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Multi-Objective Optimization of the Natural Gas-Air Venturi-Type Mixer for Bi-Fuel Engines

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Abstract

In the conversion of gasoline engines to natural gas, the mixer plays a crucial role in engine performance. This study presents a three-dimensional numerical simulation and geometric optimization of a natural gas-air Venturi-type mixer for a bi-fuel engine. The parametric study revealed that among the examined geometric parameters, increasing the number of orifices from 2 to 8 had the most significant impact, reducing the air-to-fuel ratio by 71% and increasing the uniformity index by 122.7%. Additionally, increasing the orifice diameter from 1 to 2.5 mm had a greater effect on the air-to-fuel ratio (44% decrease), while increasing the throat diameter from 17 to 20 mm had a greater effect on the uniformity index (17.9% decrease). A multi-objective genetic algorithm was employed to minimize deviations of the air-to-fuel ratio from the stoichiometric value and to maximize the uniformity index. The optimization results showed that in the geometry with the lowest deviation from the stoichiometric air-to-fuel ratio, this deviation and the uniformity index decreased by 99% and 39.7%, respectively, compared to the base-mode. Conversely, in the geometry with the highest uniformity index, this parameter increased by 16%, while the air-to-fuel ratio deviation increased by 32.6% relative to the base-mode.

Keywords: Mixer, Numerical simulation, Optimization, Air-to-fuel ratio, Uniformity index.

1. Introduction

Gasoline and diesel internal combustion engines are widely used in various sectors, including automotive transportation, agricultural machinery, industrial tools, and electric power generators. However, environmental and economic issues arising from the consumption of these fuels have underscored the necessity of considering alternative fuels as a viable solution [1]. Among these alternatives, natural gas has emerged as an attractive option for engines due to its unique properties, such as abundance, cost-effectiveness, high auto-ignition temperature, lower density compared to air, high octane number, cleaner combustion, and favorable environmental characteristics [2].

In terms of how two different fuels are utilized, gas-fueled engines can be categorized into dual-fuel and bi-fuel engines. The main difference between these two lies in the simultaneous or alternate use of fuels. Dual-fuel engines are typically based on diesel engine designs, using natural gas as the primary fuel, with diesel serving as a pilot fuel to initiate natural gas ignition. These engines operate on the compression ignition cycle. Conversely, bi-fuel engines are derived from gasoline engine designs and can operate independently on either gasoline or natural gas. In such engines, the gas fuel supply system substitutes for gasoline, allowing either fuel to independently power the engine.

To utilize natural gas, modifications in the fuel supply system are required. For this purpose, components enabling the use of gaseous fuel in gasoline engines are installed, collectively referred to as a *conversion kit*. Conversion kits can be categorized into four generations. First-generation kits are mechanical and simple, with their key component being the natural gas-air mixer, which blends gas and air at the carburetor inlet and must be designed specifically for each engine.

One of the critical parameters affecting engine performance and emissions is the air-fuel ratio (AFR), defined at the stoichiometric point when sufficient air is present for complete combustion. Furthermore, the homogeneity of the air-fuel mixture significantly influences engine performance. A homogeneous mixture improves combustion efficiency and reduces emissions, whereas an inhomogeneous mixture can cause incomplete combustion and overall performance degradation. Consequently, the design of mixers capable of producing a uniform air-fuel mixture close to the stoichiometric ratio is of great importance.

Given the crucial role of the mixer in the performance of gas-fueled engines, numerous studies have been conducted on the design and evaluation of air-fuel mixers, with Venturi-type mixers enjoying considerable popularity among researchers. For instance, Yusaf et al. [3] numerically investigated compressed natural gas usage in gas engines by analyzing methane-air flow behavior in a mixer using Computational Fluid Dynamics (CFD). In another study, Das et al. [4] simulated a Venturi-type mixer for a dual-fuel engine using ANSYS software for five combinations of biogas and producer gas, analyzing flow behavior, pressure drop, velocity, and turbulence to assess mixing quality. Among experimental studies, Ariani et al. [5] investigated the effect of employing a mixer in the intake manifold of a diesel-compressed natural gas dual-fuel engine under partial load conditions.

A review of the existing literature on natural gas-air mixers shows that most prior research has only examined the influence of a limited number of geometric parameters and has not employed optimization approaches, particularly multi-objective optimization. In the present study, a geometric optimization of a Venturi-type mixer for a bi-fuel engine intended for electric power generation was performed using a multi-objective optimization approach. The two primary objectives of this design are to maximize mixture homogeneity and to accurately adjust the air-fuel ratio. To achieve this, geometric optimization of the mixer was investigated, using three-dimensional CFD simulations in ANSYS Fluent.

2. Geometric Model of the Mixer

In this study, the airflow and natural gas flow within a Venturi-type mixer are investigated. The Venturi mixer consists of four main sections: a converging section, a throat section, a diverging section, and a gas injection section. The geometry was modeled to examine the influence of various parameters, including throat length (L_{th}), converging angle (α), diverging angle (β), gas orifice diameter (D_o), number of orifices (N), throat diameter (D_{th}), and gas distribution ring thickness (t). Figure 1 illustrates the geometry of the mixer, the defined geometric parameters, the labeled planes, and the basemode dimensions. It should be noted that plane P3 is located at the end of the diverging section.

3. Numerical Simulation

3.1 Governing Equations

In this simulation, the working fluids are air and methane. The flow is modeled as steady-state, non-reactive, and is treated using the species transport model. The numerical solution is based on the continuity and Navier-Stokes equations in a three-dimensional, steady, incompressible form. The Standard k-ε turbulence model, previously reported as suitable for gas-air mixer simulations, is adopted in this study. To predict the distribution of gas species in the flow field, the species transport model is employed, which allows the tracking of the mass fraction (Y) of each species by solving the convection-diffusion equation. The air-fuel ratio (AFR) for internal combustion engines is calculated using Eq. (1), where the stoichiometric AFR for methane is 17.2. Additionally, the uniformity index (UI), used to evaluate mixture homogeneity, is calculated according to Eq. (2). The UI ranges from 0 (no mixing) to 1 (perfect mixing).

$$AFR = \frac{\dot{m}_a}{\dot{m}_f} \quad (1)$$

$$UI = 1 - \frac{1}{2}\frac{\sum_{j=1}^{n}|Y_j - Y_{mean}|a_j}{aY_{mean}} \quad (2)$$

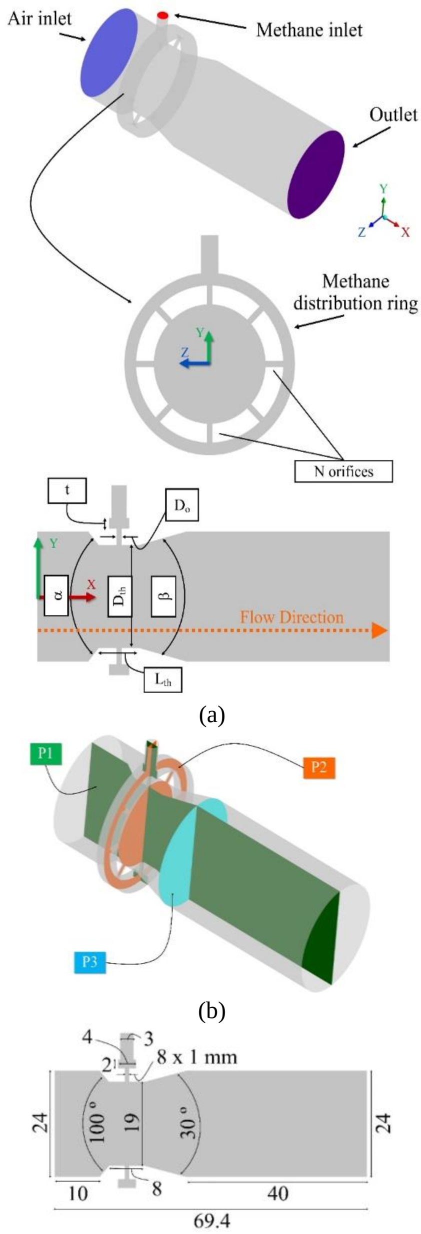

Figure 1. Geometry of the Venturi-type mixer: (a) Isometric and sectional views, (b) Labeled planes, and (c) Basemode dimensions in millimeters.

3.2 Solution Method

The flow is assumed incompressible; thus, a pressure-based solver is used, as is common for incompressible flows. The finite volume method is implemented in ANSYS Fluent 2019. A coupled scheme is applied for pressure-velocity coupling, a cell-based least-squares method is used for gradient discretization, second-order discretization for pressure, and second-order upwind schemes for momentum, turbulent kinetic energy, and turbulence dissipation rate.

Boundary conditions were obtained by simulating the target engine in GT-Suite under one-dimensional modeling. The engine operates at 1500 rpm, with ambient pressure assumed at 100 kPa. Air enters the mixer from the environment, regulated by a throttle valve. The inlet airflow boundary condition is mass flow inlet, with a mass flow rate of 0.001827 kg/s and an air mass fraction of 1. Natural gas (methane) is supplied directly from Iran's urban gas network, with an inlet pressure of 1723.69 Pa (0.25 psi) and an air mass fraction of 0. The outlet is defined as a pressure outlet with a relative pressure of -293.35 Pa. All solid walls are considered rigid with a no-slip condition, and diffusive flux on the walls is assumed zero.

4. Results and Discussion

4.1 Validation of Results

To validate the numerical results, the present simulation outcomes were compared with those from the numerical study by Yusaf et al. [3]. For this purpose, the mixer geometry was recreated according to the reference study, which featured four methane injection orifices. Figure 2 presents the comparison between the methane mass fraction along the mixer length (from the beginning of the converging section to the end of the diverging section) obtained in this study and the reference data [3]. A satisfactory agreement is observed, with a maximum relative error of 21.8%. This discrepancy may be attributed to minor geometric differences between the two models.

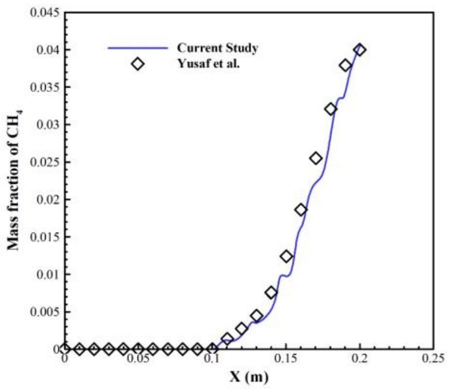

Figure 2. Comparison of methane mass fraction values along the mixer with the reference numerical results [3].

4.2 Basemode Case Analysis

The basemode mixer simulation shows that the methane mass flow rate at the inlet is 0.000195 kg/s, the AFR is 9.385, and the UI at plane P3 is 0.78. The AFR for the basemode case is lower than the stoichiometric value of 17.2 for methane; hence, the basemode mixer delivers a fuel-rich mixture to the cylinder. To achieve improved performance for the target mixer, a geometric optimization is deemed necessary.

5. Optimization

The design variables in this study are the geometric parameters: L_{th}, α, β, D_o, N, D_{th}, and t. The objective functions are minimization of AFR deviation from the stoichiometric value for methane (Eq. 3), and maximization of the uniformity index of methane mass fraction at plane P3.

$$DFS = |AFR - 17.2| \tag{3}$$

In Equation (3), DFS denotes the deviation of the AFR from the stoichiometric value of 17.2. Overall, the objective functions and the design variables are defined according to Equation (4).

$$\begin{cases} Min:\ DFS = f_1(L_{th},\ \alpha,\ \beta,\ D_o,\ N,\ D_{th},\ t) \\ Max:\ \ UI = f_2(L_{th},\ \alpha,\ \beta,\ D_o,\ N,\ D_{th},\ t) \\ Subject\ to:\qquad 4 \le L_{th} \le 10\ (mm) \\ \qquad 60 \le \alpha \le 120\ (degree) \\ \qquad 20 \le \beta \le 50\ (degree) \\ \qquad 1 \le D_o \le 2.5\ (mm) \\ \qquad N = 2, 3, 4, 5, 6, 7\ and\ 8 \\ \qquad 17 \le D_{th} \le 20\ (mm) \\ \qquad 1 \le t \le 2.5\ (mm) \end{cases} \tag{4}$$

Optimization was conducted under the boundary conditions corresponding to 1500 rpm, with an initial population size of 100. The multi-objective genetic algorithm converged after 483 iterations. Figure 3 shows the Pareto-optimal solutions, together with all evaluated points and the Pareto front. The Pareto set comprises 31 solutions, from which the designer can choose according to specific performance priorities. Three points, labeled A, B, and C in Figure 3, were selected for further analysis.

Points A and C are located at the extremes of the Pareto front, while B is the intermediate point chosen via the K-means clustering method. Table 1 presents the design variables and objective function values for the basemode, A, B, and C cases. Point A yields the best AFR deviation (DFS) but the lowest UI, whereas point C yields the highest UI but the largest AFR deviation.

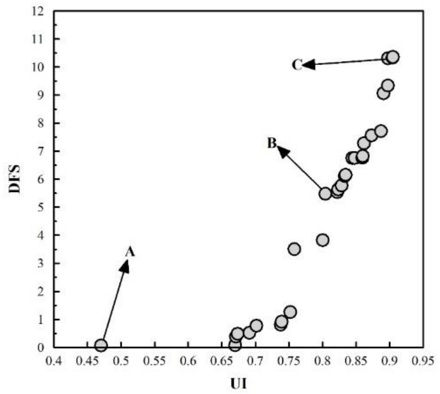

Figure 3. Pareto optimization results.

Table 1. Information for the points labeled on the Pareto front plot.

Mode	Basemode	A	B	C
L_{th} (mm)	8.000	9.430	6.713	9.384
a (degree)	100.0	116.2	72.3	81.4
b (degree)	30.0	39.6	26.3	21.5
D_o (mm)	1.000	1.356	1.012	1.245
N	8	2	6	8
D_{th} (mm)	19.000	19.501	17.060	17.059
t (mm)	2.000	2.331	1.193	2.337
AFR	9.385	17.273	11.717	6.839
DFS	7.815	0.073	5.483	10.361
UI	0.7802	0.4702	0.8043	0.9048

Methane distribution in planes P1, P2, and P3 for points A, B, C, and the basemode case is illustrated in Figure 4. As seen in Figure 4c, the mixer geometry corresponding to point C produces the most homogeneous methane-air mixture, while point A produces the least homogeneous mixture. The UI values in Table 1 confirm these observations. Conversely, point C shows the largest deviation from the stoichiometric AFR, while point A provides the closest AFR to the stoichiometric value.

6. Conclusion and Summary

In this study, the performance of a Venturi-type natural gas-air mixer in a spark-ignition bi-fuel engine was investigated using CFD simulations. To determine the required boundary conditions for the three-dimensional mixer simulations in ANSYS Fluent, a one-dimensional engine model was first developed in GT-Suite. The basemode mixer performance was then analyzed.

Subsequently, employing a multi-objective genetic algorithm, the mixer geometry was optimized with two primary objectives, minimization of the AFR deviation from the stoichiometric value and maximization of the uniformity index.

Analysis of selected points from the Pareto front revealed that the geometry yielding the lowest AFR deviation (0.073) had the poorest UI (0.4702). Conversely, the geometry with the highest UI (0.9048) exhibited the largest AFR deviation (10.361). Therefore, the optimal design selection depends on the specific operational requirements and performance priorities of the engine.

On one hand, precise AFR control is essential for achieving stable combustion, desired power output, and reduced emissions. On the other hand, mixture inhomogeneity can cause uneven cylinder distribution, irregular combustion, knocking, vibration, and, in the long term, engine damage. Hence, the uniformity index plays a crucial role in ensuring smooth operation and engine reliability. However, considering that the engine studied here is single-cylinder, reducing AFR deviation was prioritized over maximizing mixture homogeneity.

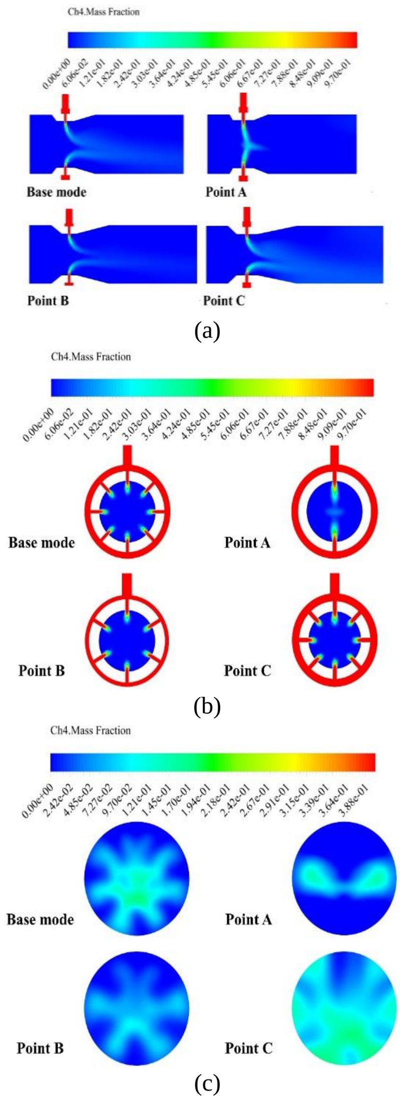

Figure 4. Methane mass fraction in planes (a) P1, (b) P2, and (c) P3 for points A, B, C, and the basemode case.

7. References

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