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Implementation of a Theoretical Carburetor Model in One-Dimensional Engine Simulation Software

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ABSTRACT

The main circuits of a small engine carburetor can be represented as a complex, dynamic, two-phase flow fluid network. This paper presents the theoretical characterization of a dynamic one-dimensional model of fuel and air flow in small engine carburetors and its implementation into a one-dimensional engine simulation software package. This implementation allows for studying the effect of changes in individual carburetor parts on engine performance. The characterization of the model indicated that the dynamic behavior of the entire flow network can be captured by the solution of the instantaneous momentum balance equation on the single-phase liquid elements of the network, simplifying the dynamic model considerably.

The second part of this work discusses the implementation into the one-dimensional engine simulation package, and shows examples of the studies that the coupled implementation allow for. Compared to a steady-state model, the implementation of the dynamic flow network captures enrichment effects due to the inertia of the fluid and the continued flow after the intake valve is closed. Parametric studies showed that the most relevant parameters in the carburetor performance are the diameters of the venturi and main fuel orifice.

INTRODUCTION

Although each carburetor design is slightly different and additional circuits may be present in individual models, it is possible to define some basic elements that can be used as building blocks to describe any carburetor design. Figure 1 shows the main circuit of a small engine carburetor, containing the following basic elements.

- **Reservoirs:** Volumes that act as fuel storage, where the fuel level is determined by the hydrostatic weight of the column of fuel and the static pressure at an orifice where fuel is allowed to escape.
- **Metering orifices:** Small orifices that restrict the flow and reduce the mass flow rate. They are characterized with experimentally determined discharge coefficients. Previous models of carburetors had orifices with adjustable cross sectional area, but current small engine carburetors use constant area orifices.
- **Single-phase flow tubes:** Carburetor passages that may be characterized by single-phase momentum balance equations. They include pressure losses due to friction and accessories, such as bends, expansions and contractions.
- **Two-phase flow tubes:** Small tubes where fuel and air can mix. Appropriate correlations must be used for the characterization of the pressure loss along these tubes.

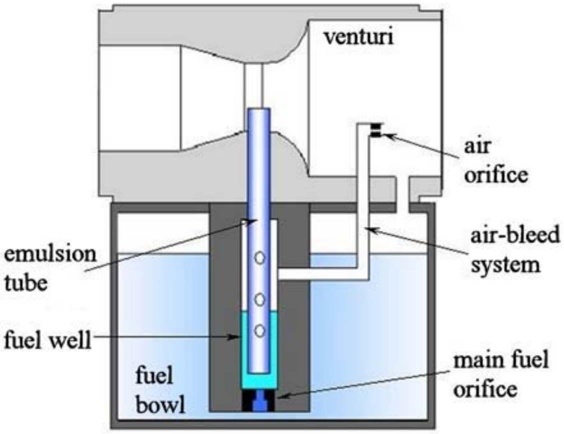


Figure 1. Main circuit of a typical carburetor used in small engines.

The carburetor phenomena can be classified into two functions: metering of the right amount of fuel as function of engine operating conditions, and providing the best mixture quality in the form of droplets, vapor and thin films on the manifold walls. This paper addresses the metering problem by developing and characterizing a theoretical representation of the dynamic flow in the main circuit of small engine carburetors.

The theoretical modeling of carburetor fuel flow is based on the representation of the carburetor circuits as a flow network. One-dimensional momentum balance equations describe the flow across the branches of the flow network, and mass balance equations enforce mass conservation at the network nodes. In the development of this project, a comprehensive literature review on the previous computational models of fuel and air flow in the main circuit of small engine carburetors was initially performed. An improved model was developed, which extended the previously published models by incorporating a detailed review of two-phase flow pressure drop, the effect of the fuel well on the control of air-bleed flow, and dynamic flow. The details of the model can be found in references [1] and [2].

Widely used one-dimensional engine simulation codes lack models for carburetors. The common strategy for modeling fuel delivery from carburetors is based on using a fuel injector with constant air-fuel ratio. The incorporation of a dynamic carburetor model into an engine simulation software package will allow for the study of the combined effect of carburetor parts on engine performance. This paper presents the incorporation of the newly developed dynamic flow network model of the main circuit of small engine carburetors into the commercial simulation code, GT-Power.

DYNAMIC MODEL

The instantaneous one-dimensional momentum balance equation for incompressible flow across a constant area pipe can be written as

$$\frac{\partial u}{\partial t}+\frac{1}{\rho}\frac{\partial P}{\partial x}+\frac{4\tau_w}{\rho D}+g=0, \qquad (1)$$

where τ_w is the shear stress at the wall and D is the tube diameter. By using the Darcy friction factor, f, defined as $f = 8\tau_w/(\rho u^2)$, Eqn. (1) may be written as [3]:

$$\frac{\partial u}{\partial t}+\frac{1}{\rho}\frac{\partial P}{\partial x}+f\frac{1}{D}\frac{u^2}{2}+g=0. \qquad (2)$$

Equation (2) may be used for the characterization of the instantaneous one-dimensional flow in a vertical pipe of constant cross-sectional area.

CHARACTERIZATION OF THE DYNAMIC BEHAVIOR OF A SINGLE-PHASE VERTICAL PIPE

Before applying the instantaneous momentum balance equation to the carburetor model, two concerns were addressed: *i*. What are the characteristics of having this differential equation in the flow network? and *ii*. Which parts should be characterized with this equation: Only the tubes with liquid flow, or those with two-phase flow too?

Figure 2-a shows the configuration of a vertical pipe used for the analysis of a single-phase flow pipe under dynamic conditions. The instantaneous flow rate in this element is given by the solution of Eqn. (2) under the boundary conditions P_{up} and P_{down}. In order to assess the behavior of this element in the flow network, a step function was applied to the static pressure P_{up}, while P_{down} remained constant. The instantaneous relative velocity (the ratio between the instantaneous velocity and the maximum velocity, $v/v_{\max}$) is shown in Figure 3; in this figure, at time t = 1, a sudden decrease in P_{up}, is applied to the pipe. The solution was performed in the Engineering Equation Solver (EES) [4]. The general solution has the shape of an exponential function, reaching a final velocity equal to the solution of Eqn. (2) at steady state conditions. Different flow parameters like density, viscosity, length and diameter were changed in order to see the effect on the time response of the fluid flow inside this pipe.

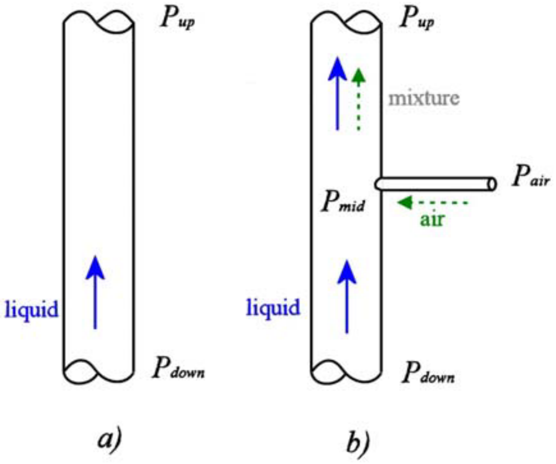


Figure 2. Basic configurations for the study of the dynamic behavior of a flow network element. (a) Single-phase pipe, (b) Two-phase flow network

The comparison between all these cases was based on the calculation of the time constant, τ, defined as the time when the relative velocity reached a value of 0.632 after the step function was applied. Figure 4 shows a comparison of the time constant for the different parameters studied. A greater time constant means that the flow will require more time to reach a steady-state solution. From this figure it was found that:

- Density is the most relevant parameter in the dynamic response of the vertical pipe flow. There is a difference of two orders of magnitude in the time constant of a pipe using a fluid of density 1000 kg/m^3 and the same pipe using air (1 kg/m^3).
- An increase in the diameter or in the length of the pipe results in an increase in time constant. Both of these parameters increase the amount of fluid that must be carried through the pipe, therefore increasing its inertia. The behavior with respect to the diameter may be counter-intuitive. But if laminar flow is assumed, an analytical solution to Eqn. (2) can be written as

$$u(t)=u_{final}\left[1-\exp\left(\frac{-32\mu}{\rho D^2}t\right)\right]. \qquad (3)$$

In this equation, an increase in diameter produces a larger exponential term, and, therefore, a smaller velocity.

- Viscosity plays a minor role, but it was the only parameter that decreased the time constant with an increase in its value. The effect of viscosity on Eqn. (2) is only seen in the friction factor; an increase in viscosity results in a larger friction factor, and therefore it reduces the velocity gradient. Finally, a smaller du/dt results in a larger time constant.

The fact that the dynamic flow in a vertical pipe is affected the most by the density of the fluid has two effects:

1. It implies that in the carburetor flow network, the response of the air passages is two-orders of magnitude faster than that of the liquid parts. Therefore, the air passages might be modeled with an algebraic momentum balance equation and still the dynamic behavior of the flow network will be captured; and
2. The branches in the flow network that have two-phase flow might need to be also characterized with a differential equation, since their time response might be of the same order or magnitude as that of the liquid branches.

CHARACTERIZATION OF THE DYNAMIC BEHAVIOR OF A FLOW NETWORK WITH TWO-PHASE FLOW

Figure 2-b shows the configuration of a simple flow network with a two-phase flow pipe: liquid flows through the lower part of the vertical tube, which is at P_{down}; air travels through the horizontal tube, from an inlet at pressure P_{air}, and enters the vertical pipe at its midpoint, whose static pressure is P_{middle}; a two-phase flow mixture is created in the upper section of the vertical tube, whose end-pressure is P_{up}. For this analysis, the static pressures P_{down} and P_{air} were kept constant, while a step function was imposed on the static pressure P_{up} (i.e., a sudden decrease in P_{up} at $t = 1$).

The horizontal pipe, with only air, was represented with an algebraic momentum balance equation (i.e., the solution of Eqn. (2) at steady state conditions $du/dt=0$). The flow inside the lower section of the vertical tube was represented with the differential equation, Eqn. (2). The fluid properties of the two-phase flow section of the network were calculated using a homogeneous equilibrium model. In this model, the two fluids are assumed to create a mixture of homogenous properties that can be characterized by a mean density and a mean viscosity [5]. The pressure drop across the homogeneous two-phase flow pipe is calculated by using momentum balance equations for single-phase flow. This model has been used in several previous carburetor models (e.g., references [6-9]).

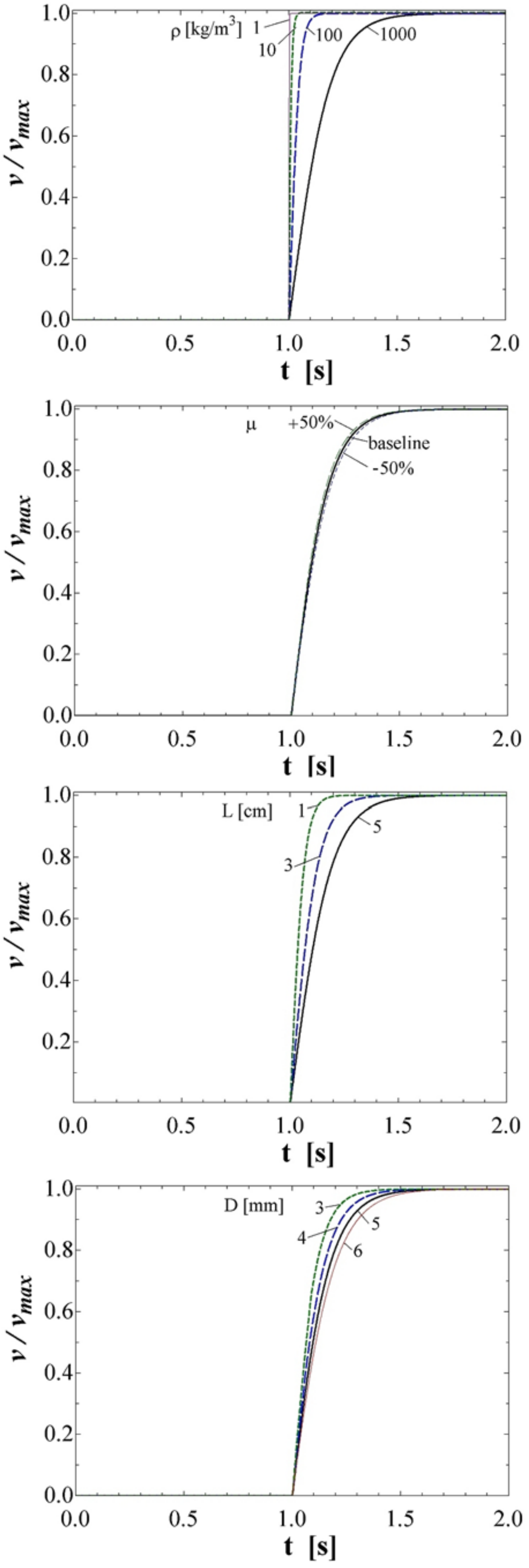


Figure 3. Comparison of the time response of a single-phase vertical pipe, as function of different flow parameters: fluid density, fluid viscosity, pipe length and pipe diameter.

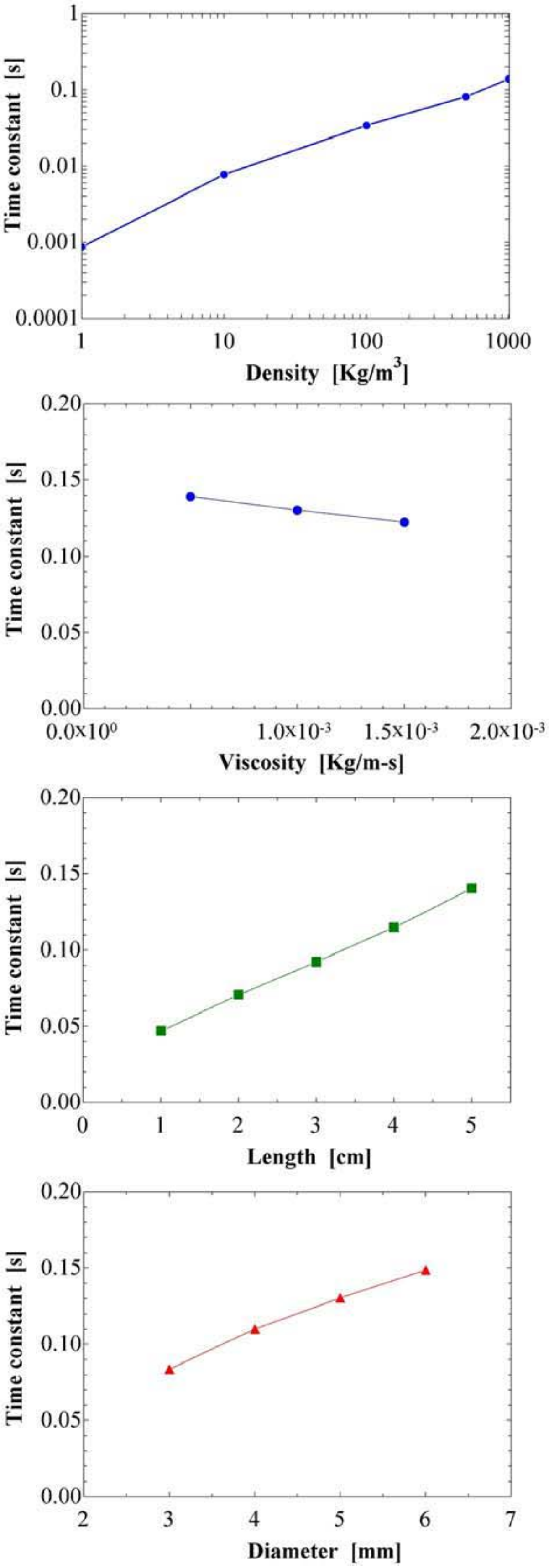


Figure 4. Comparison of the time constant of a single-phase vertical pipe, as function of different flow parameters: density, viscosity, length and diameter.

In order to study the dynamic behavior of the entire two-phase flow network, two characterizations of the upper section of the vertical tube were performed:

1. **Algebraic momentum balance equation.** Under steady state conditions, Eqn. (2) becomes the traditional modified Bernoulli equation with frictional losses. The two-phase flow section was characterized with this algebraic equation using the density and viscosity calculated with the homogeneous two-phase flow model. The lower part of the network was characterized with the instantaneous momentum balance equation in a vertical pipe with liquid only.
2. **Differential momentum balance equation:** The algebraic equation for the two-phase flow part of the network was replaced with the differential instantaneous momentum balance equation, Eqn. (2), using the density and viscosity calculated with the homogeneous two-phase flow model.

Figure 5 shows the effect of liquid density, liquid viscosity, tube length and tube diameter on the time constant of these two characterizations of the two-phase flow element. The trends found in this figure are the same as those seen in the dynamic single-phase vertical pipe, explained in the previous section. The time constant increased with liquid density, pipe diameter and pipe length; these parameters increase the inertia of the fluid and therefore delay the instantaneous response. The time constant decreased with liquid viscosity, as an increase in viscosity results in a larger friction factor, and therefore a decrease in the ability to respond to the pressure variation.

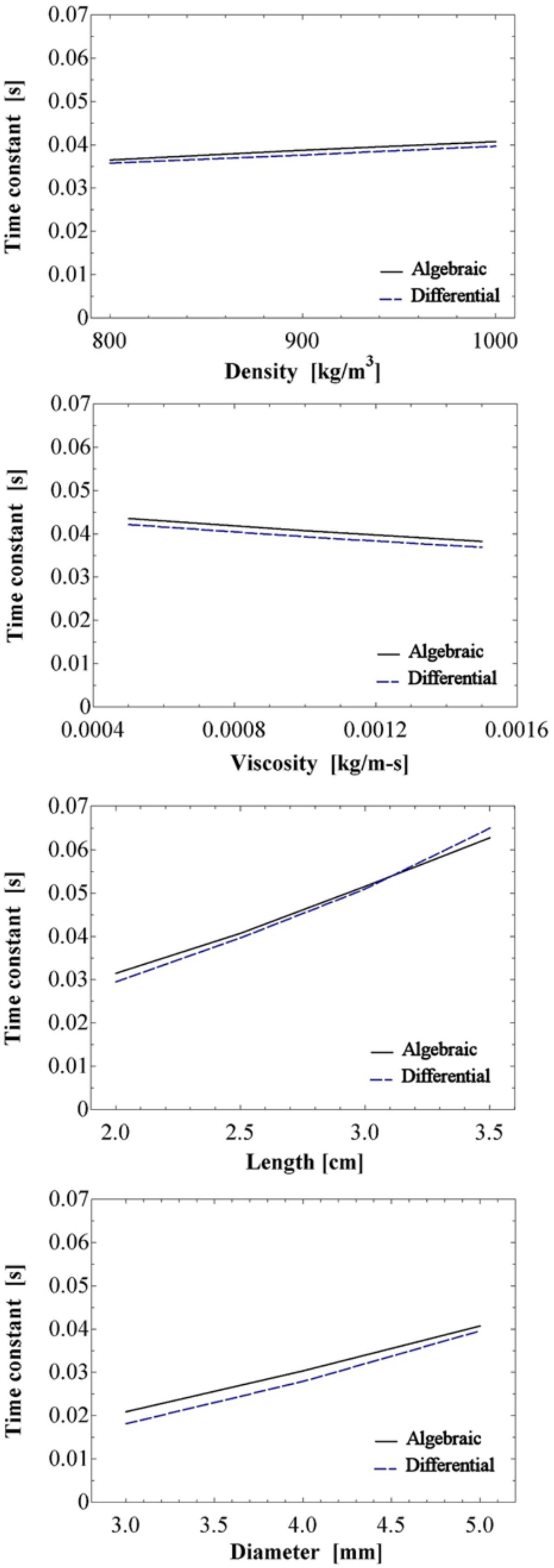


Figure 5. Effect of liquid density and viscosity, and tube length and diameter on the time constant of a flow network with two-phase flow pipe

The dependence of the model solution on the different parameters can be defined as a sensitivity [10]. The relative sensitivity, ε_i, of variable A with respect to parameter k_i is defined as

$$\varepsilon_i = \frac{\partial A}{\partial k_i}\frac{k}{A}. \tag{4}$$

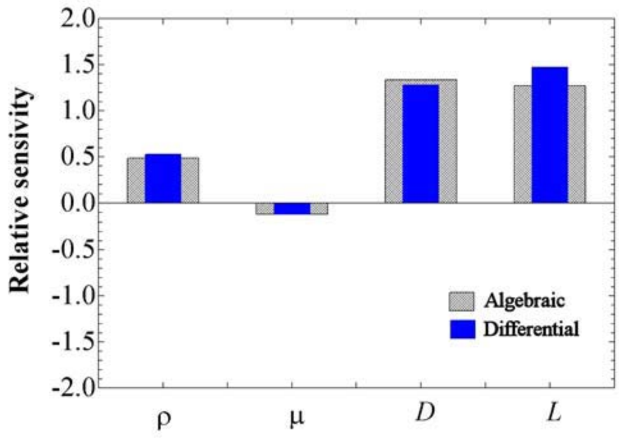


Figure 6. Sensitivity analysis the time constant of a flow network with a two-phase flow tube, with respect to different flow parameters: density, viscosity, pipe length and pipe diameter.

When the results of Figure 5 are shown in a diagram of relative sensitivity (Figure 6), it is clearly seen that both characterizations of the two-phase flow element captured the same trends and influence of the flow parameters. This conclusion indicates that the dynamic behavior of the flow network studied may be captured with the use of a differential equation in the liquid section of the vertical tube and with an algebraic momentum balance equation for the two-phase flow pipe. The use of only one differential equation in the representation of the carburetor flow network reduces substantially the mathematical complexity of the model and its solution, without compromising the ability to capture the dynamic behavior.

In order to assess the effect of using the homogeneous two-phase flow model, the flow network shown in Figure 2-b was modeled with separated-flow models for the density and frictional terms in Eqn. (2). The density was modeled with a correlation developed by Rouhani and cited by Diener and Friedel [11], and the frictional pressure drop was modeled with a correlation developed by Friedel and described by Whalley [12].

Figure 7 shows the difference in mean two-phase flow density predicted by the homogeneous model and Rouhani's correlation as function of mass fraction of the gas in the liquid-gas mixture. The homogeneous model predicts the highest change in mean density. For the main circuit of the carburetor, the mass fractions are below 0.1.

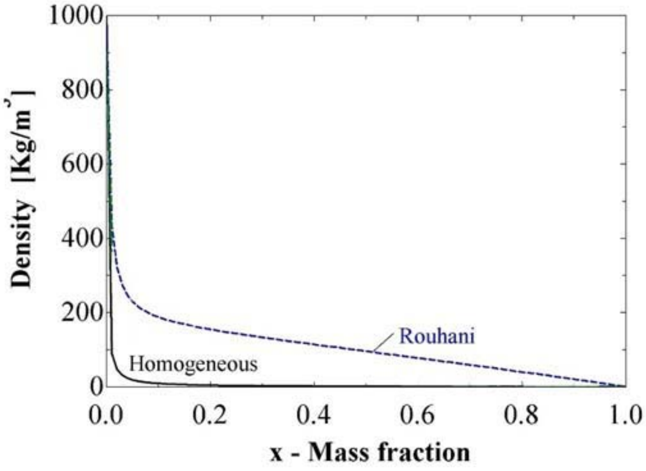


Figure 7. Effect of two-phase flow models on the mean density in a two-phase flow vertical pipe.

Figure 8 shows the predicted frictional pressure drop in a vertical pipe using the homogeneous model and Friedel's correlation. The separated-flow correlation predicts a larger frictional pressure drop than the homogeneous model.

Figure 9 shows the comparison of the time response of a flow network with a two-phase flow pipe modeled with the homogeneous and the separated models. Despite the differences in the density and frictional losses, it was found that the dynamic behavior is very similar among both two-phase flow models. This result strengthens the conclusion that the dynamic behavior of this vertical two-phase flow network can be captured by modeling the liquid section with a differential momentum balance equation.

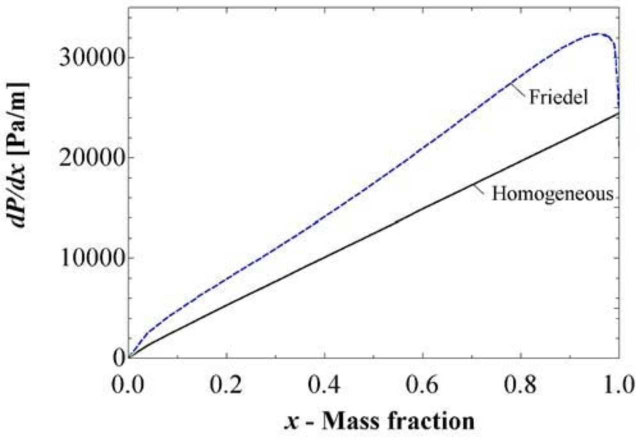


Figure 8. Effect of gas mass fraction on the pressure drop in a vertical tube of 50 cm in length and 6 mm in diameter, using homogeneous model, and Friedel's correlation.

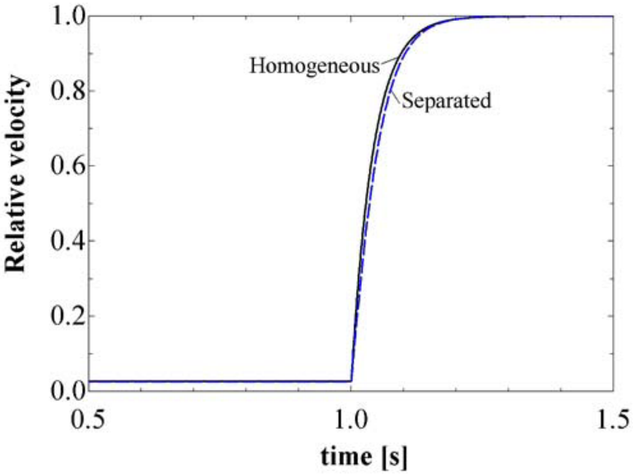


Figure 9. Comparison of the time response of a flow network with two-phase flow pipe, with homogeneous and separated models.

IMPLEMENTATION OF THE CARBURETOR MODEL INTO A ONE-DIMENSIONAL ENGINE SIMULATION SOFTWARE PACKAGE

Both steady-state and a dynamic carburetor models were implemented into the one-dimensional engine simulation software GT-Power [14]. This implementation required that the carburetor model be written in Fortran, and coupled to the GT-Power simulation via user-defined functions. It also required the implementation of the numerical methods for solving non-linear systems of equations. A modified Newton-Raphson method [15] was implemented for solving the non-linear system of equations, and a Crank-Nicolson scheme for the integration of the differential momentum balance equation in the dynamic model.

A schematic of the implementation of the carburetor model in GT-Power is shown in Figure 10-a and -b: GT-Power is responsible for calculating the instantaneous airflow through the intake manifold. This ensures that the gas dynamics inside the intake manifold are captured in the simulation. At each time step, the mass flow rate, the total and static pressures, and the mean air velocity at the inlet of the venturi are captured by 'sensors' and sent to the carburetor model. These are the variables that determine the boundary conditions for the fuel flow network under a steady-state assumption. In addition to these variables, the dimensions of all of the carburetor parts are sent from GT-Power to the carburetor model. This control from GT-Power allows the user to see the effect of geometry on the fuel delivery from the carburetor.

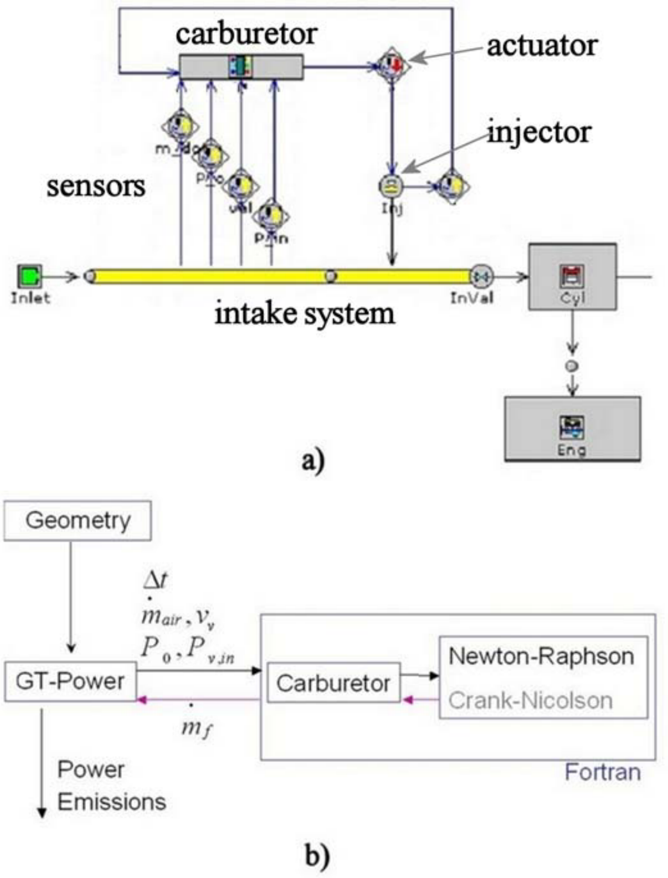


Figure 10. Schematic of carburetor model implementation into GT-Power implementation: (a) GT-Power engine lay-out console. (b) Information flow chart showing the variables that are passed from GT-Power to the carburetor model and vice versa.

The carburetor model solves the non-linear system of equations and calculates the instantaneous fuel flow. This value is returned to GT-Power and is used to control the instantaneous fuel delivered by a fuel injector. GT-Power continues the simulation for in-cylinder combustion.

In the case of the dynamic carburetor model, the instantaneous fuel delivered by the controlled fuel injector at the current time step is sent to the carburetor model. This value is used to perform the integration and calculate the fuel flow in the next time step.

RESULTS

As an example of the results that can be obtained with this implementation, Figure 11 shows the mass flow rate of air predicted by GT-Power for a single-cylinder engine as function of crank-angle. The negative value of the mass flow rate during the compression stroke indicates that, for this configuration of inlet valve timing and gas dynamics, there is a back-flow in the intake manifold.

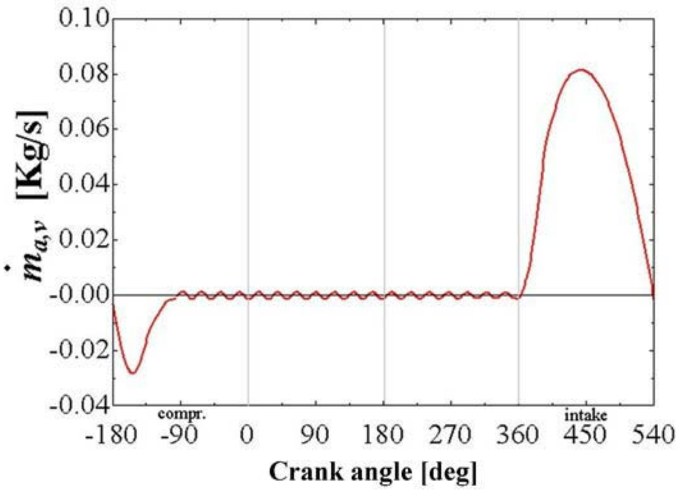


Figure 11. Airflow across venturi for a single-cylinder engine, calculated in GT-Power

The instantaneous fuel flow can be predicted with or without considering the dynamic effects: if the derivative *du/dt* is set to zero in Eqn. (2), the model does not include dynamic effects and is considered a quasi-steady state model. The results of the quasi-steady model, shown in Figure 12, indicate that the instantaneous fuel flow would follow the airflow exactly. In order to check the implementation of the numerical methods, the EES implementation was used as benchmark software. The dashed line in Figure 12 represents the results in EES using the airflow conditions calculated from GT-Power. The solid line is the result of the carburetor model written in Fortran and implemented in GT-Power. The results between the two codes were very similar.

Figure 12 indicates that the main circuit in the carburetor is active under the high airflow conditions during the intake stroke and after the intake valve has closed. This secondary flow represents the backflow under some intake conditions. This rapid change in conditions represented a fuel flow in the carburetor fuel tube that changes from a Reynolds number of 0 to ~5000.

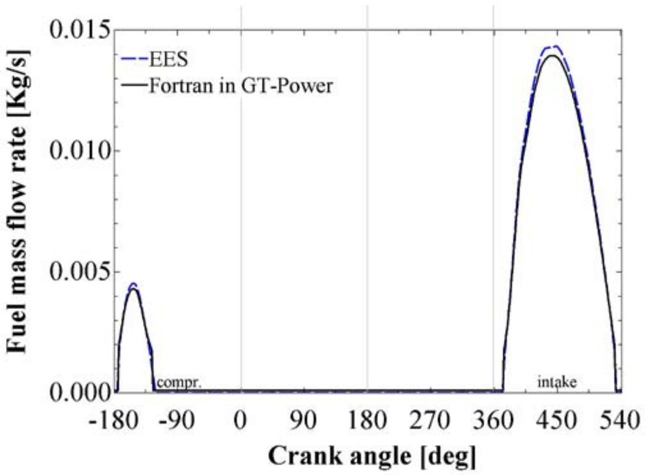


Figure 12. Fuel flow predicted by the implementation of the carburetor model into GT-Power and EES. Steady-state model applied using quasi-steady-state approximation.

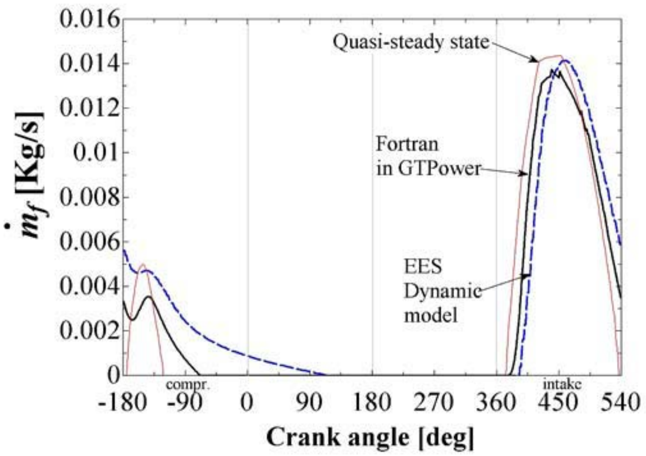


Figure 13. Fuel flow predicted by the implementation of the carburetor model into GT-Power and EES. Dynamic model applied using instantaneous momentum balance equation.

The results of the implementation of the dynamic model in GT-Power are shown in Figure 13. The fuel flow calculated with the dynamic model in EES is shown as a dashed line and the fuel flow calculated from the Fortran routines is shown with a solid line. A slight difference exists between both solutions, but the trends captured in the two cases were the same:

- There is a time delay between the pressure signal at the throat of the carburetor venturi and the instantaneous fuel flow. This delay is the effect of the inertia of the fluid.
- There is a reduced peak fuel flow in the dynamic model, when compared with the quasi-steady state solution. This is the result of the same time delay between the pressure signal and the liquid response.
- Once the pressure at the venturi throat has increased, so there would be no liquid flow under a quasi-steady state solution, there is a roll-off behavior of continuing fluid flow.

The use of the dynamic model of the carburetor resulted in a fuel-enrichment of the mixture delivered to the engine compared with the steady-state model.

This implementation can be used to perform sensitivity analysis of the engine performance with respect to the different parameters in the carburetor. As an example of such an analysis, Figure 14 shows the effect of main fuel orifice diameter, venturi throat diameter, emulsion tube diameter and emulsion tube length on the average air-fuel ratio of a simplified single cylinder engine. The default combustion models in GT-Power were used for these calculations. It was found that the main fuel orifice and venturi throat diameter are the main parameters responsible for the amount of fuel delivered to the engine. It is possible now to capture such effects, which was not possible with the use of an injector using a constant air-fuel ratio in the simulations.

These results showed the feasibility of studying the effect of carburetor parts on air-fuel ratio and engine performance. However, as the objectives of this study did not take into consideration the actual combustion processes inside the cylinder, the limitations of the coupled solution are limited to the prediction of the amount of fuel delivered to the engine. As the combustion models were not modified in the modeled engine in GT-Power, the final engine performance (i.e., emissions and power output) of all of the conditions simulated in Figure 15 followed a single curve as function of air-fuel ratio. This figure shows how the NOx levels and IMEP values given by GT-Power during all of the sensitivity analysis cases in Figure 15 fall on the same curve of air-fuel ratio.

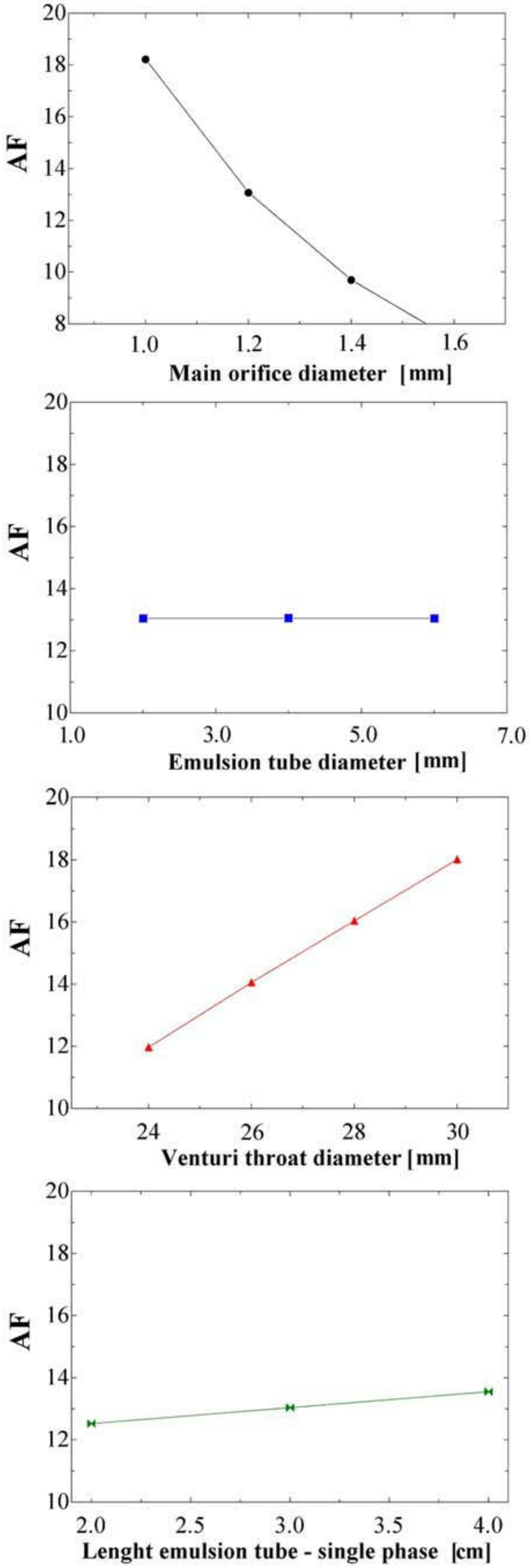
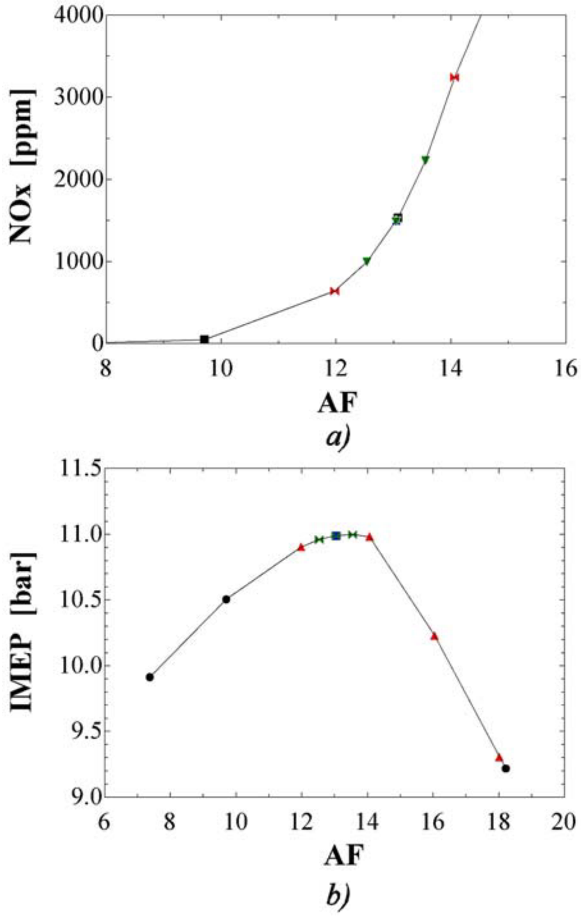


Figure 14. Effect of main orifice diameter, venturi throat diameter, emulsion tube diameter and emulsion tube length on integrated air-fuel ratio.



a)

b)

Figure 15. Results from engine performance parameters from GT-Power as function of air-fuel ratio with carburetor model. (a) NOx emissions. (b) IMEP.

DISCUSSION

The analysis of the dynamic flow inside the carburetor resulted in a theoretical characterization of a dynamic flow network with multiphase flow parts. The characterization of the time response in terms of a time constant showed that the use of a differential momentum balance equation for the characterization of the instantaneous flow inside a vertical pipe was appropriate for capturing the dynamic behavior of the entire flow network. This conclusion resulted in a significant reduction in level of complexity of the mathematical model of the emulsion tube; if the two-phase flow sections required a differential equation for its characterization, this would have resulted in a more complex system to model and solve.

The sensitivity analyses showed that the diameter and the discharge coefficient of the main fuel orifice and the venturi throat are the parameters that affect the fuel flow predicted by the steady-state and the dynamic models the most. This is knowledge that has been available to technicians and engine enthusiasts for a long time, but the fact that it is possible to have these results in a mathematical model will allow for the study of these parts on engine performance. The importance of these two parts encourage the detailed analysis of their behavior in a CFD package.

The implementation in GT-Power was a step towards the usability of this model for the prediction of the effect of carburetor parts on engine performance. A constraint found in GT-Power, and that must be addressed in the continuation of these studies, is the use of different in-cylinder combustion models that may be more sensitive to the air-fuel mixture quality delivered by the carburetor.

Future work should be focused towards the experimental validation of the results and the implementation of additional carburetor circuits, such as the idle and transitional systems. Possible extensions of the dynamic model are its use on sudden changes in load and the coupling with the gas dynamics of specific intake manifold designs.

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REFERENCES

1. D.A. Arias, and T.A. Shedd, Numerical and Experimental Study of Fuel and Air Flow in Carburetors for Small Engines. SAE Technical Paper 2004-32-0053, 2004.
2. D.A. Arias, *Numerical And Experimental Study of Air And Fuel Flow In Small Engine Carburetors*. Ph.D. Thesis, University of Wisconsin – Madison, Mechanical Engineering Dept., 2005.
3. F. J. Moody. *Introduction to Unsteady Thermofluid Mechanics.* John Wiley & Sons, 1990.
4. S. A. Klein. Engineering Equation Solver (EES). http://www.fchart.com. F-Chart Software © 1992-2005.
5. G. B. Wallis. *One-dimensional Two-Phase Flow.* McGraw Hill, 1969.
6. D. L. Harrington and J. A. Bolt. Analysis and digital simulation of carburetor metering. SAE technical report, (700082), 1970.
7. K. Shinoda, H. Koide, and A. Yii. Analysis and experiments on carburetor metering at the transition region to the main system. SAE technical report, (710206), 1971.
8. Y. Asano, T. Chuma, H. Haga, and T. Mochida. Effects of air-bleed systems on carburetor performance. Bulletin of JSME, 11(46):691–698, 1968.
9. M. Furuyama. Fuel supply characteristics in the transition region of fixed-venturi type carburetor. Bulletin of JSME, 24(191):842–848, 1981.
10. J. Warnatz, U. Mass, and R. W. Dibble. *Combustion.* Springer, 2nd edition, 1999.
11. R. Diener and L. Friedel. Reproductive accuracy of selected void fraction correlations for horizontal and vertical upflow. *Engineering Research*, 64(4-5):87–97, 1998.
12. P. B. Whalley. *Boiling, Condensation and Gas-Liquid Flow*. Oxford University Press, New York, 1987.
13. P. J. Moss. Pulsation enrichment in carburetters. *Automotive Engineer*, pp. 53–56, 1980.
14. Gamma Technologies. GT-Power: User's manual and tutorial. Gamma Technologies, 2001.
15. W. H. Press, S. A. Teukolsky, W. T. Vetterling, and B. P. Flannery. *Numerical recipes in Fortran 77*, Vol1. Cambridge University Press, 2nd edition, 1992.

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NOMENCLATURE

D	Diameter [m]
f	Friction factor
g	Gravitational constant [m^2/s]
P	Pressure [Pa]
t	Time [s]
u, v	Mean velocity [m/s]
x	Mass fraction of gas in the two-phase flow mixture
ε_ι	Relative sensitivity
μ	Viscosity [kg/m-s]
ρ	Density [kg/m^3]
τ	Time constant [s]
τ_W	Wall shear stress [Pa]