Journal of KONES Powertrain and Transport, Vol. 20, No. 3 2013

ISSN: 1231-4005
e-ISSN: 2354-0133
DOI: 10.5604/12314005.1136993

INFLUENCE OF THE EXHAUST GAS RECIRCULATION ON THE OXYGEN CONTENTS AND ITS EXCESS RATIO IN THE ENGINE COMBUSTION CHAMBER

Stefan Postrzednik, Zbigniew Żmudka, Grzegorz Przybyła

Silesian University of Technology
Institute of Thermal Technology
Konarskiego Street 22, 44-100 Gliwice, Poland
tel.: +48 32 2371231 fax.: +48 32 2372872
e-mail: postef@polsl.pl

Abstract

Exhaust gas recirculation (EGR) can be realized on the external or internal way. The main goal of EGR is to decrease the combustion temperature of the combustible mixture by increasing the relative heat capacity of the working medium. An additional effect is to reduce the oxygen content in the gas mixture flowing into the combustion chamber. To quantify the exhaust gas recirculation some descriptive parameters are defined, including: – exhaust gas recirculation rate R, – multiple of exhaust gas recirculation K, – relative exhaust gas recirculation W. The analysis concerns the effective oxygen excess ratio λ_{ef} in the combustible mixture and the so-called internal oxygen excess ratio λ_w,, which additionally takes into account the supplied oxygen with the recirculation stream. It was found that with increasing of the exhaust gas recirculation degree systematically decreases the content of the oxygen $[O_2]_d$ in the combustible mixture, while increases the oxygen excess ratio λ_{ef} observed in the exhaust gases. Changes of this all parameters influence the combustion coditions in the engine cylinder, and next the achieved specifaic work, energy efficiency and emission of the combustion engine. An advanced system with independent, early exhaust valve closing enables realization of an internal EGR. Effectiveness of internal recirculation is lower than the external recirculation.

Keywords: *exhaust gas recirculation, oxygen content, stoichiometric conditions, oxygen excess*

1. Introduction

Exhaust gas recirculation (EGR) means the recycling of the flu gases generated in the device into the combustion chamber of the same unit. The main goal of EGR is to decrease the combustion temperature of the combustible mixture by increasing the relative heat capacity of the working medium. An additional effect is to reduce the oxygen content $[O_2]_d$ in the gas mixture flowing into the combustion chamber.

Both of these effects contribute effectively to a significant reduction of nitrogen oxides $[NO_x]$ emission, produced during fuel-mixture combustion. In this respect, exhaust gas recirculation is a fundamental element of the so-called low-emission fuel combustion.

Exhaust gas recirculation can be realized on the external (bypass of exhaust gases) or internal way. Using the external EGR the mass stream and parameters (e.g. temperature T) of recycled flu gases can be more precise programmed and conditioned.

An advanced new system with e.g. independent early exhaust valve closing enables realization of an internal EGR. Therefore, effectiveness of internal recirculation is significantly lower than the external recirculation. The analysis concerns the effective oxygen excess ratio λ_{ef} in the combustible mixture and the so-called internal oxygen excess ratio λ_w, which additionally takes into account the supplied oxygen with the recirculated stream. On this way effective changes of the exhaust gas recirculation influence the content of the oxygen $[O_2]_d$ in the combustible mixture too. This both parameters influence the combustion conditions in the engine cylinder, and next the achieved specific work, energy efficiency and emission of the combustion engine.

2. Quantitative parameters of the EGR

Exhaust gas recirculation (EGR) is an important element of low-emission fuel combustion. To quantify the exhaust gas recirculation some descriptive parameters are defined, including:

– exhaust gas recirculation rate:

$$R_s \overset{df}{=} \frac{\dot{n}_{s,r}}{\dot{n}_{s,c}}, \qquad 0 \le R_s \le 1 , \tag{1}$$

– multiple of exhaust gas recirculation:

$$K_s \overset{df}{=} \frac{\dot{n}_{s,r}}{\dot{n}_s}, \qquad 0 \le K_s \le \infty , \tag{2}$$

– relative exhaust gas recirculation:

$$W_s \overset{df}{=} \frac{\dot{n}_{s,r}}{\dot{n}_a}, \qquad 0 \le W_s \le \infty , \tag{3}$$

where:

$\dot{n}_{s,r}$, *kmol/s* – stream of recirculated exhaust gas (recirculation) within the system,

$\dot{n}_{s,c}$, *kmol/s* – the total flue gas stream flowing from the combustion chamber,

$\dot{n}_s$, *kmol/s* – exhaust gas stream (net) flowing out from the system to the environment.

The above listed definitions of EGR parameters illustrate a model system shown in the Fig. 1.

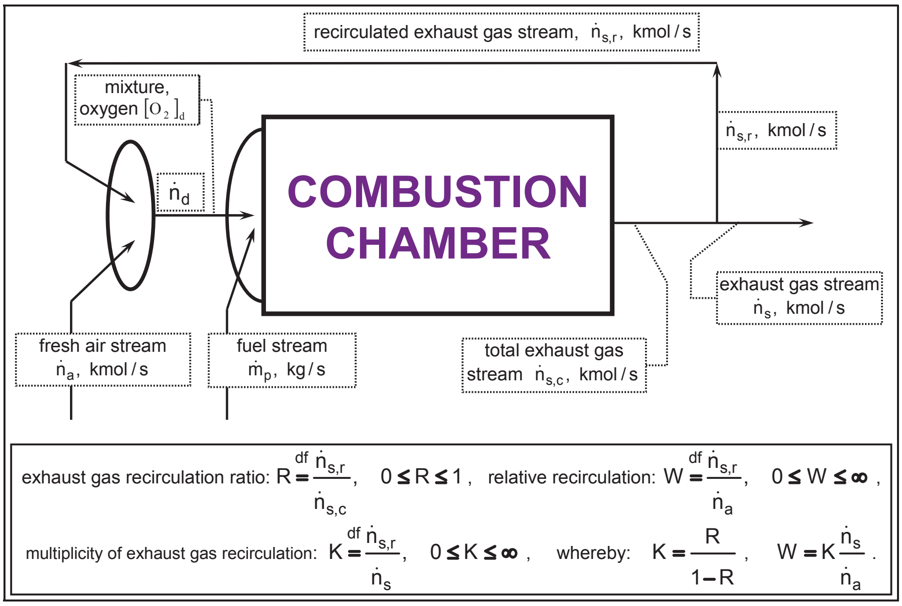

Fig. 1. Typical system with the external exhaust gas recirculation

The substance balance of the exhaust gas node (at the outlet, fig. 1) follows the relationship:

$$\dot{n}_{s,c} = \dot{n}_{s,r} + \dot{n}_s \ , \tag{4}$$

from which, having regard to the definitions (1), (2), (3), the relationships are obtained:

$$K_s = \frac{R_s}{1 - R_s}, \quad W_s = K_s \left(\frac{\dot{n}_s}{\dot{n}_a} \right). \tag{5}$$

Exhaust gas stream $\dot{n}_s$, *kmol/s* flowing out from the system to the environment is closely associated with a stream of fresh air $\dot{n}_a$, *kmol/s* flowing into the system, and therefore usually holds the following inequality $\dot{n}_s > \dot{n}_a$, from where results relation: $W_s > K_s$.

Likewise, with regard to the dry flue gases can be enter the appropriate quantitative parameters of the flu gas recirculation, which include:

– dry exhaust gas recirculation rate:

$$R_{ss} \overset{df}{=} \frac{\dot{n}_{ss,r}}{\dot{n}_{ss,c}}, \qquad 0 \le R_{ss} \le 1, \tag{6}$$

– multiple of dry exhaust gas recirculation:

$$K_{ss} \overset{df}{=} \frac{\dot{n}_{ss,r}}{\dot{n}_{ss}}, \qquad 0 \le K_{ss} \le \infty, \tag{7}$$

– relative dry exhaust gas recirculation:

$$W_{ss} \overset{df}{=} \frac{\dot{n}_{ss,r}}{\dot{n}_{as}}, \quad 0 \le W_{ss} \le \infty, \tag{8}$$

where:

$\dot{n}_{ss,r}$, *kmol/s* – substance stream of the recirculated dry exhaust gas within the system,

$\dot{n}_{ss,c}$, *kmol/s* – dry flue gas stream flowing from the combustion chamber,

$\dot{n}_{ss}$, *kmol/s* – dry exhaust gas stream (net) flowing out from the system to the environment.

The stream $\dot{n}_{ss}$, *kmol/s* of dry exhaust gas flowing out from the system into the environment is closely linked with the dry air stream $\dot{n}_{as}$, *kmol/s*, flowing into the system, and therefore usually the following inequality holds: $\dot{n}_{ss} < \dot{n}_{as}$, from where results relation: $W_{ss} < K_{ss}$.
The quotient of these mentioned streams is as follows:

$$\frac{\dot{n}_{ss}}{\dot{n}_{as}} = \frac{1}{\lambda_{ef}}\left[\left(\frac{n''_{ss,\min}}{n'_{as,\min}}\right) + (\lambda_{ef} - 1)\right], \tag{9}$$

where:
λ_{ef} – effective air (oxygen) excess ratio, and stoichiometric quantities:
$n'_{as,\min}$, $n''_{ss,\min}$ depend only from the composition (elements) of the combusted fuel,
e.g. for the standard composition (for given mass contents: c = 0.84, h = 0.14; o = 0.01, n = 0.005, s = 0.005) of engine fuel can be taken: $\dfrac{n''_{ss,\min}}{n'_{as,\min}} = \dfrac{0.4647}{0.4996} = 0.9302.$

For the global system (fig. 1, when recirculation located inside the control surface) it may be written an effective λ_{ef} air (oxygen) excess ratio, at assuming that the combustion is complete and total, then:

$$\dot{n}_{ss} = \dot{n}_{ss,\min} + (\lambda_{ef} - 1) \cdot \dot{n}_{as,\min} \tag{10}$$

and using (7), the substance stream of the recirculated dry exhaust gas results:

$$\dot{n}_{ss,r} = K_{ss} \cdot \left[\dot{n}_{ss,\min} + (\lambda_{ef} - 1) \cdot \dot{n}_{as,\min}\right]. \tag{11}$$

The oxygen content $[O_2]_s$ in combustion products leaving the system can be calculated using relation:

$$[O_2]_s \overset{df}{=} \frac{\dot{n}_{O_2,s}}{\dot{n}_{ss}} = \frac{0{,}21 \cdot (\lambda_{ef} - 1)}{\dfrac{\dot{n}_{ss,\min}}{\dot{n}_{as,\min}} + (\lambda_{ef} - 1)}, \tag{12}$$

and principally is independent from the recirculation ratio R occurring in the system.

Taking into account that the combusted fuel is a standard engine fuel: (mass contents: c = 0.84, h = 0.14, o = 0.01, n = 0.005, s = 0.005), then on base of eq. (12) can be written:

$$[O_2]_s = \frac{0.21\cdot(\lambda_{ef}-1)}{0.9302+(\lambda_{ef}-1)}. \tag{13}$$

With the increase of the effective oxygen excess ratio λ_{ef} the oxygen content $[O_2]_s$ in dry flue gas leaving the system increases, approaching to the value of oxygen standard content (0.21) in the air, supplied from the environment into the system.

By measuring the oxygen content $[O_2]_s$ in the flue gas and next using the achieved eq. (13) the values of actual effective air excess ratio λ_{ef} can be determined.

It can be proved on the basis of the given definitions, that at the end occur some beneficial identities:

$$\mathrm{R_{ss}} \equiv \mathrm{R_s} = \mathrm{R}, \qquad 0 \le \mathrm{R} \le 1, \tag{14}$$

$$\mathrm{K_{ss}} \equiv \mathrm{K_s} = \mathrm{K}, \qquad 0 \le \mathrm{K} \le \infty, \tag{15}$$

furthermore also:

$$R = \frac{K}{1+K}, \tag{16}$$

whereas:

$$\mathrm{W_s} = \mathrm{W_{ss}} \cdot \frac{1+\mathrm{X_{z,s}}}{1+\mathrm{X_{z,a}}}, \tag{17}$$

where:

$X_{z,a}$, $X_{z,s}$ – molar humidity degree (respectively of the air and exhaust gas),
and as usually: $X_{z,s} > X_{z,a}$ so therefore: $W_s > W_{ss}$.

3. Effective and internal oxygen excess ratio and its content

The combustion of fuel inside the system with realized exhaust gas recirculation does not occur in an atmosphere of pure air but in a mixture of air and exhaust gas. The essential is the presence of oxygen in the real combustible mixture. Oxygen flows into the combustion chamber with both fresh air stream in an amount of [$0.21\cdot \dot{n}_{as}$], as well as a component of the recirculating exhaust gas stream, in an amount $\dot{n}_{O_2,sr}$.

An illustration of this is the diagram shown in the Fig. 2. The total oxygen stream $\dot{n}_{O_2}$ which flows into the combustion chamber is: $\dot{n}_{O_2} = \dot{n}_{O_2,a} + \dot{n}_{O_2,sr}$, and then:

$$\dot{n}_{O_2} = 0.21\cdot\dot{n}_{as} + [O_2]_s\cdot\dot{n}_{ss,r}\,. \tag{18}$$

It is expedient to define the so-called. internal oxygen excess ratio λ_w , which is:

$$\lambda_w \overset{df}{=} \frac{\dot{n}_{O_2}}{0.21\cdot\dot{n}_{O_2,a}} = \frac{\dot{n}_{O_2}}{0.21\cdot n'_{as,\min}\cdot\dot{m}_p}, \tag{19}$$

and determines the fuel burn conditions in the combustion chamber.

After taking into account formulas (9), (18) and using the recirculation rate R – form. (1), the equation (19) takes next the finally form:

$$\lambda_w = \lambda_{ef} + \frac{R}{1-R}(\lambda_{ef}-1) \tag{20}$$

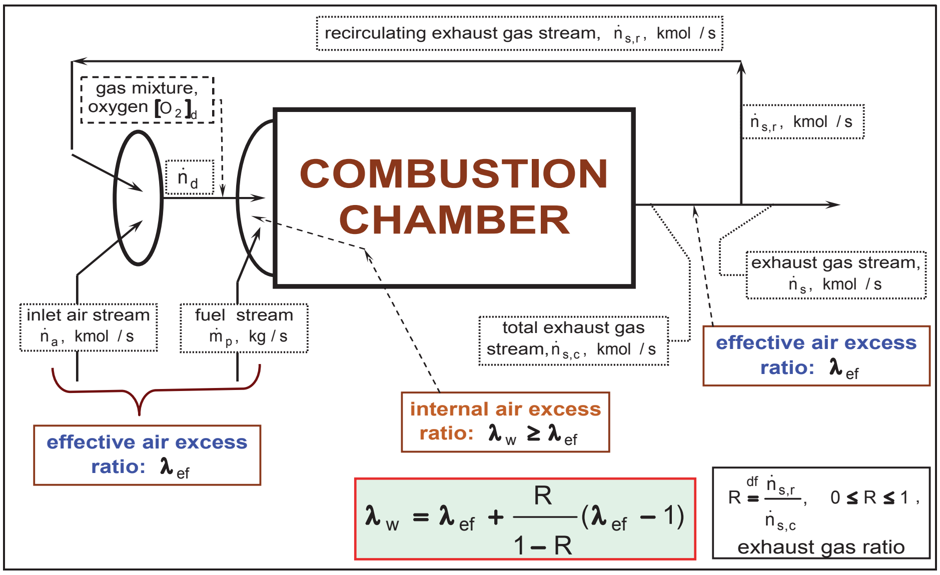

Fig. 2. Air (oxygen) excess ratio in the system with external exhaust gas recirculation

and next:

$$\lambda_w = \lambda_{ef} + K(\lambda_{ef} - 1). \tag{21}$$

From eq. (20) and (21) it shows clearly that the oxygen excess ratios: $\lambda_w > \lambda_{ef}$. Further an increase of the exhaust gas recirculation rate R, while keeping constant value of the effective air (oxygen) excess ratio λ_{ef} = idem, the value of the internal oxygen excess ratio λ_w gradually increases, as illustrated in the Fig. 3.

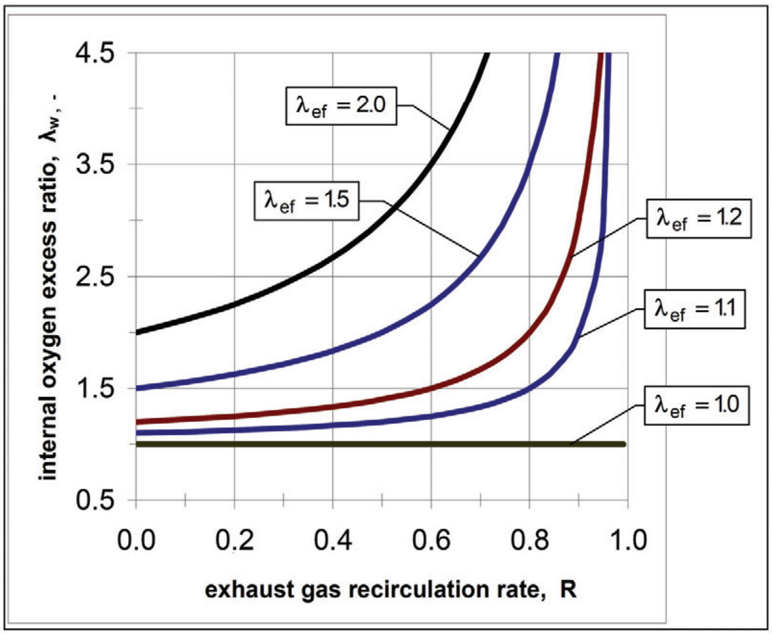

Fig. 3. Influence of the exhaust gas recirculation rate R on the internal oxygen excess ratio λ_w

Usually with an increase of the exhaust gas recirculation stream is reduced flow $\dot{n}_{as}$ of fresh combustion air, and thus the stream $\dot{m}_p$ of the fuel consumed (while maintaining the unchanged value of the effective air (oxygen) excess ratio λ_{ef} = idem). This is somewhat connected with a decrease of the effective engine torque M_e and next its power N_e, which is proportional to the stream $\dot{m}_p$ of the fuel consumed.

Exhaust gas recirculation causes that the combustion chamber is fed with the stream of gaseous mixture $\dot{n}_d$, *kmol/s*, which is formed after mixing the two streams: fresh air $\dot{n}_a$ and exhaust gas recirculating $\dot{n}_{s,r}$, that is:

$$\dot{n}_d = \dot{n}_{s,r} + \dot{n}_a , \qquad \dot{n}_{ds} = \dot{n}_{ss,r} + \dot{n}_{as} , \tag{22}$$

whereby the second equation relates to the dry gas occurring in the system.

Interesting is the oxygen content $[O_2]_d$ in the gas stream at the inlet to the chamber, which is:

$$[O_2]_d \overset{df}{=} \frac{\dot{n}_{O_2,d}}{\dot{n}_{d,s}} = \frac{0.21 \cdot \dot{n}_{a,s} + [O_2]_s \cdot \dot{n}_{ss,r}}{\dot{n}_{ss,r} + \dot{n}_{a,s}}, \tag{23}$$

where equations (12), (22) should be taken into account.

In order to use the degree (rate) of recirculation R, then using the formula (23) can be written:

$$[O_2]_d = \frac{0.21 \cdot \left[1 + \frac{R}{(1-R)} \frac{(\lambda_{ef} - 1)}{\lambda_{ef}}\right]}{1 + \frac{R}{(1-R)\,\lambda_{ef}} \cdot \left[\left(\frac{n''_{ss,\min}}{n'_{as,\min}}\right) + (\lambda_{ef} - 1)\right]}, \tag{24}$$

and then for the standard engine fuel (at mass contents: c = 0.84, h = 0.14, o = 0.01, n = 0.005, s = 0.005, and $\frac{n''_{ss,\min}}{n'_{as,\min}} = \frac{0.4647}{0.4996} = 0.9302$) the relationship can be written:

$$[O_2]_d = \frac{0.21 \cdot \left[1 + \frac{R}{(1-R)} \cdot \frac{(\lambda_{ef} - 1)}{\lambda_{ef}}\right]}{1 + \frac{R}{(1-R)} \cdot \frac{(\lambda_{ef} - 0.0698)}{\lambda_{ef}}}. \tag{25}$$

Equation (25) shows that an increase of the recirculation relation rate R, the oxygen content $[O_2]_d$ at the inlet to the combustion chamber steadily decreases and approaches the values of oxygen $[O_2]_s$ - commonly found in the flue gases leaving the system.

Using equation (25) in the Fig. 4 is shown the oxygen content $[O_2]_d$ in the mixture of gases at the flow into the combustion chamber (see Fig. 2), as a function of the exhaust gas recirculation rate R, while retaining the value of the effective air (oxygen) excess ratio λ_{ef}.

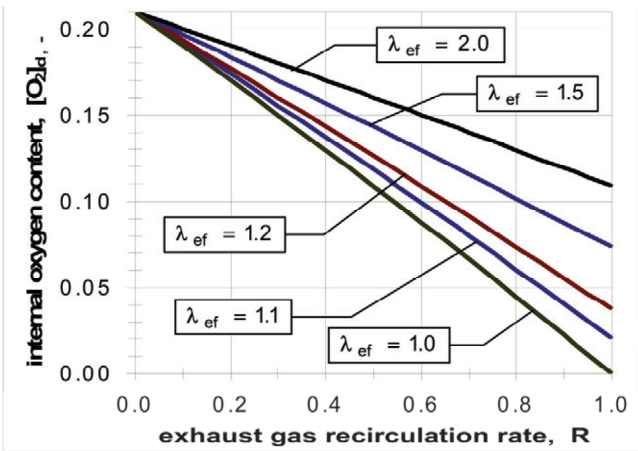

Fig. 4. Influence of the exhaust gas recirculation rate R on the internal oxygen content $[O_2]_d$

By measuring the oxygen content $[O_2]_d$ in the gas mixture at the inlet channel and next using the achieved eq. (25) the values of actual exhaust gas recirculation rate R can be effective determined.

The obtained relationships inform that the increase of the exhaust gas recirculation rate R causes systematically increasing (Fig. 3) of the internal oxygen excess ratio λ_w, while at the same time decreasing the oxygen content $[O_2]_d$ of combustible mixture (Fig. 4) – all for the effective air (oxygen) excess ratio λ_{ef}.

In case of internal recirculation the flue gas temperature equals (or is higher) the temperature of flue gas leaving the combustion chamber, so the effectiveness of the internal recirculation is not so high than the external recirculation. In the case of an external recirculation temperature of $T_{s,r}$ is close to the temperature of exhaust gases leaving the engine, or may be lower if cooling of recirculating gas is performed.

4. Closing remarks

The fuel combustion process, in the system with implemented within the exhaust gas recirculation takes place in a mixture of air and exhaust gases; for the combustion process is important the effective content (concentration) of oxygen $[O_2]_d$ in the combustible mixture. Oxygen flows into the combustion chamber with both fresh air stream, as well as a component of the recirculating exhaust gas stream.

The analysis also concerns the effective excess oxygen ratio λ_{ef} in the combustible mixture, which is observed directly in the exhaust manifold. The increase of the exhaust gas recirculation rate R causes systematically decreasing the oxygen content $[O_2]_d$ in combustible mixture, for all values of the effective air (oxygen) excess ratio $\lambda_{ef} \geq 1$.

For a full evaluation of stoichiometric conditions it was advisable to define the internal oxygen excess ratio λ_w too, in which account is also the oxygen supplied with the recirculating exhaust gas stream. It was found that an increase of the exhaust gas recirculation rate R reduces the content of oxygen in the combustible mixture and simultaneously increases the internal oxygen excess ratio λ_w.

Acknowledgement

The work was supported and performed by using of statutory funding for research on faculty ISiE of STU.

References

[1] Hribernik, A., Samec, N., *Effect of Exhaust Gas Recirculation on Diesel Combustion*, Journal of KONES, IC Engines, No. 1-2, 2004.

[2] Kowalewicz, A., *Systemy spalania szybkoobrotowych tłokowych silników spalinowych*, WKiŁ, Warszawa 1990.

[3] Merkisz, J. J., Pielecha, I., *Alternatywne paliwa i układy napędowe pojazdów*, Wydawnictwa Politechniki Poznańskiej, 2004.

[4] Müller, M., Olin, P., Schreurs, B., *Dynamic EGR Estimation for Production Engine Control*, SAE Paper 2001-01-0553.

[5] Pietras, D., Sobieszczański, M., *Problemy regulacji silnika o zapłonie iskrowym z recyrkulacją spalin*, Silniki spalinowe, Nr 2, (119), 2004.

[6] Postrzednik, S., Żmudka, Z., *Termodynamiczne oraz ekologiczne uwarunkowania eksploatacji tłokowych silników spalinowych*, Wydawnictwo Politechniki Śląskiej, Gliwice 2007.

[7] Roth, D., Sauerstein, R., Becker, M., Meiling, R., *Application of hybrid EGR systems to turbo charged GDI engines*, MTZ, 04, 2010.