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Collection and evaluation of data and development of test procedures in support of legislation on mobile air conditioning (MAC) efficiency and gear shift indicators (GSI)

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Summary

The work described in this report has been conducted in the project „Collection and evaluation of data and specification/development of test procedures“ in support of legislation on mobile air conditioning (MAC) efficiency and gear shift indicators (GSI) within the „FRAMEWORK SERVICE CONTRACT ENTR/05/18“ of the European Commission. This report covers the work for:

Task1; Support for the impact assessment. Information was gathered for the support which can be found in Annex B of this report

Task 2; Development of test conditions and physical test procedures for MAC and GSI efficiency. This is the main part of the work and this report.

The work for Task 3 consists of a study into the costs and effectiveness of a GSI and a Fuel Consumption Meter, a study into the impact of refrigerant loss on the efficiency of a MAC system and general consultancy support for the EC. The two topics have been reported separately.

<u>Background</u>
According to the Communication of the Commission on the review of the Community Strategy to reduce CO2 emissions from passenger cars and light-commercial vehicles adopted in February 2007;

"[t]he Commission will propose a legislative framework, if possible in 2007 and at the latest by mid 2008, to achieve the EU objective of 120 g CO2/km, focusing on mandatory reductions of the emissions of CO2 to reach the objective of 130 g CO2/km for the average new car fleet by means of improvements in vehicle motor technology, and a further reduction of 10 g CO2/km, or equivalent if technically necessary, by other technological improvements and by an increased use of bio-fuels, specifically:
a) setting minimum efficiency requirements for air-conditioning systems …
…
d) the use of gear shift indicators, taking into account the extent to which such devices are used by consumers in real driving conditions;"

Consequently, the Commission has to prepare a proposal for legislation on MAC and GSI efficiency in due course. Following a public consultation, which finished on 13 May 2008, the Commission is investigating several regulatory options.

<u>Objective</u>
A procedure is required to develop vehicle based efficiency figures of the whole system including the MAC and also for the GSI system. These figures could be the basis for the legislation to be developed and should thus be able to incentivise the application of more efficient MAC systems. To prepare for the development of legislation the Commission therefore decided that at first the options for a physical test procedure for MAC and GSI should be investigated. In parallel the data required for an impact assessment should be gathered.

The test procedure to be developed for MAC should be based on a whole system approach and should be a physical test procedure. The reason for this first basic choice

is that it is found important to derive real efficiency figures and it is generally known that not only the MAC system itself but also many other vehicle aspects (cabin, glazing, packaging, etc.) influence the additional FC caused by MAC use and should be therefore taken into account for the procedure. The Commission decided that at a later stage options for virtual testing could be investigated.

The goal of this study is therefore to investigate and develop options for a physical test procedure for the MAC and the GSI. Computer simulations and a laboratory emission testing programme supported the development of the procedure and the options.

Earlier investigation showed that a complete environmental simulation in an expensive climate chamber, applying a dynamic driving cycle (NEDC) resulted in poor repeatability and reproducibility (Vermeulen R., TNO 2005) of a procedure. For the current exercise, therefore the focus was especially on repeatability, reproducibility and workability, yet maintaining the capability to deliver representative figures able to discriminate MAC systems of different quality grades.

Results

The result of the exercise is a proposal for a physical test procedure. The main elements have been drafted and are described in this report. The proposed procedure for the efficiency or additional fuel consumption of a MAC consists of a test programme to be executed on a (M1) vehicle on a chassis dynamometer in an emission test laboratory. The vehicle drives a driving cycle with two steady-state speed periods and an idle period under moderate ambient conditions (temperature and humidity). The vehicle is once driven with the MAC off and once driven with the MAC on, at prescribed settings. The difference in Fuel Consumption between the tests is the additional fuel consumption due to the MAC activity. In the study also the exact conditions and margins were defined for the procedure. Special attention was paid to the setting of the MAC, the measurement of cabin temperature, stability of test room ambient conditions, repeatability of the driving cycle and the reproducibility between test houses. To further improve the repeatability and reproducibility correction functions are proposed for some less stable parameters. Optionally, the difference in cooling demand for different quality and size of the glazing can be accounted for in the procedure by means of an additional fuel consumption, by adjusted test room conditions or MAC settings.

In parallel, a group of specialists from the industry ran a separate programme and has tested the procedure on the general criteria. This programme also concluded that the procedure is repeatable and reproducible. This programme was even able to test different technology grades for MAC systems and different engine types and this lead to the conclusion that the procedure is able to discriminate MAC systems of different technology grades.

Recommendations

Since the number of tested vehicles was limited in this project and most of the test resources was needed to elaborate details of the test procedure only a few “valid” tests according to the finally suggested test procedure are available. Thus it is suggested to launch a pilot phase of the MAC type approval tests. Labs participating in this pilot phase should have a test stand with conditioning of temperature and humidity. It is suggested to include one or more partners from the current project in this, JRC as official EC test facility and partners from industry as well as type approval authorities and technical services.

The results of the pilot phase could be used to define a suitable classification method for the MAC systems according to the test results. The results of the current project suggest either to inform the customer on the additional MAC fuel consumption in [l/100km] and/or to use the result in [kg/h] to classify the MAC system into energy efficiency classes. A scheme for the classification has to be based on a representative number of test results which is not available yet.

Contents

1 Symbols and abbreviations

be............... specific fuel consumption of an engine [g/kWh]
CAP........... Cooling Capacity of the MAC system
COP........... Coefficient of performance (CAP / MAC-compressor work)
CVS........... Constant Volume Sampling (dilution system for emission measurements)
DPF........... Diesel particle filter
FC .... Fuel consumption
MAC .... Mobile Air Conditioning
P_{ce}.............. Power demand of the MAC compressor [kW]
P_{el} .... Power demand of the electric devices in a MAC system [kW]
RH............. Relative Humidity [%]
SOC........... State Of Charge of the battery of the vehicle.
SPF............ Seasonal Performance (simulation of the MAC fuel consumption over a year)
T_a............... Temperature in the test cell at the inlet of the blower for the air flow to the vehicle
T_{C3}............. Temperature measured in the cabin in the altitude of the drivers head at three positions (behind driver, behind co-driver and in the middle between the front seats
T_{V3}............. Temperature measured in the cabin at the central position of each vent outlet in the front area of the vehicle. Other vent outlets shall be closed.
η_{el} .... efficiency for production of electric energy on board [-]
φ_a............... Relative humidity in the test cell at the inlet of the blower for the air flow to the vehicle

2 Introduction

The work described in this report has been conducted in the project „Collection and evaluation of data and specification/development of test procedures“ in support of legislation on mobile air conditioning (MAC) efficiency and gear shift indicators (GSI) within the „FRAMEWORK SERVICE CONTRACT ENTR/05/18“ of the European Commission.

Structure of the report
First, in chapter 3 the proposed test procedure for the Mobile Air Conditioning systems (MAC) is described. This chapter is followed by proposed options for the assessment of the efficiency of Gear Shift Indicators (GSI) in chapter 4. The proposed test procedures are developed based on the input of a few exercises, model simulations runs and emission laboratory vehicle tests. These activities, the results and the elaboration of the results to come to clear choices for the procedures are described in the chapters afterwards, namely; chapter 4 describes the analysis of parameters that influence the additional fuel consumption due to MAC use. Chapter 5 describes the development of correction factors and chapter 6 describes the vehicle Fuel Consumption tests performed in an emission laboratory.

A lot of very useful input for the MAC test procedure was provided by ACEA (see Annex A) and Saint-Gobain Sekurit.

In Annex B the results are summarized for the exercise “Support for the Impact Assessment”.

Background
According to the EC Framework Service Contract ENTR/05/18 the objective and the subject of this project can be described as follows:

According to the Communication of the Commission[1] on the review of the Community Strategy to reduce CO_2 emissions from passenger cars and light-commercial vehicles adopted in February 2007

> *“[t]he Commission will propose a legislative framework, if possible in 2007 and at the latest by mid 2008, to achieve the EU objective of 120 g CO_2/km, focusing on mandatory reductions of the emissions of CO_2 to reach the objective of 130 g CO_2/km for the average new car fleet by means of improvements in vehicle motor technology, and a further reduction of 10 g CO_2/km, or equivalent if technically necessary, by other technological improvements and by an increased use of bio-fuels, specifically:*

[1] *Communication from the Commission to the Council and the European Parliament 6 Results of the review of the Community Strategy to reduce CO2 emissions from passenger cars and light-commercial vehicles*; /* COM/2007/0019 final */, 2 February 2007.

a) setting minimum efficiency requirements for air-conditioning systems …

…

d) the use of gear shift indicators, taking into account the extent to which such devices are used by consumers in real driving conditions;"

Consequently, the Commission has to prepare a proposal for legislation on MAC and GSI efficiency in due course. Following a public consultation, which finished on 13 May 2008, the Commission is investigating several regulatory options.

Subject of the service request

The contractor will support the Commission;

1. for preparing the underlying impact assessment (Task 1) and
2. by developing test conditions and procedures for MACs and GSIs (Task 2)
3. by providing a package of consultant services, the details of which will be specified on demand (Task 3)

This report deals mainly with 2. Information gathered for preparing the impact assessment is summarized in the Annex B of this report. The result of the services meant under 3 are reported separately.

Objective

A procedure is required to develop vehicle based efficiency figures of the whole system including the MAC and also for the GSI system. These figures could be the basis for the legislation to be developed and should for instance be able incentives more efficient MAC systems. To prepare for the development of legislation the Commission therefore decided that at first the options for a physical test procedure for MAC and GSI should be investigated. In parallel the data required for an impact assessment should be gathered.

The test procedure to be developed for MAC should be based on a whole system approach and should be a physical test procedure. The reason for this first basic choice is that it is found important to derive real efficiency figures and it is the general idea that not only the MAC system itself but also many other vehicle aspects (cabin, glazing, packaging, etc.) influence the additional FC caused by MAC use and should be therefore taken into account for the procedure. The Commission decided that at a later stage options for virtual testing could be investigated.

The goal of this study is therefore to investigate and develop options for a physical test procedure for the MAC and the GSI. Computer simulations and a laboratory emission testing programme supported the development of the procedure and the options.

Earlier investigation showed that a complete environmental simulation in an expensive climate chamber, applying a dynamic driving cycle (NEDC) resulted in poor repeatability and reproducibility (Vermeulen R., TNO 2005). For the current exercise therefore the focus was especially on repeatability, reproducibility and workability, yet maintaining the capability to deliver representative figures able to discriminate MAC systems of different quality grades.

Fundamentals
The fuel consumption of a Mobile Air Conditioning (MAC) system depends on numerous factors, which would make full representative testing, with e.g solar radiation and a wind tunnel, quite extensive. Furthermore, it has been shown in a past project that the repeatability of the test results might be poor (TNO, 2005). Figure 1 gives a brief overview of the factors influencing the fuel consumption of the MAC system. As shown in Figure 12 they can be separated in two parts:

All factor that stay the same for all cars, like
- ambient conditions
- driving cycle
- user habit

All factors that have to be determined for each vehicle individually, like
- cooling demand
- efficiency of cold production
- vehicle data

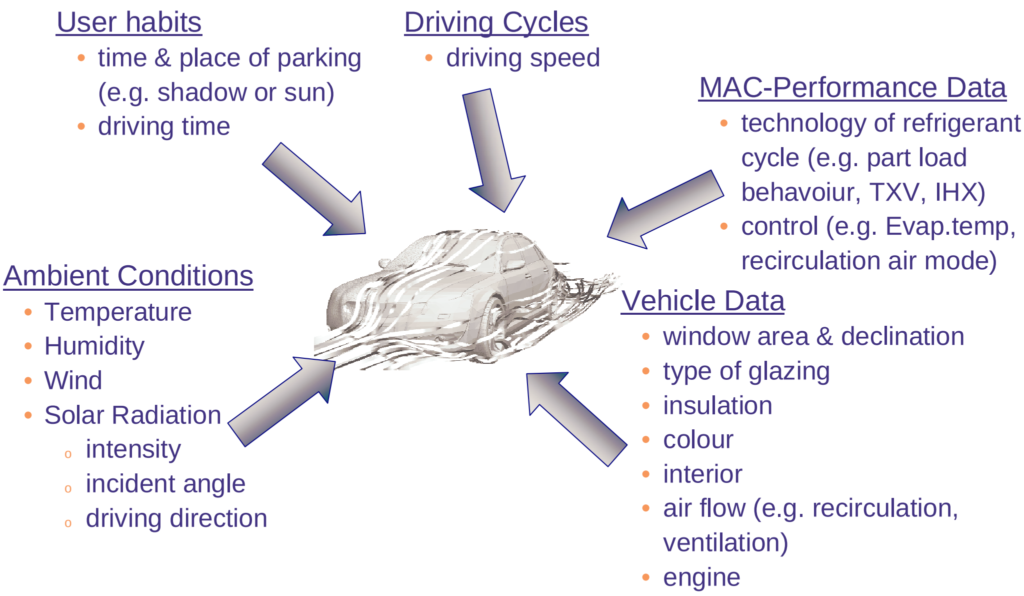


Figure 1: Overview of factors influencing the fuel consumption of the MAC system

Vehicles were investigated in this study in a computer simulation exercise to obtain input for the development of the procedure with a goal to reflect average conditions for European weather. The vehicle specific data was analysed at the beginning in a ‘seasonal performance’ simulation for different regions in Europe to identify those factors which have a high influence on the resulting fuel consumption from the MAC system. To keep the test procedure repeatable, reproducible and cost efficient, the selected test conditions had to be a simplification of the complex reality. With exception of the colour of the vehicle however, the suggested test procedure should in principle cover all parameters shown in Figure 1.

3 Proposal for a MAC test procedure

The MAC test procedure should be able to incentivise energy efficient technologies for all relevant parts of the air conditioning system of the vehicle and all related systems that influence the efficiency at the whole vehicle level. In parallel the test procedure shall be repeatable, reproducible, fair, workable, robust, able to discriminate technology grades and cost efficient.

The proposal, resulting from the underlying investigation of whole vehicle emission tests, simulations and desk research is a physical test with the entire vehicle on a chassis dynamometer;

The basic elements of the proposed procedure are;
- A physical emission (Fuel Consumption) test on the whole vehicle
- Driving the driving cycle on a chassis dynamometer, in an emission laboratory
- Driving a driving cycle with two times the same three steady state phases; the first three run with the MAC switched on and the next three run with the MAC switched off
- Testing under moderate ambient conditions in a test room that does not require expensive facilities
- Application of correction functions for some less stable parameters, for the improvement of the repeatability
- Application of correction values or altered MAC settings for differences in system properties (like glazing)
- Calculation of the additional fuel consumption due to MAC activity from the difference in fuel consumption between the test part with MAC on and with MAC off.

Since the additional fuel consumption from the MAC system is calculated as difference of two measured values which are approximately one order larger than the considered MAC fuel consumption value, a high accuracy is necessary in the test procedure to obtain correct results. To improve the accuracy some correction factors are proposed to be applied to the measured results to increase the repeatability (correction for deviations of the vehicle speed and of the temperature and humidity during the test). The influence of sun radiation is suggested to be depictured as function of the glazing size and thermal properties by a variation of the air flow through the MAC or by simple look-up tables for additional fuel consumption due to additional load to the MAC system to cool the heat entrance due to sun radiation.
In the following chapters the single parts of the test procedure are described. The background and details to the measurement results can be found in the chapters 5 to 7.

The test bed shall fulfil the definitions given in the EC type approval regulations for emission standards of passenger cars and light duty vehicles (EC 692/2008 in the actual amendment).

3.1 Soak Phase and Pre Conditioning Cycle

The MAC test cycle includes a preconditioning phase (see also chapter 3.3). Additionally, it is recommended to soak the vehicle in advance of the actual tests. This seems to be necessary to achieve a defined vehicle temperature (i.e. the test cell temperature) and the vehicle running in an extra preconditioning cycle shall bring relevant vehicle parts to a defined status (e.g. loading of the DPF).

The following preconditioning and soak procedure is suggested before the MAC test starts:
- Set up the vehicle at the test bed which has a temperature controlled between 20°C and 30°C.
- Position the driving resistance values and the fly wheel mass of the roller test bed according to the standard test procedure for cars (EC 692/2008).
- Set the MAC system of the vehicle to "automatic position"
- Start the vehicle and run one NEDC
  (+ additional EUDC if necessary for better DPF regeneration?
- Conditioning of the SOC of the battery. It may be a reasonable approach, to start one test with maximum SOC and another test with minimum SOC and average the test results or to apply a correction based on the energy flow from and to the battery measured during the test[2]. This needs more test results (→ questionnaire to JRC, TÜV's, DEKRA, ACEA and other stakeholders?
  Furthermore illogical strategies for battery loading should be forbidden, e.g. charging during preconditioning at 90 km/h but discharge at 50km/h and 100km/h.
- Parking of the vehicle longer than 8h at the test cell temperature defined for the vehicle in the MAC test (25°C +/- additional x°C for low glazing quality, see chapter 6.8).
- Start the test cycle

3.1.1 *Option for an incentive to deactivate the MAC at low temperatures*

A relevant part of the energy consumption from MAC systems is caused in Europe at temperatures below 18°C. According to (Weilenmann et.al., 2010) "two-thirds of CO_2 and fuel consumption from MAC activity could be saved without discomfort by switching off the MAC below 18 °C. The saving would be 7%, 1.5%, and 0.75% of overall fuel consumption for urban, rural, and motorway driving, respectively".
To include temperatures below 18°C directly in the test procedure described in chapter 3.3 would demand more test repetitions at different temperature levels and make the tests more expensive. Since a check if the MAC system is active does not need any exhaust gas measurements but only the inspection of the MAC status, the MAC strategy at lower ambient temperatures may be tested during the preconditioning phase. The alternative set up could look like described here:

- Set up the vehicle at the test bed which has a temperature controlled between 18°C and 21°C and a relative humidity above 60%.
- Position the driving resistance values and the fly wheel mass of the roller test bed according to the standard test procedure for cars (EC 692/2008).
- Set the MAC system of the vehicle to "automatic position"
- Parking of the vehicle up to 4h.
- Start of the vehicle and run one NEDC
  (+ additional EUDC if necessary for better DPF regeneration?)
- Conditioning of the SOC of the battery

[2] If the test starts with full loaded battery and ends with empty battery, the MAC system could be driven to a large extent from energy of the battery during the test. Thus the SOC will be very relevant to obtain tamper-resistant test results. The options to include the SOC into the test procedure have not been tested yet.

– Parking of the vehicle longer than 8h at 50% relative humidity and a test cell temperature defined for the vehicle in the MAC test (either 25°C or 25°C + additional x°C for low glazing quality, see chapter 6.8).

If the MAC system is not activated after starting the engine for preconditioning and during the entire NEDC the MAC system is awarded by for instance a bonus factor. A suggestion would be to use a bonus in the range of -20% up to -50% of the additional fuel consumption for the results obtained from the standard MAC test procedure, see also chapter 3.4. The bonus for cold temperature efficiency is provided, if the manufacturer also guarantees that the basic strategy of the AC system will avoid activation at ambient temperatures below 18°C and at sun radiation below 200W/m² at humidity levels which do not lead to fogged windows. An active de-humidification of the intake air with the MAC system can be necessary in several ambient conditions (e.g. high humidity with fogged or frosted windows). Such a behavior is important for safety issues and shall not be prohibited by the regulation. However, such functions can either be activated manually by the driver or can be controlled by sensors.

It is not clear yet if such a strategy would have reasonable disadvantages for the users. It may be necessary then to bypass the evaporator of the AC system if the AC system is off since negative impacts on the odor of the air flow have been reported if the AC system was off and the mass flow still on. All of the tested vehicles had the MAC system active also at temperatures below 18°C. Thus not enough experience was gained in the study for a final decision. We suggest to collect literature and experience on this topic before this step is decided by means of a questionnaire to the relevant stakeholders.

3.2 In-car and test cell settings for temperature, humidity and mass flow

For the MAC test, the positions of the sensors as well as the target values for temperature and humidity have been defined.

Position of the temperature sensors

During the soak time as well as during the MAC test cycle the temperature in the test cell (T_a) and the relative humidity have to be controlled. It is suggested to locate the corresponding temperature sensors at the inlet of the blower for the air flow to the vehicle (Figure 2).

During the MAC test the vehicle cabin temperature (T_{C3}) or alternatively the vent outlet temperatures (T_{V3}) have to be measured. In the vehicle tests at TUG (chapter 7) both options have been evaluated with similar results. A measurement at the centre of the vents outlet has the advantage, that the positioning of the vents flap does not influence the resulting temperature. The measurement of the temperature in the cabin at T_{C3} has the advantage that the vehicle size and also the vehicle design are integrated into the test procedure. This certainly is also a disadvantage, since the positioning of the vents flaps influences the measured T_{C3} by more then +/-1°C and manufacturers may need additional effort in R&D to optimise the vents towards highest efficiency for the three defined T_{C3} positions.

Since the three vehicles tested in this study are by far not representative for the existing makes and models, we suggest gaining experience with both options by tests at some more cars at labs which can control the test cell accordingly (e.g. ACEA, JRC, TUG, etc.). Then the decision can be supported by a (→ questionnaire to ACEA, JRC and stakeholders?).

Target values for temperature, humidity and mass flow

Cabin temperature:
There are two options which have to be decided before the definition of the final test procedure.

Option 1:
A temperature (T_{C3}) should be measured at the height of the drivers head at three positions (behind the driver, behind the co-driver and in the middle between the front seats). It is proposed to use a target value, namely the maximum of the three measured temperatures (T_{C3}) values should be below 21°C. The target value could be adapted by eventual correction factors for the vehicles glazing (option c in chapter 6.8, which is not recommended).

Option 2:
A temperature (T_{V3}) should be measured at the central position of each vent outlet in the front area of the vehicle. Other vent outlets shall be closed. As target value the maximum of all measured temperature values (T_{V3}) should be below 15°C, adapted by eventual correction factors for the vehicles glazing (option c in chapter 6.8, which is not recommended).

Mass flow

The settings of the mass flow of the air conditioning system shall be adjusted to achieve more than 230 kg/h, adapted by eventual correction factors for the vehicles glazing (option b in chapter 6.8).

Option:
At least for Option 2 above (T_{V3}) a differentiation of the demanded mass flow according to vehicle size may be useful since larger cabins need more cooling capacity to reach the same comfort than small cabins. With the differentiation one MAC system would have higher additional fuel consumption in a larger car than in a smaller car. Since the MAC test procedure will not influence the customers to buy smaller cars, the differentiation would not add incentives for more energy efficiency but would adapt the absolute values closer to the reality.

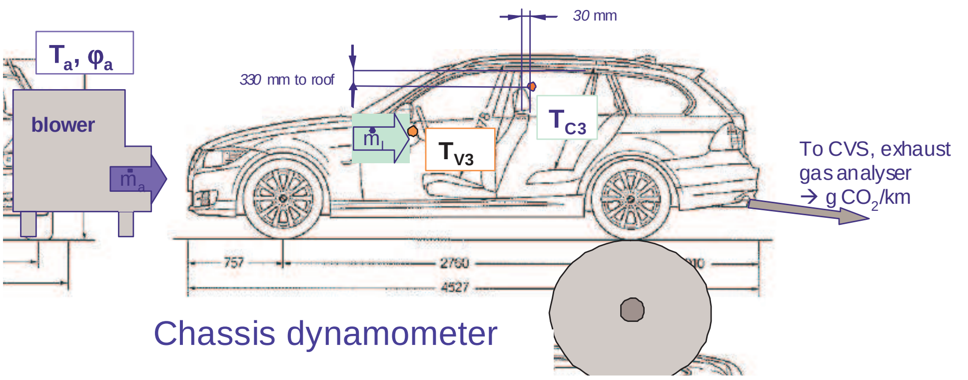


Figure 2: schematic picture of the location of the sensors for the temperature and humidity in the MAC test

The tests showed that all relevant temperatures and also the relative humidity show a variation over the test cycle. This is a normal behaviour of controllers. Since a smaller temperature difference between the test cell and the cabin as well as a lower humidity significantly reduce the energy consumption of the MAC system, either narrow tolerances or more generous tolerances with the application of correction functions are necessary (chapter 6). It is recommended to define tolerances to a state of the art for typical roller test beds and to apply correction factors since the demand of a narrow control would result in very costly test bed equipments.

Suggested tolerances:

- Humidity in the test cell φ_a = 50% +/- 5% (Option > 50%)
- Temperature in the test cell T_a = 25°C +/- X°C (Option T_a >25°C). For T_a a correction factor for vehicle glazing is also optional but not recommended
- Temperature in the vehicle T_{C3} =21°C +/- X°C (Option T_{C3} <21°C)
- Temperature in the vehicle T_{V3} = 15°C +/- X°C (Option T_{V3} <15°C)
- X°C…could be 1.5°C to 2°C and could be part of a questionnaire to organisations performing type approval tests for cars. At TUG +/-1.5°C for the average over the single steady state phases of the MAC test were feasible, ACEA tended towards 2°C. When the correction factors for the COP were applied no significant difference was found in the repeatability for tests with >1.5°C deviation and tests with <1.5°C (see chapter 7).

The given tolerances correspond to the mean values for each velocity step in the MAC step test cycle (chapter 3.3). Additionally it could be defined, that the temperature in the cabin should never be above the test cell temperature during the cycle as long as the MAC is on during the test. Such conditions resulted in worse repeatability at one tested vehicle.

We propose to gain experience at different laboratories which of the options listed above is easier to handle at their labs. At TUG both options proved to be feasible. However, a target value with +/- tolerance was the favoured option there since it needed less experience to find the correct settings

It is also open if each MAC system could manage +/- 2°C tolerance for the cabin temperature. At the moment e.g. no car with a manual MAC system was tested. The feasible tolerances and the correction factors suggested later to compensate for the variability of the temperature reached are thus suggested to be tested and if necessary adapted during a pilot phase of MAC testing-

(→ questionnaire to JRC, TÜV's, DEKRA, ACEA and other stakeholders?)

3.3 MAC test cycle

The proposed test cycle consists of three steady state phases after a preconditioning phase. The preconditioning phase starts with MAC-on to meet the target temperature in the cabin (T_{C3} or T_{V3}). Then the first part is measured with the MAC-on, the second part with MAC-off (Figure 3). With this continuous sequential test the exhaust gas analysers have the same calibration for the tests at MAC-on and MAC-off. This increases the accuracy of the measurements. The constant speeds were selected since they allow a more accurate driving than transient tests and they allow a simple correction for a variation in the speed (chapter 6.1). The speeds selected represent to a certain extend European shares in driving conditions (chapter 3.4).

For the evaluation of the test results only the phases with constant speed are used. The additional fuel consumption due to the MAC is determined by the difference between the MAC-on and the MAC-off phase of the test cycle (see chapter 3.4).

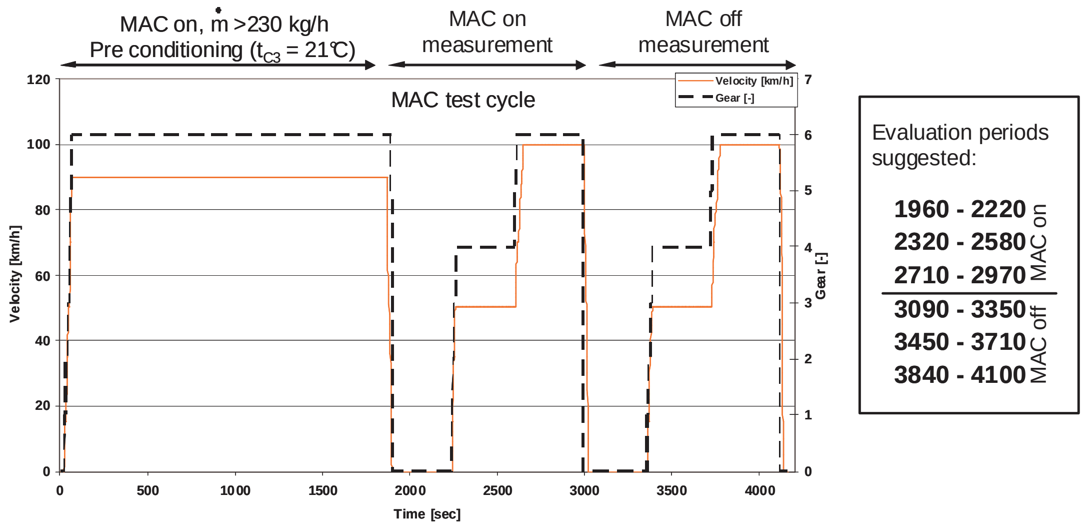


Figure 3: MAC test cycle with preconditioning and two test phases. (option: use 100km/h also for preconditioning to make tampering with battery charging/de-charging more difficult?)

During the measurement regeneration processes of the after treatment systems, non continuous OBD activities influencing the fuel consumption and any other non continuously running activities with influences on the engine work or on the engine combustion process shall be prohibited.
To improve the repeatability of the test results it is recommended to apply 2 or 3 consecutive MAC tests, each of them following the cycle shown in Figure 3. To take the SOC of the battery at the beginning and the end of the test into consideration, the test could be started at different levels of SOC or a correction can be applied based on the measured energy flows from and to the battery assuming average efficiencies for the power generation on board and for charging and discharging the battery.
The option with different levels of SOC at test start would lead to the following sequence after the preconditioning:

– MAC test 1
– Discharge battery to minimum SOC followed by MAC test 2
– Charge battery to maximum SOC followed by MAC test 3

The alternative option with a correction for the SOC needs the measurement of the Ah at the battery. A decision for the option shall be based on (→ questionnaire to stakeholders?)

3.4 MAC test evaluation

The results for the single speed steps of the MAC test cycle shall be weighted according to the shares in real world driving (chapter 5.3.5):

- Idling = 15%
- 50 km/h = 65%
- 100 km/h = 20%

Since the MAC test cycle has in total 6 phases to be evaluated, bag values can not be applied at typical CVS systems and the fuel consumption has to be calculated from the instantaneous signal of the analysers and from the CVS volume flow. The procedure shall basically follow the regulation for the emission tests for vehicle type approval to calculate the mass flows of CO_2, HC and CO. The fuel consumption shall then be calculated from the carbon balance. The concentration of CO_2, HC and CO in the dilution air shall be subtracted according to the average concentration measured for the dilution air in the bag for the entire cycle. The Dilution factor shall be calculated also from the instantaneous signals.

The time windows within the MAC test suggested for the evaluation of the test result are shown in Table 1. Table 1 also shows suggested limits for the maximum variation of the instantaneous CO_2 signal during each time window. An ideal test would show 0% standard deviation over the constant speed phase since prior to each test phase a period for stabilisation of the vehicles running conditions and for compensation of the running time of the exhaust gas from the engine to the analyser and for the response time of the analyser is defined. Disturbances of the ideal conditions lead to variations of the CO_2 signal and thus also to increased standard deviations. The standard deviation thus could be used as measure to exclude specific tests from the evaluation.

Table 1: phases of the MAC test cycle to be used fort he preconditioning (i.e. adjustment of the MAC settings) and for the testing of the MAC

Test part	MAC status	Start [s]	End [s]	Result [kg/h]	max. CO_2 standard deviation (1)
Preconditioning	On, adjust T_{V3}	90	1800	-	Not relevant
Idling	On	1960	2220	$FC_{i\text{-}MAC}$	10% from average
50 km/h	On	2320	2580	$FC_{50\text{-}MAC}$	5% from average
100 km/h	On	2710	2970	$FC_{100\text{-}MAC}$	5% from average
Idling	Off	3090	3350	FC_{i}	10% from average
50 km/h	Off	3450	3710	FC_{50}	5% from average
100 km/h	Iff	3840	4100	FC_{100}	5% from average

(1).... The tests showed no clear influence of the standard deviation of the instantaneous CO2 signal on the test result. Since non standard events, such as a DPF regeneration, will lead to an increase of the standard deviation of the CO2 signal during the constant speed phases, the standard deviation could be used as measure to exclude specific tests from the evaluation. Experience from more tests should be gained before a final decision on this topic is made (setting of the limits and if limits are necessary at all).

With the resulting fuel consumption in the single phases of the MAC test the evaluation for the entire test result is as follows (details see chapter 6):

$$\boxed{FC_{MAC_i} = 3.6 \times C_{COP\,i} \times (C_{Pe_i} \times FC_{i\,Measured-AC-on} - FC_{i\,Measured-AC-off})}$$

With;

i ....speed step i (0 km/h, 50 km/h, 100 km/h)

FC_{MACi}................additional fuel consumption of the MAC system [kg/h] in step i, including all correction factors

$FC_{iMeasured\text{-}AC\text{-}on}$ ...average fuel consumption measured at speed step i in the phase with AC on [g/s]

$FC_{iMeasured\text{-}AC\text{-}on}$ ...average fuel consumption measured at speed step i in the phase with AC off [g/s]

The correction factor C_{Pei} takes variability of the vehicle speed into consideration:

$$C_{Pe\,i} = \frac{P_{B_{AC-On\_Speed\_i}}}{P_{B_{AC-Off\_Speed\_i}}}$$

$P_{B_{AC-On\_Speed\_i}}$ ....average braking power of the rollers in speed step i with MAC-on (0 km/h, 50 km/h, 100 km/h)

$P_{B_{AC-Off\_Speed\_i}}$ ......average braking power of the rollers in speed step i with MAC-off (0 km/h, 50 km/h, 100 km/h)

The braking power of the rollers should be calculated from the measured braking force and the measured speed of the rollers ($P_B = v * F$). Alternatively the power can be calculated according to the driving resistance polynomial used in the set up for the tested vehicle: $P_B = v * (R_0 + R_1 * v + R_2 * v^2)$. The average power is the average of the recorded instantaneous data for the entire evaluation phase of speed step i. Details are shown in chapter 6.1.
The correction factor C_{COPi} takes variability of the CAP into consideration which occurs due to the variability from the temperature in the cabin and in the test cell as well as from the humidity in the test cell:

$$C_{COP\,i} = C_{COP\,i-T1} \times C_{COP\,i-RH} \times C_{COP\,i-TC3}$$

With;

$C_{COP\,i-T1}$ ....Correction factor for variation of test cell temperature T_1 with T_{C3} and RH being exactly at the target values

$C_{COP\,i-RH}$ ...Correction factor for variation of test cell humidity RH with T_{C3} and T_1 being exactly at the target values

$C_{COP\,i-TC3}$ ..Correction factor for variation of cabin temperature T_{C3} with RH and RH being exactly at the target values

It is suggested to calculate the single correction factors for the possible variability in the boundary conditions and to plot them in a simple look-up table as shown in chapter 6.7.

3.5 Classification of the tested MAC

The result obtained for the tested MAC system is the additional fuel consumption in the MAC test cycle in kg/h. This value can be converted into [l/100km] with the weighted test cycle speed and the density of the test fuel. [l/100km] may be a value which can be easier communicated to the customer. However, [kg/h] reflect the real situation better, since the additional energy consumption per hour is on the same level from idling to highway driving while the additional l/100km are increasing strongly towards lower vehicle speed. Since the energy content per kilogram is similar for gasoline and diesel the unit [kg] allows a better comparison between gasoline and diesel car results.

For a classification of the system basically two measures can be used:
1. Absolute value [kg/h] or [l/100km]
2. Relative increase of the fuel consumption with MAC-on [% against MAC-off result)

In 2 the vehicles with a high basic fuel consumption value will achieve better results than in 1. Diesel vehicles have a lower basic fuel consumption, thus the relative increase due to the MAC is rather similar for diesel cars and for gasoline cars while the absolute increase due to the MAC is lower for diesel cars (see chapters 7 and Annex A.

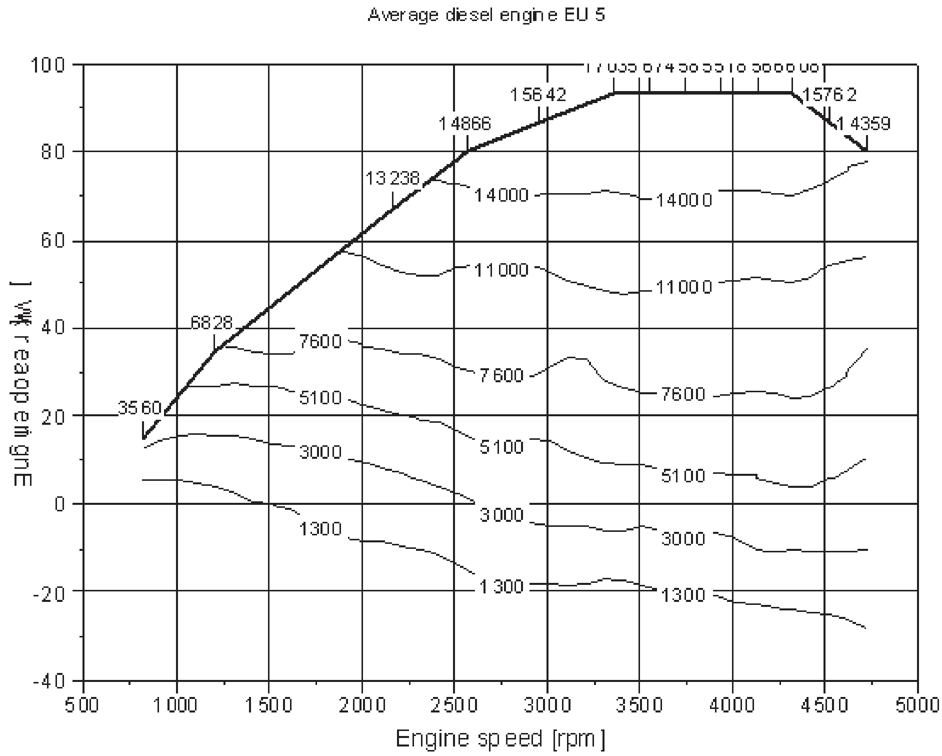
If the absolute values of the additional fuel consumption are used for the classification of the MAC system a different classification scheme for diesel and gasoline cars should be applied. Using the relative increase of the fuel consumption due to the MAC system may simplify the procedure since rather similar results are obtained for diesel and gasoline cars. As a consequence the MAC test may be performed only for the model version with gasoline engine. With different versions of gasoline engine always one – e.g. the one with the lowest specific engine power - could be selected for the MAC test as long as all models are using the same MAC system with similar control algorithms. The glazing thermal properties could be taken into consideration with an fuel consumption addition as function of the size, the design and the thermal properties of the vehicles glazing. Such a method would just evaluate the MAC quality without taking the efficiency of the entire vehicle into consideration. However, more test results should be collected with the final version of the test procedure to be able to base the decision on a reasonable experience. A pilot phase of MAC testing and data collection is urgently recommended also to gain experience with the test procedure itself and potential shortcomings and needs for improvements before the test procedure is finally decided.

4 Proposal for a GSI test procedure

A “Gear Shift Indicator (GSI)” suggests the driver to shift up or down, depending on the actual driving situation. Typically the indicator scheme takes the actual vehicle speed and the actual gas pedal position into consideration to calculate the proper gear position. Besides a low fuel consumption value also the remaining power for acceleration is influencing the strategy for the suggested gear.

Figure 4 shows average fuel consumption maps for EURO 5 passenger car engines from the cars measured for the “Handbook on Emission Factors” (Hausberger, 2010). The maps were obtained from transient tests on the roller test bed using the model PHEM (Passenger car and Heavy duty Emission Model) from TUG, e.g. (Luz, 2009).

The fuel efficiency maps show, that at a given engine power demand [kW], a lower engine speed [rpm] results in lower fuel consumption. On the other hand the available engine power is lower at low engine speeds. Additionally noise and vibration effects limit the range of useful engine speed at very low rpm levels.



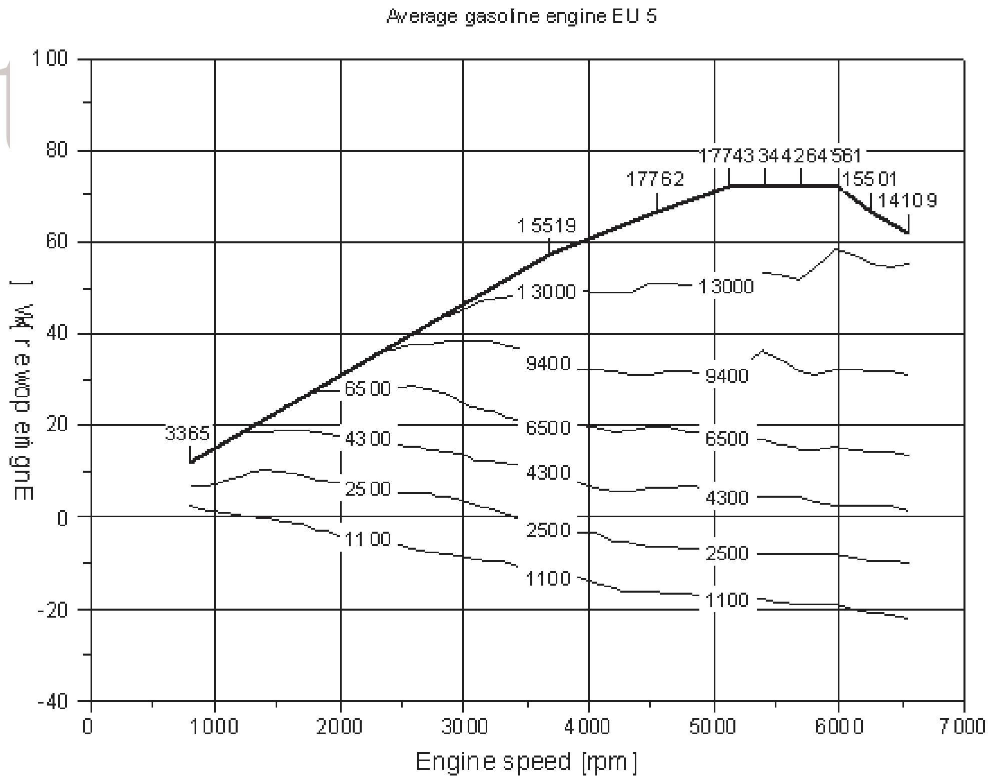


Figure 4: Fuel consumption map in [g/h] for average EURO 5 engines used in the simulation (left picture diesel engine, right picture gasoline engine)

Using the gear shifts indicated by a GSI instead of the defined gear shifts in the NEDC test cycle seemed to be a simple test option. Such a "GSI-test" could be an additional test to classify the GSI as Eco-Innovation. The GSI-test should not be a replacement of the actually prescribed test procedure for the regulated exhaust gas components (HC, CO, NOx, PM and PN) but could be an additional test.
Since the gear shift points can influence the regulated pollutants to a large extent, limiting also the pollutant emissions when following the GSI would be advantageous to achieve low pollutant emissions when a driver follows GSI suggestions.
This consideration is supported by the test results at the diesel EURO 5 car (e.g. Figure 5)[3]. The NOx emissions were considerably higher in the NEDC when following the GSI instead of the original gear shift points. The GSI of the tested vehicle however, did also not reduce the fuel consumption and CO_2 emissions against the basic NEDC.
<u>"Option 1"</u> for a test procedure thus would be:

- Test the vehicle in the NEDC following all rules of the EEC 70/220 with exception of the gear shift points which have to follow the GSI suggestion ("GSI-Test"). The GSI suggested shifting points should be integrated into the driver's aid on the test bed monitor to allow efficient gear shift manoeuvres for the test bed driver.
- Compare the test results with the type approval values of the model. The CO_2 reduction achieved with GSI could be used as value for the CO_2 credit as eco innovation. Alternatively a different reference value could be defined, since the

[3] The tested gasoline cars did not indicate advised gear shifts when driven on the roller test bed. This may be caused by the fact that the rear wheels on the chassis dyno have zero speed. The reasons are not clarified yet.

original NEDC gear shift points are defined already at quite low engine speeds (see discussion below). All pollutant emissions, especially NO_x, should be restricted in this GSI-Test with the same limit values as in the type approval test.

Although the GSI suggestions did not reduce the fuel consumption and CO_2 emissions of the diesel car in the NEDC against the type approval gear shift points. in most of the real world cycles, the GSI lead to lower fuel consumption values compared to the original gear shift strategy (Figure 7).

The reasons for the rather unexpected results for the GSI in the NEDC are:
- The NEDC has already very early gear shift points
- The GSI shifting algorithm is not optimized enough towards a low FC.
- If the driver has to follow the GSI suggestions the speed curve of the test cycle and the time of gear shifting do not fit together. Typically the acceleration is slightly negative in the second of a gear shift. Since a defined speed curve - such as the NEDC - can not take gear shift points of a GSI into consideration, typically the driver has to change the gear in phases of high acceleration. This causes a deviation against the target speed and subsequently a high acceleration demand to meet the target speed again after a gear shift manoeuvre. Such phases can result in rather poor engine efficiencies. To enable the driver to make better gear shift manoeuvres, the gear shift points according to GSI suggestions should be integrated before the test on the screen of the drivers aid on the roller test bed.

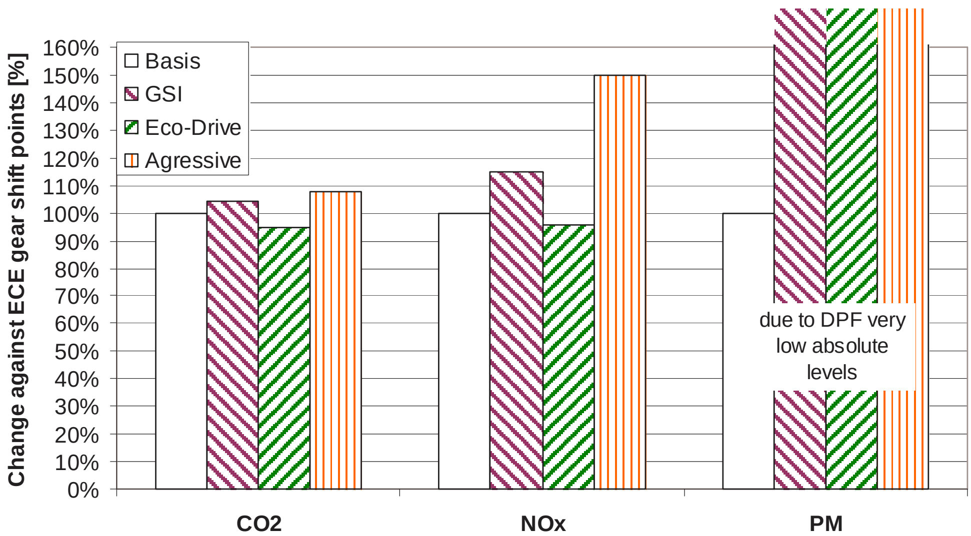


Figure 5: CO_2, NO_x and PM emissions measured in the NEDC cycle with a EURO 5 diesel car following the gear shift points defined in type approval, suggested by the GSI and with gear shifts of an "EcoDriver"

Disadvantages of Option 1 are therefore the already very fuel efficient basic gear shift strategy of the NEDC and certainly the additional costs for the tests on the roller test bed. Further more low pollutant emissions should be obtained with a vehicle not only for a driver following the GSI but also for drivers with a less efficient gear shift style. Thus the control of off-cycle pollutant emissions need a more general test approach and

should not be limited to the GSI test. As a result also a cheap “Option 2” for the GSI testing was elaborated.
The following basic considerations lead to the test procedure suggested later on:

- Since GSI algorithms of modern vehicles can be quite complex and give different gear shift points for different driving cycles, proper test cycles are necessary to fully evaluate the effects of following the GSI suggestions in real world driving.
- If only CO_2 emissions and fuel consumption are relevant for the test, these values could be interpolated from a standardised engine fuel consumption map for a “base case gear shift strategy” and for the GSI strategy without any measurements. This would allow a cheap application of the test for any test cycle[4].
- If the GSI is tuned towards optimum fuel consumption, an increased fuel efficiency should be gained for the majority of drivers which follow the indicated gear shift manoeuvres instead of their usual gear shift behaviour.
- The more the GSI strategy of the car is tuned to allow always good acceleration behaviour of the vehicle in the actual gear – i.e. rather higher engine speed levels - the smaller the part of the drivers, which may benefit from following the GSI suggestions, will be.
- A minimum requirement for the GSI is, that the share of drivers which use an aggressive and less fuel efficient gear shift behaviour – i.e. drive at high engine speeds - should always reach a lower fuel consumption when following the GSI.
- As long as the GSI influences at least the part of drivers using an aggressive gear shift behaviour, a fuel saving effect should be visible in real world driving. Otherwise the GSI shall not be stated as Eco-Innovation.

Figure 6 shows the possible classification in a schematic picture. Shifting the gears due to the GSI at different engine speeds changes the fuel consumption against

Aggressive:	Δ Aggressive
Average:	Δ Av
EcoDriver:	Δ E

Compared to a well trained EcoDriver an increase in the fuel consumption may be tolerable, since a general acceptance of the suggested gears from a GSI could be lower if it results in extreme low engine speeds. The deviation from an “average driver” and/or an aggressive driver could be used as measure to quantify the efficiency of the GSI. Thus, a test cycle should be suitable to represent drivers in the range of average to aggressive driving style.

[4] Pollutant emissions depend very much on the application of emission control strategies which varies between different makes and models. Thus NOx, HC, CO, PM and PN can not be interpolated from a standard engine map. The relative trends in the fuel efficiency over engine speed and engine power however are rather similar for all makes and models if SI and CI engines are separated.

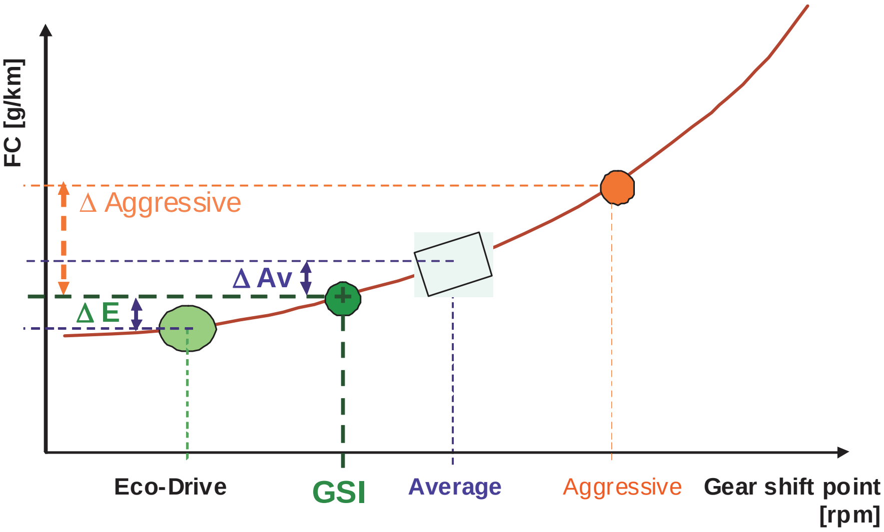


Figure 6: Schematic picture of the effect from GSI on the engine speed and on the specific fuel consumption during driving

To test the applicability of such a procedure, several tests and simulation runs were performed. Two main objectives were followed, namely:

1. to find test cycles which can represent EcoDrive, Average and Aggressive driving. Since the GSI suggestions quite often will depend not only on the engine speed but also on the actual engine load, the speed trace of a test cycle should be in line with the gear shift strategy to be able to correctly classify the GSI suggestions.
2. to test if simple virtual testing of the GSI can show the potential for fuel efficiency improvements

4.1 Test cycles for GSI

Table 2 summarises kinematic parameters and fuel consumption values in the test cycles used for urban, road and motorway driving. The cycles are the UDC and the NEDC, the parts from the “Common ARTEMIS Driving Cycle (CADC)”, the “Integrated Austrian Traffic Situations (IATS) and selected cycles from the “Handbook on Emission Factors (HBEFA)”. The cycles are described in (Zallinger, 2010) and are average speed curves based on hundreds of hour’s recordings in real world traffic situations.

The green marked cells in Table 2 represent speed patterns with anticipatory driving, thus low deceleration levels and a high share of cruising. From all tested cycles these are the closest to an Eco Driving style. Orange marked cells have high deceleration levels and low share of cruise. Cruise is here defined as phases where the speed does not change more than +/-3%.

Table 2: kinematic parameters and fuel consumption in the test cycles

Cycle	Velocity	Engine power	Engine speed	Fuel consumption	a_{pos}	a_{neg}	Idle time	Cruise time
	(km/h)	% from P_rated	n_norm	[g/km]	[m/s²]	[m/s²]	[% time]	[% time]
CADC urban	17.5	2%	12%	80.1	0.65	[illegible]	26%	[illegible]
HBEFA urban	32.0	[illegible]	28%	68.3	0.47	[illegible]	8%	19%
IATS urban	27.0	3%	19%	63.5	0.54	-0.53	14%	24%
UDC	18.8	1%	16%	74.7	0.56	-0.68	31%	30%
CADC road	60.3	7%	33%	49.0	0.41	[illegible]	2%	[illegible]
HBEFA road	80.6	[illegible]	39%	45.4	0.24	[illegible]	0%	[illegible]
IATS road	66.8	8%	34%	44.8	0.30	-0.32	2%	32%
NEDC	33.6	4%	19%	53.8	0.47	-0.72	24%	37%
CADC motorway 130	116.4	24%	60%	56.0	0.23	[illegible]	0%	[illegible]
CADC motorway 150	120.3	26%	63%	57.5	0.24	-0.30	0%	52%
HBEFA motorway	118.7	[illegible]	51%	52.6	0.13	[illegible]	0%	[illegible]
IATS motorway	98.9	19%	41%	49.8	0.24	-0.29	2%	49%
EUDC	62.6	8%	24%	41.6	0.33	-0.86	10%	52%

As a summary of the analysis the CADC represents a rather aggressive driving behaviour compared to the HBEFA, which is closer to Eco Driving. As a result of this analysis following tests were performed on the roller test bed at TUG with standard and adapted gear shifting:

NEDC: Shifting according to GSI, Ecodrive, Standard, Aggressive
CADC: Shifting according to GSI, Standard, Aggressive
HBEFA: Shifting according to GSI, Ecodrive, Standard

The measurement results should show if the speed curve influences the relative position of the fuel consumption reached with gear shifts following the GSI compared to standard, EcoDrive and Aggressive Driving. It can be seen that the gear shift points according to the “Eco-Driving style” achieved approximately 4% to 5% lower CO_2-emissions than the basic gear shift points in all test cycles. The aggressive gear shift strategy lead to 4% to 8% higher fuel consumption values than the basic strategies. The GSI strategy leads to lower CO_2 emissions compared to the basic strategies only in the HBEFA cycles and in the road and motorway parts of the CADC.
Obviously the original gear shift points from the CADC are a more fuel efficient for diesel cars with 6 gears than those from the HBEFA cycles. When the HBEFA gear shift strategy is applied in the CADC the fuel consumption increases by 9% in the CADC 1/3 mix.

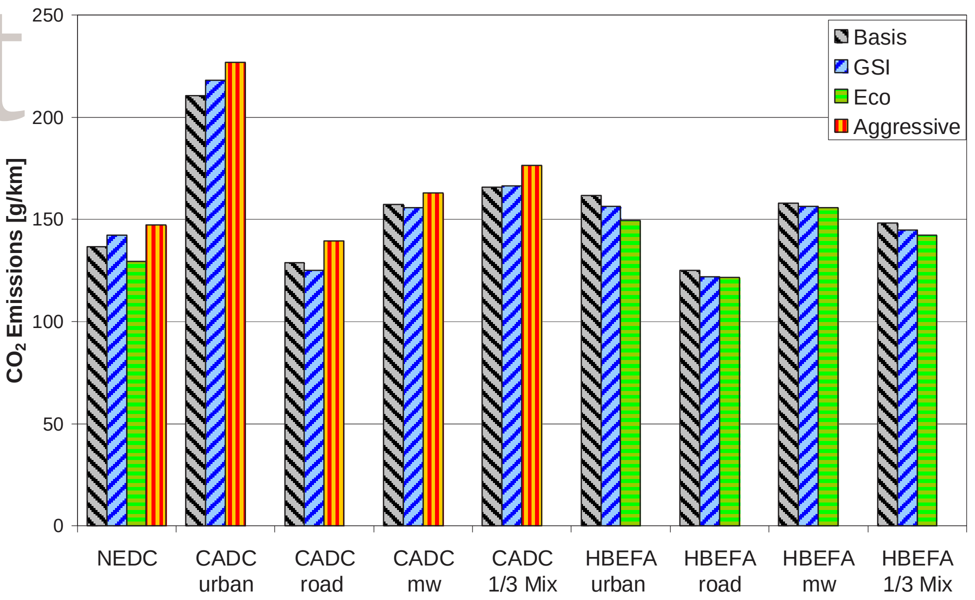


Figure 7: Test results at TUG for CO_2 with different gear shift strategies in different test cycles

A selection of these cycles was measured at KTI too. At LAT the GSI from the tested vehicles did not work on the roller test bed. At the KTI the tests were performed with different settings for the driving resistance values than at TUG. While TUG used the values from a coast down test to depicture the real world data at KTI the default values described in EEC 70/220 for the reference mass of the vehicle were used (see chapter 7.1.4. The settings at KTI resulted in significantly higher driving resistances at higher vehicle speeds than those from TUG. Figure 8 shows the measured CO_2 emissions. Although the absolute levels are higher than those found at TUG due to the higher driving resistances the relative trends between the basic gear shift strategy and the GSI suggestion are similar.

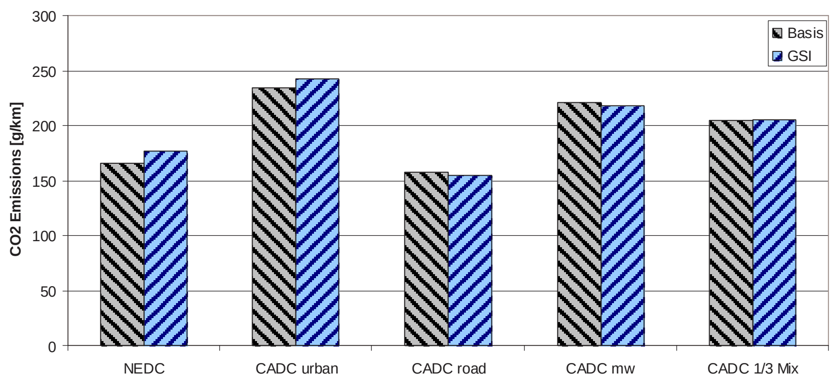


Figure 8: Test results at KTI with different gear shift strategies in different test cycles

From the cycle/vehicle combinations tested it can be recommended to use the CADC with the standard gear shift strategy as reference cycle to evaluate the GSI effect on the fuel consumption[5]. Since the CADC already represents a rather aggressive driving style compared to NEDC, HBEFA and IATS, it is suggested to use the standard gear shift strategy of the CADC as reference. The CADC was developed from real world driving data from the Modem/Hyzem study and German and Swiss data by application of a cluster analysis (Andre, 2001). As a result 13 significantly different types of driving situations were extracted from the data and combined to the CADC. Thus the CADC “envelopes” the real world driving situations, what could be seen as a good attribute for a GSI test cycle if many different real world situations should be covered. The NEDC is also an option but would not cover the GSI behaviour at higher engine loads and covers only a few different acceleration levels. The HBEFA cycle would be more representative for fuel efficient drivers than the CADC. Since a GSI shall rather support drivers with bad gear shift behaviour than drivers with a very economic driving style, the HBEFA may be too smooth for a GSI test cycle. If more effort should be put into the test procedure the development of a special GSI test cycle would be the best option. Such a test cycle should cover a part representing EcoDrive, a part with average driving and a part with aggressive driving (each covering the speed course and the corresponding gear shift behaviour). Due to the lack of measured driving behaviour data which distinguishes between driving styles such a cycle development would need several months.

The emission levels of CO, HC and PM and PN of the tested vehicle were on a very low level due to the modern engine concept and the exhaust gas after treatment (DOC and DPF). No trend in the emission behaviour between standard gear shift points and GSI suggested shifting points was visible for those exhaust gas components. For NO_x however the GSI gear shift strategy lead to reasonable higher emissions than the basic gear shift strategy in almost all test cycles (Figure 9). This shows the high influence of different engine speeds at similar engine loads. At modern diesel engines most likely different EGR level settings combined with different injection timing at different engine speeds are the main reasons for the different NO_x results.
If we assume, that many drivers will follow the GSI suggestions a method to test the emissions also with GSI recommended gear shift points could save pollutant emissions. For the tested EURO 5 diesel car the average change of pollutant emissions when following the GSI suggestions over all test cycles was +16% for NO_x. Compared to the aggressive gear shift style the GSI suggestions save NO_x emissions in the range of 18%. Compared to the standard gear shift points in the test cycle the aggressive gear shift manoeuvres lead to approx. 40% higher NO_x emission levels. As mentioned before, the GSI test seem not to be the best candidate to regulate the off-cycle emissions of vehicles.

[5] It has to be pointed out, that none of the tested cycles was developed to explicitly represent Eco Driving or aggressive driving. More representative cycles however are not available yet. If for the test procedure representative speed traces would be required, a data collection and analysis or a new driving behaviour study will be necessary. Eventually the new test cycle under development for the WHLTP could be used instead of the CADC. Also the gear shift strategies applied to the Eco Driving and to the aggressive driver are not based on a large number of test drivers but simulated with the driver’s gear shift model in PHEM.

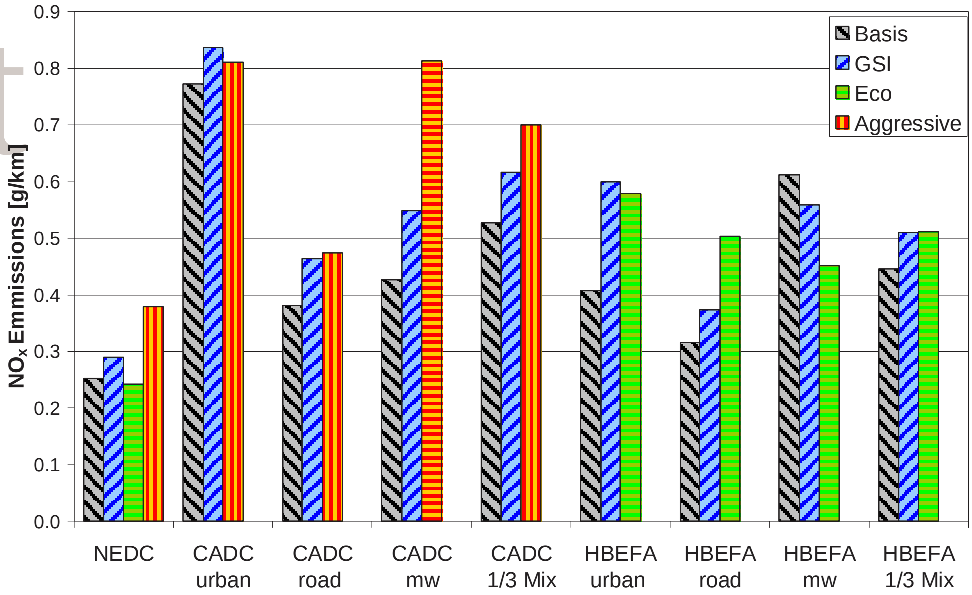


Figure 9: Test results at TUG for NO_x with different gear shift strategies in different test cycles

4.2 Test method for GSI (Option 2)

A virtual test method can be designed independently from the test cycle and is described in this chapter.

The method works as follows:
1) Calculate the actual engine power demand over a driving cycle from the driving resistances of the vehicle and losses in the drive train.
2) Calculate the engine speed by the tire diameter and the transmission ratios of the axis for the engaged gear one time for the original version of gear shift points and one time for the GSI gear shift algorithms. The GSI shift points have to be provided by the manufacturer.
3) With known engine speed and engine power demand the actual fuel consumption can be interpolated from the engine fuel efficiency map. The interpolation shall be done in 1 Hz over the test cycle. The entire fuel consumption is the integral value over the cycle. Existing vehicle longitudinal models are capable of such calculations. For the simulations the routines of the model PHEM (Passenger car and Heavy duty Emission Model) from TUG were used. For a final test procedure further simplifications were done against the model PHEM. The basic functions and formulas can be found in (Luz, 2009). The simplifications are:

a) Driving resistances are simulated from the driving resistance polynomial which is used for the type approval test of the vehicle on the roller test bed and a constant efficiency of the gear box of 98% and a fixed share of 5% of the rotational inertia on the longitudinal inertia:

$$P_e = v \times (R_0 + R_1 \times v + R_2 \times v^2) \times 0.00102 + m \times 1.05 \times a$$

With;
v ....velocity in [m/s]

Pe ....effective engine power demand [kW]
m ....vehicle mass according to type approval value [t]
a............acceleration of the vehicle [m/s²]
R_i..........driving resistance coefficients from coast down tests for the vehicle type approval testing according to EC 692/2008

b) The engine speed is calculated without slip between tires and (rolling-) road surface

$$n = \frac{v}{(D_{tires} \times \pi)} \times I_{axis} \times I_{gear}$$

with: n .... engine speed [1/s]
v .... velocity in [m/s]
Iaxis.......... transmission ratio of the axis [-]
Igear..... transmission ratio of the actual gear [-]
D_{tires}..... diameter of the driven tires [m]

c) No transient effects on the specific fuel efficiency are assumed and the fuel consumption value is simply interpolated from the engine map with the Shepart method:

$$FC_{(P_e,n)} = \frac{\sum_{i=1}^{e} \left(\frac{1}{R_i^{\,2}} \times FC_{i-map} \right)}{\sum_{i=1}^{e} \left(\frac{1}{R_i^{\,2}} \right)}$$

with: e......................number of points stored in the standardised engine map
$FC_{(Pe,n)}$ ....fuel consumption value interpolated from the engine map [g/s]
$FC_{i\text{-}map}$ ....fuel consumption value stored in the map point i [g/s]
R_i ....distance between map point i and point to be interpolated
To be calculated from the normalised coordinates in the engine map (Pe/P_{rated} and n_{norm})
n_{norm} is normalised between idling speed (=0) and rated speed (=1)

The engine emission map is defined as normalised map with

- engine speed normalised as idling = 0 and rated engine speed = 1
- engine power normalised as idling = 0 and rated power = 1
- fuel consumption is normalised as [(g/h)/rated vehicle power] to be useful for each rated engine power after de-normalisation.

Figure 10 shows as example the engine load points in the CADC road cycle for the three different gear shift strategies. Each dot in the map represents the engine load of one second of the test cycle. The “Basis” results from the gear shift strategy defined in the CADC for diesel cars with 6 gears, the “Aggressive” was simulated with the “aggressive driver” from PHEM and the “GSI” results from the gear shifts according to the GSI of the tested EURO 5 diesel car. It can be seen that in the CADC road cycle the GSI has on average lower engine speed levels than defined in the basic CADC cycle and clearly lower engine speeds than the aggressive driver. The engine power demand is independent from the gear shift strategy when using the approach as described before.

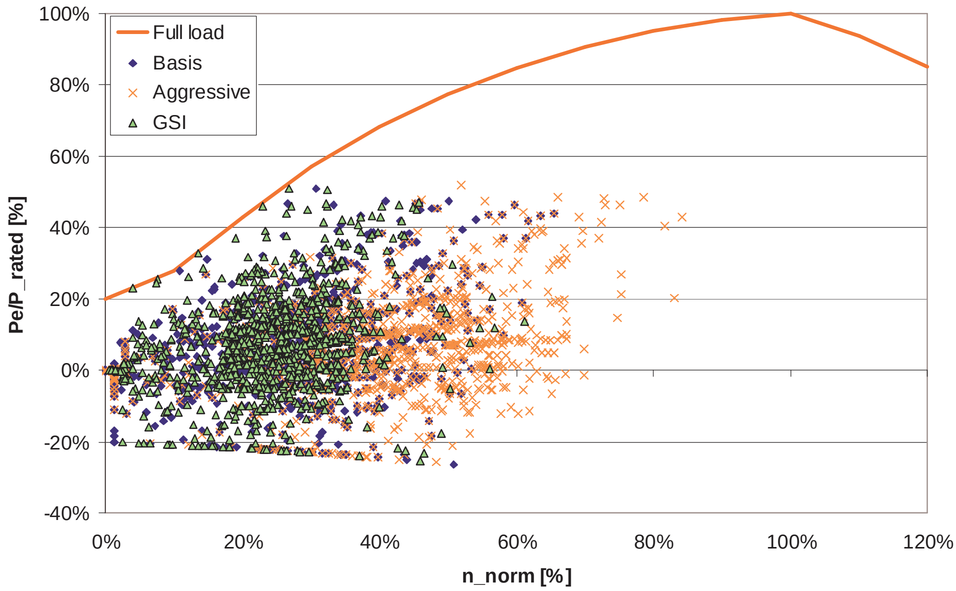


Figure 10: Normalised engine speed and normalised engine power in the engine map for the CADC road with three different gear shift strategies (each dot represents the engine load in one second)

The lower engine speeds of the GSI shifting strategy results also in a lower fuel consumption when interpolated from the average EURO 5 engine map (Figure 11). Integrated over the CADC road cycle the GSI has 7% lower fuel consumption than the average and 25% lower fuel consumption than the aggressive driving style.

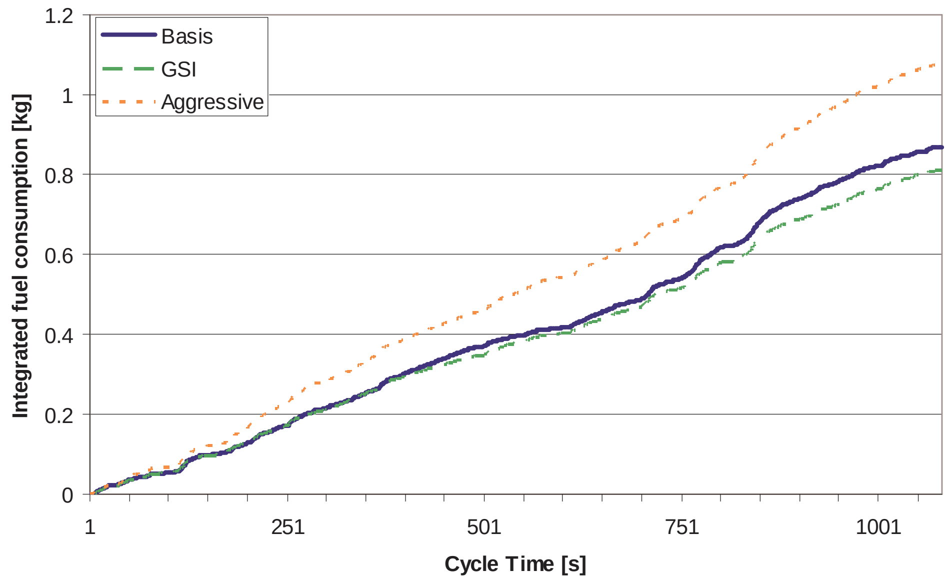


Figure 11: Results from the interpolation in the average diesel Euro 5 engine map for the CADC road with the three different gear shift strategies

For the urban parts of the CADC the GSI suggestions from the tested car did not lead to a reduced fuel consumption compared to the basic strategy. Compared to the aggressive driver a 17% reduction was found. In the motorway part the GSI strategy led to 5% lower fuel consumption against the basic strategy and -17% against the aggressive strategy. For the entire CADC 1/3 mix cycle this results in a 1% fuel saving against the basic gear shift strategy of the CADC and -19% compared to an aggressive gear shift style (Table 3).

Table 3: results for the CADC sub parts and for the CADC 1/3 mix for three different gear shift strategies simulated for the diesel car tested combined with the “average” EURO 5 engine map

	fuel	versus basis	versus aggressive
	[g/km]	[% change]	
Urban (reference)	81.0	0%	-21%
Urban GSI	85.9	6%	-17%
Urban aggressive	103.0	27%	0%
Road	50.3	0%	-19%
Road GSI	46.9	-7%	-25%
Road aggressive	62.2	24%	0%
Motorway (reference)	55.1	0%	-13%
Motorway GSI	52.1	-5%	-17%
Motorway aggressive	63.1	14%	0%
CADC	62.1	0%	-18%
CADC GSI	**61.7**	**-1%**	**-19%**
CADC aggressive	76.1	22%	0%

The results indicate that the simulation is quite accurate when comparing the results with the measured fuel consumption in Figure 5 although the simulation was based on an “average” engine map for diesel EURO 5 cars. The accuracy of such an approach is known from many validation runs, e.g. (Zallinger, 2010), (Hausberger, 2010). For comparing the effects from GSI against a standard gear shift strategy the accuracy of the simulation shall be better than from roller test bed measurements since:

- No driver influence is given (on the test bed the driver has to learn how to follow the GSI).
- No inaccuracies from the measurement equipment and test bed controller
- The effect of gear shifts at inadequate acceleration levels when following a defined speed course with GSI shifts is very low in the simple model while it can have rather big influences on the roller test bed.

The decisions open for a test procedure are:
- Shall the CADC be used or is another test cycle more appropriate?
- Shall the interpolation be based on a standard engine map (differentiated into gasoline and diesel) or shall each vehicle be tested by the map of his own engine

(the latter would be more complicated since regulations would be necessary how to set up the fuel consumption map)?

- Which ratios of the fuel consumption with GSI against average or against aggressive driver are set as target? E.g. “lower than CADC 1/3 mix basic strategy” could be an option, since at least all drivers which show now a worse gear shift behaviour and which are willing to improve would benefit from such a GSI. Certainly also up to e.g. -5% could be set as target.
- How can the GSI gear shift points delivered from the manufacturer for the vehicle be validated? We suggest testing randomly one ore more models per manufacturer and year on the roller test bed and record the engine speed when following the GSI suggestions. For the average engine speed a tolerance level can be defined which is allowed between the measured and the delivered gear shift data for the GSI to be valid.

5 Parameters influencing MAC fuel consumption

In this chapter a “Seasonal Performance (SPF)” simulation tool is used to determine the most influential parameters on the annual fuel consumption that is caused by the MAC system. The results of this chapter were used as input in the development of the test procedure to ensure that the important technical parameters which also have further potential for improvements in their energy efficiency are considered. The test procedure was then designed in a way, that it should give incentives to use advanced MAC technology to a large extent to get a good test result.

Figure 12 shows the basic approach for the SPF analysis performed here. The ambient conditions were investigated for three different cities in Europe. Then an average vehicle with “average” MAC system was simulated in the defined ambient conditions over 12 months in a year. The resulting fuel consumption was then analysed to obtain shares of different parameters on the yearly fuel consumption of the MAC system.

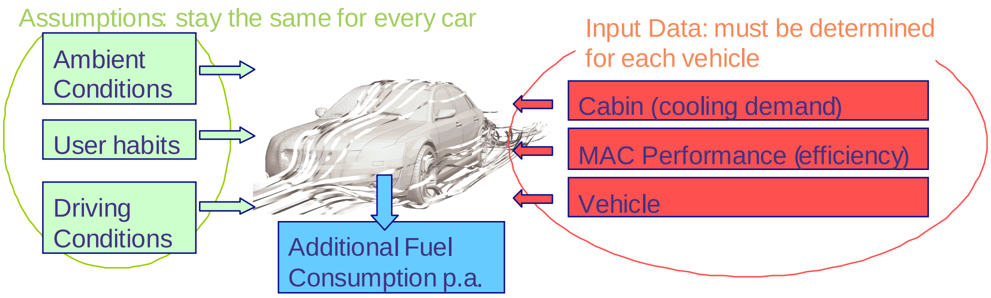


Figure 12: Overview of factors influencing the fuel consumption of the MAC system

5.1 The tool for “Seasonal Performance (SPF)”

The SPF is based on the GREEN-MAC-LCCP (Global Refrigerants Energy & Environmental Mobile Air Conditioning Life Cycle Climate Performance), see GM (2007).

The GREEN-MAC-LCCP provides a holistic approach in estimating all greenhouse gas contributions emitted during the lifetime of a refrigerant and a MAC operating system. For the investigations within this project only the part of “operating the MAC system” (i.e. the fuel consumption) is of interest. Therefore the GREEN-MAC-LCCP was stripped down to a so called “Seasonal Performance (SPF)” simulation.

However, the approaches for modelling the MAC system and the fuel consumption are the same for SPF and GREEN-MAC-LCCP. The basics of the SPF are described briefly in the following paragraphs.

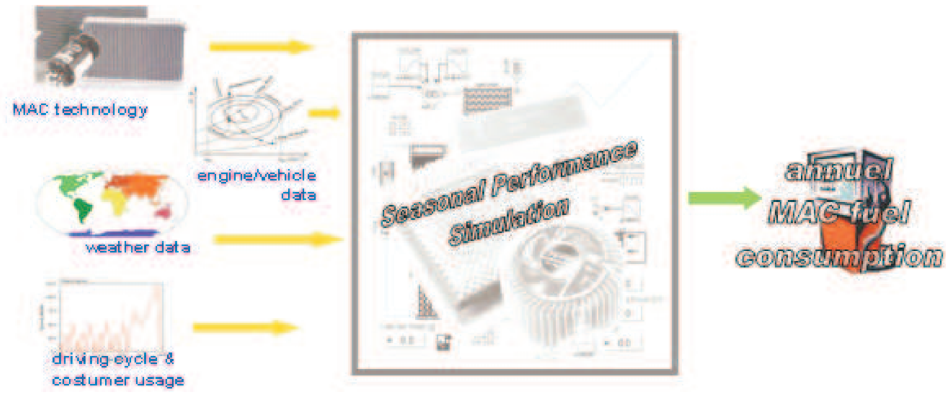


Figure 13: Overview of the Seasonal Performance Simulation model with inputs and output

The overview of the Seasonal Performance Simulation model is shown in Figure 13. The output of the simulation is the annual fuel consumption that is caused by the MAC system. The input values can be divided in the following groups:

MAC technology: This part reflects the capacity and the COP of the refrigerant cycle. Data for Capacity and COP have to be entered as a function of ambient temperature, humidity and compressor speed (see Figure 14). It is foreseen that in idling condition (here 900 rpm) data have to be entered that consider an increased air temperature at the condenser inlet, i.e. the column “900 (+15 K)”. These data represent recirculating air from the engine compartment in standstill of the vehicle.

For the studies within this project the baseline system from Green MAC LCCP was used. This is the enhanced system from SAE ARCRP (2002) with variable displacement compressor, condenser with integrated receiver, thermostatic expansion valve (TXV) and without internal heat exchanger (IHX).

	SetPoint	Humidity	900 (+15K)	900	1500	2500	4000
15℃	10℃	80%					
	3℃	80%					
25℃	10℃	80%					
	3℃	80%	Capacity & COP				
	10℃	50%					
	3℃	50%					
35℃		40%					
45℃		25%					

Figure 14: Input table of SPF and GREEN MAC LCCP for capacity and COP

vehicle data (i.e. engine): In the GREEN MAC LCCP an “incremental engine efficiency” is defined that describes the efficiency of an additional engine load. This incremental engine efficiency is supposed to be independent from engine size and engine speed. This means that the MAC system causes the same additional fuel consumption (absolute) for different engines of same type (in the same vehicle). The incremental engine efficiency is 40 % for gasoline engines and 45 % for diesel engines.
weather data: The modelling of the weather data in SPF differs from the approach in the GREEN MAC LCCP, i.e. the weather data basis Meteonorm (2008) was implemented in the SPF. Therefore the actual weather data at every hour of the year can be considered.

driving cycle & customer usage: The modelling of the customer usage in SPF also slightly differs from GREEN MAC LCCP, since costumer usage models according to Pinkofsky (2006) were implemented. These models consider the volume of traffic depending on daytime, weekday and month. Furthermore this block includes the probability that the MAC system is turned on by the customer. This data is taken from GREEN MAC LCCP.
For the investigations shown in this chapter the New European Driving Cycle (NEDC) was used.

Overall, the Seasonal Performance calculates the MAC fuel consumption for every hour of the year and therefore allows a detailed analysis of the results, e.g. regarding the most important operating conditions of the MAC system.

5.2 Ambient conditions in Europe

The test conditions should reflect the climatic and operational conditions as well as consumer habits typical for Europe. Therefore three typical climates of Europe (hot, mild and cold) have been investigated using the example of Athens, Frankfurt and Helsinki.

Within these investigations the annual fuel consumption of the MAC system has been determined by means of the Seasonal Performance (see chapter 5.1) and the most influential ambient conditions were elaborated. Figure 15 shows the results for the city of Athens. The figures show that the most important temperatures are in the range between 25 and 30°C, e. g. 7.5 % of the annual MAC fuel consumption is caused at a temperature of 30°C and 4 % at a ambient temperature of 20°C (see left figure). In total 36 % of the annual MAC fuel consumption is caused at temperatures between 25 and 30°C whereas 26 % of the annual MAC fuel consumption is caused at temperatures above 30°C (see right figure). Therefore it is supposed to use a temperature of 30°C to reflect the typical conditions for Athens. For complete test conditions the humidity and solar radiation still has to be determined. Considering all the time whenever it is 30°C at Athens this results in an average humidity of 40 % and an average solar radiation of 700 W/m².

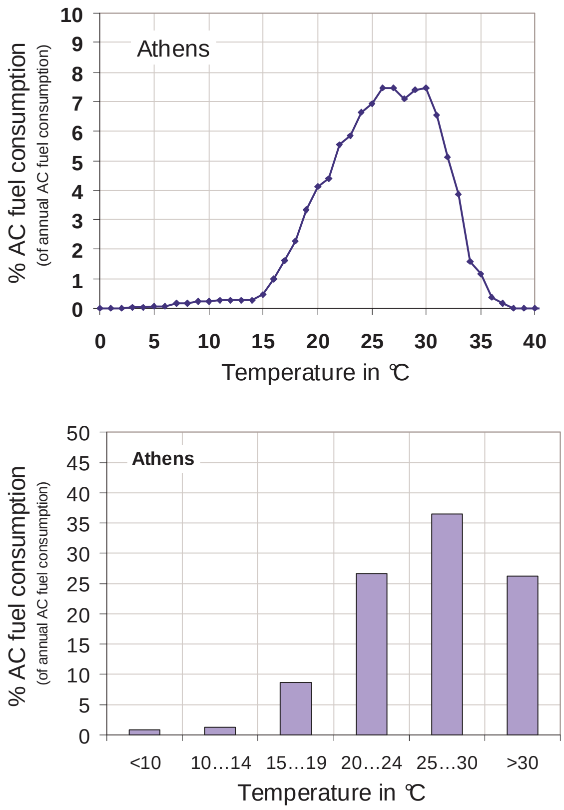


Figure 15: % AC fuel consumption of annual AC fuel consumption depending on ambient temperature for Athens

Figure 16 shows the results of the simulations for Frankfurt. Here the most influential ambient temperature is 21°C where 9 % of the annual fuel consumption is caused (see left figure). The right figure shows that more than 40 % of the annual fuel consumption is caused at temperatures between 20 and 24°C. The average humidity of all conditions whenever the temperature is 21°C is 62 %. However, the suggested condition for Frankfurt is 20°C and 65 % with a solar radiation of 500 W/m².

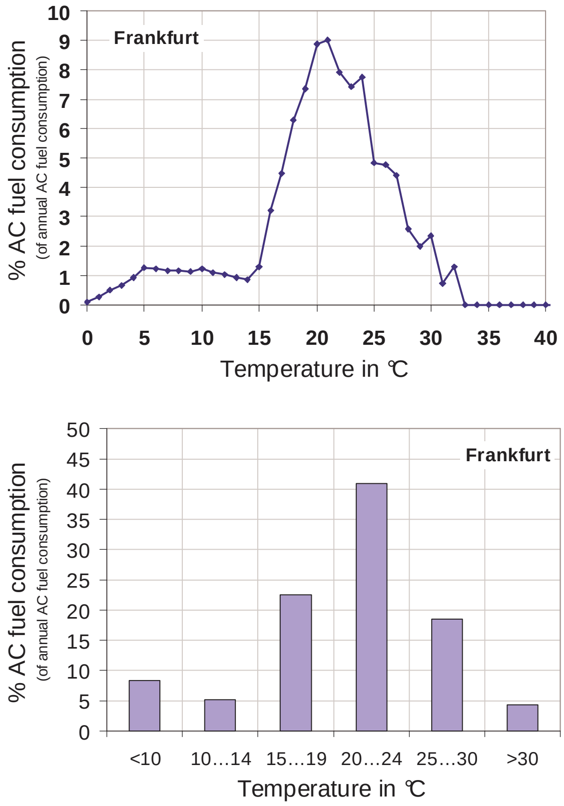


Figure 16: % AC fuel consumption of annual AC fuel consumption depending on ambient temperature for Frankfurt

Figure 17 shows the results of the simulations for Helsinki – the coldest of all investigated climates. It should be pointed out here that the absolute value of the annual fuel consumption differs for each city, i.e. the annual fuel consumption that is caused by the MAC system is much lower in Helsinki than in Athens. In Helsinki the most influential ambient temperature is 17°C where 7 % of the annual fuel consumption is caused (see left figure). The right figure shows that more than 75 % of the annual fuel consumption is caused at temperatures lower than 20°C. Therefore a temperature of 15°C is suggested as typical condition for a cold climate. The average humidity of all conditions whenever the temperature is 15°C in Helsinki is 75 % with a solar radiation of 300 W/m².

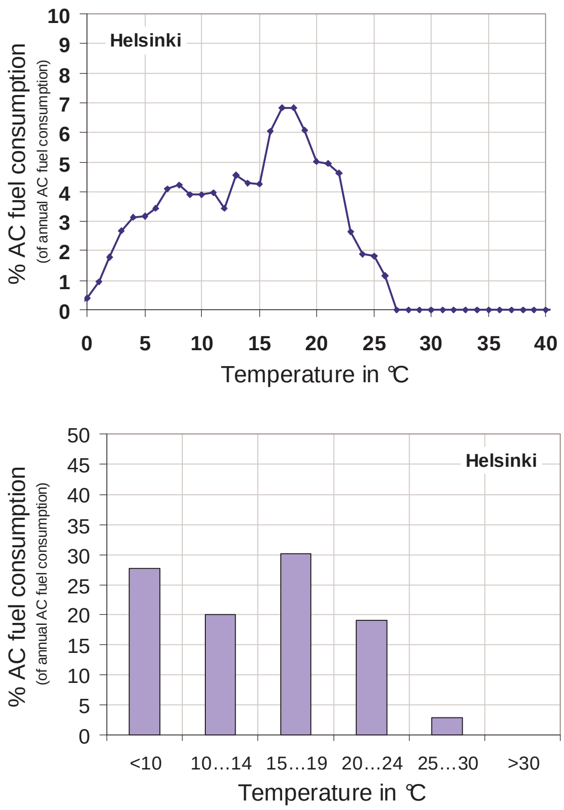


Figure 17: % AC fuel consumption of annual AC fuel consumption depending on ambient temperature for Helsinki

From the results of the simulations the following ambient conditions are suggested to represent “hot”, “mild” and “cold” climatic conditions.

– 30°C, 40 %, 700 W/m²
– 20°C, 65 %, 500 W/m²
– 15°C, 75 %, 300 W/m²

These are the conditions where most of the annual fuel consumption of the MAC system is caused in Athens, Frankfurt and Helsinki. A weighted average around all European countries was not available during this project. However, the test conditions do have more important conditions to fulfill than just representing the European average:

– The “ambient temperature” is depictured by the temperature of the test cell. Thus the temperature should be in the range of typical temperature controllers in chassis dynamometers
– The cabin temperature should be “representative” for typical users. This results in approximately 21°C cabin temperature which corresponds to approximately 15°C temperature at the vent outlet in chassis dynamometer test conditions.
– Both temperatures will show variability during the tests in the range of +/-2°C (depending on the controllers). The cabin temperature should be lower than the ambient temperature to ensure that the AC system is active during the entire test procedure.
– In real world operation the AC systems are running also at ambient temperatures below 21°C. This is for defogging of the air and also to prevent smelly air flows.

– According to (Weilenmann et.al., 2010) “two-thirds of CO_2 and fuel consumption from MAC activity could be saved without discomfort by switching off the MAC below 18 °C. The saving would be 7, 1.5, and 0.75% of overall fuel consumption for urban, rural, and motorway driving, respectively”. It is not clear how an incentive for automatic switch off at low ambient temperatures could look like and which disadvantages such a strategy could have for the users. A simple check during vehicle preconditioning may be sufficient, if the AC is not activated when the vehicle is started at temperatures below approx. 18°C. Such a behavior may be rewarded by a reduction factor for the measured fuel consumption.

5.3 Boundary conditions for a test procedure

The aim of Task 2.2 of the project is to develop a procedure for testing MAC/GSI performance. The objective of the test procedure can be described as follows (Steiniger, 2009):

– Cost-efficient
– Should incentivise reduction of emissions resulting from MAC use in real driving:
  – No “academic” exercise trying to provide the best possible accuracy for environmental impact of MAC use
  – But designed such that technical measures reducing/not reducing “MAC emissions” established by the test procedure reduce/do-not-reduce “MAC emissions” in real driving
– Suitability for “virtual testing”, which may be developed in parallel or at a latter stage (i.e. availability of virtual testing is no condition for future legislation!)
– Assessment of the whole vehicle, including the impact of non-MAC components such as glassing, insulation or packaging of components

In the following paragraphs the results of further Seasonal Performance simulations are described. The aim of the simulations is to estimate the impact of the different parameters on the fuel consumption of the MAC system and thereby identify the most important factors influencing the fuel consumption of the MAC system.

5.3.1 Cabin (Cooling Demand)

The design of the cabin affects the cooling demand of the MAC system. Therefore the cabin should be taken into account within a test procedure for the performance of MAC systems. Concerning the cabin one of the most important factors is the solar radiation. According to Grundstein (2009) the maximum vehicle cabin temperature can reach a value of 58°C at an ambient temperature of 20°C when the car is parked in the sun at a solar irradiation of 800 W/m². Furthermore, Grundstein states that it will take less than three hours to reach this temperature.

However, the influence of solar irradiation is not included in the GM LCCP. Therefore the solar radiation has been implemented in the Seasonal Performace by means of a simple model. This model assumes that the additional load (in W) that is caused by the solar radiation is 110 % of the actual solar radiation (in W/m^2) that is taken from the weather data base.
Figure 18 shows the influence of the solar radiation on the annual MAC fuel consumption (Remark: The fuel consumption is referred to “no solar load” for each city). It can be seen that the fuel consumption of the MAC system in Helsinki will raise by 40 % when solar radiation is taken into account. In Athens the fuel consumption will

increase by approximately 25 %. Certainly, the absolute level of the additional fuel consumption is higher in Athens than in Helsinki due to the higher energy transmitted into the cabin at higher solar radiation. But due to the low basic energy demand of the MAC at the ambient temperatures in Helsinki the additional solar energy results in high relative changes of the MAC energy consumption. The ambient conditions used for this assessment for each city were used as described before.

Barathan et al. (2007), Rugh et al. (2007) and Rugh (2009) from NREL did investigate the potential for reduction of thermal loads by means of different technologies. They came to the following conclusions:

- With a solar transmission of 50% the total thermal load can be reduced by approx. 20%.
- Changed inclination of the windshield has only slight positive or negative effects.
- With insulation and solar reflective painting only small improvements can be achieved.
- Ventilation has positive effects especially when the hot air is sucked out, but requires additional air outlets.

To investigate the potential of a load reduction concerning the annual fuel consumption some simulation with reduced solar load were conducted, i. e. it was assumed that the additional load (in W) that is caused by the solar is 80 % (instead of 110 %) of the actual solar radiation. The results of theses simulations are also shown in Figure 18. The results show that the annual fuel consumption of the MAC system can be reduced by approximately 5 to 15 % if solar load is reduced by 30 %.

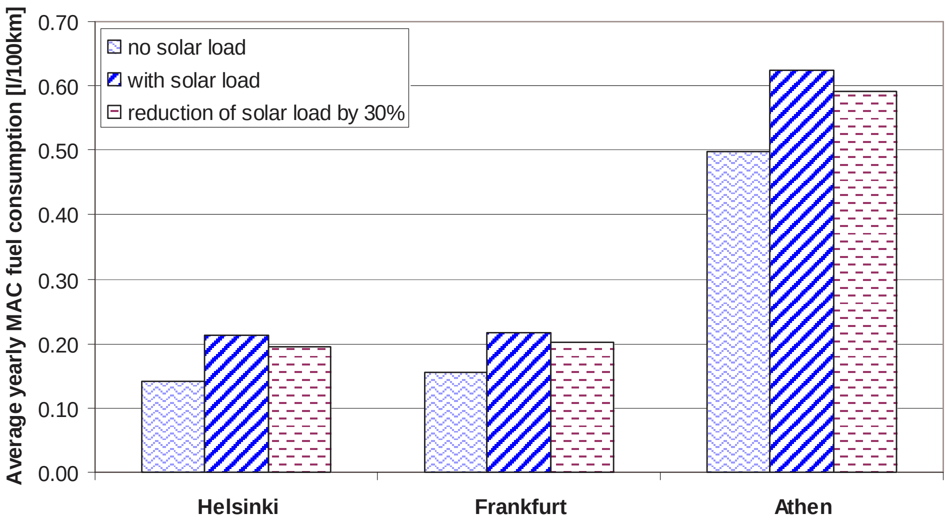


Figure 18: Influence of solar radiation on annual fuel consumption of MAC simulated with the SPF model

Due to the reasonable influence of the glazing on the demanded CAP it is recommended to take the solar radiation into account within the test procedure. The solar load could be depictured by increased mass flow of fresh air over the MAC system or by a higher test cell temperature or by additions to the fuel consumption of the MAC as function of the size and thermal properties of the glazing. Using solar lamps would be very costly and the application of an electrical heater showed a rather worse repeatability (see chapter 7). Furthermore such an approach would need a “standardized heater” with a “standardized blowing direction and volume flow” in a “standardized position” in the vehicle. Since automatic MAC systems do have temperature sensors in the vehicle cabin and the location of these sensor(s) can vary between makes and models it may be hard to find a commonly agreed design of such a heater. Furthermore advanced MAC systems also use solar sensors which adapt the temperature at the vents of the MAC to the solar radiation to improve the comfort feeling for the passengers. With solar radiation the passengers need a lower temperature to have the same comfort feeling (S. Hessel / F. Manz, 2009).

5.3.2 Efficiency of Refrigerant Cycle (COP)

The cooling capacity that is necessary to cool down the cabin is produced by the refrigerant cycle. The required driving power of the compressor of the refrigerant cycle is affected by the Coefficient of Performance (COP) of the refrigerant cycle. The COP is defined as benefit (cooling capacity) divided by the effort (compressor driving power). An improvement of 10 % in the COP directly leads to a reduction of 10 % of the fuel consumption of the compressor. The energy consumption from the blower and from the fan of the MAC system are not affected by the COP value.
Hill (2006) states that a COP-improvement of approximately 30 % of the baseline system (the same one as used in this study, compare chapter 5.1) seems to be realistic, especially in part load conditions of the refrigerant cycle. He gives the following measures to improve the COP:

- Compressor:
  - Improved piston compressors,
  - Alternative technologies
- Heat Exchangers:
  - Improved Effectiveness evaporators and condensers,
  - Internal heat exchanger
- Controls:
  - Optimized superheat controls
  - Optimized Sub-cooling controls
  - Optimized compressor control
  - Flash gas removal
- Others:
- Optimized plumbing
- Control of re-circulation

However, all of these measures are covered with the description of the COP and there is no need to consider these measures separately if the COP is considered within the test procedure.
Therefore SPF simulations were conducted with a COP of the refrigerant cycle that is improved by 15 % at full load conditions and by 30 % at part load conditions. The results of these simulations are shown in Figure 19. It can be seen that the annual fuel consumption would be reduced by approximately 20 % (Figure 19).

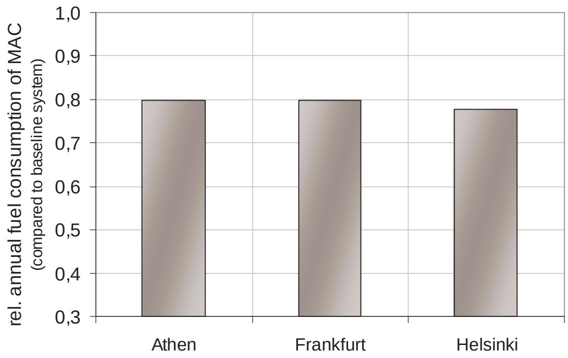


Figure 19: Reduction of annual MAC fuel consumption if COP is improved by 15 % at full load conditions and by 30 % at part load conditions

The final conclusion of this paragraph is that the refrigerant cycle technology still shows room for improvement even if the baseline system is already a so call “enhanced” system (Hill, 2006). Therefore the efficiency of the refrigerant cycle (COP) is one of the key facts that have to be considered within test procedure.

5.3.3 *Engine*

As discussed in chapter 5.1 the additional load of the engine (i.e. compressor load) is described by means of“incremental engine efficiency”. This incremental engine efficiency is supposed to be independent from engine size and engine speed. This means that the MAC system causes the same additional fuel consumption (absolute) for different engines of same type (in the same vehicle). In Green MAC LCCP several OEMs are cited as reference. This assumption certainly is a simplification. However, the courses of the average engine power and engine speed in Europe combined with the course of the energy demand from the MAC system are not available. Thus a detailed simulation with engine fuel efficiency maps would not add accuracy to the assessment of the average fuel consumption behaviour of MAC systems in Europe.

In Green MAC LCCP The incremental engine efficiency was set to 40 % for gasoline engines and 45 % for diesel engines. Therefore the additional fuel consumption that was calculated for the MAC systems is 8 % lower for a vehicle with a diesel engine than for a vehicle with a gasoline engine. This value was tested also with the roller tests at TUG and LAT. In this comparison the gasoline vehicle had a lower additional fuel consumption than the diesel vehicle (see chapter 7.3). However, the test conditions at LAT were out of the defined boundary limits for the MAC test procedure and the gasoline car was a sedan while the diesel car was the estate version.
The tests at PSA showed higher additional fuel consumption for the gasoline cars compared to the diesel cars (see annex). A general conclusion is, that the additional fuel consumption should be evaluated in kg instead of litres since the energy content per kg is roughly similar for gasoline and diesel while the energy content per litre is more than 10% lower for gasoline than for diesel. Alternatively the evaluation can be performed separately for gasoline and diesel cars.

5.3.4 *Vehicle Data*

Besides the cooling demand (see chapter 5.3.1) and the efficiency of the refrigerant cycle (see chapter 5.3.2) the fuel consumption of the MAC system also depends on the design of the vehicle. One important design factor is the construction of the front end module (cooling package). Often the design allows that hot air from the passenger compartment or from the engine recirculates in standstill of the car and therefore causes an increases air inlet temperature of the condenser which results in higher cooling demand and a lower COP of the refrigerant cycle. This phenomenon is also considered within the Seasonal Performance simulation (compare chapter 5.1, Figure 14). To investigate the influence of this phenomenon it was assumed that no COP reduction by recirculating air occurs, i.e. the COP in column 1 and 2 of Figure 14 are the same. Figure 20 shows the reduction of the annual MAC fuel consumption if recirculating air in standstill of the vehicle can be avoided. This measure shows a potential of 10 to 13 % for the investigated cities. This means that the recirculating has a rather great influence on the annual fuel consumption of the MAC. One reason for this is that the NEDC (which was used within these investigations, see explanation if simulation model in chapter 5.1) has a rather great portion of idling conditions (see chapter 5.3.5).

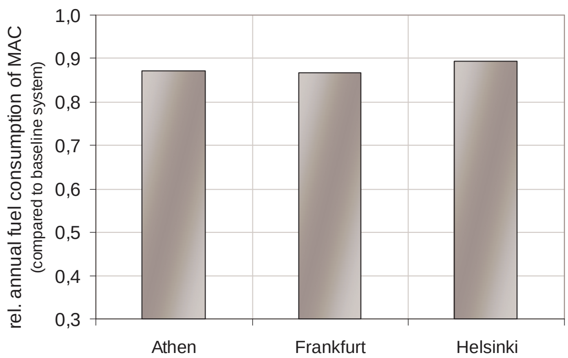


Figure 20: Reduction of annual MAC fuel consumption if recirculating air in standstill of the vehicle can be avoided

According to these results it is recommended to consider standstill conditions within the MAC testing procedure – if applicable. However, the relevant wind speed of the blower at the test cell has to be considered to obtain a defined condition for the ambient air flow in all test cells.

5.3.5 *Driving Conditions*

The efficiency of the compressor of the MAC system depends on the rotational speed. Typically at high speeds the efficiency drops. To take these effects into consideration variable engine speeds (different vehicle speed levels) should be tested. To end up with a reasonable test cycle duration three speed steps were selected.

The JRC/WHLTP EU Data based on driving behaviour data on 213 900 km in Europe (status with data from CH, Be, Fr, De, Slo, It) leads to the speed distribution shown in Figure 21.

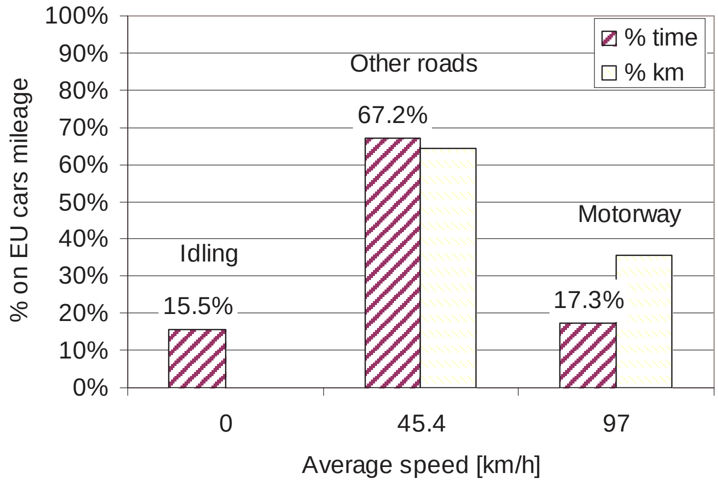


Figure 21: Shares and average speeds for idling, for driving on motorways and for driving on other roads

An important topic is the selection of the gears for these speeds. A lower gear leads to a higher rotational speed of the compressor of the MAC system and thus to a higher additional fuel consumption.

To reward vehicles with a 6th gear it is suggested to define the highest gear to be used at 100km/h. The 50km/h could be driven typically in the 3rd or in the 4th gear. Using the 3rd gear would lead to a higher fuel consumption. The 4th gear will on average cover the engine speed range between idling and 100km/h in the highest gear. 50km/h in the 3rd gear would extend the covered rpm on average to slightly higher values (Figure 22).

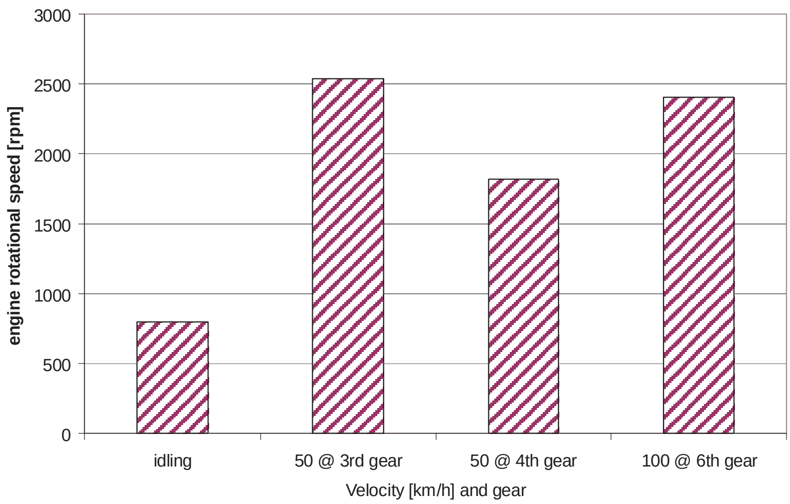


Figure 22: Typical engine speeds for a medium gasoline car with 6 gears in the MAC test cycle

5.3.6 Preconditioning and soak

From today's point of view it seems to be obvious to use the NEDC as preconditioning cycle as defined in the EC regulation for the type approval tests of passenger cars and light duty vehicles. The preconditioning is necessary to bring the vehicles in a defined status before the test starts. Important is the status of the exhaust gas after treatment devices. Especially the pressure drop over the particle filter (DPF), which depends on previous driving conditions and on eventually occurring DPF regeneration phases, and the condition of the lean NO_x catalyst, which may cause a desulphurisation process, are important. A variability in the basic fuel consumption of the vehicles due to other influences than the MAC system should be kept as low as possible. Thus a complete regeneration of the existing after treatment systems (if relevant) could be included in the preconditioning cycle. The soak time is necessary to bring the vehicle on a repeatable temperature, i.e. the test cell temperature.

6 Calculation of correction factors

Since a tolerance for the vehicle speed, the temperatures and for the humidity has to be allowed to run MAC tests on existing chassis dynamometers, variations and drifts against the target values will influence the fuel consumption from the MAC system. Following tolerances are suggested (see also chapter 3 for the definition of “X”)

- Velocity (+/- 2 km/h)
- Humidity in the test cell φ_a = 50% +/- 5% (Option > 50%)
- Temperature in the test cell T_a = 25°C +/- X°C (Option T_a >25°C). For T_a a correction factor for vehicle glazing is also optional but not recommended
- Temperature in the vehicle T_{C3} =21°C +/- X°C (Option T_{C3} <21°C)
- Temperature in the vehicle T_{V3} = 15°C +/- X°C (Option T_{V3} <15°C)

For variations of the temperatures and of the humidity correction functions hare shown in the chapters 6.1 to 6.7.
The glazing of a vehicle has a reasonable influence on the cooling demand (CAP) of the vehicle (e.g. chapter 5.1). Options to reduce the heat entrance due to sun radiation are smaller window surfaces, steeper angles of the windows and better glazing quality (mainly transmission and reflection factors for direct sun and thickness of the window). Thus it is recommended to take the relevant glazing parameters into consideration.
Since a heat blower in the vehicle cabin proved in the practical tests to be not a reliable tool to depicture additional heat from sun radiation, a correction value for the fuel consumption or of the mass flow from the AC system is recommended (chapter 6.8).

6.1 Correction for variations in vehicle speed

Variations in the vehicle speed directly affect the braking force of the rollers and thus the positive engine work over the cycle. If the additional fuel consumption due to the MAC system is calculated by the difference of the fuel consumption measured with AC-on and with AC-off, both tests have to deliver the same engine work. Otherwise a part of the difference in fuel consumption would be due to a different engine work.
The braking force F_B in [N] of a chassis dynamometer typically is simulated as function of the vehicle speed in [m/s] as follows:

$$F_B = R_0 + R_1 * v + R_2 * v^2$$

The actual braking power in [W] is

$$P_B = F_B * v$$

To control the actual braking force typically the torque is measured at the electric motor which brakes the rollers. From the torque signal the braking force and with the measured velocity of the rollers also the barking power can be calculated. For the correction function this measured values should be used.
For the correction of variations in the average braking power it is suggested to correct each constant speed phase of the 3-Step test with AC-on against the average braking power with AC-off:

$$\boxed{C_{Pe} = \frac{P_{B_{AC-On\_Speed\_i}}}{P_{B_{AC-Off\_Speed\_i}}}}$$

The fuel consumption measured in the speed phase i with AC on is then:

$$FC_{AC-on_i\,Pe-corrected} = C_{Pe_i} \times FC_{AC-on_i\,Measured}$$

This correction assumes constant engine efficiency, what is a reasonable simplification in the small speed tolerances allowed. This correction is suggested independently of further corrections for the ambient conditions. Further corrections are then based on the P_e-corrected fuel consumption.

6.2 Calculation of the CAP

As described before important coefficients for the additional fuel consumption due to the MAC system are the:

- CAP refrigerating capacity (i.e. kW cooling capacity)
- COP Coefficient of performance (refrigerating capacity / power input to the compressor)
- Mass flow of air

If the actual boundary conditions during a test are different to the target settings defined in the type approval procedure (e.g. higher test cell temperature), the results of the test can be corrected by the ratio of a calculated energy consumption from the AC-system at the target settings to a calculated energy consumption from the AC-system at the measured settings.

To calculate such correction factors a simplified AC model was developed which depictures the behaviour of AC systems in the given tolerances for temperatures and humidity reasonable well. The model simulates CAP, COP and additional energy demand from electric consumers (blower, fan and clutch, if existing). The simulation of the CAP is described in this chapter.

The CAP is influenced especially by the ambient conditions, by additional heat loads in the vehicle (excess heat from the engine, solar radiation,…), by the settings of the MAC system, i.e. the mass flow of ambient air over the evaporator of the MAC system and by the temperature at the evaporator. The share of air recycled from the cabin is relevant in the schematic system shown in Figure 23, since it influences the mass of fresh air at a given total mass flow.

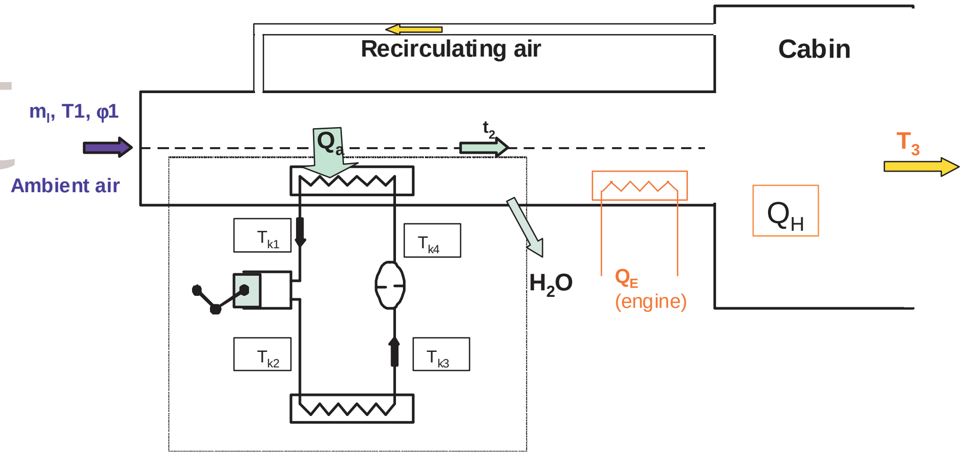


Figure 23: Schematic picture of the relevant parts of the MAC system

To calculate the CAP (Q_a) a heat balance can be made over the entire vehicle:

$$Q_a = m_l * [h_{3(1+x)} - h_{1(1+x)} + (x_1 - x_2) * C_w * t_w] + Q_E + Q_H$$

blower position — Entahlpy (3 -1) — Condensation — heating — „sun radiation“

$\dot{m}_l$ — T1, φ1, Ti, φi — T1, T2, φ1

T_{k4}…setting of MAC evaporator temperature

The values for $h_{(1+x)}$ result from the temperature and the relative humidity of the humid air:

$$h_{i\,(1+x)} = c_{p_{\text{dry air}}} \times t_i + x_i \times (r_0 + c_{p_{\text{steam}}} \times t_i)$$

$$x_i = \frac{R_{\text{dry air}} \times p_{\text{steam}}}{R_{\text{steam}} \times p_{\text{dry air}}}$$

The partial pressure of steam and dry air is gained from the relative humidity and from the total pressure of the humid ambient air.

$$p_{\text{steam}_i} = \varphi_i \times p'_{\text{steam } i}$$ with p'_{steam} as the saturation pressure.

Table 4 summarises the data used in the simulation.

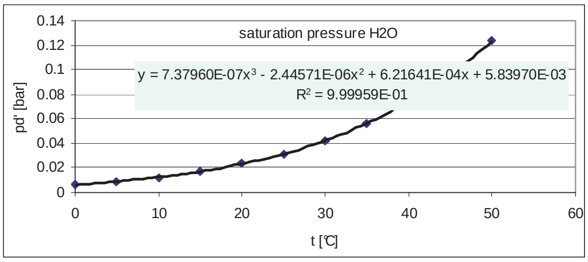
Table 4: Data used for the moist air from the test cell to the vehicle cabin

Parameter	Value	Unit	Comment
R_{steam}	461.5	J/kgK	
$R_{dry\text{-}air}$	287	J/kgK	
$cp_{dry\text{-}air}$	1	kJ/kgK	
Cp_{steam}	1.85	kJ/kgK	
c_{water}	4.19	kJ/kgK	
r_0	2500	kJ/kg	
Q_H	0.2	kW	Heat load from the driver and from heat transfer from the engine and from the test cell into the cabin
% recirc. Air	0%	m_{rec}/m_{tot}	Actual MAC systems do rather not use

			recirculated air at ambient temperatures of 25℃. Thus 0% is suggested (1).

(1).. If at a test a higher test cell temperature is used in combination with recirculated air, the correction factor for 0% would lead to too much reduction of the measured MAC fuel consumption value and vice versa. The alternative would be to calculate extra correction factors for different shares of recirculated air but this option would make the procedure much more complex since also the share of recirculated air would have to be measured, what is a very difficult task. With 0% recirculated air at least only MAC systems with recirculating air can benefit from the settings of the correction. The application of recirculating air is assumed to reduce the energy consumption in real world operation.

Since look up tables make a handling in Excel rather slow the condensation curve for water is approximated by a polynomial function (Figure 24).



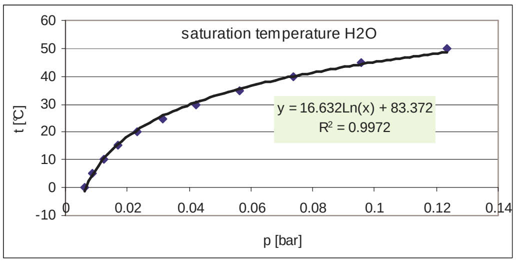


Figure 24: polynomial functions as simplified condensation curve for water

The energy from excess heat of the engine (Q_E in Figure 23) is important for the entire energy balance but can hardly be measured in the type approval. Thus it is suggested to omit Q_E in the simulation. If Q_E does not change significantly in the tolerances allowed for the boundary conditions it still influences the ratio of the Q_a-target/Q_a-measured. For this reason a heat transfer from engine and ambient of approx. 100W is included in Q_H. For the vehicles tested so far, this approach lead to reasonable results.

6.3 Calculation of the COP

The COP can be measured on an AC test stand or calculated from the air conditioning (AC) cycle. Figure 25 shows a simplified AC cycle. The CAP (Q_a) is entering the AC system during the evaporation of the coolant from 4 to 1. Then the coolant enters the compressor (1 to 2). After the compressor a higher temperature and pressure level is reached. With condensation from 2 to 3 the heat is released to the ambient. To close the cycle the coolant is then sucked through a throttle from 3 to 4.

Figure 25: Schematic picture of the air conditioning cycle system

In the following the simulation of the simplified AC cycle is described.
If the temperature levels and the efficiencies of the compressor are known, the COP can be calculated for this simplified cycle:

$$\mathrm{COP} = \frac{\dot{\mathrm{Q}}_{\mathrm{a}}}{\mathrm{P}_{\mathrm{ce}}} = \frac{h_4 - h_1}{(\mathrm{h}_2 - h_1)/(\eta_m)}$$

The Enthalpy h_2 can be assessed by the isentropic standard cycle (s = constant from over the compressor, i.e. point 1 to 2 in the AC cycle).

$$\mathrm{h}_2 = \frac{h_{2s}}{\eta_{s-i}}$$

The power demand from the AC compressor can be estimated as:

$$\mathrm{P}_{\mathrm{ce}} = \dot{\mathrm{m}}_{\mathrm{R134a}} \times (\mathrm{h}_2 - \mathrm{h}_1)$$

The mass flow of the coolant can be calculated from the CAP demand:

$$\dot{\mathrm{m}}_{\mathrm{R134a}} = \frac{\dot{\mathrm{Q}}_{\mathrm{a}}}{(\mathrm{h}_4 - \mathrm{h}_1)}$$

The real AC cycles are more sophisticated, using e.g. under cooling at point 3 of Figure 25 and overheating at point 1.

Again functions instead of look up tables are used in the Excel file. The AC cycle is calculated for the properties of R 134a (Figure 26).

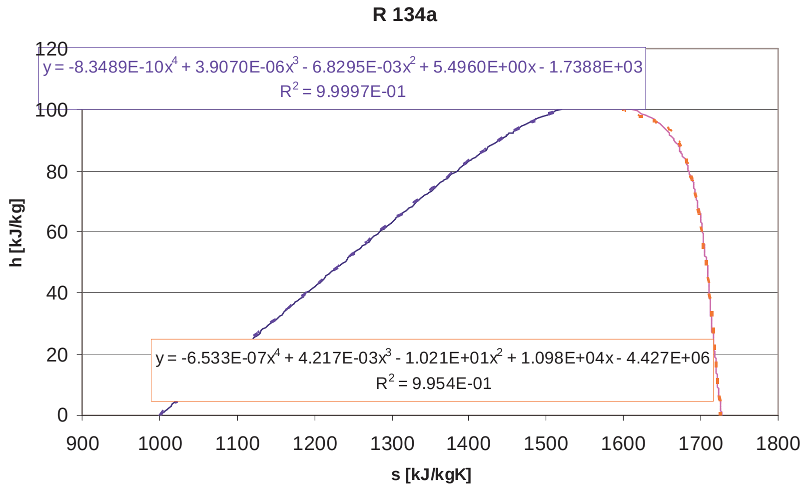


Figure 26: polynomial functions as simplified limiting curve for the coolant

The values shown in Table 5 were used for the parameterisation of the simplified COP simulation. The values were calibrated with the test results of a EURO 5 diesel car from the chassis dynamometer.

Table 5: data used in the simulation of the AC cycle

Parameter	Value	Unit	Remarks
c_p in superheated steam (1→2s)	1.1	[kJ/kgK]	Simplification to avoid look up table
Eta_s-i	0.68	-	Variable over rotational speed (is constant value sufficient for correction factors?)
Eta_m	0.85	-	Variable over rotational speed (is constant value sufficient for correction factors?)
tk3 - t1	25	°C	Inclination of temperature in condenser against ambient
t2 - tk1	12	°C	Inclination of temperature in evaporator against t_2
Q_H	0.2	kW	Heat load from the driver and from heat transfer from the engine and from the test cell into the cabin

6.4 Calculation of the electric energy consumption

The data of electrical energy demand in Figure 27 was assessed from the measured AC fuel consumption data at the BMW 318d with different blower settings (the values should be validated and supplemented by BMW and also by other manufacturers).

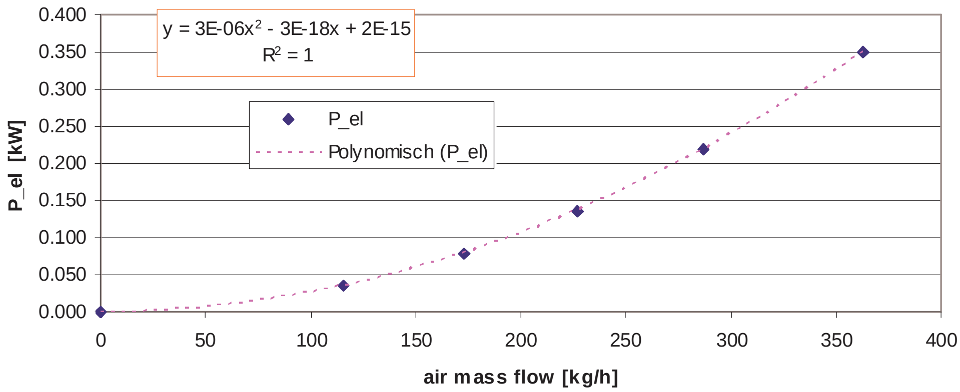


Figure 27: Consumption of electric energy from the blower and from the fan of MAC system as function of the air mass flow

6.5 Simulation of the additional fuel consumption

To obtain reasonable correction factors for small variations in the ambient conditions and in the cabin temperature against the target settings, the energy consumption of the compressor as well as of the electric system for the blower and other components should be considered.

The energy consumption of the AC compressor is directly influenced by variations of temperatures and humidity while the energy consumption of the blower is depending on the blower stage selected in the AC settings. The small changes in the density of the air at changing ambient conditions will have only minor influence on the power demand of the blower.

The electric energy has to be produced by the engine driving the generator. The compressor is assumed to be driven directly be the engine.
When the power demand of the compressor (P_{ce}) and the electric power demand (P_el) have been calculated for the corresponding boundary conditions the additional fuel consumption FC_{AC} is:

$$FC_{AC} = be \times \left(P_{ce} + \frac{P\_el}{\eta_{el}} \right)$$

With FC_{AC}.......additional fuel consumption due to the AC system on [g/h]
The incremental fuel consumption was already discussed in chapter 5.3.4. It takes the additional fuel consumption due to an additional load to the engine into consideration. A more sophisticated model could make use of engine emission maps like shown as approach for the GSI testing (chapter 4). Then different incremental engine efficiencies at idling, 50km/h and 100km/h could be considered. Since the final correction factors should be simple look-up tables the simulation was based on average values which were applied for all three speed steps of the MAC test cycle (Table 6).

Table 6: incremental specific fuel consumption values used in the model

Parameter	Value	Unit	Remarks
be	230	g fuel/kWh	Incremental fuel efficiency

Eta-el	50%	kwh_el/kWh_mech	Efficiency for supply of electric energy via generator and battery.

6.6 Calculation of the correction factor for the boundary conditions

If the FC_{AC} is calculated for the target settings of the test ($FC_{AC\text{-}t}$) as well as for the measured conditions ($FC_{AC\text{-}m}$), the correction factor for the variation in the boundary conditions is:

$$\boxed{C_{COP\,i} = \frac{FC_{AC-t_i}}{FC_{AC-m_i}}}$$

With i……… test-phase (idling, 50km/h or 100km/h)

The additional fuel consumption for the MAC system with the correction for Pe and for the boundary conditions is:

$$\boxed{FC_{MAC_i} = 3.6 \times C_{COP\,i} \times (C_{Pe_i} \times FC_{i\,Measured-AC-on} - FC_{i\,Measured-AC-off})}$$

With;

FC_{MACi}................corrected additional fuel consumption of the MAC system [kg/h]
$FC_{iMeasured\text{-}AC\text{-}on}$ ...average fuel consumption measured at speed step i in the phase with AC on [g/s]
$FC_{iMeasured\text{-}AC\text{-}on}$ ...average fuel consumption measured at speed step i in the phase with AC off [g/s]

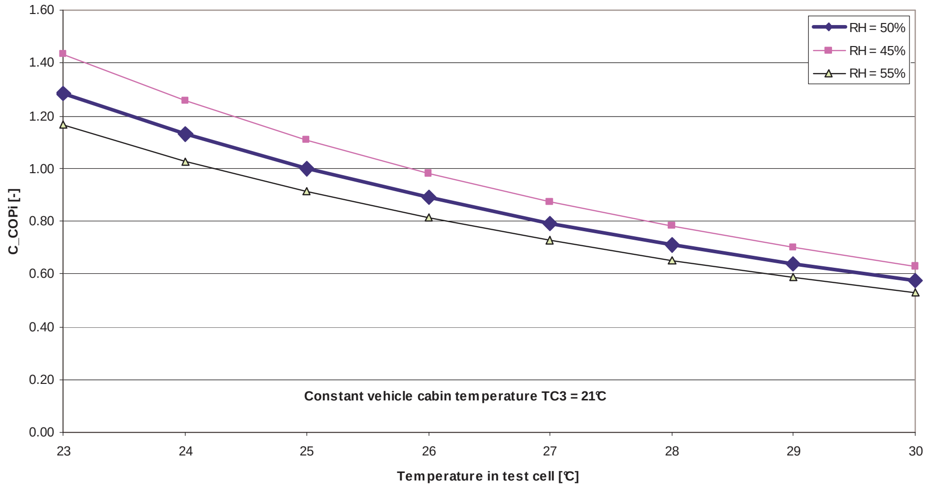
6.7 Results of the correction functions for ambient conditions

The basic relations of the entire MAC system with the simplified AC cycle have been programmed in an Excel sheet. With the results of the measurements the program was parameterized. The modeling gives correction functions for variations of test cell temperature and humidity as well as for a not constant cabin temperature level during the test.
Options for the corrections are:
– Calculate C_{COPi} from the functions as described above or from COP maps and the additional data on electric energy consumers.
– Calculate correction factors for C_{COPi} and provide them in look up tables

An advantage of the latter is that the values could be checked by the industry rather easily and thus could be adapted according to input from different manufacturers if necessary. The advantage from the first is that non linear effects from simultaneous variations of all relevant parameters would be covered. However, the responds from the model are not very non-linear in the relevant ranges (Figure 28 and Figure 29).
Figure 28 shows a simulation result with the described tool for variations in the temperature and humidity. E.g. 1 percentage point change in the relative humidity results here in approx. 2% change of the fuel consumption (increasing humidity

increases the Enthalpy of the entering air flow and can increase also the condensation of water). A change of 1°C leads here to approx. 10% change in the resulting fuel consumption.

Certainly the sensitivity of the system depends on the MAC settings (temperature, blower stage, share of recycled air), on the COP of the AC cycle and on the efficiency of the combustion engine in the actual load point.
The following graphs show results for variations in temperatures and in the relative humidity.



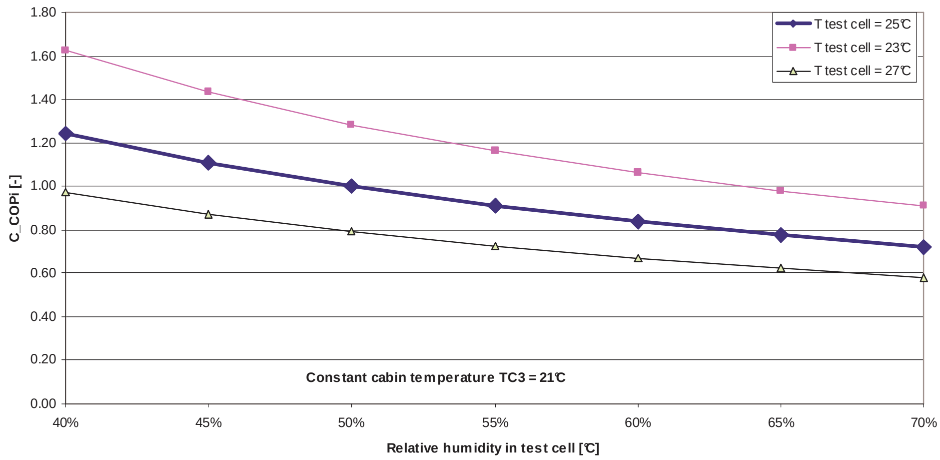


Figure 28: Simulated correction factor for a variability in the ambient temperature (left) and in the humidity of the ambient air (right) for the additional fuel consumption from a MAC system

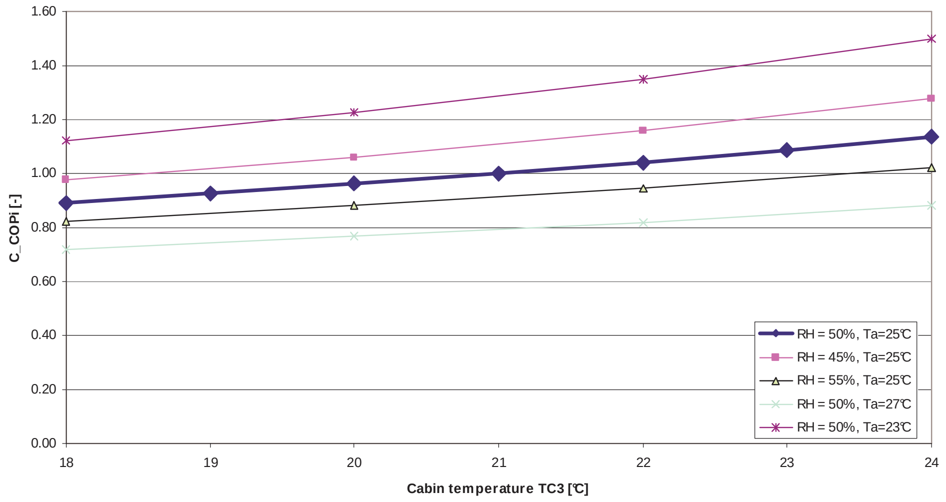


Figure 29: Simulated correction factor for a variability in the cabin temperature TC3 for the additional fuel consumption from a MAC system

It is suggested to prescribe such correction functions for a fair procedure and to prevent misuse of the settings during type approval.
For look up tables the simplest approach is a multiplicative approach, since otherwise a rather large table is needed to cover all combinations of variations in T_1, T_{C3} and RH :

$$\boxed{C_{COP\,i} = C_{COP\,i-T1} \times C_{COP\,i-RH} \times C_{COP\,i-TC3}}$$

With;

$C_{COP\,i-T1}$ ....Correction factor for variation of test cell temperature T_1 with T_{C3} and RH being exactly at the target values

$C_{COP\,i-RH}$ ....Correction factor for variation of test cell humidity RH with T_{C3} and T_1 being exactly at the target values

$C_{COP\,i-TC3}$ ....Correction factor for variation of cabin temperature T_{C3} with RH and T_1 being exactly at the target values

Certainly this multiplicative approach is a simplification since not all variations are in linear relation. However, the simulation approach shown before suggests that this approach has only small deviations against the direct simulation (Figure 30).

Since the correction in general can not depict the behaviour of each single AC system exactly, the deviations seem to be acceptable.

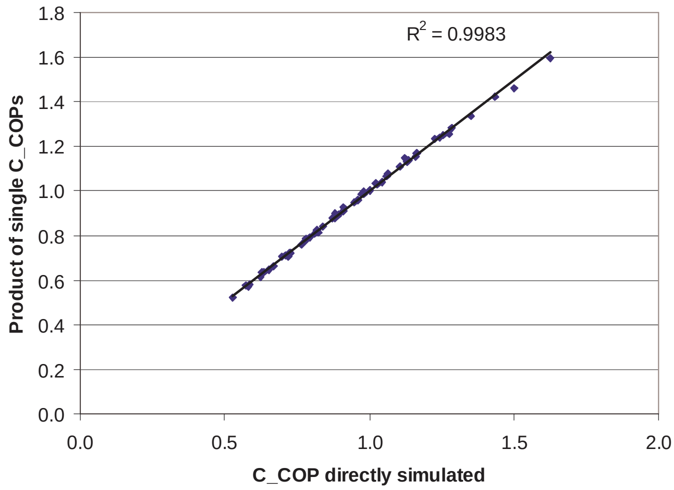


Figure 30: correction factor for a simultaneous variability in T_1, RH and T_{C3} for the additional fuel consumption from a MAC system (directly simulated with COP model shown before on x-axis and product from the 3 single C_{COP} on y-axis)

We suggest to collect suggestions for the single correction factors $C_{COP\,i-T1C3}$, $C_{COP\,i-RH}$ and $C_{COP\,i-T1}$ from ACEA. Maybe this is the simplest way to get an agreement for the correction method and for the values (→ questionnaire to ACEA). Table 7 summarises the values calculated from the simplified model described before.

Table 7: draft for look up table for correction factors)

t1 [℃]	RH1 [%]	t3 [℃]	C_{COPi_T1}	C_{COPi_RH}	C_{COPi_TC3}
25	50%	21	1.000	1.000	1.000
23.00	50%	21.00	1.285	1.000	1.000
24.00	50%	21.00	1.131	1.000	1.000
25.00	50%	21.00	1.000	1.000	1.000
26.00	50%	21.00	0.888	1.000	1.000
27.00	50%	21.00	0.793	1.000	1.000
28.00	50%	21.00	0.710	1.000	1.000
29.00	50%	21.00	0.637	1.000	1.000
30.00	50%	21.00	0.574	1.000	1.000
25.00	40%	21.00	1.000	1.242	1.000
25.00	45%	21.00	1.000	1.108	1.000
25.00	50%	21.00	1.000	1.000	1.000
25.00	55%	21.00	1.000	0.912	1.000
25.00	60%	21.00	1.000	0.838	1.000
25.00	65%	21.00	1.000	0.776	1.000
25.00	70%	21.00	1.000	0.722	1.000
25.00	50%	18.00	1.000	1.000	0.893
25.00	50%	20.00	1.000	1.000	0.962
25.00	50%	22.00	1.000	1.000	1.042
25.00	50%	24.00	1.000	1.000	1.136

First validation of the correction functions

The correction functions suggested here may be adapted after further tests in a pilot phase of the MAC test procedure. This would need tests on more vehicles with systematic variations of the test cell temperature and humidity as well as of the cabin temperature (or vent outlet temperature) against the basic settings. It has to be pointed out, that the correction factors can only be a reasonable average for the existing MAC systems since certainly each MAC system will react somewhat different on changes in the running conditions. However, if the tests can be kept within the defined tolerances, the correction factors should help to improve the repeatability for all systems.
In addition to the validation of the correction functions with the test results (chapter 7.5) a comparison with results from other model sources has been performed. One model source was the tool used for the “seasonal performance (SPF)” simulation described in chapter 5.1, the second tool was the calculation of the Enthalpy differences of the air mass flow through the MAC system at different settings of the temperatures and humidity as described in the Annex. The results from the latter were provided by the colleagues from PSA. The seasonal performance tool allows only the variation of the ambient temperature and humidity while the cabin temperature is not an explicit variable. Thus differences in the results can also be an effect of differences in the assumptions for the MAC settings.
In general the different sources give quite similar results (Figure 32). One outlier occurs from the SPF tool for 30°C test cell temperature. Since this temperature is far beyond the tolerances defined for the test cell temperature no detailed analysis of the resons for this deviations have been performed.

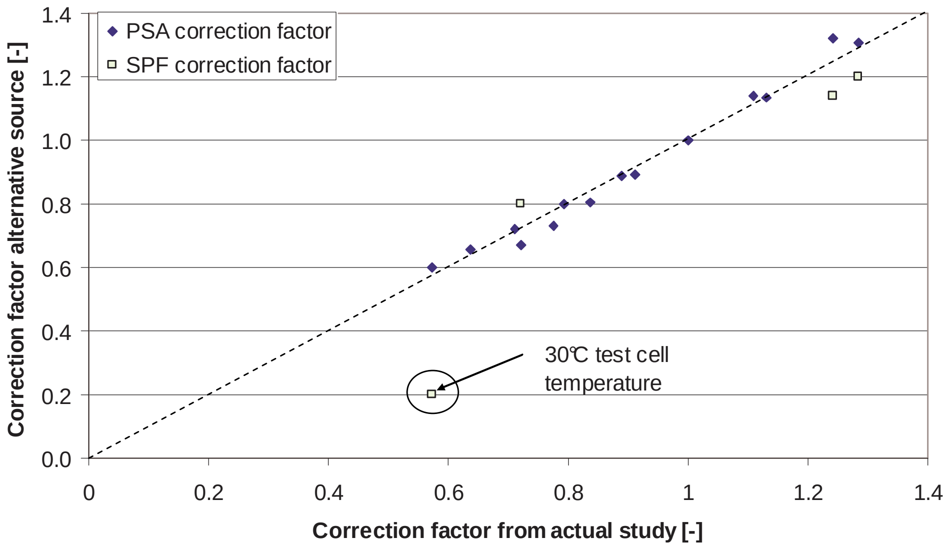


Figure 31: Comparison of the correction factor s from different sources for a simultaneous variability in T_1, RH and T_{C3} for the additional fuel consumption from a MAC system

6.8 Correction for vehicle glazing

The influence of the glazing on the fuel consumption of the MAC system was already discussed in chapter 5.3.4. The effects of the design and quality of glazing on the heat load in the cabin are rather complex. Furthermore the subjective comfort of the persons is affected by the transmitted energy from the sun radiation. Thus as reaction to radiation typically the mass flow and/or the temperature at the vent outlets are adapted at modern MAC systems. The quality of glazing affects the specific energy balance per m² window (Figure 32). The energy from sun radiation is partly reflected while a reasonable proportion enters the car via transmission and via radiation to the cabin from the hot glass.

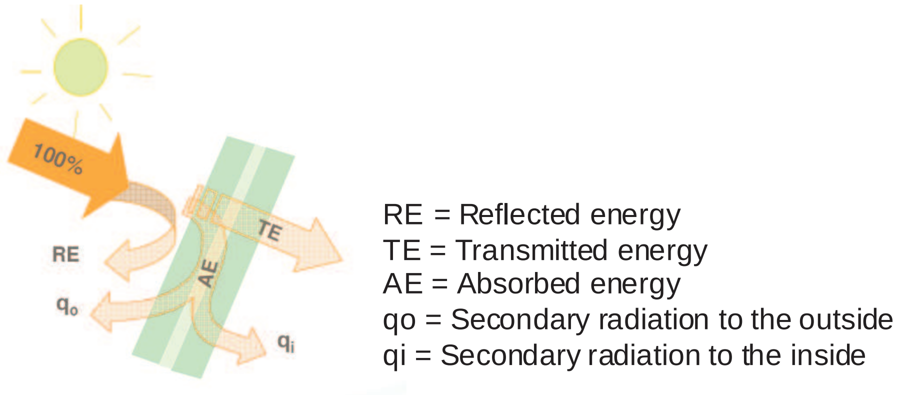


Figure 32: schematic picture of energy flows on a window
(source S. Hessel, F. Manz; Saint-Gobain Sekurit)

6.8.1 *Options to consider the sun radiation in the test procedure*

The installation of spotlights with a similar energy spectrum as the sunlight on roller test beds for type approval would be quite costly, thus an alternative option is suggested.

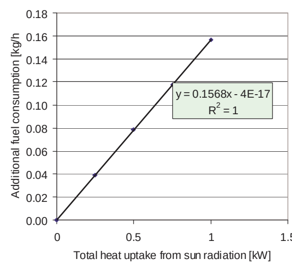
Step 1) calculate the total heat uptake of the cabin from the sun radiation through the glazing for steady state conditions (see chapter 6.8.2)
Step 2) Correction of the test procedure settings according to the total heat uptake. There are 3 options, where some experience in practical testing should be gained before a final decision.

Step 2) Option a: “table for additional fuel consumption”

The additional fuel consumption is simply calculated for the additional heat source (i.e. the total heat uptake according to step 1) with the formulas shown in chapter 6. This simulated fuel consumption could be added to the basic test results for the additional MAC fuel consumption without sun radiation. Test runs of the model for the diesel car measured showed a linear dependency for the additional fuel consumption (Table 8).

Table 8: Additional fuel consumption of the MAC system for different heat uptake from sun radiation for a diesel car (Option a)

sun energy entrance [kW]	Simulated FC [kg/h]	Additional FC [kg/h]
1	0.472	0.157
0.75	0.433	0.118
0.5	0.393	0.078
0.25	0.354	0.039
0	0.315	0.000



Step 2) Option b: “table for adaptation of the AC mass flow”

The additional fuel consumption is again calculated for the additional heat source (i.e. the total heat uptake according to step 1) with the formulas shown in chapter 6. Then the additional mass flow of air through the AC-system is simulated, which leads to the same amount of additional fuel consumption.

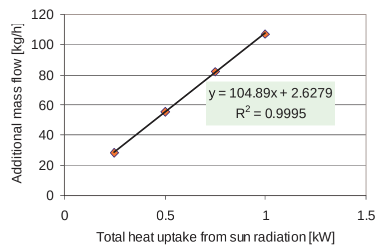
An adaptation of the mass flow most likely is closest to the realistic behaviour of the AC system at automatic settings if a solar sensor is involved. A disadvantage is that the mass flow can be adjusted only stepwise and thus the options for variations are limited. Since we may be faced to five or more typical glazing qualities in at least three different categories of glazing sizes and angles (Van, SUV, car), the actually available steps to adjust the mass flow will certainly not allow to depicture all variations. Furthermore the mass flow can hardly be measured during the type approval tests of the AC system.

An option is, to calculate the additional mass flow and define the settings in the type approval test of the AC system then as “Mass flow ≥ setting”. This would allow the manufacturer to adjust the mass flow steps to the actual vehicle/glazing combination in type approval.

For the tested diesel car the additional mass flow has been calculated for different additional energy entrance from sun radiation (Table 9). The energy demand of the blower increases disproportionately high with increasing mass flow, the CAP increases linear with the mass flow. In total a nearly linear effect shows up. This would allow to apply a simple formula for the additional mass flow in the settings of the test procedure.

Table 9: Additional mass flow of air through the MAC system for different heat uptake from sun radiation for a diesel car (Option b)

sun energy entrance	Additional mass flow
[kW]	[kg/h]
1	106.76
0.75	82.30
0.5	55.32
0.25	28.35
0.0	0.00



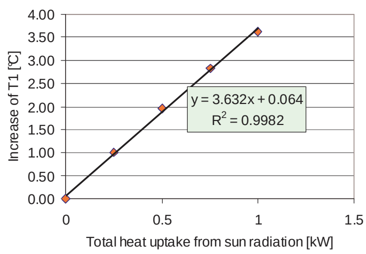
In practice the increase against a minimum value of energy entrance may be a useful approach (e.g. D against 0.25kW)

Step 2) Option c: “table for adaptation of test cell temperature”

The additional fuel consumption is again calculated for the additional heat source (i.e. the total heat uptake according to step 1) with the formulas shown in chapter 6. Then the test cell temperature is simulated, which leads to the same amount of additional fuel consumption (alternatively the cabin temperature target could be reduced but high additional energy loads from sun radiation would lead to very unrealistic low cabin temperature settings).

For the tested diesel car the cabin temperature has been calculated for different additional energy entrance from sun radiation (Table 10). The additional temperature follows a linear dependency, thus a simple correction function for the settings of the cabin temperature in the test procedure could be achieved.

Table 10: Settings of the test cell temperature for different heat uptake from sun radiation for a diesel car (Option c)

sun energy entrance	Increase of test cell temperature	
[kW]	[kg/h]	
1	**3.63**	
0.75	**2.82**	
0.5	**1.95**	
0.25	**1.00**	
0.0	**0.00**	
In practice the increase against a minimum value of energy entrance may be a useful approach (e.g. D against 0.25kW)		

For the options b and c the amount of air from the cabin recirculated is an important factor. If 100% of the mass flow are recirculated, a higher temperature in the test cell according to option c will have nearly no effect (only heat transmission) and also the additional mass flow in option b would mainly lead to a higher power demand from the blower but not to an increased CAP.
Thus the factors shown for the options b and c would have to be corrected by the correction factor for recirculated air C_{ra} if correct results should be achieved:

$$C_{ra} = \frac{1}{(100\% - \%\,\text{air recirculated})}$$

As a consequence the % air recirculated have to be limited, e.g. to 80%. Options to control the % air recirculated during type approval by independent parties are however not known yet. This suggests option a as the most robust approach from today's point of view.

6.8.2 *Calculation of the heat entrance into the cabin*

The method of the simulation was provided by Saint-Gobain Sekurit, Mr. Florian Manz and Mr. Volkmar Offermann in form of an Excel tool. The basic approach and some results for two vehicle categories are provided in the following

6.8.3 *Energy balance*

The total heat balance for the specific heat flux for a window pane is:

$E_{\text{total sun radiation}} = E_{\text{absorbed}} + E_{\text{transmitted}} + E_{\text{reflected}}$

A part of the heat is reflected from the window, and the other part is absorbed and a part is transmitted to the interior (see Figure 33). The absorbed energy itself leads to a heat flow to from the window to the cabin and to the outside of the car. If the situation is static (i.e. the temperatures do not change over time), the following equation valid:

$E_{\text{absorbed}} = E_{\text{re-emitted, i}} + E_{\text{re-emitted, e}}$

The flux of specific energy from sun radiation into the cabin of a car consists of the directly transmitted radiation $E_{\text{transmitted}}$ and the flux which is emitted due to the absorbed energy in the window pane $E_{\text{re-emitted, i}}$.

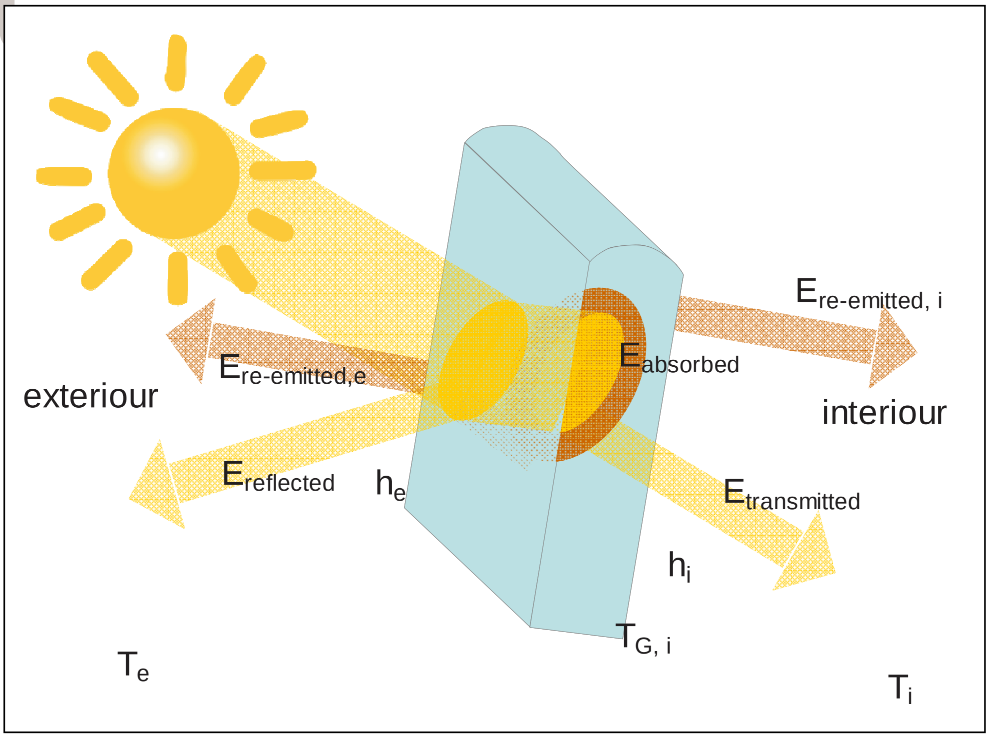


Figure 33: schematic picture of the energy balance over a window.

The flux of specific energy from sun radiation into the cabin of a car consists of the directly transmitted radiation $E_{\text{transmitted}}$ and the flux which is emitted from the amount of absorbed energy in the window pane $E_{\text{re-emitted, i}}$.

$$\boxed{\mathbf{E}_{\textbf{interior}} = \mathbf{E}_{\textbf{transmitted}} + \mathbf{E}_{\textbf{re-emitted, i}}}$$

The specific heat flow to the inside of the car $E_{\text{re-emitted, i}}$ is calculated as convective heat transfer (from the inner side of the glass surface) as follows:

$$E_{\text{re-emitted, i}} = h_i \cdot (T_{G,i} - T_i)$$

h_i...........heat transfer coefficient for the interior of the car

$T_{G,i}$.........temperature at the inner surface of the glass

T_i.............interior temperature

The specific heat transfer form the inner surface to the ambient is calculated as follows:

$$E_{\text{re-emitted, e}} = h_e^{'} \cdot (T_{G,i} - T_e)$$

Where $h_e^{'}$ is the heat transfer coefficient from the inner surface to the ambient:

$$h_e^{'} = \frac{1}{\frac{1}{h_e} + \frac{d}{\lambda}}$$

h_e...........heat transfer coefficient from the outer surface of the glazing to the ambient

h_e'..........heat transfer coefficient from the inner surface to the ambient (dependent on he and the heat conduction through the glazing)

d...........thickness of the window pane

λ.........heat conduction coefficient of the window pane

6.8.3.1 *Calculation of the heat transfer coefficients*

The heat transfer coefficient for the exterior of the car is obtained from **ISO 13837** for the four wind velocities of 4, 14, 28 and 42 m/s (according to vehicle velocities of 0, 50, 100 and 150km/h).

h_{e1}=21 W/(m^2K) for a velocity of 4 m/s (parking)

h_{e2}=61 W/(m^2K) for a velocity of 14 m/s

h_{e3}=106 W/(m^2K) for a velocity of 28 m/s

h_{e4}=146 W/(m^2K) for a velocity of 42 m/s

The heat transfer coefficient for the interior is calculated as shown in equation below according to **ISO 13837:**

$$h_i = 3{,}6 + \frac{4{,}4 \cdot \varepsilon_i}{0{,}837} W/(m^2 K)$$

ε_i........corrected emissivity (defined and measured in accordance with **EN 673**)

For ordinary glass: εi=0.837 and h_i=8W/(m^2K)

6.8.3.2 *Calculation of the absorbed energy*

The absorbed energy is calculated with the transmittances of the glazing material (according to **ISO 13837**):

Definition of the transmittance:

$$T = \frac{transmitted\ flux}{incident\ flux}$$

with the energy balance:

$E_{\text{total sun radiation}} = E_{\text{absorbed}} + E_{\text{transmitted}} + E_{\text{reflected}}$

this leads to:

$$1 = \frac{E_{absorbed}}{E_{totalsunradiiation}} + \frac{E_{transmitted}}{E_{totalsunradiiation}} + \frac{E_{reflected}}{E_{totalsunradiiation}}$$

$$1 = \alpha_e + T_{Ds} + R_{Ds}$$

T_{Ds}...solar direct transmittance obtained for different types of glass according to **ISO 13837**

R_{Ds}...solar direct reflectance obtained for different types of glass according to **ISO 13837**

$$E_{absorbed} = \alpha_e \cdot E_{totalsunradiation}$$

$$E_{reflected} = R_{Ds} \cdot E_{totalsunradiation}$$

$$E_{transmitted} = T_{Ds} \cdot E_{totalsunradiation}$$

The total solar transmittance T_{Ts} of a glazing is the sum of the solar direct transmittance and the secondary heat transfer factor q_i of the glazing towards the inside.

$$T_{Ts} = T_{Ds} + q_i$$

q_i describes the part of the absorbed thermal energy in the glazing which enters the cabin.

q_i is calculated according to **ISO 13837** as follows:

$$q_i = \frac{h_i}{h_e + h_i} \cdot \alpha_e$$

h_e and h_i are the heat transfer coefficients from the outer surface of the glazing to the ambient and the inner surface of the glazing to the interior.

The amount of total sun radiation will be calculated with a solar intensity in W/m^2 obtained from meteorological data and under consideration of the resulting angles of the sun radiation to the various window panes in the car. The resulting angles are calculated from the installation angles of the panes in the car and the declination of the sun radiation obtained from meteorological data. In our case the place of Paris was chosen to calculate the angle of incidence of the sun radiation. The radiation angles are calculated for every month over the year 5 times a day at every 21rd of the month. To calculate the resulting angles of the sun radiation on the window panes 16 driving directions have been evaluated. The result is an average resulting angle of the sun radiation on the glass for different installation angles (in steps of 10 degrees) of the specific window pane.

6.8.4 *Application of the calculation scheme*

In principle there are two options for the application of the calculation of the total energy uptake:

- define all formulas and/or provide the Excel tool to calculate the energy uptake for each vehicle individually
- provide look up tables for the specific total energy uptake [W/m²] for different vehicle categories (to consider different angles of the glazing) and for different glazing qualities.

To be discussed (→questionnaire to manufacturers + ACEA?)

To test the applicability of option b) results from the Excel tool for three different vehicle categories (different size and angles of the glasses) with different glazing qualities are shown.

Three makes and models were selected to cover typical estate, Van and SUV cars. Beside the different glazing surface also the angles of the windows typically differ between these categories. Table 11 shows the basic data assumed for the simulation.

Table 11: general data for the simulation of the influence of sun radiation at 3 vehicles

	Estate	Van	SUV
Total glazing surface [m2]	1.73	2.84	2.1
Sun intensity [W/m^2]	700		
Outside temperature [℃]	25		
Temperature cabin [℃]	21		

For the glazing five typical combinations of glass qualities with the corresponding glass properties were selected according to the experience of Saint-Gobain Sekurit.
The heat transfer was simulated for idling, 50km/h and 100km/h respectively since the heat transfer coefficient increases with the wind speed. Then the weighted average of the results for the single speed steps was calculated according to the weighting factors suggested in chapter 3.

Table 12: combinations of glazing qualities in the 5 variants used in the simulation

<table>
<tr><th>window</th><th>Variant 1</th><th>Variant 2</th><th>Variant 3</th><th>Variant 4</th><th>Variant 5</th></tr>
<tr><td>windscreen</td><td colspan="2">2.1mm lite green-0.76mm PVB-1.6mm clear</td><td colspan="3">IRR coating 2.1mm clear-0.76mm PVB-1.6mm clear</td></tr>
<tr><td>front door quarter (1)</td><td colspan="3">3.15mm lite green</td><td colspan="2" rowspan="2">2.1mm clear-PVB-PET-PVB-2.1mm clear</td></tr>
<tr><td>front door side lite</td><td colspan="3">3.85mm lite green</td></tr>
<tr><td>rear door side lite</td><td rowspan="2">3.85mm lite green</td><td rowspan="2">3.85mm dark grey</td><td rowspan="2">3.85mm lite green</td><td rowspan="2">2.1mm clear-PVB-PET-PVB-2.1mm clear</td><td rowspan="2">3.85mm dark grey</td></tr>
<tr><td>Rear (2)</td></tr>
</table>

(1)….the vehicle used as base case had a front door quarter. This is not very typical for cars but has no relevant influence on the results
(2)… rear door side lite, rear door quater, site lites, backlite

Figure 34 and **Error! Reference source not found.** shows the results as absolute values as well as the specific energy per m². The absolute energy entrance from sun radiation is influenced very much by the total glazing surface and by the glazing quality. The average energy entrance is 635 W, the maximum value is 945 W for a Van with low glazing quality (+49% against average), the minimum value is 392 W for the estate with high glazing quality (-38% against the average).

The specific values indicate that the effects of different declinations of the glazing have a rather small influence since the W/m² are not differing very much between the 3 vehicle categories (less than 5% difference between average and maximum values). The glazing quality however has a huge influence. The W/m² differ between the best quality and the lowest quality on average of the three vehicle classes by 30%.

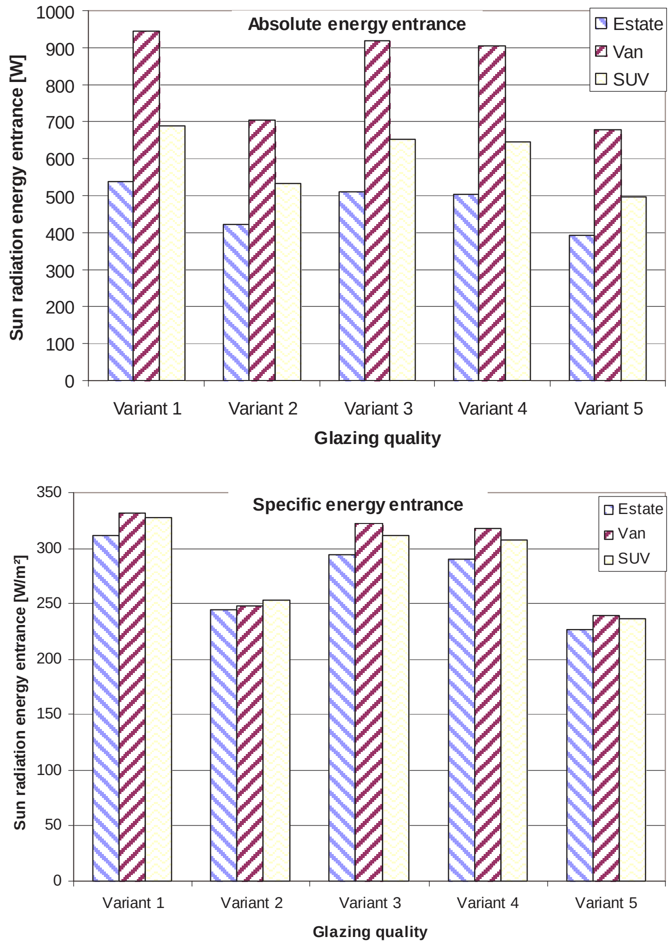


Figure 34: Results for the energy entrance into the cabin due to 700W solar radiation for 3 vehicle categories and for 5 types of glazing (left picture shows the absolute values, the right picture shows the specific energy per m²)

This indicates that the simplest solution would use a look up table with specific values for the energy entrance in W/m² as function of the glazing quality. The glazing quality of the backlite has a high influence on the result, since it allows the usage of rather dark glasses, the surface can be high and the average angle for the sun radiation leads to a higher share of transmitted energy than for the door glazing (compare variants 2 and 5 with dark grey glass and the other glazing variants). Also the wind screen has a reasonable influence due to the surface and angle.

Table 13: specific energy entrance from a sun radiation of 700 W calculated for three vehicle categories and five glazing variants

Glazing	Estate	Van	SUV	average
	[W/m²]	[W/m²]	[W/m²]	[W/m²]
Variant 1	310.8	332.0	327.2	**323.3**
Variant 2	245.0	248.0	253.3	**248.8**
Variant 3	293.9	322.5	311.2	**309.2**
Variant 4	290.2	318.2	307.4	**305.3**
Variant 5	226.6	239.1	235.8	**233.8**
Average 1	**273.3**	**292.0**	**287.0**	**284.1**

The glazing quality depends mainly on following values:
1. transmission of direct sun, T_{ds} (measured according to ISO 13837)
2. reflection of direct sun, R_{ds} (measured according to ISO 13837)
3. thickness of the layers (glass, PVB, PET, PVB, glass)

The glazing properties of T_{ds} and R_{ds} can be combined in the
4. total heat transmission (TTS) (according to ISO 13837)

If a simple look-up table shall be produced, the specific values given in **Error! Reference source not found.** as function of the glazing variants would have to be converted in tables as function of the parameters 1., 2. and 3 or parameter 4. Also the angles of the windows are relevant and should be taken into consideration.

Since a 4 dimensional look up table is too difficult to handle, it is suggested as **a first option** to set up the table as function of T_{TS} and of the angle of the window. For each window category (wind screen, front door,..) a separate table should be established to be able to depicture the different angles of the different windows correctly. The table should deliver the W/m². Together with the size of the window [m²] the heat entrance could be calculated straight forward for each window. The total heat load is then the sum of the single windows heat load.
The T_{TS} values could be gained according to ISO 13837 from the results for 0km/h, 50km/h and 100km/h wind speed. The weighting of these single values could follow the weighting factors of the MAC test amended by an increased share of 0km/h wind speed to represent the heat entrance during the parking time of the vehicle with engine off:

$$T_{TS} = A_P \times T_{TS-0km/h} + A_D \times \left(0.15 \times T_{TS-0km/h} + 0.65 \times T_{TS-50km/h} + 0.2 \times T_{TS-100km/h}\right)$$

With AP ......weighting factor for heat entrance during parking
AD......weighting factor for heat entrance during driving the vehicle

A_P and A_D need to be defined from average vehicle operation data if option 1 shall be applied. The sum of A_P and A_D should be 1.
The look-up table could be combined with the additional fuel consumption value per kW energy entrance according to option a) (Table 8) to obtain a very simple method to take the glazing quality into consideration. Figure 35 shows a schematic picture of this approach.

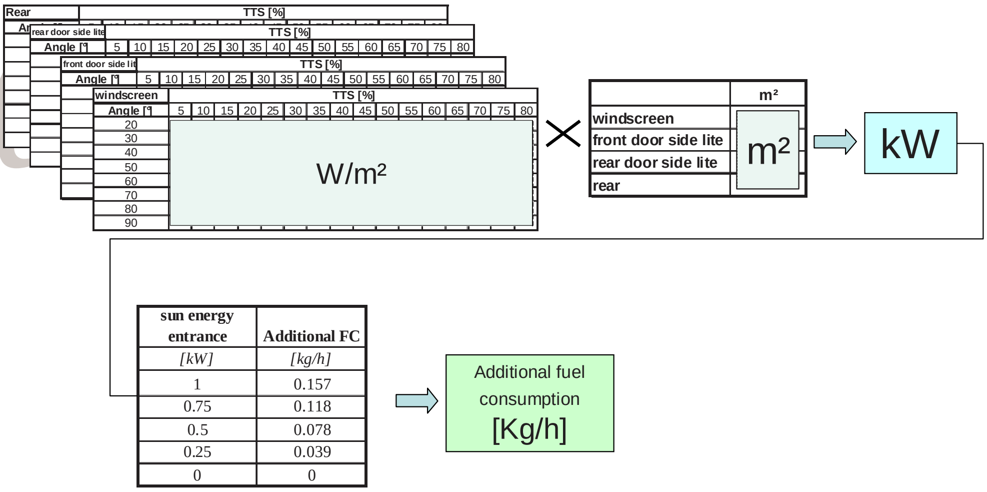


Figure 35: Schematic picture of the application of the look up tables for the heat entrance per screen and for the additional fuel consumption per kW heat entrance due to the sun radiation as function of the glazing quality described by the T_{TS} value

With the preliminary tables the minimum value of 392 W heat entrance shown before would result in additional 0.06 kg/h fuel consumption for the MAC, the average of 635W in +0.15 kg/h and the maximum energy entrance of 944 W would lead to additional 0.15 kg/h fuel consumption. For such a van with low glazing quality the “penalty” for the MAC fuel consumption would be approximately 35%. The final value depends on the average sun radiation intensity underlied for the look up tables and can differ from the values stated here for 1000W/m². With the 700W/m² suggested below to represent average European conditions the penalty would be some 25%.

This first option was elaborated together with experts from Saint-Gobain Sekurit[6] and was discussed also with experts from NSG[7]. Although this approach certainly includes several simplifications it is assumed at the moment to give fair results for improved glazing qualities to achieve a better fuel consumption value for the entire system of the MAC system and the vehicles glazing.

As **second option** the mass flow of fresh air through the MAC system could be linked to the look up tables for the W/m² from the different windows. An increased mass flow is a physically correct response of the MAC system to manage a higher energy entrance from sun radiation (see Table 9 in option b) in chapter 6.8). The advantage of this second option is that a more efficient MAC system together with a more efficient engine would get better test results compared to a systems with a worse efficiency. In the 1st option the efficiency is independent from the MAC and engine of the tested vehicle. Disadvantages of option 2 are, that in this case separate tests would be necessary for all different combinations of glazing quality available per vehicle model, that the mass flow hardly can be inspected on the roller test bed from type approval authorities and that maintaining different mass flow settings makes the type approval procedure more complex for the test bed personnel.

The **third option** would be the combination of the look-up tables with the increase of the test cell temperature according to Table 10 in option c) in chapter 6.8. This however

[6] Florian Manz and Volkmar Offerman
[7] Joe Boote

has also the disadvantage of more test demand for different glazing at one vehicle model and additionally needs a broader range of temperature control on the test stand and also extended ranges for the correction factors for the variability of this temperature. This third option therefore is not recommended.

A **fourth option** could be the direct application of the calculation tool for the heat load from sun radiation. This option seems to be quite unfavourable since a calculation tool is more difficult in the application than a simple look-up table and it would need reasonable more efforts to produce a generally accepted and validated calculation tool compared to a general accepted set of tables.

We suggest to test option 1 and/or option 2 in a pilot phase, depending on the preferences of the Commission. Before the start of the pilot phase also the values for the look-up tables can be produced from the simulation tool if option 1 should be followed. Other experts should then get the possibility to check the tables and produce results from other model approaches. If differences in the results arise, a working group would have to elaborate a common suggestion for the look up table data. However, since the physics of the simulation seem to be common knowledge, huge differences in the results are not expected.

To produce the look-up tables it is suggested to use a sun radiation on the windows from European cars of 700W/m². This value was agreed by the experts from Saint-Gobain and NSG but no study is available at the moment from which a more sounded value can be gained. The 700 W/m² is assumed to be higher than the EU average but since the MAC test cycle does not take the initial coold down of vehicles after parking into consideration the higher W/m² compensate for this influence. Since a small variation of this value would not influence the ranking of different glazing qualities, the existing uncertainty seems to be acceptable.

7 Vehicle tests performed and results

In this chapter the test facilities and test results are described.

7.1 Description of the test facilities

In this chapter the test facilities used are shortly described together with an analysis of the variation found for the temperatures and humidity during the tests.

7.1.1 TUG roller test bed

Manufacturer: Zöllner

Technical Data:

Brake:	56 kW DC Machine+ 240 kW AC Machine
Max. velocity:	200 km/h
Temperature control:	-30°C to +40°C
Simulated vehicle mass:	567 to 2325 kg (55kg increment)
CVS flow rate:	6, 10 or 20 m^3/min
Controlled humidity.	

Operating Modes

The test bed can be operated in transient and in steady state mode in four quadrant operation.
Steady state mode (driving-performance test): This operation mode can be controlled either for constant braking force or for constant velocity.
Transient mode: In this mode the driving resistances of the vehicle are simulated as function of the vehicle speed and acceleration as defined by the input data on vehicle mass and the driving resistance values. The speed pattern to be driven can be selected freely and is specified for the driver by the control device on the monitor. Typical application is type approval testing for cars and light commercial vehicles.
In the actual test series the transient mode of the test stand was used.

Exhaust gas analysis

The exhaust gas emissions are measured via a full flow CVS system and analysers from AVL (CEB II). Emissions are measured as bag values and as instantaneous data (up to 3 Hz). Special analysers can be added, such as a particle number measurement system and a FTIR.

Chassis dynamometer settings during all tests at the diesel car tested at TUG:
Equivalent mechanical inertia: 1605 kg

Driving resistance from coast down test on test track around the city of Graz:

$F [N] = 210.13 + 0.17 * v + 0.47 * v^2$

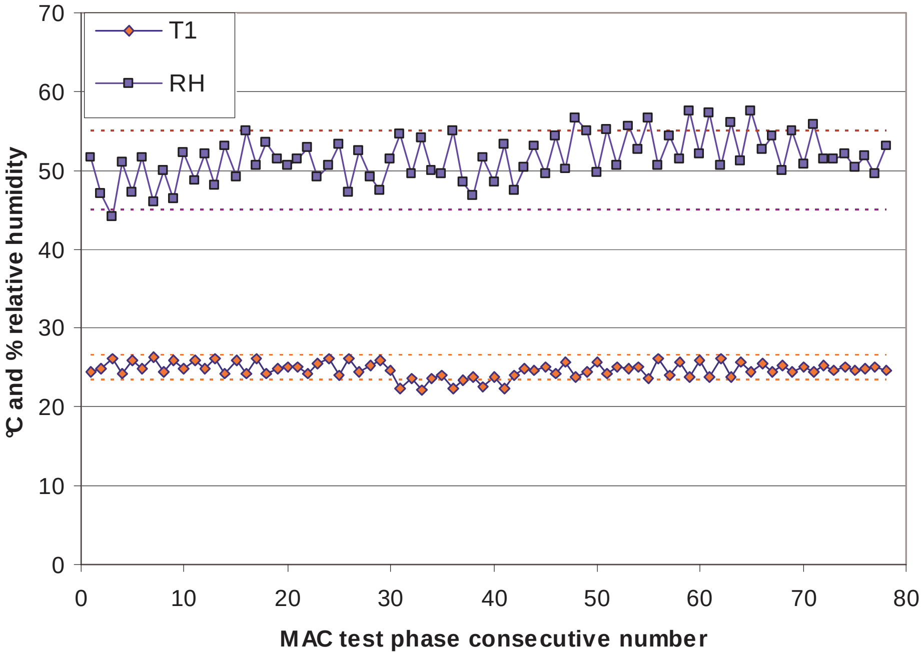
v….velocity in m/s

Variations in test cell temperature and humidity occurred during the tests
In 13 of the MAC tests the target settings were T1 = 25°C and RH = 50% relative humidity. In other tests sensitivity runs for T1 and RH for the effect on the MAC fuel consumption were done or other test cycles were driven. Table 14 summarises the variability of temperature and humidity on the test bed during these MAC tests.

Table 14: Variability of temperature and humidity on the test bed during the MAC tests

	T1	RH
	[℃]	[%]
Average	24.7	51.5
Maximum	26.2	57.4
Maximum	22.2	44.0
Standard deviation	1.0	2.9
% standard deviation	4%	6%

The average values met the tolerances suggested in chapter 3. Meeting the tolerances in the relative humidity of 50% +/- 5% proved to be the more demanding task than meetings the tolerances for the temperature (25°C +/- 1.5°C). Two of the MAC tests would have failed the tolerance criterions due to exceeding the temperature limits in single speed phases, five tests would have failed due to exceeding the RH limits in single speed phases (Figure 36).



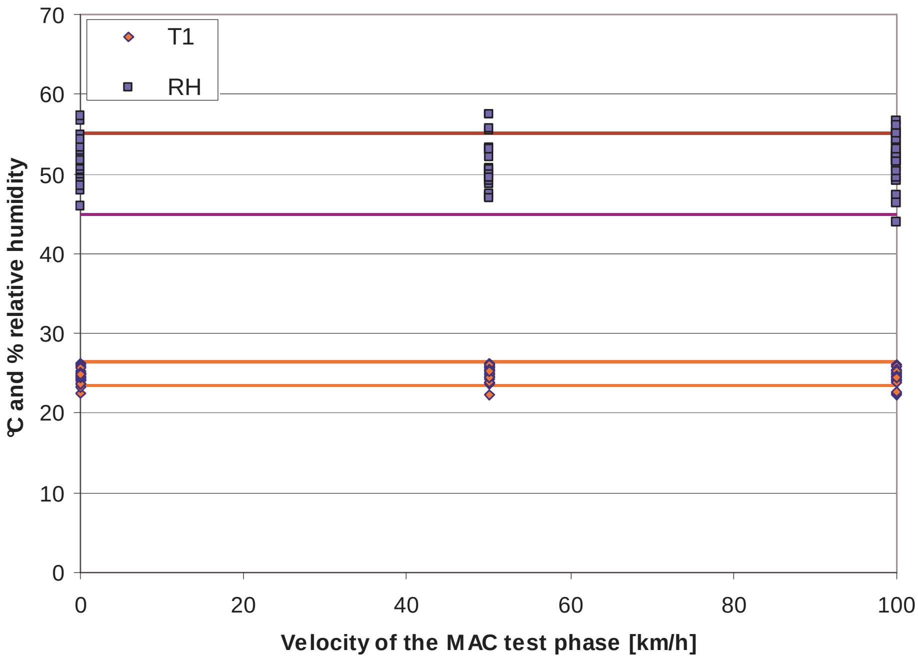


Figure 36: variability in temperature and relative humidity in the test cell during the MAC tests with settings of 25°C and 50° RH (left picture: plotted over the MAC phase number, right picture: plotted over the target speed of the test phase)

Figure 37 shows an example for the variability of the RH and test cell temperature over one single MAC test cycle. Both, the temperature and the humidity are controlled over periods of approximately 1000 seconds. For a better matching with the tolerances in a MAC test the periods obviously need to be shortened.

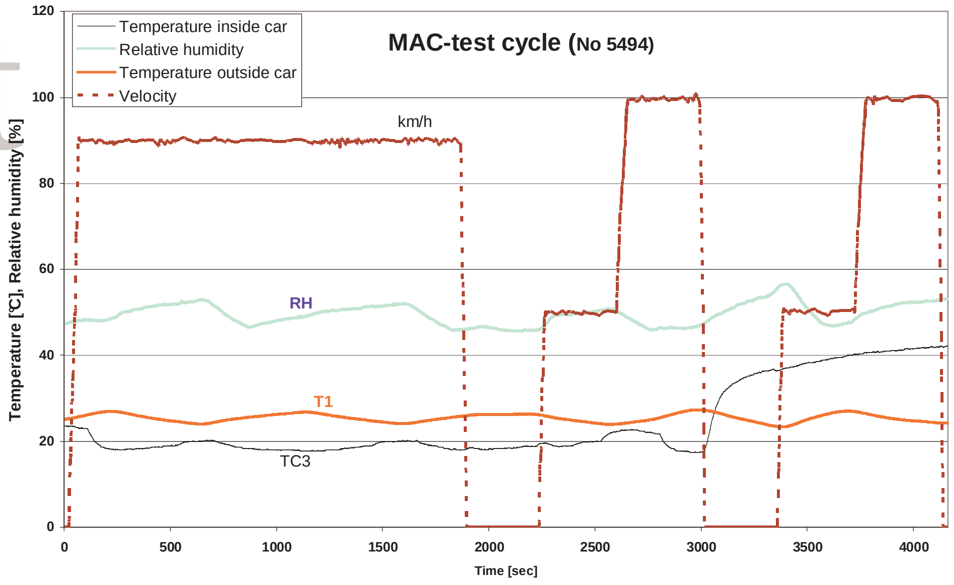


Figure 37: variability in temperature and relative humidity in the test cell during the one single MAC test with settings of 25°C and 50° RH

7.1.2 *LAT roller test bed*

Figure 38: Roller test bed at LAT

<u>Chassis dynamometer</u>

Type:	regenerative
Control:	Ward-Leonard
Inertia simulation:	mechanical
Maximum inertia:	1700 kg
Operation:	Predefined and custom transient driving cycles, steady state

<u>Exhaust gas analysis</u>

System:	full flow CVS

CVS flow rate: 700 Nm^3/h
Operating modes:
- Diluted exhaust gas sampling in bags
- Instantaneous emission analysis

Standard analyzers
- Regulated components (2+1 lines): CO (NDIR), CO_2 (NDIR), HC (FID), NO_x (CLD)
- Non regulated components: SO_2 (NDIR), N_2O (NDIR), H_2S COS (mass sp.), O_2 (par.)
- Equipment by Signal, Horiba, AVL, Hartmann Braun, ABB
- Real time sensors: NO_x, O_2

Particle measurements
PM gravimetric analysis (high precision balances in clean room), Opacimeter, smoke meter, Gravimetric Impactor, Scanning Mobility Particle Sizer (SMPS), Electrical Low Pressure Impactor (ELPI), Nano-DMA for measuring particles down to ~2 nm, Mass Monitor (DMM), Active surface monitor (diffusion charger) (ASMO), Thermodenuder, Tapered Element Oscillating Microbalance (TEOM)

Ultra fast response analyzers
- T_{90}: ~3 ms
- NO_x (NO, NO_2) (CLD): Cambustion fNOx400
- HC (FID): Horiba MEXA-1210FRF

Ambient conditions control
The test bed area at LAT is possible to be controlled in terms of temperature by heating or ventilating the room. Below are some measurements of temperature and relative humidity during the execution of driving cycles.

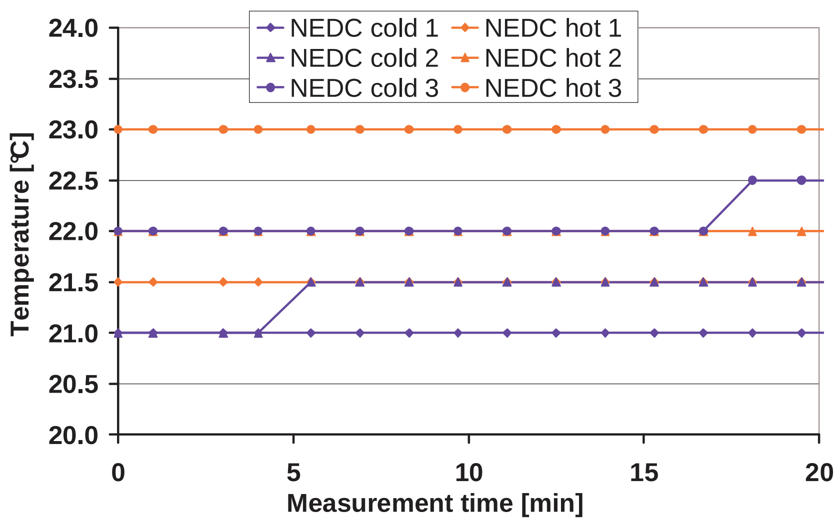


Figure 39: Temperature variation at the chassis dynamometer of LAT

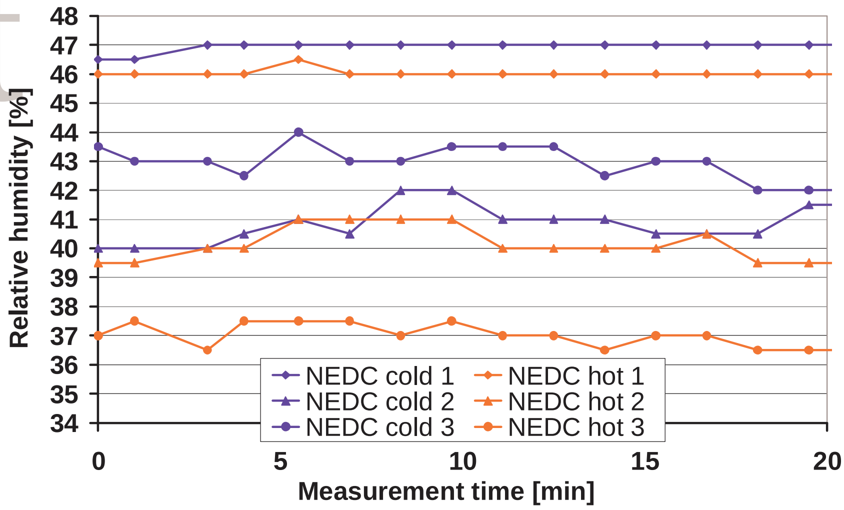


Figure 40: Humidity variation during at the chassis dynamometer of LAT

From the above, it is concluded that the variation of ambient conditions at the LAT test bench during one driving cycle is:

Temperature: less than ± 0.5ºC
Humidity: less than ± 1%

7.1.3 Measurements on GSI at LAT

The test vehicles were chosen equipped with a gear shift indicator. During testing, a video camera was installed to provide the driver with the gear shift information directly on the driver's aid screen installed outside the vehicle.

Figure 41: Camera to transfer the signal of the gear shift indicator to the driver's aid

The target of this testing was to assess the potential benefit of a GSI on the fuel consumption of the vehicle. During the testing of this system it was observed that the GSI did not suggest any gear shift modification besides the one predefined by the NEDC and the CADC cycles. More specifically, at the beginning of each measurement, the GSI proposed an initial shifting from 2^{nd} to 3^{rd} gear and then stayed neutral. The way this was interpreted was that the system was suggesting that the normal driving cycle gear shifting is acceptable by the installed GSI strategy. Consequently, no effect on fuel consumption was observed as expected and for this reason no GSI measurements were performed on the second test vehicle.

7.1.4 *KTI roller test bed*

Technical data of the Chassis dynamometer
Brake: Eddy current brake (TELMA). The load is controlled by a digital control system and software.
Max. brake torque: 800 Nm
Max brake power: 300 kW
Max. velocity: 140 km/h
Simulated vehicle mass by mechanical inertia (flywheels): 455 kg – 2270 kg
The speed pattern of driving cycle controlled by a computer and displayed on a monitor for the test driver.

Figure 42: The tested vehicle in the test cell at KTI.

Technical data of the test cell
Thermoisolated test cell with refrigerating, heating and air ventilation devices.
Temperature control: -30 °C to +35 °C
Heating power: 7 kW
Volume: 200 m^3
Humidity control: not controllable
Wind velocity of the blower: controllable as function of the vehicle velocity up to max. 20 m/s.
Exhaust gas sampling and analysis
Analyzers:
PIERBURG AMA 2000: $CO_{(H)}$, $CO_{(L)}$, CO_2, $NO_{(X)}$, THC/CH_4, O_2
HORIBA MEXA 7100 D: $CO_{(H)}$, $CO_{(L)}$, CO_2, $NO_{(X)}$, THC/CH_4

CVS system: full flow
CVS rates: 2-8,5 Nm^3/min
Diluted exhaust gas sampling into max. three different bags. Real time sampling with 1 Hz sampling rate, data recording by software.
PM and smoke measuring and devices:
SARTORIUS high precision balance in a clean room for PM gravimetric measuring.
AVL-439 opacimeter for smoke opacity tests.

Chassis dynamometer settings during all tests at the diesel car tested at KTI:

Equivalent mechanical inertia (according to the reference mass of vehicle): 1470 kg

Load: according to 70/220/EEC ANNEX III, *Appendix 2.*

Coefficient a = 7,4 N; Coefficient b = 0,0502 $N/(km/h)^2$

7.2 Repeatability

In this chapter the repeatability of the tests is discussed. Since the final design of the MAC test evolved over the test campaign, the number of MAC tests following the finally proposed design is quite limited. The tight time schedule did not allow extensive repetitions of tests when the final test procedure was proposed. Thus it is recommended to gain experience with the test procedure and to collect data on the repeatability as well as on pros and cons for different options in the test procedure described in chapter 3 before a final decision is taken.

7.2.1 Repeatability of constant speed tests

The standard deviation in the phases with AC-off in 24 MAC tests is shown in Table 15. A main contributor to the standard deviation is the variation of the vehicle speed against the target speed. Without correction of this variation the standard deviation of the measured fuel consumption was 6% in the weighted average of the MAC test. When the measured fuel consumption is corrected for the average breaking power of the test stand according to the formulas in chapter 6.1 the standard deviation of the weighted fuel consumption drops to 2%. The variation is typically highest in the idling phase.

Table 15: standard deviations from 24 MAC tests in the phases with MAC-off at the roller test bed from TUG

% StDev	Velocity	FC measured	Braking power	FC Pe-corrected
	Standard deviation [% from mean value]			
0 km/h AC-off	-	6%	-	6.0%
50 km/h AC-off	0%	3%	1%	2.7%
100 km/h AC-off	6%	15%	16%	1.5%
Total weighted Average	2%	6%	9%	2.1%

The standard deviations found are caused by inaccuracies from the analysers (2% from ultimate value), from the calibration gas for CO_2 (1%), from the variability of the engine running conditions and from parameters influencing the losses in the drive train of the vehicle (e.g. temperatures of the tires and of lube oil in the gear box etc.) as well as from differences in the battery SOC at the beginning and the end of the test cycle.

From 24 MAC tests performed at the test stand at TUG, the difference between the highest and lowest weighted average of the MAC test for AC-off was 6%. Approximately the half of this difference may be attributed to variability from the analyser calibration which will not influence the accuracy of the MAC test since the phases with MAC-on and MAC-off are driven consecutively with the same analysers calibration. The remaining difference may be attributed to parameters which can also influence the test results for the MAC. Increasing the number of tests and taking the average of these tests reduces the differences between the results (Figure 43). The most efficient drop is from 1 test towards 2 tests. With 2 repetitions the difference between maximum and minimum of the weighted test average drops from 6.4% to 4.8% and the standard deviation of the results drops from 1.9% to 1.4% of the average fuel consumption measured. Thus it is recommended to apply two repetitions of the entire MAC test.

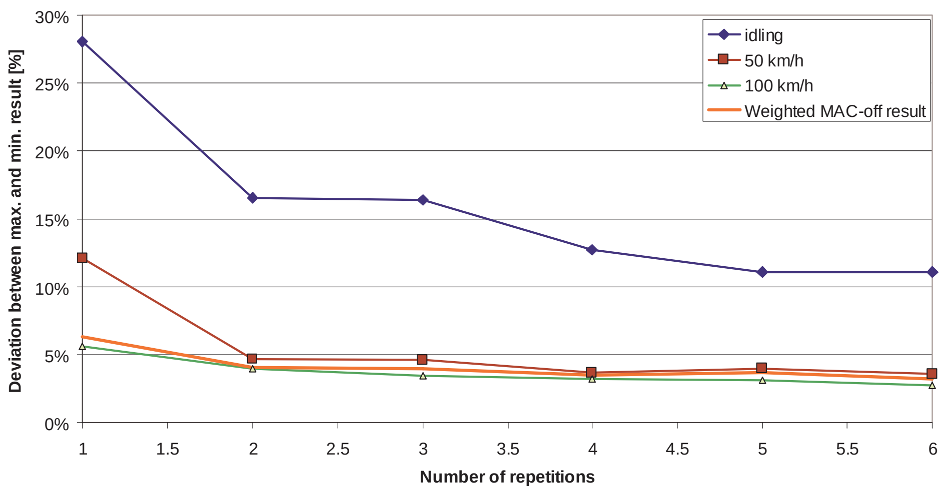


Figure 43: Deviation of the average fuel consumption for different number of tests fur averaging of the phases of the MAC test cycle

7.2.2 *Potential effects of the SOC of the battery*

With an effective testing time of 0.28 hours, the blower and fan of the MAC system may need less than 100Wh and the power demand of the compressor can be less than 300Wh. As long as the compressor is driven by the engine the compressor work can not be covered by the electric energy but future systems may use electric engines for the compressor. If a manufacturer charges the battery during the 0.5 h preconditioning and if we assume up to one kWh electric energy to be available in a battery, future MAC systems thus may drive the entire energy demand of the MAC system in the test from the battery.

7.2.3 *Repeatability of the MAC tests*

At TUG in total 24 MAC tests and were performed. In the first runs single steady state tests were performed to obtain results for the repeatability of temperature settings and to test the best locations for measuring temperatures and humidity in the test cell and in the vehicle. Then a two speed test cycle (idling and 65 km/h) was tested with and without heater in the vehicle. The heater was tested as option to depicture the effect of the sun radiation in the test procedure.

Figure 44 shows results from this two speed test cycle. With heater the MAC was set to maintain a cabin temperature of 21°C, without heater a cabin temperature of 15°C was set to replace the energy of the 1 kW heater by a lower cabin temperature. The repeatability with heater was clearly worse than without heater. The heater influences the air flow from the vents through the cabin and increases also the turbulence of the air flow. This leads to less stable conditions in the vehicle cabin.

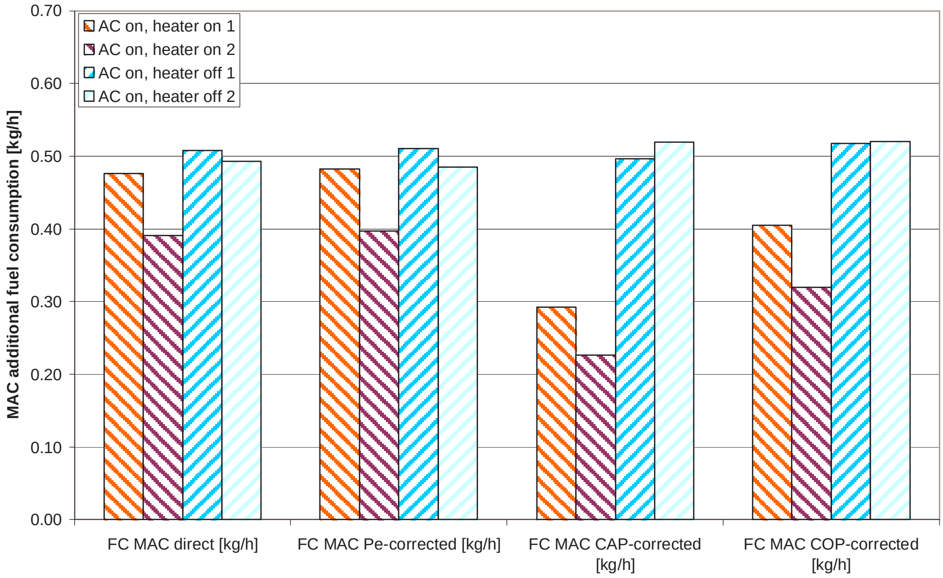


Figure 44: Results from a 2-speed test cycle at the diesel EURO 5 car at TUG with and without a 1 kW heater in the vehicle

Then the test campaign focussed on the MAC test cycle with the three speed ranges (idling, 50 km/h and 100 km/h). At the beginning options with 40° to 60% humidity and 15°C vent outlet up to 22°C cabin temperature were tested with different locations of the temperature sensors in the vehicle cabin. Finally the following settings were found to give good results and to be also reasonable in line with average European ambient conditions:

– 21°C as average of 3 temperature sensors in the cabin as defined in chapter 3.2 with 25°C test cell temperature and 50% humidity
– 15°C as maximum of the temperature of the air flow at vent outlet as defined in chapter chapter 3.2.with 25°C test cell temperature and 50% humidity

Figure 45 shows results for settings with 21°C in the cabin. Unfortunately the results were obtained at different settings of the blower and also the positions of the thermocouples were not identical in all tests. This explains a part of the differences found in the results. The application of all correction factors suggested in chapter 6 improves the repeatability of the test results significantly.

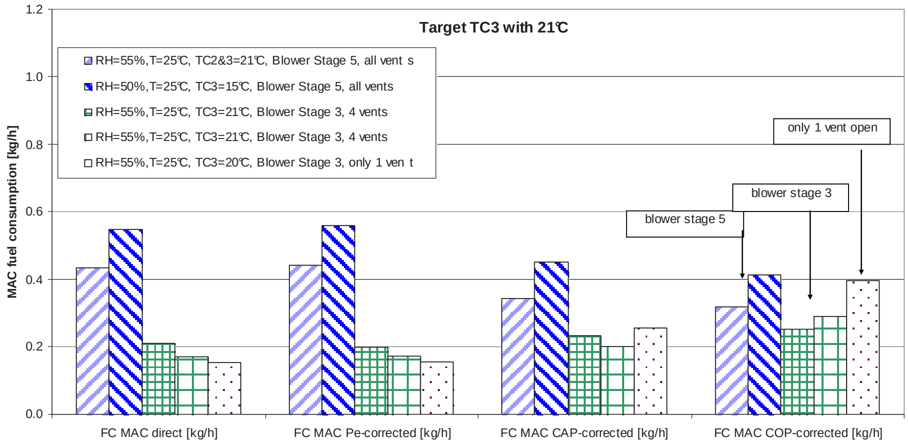


Figure 45: Results from the MAC test cycle at the diesel EURO 5 car at TUG with setting of the cabin temperature T_{C3} to 21°C (note, none of the tests followed the finally suggested procedure exactly!)

The results achieved with the setting of the vent outlet temperature to 15°C are shown in Figure 46. These tests were driven at 40% RH. Additionally the test vehicle showed differences in the vent outlet temperatures between the vents of more than 4°C. Thus all vents with exception of the one with the lowest outlet temperature were closed in these tests. The reason for the different vent outlet temperatures was not clarified yet. Again the application of the correction factors improves the repeatability results significantly.

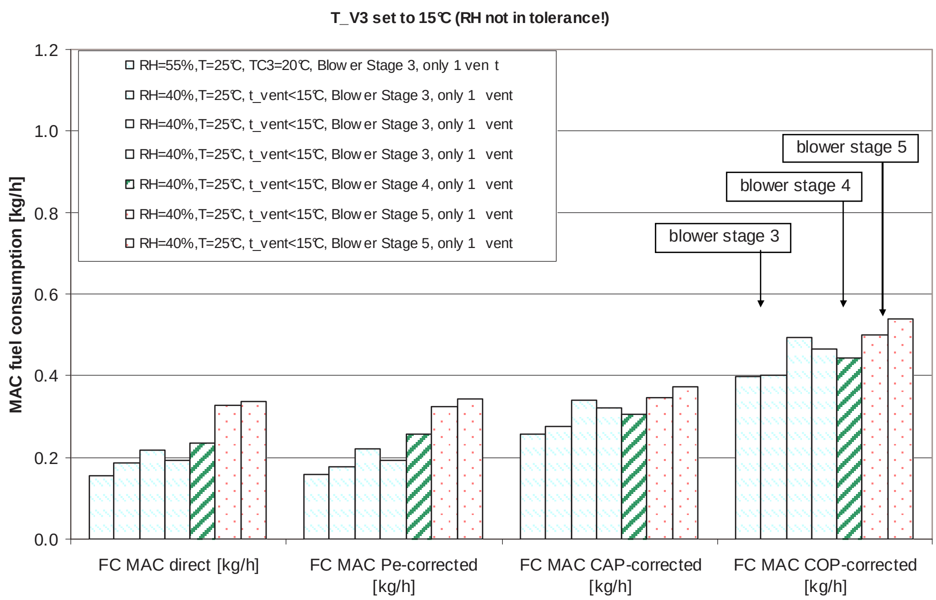


Figure 46: Results from the MAC test cycle at the diesel EURO 5 car at TUG with settings of the vent outlet temperature to 15°C (note, none of the tests followed the finally suggested procedure exactly!)

It was planned to test the reproducibility of the measurements at TUG with the same vehicle and the same settings at KTI. Unfortunately the settings used at KTI did not at all match the defined settings. The driving resistance values were applied according to the default values in EEC 70/220 at KTI while the coast down results were used at TUG. The cabin temperature was set to 16°C at KTI while 21°C were used at TUG.

Beside three basic MAC tests also sensitivity tests for a higher test cell temperature and for an electrical heater in the vehicle were performed at KTI (Figure 47).
The directly measured additional MAC fuel consumption was corrected according to the procedure described in chapter 3.4 to 21°C cabin temperature and test cell conditions of 25°C and 50% humidity. The correction method results in an improved repeatability with 13% standard deviation against the average test result (17% without COP correction). Excluding the first test from the analysis the standard deviation after COP-correction is 6%.

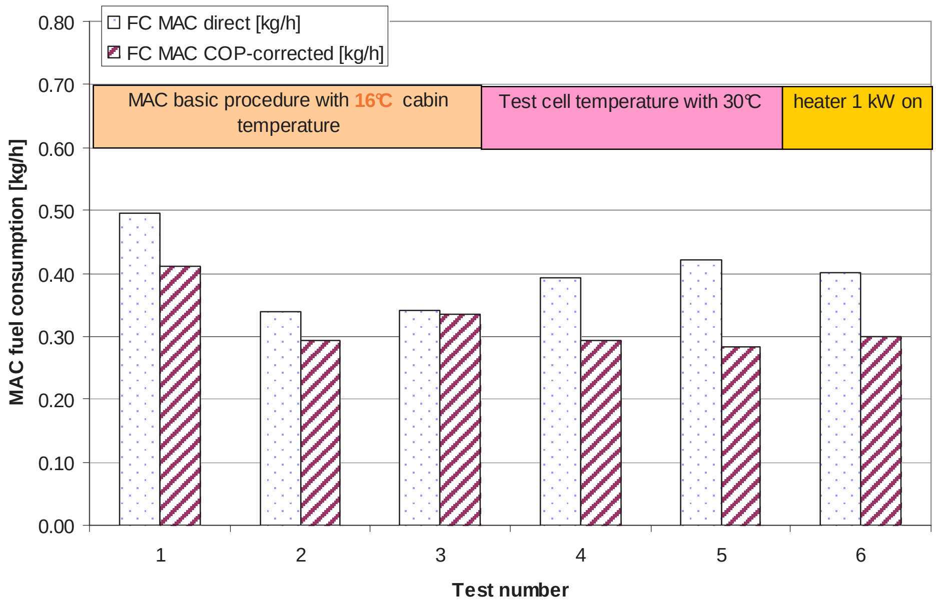


Figure 47: Results from the MAC test cycle at the diesel EURO 5 car at KTI with settings of the vehicle cabin temperature to 16°C (note, none of the tests followed the finally suggested procedure exactly!)

The test results at LAT for the BMW 316i are shown in Figure 48. At LAT the test cell temperature exceeded the 26.5°C at all tests. The humidity was lower then 45% in 17 of the 18 tests.
The tests following the option with setting TC3 = 21°C, no heater and a mass flow > 230 kg/h (Stage 4 of the blower settings for this car) showed a good repeatability with a standard deviation of 7% after the application of all correction factors. Without the correction factors the standard deviation of the measured fuel consumption was 12% (Table 16). This shows that the correction factors improve the repeatability of the tests at the gasoline car too. Main difference for the correction factors could be a slightly different incremental engine fuel efficiency. However, the Excel tool for the MAC simulation was used for this car without adaptations against the diesel car data.
The variations of the blower stage were also corrected sufficiently with the MAC model, taking the uncertainties in the mass flow values into consideration. However, in the suggested version of the test procedure no correction for the mass flow is foreseen, since the manufacturers can set one blower stage to the target value of the test if necessary.
The influence of the heater in the tests was astonishing. With the 1 kW heater in the vehicle the additional fuel consumption from the MAC system did not change against the basic settings without heater.

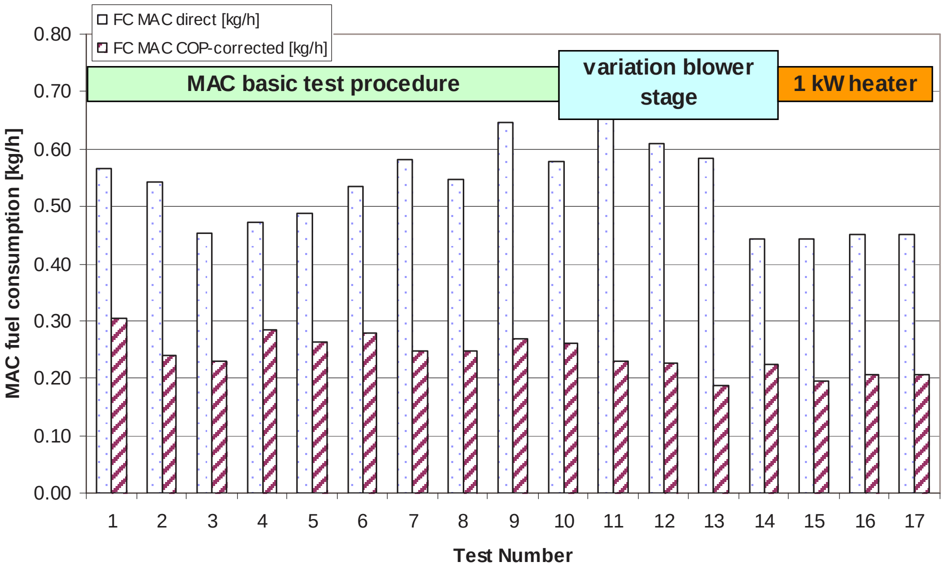


Figure 48: Results from the MAC test cycle at the gasoline EURO 5 estate at LAT with settings of the cabin temperature to 21°C (note, none of the tests met both tolerances for RH and T1!)

Table 16: results from the eight MAC tests at the gasoline EURO 5 estate car at LAT

Regular BMW 316i	T-a	RH	T-cabin	FC measured	FC COP-corrected
	[℃]	[%]	[℃]	[kg/h]	
Average	31.6	38%	19.9	0.54	0.26
Stabw	1.3	2%	0.8	0.06	0.02
% stabw	4%	5%	4%	12%	7%
Max	33.5	41%	21.0	0.65	0.28
Min	29.6	35%	19.0	0.45	0.23

Figure 53 and Table 17 summarise the results for the BMW 116i tested at LAT. For this car the standard deviation of the measured fuel consumption was high (32%) and did not improve when the correction factors were applied. During the tests of the BMW 116i the variation of the humidity was higher than during the tests with the 316i and the average test cell temperature was also higher during the tests of the 116i (36°C!). Since the temperature and also the humidity were far out of the tolerance defined in chapter 3 the poor behaviour of the correction factors could be explained by the increased amount of condensing water. Since the condensing mass of water is depending on the air temperature at the evaporator (which is unknown), a reasonable uncertainty is introduced in the MAC simulation tool at high test cell temperatures if the test cell air is not very dry. No efforts were undertaken to adapt the MAC simulation tool to the temperature ranges which occurred in the tests of the BMW 116i.
The problems with the test series at the 116i indicate that a tolerance for T_1 and RH as defined in chapter 3 is reasonable to improve the repeatability of test results.

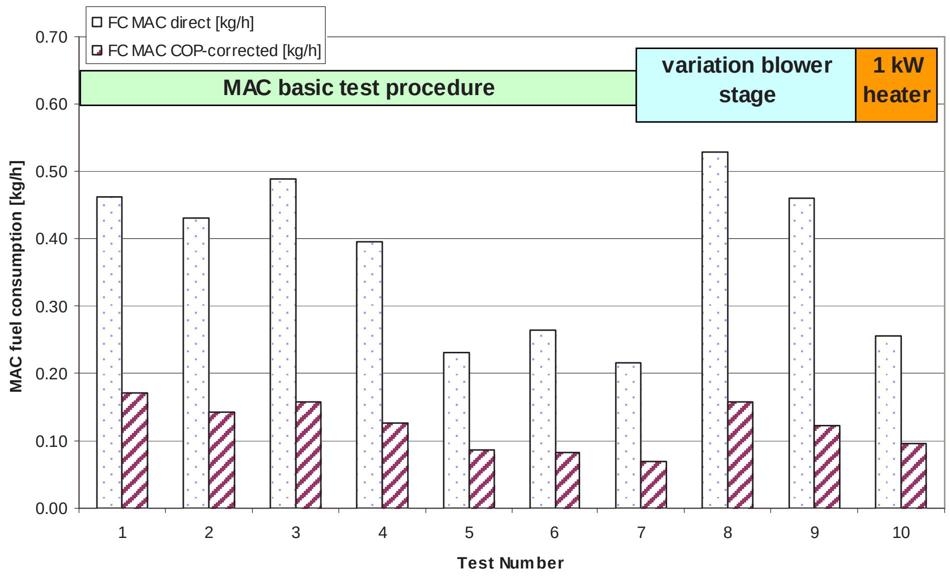


Figure 49: Results from the MAC test cycle at the gasoline EURO 5 compact car at LAT with settings of the cabin temperature to 21°C (note, none of the tests met both tolerances for RH and T1!)

Table 17: results from the eight MAC tests at the gasoline EURO 5 compact car at LAT

Regular BMW 116i	T-a	RH	T-cabin	FC measured	FC COP-corrected
	[°C]	[%]	[°C]	[kg/h]	
Average	35.7	38%	20.3	0.35	0.12
stabw	1.4	5%	1.0	0.12	0.04
% stabw	4%	14%	5%	32%	34%
Max	38.4	47%	21.4	0.49	0.17
Min	34.4	30%	18.3	0.22	0.07

7.3 Influence of the engine technology and vehicle size

Comparing the results from the BMW 318d, 316i and 116i the influence of the engine type (diesel and gasoline) and of the vehicle size (316 versus 116) can be analysed. Unfortunately the ambient conditions were extreme hot during the tests at LAT so no “valid” MAC test is available for the gasoline cars. Despite of exceeding the tolerances for RH and T_1 the trends of the test results meet the expectations (Figure 50, Table 18). The difference between the diesel engine and the gasoline engine may be within the accuracy of the tests (when the tolerances are clearly exceeded). In general the higher engine efficiency of a diesel engine should result in lower MAC fuel consumption if all other parameters are similar. However, the additional load of the MAC improves the fuel efficiency of the gasoline engine to a larger extent than for the diesel engine (steeper gradients of the efficiency at low engine loads for the gasoline engine). This improved efficiency affects the basic fuel consumption to overcome the driving resistances of the vehicle also, thus the effect of the combination of these two effects may lead to less additional MAC fuel consumption for a gasoline engine if the MAC adds a high load as happened at the high test cell temperatures at LAT. As a preliminary conclusion we can suggest, that the correction factors do not need to distinguish between gasoline and diesel engines.

The lower fuel consumption found for the compact car is reasonable, however, a reduction of more that 50% against the estate car seems to be quite high. The uncertainties at the tests of the 316i were discussed before and may be the reason for this result.

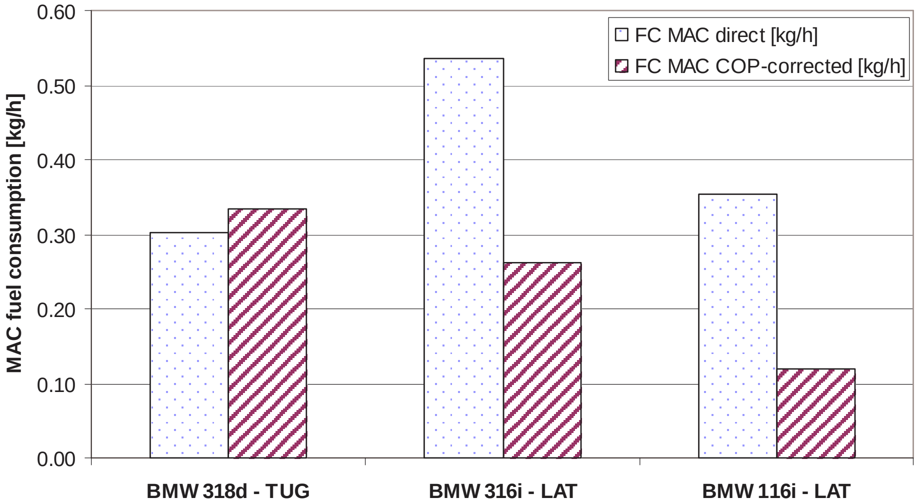


Figure 50: Results from the MAC test cycle at the estate car (diesel and gasoline from same model) and the compact car (gasoline from same make as the estate) (note, LAT tests did not meet the tolerances for RH and T_1)

Table 18: Results from the MAC tests at the diesel and gasoline EURO 5 cars (all values “COP-corrected” to 21°C cabin temperature with 25° test cell temperature at 50% RH)

	T-a	RH	T-cabin	FC measured	FC COP-corrected
	[°C]	[%]	[°C]	[kg/h]	
Estate diesel (TUG)	24.8	50%	20.7	0.30	0.33
Sedan gasoline (LAT)	31.6	38%	19.9	0.54	0.26
Compact car gasoline (LAT)	35.7	38%	20.2	0.35	0.12
Estate diesel (KTI)	25.6	39%	13.8	0.39	0.35

7.4 Reproducibility

The application of the suggested COP correction should give similar results for the same vehicle at different test facilities.
It was planned to test the reproducibility of the measurements at TUG with the same vehicle and the same settings at KTI. Unfortunately the settings used at KTI did not at all match the defined settings. The driving resistance values were applied according to the default values in EEC 70/220 at KTI while the coast down results were used at TUG. The cabin temperature was set to 16°C at KTI while 21°C were used at TUG.

Although the test conditions were very different at TUG and KTI the resulting MAC fuel consumption was rather similar after the application of the correction functions described in chapter 3.4.
The test results for the finally COP corrected MAC fuel consumption at TUG were on average 0.27 kg/h while the test results at KTI was on average 0.35 kg/h.

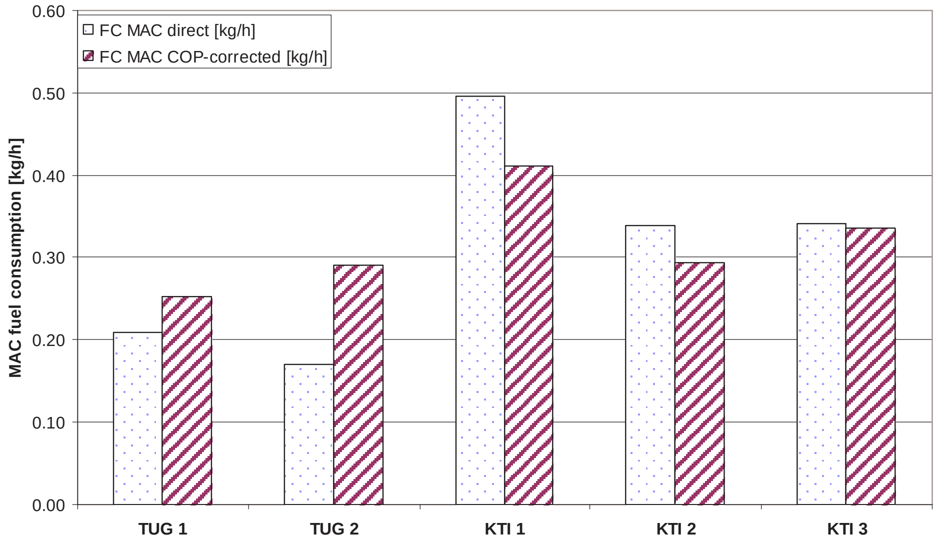


Figure 51: Results from the MAC test at the diesel EURO 5 car at KTI and at TUG (tests with blower stage 3, COP corrected to 21°C cabin temperature and test cell temperature of 25°C at 50% humidity). Note: the settings of the cabin temperature was 21°C at TUG tests and 16°C at KTI tests

A final validation of the reproducibility is suggested for a pilot phase of the MAC type approval tests. Labs participating in this pilot phase should have a test stand with full conditioning of temperature and humidity. It is suggested to include one or more of the current project partners in, JRC as official EC test facility and partners from industry as well as type approval authorities and technical services.

7.5 Validation of the correction factors

At TUG several sensitivity runs for variability in cabin temperature T_{C3}, in test cell temperature T_1 and in the relative humidity in the test cell (RH) were performed to validate the magnitude of the correction factors simulated with the MAC simulation tool described in chapter 6. For the validation the fuel consumption measured in the tests with variation in one parameter were corrected for the (slight) differences in the other two parameters. For example the differences in T_1 and RH in the left picture in Figure 52 were corrected with the corresponding correction factors while the difference in T_{C3} was not corrected to obtain the influence from T_{C3} without “noise” from variations in the other parameters. In general the agreement between the correction factors and the measured trends in the fuel consumption are good. It has to be pointed out, that the influences of RH, T_1 and T_{C3} are calculated via the Enthalpy of the air flow before and after the evaporator of the MAC system. The enthalpy of the air is a physical value which is not calibrated. Only the efficiency of the compressor and the power demand of the blower from the MAC system were calibrated to meet the absolute fuel consumption level measured at this vehicle for the MAC system. Figure 52 and Figure

53 show the average fuel consumption measured in repetitions with the same settings and the corresponding reciprocal value of the correction factor for the average of the test conditions. For the variation in the cabin temperature only one test per temperature was available with comparable settings of the blower and vents, thus no single test results are shown there.

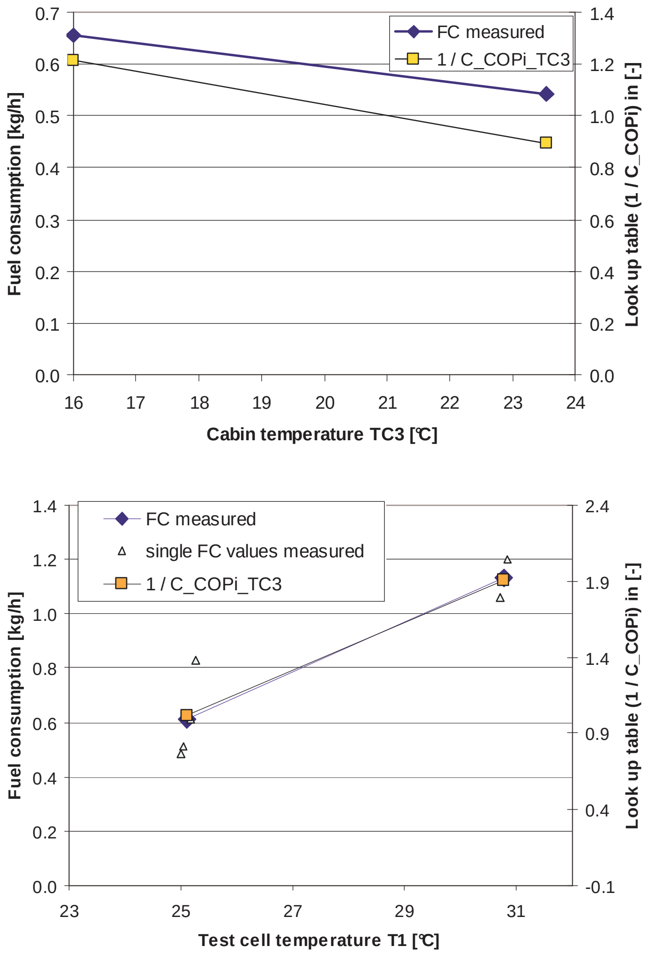


Figure 52: Variation in the measured fuel consumption of the MAC in the MAC test cycle compared to the corresponding correction factor interpolated from Table 7 (left picture variability of T_{C3}, right picture variability of T_1).

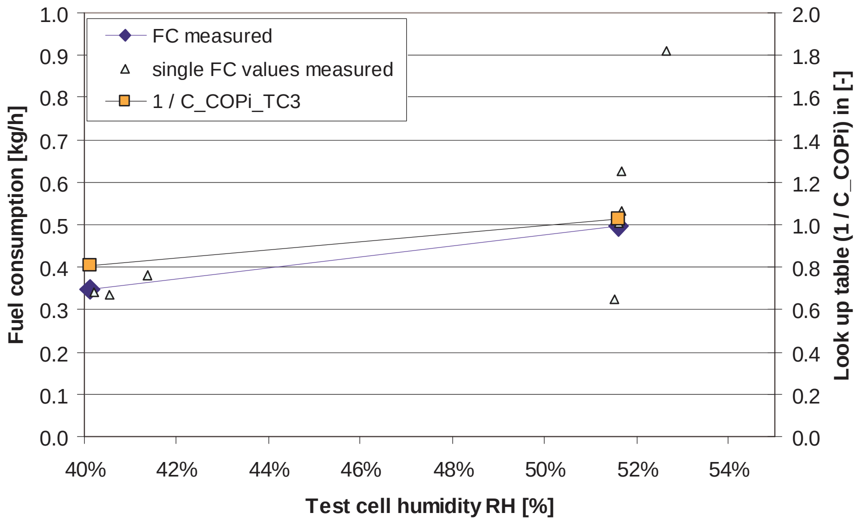


Figure 53: Variation in the measured fuel consumption of the MAC in the MAC test cycle compared to the corresponding correction factor interpolated from Table 7 for variations in RH

8 Summary and conclusions

Omit this chapter???

A Seasonal Performance Simulation for the annual fuel consumption of the MAC system has been conducted to identify the most important factors which affect the annual fuel consumption of the MAC system. Based on the results of these investigations the following test procedure is suggested.

Summary of the suggested test procedure

The procedure is based on a physical test procedure to be performed in an emission test laboratory on a chassis dynamometer. The procedure comprises a test cycle with a preconditioning phase, idling and two constant speed levels for the situation MAC off and MAC on. The steady state cycle allows the best repeatability as opposed to a transient driving cycle. This is very important because the additional MAC fuel consumption is gained by subtracting the fuel consumption measured at MAC-off from the fuel consumption measured at MAC-on, i.e. by subtracting two large numbers to get one small number. The ambient conditions for the test cell could be in the range of 25° with 50% relative humidity. Interior temperature setting for the MAC-on should be in the range of 21°C (comfort temperature according to DIN1946-3) corresponding to 15°C vent outlet temperature. The setting for the blower are suggested to be >230kg/h. All of these boundary conditions heavily influence the cooling demand and the energy consumption of the MAC system. Thus the defined values have to be met during the tests rather accurately. For deviations of the main parameters against the target values correction functions have been developed. The application of the test method with application of the correction functions resulted in a reasonable repeatability and reproducibility.

Since the number of tested vehicles was limited in this project and most of the test resources was needed to elaborate details of the test procedure only a few “valid” tests according to the finally suggested test procedure are available. Thus it is suggested to launch a pilot phase of the MAC type approval tests. Labs participating in this pilot phase should have a test stand with full conditioning of temperature and humidity. It is suggested to include one or more of the current project partners in, JRC as official EC test facility and partners from industry as well as type approval authorities and technical services.

9 Literature

draft

Andrè M.: Driving cycles derived from real-world in vehicle measurements for passenger cars and light duty vehicles. Principles, database and main results – Particular case of the ARTEMIS driving cycles, Bron 2001

Bharathan, D., Chaney, L., Farrington, R., Lustbader, J., Keyser, M., Rugh, J. (2007): *An Overview of Vehicle Test and Analysis from NREL's A/C Fuel Use Reduction Research.* Proceedings of 2007 Vehicle Thermal Management Systems Conference & Exhibition, Nottingham England, NREL

Contract (2009): Specific Contract No SI2.543231: "*Collection and evaluation of data and specification/development of test procedures in support of legislation on mobile air conditioning (MAC) efficiency and gear shift indicators (GSI)"* implementing „Framework Contract No. ENTR/05/18", European Commission, Enterprise and Industry Directorate - General

DIN1946-3: *Raumlufttechnik, Teil3: Klimatisierung von Personenkraftwagen und Lastkraftwagen*, Deutsche Norm, April 2003
GM (2007), GREEN-MAC-LCCP: Global Refrigerants Energy & Environmental Mobile Air Conditioning Life Cycle Climate Performance, © Copyright 2007 General Motor Corporation, http://www.epa.gov/cppd/mac/compare.htm, 01.04.2008

Grundstein A., Meentemeyer V., Dowd J. (2009): *Maximum vehicle cabin temperatures under different meteorological conditions.* Department of Geography University of Georgia 2009

Hausberger, S. (2003). Simulation of Real World Vehicle Exhaust Emissions (Bde. VKM-THD Mitteilungen; Heft/Volume 82 - ISBN 3-901351-74-4). (V. d. Graz, Hrsg.) Graz.

Hausberger, S. (2002). Update of the Emission Functions for Heavy Duty Vehicles in the Handbook Emission Factors for Road Traffic. Graz: Institute for Internal Combustion Engines and Thermodynamics.

Hausberger, S., Rexeis, M., Zallinger, M., & Luz, R. (2009). Emission Factors from the Model PHEM for the HBEFA Version 3. Graz: Report Nr. I-20/2009 Haus-Em 33/08/679.

Hessel S., Manz F.: Impact of Heat Transmission on MAC Systems; presentation from Saint-Gobain Sekurit; May 2009

Hill (2006): *Improved Mobile Air Conditioning Cooperative Research Program.* VDA Wintermeeting, Saalfelden 2006

Luz R., Hausberger S.: User Guide for the Model PHEM, Version 10; Institute for Internal Combustion Engines and Thermodynamics TU Graz, Graz, 2009

Meteonorm 6.0 (2008), *Climatological Databae,* METEOTEST, Bern, Switzerland

Pinkofsky L. (2006): *Typisierung der Ganglinien der Verkehrsstärke und ihre Eignung zur Modellierung der Verkehrsstärke*. Dissertation TU Braunschweig. Shaker Verlag. ISBN 3-8322-4859-5.

Rexeis, M. (2009). Ascertainment of Real World Emissions of Heavy Duty Vehicles. Dissertation, Institute for Internal Combustion Engines and Thermodynamics. Graz:

Rexeis, M., Hausberger, S., & Riemersma, I. (2005). Heavy duty vehicle emissions. In Report of WP 400 in ARTEMIS (Assessment and Reliability of Transport Emission Models and Inventory Systems). Graz: DGTREN Contract 1999-RD.10429.

Rugh, J., Chaney, L., Lustbader, J., Meyer, J., Rustagi, M., Olson, K., Kogler, R. (2007): *Reduction in Vehicle Temperatures and Fuel Use from Cabin Ventilation, Solar-Reflective Paint and a New Solar-Reflective Glazing*. Proceedings of 2007 SAE Congress, Detroit, MI. Paper # 2007-01-1194. Society of Automotive Engineers

Rugh, J. (2009): *Impact of Sungate EP on PHEV Performance. Final Report in Support of Technical Services, Golden Colorado, Agreement TSA-08-206. NREL*
SAE ARCRP (2002) - SAE Alternate Refrigerant Cooperative Research Project

Schmitzberger M. (2007): Emissionsverhalten von SUV's und Potential einer Hybridisierung; Diploma Thesis at the Institut for INternal Combustion Engines at the TU-Graz; 2007

Smokers et al., TNO 2006, Review and analysis of the reduction potential and costs of technological and other measures to reduce CO_2-emissions from passenger cars for EC contract SI2.408212, TNO report 06.OR.PT.040.2/RSM, October 31 2006.

Steininger N. (2009), European Commission: *Mobile Air-conditioning (MAC): Development of an Emission test procedure*. Presentation at Kick-off meeting for the development of a European MAC emission test procedure, Brussels, 03.12.2009

Vermeulen, R., 2005: *Development of a procedure for the determination of the additional fuel consumption of passenger cars due the use of mobile air conditioning equipment.*TNO report 05.OR.VM.014.1/RJV

Weilenmann M., Vasic A-M., Stettler P., Novak P.: Influence of Mobile Air-Conditioning on Vehicle Emissions and Fuel Consumption: A Model Approach for Modern Gasoline Cars Used in Europe Empa, Materials Science and Technology, Environ. Science Technology 2005, 39, 9601-9610

Weilenmann M., Alvarez R., Keller M.: Fuel Consumption and CO2/Pollutant Emissions of Mobile Air Conditioning at Fleet Level – New Data and Model Comparison; Empa, Materials Science and Technology, Environ. Science Technology 2010

Zallinger, M. (2010). Mikroskopische Simulation der Emissionen von Personenkraftfahrzeugen. Graz: Dissertation, Institute for Internal Combustion Engines and Thermodynamics, Graz University of Technology.

Zallinger, M., Hausberger, S., Ajtay, D., & Weilenmann, M. (2005). Instantaneous emission modeling applications. In Final Report of WP 300 in ARTEMIS (Assessment and Reliability of Transport Emission Models and Inventory Systems). Graz: DGTREN Contract 1999-RD.10429.

A “Evaluation results from ACEA”

This annex gives the unabridged content of a report written by Bruno Rose from PSA Peugeot Citroën and from Enrique PERAL-ANTUNEZ from Renault on:

Evaluation of a “simple test” method to measure the fuel consumption of MAC system. Assessment of repeatability, comparison of test benches and operators, MAC technologies and engine types.

A.1 Annex-Introduction

In Europe and in the US, the authorities are working with stakeholders to set up a regulation of the MAC system influence on the overall vehicle fuel consumption.
PSA and Renault have been working together on MAC fuel consumption for engineering purpose since the late 90s. From this experience, the two companies have clearly identified that standard OEMs testing procedures are not applicable for this type approval purpose.
ACEA has developed a simplified approach in order to adapt OEMs knowledge for this purpose, taking into account the technical and economical constraints of type approval activities.
This report summarizes all tests done by PSA and Renault on their benches in order to assess the reliability of the ACEA test procedure.

A.2 Annex-Summary of test conditions

Results presented in this report have been measured using the test conditions presented in this chapter which corresponds to the latest draft of the ACEA test proposal. Nevertheless, this report is NOT the official ACEA test procedure for MAC fuel consumption measurement. The values given in this section are only for information and for better understanding of the results hereafter.

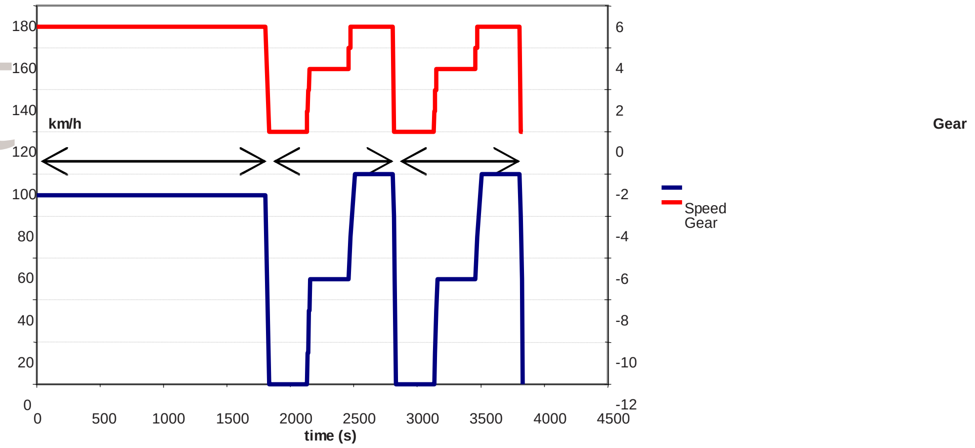
A PSA & Renault analysis of the pros and cons of these test conditions is provided in annex.

A.3 Driving cycle

The driving cycle consists in 3 phases :

- A 30mn engine heat-up phase with the AC system running. This phase is also used to reach stabilized thermal conditions in the cabin and the HVAC system according to the specification of section 1.3. Driving setting for this engine heat-up phase are 90kph @ maximum gear.
- A 15mn phase AC ON with 3 stabilized speeds (Idle, 50kph @ maximum gear minus two, 100kph @ maximum gear)
- A 15mn phase AC OFF with 3 stabilized speeds (Idle, 50kph, 100kph)

It is done once, so the total test duration is 1hour.



The fuel consumption measurement is done on a 5min time range on every single speed-phase, in steady-state conditions only.

The wheel resistance (Dyno torque) is set to a value corresponding to the proper driving speed for each vehicle (flat road).

AC OFF condition shall be understood as *all AC related commands set to OFF mode*. (including blower and control panel display)

A.4 Ambient conditions

The temperature in the test chamber is controlled at 25°C +/- 2°C
The relative humidity in the test chamber is controlled at 40% +/- 5%
The test is done without solar load.
The wind at the front nozzle is controlled at the same speed as the wheels during the driving cycle, except at idle condition where a minimum wind speed of 5kph is required for the controllability of the bench.

A.5 AC commands settings in AC ON Mode

The blower command is set to the first step providing an air flow in the vehicle above or equal to 230kg/h.
All vehicle tested have been preliminary characterized for airflow according to in-house PSA-Renault test procedures with the following settings:
- Full vent
- Maximum cold
- Outside air

The mixing flap or the temperature setting command (depending on automatic or manual versions) is set to a level leading to a maximum T° of 15°C for all the vent outlets. This is tuned during heat-up phase and checked continuously during all the AC ON measurement phase. The test is not accepted if this criteria is not reached.

The air distribution is set to full vent mode (face mode).

The recirculation flap is set in outside air mode for manual versions, and moving according to OEM strategy for automatic versions. No forced recirculation mode is allowed.

A.6 List of vehicles tested

The following vehicles have been tested. Two groups are presented: diesel and gasoline engines. The vehicles in each group are sorted here according to the installed MAC technology, by **increasing order of expected fuel consumption**.

No name is mentioned, but the list below includes several brands from European and non-European car manufacturers. All these vehicles are sold on the European market.

Vehicle	Vehicle type	Engine	A/C technology
Vehicle D1	*B-Segment*	**1.6L Diesel**	Automatic, **External** control compressor (120cc) with clutch, TXV
Vehicle D2	*D-Segment luxury*	**2.0L Diesel**	Automatic, **External** control compressor (140cc) clutchless, TXV
Vehicle G1	*B-Segment*	**1.4L Gasoline**	Manual, **External** control compressor (120cc) with clutch, TXV
Vehicle G2	*C-Segment*	**2.0L Gasoline**	Automatic, **External** control compressor (140cc) clutchless, TXV
Vehicle G3	*B-Segment*	**1.4L Gasoline**	Manual, **Internal** control compressor (120cc) with clutch, TXV
Vehicle G4	*B-Segment*	**1.3L Gasoline**	Manual, **Fixed** displacement compressor (Scroll 60cc), TXV

A.7 Test results

A.7.1 Results processing

Calculation of MAC over fuel consumption
The MAC over fuel consumption is calculated by making the difference between the AC ON and the AC OFF overall fuel consumption measurement on each phase.

The value is presented in l/h (Idle condition) and l/100km (other phases and cycle average) as the *absolute MAC over fuel consumption*. The *relative MAC over fuel consumption* is given in %, and is calculated by dividing the first result by the overall vehicle fuel consumption measured on each phase in AC OFF mode.

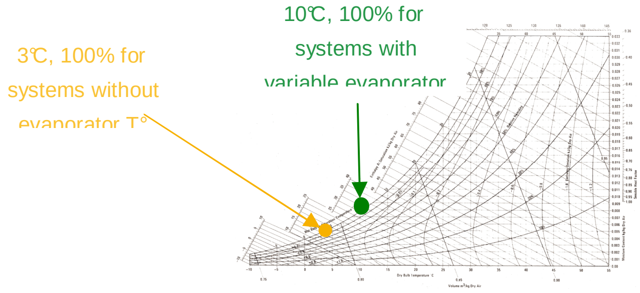
Calculation of a cycle average value for fuel consumption
The average fuel consumption for a real customer usage at the test ambient conditions (25°C, 40%) is estimated from the single measurements of each phase (Idle, 50kph, 100kph) by mean of a weighted average formula giving a result in l/100km.
The weight of each phase is chosen as follows to represent a standard customer usage :
- Idle : 23%
- 50kph : 46%
- 100kph : 31%

Taking fcI in l/h, fc50 and fc100 as respective fuel consumption measurement for each phase, the formula used is:

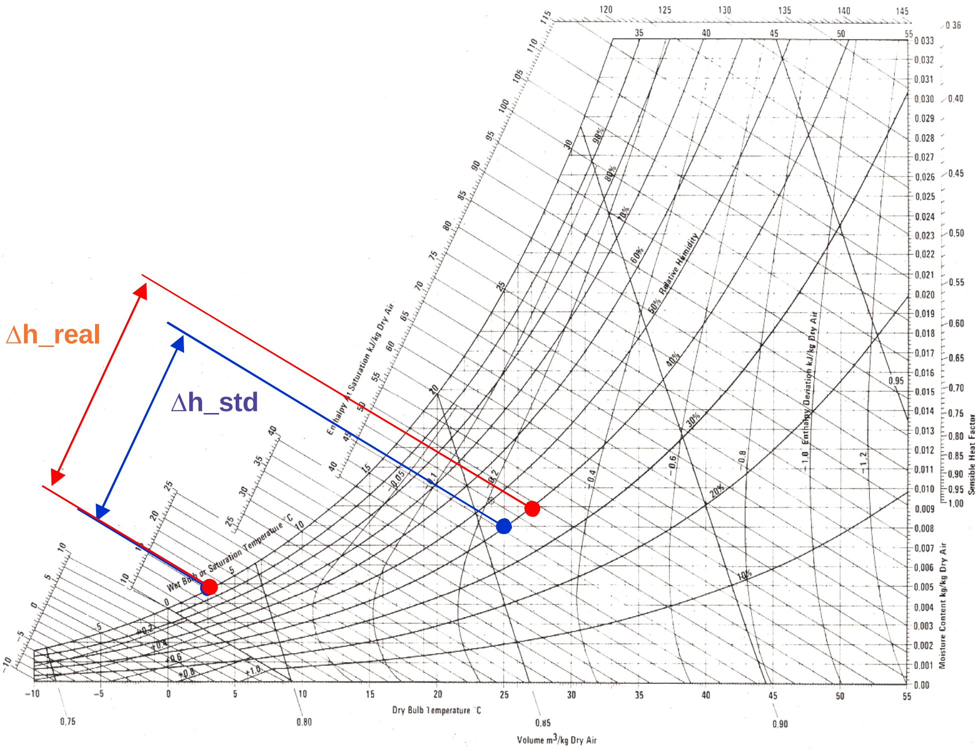
$$fc\_avg = \frac{0.23 \times fcI + 0.46 \times fc50 \times \frac{50}{100} + 0.31 \times fc100 \times \frac{100}{100}}{0.46 \times 50 + 0.31 \times 100} \times 100$$

Correction method for ambient conditions variations
In order to reduce the deviation of the results due to fluctuation of the ambient Temperature and Humidity during the test, a correction factor has been applied to all results. The method to calculate the correction factor is based on the air enthalpy variation through the evaporator, and is presented below:

- Define evaporator outlet standard conditions as shown in the next graph

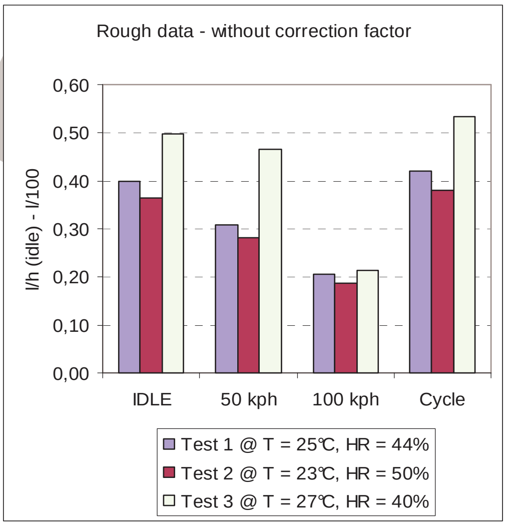


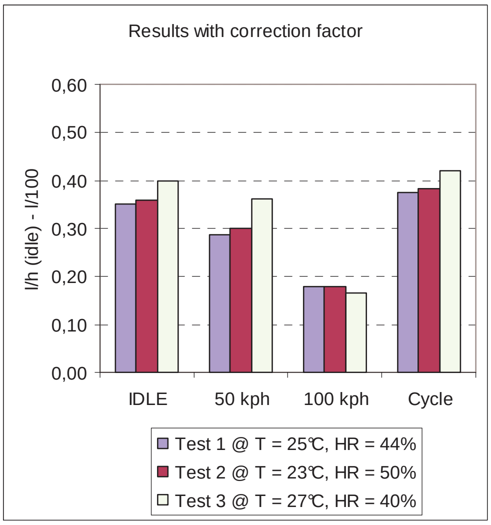
- Calculate standard enthalpy variation through evaporator with 25°C, 40%
- Calculate real enthalpy variation through evaporator with actual T°, RH on each phase (Idle, 50kph, 100kph)
- Determine correction factor : K_corr = **Δh_std** / **Δh_real**



Finally, the fuel consumption measurement of each phase is multiplied by the correction factor.

In order to measure the effect, and evaluate the potential benefit from this correction method, we repeated three times the same test on the same car, with intentional deviations on the temperature and humidity in the test chamber. The two graphs below show the outcome of the tests and correction factor:



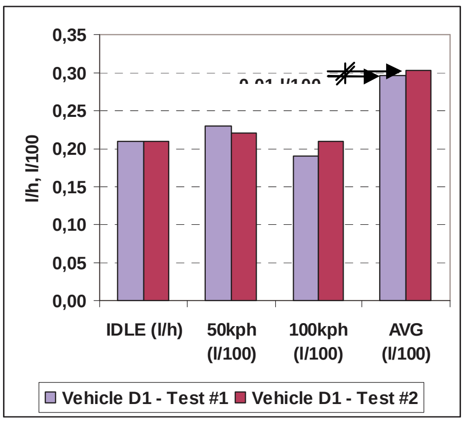


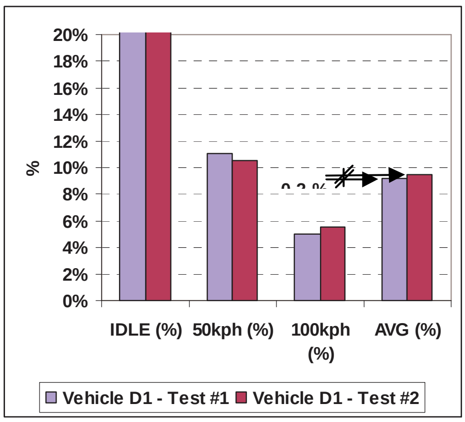
After multiplying by the correction factor, the maximum deviation is less than 0.05l/100km. In the following, the same approach is applied to all results obtained.

A.8 Assessment of repeatability

A.8.1 Repeatability on the same test bench

Vehicle D1 has been measured twice on the same test bench, with the same operators.



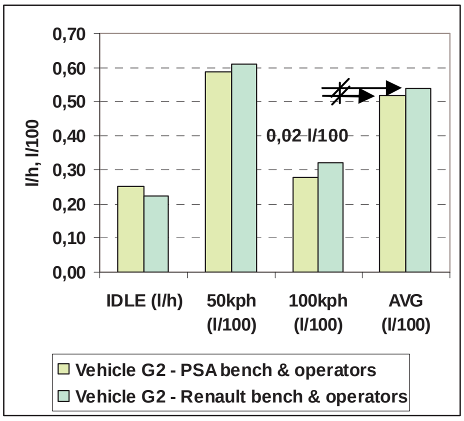


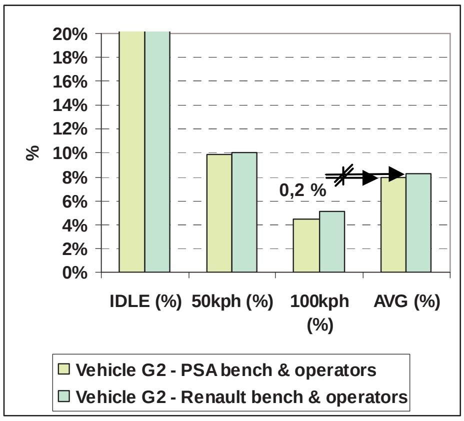
Under these conditions, the maximum deviation between the two tests is 0.01 l/100km and 0.2% of AC OFF overall fuel consumption.

A.8.2 Repeatability on the different test benches & operators

Vehicle G2 has been measured:

- by PSA personnel in PSA test facility
- by Renault personnel in Renault test facility





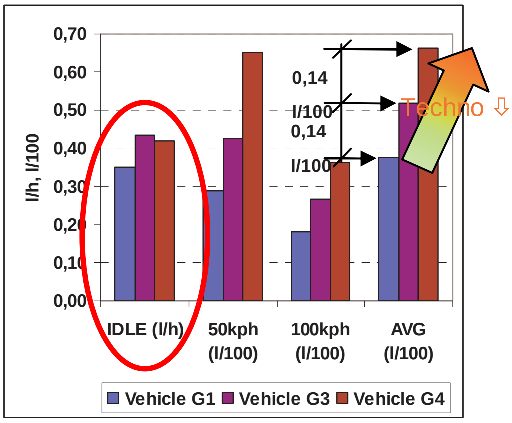
Under these conditions, the maximum deviation between the two tests is 0.02 l/100km and 0.2% of AC OFF overall fuel consumption.

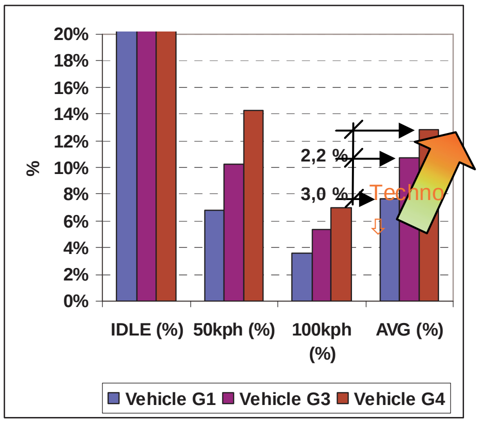
A.9 Comparison of MAC technologies

Vehicles G1, G3 and G4 have been chosen to measure the influence of MAC technology because their engines are quite similar and they belong to the same segment.

- G1 : External control variable type compressor
- G3 : Internal control variable type compressor
- G4 : Fixed type scroll compressor

The effect of each technological step can be seen on the next graph:



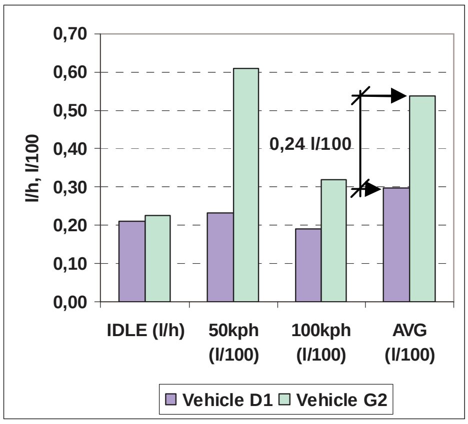


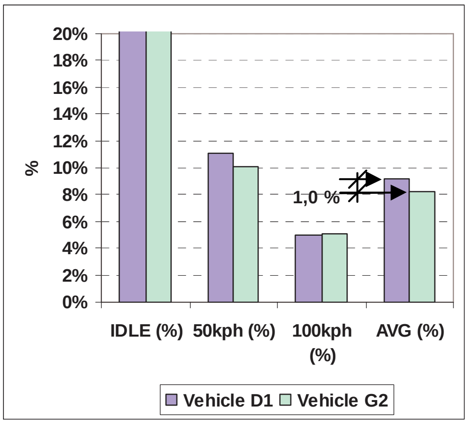
In this case (comparable vehicles and engines), the influence of MAC system appears clearly in the absolute figures as well as in the relative ones. The effect is in the order of 0.14l/100 and 2-3% for each technological step.

One important observation can be made on the Idle condition test result, which cannot show alone any significant difference between the three definitions.

A.10 Influence of engine types

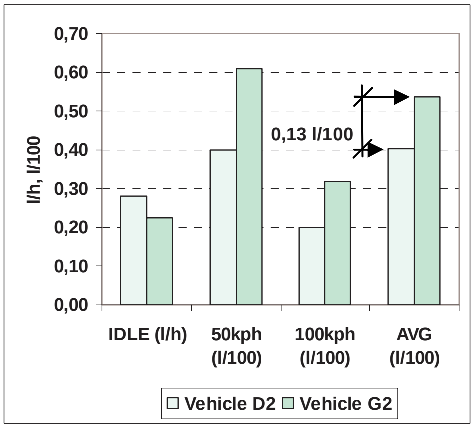
Measurement of D1 and G2 vehicles can be used to compare the influence of very different engines with very similar MAC systems from a technological standpoint.

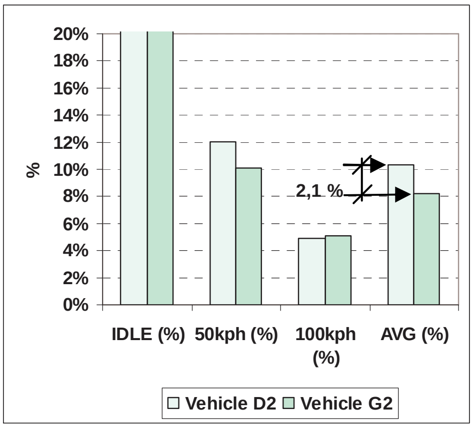




The results above show that, even if the gap on the absolute figures is quite important (G2 is almost the double of D1), the relative figures are pretty close. This demonstrates the engine efficiency and fuel type is in the first order of magnitude in the results we get from the measurements.
On the other hand, the results in % give a good image of the efficiency of the MAC system itself, and the quality of its integration to the whole car.

Comparison of D2 and G2 vehicles gives quite the same kind of information, but in this case, vehicles and engines are even closer in size (both are 2.0l engines but different types) and the MAC system is also equivalent in terms of maximum performance.





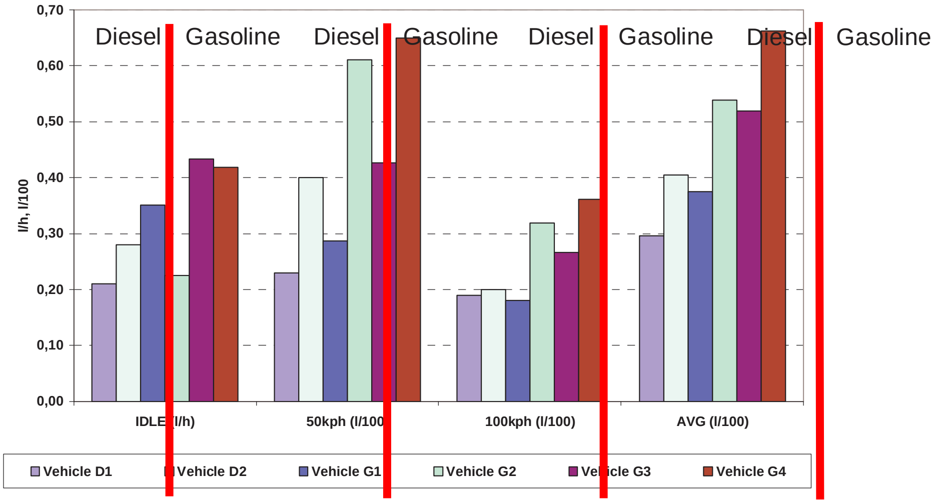
In this case, we get a smaller gap for the absolute figures because MAC performance is closer, but the gap for relative figure becomes larger because the relatively low fuel consumption of the vehicle in AC OFF mode amplifies the number in %.

These results show that even if the test procedure itself is reliable, the way we should communicate the result is still an open point.

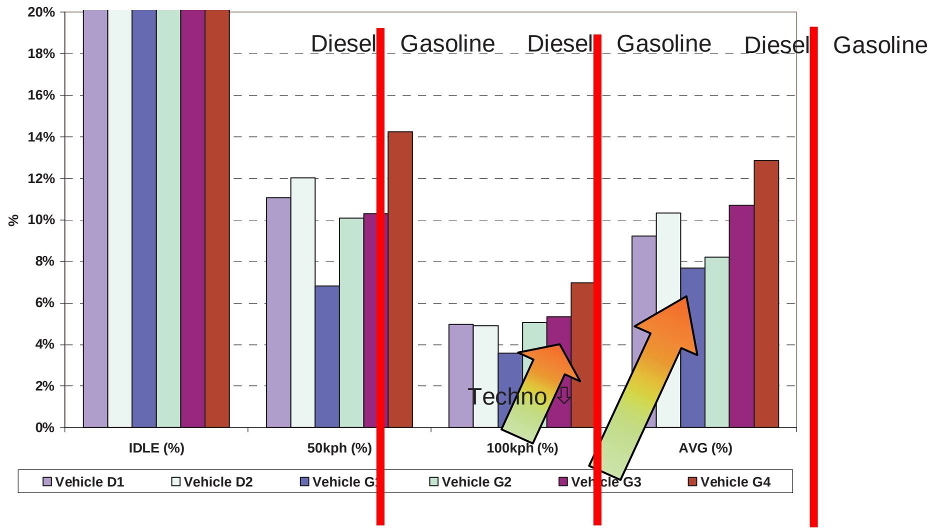
draft

A.11 Global picture

In this section, we look at the global picture presented with absolute and relative figures. The objective is to understand how we cold compare different types of vehicles and engines and carrying different MAC systems on a fair basis.



Looking at the absolute values we can see a wide spread of results (from 0.3 l/100km to 0.66 l/100km). This is in favor of diesel engines obviously, but it hides partly the direct effect of MAC technologies, and more generally, of the efficiency of the MAC system itself.



The relative figures can better show the effect of the MAC system itself. The trend in each engine category taken separately is consistent with the MAC system installed.

Nevertheless, this introduce a small disadvantage for diesel engines whose figures are amplified by the relatively good vehicle overall fuel consumption in AC OFF mode.

A.12 Conclusions

According to PSA and Renault testing engineers, measurement consistency is very good between both test benches. We get very similar results (less than 0,02 l/100km and less than 0,2 % accuracy), even knowing that :

- Test benches are different
- Measurements facilities are different
- Car drivers are different

According to those who made the measurements, the test procedure is simple, and then probably acceptable by Technical Type Approval services.

The proposed correction method can compensate the deviation of measurements due to the fluctuation of Temperature and RH in the test facility. This correction method is easily applicable by type approval authorities.

The measurement procedure is accurate enough to represent the differences between MAC technologies.

Even in the test procedure itself is reliable, the question of the final relevant figure to communicate remains open:

If the result is given in l/100km (absolute over consumption) MAC fuel consumption measurement will be directly related to engine efficiency, and the influence of MAC technology itself might be hidden by engine influence.
→ *Real customer consumption will be assessed (l/100km or gCO2/km), but* ***AC labeling may be misunderstood*** *in this case.*

If the result is given in % (relative over consumption) very similar AC technologies give very similar results, but it can disadvantage diesel vehicles where the A/C off fuel consumption is low. The result is more related to AC technology and AC good engineering, especially if Diesel and Gasoline are considered separately.
→ *This would be the* ***best approach if the purpose is MAC efficiency evaluation****,*
but ***AC labeling scale should be different between Diesel and Gasoline cars.***

A.13 ANNEX to the Annex

A.13.1 PSA & Renault analysis on the most suitable test conditions and parameters for a MAC efficiency homologation test.

The information presented in this annex section is the result of the PSA & Renault analysis about the best test conditions and parameters to be used in the test procedure in order to deal with both regulatory requirements and real MAC efficiency evaluation.
The following argumentation have been discussed and agreed within ACEA in a general basis, nevertheless, final values for some of the proposed test conditions or parameters are not exactly established yet.

A.13.2 Annex 1 : Driving cycle, steady state vs dynamic cycle.

Repeatability is a key issue for MAC fuel consumption testing because the value to measure is relatively low compared to the overall vehicle fuel consumption. The measurement during transient phases (acceleration) is influenced by the way the operator drives the car

Measurement on a steady-state cycle can be drivers-independent (automatic cruise control can even be used if the vehicle is equipped)

MAC systems have very few moving parts, which make inertial forces negligible compared to the steady-state compressor torque, which means that there is a low influence of vehicle dynamics on MAC efficiency.

For all the above reasons we consider that **steady-state cycle is the best option for a good repeatability and no benefit exists in testing MAC efficiency on a dynamic cycle.**

A.13.3 Annex 2: Driving speeds.

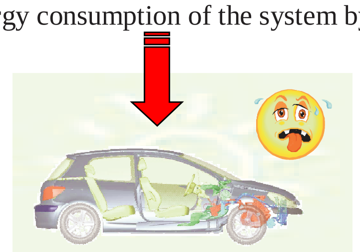
Most representative customer driving conditions should be included in the test cycle. It is needed to analyze the MAC behavior in running conditions and not only in idle phase, to be able to assess technology improvements & good engineering of the front end module, as well as control strategy.

In addition, some of these improvements have higher influence at driving speeds higher than 65kph.

For all the above reasons **we propose idle, 50kph and 100kph as driving speeds** during the homologation test, corresponding to standard speeds for urban and highway driving in customer profile, which adds no significant additional test complexity.

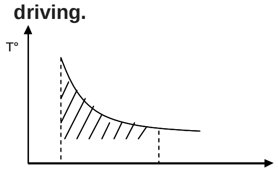
A.13.4 Annex 3 : Solar load influence in MAC fuel consumption.

Additional heat entering the cabin will require more effort to cool the interior of the car. This can influence the energy consumption of the system by 3 different mechanisms:



1. Cooldown energy

After soaking, the heat stored inside the cabin and cabin material must be rejected. This happens only during the first minutes of driving.



2. Blower level and Vents T°

On an automatic HVAC, sunload is measured by a sensor on the dashboard and blower level is increased and T° at the vents decreased when sunload is high.

On manual versions, the occupants usually act the same way manually, through the control panel.

3. Increased evaporator load

If the temperature of the air inside the cabin is higher, in recirc mode, the load on the evaporator will be higher.

This effect does not exist when the comfort level is kept constant by effect n°2 and/or when the system is running in outside air mode.

An explanation of the influence of each one of these effects on MAC fuel consumption is given as follows:

The influence of **cooldown energy** on annual fuel consumption is relatively low, as cooldown phases represent less than 5% of most customer driving usage. In addition, repeatability is usually a problem when dealing with soaking (even for thermal comfort testing, soaking is a source of problems)

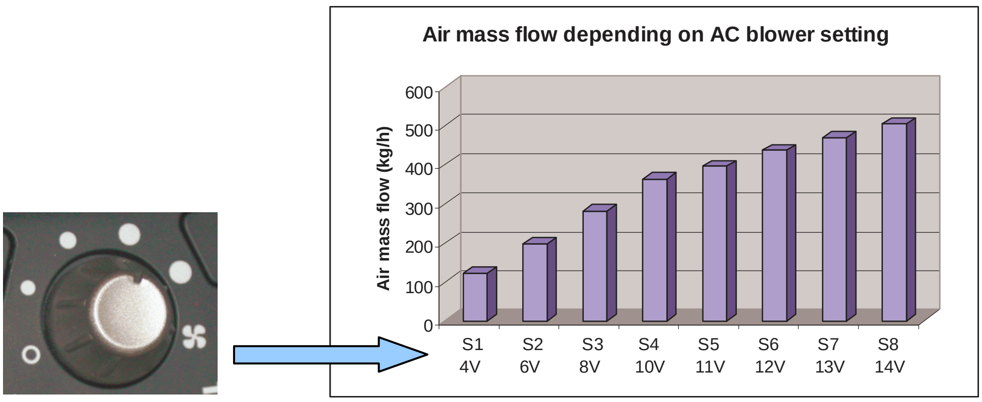
For these reasons, we recommend to measure fuel consumption in stabilized conditions and avoid any kind of soaking
The effect on **blower level and Vents T°** is the main one of sunload on the fuel consumption under standard customer usage conditions
The proposed blower level at 230kg/h and vents T° at 15°C corresponds to standard ECU calibrations of most AUTO mode with 25°C ambient and 700W sunload.
Then, even if the test is done without sun lamps, most of influence of the sunload can be taken into account if blower level and vents T° are chosen accordingly.
The increased evaporator load in recirculation mode is not representative in these type approval test conditions, since recirculation mode is not usually used below 30°C ambient temperature by customers.
For a stabilized comfort level inside the cabin, recirculation mode even has no influence

Any other artificial alternative to increase the cabin temperature (like a heater on a seat for instance) will never be representative of real sunload under standard customer usage, and will not influence fuel consumption on most vehicles. It neither deals with glazing effect.

For all the above reasons **we recommend to avoid any kind of solar load and any kind of artificial increase of cabin temperature** in the MAC efficiency homologation test.

Note: Glazing effect could be considered, if necessary, by an external calculation.

A.13.5 Annex 4 : MAC airflow into the cabin.
Fixing a constant mass flow should not be considered, as an accurate value is impossible to be assured. We rather propose to establish a minimum value to be respected during the whole test, related to the “x” positions of the blower setting in the control panel for each car. The air mass flow corresponding to those positions are well known by carmakers.
We propose to include the report of the vehicle mass flow characterization, done by the OEMs, into the MAC efficiency homologation test report, for authorities verification.
An example of the air mass flow (kg/h) blown into the cabin depending on the blower setting (blower speed or blower voltage) fixed on the control panel is shown in the figure below:



As already explain in the annex 3, the air flows proposed in the test method (230kg/h at 15°C) correspond to a situation with solar load of 700W, which represents the EU summer average conditions.

A.13.6 How to measure cooling capacity. Vent vs cabin temperatures.

For homologation purpose, measuring the cabin temperature (at driver's head level, for example) should not be considered. In this case the measured numbers are completely influenced by the exact position and orientation of the temperatures sensors in the cabin, and the orientation of the vent outlet flaps.

In addition, the temperature distribution inside the cabin can be very different from one car to another, which means that many sensors will be needed to represent average T° accurately.

Finally, a correct representation of cabin thermal behavior (T° stratification, heat radiation,…) requires a full size climatic windtunnel, because it is not feasible in existing type approval facilities.

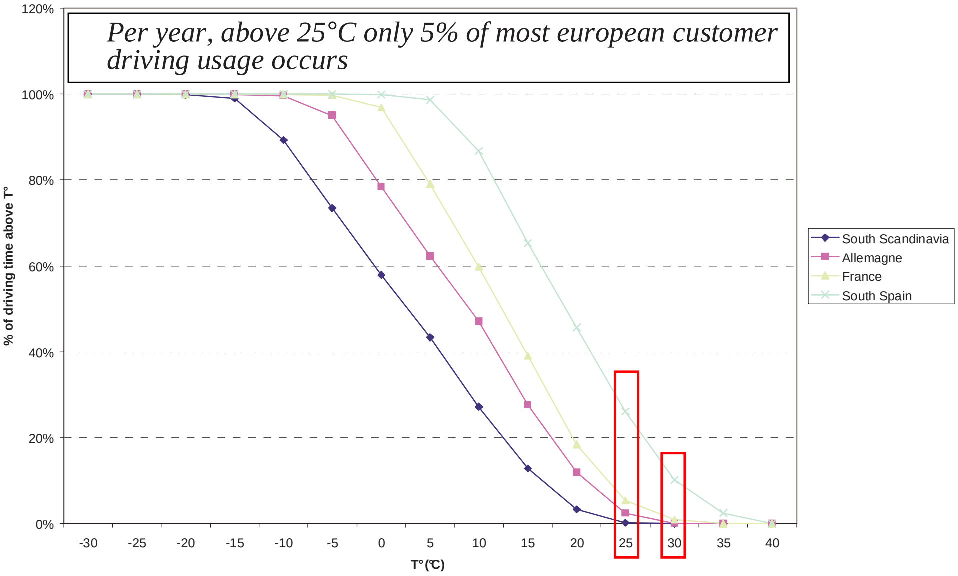
In the other hand, we propose to control the cooling performance of the MAC system by measuring directly the temperature of the air in the **vent outlets (15°C as maximum value).**

Our correlation shows that 15°C in vent outlets correspond to a setting signal of around 20,5 – 21°C on the control panel for most AUTO A/C systems in the proposed test conditions.

Note: Most European cars have ΔTemperature between vents < 4°C

A.13.7 Ambient temperature & humidity.

We have defined 25°C as ambient temperature because it is representative of most of European summer conditions.

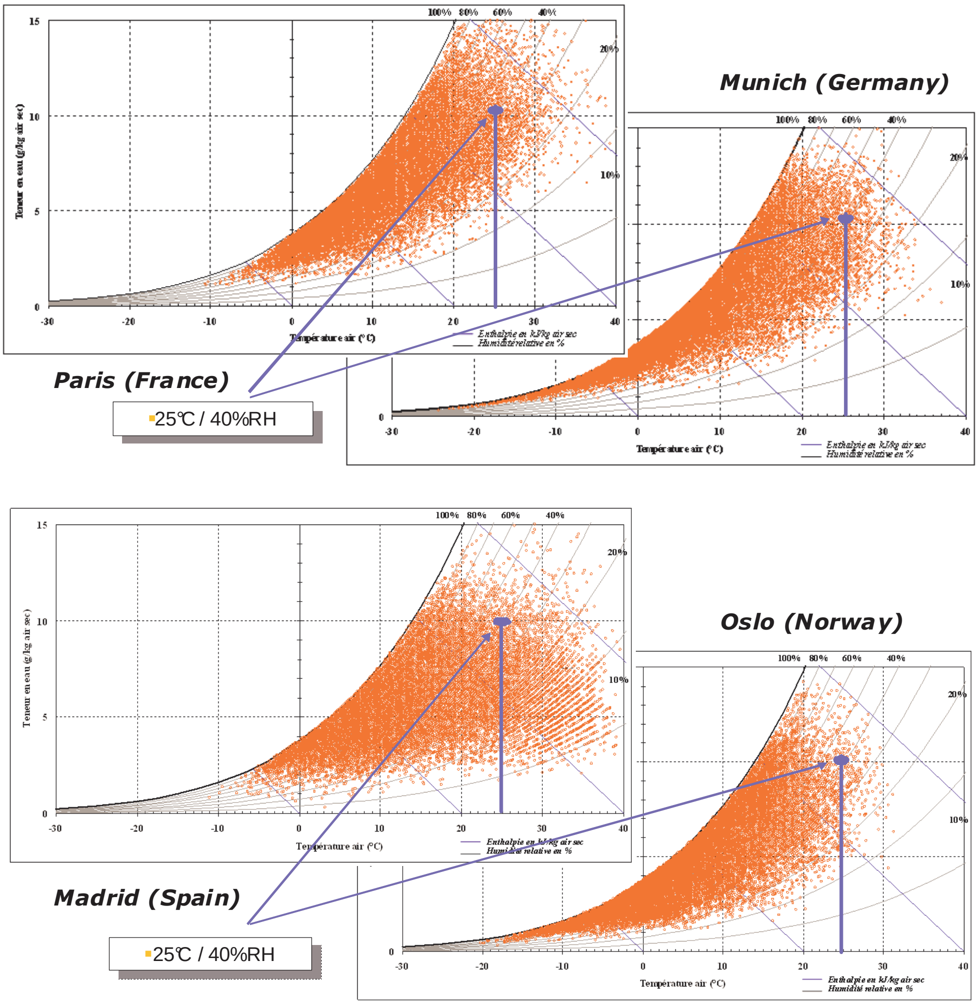


The figure below shows the ambient temperature conditions during a whole year when real driving usage occurs, for 4 different locations in Europe (south of Spain, France, Germany, and north of Scandinavia). We can see that above 25°C only 5% of real customer driving usage occurs.

In addition, 25°C deal also with practical reasons, because current type approval facilities (chassis dyno-test bench) could not achieved 30°C with good enough accuracy. If a climatic chamber should be considered, an unaffordable investment should be done by most carmakers and type approval authorities, and this is not realistic considering EU roadmap.

Regarding the ambient humidity, a value of 40%RH has been established after checking the information coming from the European weather database.

In the figures below, each one of the red dots show the real weather condition (temperature and relative humidity) of every single day for the time period between 1996 and 2005, in 4 different European locations (Paris, Munich, Madrid and Oslo). In these moist air diagrams we have pointed out, with a bleu dot, the situation corresponding to 25°C and 40%RH. As we can see, this ambient setting point is representative enough of most of European summer conditions.



A.13.8 *Possibilities to increase thermal load in the test.*

After previous explanations, we do consider that thermal load proposed in the test method is representative of real life conditions for an European average.
Nevertheless, we could identify three different ways to increase thermal load in the test, if it is requested by regulation authorities:

One possible way could be by increasing ambient temperature. This option has some technical limitations, because as we have already explained, not all existing type approval facilities can reach values around 30°C, and therefore a huge investment would be required.
Another alternative way could be by decreasing air temperature blown into the cabin. This could be feasible, but not fully representative of customer conditions, because we should have 17 – 18°C as setting temperature in control panel.
The third and the best way is **by increasing the air flow blown into the cabin**. Actually this is the easiest way to do it because of the linear influence of the air flow in the global thermal load.

B Support for the impact assessment

For this task from LOT 1 four questions were defined to gather the required input for the impact assessment on MAC and GSI. Based on available data from own research, a questionnaire send to the stakeholders and on data from other sources (e.g. ADAC, CERAM) the contactor should provide;

1. Cost estimates for possible physical test procedures for MAC/GSI efficiency
2. Estimates of the number of manufactured vehicle per physical MAC/GSI test
3. Identification of 3-4 categories of existing typical "advanced" MAC systems
4. Estimates of the costs of a GSI system

A questionnaire was send to the stakeholders on the 24th of December 2009 to collect information from the stakeholders about the topics mentioned above. Before the beginning of February 2010 responses were received from ACEA/JAMA/CLEPA jointly, from Saint-Gobain Sekurit and the Alliance of Automobile Manufacturers.

B.1 Cost estimates for possible physical test procedures for MAC/GSI efficiency

Cost estimates for possible physical test procedures for MAC/GSI efficiency to be performed at type approval. In particular for MAC, several test options, including simplified versions replacing certain expensive experimental features (e.g. solar panels) by analytical estimates (e.g. for the heat take-up of the vehicle at a given solar radiation strength) and less costly physical devices (e.g. electric heater inside the vehicle) should be considered.

A physical test procedure for MACs is a procedure testing a physical system (vehicle with a MAC) or sub-system elements (MAC, cabin, powertrain). This in contrast to a virtual test procedure, which often contains a physical or empirical model of the entire system or sub systems but still, could be based on existing data of the whole system or sub system.

A physical test procedure can test the whole system (vehicle with MAC) in an environment (driver, ambient conditions) or can test sub system *components* (cabin cold demand, MAC cold delivery efficiency, engine efficiency of power delivery) within an environment.

Several physical test procedures for MAC/GSI efficiency are possible:

Existing formal and non-formal procedures, whole system approach:
- ADAC test; driving cycle on a chassis dyno with heater in the cabin
- USA EPA SC 03; test with full simulation of ambient conditions or simulated additional load on chassis dyno;
- Visteon test; 50, 100, idle
- TNO NEDC test with full simulation of ambient conditions;
- TUV Nord/ EMPA; tests with a small solar panel in front of the windshield.
- Common spec book GAR; 5x MVEG-B on and of with 3 different ambient conditions (= 30 tests)

Other options, suggestions for the whole system approach:
- Steady state test on the chassis dyno with one or more test speeds, e.g. idle and 60, or idle, 50 and 100 km/h;
- ACEA steady state tests

Existing procedures on sub system level:
- Common spec book GAR; system bench testing with different conditions
- COP of MAC system or compressor (ISO 917:1989 Testing of refrigerant compressors)
- Heat transfer efficiency of heat exchangers

Similarity for most procedures using the whole system approach is found in the fact that a chassis dynamometer is used to test the whole vehicle including the MAC. The vehicle drives a cycle with the A/C on and off. The effect of the A/C on fuel consumption is determined by the difference measured between the two tests;

dFC = FC A/Con – FC A/Coff

For physical testing using the whole vehicle approach the procedures differ in how the vehicles are driven (driving cycle) and more important in how the ambient conditions are simulated.

Virtual testing of the MAC efficiency may be considered at a later stage.

The baseline for a test procedure for MACs is a driving cycle with the MAC system ‘OFF’ in a standard laboratory environment. The ambient conditions like temperature and humidity of the lab room should be the same as during the regular TA test.

For the condition ‘MAC on’ there are several options to simulate the ambient conditions that could be relevant for MAC testing;
- Electrical heater; simulated heat flux to be calculated/modeled
- Solar panels; from a small area pointing at the wind shield as used by EMPA to full simulation of solar load above the complete vehicle (US EPA SC03)
- Room temperature control; from intermediate temperatures around normal test cell temperatures of 20-30 to 40-50C.
- Driving wind simulation; from a fan in front of the vehicle to a complete wind tunnel.
- Humidity control.

Costs
The base cost estimates for a standard fuel consumption test under standard conditions of a vehicle driving a certain cycle on a chassis dynamometer vary little across the test labs. Additional attributes like solar simulation and full simulation of wind (wind tunnel) for the tests with ‘A/C on’ would increase the costs per tests significantly. A heater would increase testing costs only marginally.

Costs for testing on a chassis dynamometer in an emission laboratory constitute
- Chassis dyno
- Test equipment aside dyno; CVS, analysers, temperature sensors.
- Ambient control
- Personnel

The additional costs for a test depend on if a cold start or if conditioning of the cabin and vehicle is required. For instance, if a test has to be driven just after the regular NEDC it is a warm started test and hence, the ’MAC on’ and ’MAC off’ test both should be warm started and one needs an additional test for MAC off. If the test needs to be cold started no additional MAC off test is required but the vehicle has to be taken of the chassis dynamometer to cool down. For a conditioned test it is assumed that a vehicle also has to taken of the chassis dynamometer after the regular NEDC. For a step test anyway two extra cycles are required. They both can again be driven after the NEDC if no special ambient conditioning is required. Special conditioning would increase costs because the vehicle either takes conditioning time blocking the chassis dynamometer for other tests or be taken of the CD to allow other cars to be tested.

A test executed in a standard emission laboratory in combination with and executed just after the TA test without any special ambient control and without all preparations will be additional around EUR 1000,= per NEDC (ADAC, TNO) for almost half an hour of testing. A full stand alone test with preparations and conditioning costs more. ACEA reported EUR 1650 per hour and 1,2 hours required per NEDC to perform a cold NEDC. Emission tests performed in special climate chambers cost considerable more. But, maybe more important, facilities sharing full climate simulation (sun, wind, temperature and humidity), emission testing and a chassis dynamometer are scarce.

But costs do not only consist of testing costs. Other costs could be considered as well;
- Vehicle rent, transportation, depreciation of the test vehicle. This is required anyway for the regular NEDC test so it will not be an additional burden
- Special preparation, instrumentation (e.g. thermo sensors and data-acquisition)
- Coast down test on the chassis dynamometer if coast down values are used to adjust the chassis dynamometer. This is done anyway for the regular NEDC test so it will not be an additional burden.
- Administration, reporting of particularly the MAC settings and results.

B.2 Estimates of the number of manufactured vehicle per physical MAC/GSI test

Estimates of the number of manufactured vehicle per physical MAC/GSI test, which could e.g. be based on type approval data taking into account possibly necessary partitioning of vehicles (e.g. having different MAC systems installed) covered by a single type approval for the purpose of a MAC/GSI test.

This is largely depending on the manufacturer and on the properties one wishes to differentiate systems. Currently, manufacturers have many TA variants/parents, which is one of the main parameter determining the amount of TA tests. The variant are amongst others defined by chassis shape, engine capacity and tolerance for deviation in CO_2 emission from the parent. A MAC adds variants as certain TA variants may have different MAC systems. The amount of MAC variants depends on the feature or parameter one wishes to distinguish.

The industries response to the questionnaire states an order of magnitude of hundred(s) Type-Variant-Versions per manufacturer which would lead to thousands Type-Variant-Versions-MACS if there would be 10 type of MACs as is the case for the leakage TA.

It is assumed that the amount of tests required especially for MAC can be drastically reduced an order of magnitude or more if families of the regular emission and fuel consumption TA and MAC families of the leakage TA are aggregated into larger groups or families to be defined for the MAC FC test. Several properties of vehicles and MAC systems were already investigated for their impact on fuel consumption in this study. These properties can be taken into account for the selection of MAC families of vehicles and MAC systems.

In Annex A similar MAC systems were tested in vehicles with different engines. This lead to a difference in absolute additional FC but only in a marginal difference (1-2%) of the relative difference.

An option to decrease the number of tests, yet to deliver accurate enough figures for each vehicle type, is to allow correction models for the features that affect the additional FC and can easily be modelled or estimated.

Identification of 3-4 categories of existing typical "advanced" MAC systems
Identification of 3-4 categories of existing typical "advanced" MAC systems (relative to a simple baseline version) according to certain technical features (e.g. automatic control, air recirculation, variable displacement) and combinations thereof.

Summary
Typically, when the additional fuel consumption is concerned not only the MAC system itself determines the additional FC but also other features, like the vehicles cabin and glazing. When the MAC system itself is concerned it can generally be qualified or graded in two dimensions, namely by components and by engineering effort. The engineering effort is hard to quantify but probably has a large effect on the additional FC. The effort obviously ranges from none (only taking other design criteria into account, like costs and packaging) to the best possible for optimisation of the system with regards to the minimisation of the additional FC. When grades of MAC systems have to be defined, based on type of components, the choices are somewhat arbitrary;

Table; Overview of three MAC grades based on components.

Basic/simple	- VDC* with internal control / (FDC** still used) - Orifice tube as expansion device - Accumulator - Manual control (fans, air flow control flaps) - Simple heat exchangers
In between /regular	- VDC* with internal control - thermal expansion valve - integrated or non-integrated receiver/dryer - IHX/ no-IHX - simple control
Advanced	- VDC* with external control - Thermal expansion valve or other type of controlled expansion device - Integrated receiver/dryer

	- IHX, Internal heat exchanger - Automated control (PWM fans and blower, air flow control flaps) - More efficient heat exchangers (micro channel, others.)

VDC* = Variable Displacement Compressor

FDC** =Fixed Displacement Compressor

Elaboration

The additional fuel consumption due to usage of a MAC is determined by a range of parameters:

- Heat load to the cabin and cabin properties, like glazing, cabin size, colour of the car, dashboard colour, etc... These parameters influence the demand of cold at a given comfort level.
- The efficiency of cold generation. Elements that influence the efficiency are amongst others the engine, the compressor, the heat exchangers, fan operation and even the shape of the front end of the car.
- The amount of cold production per unit of time or ‘system size’ or the systems cold production capacity.
- The mass of the system. It increases the driving resistance.

Of a complete vehicle three sub-systems can be distinguished;

1. Engine
2. MAC
3. Cabin

A complete system can thus be qualified or graded through different features. Design criteria for MACs are dominated by comfort, costs and packaging however. At the moment energy use of a MAC is still of secondary importance for the design. The MAC system alone and the cabin are the main systems that can be adapted to achieve a lower fuel consumption due to MAC use. TNO 2006 reported, based on information from the industry that by using advanced technology a reduction of the fuel consumption is possible. Other investigations and a response to the questionnaire mentioned a significant influence of reflective glazing on the additional fuel consumption. Besides advanced technology good engineering practice stimulated by the goal to increase the efficiency may lead to better systems and better integration of the system in the vehicle. A good example is the efficiency of the condenser; often a bumper or a chassis beam hinders the air flow through the condenser and resultantly the efficiency of the heat exchange drops. In some cases even the heat exchanger of the engines cooling system (radiator) is placed in front of the condenser and reduces its’ efficiency. The overall design in this case is determined by other factors than the efficiency of the MAC.

In general, one could state that reducing the heat load results in a reduced demand for cold air and that a well integrated advanced MAC system could supply that cold air with more efficiency where the combination of both leads to a reduction of fuel consumption. Some properties of these systems do not have measureable quantities and thus make it hard to define grades or classes.

In the industries response to the questionnaire it was stated that a simple definition of MAC components not necessarily describes the system design and package and that therefore the efficiency is subject to a complex interaction of all components. This does not mean that one cannot distinguish hardware that allows for a better efficiency. Examples of more efficient hardware are; the variable displacement compressor opposed to the fixed displacement compressor. The external controlled compressor opposed to an internal controlled one. A controlled expansion valve opposed to the orifice tube. Pulse width modulation (more flexible) fan control or not. Integrated receiver versus non-integrated receiver. Each of these components can be arranged with the other and so a large number of combinations can be defined.

CLEPA reports various MAC elements with advanced technologies:
- Non integrated versus integrated receiver;
- Thermostatic Expansion Valve (TXV) versus the orifice
- Variable internal / external controlled piston compressor, fixed piston compressor, fixed rotary, vane compressor, scroll, swash plate, swivel plate (Doowan, ObrisT);
- Single versus multiple evaporator;
- With versus without Internal Heat eXchanger (IHX);
- With versus without Pulse-Width Modulation (PWM) blower & fan;

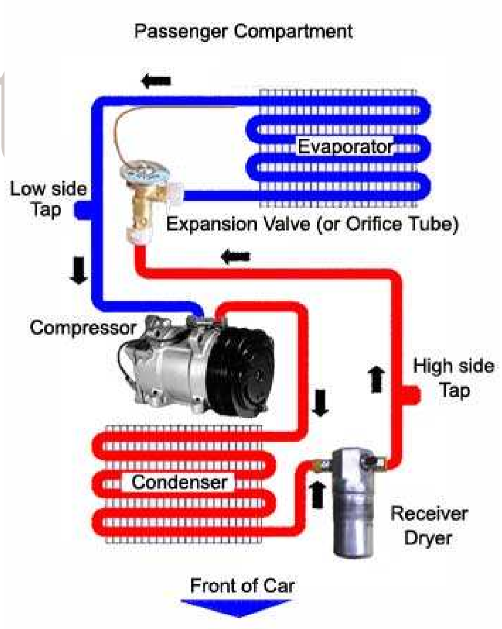
In the response to the questionnaire ACEA /JAMA/CLEPA also mentions grades for;

- Homogeneity of air flow at the condenser; from poor to very good
- Hot air circulating from the engine through the condenser; from high to low
- Pressure drop of the piping; from high to low
- Amount of reheat needed; from high to low
- Electrical efficiency; from basic to enhanced
- Component efficiency; from basic to enhanced
- Condensor efficiency; from basic to sub-cooled with parallel flow or micro channels
- Size harmonization; from none to perfect
- Heat tightness of the cabin; from poor to very good
- Inlet air circulation control; from manual to automatic

Obviously, there is a difference in hardware technology grades throughout the size or price range of passenger cars; for smaller passenger cars a MAC should be simple and low cost yet still deliver a certain amount of comfort. For larger more expensive executive type cars a MAC is expected to control climate under all conditions swiftly, accurately, silently and comfortably, often with separate control for every passenger. Other issues play a role as well. Drivability for instance is an issue for the lower motorized cars. A compressor switching on at full load at idle or when just driving away may be annoying if it leads to an engine stall.

Concluding, for the definition of grades or classes of advanced MAC systems two different matters determine the additional FC; these are the **components** and the **engineering effort** for optimization with regard to FC. To determine costs of advanced MAC systems one should therefore not only look into additional component costs but also take into account the effort for engineering. For the definition of classes or grades engineering effort will not be taken into account as one can only classify the effort qualitatively. Engineering effort may become more or better if there are incentives to improve the MAC efficiency. Resultantly, for MACs grades can only be defined roughly, based on general component qualifications.

The most common components that make up MAC systems are the following: Compressor, Condenser, Evaporator, Throttling/Expansion device, Receiver-Dryer/Accumulator, air flow control (flaps) and sensors.



MAC grades based on components
A basic system would have manual control, a variable displacement compressor, although some still have a fixed displacement one, an orifice tube, a non-integrated receiver dryer and no intermediate heat exchanger (IHX). These systems are often found in smaller less expensive cars.

Going from basic to the most advanced system each grade would have more and more improved components over the previous system. But the combination of components to select for each next grade is somewhat arbitrary as the selection of the type of components does not necessarily depend on each other.

Going to ‘advanced’ in general the systems control gets better. This can be observed for the compressors displacement (internal or external control), the expansion valve, the fans, air flaps and the sensors used.

The most advanced system would have an external controlled compressor, a thermal expansion valve, possibly an integrated receiver dryer and an intermediate heat exchanger, automated control with ambient temperature and humidity sensors, PWM controlled fans and a controlled air flow system (flaps).

A system in between would have combinations of all components different or additional to the basic system.

Estimate of differential system costs and fuel/CO_2 emission savings

Estimate of differential system costs and fuel/CO_2 emission savings (relative to a simple baseline version) of the identified "advanced" MAC system categories

Summary
Until now, little information regarding costs of advanced MAC systems is available. Cost information is available from a respondent to the questionnaire and from the SAE ARRS 2003. Both estimates differ quite a lot, but it must be stated that for both sources it was not very well defined what the costs of a baseline system is composed of and what the costs of an advanced system is composed of. The respondent to the questionnaire noted a 50% reduction in FC at the cost of EUR 200. The SAE ARRS presentation noted a 25-30% reduction of FC at the costs of EUR 40. These figures differ substantially and it is not clear why they differ.

Elaboration
Little information regarding costs of advanced MAC systems is available.
One respondent to the questionnaire supplied a simple cost benefit calculation with costs for different MAC systems:

Table; response to the questionnaire. MAC system costs and FC reduction.

System type	Additional FC in ADAC Test	System costs
0. without MAC	0%	0 €
1. controlled compressor	5%	900 €
2. uncontrolled compressor with air flap control	10%	700 €
3. uncontrolled compressor without air flap control	20%	500 €

The other latest collected information stems from the findings of the SAE platform Interior Climate Control Standards Committee which have been used and summarized in [Smoker et al., TNO, 2006]. For systems with the same type of refrigerant it is shown that the improved/ advanced systems may be 25 % to 30% more efficient against extra costs of EUR 40.

An SAE platform (SAE Interior Climate Control Standards Committee) established the SAE Alternative Refrigerants Cooperative Research Programme SAE ARCRP in 2001 to develop and assess new solutions for MACs. Every year they have a symposium and the members of the Committee (manufacturers, suppliers and Governmental Organisations) present results with respect to the technological development of some alternative systems and their potentials and problems. For the SAE ARSS 2003 the potential of several systems was assessed and held against the baseline R134a system. Besides, costs estimations were made for those systems. The table below summarizes the findings for the reduction of so called indirect emissions (additional tail pipe CO_2 emission or additional FC) due to MAC use.

Table; estimated costs and FC reduction for an improved HFC-134a MAC compared to a baseline system (with a variable displacement compressor). Source: SAE ARSS 2003.

System choice	FC Reduction indirect	Costs* [Euro]
HFC-134a VDC	Baseline	Baseline
Improved HFC-134a	25-30%	40

*Manufacturer costs, including safety, ‘direct and indirect’ features.

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