# Linear Modeling, Simulation and Experimental Verification of a Pressure Regulator for CNG Injection Systems

*Dirk Hübner, Harald Ortwig*

Trier University of Applied Sciences
Department of Mechanical Engineering
P.O. Box 1826, 54208 Trier, Germany
d.huebner@fh-trier.de, ortwig@fh-trier.de

## ABSTRACT

The number of motor vehicles powered by internal combustion engines keeps growing despite shrinking oil reserves. As a result, compressed natural gas (CNG) is gaining currency as an emerging combustion engine fuel.

To this day, CNG systems – e.g., in passenger cars – are not fully integrated into the development process as conducted by vehicle or engine manufacturers. Instead, they are usually "adapted in" at a downstream stage by small, specialized companies.

The present paper initially outlines the state of the art in advanced gas injection technologies. Especially the development towards sequential injection systems is described.

A pressure regulator for CNG driven combustion engines is examined in detail, given its role as a highly sensitive and critical system component. Based on a precise theoretical analysis, a linear model of this pressure regulator is derived and subjected to dynamic simulation.

The analytical approach is accompanied by an experimental investigation of the device. On a test rig developed at the Trier University of Applied Sciences, the static and dynamic features of the pressure regulator can be measured with the requisite precision.

The comparison of measured and simulated data yields a validation of the dynamic simulation model. With the approaches developed it is now possible for the first time to model, simulate and optimize single- or multi-stage pressure regulators for CNG driven engines with less effort and higher accuracy.

***Keywords:*** CNG, Compressed Natural Gas, Modeling, Simulation, Injection System, Pressure Regulator, Applied Optimization

## 1. INTRODUCTION

In the face of today's traffic development trends, shrinking oil reserves, rising prices of gasoline and diesel fuel, the restrictions imposed by national and international environmental policies/ legislation and the present-day awareness of the ecological implications of exhaust emissions, vehicle users are looking for alternatives to conventional motor cars powered by spark-ignition or diesel engines. Market demand in this segment might be met by vehicles using alternative engine fuels.

The alternative fuel types currently favored by political decision-makers, automotive manufacturers and customers chiefly include the following:

- natural gas (CNG or LNG)
- butane/propane mixtures (LPG)
- hydrogen ($CH_2$ or $LH_2$)

Hydrogen-powered engines have not yet found their way into high-volume manufacturing due to a variety of technical obstacles. On the other hand, natural gas (CNG) or butane/propane mixtures (LPG) have been around for years as a fuel for converted spark-ignition engines originally optimized for gasoline operation. Conversion kits are marketed by various manufacturers but usually fail to meet OEM equipment standards in terms of system integration and performance characteristics.

Growing demand for alternative-energy vehicles has spurred the automotive industry's interest in offering such vehicles "ex works", in line with customer preference. As a result, vehicles designed for natural gas operation directly by the manufacturer have become available in the marketplace in recent years. However, these vehicles too rely on the use of conversion kits sourced from specialized suppliers. Their advantage over a conventional conversion lies in the close cooperation with vehicle and engine manufacturer, which makes it possible to achieve higher levels of system integration and hence, superior performance characteristics.

Since the market for natural-gas powered vehicles is expanding despite the downtrend in overall new registration figures in Europe, and since components meeting the automotive industry's exacting demands are still not readily available, there is scope for a major development effort in this field. The greatest development need can be identified at the level of the fuel system /1/. Demand for high-quality solutions is voiced by original equipment manufacturers (OEM) and customers alike.

The present paper addresses the fluid and control technology aspects of a fuel system in which natural gas is stored in pressure vessels (tanks) at up to 200 bars. In the following sections we shall be examining the general structure of such a CNG system, as well as development trends in the field of natural gas injection equipment. This will be followed by a numerical model of a pressure regulator stage plus the associated components on the low-pressure side. The model of the regulator stage, prepared on the basis of extensive analyses, will be mapped and simulated using a software capable of dynamic system simulation. Concurrently, the modeled pressure regulator stage will be examined experimentally. The model can thus be validated through a comparative evaluation of the measurements and simulation results. It will thus become possible at some future point to identify influencing factors and to give design recommendations on the basis of simulation-based parameter variations.

## 2. CNG SYSTEM OVERVIEW

The schematic diagram in Fig 2.1 illustrates the structure of a natural gas fuel system.

The gas tank system is charged via a filler connection. An integrated check valve prevents any reverse flow. In practice, the maximum gas pressure varies between 200 and 250 bar.

The solenoid valves (electromagnetic valves), when energized, open the path toward the main shutoff valve from where the gas reaches the subject of our investigation, i.e., the pressure regulator. At this point, the pressure regulator controls the variable inlet pressure down to a near-constant outlet pressure of between 1 and 10 bars above atmospheric, depending on the system.

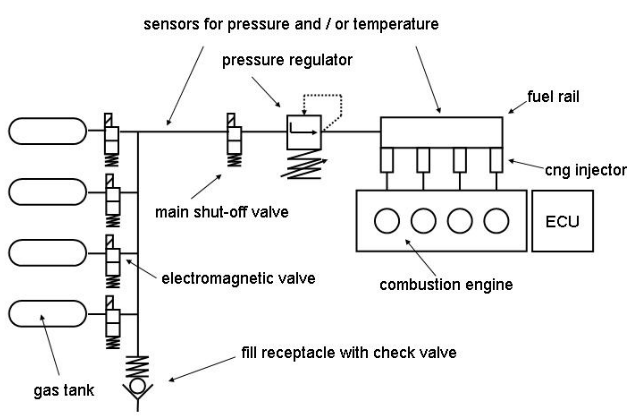


Fig. 2.1

The natural gas then reaches a fuel rail which, in 4th generation systems, comprises the gas injection valves that meter the gas mass selectively into the cylinders by their intermittent operation.

## 3. DEVELOPMENT TOWARDS SEQUENTIAL INJECTION

On first-generation natural gas systems for motor vehicles /2/, the gas is injected via a venturi nozzle operating at a constant inlet pressure. The formation of the fuel/air mixture is controlled via the airflow. As a result, the composition of the mixture cannot be selectively influenced while the engine is running. Such systems are therefore incompatible with lambda control technology (Fig 3.1).

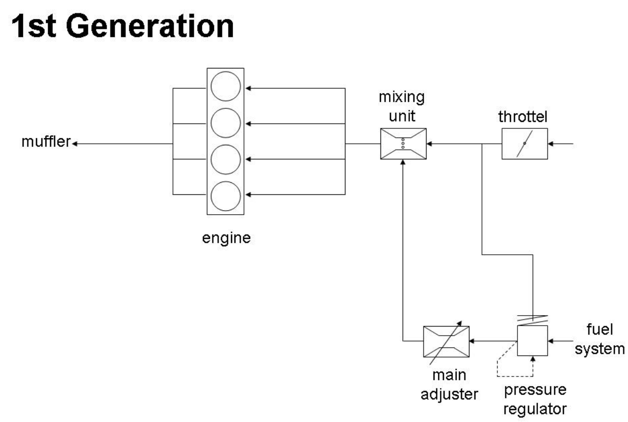


Fig. 3.1

On second-generation systems /2/, the inlet pressure upstream of the venturi nozzle is adjustable. The composition of the mixture can thus be controlled while the engine operates, and a simple lambda control system can be implemented. However, the dynamic capabilities of the system – in terms of variability of the mixture composition – remain limited (Fig. 3.2).

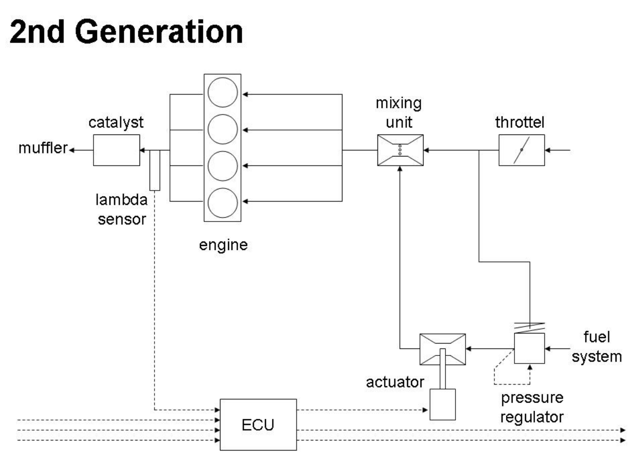


Fig. 3.2

On third-generation systems /2/, gas is metered into the cylinders via a multiport unit and downstream leaf valves. Still, the dynamic response of such a system leaves room for optimization (Fig. 3.3).

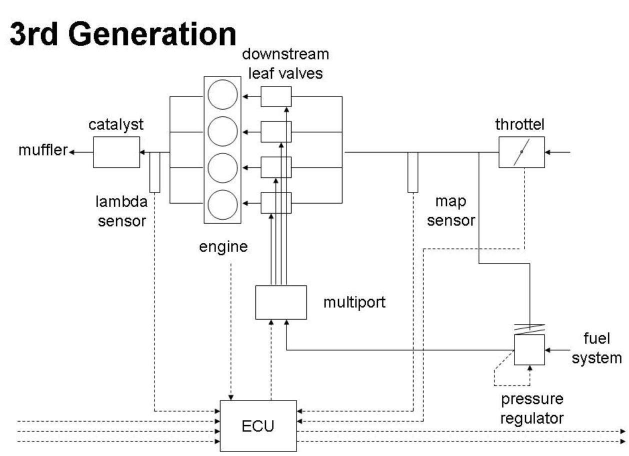


Fig. 3.3

Fourth-generation designs /2/ represent the most advanced gas charging method today in that they provide a sequential injection with individual cylinder selectivity. Fuel is injected into the intake manifold for each cylinder separately, so that the fuel composition can be controlled highly dynamically for optimum operating conditions. Although such systems have been developed at the basic design level, only individual solutions exist so far and each system proposed has usually been optimized for its specific application. It is in such systems that the pressure regulators to be examined here are habitually employed (Fig. 3.4).

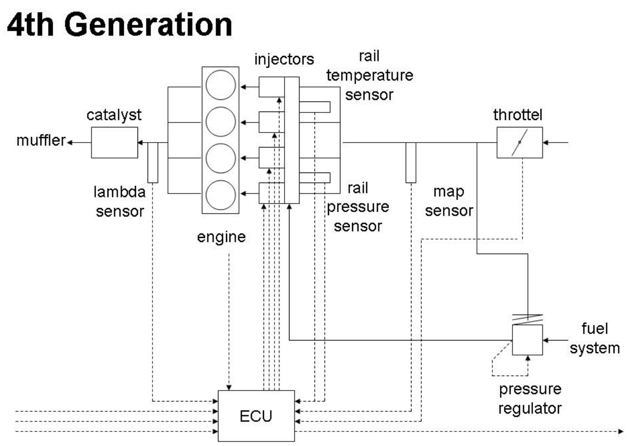


Fig. 3.4

## 4. MODELING

In deriving the model, all aspects known to date were duly taken into account. The model was created on the basis of the inter-relationships given in Fig. 4.1.

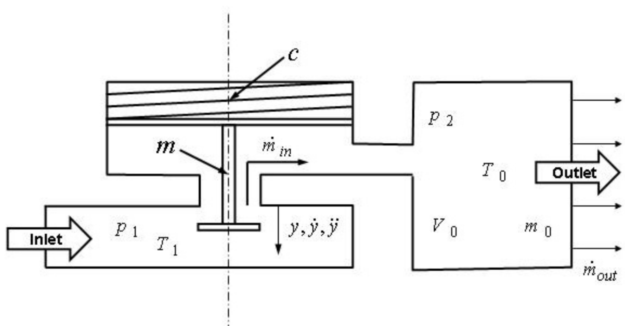


Fig. 4.1

Outlet pressure $p_2$

Assuming an isentropic change of state in the system's low-pressure volume /3/, the pressure change will follow the equation given below:

$$d\dot{p}\cdot\left(\frac{V_0}{m_0}\right)^{\kappa}+\kappa\cdot p_0\cdot\left(\frac{1}{m_0}\right)^{\kappa}\cdot V_0^{\kappa-1}\cdot d\dot{V}-\kappa\cdot p_0\cdot V_0^{\kappa}\cdot m_0^{-\kappa-1}\cdot d\dot{m}=0$$

If we further surmise that the volume change remains negligible, the following equations are obtained:

Pressure change over time

$$d\dot{p}=\kappa\cdot p_0\cdot\frac{d\dot{m}}{m_0}=\kappa\cdot\frac{R\cdot T_0}{V_0}\cdot d\dot{m}$$

Mass flow differential $d\dot{m}$

$$d\dot{m}=\dot{m}_{in}-\dot{m}_{out}$$

Mass flow $\dot{m}$

$$\dot{m}=\mu\cdot A\cdot\sqrt{2\cdot\rho_1\cdot p_1}\cdot\sqrt{\frac{\kappa}{\kappa-1}\cdot\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{\kappa}}-\left(\frac{p_2}{p_1}\right)^{\frac{\kappa+1}{\kappa}}\right]}$$

Cross-sectional area of flow $A$

$$A=A_0+k\cdot y$$

Valve gate travel $y$

$$y=\int\dot{y}\cdot dt=\iint\ddot{y}\cdot d$$

Acceleration $\ddot{y}$

$$\ddot{y}=\frac{dF}{m}=\frac{\sum F_y}{m}$$

Sum of forces $\sum F$

$$\sum F=\sum F_{pressure}+\sum F_{spring}+\sum F_{friction}\sum F_{stream}$$

Temperature $T_0$

$$\left(\frac{\Delta T}{\Delta p}\right)_h=\alpha\cdot\left(\frac{273}{T}\right)^2$$

For modeling purposes, the temperature in the low-pressure section is computed using the value $\alpha$ = 0.265 K/bar /4/ and $(273/T)^2 \approx 1$.

In the following simulation of the regulator stage of the examined diaphragm-type pressure regulator, the secondary factors "intrinsic resistance" and "deflection-related active surface area of the diaphragm" have been neglected.

From the analytical work we can derive the action diagram shown in Fig. 4.2.

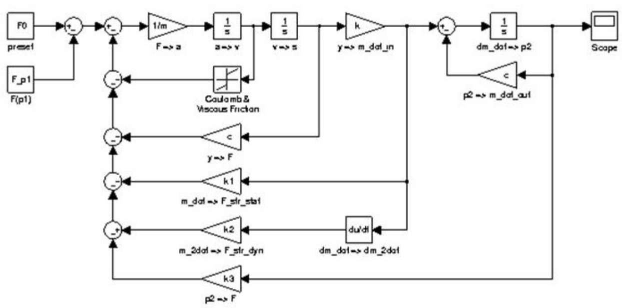


Fig. 4.2

## 5. EXPERIMENTAL ANALYSES

Fig 5.1 gives a schematic view of the experimental set-up.

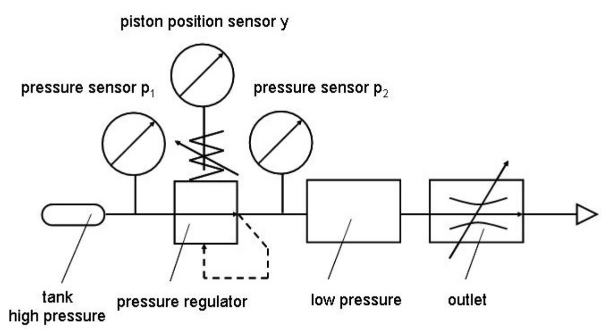


Fig. 5.1

As will be evident from the above, the gas flows from the pressure vessel (tank) to the pressure regulator via a pipe. A line pressure sensor determines the regulator inlet pressure (i.e., the pressure level on the high-pressure side). On the pressure regulator itself, an inductive position encoder determines the piston position. The throttling point, designated "outlet" in this diagram, lies at the end of the measuring section. In between we have a low-pressure volume (which, in the application case of a CNG-powered motor vehicle, corresponds to the rail) as well as the sensor measuring the regulator outlet pressure.

In our measurements, both steady-state and dynamic behavior were determined. For clarity's sake, Fig. 5.2 illustrates the respective measuring processes designated by the words "static" and "dynamic", respectively.

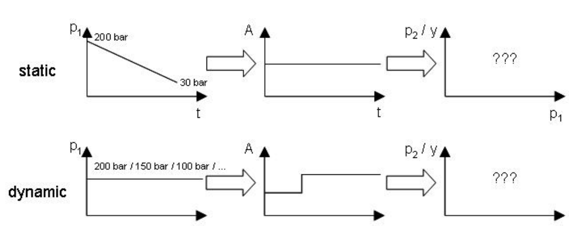


Fig. 5.2

## 6. COMPARSION OF MESSURED AND SIMULATED RESULTS

In the following sections, the measurement and simulation results are compared for the respective boundary conditions, i.e., a "static" and "dynamic" situation.

### 6.1 Static

In the "static" measurements, bore diameters of 1, 2, 3 and 4 mm were used to represent the cross-sectional area of flow in the low-pressure section.

The equation for calculating the outlet pressures and piston positions was derived from the force equilibrium equation. Time-related variables have no influence on the description.

Fig. 6.1.1 shows the outlet pressure while Fig. 6.1.2 plots the piston position over the inlet pressure for the respective cross-sections of the low-pressure system.

Figs. 6.1.3 and 6.1.4 give the corresponding simulation results.

A high coincidence quality is evident from these results. Existing deviations are attributable to non-linearities which are unknown to date and could therefore not be taken into account.

### 6.2 Dynamic

The "dynamic" measurements were conducted with a near-constant inlet pressure but with abrupt variations of the cross-sectional area of flow in the low-pressure section.

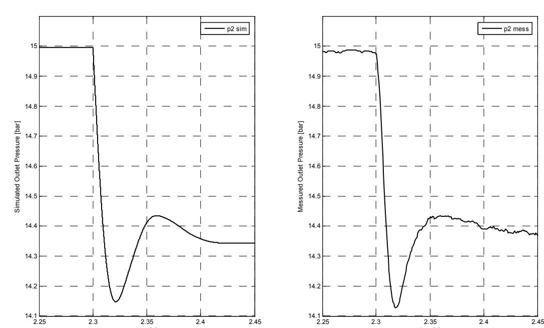


Fig. 6.2.1

Figs. 6.2.1 and 6.2.2 show the measured step responses of the outlet pressure and piston position at an inlet pressure of 65 bars. The corresponding dynamic simulation results appear directly alongside these diagrams.

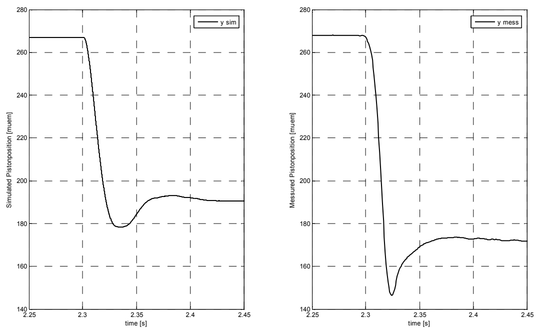


Fig. 6.2.2

## 7. CONCLUSION

The objective of this study was to conduct an experimental analysis and to model and simulate the behavior of one stage of a mechanical pressure regulator for CNG-operated spark-ignition engines.
Using the equations developed herein, it is possible to describe the control behavior of single- or multiple-stage pressure regulators (whether all-mechanical or electromechanical), to simulate such devices, and to optimize their performance by means of computer-based parameter variations, all without having to resort to time-consuming and costly development tests.

The investigations were carried out for a two-stage diaphragm-type pressure regulator. This device was selected in view of its widespread use in vehicles powered by CNG-powered spark-ignition engines. Moreover, it exhibits certain design features which facilitate the modeling process.

A comparison of the measurements and simulation results for the first regulator stage reveals a very good coincidence in the pressure profiles. The model may thus be deemed successfully validated.

The actual pressure profile in the low-pressure chamber of the relevant stage has not been further examined herein. It should in any event form the subject of subsequent research. Moreover, the model is not yet sufficiently detailed as regards its capabilities to take into account pressure losses and other influencing factors. Further work should also include mass flow and temperature measurements, since these would allow an evaluation of the response characteristics of individual system components, thus refining the model even more.

Investigations on pressure regulators offered by different manufacturers will permit comparisons and evaluations of different makes and versions. The improvement potentials residing in this technology could thus be fathomed, and significant improvements in the operating reliability and efficiency of CNG-powered spark ignition engines would become achievable.

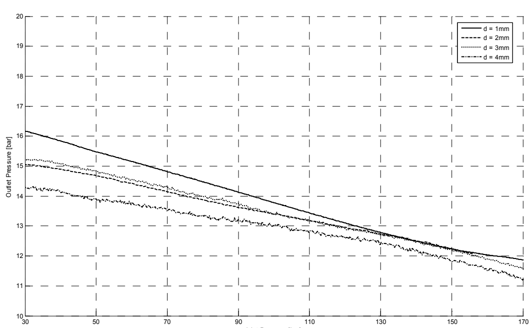


Fig. 6.1.1

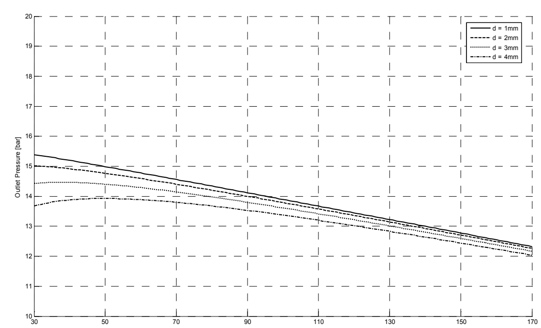


Fig. 6.1.3

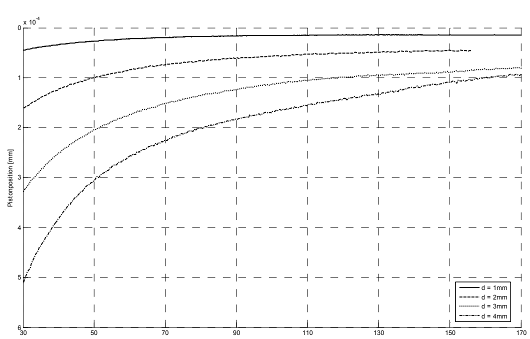


Fig. 6.1.2

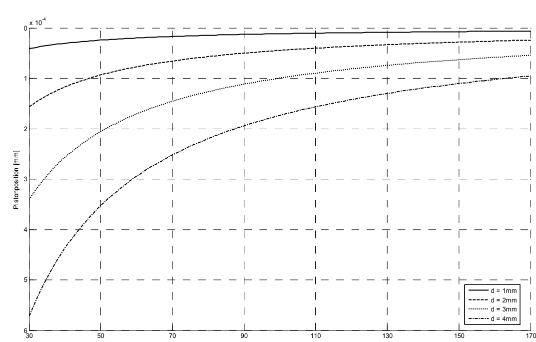


Fig. 6.1.4

## 8. REFERENCES

[1] Iraklis Avramopoulos, Andreas Sprysch, Ulrich Holthaus, What demands does serial development place on natural-gas-vehicles? Examples of solutions and a look at their cost-benefit characteristics, Gasfahrzeuge 2004, Oliver Dingel, pp. 4 - 20

[2] http://www.engva.org, European Natural Gas Vehicle Association, http://engva.org/Pages/235/documents/Basic%20LDV%20Info%202.pdf

[3] Prof. Dr.-Ing. W. Backé, Grundlagen der Pneumatik *[Fundamentals of Pneumatics],* 7th edition, 1986

[4] Cerbe / Hoffmann, Einführung in die Thermodynamik *[Introduction into Thermodynamics]*, 10th ed., Carl Hanser Verlag, Munich/Vienna, 1994

## 9. CONTACT

**Dirk Hübner** holds a Dipl.-Ing. degree from Trier University of Applied Sciences, Germany. He is a member of scientific staff in the Fluid Power Department of that university since 2004.

His scientific interests include modeling, simulation, and applied optimization of CNG injection systems.

Dipl.-Ing. Dirk Hübner
Trier University of Applied Sciences
Mechanical Engineering
Schneidershof
D-54293 Trier
Germany
Phone: +49.651.8103 211
Fax: +49.651.8103 377
E-mail: d.huebner@fh-trier.de

**Harald Ortwig** holds a Dipl.-Ing. (MSc) and Dr.-Ing. (PhD) degree in mechanical engineering from the Technical University of Aachen, Germany. Currently, he is Professor and head of the Fluid Power Department at Trier University of Applied Sciences, Germany.

In addition to these duties, he undertakes consultancy work and expertises. He is acknowledged as a publicly certified expert in fluid power and failure analysis and in the valuation of machines and industrial equipment, apart from being a member of the SAE Fluid Power Committee and Fluid Power Net International.

Professor Dr.-Ing. Harald Ortwig
Trier University of Applied Sciences
Mechanical Engineering
Schneidershof
D-54293 Trier
Germany
Phone: +49.651.8103 367
Fax: +49.651.8103 377
E-mail: ortwig@fh-trier.de