

Development of High Level Modeling Method for Rapid Modeling Process

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ABSTRACT

A novel plant modeling method called High Level Modeling (HLM) to design and develop control-oriented plant model is introduced. The HLM method is specifically designed to expose the design intention at physics level of a target physical system in a straightforward manner so that plant models can be efficiently understood and peer-reviewed from the physical standpoint. The method also enables construction of system equations from the design information based on simple formal rules which guarantees conservation laws. To show the effectiveness of the HLM approach through a concrete use case, it is applied to turbocharger radial compressor modeling, and analysis on the model equations is performed by deriving compressor speed lines and efficiency.

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1. INTRODUCTION

In-vehicle control system is becoming larger and more complicated to meet more challenging and seemingly opposing requirements on fuel economy, emissions and driveability. To keep the quality of control system while avoiding the bloat in development time and cost against this backdrop, Model-Based Development (MBD) technologies are being widely adopted throughout the entire control system development process including control design [1,2], calibration [3] and verification and validation [4,5].

One of the aims of MBD is to enable the control system development to start efficiently by using models of a target system, i.e. plant model, such as an internal combustion engine even before actual physical system is made available. In this MBD approach, it is crucial to be able to build plant models responsively according to the development needs [6]. However, there still remain many challenges to realize such a rapid plant modeling. One of the major challenges is that existing models are likely to become useless when new and more sophisticated requirements on plant models emerged. Another challenge is that modeling process cannot begin before specific requirements are known, but requirements can only be made specific enough by examining issues of ongoing control system development process. This means that plant modeling process can begin in earnest after control system development project starts. Yet another significant challenge is that plant models must be completed and provided to rapidly progressing projects on time. The

implication of this situation is that 1) model parameterization should be kept as simple as possible, 2) plant models must be efficiently useable on time without error-prone manual implementation work as well as too many time-consuming and dedicated measurements for parameter optimization, and 3) simulation must run at a reasonable speed or even at real-time at times.

High-level goals for plant modeling process that can be set out from these challenges are 1) physics-level design intent is exposed in a straightforward fashion for efficient model comprehension, review and design change, 2) system equations are systematically composed from the model design information to serve for model analysis, simplification and control design and 3) model implementation for numerical simulation, which could be either open loop or closed loop with controller, is systematically generated from equations.

There are a couple of existing plant modeling technologies that are in industrial use today. Modelica is one of prominent plant modeling technologies and there are many software packages that support Modelica [7]. While it is useful for system level numerical simulation [8,9], its object-oriented paradigm originates in computer science, not in requirements on plant models mentioned earlier, and the use of inheritance mechanism and procedural programming constructs could make comprehension of textual representation inside Modelica components from physical standpoint impractical. VHDLAMS is another plant modeling

technology [10], but its intended use is primarily for electronics and models in other domains such as internal combustion engine are often very simple [11]. In addition, textual representation at its language level poses similar issues with Modelica. Simulink® is another common choice for plant modeling in control design [12]. However, Simulink basically deals with a system of ordinary differential equations (ODEs) and does not impose any physical principles on plant modeling in there. Because a physical system is generally represented in differential algebraic equations (DAEs), first it must be transformed to ODEs before implementation in Simulink. This may not be a problem if a system is simple enough to deal with or plenty of time is given for modeling, but due to the aforementioned complexities in recent control system development of automotive engines, this approach is becoming too time-consuming and difficult to effectively serve the needs of automotive industrial requirements. To overcome this difficulty, using Simscape™ instead of Simulink might be helpful, but what it provides are physical connectivity principle for components and a DAE solver, which are useful for system level simulation but not for creating physical equations inside a component at a design phase of a plant model (This holds true for Modelica and VHDL-AMS as well). The use of imperative programming languages like C or FORTRAN for plant model design is out of question as they are solely for numerical computation and cannot represent physical design intent at their language level at all.

Thus, we conclude that there is a need for another modeling framework in order to achieve the aforementioned goals, and in this paper we propose to use a physics-oriented modeling method called High Level Modeling (HLM) [13,14], which is suitable for the MBD approach for the control system development. In order to show the effectiveness of the HLM approach through a concrete use case, it is applied to turbocharger radial compressor modeling, and analysis on the model equations is performed by deriving compressor speed line and isentropic efficiency. While this paper highlights analysis of equations, a process starting from physics-level plant model design using HLM, followed by simplification, implementation and simulation is discussed elsewhere [15].

2. HIGH LEVEL MODELING METHOD

In this section, the HLM method is briefly reviewed. In the HLM method, a physical system in question is componentized according to actual physical configuration, and conserved quantities (CQs) are assigned to each component. Then the CQs in different components are connected to represent their flows. Since some physical quantities are not CQ or CQ flow, they are introduced as intermediate variables. If an algebraic relationship exists among these quantities, it can be included as a constraint. Such design information of a physical system can be

naturally expressed in a diagram form, which we call High Level Model Description (HLMD).

In an HLMD (see Figure 1 as an example whose detail will be explained later), a component is shown in a round square, a CQ in a circle, a CQ flow in an arrow with its direction specifying mathematical sign convention (not an input and output relation), a constraint in a rectangle (not appearing in Figure 1). If equations are the only information available about a system in question, one has to decipher the underlying physics from them. But, if an HLMD is presented, physics level design intent is clearly shown.

Once an HLMD is developed for the target system, a set of system equations can be constructed from it. As a simple example, suppose a CQ is denoted as f , a CQ flow coming into it as ϕ_1 , another CQ flow coming out of it as ϕ_2 , then a conservation law is constructed as follows.

$$\frac{d}{dt} f = \phi_1 - \phi_2$$

By applying this rule to all CQs defined in an HLMD, it is ensured that CQs are actually conserved. By specifying constitutive equations and constraints in addition to this rule, a complete set of system equations is built. A constitutive equation may be defined directly for a CQ or a CQ flow, but it may also be defined by introducing an intermediate variable.

At equation level in general, a plant model Σ corresponding to an HLMD is represented with at most five types of functions, each expressing the conservation laws f , the definition of intermediate variables g , the constraints h , the definition of conserved quantity flow rates ϕ and the definition of shape or mechanical configuration φ

$$\Sigma(f, g, h, \phi, \varphi) = \begin{cases} \frac{d}{dt} f(e, V) \\ g(E, X, e, V, u) = 0 \\ h(E, X, e, V) = 0 \\ \phi(E, X, e, V) = 0 \\ \varphi(E, X, V) = 0 \end{cases}$$

where

$E \in R^n$: CQs

$X \in R^m$: Intermediate variables

$e \in R^i$: CQ flow rates

$V \in R^w$: Properties of the components

$u \in R^j$: External inputs to the system.

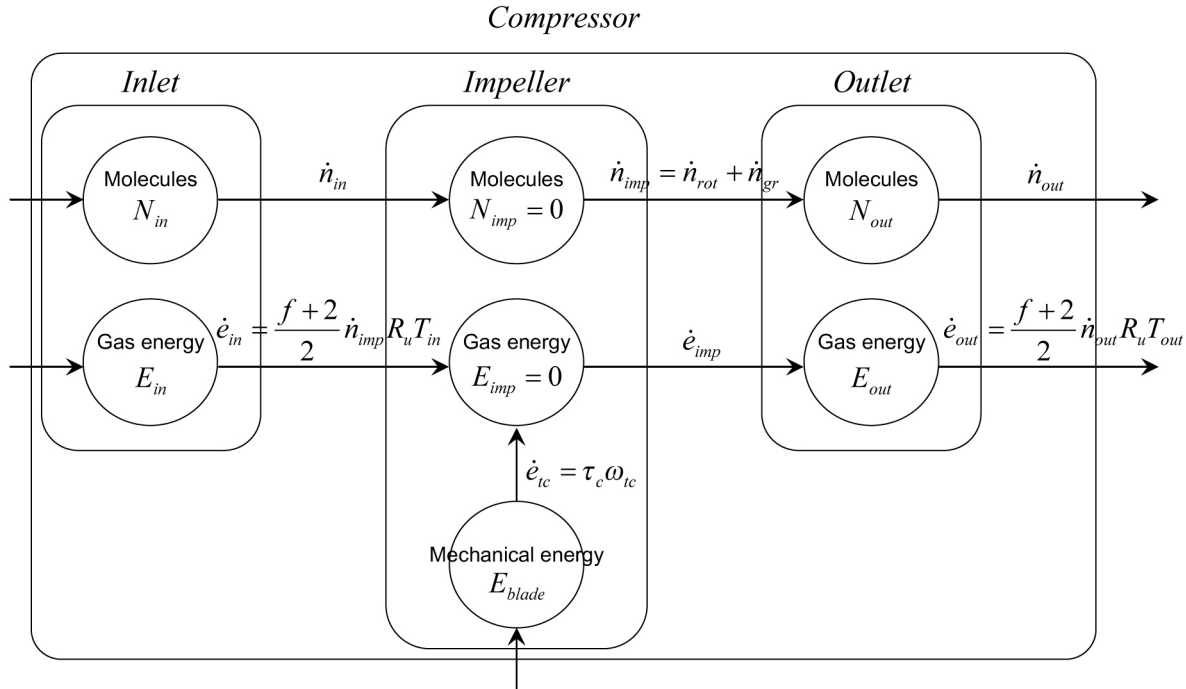


Figure 1. An HLMD of a turbocharger compressor component.

In the next section, the HLM approach is further delineated with a concrete example.

3. TURBOCHARGER COMPRESSOR MODELING

3.1. Background

As mentioned previously, internal combustion engines are becoming more complex than ever before in order to reduce fuel consumption and air polluting emissions without compromising driveability. Exploiting a turbocharger as a pressure charging mechanism is one practical way to grapple with such challenges [16]. Turbocharged engines have been around for many years, but due to the ever increasing complexity of engine system, relevant new studies including control oriented modeling are abundant, for example [12, 17, 18, 19, 20].

Control-oriented models of a radial compressor typically found in literatures [20, 21, 22] are often based on the following two key quantities [23]. One is the dimensionless head parameter

$$\Psi = \frac{c_p T_{in} \left(\left(\frac{p_{out}}{p_{in}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right)}{\frac{1}{2} U_c^2} \quad (1)$$

where c_p is the specific heat of dry air at constant pressure, T_{in} and p_{in} are the gas temperature and pressure at

compressor inlet, respectively, p_{out} is the gas pressure at compressor outlet, γ is the specific heat ratio of dry air and U_c is the impeller blade tip speed.

The other is the normalized compressor flow rate

$$\Phi = \frac{\dot{m}_c}{\rho_a \pi \left(\frac{d_c}{2} \right)^2 U_c} \quad (2)$$

where $\dot{m}_c$ is the mass flow rate at compressor, ρ_a is the density of dry air, d_c is the impeller blade diameter.

For the above two equations, U_c is computed as follows.

$$U_c = \frac{d_c}{2} \omega_{tc} \quad (3)$$

where ω_{tc} is the rotational speed of turbocharger.

Now, let us obtain $\dot{m}_c$ from T_{in} , p_{in} , p_{out} and ω_{tc} . This is a brief review of [23], and of type “Model I” [22] which is suitable for the integration with a mean value engine model for control design purpose. From Eq.(2),

$$\dot{m}_c = \Phi \rho_a \pi \left(\frac{d_c}{2} \right)^2 U_c \quad (4)$$

and Φ is computed from another parametric function

$$\Phi = \frac{(k_5 + k_6 M)\Psi - (k_1 + k_2 M)}{\Psi + (k_3 + k_4 M)} \quad (5)$$

where k_i ($i = 1, \dots, 6$) are adjusted to fit measured compressor speed lines, and M is the inlet Mach number which is calculated as the ratio of the impeller tip speed to the sonic intake velocity

$$M = \frac{U_c}{\sqrt{\gamma R_{air} T_{in}}} \quad (6)$$

where R_{air} is the specific gas constant of dry air.

Ψ in Eq.(5) is calculated from Eq.(1). U_c in Eqs.(4) and (6) are computed with Eq.(3). Parameters excluding gas properties are six adjustable parameters k_i and one geometry parameter d_c .

In addition to the speed lines, the compressor isentropic efficiency η_c plays an important role as it determines the gas temperature at compressor outlet, T_{out} .

The efficiency is approximated with the following parametric polynomial

$$\eta_c = a_1 \Phi^2 + a_2 \Phi + a_3 \quad (7)$$

$$a_i = \frac{a_{i1} + a_{i2} M}{a_{i3} - M} \quad i = 1, 2, 3 \quad (8)$$

where there are nine parameters a_{i1} , a_{i2} , a_{i3} ($i = 1, 2, 3$) that are adjustable. Φ and M in Eqs.(7) and (8) are the same as those for the speed lines.

A slightly simpler quadratic approximation for the compressor efficiency is suggested in [21, 24]

$$\eta_c = \eta_{c,max} - \chi^T Q_\eta \chi \quad (9)$$

$$\chi^T = [\dot{m}_c - \dot{m}_{c,max}, \Pi_c - \Pi_{c,max}] \quad (10)$$

$$\Pi_c = \frac{p_{out}}{p_{in}}, \quad \Pi_{c,max} = \frac{p_{out,max}}{p_{in,max}} \quad (11)$$

$$Q_\eta = \begin{bmatrix} q_{11} & q_{12} \\ q_{12} & q_{22} \end{bmatrix} \quad (12)$$

where six parameters $\eta_{c,max}$, $\dot{m}_{c,max}$, $\Pi_{c,max}$, q_{11} , q_{12} , q_{22} are adjustable.

The models presented here are equation-based, but inferring the underlying physical design, such as conservation

law, as well as explaining each parameter in the equations from the physical standpoint are not straightforward. This makes it difficult to decide if the model can meet requirements for a specific control design goal, and it is even more difficult to change the model according to requirements which are not captured in the model without breaking the physical correctness that have already been captured in the model.

3.2. Compressor Modeling

3.2.1. Plant Model Design using HLMD

As already mentioned, an HLMD of a control-oriented turbocharger compressor component model is shown in Figure 1. From the diagram, some observations from the standpoint of physics are possible. Physical quantities and equations irrelevant to the current discussion (such as the equation for the rotational energy of the turbocharger rotor shaft) are omitted in the diagram.

- A compressor is componentized as three subcomponents, an inlet, an impeller and an outlet.
- Two different CQs are considered. One is the number of gas molecules N , and the other is the energy E .
- Gas runs through the inlet upstream, inlet, impeller, outlet and outlet downstream.
- Energy is exchanged between the gas and the impeller blade.

Some constitutive equations are directly defined for some CQs and CQ flows.

- The mol number and the energy at the impeller N_{imp} and E_{imp} are both set to zero, indicating that the impeller is not a storage-type component.
- Molecular flow coming out of the impeller $\dot{n}_{imp}$ consists of two terms, one is the compressor-driven flow $\dot{n}_{rot}$ and the other is the pressure gradient-induced flow $\dot{n}_{gr}$

$$\dot{n}_{imp} = \dot{n}_{rot} + \dot{n}_{gr} \quad (13)$$

The two quantities $\dot{n}_{rot}$ and $\dot{n}_{gr}$ are intermediate variables since they are not defined as CQs nor CQ flows. Thus, constitutive equations need to be defined for them separately, which will be explained later.

- Gas energy flows (enthalpy flows) at the inlet and the outlet, $\dot{n}_{rot}$ and $\dot{n}_{gr}$, respectively, are specified as ideal gas

$$\dot{e}_{in} = \frac{f+2}{2} \dot{n}_{imp} R_u T_{in} \quad (14)$$