerably improves the engine performance and also a very low cost because it uses the waste energy from the engine. The additional power (mechanical or electrical) has a high value and a high cost because it requires a source of energy (PTI) or acts as an auxiliary power source (PTO).

The following formula is proposed:

$$\eta_{TC,Eq} = \frac{\overline{\dot{m}}_{Co} \cdot e_{P,Com}\left(\overline{T}^*_{Ci}, \frac{p^*_{Co,eq}}{p^*_{Ci,eq}}\right)}{\overline{\dot{m}}_{Ti} \cdot e_{P,Exp}\left(\overline{T}^*_{Ti}, \frac{p^*_{Ti,eq}}{p_{To,eq}}\right) + \frac{P_{PTI/PTO}}{\eta_{sT}}} \tag{12}$$

The additional power $P_{PTI/PTO}$ is positive in the case of PTI, negative in the case of PTO. The isentropic turbine efficiency η_{sT} cannot be measured with reasonable accuracy on an engine. A rough estimation could be the square root of the turbocharger efficiency.

5 References

[1] Measurement of fluid flow by means of pressure differential devices – Part 1: Orifice plates, nozzles and Venturi tubes inserted in circular cross-section conduits running full, EN ISO 5167-1 : 1991.

[2] Measurement of fluid flow in closed conduits – Guidelines on the effects of flow pulsations on flow-measurement instruments, ISO/TR 3313:1998(E).

[3] Roth, B. "A Work Potential Perspective of Engine Component Performance", AIAA Paper No. 2001-3300.

[4] Turbocharger Gas Stand Test Code, SAE J1826, Reaf. Mar95.

[5] Watson, N., S. G. Oldfield: The influence of exhaust valve and port design on energy transfer to the turbocharger turbine. CIMAC-Congress Paris, 1983, paper D8.4

Annex 1 - Definition of the mean values

1 Temperature

The required mean temperature for the calculation of the efficiencies is based on enthalpy conservation:

$$h\left(\overline{T}\right)=\frac{\int\limits_{Cycle}\dot{m}\cdot h(T)\cdot dt}{\int\limits_{Cycle}\dot{m}\cdot dt} \qquad \text{or} \qquad \overline{T}=h^{-1}\left(\frac{\int\limits_{Cycle}\dot{m}\cdot h(T)\cdot dt}{\int\limits_{Cycle}\dot{m}\cdot dt}\right)$$

In order to calculate the mean temperature it is necessary to first calculate the mass averaged enthalpy in the flow over a cycle, and then to invert the enthalpy function. This can be easily done within a computer program with the help of a library of thermodynamic functions.

For a simplified calculation the approximate formula below could be used:

$$\overline{T}=\frac{\int\limits_{Cycle}\dot{m}\cdot c_p\cdot T\cdot dt}{\int\limits_{Cycle}\dot{m}\cdot c_p\cdot dt}$$

2 Pressure

The required mean pressure for the calculation of the efficiencies is based on expergy conservation. The averaging process is based on a resulting function; therefore, it seems appropriate to describe the result as an “equivalent” instead of “mean” pressure. A reference constant state (p_0, T_0) is required to define the equivalent pressure; it is convenient to use the atmospheric state.

The equivalent pressure in a position i within the flow path before the engine is defined by:

$$e_{P,Com}\left(T_0,\frac{p_{ieq}}{p_0}\right)=\frac{\int\limits_{Cycle}e_{P,Com}\left(T_0,\frac{p_i}{p_0}\right)\cdot\dot{m}\cdot dt}{\int\limits_{Cycle}\dot{m}\cdot dt}$$

The definition for the flow path after the engine is:

$$e_{P,Exp}\left(\overline{T},\frac{p_{ieq}}{p_0}\right)=\frac{\int\limits_{Cycle}e_{P,Exp}\left(T,\frac{p_i}{p_0}\right)\cdot\dot{m}\cdot dt}{\int\limits_{Cycle}\dot{m}\cdot dt}$$

In order to calculate the equivalent pressure it is necessary to first calculate the mass averaged expergy in the flow over a cycle, and then to invert the expergy function. This can be easily done within a computer program with the help of a library of thermodynamic functions.

For a simplified calculation the approximate formulas, valid for constant c_p and κ, could be used:

Inlet side (before engine):

$$p_{eq} = p_0 \left(\frac{\int\limits_{Cycle} \dot{m} \cdot c_p \cdot T_0 \cdot \left(\left(\frac{p}{p_0} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right) \cdot dt}{c_p \cdot T_0 \int\limits_{Cycle} \dot{m} \cdot dt} + 1 \right)^{\frac{\kappa}{\kappa-1}} = \left(\frac{\int\limits_{Cycle} \dot{m} \cdot p^{\frac{\kappa-1}{\kappa}} \cdot dt}{\int\limits_{Cycle} \dot{m} \cdot dt} \right)^{\frac{\kappa}{\kappa-1}}$$

Exhaust side (after engine):

$$p_{eq} = p_0 \left(1 - \frac{\int\limits_{Cycle} \dot{m} \cdot c_p \cdot T \cdot \left(1 - \left(\frac{p}{p_0} \right)^{\frac{1-\kappa}{\kappa}} \right) \cdot dt}{\int\limits_{Cycle} \dot{m} \cdot c_p \cdot T \cdot dt} \right)^{\frac{\kappa}{1-\kappa}} = \left(\frac{\int\limits_{Cycle} \dot{m} \cdot T \cdot p^{\frac{1-\kappa}{\kappa}} \cdot dt}{\int\limits_{Cycle} \dot{m} \cdot T \cdot dt} \right)^{\frac{\kappa}{1-\kappa}}$$

It can be noted that with the simplified method the values in the reference state have no influence on the results. In the exact method they just have a marginal influence on the gas properties.

Annex 2 - Mean values for κ and R

The exact efficiency definitions given above should be used with the help of thermodynamic tables or functions for the calculation of the expergy. These functions are today very easy to find and to implement in software; nevertheless, approximated values are given to allow the direct calculation of the efficiency with reasonable accuracy.

1 Thermodynamic values for the air side

For the calculation of the isentropic compression enthalpy, the formula derived for constant thermodynamic properties is widely used:

$$e_{P,Com}\left(T_1,\frac{p_2}{p_1}\right)=\frac{\kappa_A}{\kappa_A-1}R_A T_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{\kappa_A-1}{\kappa_A}}-1\right]$$

For dry air at ISO conditions, the constants are: R_A = 287.05 J/(kg K) κ_A = 1.4000
For humid ISO air (relative humidity φ = 30%) the constants are: R_A = 288.10 J/(kg K) κ_A = 1.3991

The influence of the humidity can be described with good accuracy as follows:

$R_A(x)$ = 287.05 + 1.74 x
$\kappa_A(x)$ = 1.4000 – 0.00137 x

where x is the water vapour mass fraction in the ambient air (water vapour mass to total air mass, in percent).

The variation of κ_A with the temperature is very small. Table 1 gives the variation of the mean value $\overline{\kappa}_A$ as a function of the arithmetic mean temperature of the compression process. The pressure ratios required to achieve the mean temperature from different start temperatures are given as an indication.

Table 1: Mean specific heat ratio as function of the mean temperature for dry air.

Tmean	κ_A	Pressure ratio for T_1 =		
		273	298	323
298	1.400	1.8	1	-
343	1.399	4.3	2.5	1.5
375	1.398	7.1	4.3	2.7
401	1.397	-	6.4	4.0
426	1.396	-	8.9	5.7

Example

Compression process with pressure ratio 3.7, starting with humid air (x = 1.3%) at temperature T_1 = 308 K.

Exact calculation

$$e_{P,Com}(308K, 3.7, x=1.3\%)= 141162 \text{ J/kg}$$ (Reference)

First approximation

κ_A = 1.4, R_A = 287.05 J/(kg K) assumed as constants:

$$e'_{P,Com}=\frac{1.4}{1.4-1}287.05\cdot 308\cdot\left[(3.7)^{\frac{1.4-1}{1.4}}-1\right]= 140257 \text{ J/kg}$$ (error = -0.64%)

$$T_2 = 308 \cdot (3.7)^{\frac{1.4-1}{1.4}} = 447.6K$$

Second approximation

From the first approximation, the mean temperature can be estimated:

$$T_{mean} = \frac{T_1 + T_2}{2} = \frac{308 + 447.6}{2} = 377.8K$$

With a rough interpolation from Table 1, a value for dry air can be found $\overline{\kappa}_{A,x=0}$: 1.3979
The influence of the humidity then gives:

$$R_A = 287.05 + 1.74 \cdot 1.3 = 289.31 \text{ J/(kg K)}$$
$$\overline{\kappa}_A = 1.3979 - 0.00137 \cdot 1.3 = 1.3961$$

The result of the second approximation is then:

$$e''_{P,Com} = \frac{1.3961}{1.3961 - 1} \cdot 289.31 \cdot 308 \cdot \left[(3.7)^{\frac{1.3961-1}{1.3961}} - 1 \right] = 141165 \text{ J/kg} \qquad \text{(error = 0.002\%)}$$

2 Thermodynamic values for the gas side

For the calculation of the isentropic expansion enthalpy, the formula derived for constant thermodynamic properties is widely used:

$$e_{P,Exp}\left(T_1,\frac{p_1}{p_2}\right)=\frac{\kappa_G}{\kappa_G-1}R_G T_1\left[1-\left(\frac{p_1}{p_2}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]$$

For the exhaust gas the composition must be known. The values in the following have been calculated for a default liquid fuel which will be described later in this Annex.

In order to describe the mixture in the exhaust gas it is convenient to use the mass fraction of the stoichiometric combustion products in the whole mass of gas as parameter:

$$x_c=\frac{m_{combustion\ products}}{m_{combustion\ products}+m_{excess\ air}}$$

The variability of the gas properties as a consequence of the variability of the exhaust gas temperature as well as of the gas composition is much higher on the gas side than on the air side. Therefore, it is useful to define reference values for different engine categories. For better accuracy it is also convenient to define the mean value $\overline{\kappa}_G$ with reference to the temperature level as well as to the mean temperature difference during the expansion process.

Table 2 Reference values

Engine type		High-speed	Medium-speed	Low-speed
Exhaust gas temperature	$T_{1,ref}$ [K]	900	800	700
Mean temperature difference	ΔT_{1mean} [K]	100	100	100
Mass fraction of combustion products	$x_{c,ref}$	0.45	0.40	0.35
Mass fraction of water in ambient air	x [%]	0.6	0.6	0.6
Gas constant	$R_{G,ref}$ [J/(kg K)]	288.07	288.07	288.07
Specific heat ratio	$\overline{\kappa}_{G,ref}$	1.3302	1.3427	1.3562

The influence of changed parameters can be described with good accuracy as follows:

$$R_G=R_{G,ref}+1.70\cdot(x-0.6)$$

$$\overline{\kappa}_G=\overline{\kappa}_{G,ref}-0.01025\cdot\frac{T_1-T_{1,ref}}{100}+0.0075\cdot\frac{\Delta T_{mean}-100}{100}-0.0407\cdot(x_c-x_{c,ref})+$$
$$-0.0012\cdot(x-0.6)$$

Example

Medium-speed engine: expansion process with pressure ratio 3.7, starting from temperature T_1 = 900 K with gas mass fraction x_c = 0.45, and water vapour mass fraction of ambient air x = 1.3%.

Exact calculation

$$e_{P,Exp}\left(900K, 3.7, x = 1.3\%, x_c = 0.45\right) = 290712 \text{ J/kg} \qquad \text{(Reference)}$$

First approximation

$\overline{\kappa}_{G,ref}$ = 1.3427, $R_{G,ref}$ = 288.07 J/(kg K) assumed as constants:

$$e'_{P,Exp} = \frac{1.3427}{1.3427-1} \cdot 288.07 \cdot 900 \cdot \left[1-\left(3.7\right)^{\frac{1-1.3427}{1.3427}}\right] = 288379 \text{ J/kg} \qquad \text{(error = -0.80 \%)}$$

$$T_2 = 900 \cdot \left(3.7\right)^{\frac{1-1.3427}{1.3427}} = 644K$$

Second approximation

From the first approximation the mean temperature variation can be estimated:

$$\Delta T_{mean} = \frac{T_1 - T_2}{2} = \frac{900-644}{2} = 128 \text{ K}$$

Better approximations of R_G and κ_G are:

$$R_G = 288.07 + 1.70 \cdot \left(1.3 - 0.6\right) = 289.26 \text{ J/(kg K)}$$

$$\overline{\kappa}_G = 1.3427 - 0.01025 \cdot \frac{900-800}{100} + 0.0075 \cdot \frac{128-100}{100} + \\ - 0.0407 \cdot \left(0.45 - 0.40\right) - 0.0012 \cdot \left(1.3 - 0.6\right) = 1.3317$$

The second approximation is then:

$$e''_{P,Exp} = \frac{1.3317}{1.3317-1} \cdot 289.26 \cdot 900 \cdot \left[1-\left(3.7\right)^{\frac{1-1.3317}{1.3317}}\right] = 290673 \text{ J/kg} \qquad \text{(error = -0.01 \%)}$$

3 Accuracy of the simplified efficiency calculation

It can be noted from the examples above that with the first approximation errors of up to about 1% are possible for the expergy values for the compression and expansion processes. However, since the errors are comparable for both terms of the ratio, the resulting error for the efficiency is much lower, typically about 0.25 %.

4 Different fuels

Many authors have found it convenient to use, for general calculation, a conventional fuel (C_8H_{16}) which corresponds roughly to a diesel oil without sulphur. The values and formulae given above have been derived for a default fuel, characterized by slightly different mass fractions:

Hydrogen	13.92 %
Carbon	86.08 %

This particular fuel is also very similar to diesel oil and has the additional property that the molecular mass as well as the gas constant of the combustion products is the same as for the reference dry air.

The values are not valid for liquid or gaseous fuels that differ from diesel oil.

For exact solutions, the composition of the fuel used should be known and the thermodynamic properties of the exhaust gas should be calculated by means of a thermodynamic library. Many data sources are nowadays available as freeware; nevertheless, the implementation of the data for standard calculations can be a difficult task.

Alternatively, a simplified method can be used which guarantees a reasonable accuracy. With it, the gas properties for the default fuel are maintained and a correction factor depending on the actual fuel is defined. The method is based on the consideration that the specific heat can vary considerably for different species with reference to the mass, but is approximately constant when referred to the mole. Therefore the efficiency calculated using the mole flow ratio instead of the mass flow ratio is much more accurate independently of the effective fuel and gas properties.

For the reference default fuel the mass flow ratio can be typically:

$$\frac{\dot{m}_{Air} + \dot{m}_{Fuel}}{\dot{m}_{Air}} = 1.03 \qquad \text{[kg/kg]}$$

The mole flow ratio is in this case per definition the same:

$$\frac{\dot{n}_{Gas}}{\dot{n}_{Air}} = 1.03 \qquad \text{[mol/mol]}$$

For the definition of the correction factor of the turbocharging efficiency for an arbitrary fuel, it must be distinguished between two cases:

a) The fuel is introduced between the compressor and turbine.

In this case the definition of the correction factor is:

$$C_{Fuel} = \frac{\dot{m}_{Air} + \dot{m}_{Fuel}}{\dot{m}_{Air}} \cdot \frac{\dot{n}_{Air}}{\dot{n}_{Gas}}$$

The following rules are valid for the calculation of the mole ratio:

- Elements which oxidize with one oxygen molecule during the combustion (e.g. carbon, sulphur) do not change the mole ratio and do not need to be considered.
- Species that oxidize using one oxygen atom (e.g. hydrogen and carbon monoxide) produce additional moles.

The mole flow ratio can then be calculated by the following formula:

$$\frac{\dot{n}_{Gas}}{\dot{n}_{Air}} = 1 + \frac{\dot{m}_{Fuel}}{\dot{m}_{Air}} \cdot \left(7.184 \cdot x_H + 0.517 \cdot x_{CO}\right)$$

In the formula, x denotes the mass fractions of the species in the fuel, and the coefficients are the ratios of the molecular mass of the gas (28.965) and the mass of the species oxidized by one mole of O_2 (4.032 for hydrogen and 56.02 for carbon monoxide, respectively).

b) The fuel is mixed with air before compression.

In this case, which is typical for some categories of gas engines, the mass flow ratio is equal to unity and the following rules apply to the mole ratio:

- The combustion of premixed methane does not change the mole ratio; for other gaseous hydrocarbons the deviations are small, so that hydrocarbon molecules do not need to be considered.
- Molecules that oxidize using one oxygen atom (e.g. free hydrogen and carbon monoxide) reduce the number of moles.

The mole flow ratio and the efficiency correction factor can then be calculated by the following formula:

$$C_{Fuel} = \frac{\dot{n}_{Air} + \dot{n}_{Fuel}}{\dot{n}_{Gas}} = \frac{1 + \frac{\dot{m}_{Fuel}}{\dot{m}_{Air}} \cdot \left(2 \cdot 7.184 \cdot x_{H_2} + 2 \cdot 0.517 \cdot x_{CO}\right)}{1 + \frac{\dot{m}_{Fuel}}{\dot{m}_{Air}} \cdot \left(7.184 \cdot x_{H_2} + 0.517 \cdot x_{CO}\right)}$$

The symbols and coefficients have the same meaning as in the formula for case a).

Table 3 gives some figures for the correction factor for different cases.

Table 3: Efficiency correction factors for different fuels.

	Fuel introduced after compressor			Fuel introduced before compressor		
Fuel	$\frac{\dot{m}_{Air} + \dot{m}_{Fuel}}{\dot{m}_{Air}}$	$\frac{\dot{n}_{Gas}}{\dot{n}_{Air}}$	C_{Fuel}	$\frac{\dot{m}_{Fuel}}{\dot{m}_{Air}}$	$\frac{\dot{n}_{Gas}}{\dot{n}_{Air} + \dot{n}_{Fuel}}$	C_{Fuel}
Diesel oil (14% H)	1.0264	1.0265	≈1	-	-	-
HFO (11% H)	1.0281	1.0226	1.005	-	-	-
Methane	1.0223	1.0402	0.983	0.0223	1	1
Hydrogen	1.0114	1.0819	0.935	0.0114	0.9296	1.076
Carbon monoxide	1.1299	1.0672	1.059	0.1299	0.9408	1.063

It can be noted that the errors produced by using the gas properties for the standard fuel without corrections can reach, in extreme cases, the level of ±7%. The correction factor reduces these errors to ±0.3% against the exact calculation.

5 Water addition/separation

When water is added between the compressor and turbine (water injection into the cylinder, humidification methods, etc.) or separated after the charge air cooler, the corresponding change in the gas properties should be considered.

Alternatively, a correction factor can be used, derived using the same method as above:

$$C_{Water} = \frac{\dot{m}_{Air} + \dot{m}_{Fuel} + \dot{m}_{Water}}{\dot{m}_{Air} + \dot{m}_{Fuel} + 1.61 \cdot \dot{m}_{Water}}$$

Of course, $\dot{m}_{Water}$ is positive in the case of water addition, and negative in the case of water separation.

Annex 3 - Real and conventional total state

On an infinitesimal flow path the flow is one-dimensional, and the total or stagnation state (t) is defined by the well known relation:

$$h_t = h + \frac{c^2}{2}$$

The total temperature and the total pressure can be derived from the total enthalpy with the help of thermodynamic functions. For moderate values of the Mach number, they can be calculated directly:

$$T_t = T + \frac{c^2}{2 \cdot c_p}$$

$$p_t = p + \frac{1}{2}\rho \cdot c^2$$

In a cross section A of a real pipe under steady-state conditions the total state should be defined with mass averaged values using the actual velocity vector describing the 3-dimensional flow field at every point of the area:

$$\overline{h}_t = \overline{h} + \frac{\int_A \frac{\vec{c}^{\,2}}{2} \cdot \rho \cdot c_\perp \cdot dA}{\int_A \rho \cdot c_\perp \cdot dA} = \overline{h} + \alpha \cdot \frac{\overline{c}_\perp^2}{2}$$

In a similar way the other mean total values can also be defined for moderate Mach numbers:

$$\overline{T}_t = \overline{T} + \alpha \frac{\overline{c}_\perp^2}{2 \cdot \overline{c}_p}$$

$$\overline{p}_t = \overline{p} + \frac{1}{2}\alpha \cdot \overline{\rho} \cdot \overline{c}_\perp^2$$

The parameter α is always greater than 1, and can vary from 1.05 ÷ 1.1 for a well developed turbulent flow up to several units for flow with strong tangential components. The fraction (α-1) of the kinetic energy of the flow in a pipe cannot be converted into usable energy and represents therefore a potential for dissipation. Furthermore, it is very difficult to measure the real total state.

For these reasons, a conventional total state (*) is defined for efficiency calculations by setting $\alpha = 1$:

$$\overline{h}^* = \overline{h} + \frac{\overline{c}_\perp^2}{2}$$

$$\overline{T}^* = \overline{T} + \frac{\overline{c}_\perp^2}{2 \cdot \overline{c}_p}$$

$$\overline{p}^* = \overline{p} + \frac{1}{2}\overline{\rho} \cdot \overline{c}_\perp^2$$

Annex 4 - Direct calculation of the system efficiency

When both efficiencies η_T and η_{TC} are known, the system efficiency can be easily calculated with Eq. (6). Since the system efficiency η_{TS} can be measured and calculated, it can be useful to calculate the system efficiency directly and to use it to find η_{TC}.

Using the definitions (4), (5), and (6), the following expression can be written:

$$\eta_{TS} = \frac{\eta_T}{\eta_{TC}} = \frac{\bar{\dot{m}}_{CYi} \cdot e_{P,Com}\left(\bar{T}^*_{Si}, \dfrac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)}{\bar{\dot{m}}_{Co} \cdot e_{P,Com}\left(\bar{T}^*_{Ci}, \dfrac{p^*_{Co,eq}}{p^*_{Ci,eq}}\right)} \cdot \frac{\bar{\dot{m}}_{Ti} \cdot e_{P,Exp}\left(\bar{T}^*_{Ti}, \dfrac{p^*_{Ti,eq}}{p_{To,eq}}\right)}{\bar{\dot{m}}_{CYo} \cdot e_{P,Exp}\left(\bar{T}^*_{EM}, \dfrac{p^*_{EM,eq}}{p_{So,eq}}\right)}$$

In most cases the mass flow rates at the compressor outlet and engine inlet as well as at the engine outlet and turbine inlet are equal. Also, the temperature at the compressor inlet is usually the ambient temperature, and the temperature at the turbine inlet is equal to the temperature in the exhaust manifold.

When these conditions are fulfilled, the formula above can be modified as follows:

$$\eta_{TS} = \frac{1 - \dfrac{\left(\Delta e_{Ti} + \Delta e_{To}\right)}{e_{P,Exp}\left(\bar{T}^*_{EM}, \dfrac{p^*_{EM,eq}}{p_{So,eq}}\right)}}{1 + \dfrac{\left(\Delta e_{Ci} + \Delta e_{Co}\right)}{e_{P,Com}\left(\bar{T}^*_{Si}, \dfrac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)}}$$

The losses can be described as follows:

$$\Delta e_{Ti} = e_{P,Exp}\left(\bar{T}^*_{EM}, \frac{p^*_{EM,eq}}{p_{So,eq}}\right) - e_{P,Exp}\left(\bar{T}^*_{EM}, \frac{p^*_{Ti,eq}}{p_{So,eq}}\right) \approx \frac{p^*_{EM,eq} - p^*_{Ti,eq}}{\dfrac{p^*_{EM,eq}}{R_G \cdot \bar{T}^*_{EM}}} \cdot \left(\frac{p_{So,eq}}{p^*_{EM,eq}}\right)^{\frac{\kappa_G - 1}{\kappa_G}}$$

$$\Delta e_{To} = e_{P,Exp}\left(\bar{T}^*_{EM}, \frac{p^*_{EM,eq}}{p_{So,eq}}\right) - e_{P,Exp}\left(\bar{T}^*_{EM}, \frac{p^*_{EM,eq}}{p_{To,eq}}\right) \approx \frac{p_{To,eq} - p_{So,eq}}{\dfrac{p^*_{EM,eq}}{R_G \cdot \bar{T}^*_{EM}}} \cdot \left(\frac{p^*_{EM,eq}}{p_{So,eq}}\right)^{\frac{1}{\kappa_G}}$$

$$\Delta e_{Ci} = e_{P,Com}\left(\bar{T}^*_{Si}, \frac{p^*_{IM,eq}}{p^*_{Ci,eq}}\right) - e_{P,Com}\left(\bar{T}^*_{Si}, \frac{p^*_{IM,eq}}{p^*_{Si,eq}}\right) \approx \frac{p^*_{Si,eq} - p^*_{Ci,eq}}{\dfrac{p^*_{Si,eq}}{R_A \cdot \bar{T}^*_{Si}}} \cdot \left(\frac{p^*_{IM,eq}}{p_{Si,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}}$$

$$\Delta e_{Co} = e_{P,Com}\left(\bar{T}^*_{Si}, \frac{p^*_{Co,eq}}{p^*_{Si,eq}}\right) - e_{P,Com}\left(\bar{T}^*_{Si}, \frac{p^*_{IM,eq}}{p^*_{Si,eq}}\right) \approx \frac{p^*_{Co,eq} - p^*_{IM,eq}}{\dfrac{p^*_{Si,eq}}{R_A \cdot \bar{T}^*_{Si}}} \cdot \left(\frac{p^*_{Si,eq}}{p^*_{IM,eq}}\right)^{\frac{1}{\kappa_A}}$$

Annex 5 – Turbocharging system power balance

The expergy integration by means of time-resolved gas exchange simulation results allows the investigation of the evolution of the power availability at different stations in the turbocharging system. Figure 1 shows the calculated breakdown of power and power losses for the full load operation of a 4-stroke medium-speed engine with three different turbocharging systems.

	3-pulse	SPS	Constant pressure
η_{TC}	0.614	0.651	0.663
η_{T}	0.581	0.616	0.628
η_{ieT}	0.534	0.530	0.628
η_{iiT}	0.411	0.413	0.412

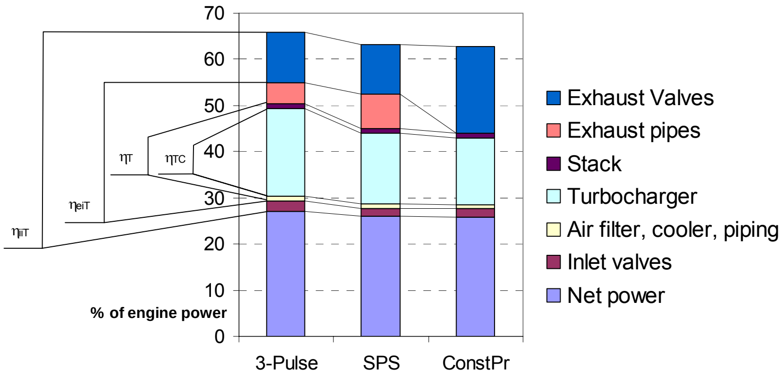


Fig. 1: Breakdown of power and power losses in the turbocharging system.

Pulse turbocharging with groups of three cylinders and a compact undivided pipe system (SPS, also known as a Modular Pulse Converter – MPC) are the most used options for engines with 6 or 9 cylinders per turbocharger. The constant pressure system would differ from SPS only because of its larger volume, but the third column in Figure 1 represents an extreme theoretical case where the pressure has been set constant from the cylinder ports to the turbine inlet.

The turbocharger efficiency η_{TC} is at its highest in the case of constant pressure and is increasingly reduced by systems which produce unsteady flow at the turbine inlet.

The turbocharging efficiency η_T shows the same tendency because the system losses are similar in all three cases.

The responsibility for the difference between η_T and η_{ieT} lies with the expergy evolution in the exhaust pipe system. The expergy level of the gas just after the cylinders is still high, but it is reduced by friction, mixing and expansion losses in the pipe system. The ratio of the two efficiencies could be defined as transmission efficiency of the expergy between cylinders and turbine. It is higher with divided systems.

The difference between η_{ieT} and η_{iiT} takes into account the valve losses, whereby only a small part of these is due to the throttling effect of the valves. The losses are mainly due to the free expansion of the gas into the exhaust port, which strongly reduces the expergy level available in the cylinders during the blow-by phase. In the extreme case of theoretical constant pressure the blow-by energy is fully dissipated by the exhaust valves, so that there are no losses in the pipes.

The indicated internal turbocharging efficiency is quite similar in all three cases. In the case of pulse turbocharging the available power is higher, partly due to the higher mass flow rates, partly at the cost of more piston work in the gas exchange process, which is reflected by a slight increase in the specific fuel consumption of the engine.

At part load the higher power availability and the better transmission efficiency lead to the well known improvement in performance with pulse turbocharging.

Annex 6 – Mass flow determination

While direct measurement of the air flow rate is usual for development engines on the test rig, it tends to be rare in other cases. Simulation programs can provide very good estimates of the flow rates in the turbocharging system when a calibrated engine model is available.

When neither measurements nor simulations are available, the air mass flow rate can be estimated using different methods.

1 Specific air flow consumption

$$\dot{m}_{CYi} = \frac{l_e \cdot P_e}{3600} \qquad \text{[kg/s]}$$

l_e	Specific air consumption	[kg/kWh]
P_e	Engine power	[kW]

This is the simplest way to estimate the air flow rate of the engine. The specific air consumption is typically:

- 5 ÷ 6.5 kg/kWh for high-speed engines
- 6.5 ÷ 8 kg/kWh for medium-speed engines
- 8 ÷ 9 kg/kWh for low-speed engines

Engine manufacturers can provide curves of l_e over engine power and often also correction factors for changed ambient conditions. Changes in engine or turbocharger conditions cannot be taken into account by this method.

2 Delivery ratio

$$\dot{m}_{CYi} = \frac{p_{IM}}{R_A \cdot T_{IM}} \cdot V_d \cdot z \cdot \frac{n}{30 \cdot \tau} \cdot \lambda_d \qquad \text{[kg/s]}$$

V_d	Cylinder displacement	[m^3]
z	Number of cylinders	
n	Engine speed	[1/min]
τ	Number of strokes	
λ_d	Delivery ratio	

λ_d	0.9 ÷ 1.0	for high-speed engines
λ_d	1.1 ÷ 1.2	for medium-speed engines
λ_d	1.0 ÷ 1.2	for low-speed engines

The delivery ratio can be calibrated on the basis of test rig measurements or simulations for a representative engine. The formula describes the dependency of the air flow rate on the density in the air manifold and the engine speed, without relying on engine power. It can take into account changes in engine and turbocharger conditions as well as in the ambient conditions, but only to a limited extent.

Modern engines operating with Miller timing can have much lower values of λ_d. Operation with variable valve timing is an additional difficulty which requires an appropriate calibration (engine mapping).

3 Flow function

$$\dot{m}_{CYi} = \frac{p_{IM}}{R_A \cdot T_{IM}} \cdot V_d \cdot z \cdot \frac{n}{30 \cdot \tau} \cdot \lambda_{Vd} + \frac{p_{IM}}{\sqrt{R_A \cdot T_{IM}}} \cdot \psi\left(\frac{p_{IM}}{p_{EM}}\right) \cdot S_{eq} \qquad \text{[kg/s]}$$

λ_{Vd} Charging efficiency referred to the swept volume

ψ Flow function:

$$\psi = \sqrt{\frac{2 \cdot \kappa}{\kappa - 1}\left[\left(\frac{p_1}{p_2}\right)^{-\frac{2}{\kappa}} - \left(\frac{p_1}{p_2}\right)^{-\frac{\kappa+1}{\kappa}}\right]} \qquad \text{for} \qquad \frac{p_1}{p_2} < \left(\frac{\kappa+1}{2}\right)^{\frac{\kappa}{\kappa-1}}$$

$$\psi = \sqrt{\kappa} \cdot \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{2\cdot(\kappa-1)}} \qquad \text{for} \qquad \frac{p_1}{p_2} \geq \left(\frac{\kappa+1}{2}\right)^{\frac{\kappa}{\kappa-1}}$$

S_{eq} Area of the engine equivalent nozzle

The charging efficiency λ_{Vd} is referred to the swept volume during the suction stroke without scavenging. It is fairly constant and can be assumed as 0.9 for 4-stroke engines. For 2-stroke engines it is zero, because the suction stroke is missing. An exception is given again by engines operating with Miller timing, where the value can be substantially reduced.

The area of the equivalent nozzle is a calibrated value, which is fairly constant over the whole engine operating range as long as the valve timings are also constant.

This formula allows the influence of the turbocharging efficiency on the engine air mass flow rate to be taken into account.

4 Turbine as a metering device

There are two ways to describe the turbine swallowing capacity:

A) The reduced mass flow rate:

$$\dot{m}_{T,red} = \overline{\dot{m}}_{Ti} \cdot \frac{\sqrt{\overline{T}^*_{Ti}}}{p^*_{Ti,eq}} \qquad \left[\frac{\text{kg}}{\text{s}} \cdot \frac{\text{K}^{1/2}}{\text{Pa}}\right]$$

B) The turbine flow coefficient:

$$\alpha_T = \overline{\dot{m}}_{Ti} \cdot \frac{\sqrt{R_G \cdot \overline{T}^*_{Ti}}}{p^*_{Ti,eq} \cdot \psi \cdot S_{resT}} \qquad [-]$$

In the latter formula S_{resT} is the area of the nozzle equivalent to the turbine, ψ is the above-defined flow function.

Both parameters are mainly dependent on the expansion ratio of the turbine. The secondary influence of the turbocharger speed can be usually neglected for a rough evaluation. It is beyond the scope of this Recommendation to define a standard for these parameters; the user must therefore rely on the data provided by the turbocharger manufacturer, which must be enhanced by the relevant definitions.

Annex 7 – Examples

For the sake of simplicity, the calculation of the mean dynamic pressure and temperature to be added to the respective static values according to Annex 3 is omitted in the following examples. In general, a pre-processing table should be applied for a specific engine which connects the raw measured data to the input section of the tables. The position and area of the measuring sections belong to this pre-processing.

1 Two stroke engine – constant pressure

Tables 1a and 1b show an example of efficiency calculations for one load point of a 2-stroke engine.

The turbocharging system is built up with more than one turbocharger, but the efficiency is calculated with mean values. Therefore, only the efficiency of the 'mean turbocharger' can be calculated, and not the individual values.

With regard to the turbocharging system losses, which reduce the system efficiency to 95%, the following remarks can be made:

- Compressor inlet
  The turbochargers are provided with their own filter-silencers; therefore, no pressure losses are present on the compressor inlet side, but the temperature at compressor inlet (mean value around the filter) is higher than at the system inlet (room temperature).

- Compressor outlet
  The pressure loss is incurred here mainly in the charge air cooler as well as in the connecting pipes between the compressor outlet and the cooler, where the air flow is decelerated and part of the available dynamic pressure at the compressor outlet is dissipated.

- Turbine inlet
  The pressure loss is incurred here mainly in the connection pipes from the exhaust manifold to the turbine inlet, where a protection grid is present.

- Turbine outlet
  The pressure loss is incurred here mainly in an exhaust gas boiler and connecting pipes. The natural draught of the stack makes a positive contribution, which reduces the pressure loss.

The engine is operated with 50% water added to the fuel (emulsion).

The comparison of Table 1a with Table 1b shows that all three methods deliver practically the same results. Nevertheless, it must be said that the first approximation is only useful for calculation of the turbocharging efficiencies. When the air and gas sides are also looked at, for instance to evaluate the compressor efficiency, at least the second approximation is necessary.

Table 1a

Calculation of turbocharging efficiencies

2-stroke engine with constant pressure

Measurements (time averaged values) → | ← Simulations (energy averaged values)

		Measurements	Simulations	
Air flow at compressor outlet	$\overline{m}_{Co}$	100		kg/s
Air flow to the cylinders	$\overline{m}_{CYi}$	100		kg/s
Air Humidity	x	1.11%		kg/kg
Air temperature at system boundary	$\overline{T}^*_{Si}$	300.15		K
Air temperature at compressor inlet	$\overline{T}^*_{Ci}$	303.15		K
Air pressure at system boundary	$p^*_{Si,eq}$	1.01		bar
Air pressure at compressor inlet	$p^*_{Ci,eq}$	1.01		bar
Air pressure at compressor outlet	$p^*_{Co,eq}$	3.65		bar
Air pressure in the inlet manifold	$p^*_{IM,eq}$	3.60		bar
Fuel mass flow	m_{Fuel}	2.00		kg/s
Additional water mass flow	m_{Water}	1.00		kg/s
Gas mass fraction	x_c	0.30		
Gas flow from the cylinders	$\overline{m}_{CYo}$	103.00		kg/s
Gas flow to the turbine	$\overline{m}_{Ti}$	103.00		kg/s
Gas temperature in the exhaust manifold	$\overline{T}^*_{EM}$	673.15		K
Gas temperature at turbine inlet	$\overline{T}^*_{Ti}$	673.15		K
Gas pressure in the exhaust manifold	$p^*_{EM,eq}$	3.40		bar
Gas pressure at turbine inlet	$p^*_{Ti,eq}$	3.38		bar
Gas pressure at turbine outlet	$p_{To,eq}$	1.04		bar
Gas pressure at system boundary	$p_{So,eq}$	1.01		bar

Fuel correction

$$C_{Fuel} = \frac{\dot m_{Co} + \dot m_{Fuel}}{\dot m_{Co}} \Bigg/ \left(1 + \frac{\dot m_{Fuel}}{\dot m_{Co}} \cdot 7.184 \cdot x_H\right) \qquad 1.0027$$

Water correction

$$C_{Water} = \frac{\dot m_{Air} + \dot m_{Fuel} + \dot m_{Water}}{\dot m_{Air} + \dot m_{Fuel} + 1.61 \cdot \dot m_{Water}} \qquad 0.9941$$

1. **First approximation**

R_A	288.10	J/(kg K)	k_A	1.3991
R_G	288.07	J/(kg K)	k_G	1.3562

Turbocharging efficiency

$$\eta_T = C_{Fuel} \cdot C_{Water} \cdot \frac{\overline{m}_{CYi} \dfrac{\kappa_A}{\kappa_A - 1} R_A \cdot \overline{T}^*_{Si} \left[\left(\dfrac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1\right]}{\overline{m}_{CYo} \dfrac{\kappa_G}{\kappa_G - 1} R_G \cdot \overline{T}^*_{EM,eq} \left[1 - \left(\dfrac{p^*_{EM,eq}}{p^*_{So,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]}$$

η_T 63.61%

Turbocharger efficiency

$$\eta_{TC} = C_{Fuel} \cdot C_{Water} \cdot \frac{\overline{m}_{Co} \dfrac{\kappa_A}{\kappa_A - 1} R_A \cdot \overline{T}^*_{Ci} \cdot \left[\left(\dfrac{p^*_{Co,eq}}{p^*_{Ci,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1\right]}{\overline{m}_{Ti} \dfrac{\kappa_G}{\kappa_G - 1} R_G \cdot \overline{T}^*_{Ti} \cdot \left[1 - \left(\dfrac{p^*_{Ti,eq}}{p_{To,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]}$$

$\eta_{TC,F}$ 66.73%

Turbocharging system efficiency

$$\eta_{TS} = \frac{\eta_T}{\eta_{TC}}$$

$\eta_{TS,F}$ 95.33%

Air and fuel specification

TAIR	300.15	
pair	101000	
phiAIR	0.5	
LiqFuel1	C	0.86
LiqFuel1	H	0.12
LiqFuel1	S	0.02
end		
LHV	41624	kJ/kg
AirMin	14.26	kg/kg

$$x = \frac{0.622}{\dfrac{p_{Air}}{\varphi \cdot p_{Saturation}(T_{Air})} - 0.378}$$

Table 1b

2. Second approximation

$$R_A = 287.05 + 1.74 \cdot x \qquad T_{mean} = \frac{\overline{T}^*_{Si}}{2} \cdot \left(1 + \left(\frac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}}\right) \quad 365.732 \qquad \kappa_A = \kappa_A(T_{mean}) - 0.00137 \cdot x$$

$$R_G = R_{G,ref} + 1.70 \cdot (x - 0.6) \qquad R_{G,ref} = 288.07 \qquad x_{c,ref} = 0.35$$

$$T_{1,ref} = 700 \qquad \Delta T_{mean} = \frac{\overline{T}^*_{EM}}{2} \cdot \left(1 - \left(\frac{p^*_{EM,eq}}{p^*_{So,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right) = 92 \text{ K} \qquad \overline{\kappa}_{G,ref} = 1.3562$$

$$\overline{\kappa}_G = \overline{\kappa}_{G,ref} - 0.01025 \cdot \frac{\overline{T}^*_{EM} - T_{1,ref}}{100} + 0.0075 \cdot \frac{\Delta T_{mean} - 100}{100} - 0.0407 \cdot (x_c - x_{c,ref}) - 0.0012 \cdot (x - 0.6)$$

R_A	288.97 J/(kg K)	k_A	1.3968
R_G	288.93 J/(kg K)	k_G	1.3599

Turbocharging efficiency

$$\eta_T = C_{Fuel} \cdot C_{Water} \cdot \frac{\overline{\dot{m}}_{CYi} \frac{\kappa_A}{\kappa_A - 1} R_A \cdot \overline{T}^*_{Si} \left[\left(\frac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1\right]}{\overline{\dot{m}}_{CYo} \frac{\kappa_G}{\kappa_G - 1} R_G \cdot \overline{T}^*_{EM,eq} \left[1 - \left(\frac{p^*_{EM,eq}}{p^*_{So,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]}$$

η_T 63.64%

Turbocharger efficiency

$$\eta_{TC} = C_{Fuel} \cdot C_{Water} \cdot \frac{\overline{\dot{m}}_{Co} \frac{\kappa_A}{\kappa_A - 1} R_A \cdot \overline{T}^*_{Ci} \cdot \left[\left(\frac{p^*_{Co,eq}}{p^*_{Ci,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1\right]}{\overline{\dot{m}}_{Ti} \frac{\kappa_G}{\kappa_G - 1} R_G \cdot \overline{T}^*_{Ti} \cdot \left[1 - \left(\frac{p^*_{Ti,eq}}{p_{To,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]}$$

$\eta_{TC,F}$ 66.75%

Turbocharging system efficiency

$$\eta_{TS} = \frac{\eta_T}{\eta_{TC}}$$

$\eta_{TS,F}$ 95.33%

3. Exact method

Turbocharging efficiency

$$\eta_T = \frac{\overline{\dot{m}}_{CYi} \cdot e_{P,Com}\left(\overline{T}^*_{Si}, \frac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)}{\overline{\dot{m}}_{CYo} \cdot e_{P,Exp}\left(\overline{T}^*_{EM}, \frac{p^*_{EM,eq}}{p_{So,eq}}\right)}$$

η_T 63.62%

Turbocharger efficiency

$$\eta_{TC} = \frac{\overline{\dot{m}}_{Co} \cdot e_{P,Com}\left(\overline{T}^*_{Ci}, \frac{p^*_{Co,eq}}{p^*_{Ci,eq}}\right)}{\overline{\dot{m}}_{Ti} \cdot e_{P,Exp}\left(\overline{T}^*_{Ti}, \frac{p^*_{Ti,eq}}{p_{To,eq}}\right)}$$

$\eta_{TC,F}$ 66.72%

Turbocharging system efficiency

$$\eta_{TS} = \frac{\eta_T}{\eta_{TC}}$$

$\eta_{TS,F}$ 95.34%

2 Four stroke engine – quasi constant pressure

Tables 2a and 2b show an example of efficiency calculations for one load point of a 4-stroke engine.

The turbocharging system is built up with one turbocharger (one turbine entry) and a compact single pipe exhaust system, also known as a modular pulse converter.

With regard to the turbocharging system losses, which reduce the system efficiency to 95%, the following remarks can be made:

- Compressor inlet
  A pipe and a nozzle for flow rate measurement are present at the compressor inlet, leading to pressure loss and an increase in the air temperature.

- Compressor outlet
  The pressure loss is incurred here mainly in the charge air cooler as well as in the connecting pipes between the compressor outlet and the cooler, where the air flow is decelerated and part of the available dynamic pressure at the compressor outlet is dissipated.

- Turbine inlet
  The points of measurement in the exhaust manifold and at the turbine inlet are coincident.

- Turbine outlet
  The pressure loss produced by the piping and the stack are artificially increased by means of a flap in order to produce normalised values (test engine).

The comparison of Table 2a with Table 2b shows that all three methods deliver practically the same results.

Since the flow is pulsating (Fig. 2), the efficiencies calculated with measured values are to be classified as apparent. The efficiencies also have been calculated with values derived by means of simulation and by applying the averaging process according to Annex 1. The deviations are in this case small (about 0.5 % points) because the effect of the pressure deviation (1%) and that of the temperature deviation (1.5%) are in opposite directions, compensating each other partially.

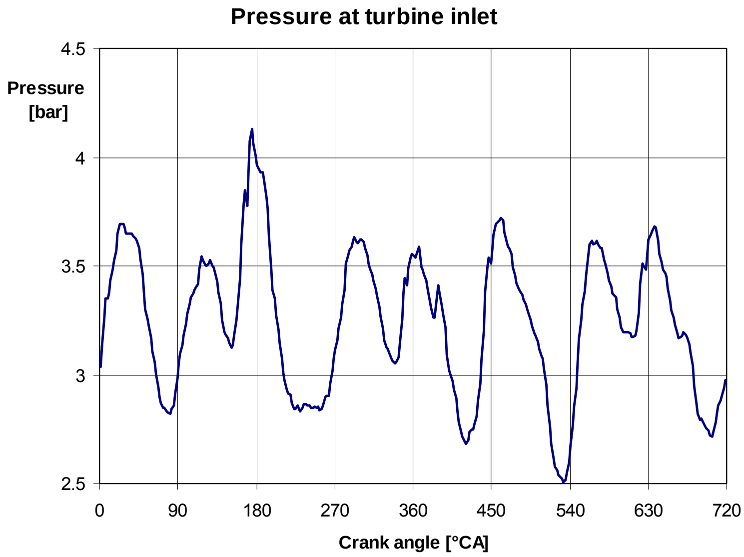


Fig. 2: Simulated static pressure trace at turbine inlet

Table 2a

Calculation of turbocharging efficiencies

4-stroke engine with single pipe exhaust system

Measurements (time averaged values) → | ← Simulations (energy averaged values)

		Measurements	Simulations	
Air flow at compressor outlet	$\overline{m}_{Co}$	12.93		kg/s
Air flow to the cylinders	$\overline{m}_{CYi}$	12.93		kg/s
Air Humidity	x	1.07%		kg/kg
Air temperature at system boundary	$\overline{T}^*_{Si}$	303.15		K
Air temperature at compressor inlet	$\overline{T}^*_{Ci}$	306.15		K
Air pressure at system boundary	$p^*_{Si,eq}$	0.998		bar
Air pressure at compressor inlet	$p^*_{Ci,eq}$	0.981		bar
Air pressure at compressor outlet	$p^*_{Co,eq}$	4.038		bar
Air pressure in the inlet manifold	$p^*_{IM,eq}$	3.947		bar
Fuel mass flow	m_{Fuel}	0.38		kg/s
Additional water mass flow	m_{Water}	0.00		kg/s
Gas mass fraction	x_c	0.429		
Gas flow from the cylinders	$\overline{m}_{CYo}$	13.31		kg/s
Gas flow to the turbine	$\overline{m}_{Ti}$	13.31		kg/s
Gas temperature in the exhaust manifold	$\overline{T}^*_{EM}$	791.15	772.05	K
Gas temperature at turbine inlet	$\overline{T}^*_{Ti}$	791.15	772.05	K
Gas pressure in the exhaust manifold	$p^*_{EM,eq}$	3.315	3.395	bar
Gas pressure at turbine inlet	$p^*_{Ti,eq}$	3.315	3.395	bar
Gas pressure at turbine outlet	$p_{To,eq}$	1.01		bar
Gas pressure at system boundary	$p_{So,eq}$	1.00		bar

Fuel correction

$$C_{Fuel} = \frac{\dot m_{Co} + \dot m_{Fuel}}{\dot m_{Co}} \Bigg/ \left(1 + \frac{\dot m_{Fuel}}{\dot m_{Co}} \cdot 7.184 \cdot x_H\right) \qquad 1.0046$$

Water correction

$$C_{Water} = \frac{\dot m_{Air} + \dot m_{Fuel} + \dot m_{Water}}{\dot m_{Air} + \dot m_{Fuel} + 1.61 \cdot \dot m_{Water}} \qquad 1.0000$$

1. **First approximation**

R_A	288.10	J/(kg K)	k_A	1.3991
R_G	288.07	J/(kg K)	k_G	1.3427

Turbocharging efficiency

$$\eta_T = C_{Fuel} \cdot C_{Water} \cdot \frac{\overline{m}_{CYi} \dfrac{\kappa_A}{\kappa_A - 1} R_A \cdot \overline{T}^*_{Si} \left[\left(\dfrac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1\right]}{\overline{m}_{CYo} \dfrac{\kappa_G}{\kappa_G - 1} R_G \cdot \overline{T}^*_{EM,eq} \left[1 - \left(\dfrac{p^*_{EM,eq}}{p^*_{So,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]}$$

$\eta_{T,App,SPS}$ = 60.90% ; η_T = 61.37%

Turbocharger efficiency

$$\eta_{TC} = C_{Fuel} \cdot C_{Water} \cdot \frac{\overline{m}_{Co} \dfrac{\kappa_A}{\kappa_A - 1} R_A \cdot \overline{T}^*_{Ci} \cdot \left[\left(\dfrac{p^*_{Co,eq}}{p^*_{Ci,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1\right]}{\overline{m}_{Ti} \dfrac{\kappa_G}{\kappa_G - 1} R_G \cdot \overline{T}^*_{Ti} \cdot \left[1 - \left(\dfrac{p^*_{Ti,eq}}{p_{To,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]}$$

$\eta_{TC,App,SPS}$ = 64.17% ; $\eta_{TC,F}$ = 64.65%

Turbocharging system efficiency

$$\eta_{TS} = \frac{\eta_T}{\eta_{TC}}$$

$\eta_{TS,App,SPS}$ = 94.91% ; $\eta_{TS,F}$ = 94.93%

Air and fuel specification

TAIR	303.15		
pair	99800		
phiAIR	0.4		
LiqFuel1	C		0.852
LiqFuel1	H		0.117
LiqFuel1	S		0.031
end			
LHV	41175.5	kJ/kg	
AirMin	14.11	kg/kg	

$$x = \frac{0.622}{\dfrac{p_{Air}}{\varphi \cdot p_{Saturation}(T_{Air})} - 0.378}$$

Table 2b

2. Second approximation

$$R_A = 287.05 + 1.74 \cdot x \qquad T_{mean} = \frac{\bar{T}^*_{Si}}{2} \cdot \left(1 + \left(\frac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}}\right) \quad 375.9444 \qquad \kappa_A = \kappa_A(T_{mean}) - 0.00137 \cdot x$$

$$R_G = R_{G,ref} + 1.70 \cdot (x - 0.6) \qquad R_{G,ref} = 288.07 \qquad x_{c,ref} = 0.40$$

$$T_{1,ref} = 800 \qquad \Delta T_{mean} = \frac{\bar{T}^*_{EM}}{2} \cdot \left(1 - \left(\frac{p^*_{EM,eq}}{p^*_{So,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right) = 104 \qquad 104 \qquad \bar{\kappa}_{G,ref} = 1.3427$$

$$\bar{\kappa}_G = \bar{\kappa}_{G,ref} - 0.01025 \cdot \frac{\bar{T}^*_{EM} - T_{1,ref}}{100} + 0.0075 \cdot \frac{\Delta T_{mean} - 100}{100} - 0.0407 \cdot (x_c - x_{c,ref}) - 0.0012 \cdot (x - 0.6)$$

R_A	288.90	J/(kg K)	k_A	1.3965	
R_G	288.86	J/(kg K)	k_G	1.3422	1.3441

Turbocharging efficiency $$\eta_T = C_{Fuel} \cdot C_{Water} \cdot \frac{\bar{m}_{CYi} \frac{\kappa_A}{\kappa_A - 1} R_A \cdot \bar{T}^*_{Si} \left[\left(\frac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)^{\frac{\kappa_A-1}{\kappa_A}} - 1\right]}{\bar{m}_{CYo} \frac{\kappa_G}{\kappa_G - 1} R_G \cdot \bar{T}^*_{EM,eq} \left[1 - \left(\frac{p^*_{EM,eq}}{p^*_{So,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]}$$	$\eta_{T,App,SPS}$ 60.84%	η_T 61.34%
Turbocharger efficiency $$\eta_{TC} = C_{Fuel} \cdot C_{Water} \cdot \frac{\bar{m}_{Co} \frac{\kappa_A}{\kappa_A - 1} R_A \cdot \bar{T}^*_{Ci} \cdot \left[\left(\frac{p^*_{Co,eq}}{p^*_{Ci,eq}}\right)^{\frac{\kappa_A-1}{\kappa_A}} - 1\right]}{\bar{m}_{Ti} \frac{\kappa_G}{\kappa_G - 1} R_G \cdot \bar{T}^*_{Ti} \cdot \left[1 - \left(\frac{p^*_{Ti,eq}}{p_{To,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]}$$	$\eta_{TC,App,SPS}$ 64.09%	$\eta_{TC,F}$ 64.61%
Turbocharging system efficiency $$\eta_{TS} = \frac{\eta_T}{\eta_{TC}}$$	$\eta_{TS,App,SPS}$ 94.92%	$\eta_{TS,F}$ 94.93%

3. Exact method

Turbocharging efficiency $$\eta_T = \frac{\bar{m}_{CYi} \cdot e_{P,Com}\left(\bar{T}^*_{Si}, \frac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)}{\bar{m}_{CYo} \cdot e_{P,Exp}\left(\bar{T}^*_{EM}, \frac{p^*_{EM,eq}}{p_{So,eq}}\right)}$$	$\eta_{T,App,SPS}$ 60.83%	η_T 61.34%
Turbocharger efficiency $$\eta_{TC} = \frac{\bar{m}_{Co} \cdot e_{P,Com}\left(\bar{T}^*_{Ci}, \frac{p^*_{Co,eq}}{p^*_{Ci,eq}}\right)}{\bar{m}_{Ti} \cdot e_{P,Exp}\left(\bar{T}^*_{Ti}, \frac{p^*_{Ti,eq}}{p_{To,eq}}\right)}$$	$\eta_{TC,App,SPS}$ 64.08%	$\eta_{TC,F}$ 64.60%
Turbocharging system efficiency $$\eta_{TS} = \frac{\eta_T}{\eta_{TC}}$$	$\eta_{TS,App,SPS}$ 94.93%	$\eta_{TS,F}$ 94.94%

3 Four stroke engine – pulse turbocharging

Tables 3a and 3b show an example of efficiency calculations for one load point of a 4-stroke engine.

The turbocharging system is built up with one turbocharger (two turbine entries) and a divided exhaust pipe system, also known as a 3-pulse exhaust system.

With regard to the turbocharging system losses, which reduce the system efficiency to about 95%, the following remarks can be made:

- Compressor inlet
  A pipe and a nozzle for flow rate measurement are present at the compressor inlet, leading to pressure loss.

- Compressor outlet
  The pressure loss is incurred here mainly in the charge air cooler as well as in the connecting pipes between the compressor outlet and the cooler, where the air flow is decelerated and part of the available dynamic pressure at the compressor outlet is dissipated.

- Turbine inlet
  The points of measurement in the exhaust manifolds and at turbine inlets are coincident.

- Turbine outlet
  The pressure loss produced by the piping and the stack are artificially increased by means of a flap in order to produce normalised values (test engine).

The comparison of Table 3a with Table 3b shows that all three methods deliver practically the same results.

Since the flow is pulsating (Fig. 3), the efficiencies calculated with measured values are to be classified as apparent. The efficiencies also have been calculated with values derived by means of simulation and by applying the averaging process according to Annex 1. The deviations are in this case large (4 to 5 % points) due to the large variation in the measured pressure. Much larger deviations (apparent efficiency above 100%, "real" efficiency below 50%) can even be expected at part load due to the pulse effect.

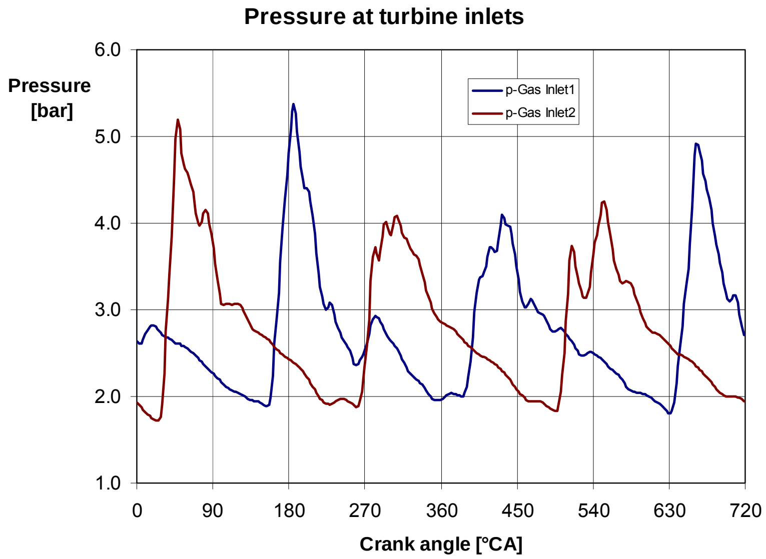


Fig. 3: Simulated static pressure traces at turbine inlets

Table 3a

Calculation of turbocharging efficiencies

4-stroke engine with divided exhaust system (3-pulse)

Measurements (time averaged values) → | ← Simulations (energy averaged values)

		Measurements	Simulations	
Air flow at compressor outlet	$\overline{m}_{Co}$	5.52		kg/s
Air flow to the cylinders	$\overline{m}_{CYi}$	5.52		kg/s
Air Humidity	x	0.62%		kg/kg
Air temperature at system boundary	$\overline{T}^*_{Si}$	299.35		K
Air temperature at compressor inlet	$\overline{T}^*_{Ci}$	299.35		K
Air pressure at system boundary	$p^*_{Si,eq}$	1.022		bar
Air pressure at compressor inlet	$p^*_{Ci,eq}$	1.006		bar
Air pressure at compressor outlet	$p^*_{Co,eq}$	3.631		bar
Air pressure in the inlet manifold	$p^*_{IM,eq}$	3.587		bar
Fuel mass flow	m_{Fuel}	0.16		kg/s
Additional water mass flow	m_{Water}	0.00		kg/s
Gas mass fraction	x_c	0.419		
Gas flow from the cylinders	$\overline{m}_{CYo}$	5.68		kg/s
Gas flow to the turbine	$\overline{m}_{Ti}$	5.68		kg/s
Gas temperature in the exhaust manifold	$\overline{T}^*_{EM}$	788.65	786.7	K
Gas temperature at turbine inlet	$\overline{T}^*_{Ti}$	788.65	786.7	K
Gas pressure in the exhaust manifold	$p^*_{EM,eq}$	2.819	3.097	bar
Gas pressure at turbine inlet	$p^*_{Ti,eq}$	2.819	3.097	bar
Gas pressure at turbine outlet	$p_{To,eq}$	1.06		bar
Gas pressure at system boundary	$p_{So,eq}$	1.02		bar

Fuel correction

$$C_{Fuel} = \frac{\dot{m}_{Co} + \dot{m}_{Fuel}}{\dot{m}_{Co}} \Bigg/ \left(1 + \frac{\dot{m}_{Fuel}}{\dot{m}_{Co}} \cdot 7.184 \cdot x_H\right) \qquad 1.0045$$

Water correction

$$C_{Water} = \frac{\dot{m}_{Air} + \dot{m}_{Fuel} + \dot{m}_{Water}}{\dot{m}_{Air} + \dot{m}_{Fuel} + 1.61 \cdot \dot{m}_{Water}} \qquad 1.0000$$

1. **First approximation**

R_A	288.10	J/(kg K)	k_A	1.3991
R_G	288.07	J/(kg K)	k_G	1.3427

Turbocharging efficiency

$$\eta_T = C_{Fuel} \cdot C_{Water} \cdot \frac{\overline{m}_{CYi} \frac{\kappa_A}{\kappa_A - 1} R_A \cdot \overline{T}^*_{Si} \left[\left(\frac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1\right]}{\overline{m}_{CYo} \frac{\kappa_G}{\kappa_G - 1} R_G \cdot \overline{T}^*_{EM,eq} \left[1 - \left(\frac{p^*_{EM,eq}}{p^*_{So,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]}$$

$\eta_{T,App,Pulse}$ = 62.61% | η_T = 58.10%

Turbocharger efficiency

$$\eta_{TC} = C_{Fuel} \cdot C_{Water} \cdot \frac{\overline{m}_{Co} \frac{\kappa_A}{\kappa_A - 1} R_A \cdot \overline{T}^*_{Ci} \cdot \left[\left(\frac{p^*_{Co,eq}}{p^*_{Ci,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1\right]}{\overline{m}_{Ti} \frac{\kappa_G}{\kappa_G - 1} R_G \cdot \overline{T}^*_{Ti} \cdot \left[1 - \left(\frac{p^*_{Ti,eq}}{p_{To,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]}$$

$\eta_{TC,App,Pulse}$ = 66.34% | $\eta_{TC,F}$ = 61.37%

Turbocharging system efficiency

$$\eta_{TS} = \frac{\eta_T}{\eta_{TC}}$$

$\eta_{TS,App,Pulse}$ = 94.38% | $\eta_{TS,F}$ = 94.67%

Air and fuel specification

TAIR	299.35		
pair	102230		
phiAIR	0.3		
LiqFuel1	C		0.852
LiqFuel1	H		0.117
LiqFuel1	S		0.031
end			
LHV	41175.5	kJ/kg	
AirMin	14.04	kg/kg	

$$x = \frac{0.622}{\frac{p_{Air}}{\varphi \cdot p_{Saturation}(T_{Air})} - 0.378}$$

Table 3b

2.	**Second approximation**		
	$R_A = 287.05 + 1.74 \cdot x$ $T_{mean} = \frac{T^*_{Si}}{2} \cdot \left(1 + \left(\frac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}}\right)$ 363.7948 $\kappa_A = \kappa_A(T_{mean}) - 0.00137 \cdot x$		
	$R_G = R_{G,ref} + 1.70 \cdot (x - 0.6)$ $R_{G,ref}$ = 288.07 $x_{c,ref}$ = 0.40		
	$T_{1,ref}$ = 800 $\Delta T_{mean} = \frac{T^*_{EM}}{2} \cdot \left(1 - \left(\frac{p^*_{EM,eq}}{p^*_{So,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right) = $ 90 97 $\overline{\kappa}_{G,ref}$ = 1.3427		
	$\overline{\kappa}_G = \overline{\kappa}_{G,ref} - 0.01025 \cdot \frac{\overline{T}^*_{EM} - T_{1,ref}}{100} + 0.0075 \cdot \frac{\Delta T_{mean} - 100}{100} - 0.0407 \cdot (x_c - x_{c,ref}) - 0.0012 \cdot (x - 0.6)$		
	R_A 288.14 J/(kg K) k_A 1.3975		
	R_G 288.11 J/(kg K) k_G 1.3423		1.3430
	Turbocharging efficiency $\eta_T = C_{Fuel} \cdot C_{Water} \cdot \frac{\overline{m}_{CYi} \frac{\kappa_A}{\kappa_A - 1} R_A \cdot T^*_{Si} \left[\left(\frac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1\right]}{\overline{m}_{CYo} \frac{\kappa_G}{\kappa_G - 1} R_G \cdot T^*_{EM,eq} \left[1 - \left(\frac{p^*_{EM,eq}}{p^*_{So,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]}$	$\eta_{T,App,Pulse}$ 62.57%	η_T 58.08%
	Turbocharger efficiency $\eta_{TC} = C_{Fuel} \cdot C_{Water} \cdot \frac{\overline{m}_{Co} \frac{\kappa_A}{\kappa_A - 1} R_A \cdot \overline{T}^*_{Ci} \cdot \left[\left(\frac{p^*_{Co,eq}}{p^*_{Ci,eq}}\right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1\right]}{\overline{m}_{Ti} \frac{\kappa_G}{\kappa_G - 1} R_G \cdot T^*_{Ti} \cdot \left[1 - \left(\frac{p^*_{Ti,eq}}{p_{To,eq}}\right)^{\frac{1-\kappa_G}{\kappa_G}}\right]}$	$\eta_{TC,App,Pulse}$ 66.29%	$\eta_{TC,F}$ 61.34%
	Turbocharging system efficiency $\eta_{TS} = \frac{\eta_T}{\eta_{TC}}$	$\eta_{TS,App,Pulse}$ 94.38%	$\eta_{TS,F}$ 94.67%
3.	**Exact method** **Turbocharging efficiency** $\eta_T = \frac{\overline{m}_{CYi} \cdot e_{P,Com}\left(\overline{T}^*_{Si}, \frac{p^*_{IM,eq}}{p^*_{Si,eq}}\right)}{\overline{m}_{CYo} \cdot e_{P,Exp}\left(\overline{T}^*_{EM}, \frac{p^*_{EM,eq}}{p_{So,eq}}\right)}$	$\eta_{T,App,Pulse}$ 62.56%	η_T 58.07%
	Turbocharger efficiency $\eta_{TC} = \frac{\overline{m}_{Co} \cdot e_{P,Com}\left(T^*_{Ci}, \frac{p^*_{Co,eq}}{p^*_{Ci,eq}}\right)}{\overline{m}_{Ti} \cdot e_{P,Exp}\left(\overline{T}^*_{Ti}, \frac{p^*_{Ti,eq}}{p_{To,eq}}\right)}$	$\eta_{TC,App,Pulse}$ 66.28%	$\eta_{TC,F}$ 61.34%
	Turbocharging system efficiency $\eta_{TS} = \frac{\eta_T}{\eta_{TC}}$	$\eta_{TS,App,Pulse}$ 94.39%	$\eta_{TS,F}$ 94.68%

4 Conclusions

The following set of formulae provides an accuracy which is sufficient for quick evaluation of the turbocharging efficiencies on an engine:

$$\eta_T = C_{Fuel} \cdot C_{Water} \cdot \frac{\overline{\dot{m}}_{CYi} \frac{\kappa_A}{\kappa_A - 1} R_A \cdot \overline{T}^*_{Si} \left[\left(\frac{p^*_{IM,eq}}{p^*_{Si,eq}} \right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1 \right]}{\overline{\dot{m}}_{CYo} \frac{\kappa_G}{\kappa_G - 1} R_G \cdot \overline{T}^*_{EM,eq} \left[1 - \left(\frac{p^*_{EM,eq}}{p^*_{So,eq}} \right)^{\frac{1-\kappa_G}{\kappa_G}} \right]}$$

$$\eta_{TC} = C_{Fuel} \cdot C_{Water} \cdot \frac{\overline{\dot{m}}_{Co} \frac{\kappa_A}{\kappa_A - 1} R_A \cdot \overline{T}^*_{Ci} \cdot \left[\left(\frac{p^*_{Co,eq}}{p^*_{Ci,eq}} \right)^{\frac{\kappa_A - 1}{\kappa_A}} - 1 \right]}{\overline{\dot{m}}_{Ti} \frac{\kappa_G}{\kappa_G - 1} R_G \cdot \overline{T}^*_{Ti} \cdot \left[1 - \left(\frac{p^*_{Ti,eq}}{p_{To,eq}} \right)^{\frac{1-\kappa_G}{\kappa_G}} \right]}$$

$$\eta_{TS} = \frac{\eta_T}{\eta_{TC}}$$

with:

$$C_{Fuel} = \frac{\dot{m}_{Co} + \dot{m}_{Fuel}}{\dot{m}_{Co}} \Bigg/ \left(1 + \frac{\dot{m}_{Fuel}}{\dot{m}_{Co}} \cdot 7.184 \cdot x_H \right) \qquad C_{Water} = \frac{\dot{m}_{Air} + \dot{m}_{Fuel} + \dot{m}_{Water}}{\dot{m}_{Air} + \dot{m}_{Fuel} + 1.61 \cdot \dot{m}_{Water}}$$

Engine type		High-speed	Medium-speed	Low-speed
Gas constant for air	R_A [J/(kg K)]	288.10		
Specific heat ratio for air	κ_A	1.3991		
Gas constant for exhaust gas	R_G [J/(kg K)]	288.07		
Specific heat ratio for exhaust gas	κ_G	1.3302	1.3427	1.3562

The nomenclature has been defined with the aim of covering the generality of the possible cases as well as possible. Some effort may be necessary to translate the generic symbols into those familiar to the test engineer for specific cases.

Exceptions and limitations as well as possibilities for improving the accuracy are described within this Recommendation.

6 Acknowledgement

By endorsing this document, CIMAC acknowledges the work accomplished by the CIMAC Working Group "Turbocharger Efficiency" through its worldwide membership. A detailed listing of participating companies, institutions and associations is given on the inside of the back cover.

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Part I: Engine De-rating on Account of Ambient Conditions
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No. 6 Lexicon on Combustion Engines, Technical Terms of the IC Engine and Gas Turbine Industries, 1977

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