

Piston Design Using Multi-Objective Reliability-Based Design Optimization

2010-01-0907

Published
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ABSTRACT

Piston design is a challenging engineering problem which involves complex physics and requires satisfying multiple performance objectives. Uncertainty in piston operating conditions and variability in piston design variables are inevitable and must be accounted for. The piston assembly can be a major source of engine mechanical friction and cold start noise, if not designed properly. In this paper, an analytical piston model is used in a deterministic and probabilistic (reliability-based) multi-objective design optimization process to obtain an optimal piston design. The model predicts piston performance in terms of scuffing, friction and noise. In order to keep the computational cost low, efficient and accurate metamodels of the piston performance metrics are used. The Pareto set of all optimal solutions is calculated allowing the designer to choose the “best” solution according to trade-offs among the multiple objectives. Our multi-objective optimization method is demonstrated with simple examples and applied in the design of an automotive piston.

1. INTRODUCTION

While satisfying the ever increasing customer expectations and regulatory requirements, product innovation and reduction in the product development time are two key enablers for success in the highly competitive automotive industry. Numerical design optimization methods combined with analytical methods can be used to obtain optimal engineering designs with a considerably reduced product development time. The application of optimization methods in piston design is still evolving. Most engineering design problems, including piston design involve complex physics

and require the satisfaction of multiple objectives simultaneously. To meet the current trends of faster product development of I.C. engines, deterministic and probabilistic optimization of the piston, considering secondary dynamics and skirt to liner contact, can be a very useful design tool. The piston design variables affecting these phenomena also affect key performance measures such as noise, friction, and durability of power cylinder components. Current trends in I.C. engine design are continuously moving towards higher specific power with lower friction in an effort to improve overall vehicle fuel efficiency. Also, high durability and low noise are major design criteria which aim towards a low warranty cost and customer satisfaction. In recent years, increasing importance has been given to analytical piston design methods in order to simultaneously reduce the turn-around time and the development cost.

The objective of this work is to obtain optimal piston designs using deterministic and probabilistic multi-objective design optimization to simultaneously optimize for friction, noise and scuffing. In deterministic multi-objective optimization where uncertainty and variation are ignored, the so-called Pareto set provides all optimal solutions. The designer can then select the solution which provides the best trade-offs among all objectives based on his/her subjective judgment. If uncertainty is considered, a “reliable” Pareto set can be obtained which sacrifices some optimality in order to accommodate the inherent uncertainty. Both deterministic and probabilistic multi-objective optimization is used in this paper to obtain an optimal piston design deterministically or under uncertainty.

Deterministic mathematical optimization has led to greatly improved performance in all areas of engineering design. It

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can however, lead to undesired choices if uncertainty/variation is ignored. In deterministic design we assume that there is no uncertainty in the design variables and/or modeling parameters and hence no variability in the simulation outputs. However, there exists inherent input and parameter variation that results in variation in outputs. Deterministic optimization typically yields optimal designs that are pushed to the limits of design constraint boundaries, leaving little or no room for tolerances in manufacturing, or uncertainty in operating conditions. Therefore, deterministic optimal designs that are obtained without taking into account uncertainty are usually unreliable.

In contrast to deterministic optimization, Reliability-Based Design Optimization (RBDO) accounts for uncertainty and variation using probability distributions to describe the stochastic nature of the design variables and model parameters. Variations are represented by standard deviations (typically assumed to be constant), and a mean performance measure is optimized subject to probabilistic constraints. RBDO can be a powerful tool which can assist in design under uncertainty, because it provides optimal designs in the presence of uncertainty in design variables/parameters and simulation models. For this reason, it has been extensively studied [1,2,3,4,5,6,7]. In optimization under uncertainty a desirable task is to minimize (or maximize) one or more objectives while (1) all constraints are satisfied and, (2) the performance of the design is insensitive to the existing variation or uncertainty. In this work, a design is called reliable if it meets all performance targets in the presence of variation/uncertainty. Usually some of the deterministic optimality is traded off in order for the design to be reliable. Multi-objective optimization is used if multiple, often competing, objectives must be optimized simultaneously. In this case, we have multiple optimal solutions which form the so-called Pareto set. In this paper, we use multi-objective optimization to obtain the Pareto set of the piston optimal designs both deterministically and probabilistically. In the latter case, we calculate all reliable optimal designs.

Section 2 of the paper gives an overview of piston design and briefly describes the development of a physics-based analytical model for piston design. Non-linear dynamic equations of motion are derived describing the piston rigid-body dynamics and skirt flexibility. The Newmark-beta time integration scheme combined with a Newton-Raphson linearization approach is used to solve the equations of motion. Subsequently, the piston performance measures of scuffing, friction, and noise are calculated as output. Section 2 also describes the piston design variables and provides information on the metamodels of the three piston performance measures. Section 3 provides an introduction to deterministic single-objective and multi-objective optimization. Our approach to solve the multi-objective optimization problem is described and demonstrated with a simple mathematical example in section 4 and is then used

in piston design. Section 5 provides some background on the classical RBDO methods and extends the deterministic multi-objective approach to account for uncertainty. Our multi-objective RBDO method is demonstrated with a simple mathematical example and is subsequently, applied in piston design. A summary and conclusions are provided in section 6.

2. PISTON DESIGN

Engine NVH (Noise, Vibration, and Harshness) is an important design issue for internal combustion engines because it plays a crucial role on how a customer perceives engine quality. Also, efforts to reduce engine vibration and radiated noise while improving durability, fuel economy and reliability have become increasingly important in the automotive industry due to requirements for high performance, low weight, low cost, and fast-to-market engine designs. It is well known that the piston assembly can be a major source of engine mechanical friction and cold start noise, if not designed properly. It provides a critical lubricated interface and reduces scuffing which improves engine durability. The most important piston design requirements are high scuff resistance, low friction and wear, low noise, and high durability and fatigue resistance. The optimal piston skirt profile is determined so that the maximum contact pressure at the piston-bore interface is kept low in order to avoid scuffing. Reduced friction and low piston slap noise are used as additional design requirements.

There have been numerous parametric studies [8,9,10] on piston design in which each design variable is changed individually and its effect on friction, noise, and/or scuffing performance is calculated. Some of these studies are purely parametric investigating the effect of change in one variable at a time. They do not consider the interaction effects among the design variables, and may therefore lead to incorrect conclusions. Even when the interaction effects are accounted through appropriate design of experiments methods, a sensitivity analysis may fail to provide a definitive design direction.

The piston operates under very high pressure and temperature conditions. Its main functions are power transmission, sealing, and pumping. The relative motion between the piston and the cylinder liner affects friction and hence the efficiency of an I.C. engine. The piston impact on the liner increases the vibration level of the engine block surface which may degrade the NVH performance. The repetitive impact of the piston on the liner can significantly reduce the life of the piston and affect negatively the customer expectation for a durable engine. The multiple requirements for NVH performance, durability and fuel economy must be accounted for in piston design. The parametric design study in [11,12,13,14] showed that various piston design variables and engine operating parameters have a significant effect on

piston friction and noise. In addition some of these parameters, such as nominal cold clearance between the skirt and liner may have a negative effect on friction, noise and scuffing. The balancing of multiple conflicting performance requirements in piston design should be therefore, addressed through multi-objective optimization where the optimal trade-offs can be shown in a Pareto set.

The piston operates under varying conditions of speed and loading. In addition, its design is subjected to variability and uncertainty in modeling and simulation, testing, manufacturing processes, material properties, design variables and parameters. A deterministic approach does not account for the variability and uncertainty. For this reason, a safety factor approach is common. However, it can significantly compromise the optimality and lead to over design. A Multi-Objective Reliability-Based Design Optimization (MO-RBDO) is used in this paper to address the trade-offs among different performance metrics in piston design and simultaneously consider the inherent variation.

There are various shapes of piston designs. Some pistons are designed with a taper, with the largest clearance at the top end of the piston which allows it to compensate for thermal expansion. Satisfying piston design for noise, friction, and scuffing can be very challenging. The piston barrel shape is a useful design tool to improve the friction, noise and scuffing performance. The intention of a barrel is for the bottom of the piston to contact the thrust side of the bore first and then allow the remaining portion of the skirt to tilt against the bore until the piston is fully transferred across to the thrust side. This prevents any hard impact which would generate noise. Also, due to the barrel shape, the side load on the piston skirt gets distributed more evenly resulting in a much wider contact pattern and lower skirt pressure. This ultimately improves the scuffing propensity. Barrel shaped piston skirts have been around for some time and are used in most engines. Apart from enabling a smaller piston to bore clearance, barrel shaped pistons reduce friction because of the oil wedge formed between the piston and the bore. Hence, the piston barrel shape must be optimized in order to meet the friction, noise and scuffing design objectives.

Early piston designs had circular cross sections across the length of the piston resulting in an uneven deformation under thermal and mechanical loading. Due to the non-uniform thickness distribution along various parts of the piston and the variation in loading, the deformation along the skirt is not uniform. Because of the piston tilt around the piston pin, an oval piston shape provides much larger contact area which reduces the contact pressure for a given side load and moment acting on the piston. Furthermore, under thermal expansion of the piston, ovality provides additional clearance to parts of the skirt.

In recent years, there is an increasing urgency among engine manufacturers to design and produce low noise engines. Piston noise, while not a significant contributor to the overall noise level, is a major cause of customer dissatisfaction because it is readily detected by the human ear. There is a small clearance between the piston and the cylinder bore of about 15-20 μm . Despite the small clearance, the piston undergoes a secondary motion within the bore, which can generate unwanted sound and vibration. The piston secondary motion is caused by the piston side thrust, which changes direction due to the connecting rod angularity. As a result, the piston moves laterally from one side of the bore to the other and also tilts within the bore clearance. It eventually collides against the cylinder wall. This impact is commonly known as “piston slap.”

Piston slap is most prominent near the top dead center and under cold engine conditions when the skirt-bore clearance is very high. The slap can not be eliminated completely although it can be significantly reduced with a good piston skirt profile. The combination of a deformed cylinder with a deformed, profiled piston (oval and barrel shapes) defines the clearance at running condition. A good profile can reduce the secondary motion to a great extent thus reducing piston slap significantly. This is because the barrel shape of the skirt lets the piston “glide” over from one side to the other as the connecting rod changes angularity and consequently changes piston secondary motion. A good skirt profile also helps the piston roll over smoothly, thus minimizing the piston tilt and hence the piston slap. An optimized profile also keeps the contact pressures within acceptable limits, reducing therefore the risk of skirt scuffing. For this reason, it is very important that the skirt profile design be as optimal as possible.

Piston scuffing is one of the main mechanisms affecting the durability of automotive engines. The effort toward lower fuel consumption and high power places a limit on the minimum weight of a piston, in order to avoid compromising the structural rigidity in the piston skirt areas, and avoid therefore, a mechanical failure of the piston skirt. The engine piston is one of the most complicated components to design due to the multiple objectives which are often conflicting. Customer demand for higher durability and less maintenance puts emphasis on piston skirt scuffing. A considerable increase in friction accompanied by a rapid rise in temperature [15, 16] is considered to be the primary reason for scuffing.

Having a test based approach to achieve an optimum piston design is very difficult due to limited availability of prototypes during the early design stages and the requirement for reduced product development time. Therefore, an accurate and efficient analytical method must be established as an alternative. An analytical model has been developed in [13, 14] and used in this study to simulate piston secondary dynamics and piston-bore contact for an asymmetric half

piston model. The piston secondary motion and piston-skirt contact pattern determine the piston NVH characteristics because they affect the skirt-to-bore impact force and therefore, how the piston impact excitation energy is damped, transmitted and eventually radiated from the engine structure as noise. The analytical model includes several important physical attributes such as bore distortion effects due to mechanical and thermal deformation, inertia loading, piston barrelity and ovality, piston flexibility and skirt-to-bore clearance. The model accounts for piston kinematics, rigid-body dynamics and skirt flexibility. An accurate and yet efficient time integration algorithm is used based on a modified Newmark-Beta method for nonlinear systems. A contact algorithm using the complementarity method, determines the contact characteristics between the flexible piston and a rigid liner during the engine cycle. The following section provides the basics of the kinematic and dynamic formulations and the solution method.

2.1. PISTON KINEMATICS AND DYNAMICS

A two-dimensional lumped parameter model of the crank-slider assembly is shown in [Figure 1](#), including the crankshaft and piston pin offsets. The model describes the translational (lateral) piston motion along the (X)-axis, and the rotational motion around the piston pin axis (Y)-axis. The lateral and rotational displacements constitute the piston's secondary motion. The piston displacement, velocity and acceleration in the primary direction are calculated using detailed kinematics of the crank-slider mechanism, including the piston pin offset. The primary kinematics is used to calculate the inertia forces on the piston.

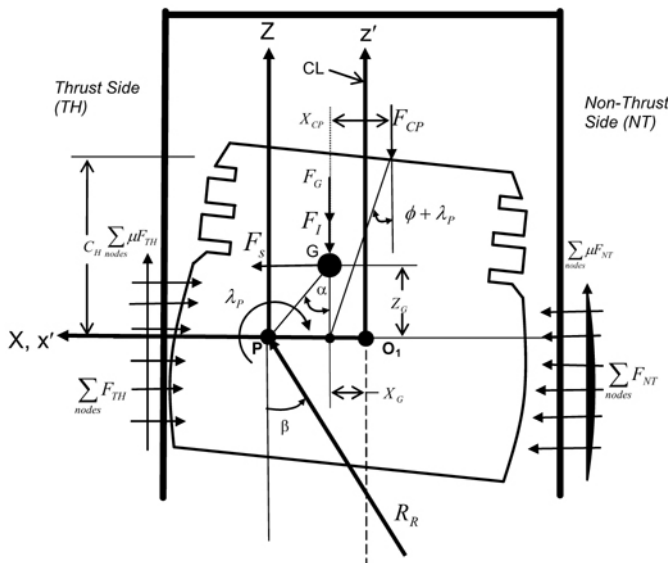


Figure 1. Forces acting on the piston

[Figure 1](#) shows all necessary forces for deriving the equations of motion for the piston primary and secondary dynamics. A detailed description of all symbols is provided in the nomenclature. The considered forces are due to combustion, gravity, inertia due to piston primary and secondary accelerations, skirt-liner contact and skirt friction. The total force on the piston in the vertical direction can be described by the following expression,

$$F_V = F_I + F_G + \frac{F_{CP}}{2} - \left(\mu \sum_{\text{Nodes}} F_{TH} + \mu \sum_{\text{Nodes}} F_{NT} \right) \quad (1)$$

Based on the piston free-body diagram, Newton's law in the lateral direction yields,

$$M_P \ddot{\delta}_P = M_P Z_G \ddot{\lambda}_P - M_P X_G \tan \beta \ddot{\lambda}_P + R_R \sin \beta + F_S \tan \beta - \left[\sum_{\text{nodes}} F \sin \psi \right]_{TH} + \left[\sum_{\text{nodes}} F \sin \psi \right]_{NT} \quad (2)$$

Similarly, Newton's law for the rotational motion about the piston's mass center yields

$$I_G \ddot{\lambda}_P = \frac{F_{CP}}{2} X_{CP} - M_P X_G^2 \ddot{\lambda}_P - M_P X_G \tan \beta \ddot{\lambda}_P Z_G + \dots + R_R \cos \beta X_G + F_S X_G + R_R \sin \beta Z_G + \dots + F_S \tan \beta Z_G - \left[\sum_{\text{nodes}} F(Z_G - Z) \sin \psi \right]_{TH} + \dots + \left[\sum_{\text{nodes}} F(Z_G - Z) \sin \psi \right]_{NT} + \dots + \mu \left[\sum_{\text{nodes}} F(X - X_G) \sin \psi \right]_{TH} - \dots - \mu \left[\sum_{\text{nodes}} F(X_G - X) \sin \psi \right]_{NT} \quad (3)$$

[Equations \(2\)](#) and [\(3\)](#) constitute the equations of motion for the piston secondary motion vector $\mathbf{x} = (\delta_P, \lambda_P)$. These equations are coupled with a finite-element representation of the skirt flexibility. The resulting equations are

$$\begin{bmatrix} [M_{11}(\mathbf{x})] & [0] \\ [0] & [0] \end{bmatrix} \begin{Bmatrix} \ddot{\mathbf{x}} \\ \ddot{\mathbf{F}} \end{Bmatrix} + \begin{bmatrix} [C_{11}(\mathbf{x}, \dot{\mathbf{x}})] & [0] \\ [0] & [0] \end{bmatrix} \begin{Bmatrix} \dot{\mathbf{x}} \\ \dot{\mathbf{F}} \end{Bmatrix} + \dots \\ + \begin{bmatrix} [K_{11}(\mathbf{x})] & [K_{12}] \\ [K_{21}] & [C] \end{bmatrix} \begin{Bmatrix} \mathbf{x} \\ \mathbf{F} \end{Bmatrix} = \begin{Bmatrix} \mathbf{R}(\mathbf{x}, \dot{\mathbf{x}}) \\ \mathbf{h}(\mathbf{x}) \end{Bmatrix} \quad (4)$$

where $\mathbf{F}$ is the vector of contact forces. Both $\mathbf{x}$ and $\mathbf{F}$ are calculated simultaneously.

The above equations are linearized using the Newton-Raphson method and then integrated in time, based on a modified Newmark-Beta method. The contact force pattern along the piston skirt is determined iteratively using a complementarity algorithm [13]. Output quantities representing performance metrics such as friction, noise and scuffing are calculated. The interested readers are referred to [11,12,13,14] for further details on the piston dynamics, the solution scheme and the contact algorithm.

2.2. PISTON DESIGN VARIABLES

We have already mentioned that the piston shape is not cylindrical although it seems so. It has a barrel shape along the piston skirt and an oval shape in any cross section. Designing the piston shape is important because it compensates for the piston's mechanical and thermal expansion. It also influences the "optimum" contact pattern between the piston skirt and the cylinder wall. The piston shape mainly helps to reduce friction, to optimize lubrication at the piston-bore interface, to guide the piston in the cylinder, and to reduce piston noise. In this research, the piston skirt barrelity and ovality are accounted for in calculating the clearance between the skirt and the liner. It was found in [17] that a parabolic piston skirt profile is better than a linear one, because the former sustains hydrodynamic lubrication in both the up and down strokes. An attempt was made in [8] to address the secondary dynamics of both articulated and conventional pistons. Barrel and oval profile also help account for cylinder bore distortion along the cylinder liner and the resultant change in operating clearance between the skirt and liner because of it. Minimizing the cylinder bore distortion has an advantage for the sealing of piston rings and the crankcase. Better sealing for piston rings has a direct impact on oil consumption and blow-by.

Based on a sensitivity study [11], seven key design variables were identified as the biggest contributors on piston friction, noise and scuffing. Table 1 provides the lower and upper bound for each design variable and the assumed standard deviation we used in the multi-objective RBDO study of section 5.1.

Table 1. Piston design variables

	Design variables	Units	Lower bound	Upper bound	Standard deviation
1	Pin Offset	mm	-1	1	0.1
2	Cold clearance	micron	0	0.025	0.005
3	Clearance at A ₂ section	micron	0.1	0.2	0.05
4	Ovality at A ₂ section	-	0.4	0.9	0.1
5	Barrelity at A ₂ section	-	1.5	4.0	0.2
6	Cylinder Bore Distortion	deg	60	120	10
7	Coefficient of Friction	-	0.05	2.0	0.05

Figure 2 shows some details of piston design. From the front view, the piston has a barrel shape which allows different level of clearance at different sections. As shown in Figure 2, we have divided the piston skirt into three main sections; top section A₂, middle section A₁, and bottom section A. The clearance between the skirt and the liner in the middle section represents the cold-clearance. The following equation is used to define the piston's barrel shape from the middle section to the top and bottom sections,

$$Y = Ae^x \quad (5)$$

where x is a coefficient defining the shape of the barrelity, e is distance of any point along the piston skirt from middle section, and A is the clearance of the top or bottom section. The clearance at section A₂ (also known as 'A₂-drop') and the barrelity at section A₂ are used as design variables in the present optimization study. The pin offset is defined as the distance between the piston centerline to the piston pin hole centerline. The piston pin offset and the cold clearance are also considered as design variables.

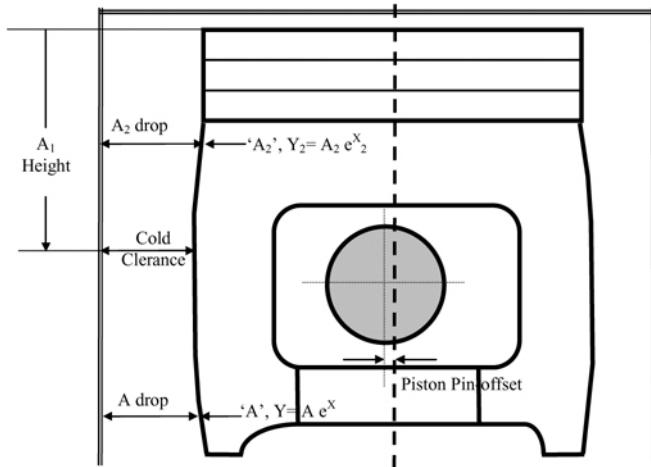


Figure 2. Piston geometric dimensions

From a top view, the piston has an oval shape which provides clearance between the piston and the liner in order to reduce the piston noise from the piston impact onto the liner. The oval shape is defined at three sections along the piston skirt similarly to the piston barrelity. Because the ovality at the top-section is critical, it is used as a design variable.

The role of cylinder bore shape on engine performance and its effect on oil consumption under high speed conditions has generated significant interest in recent year. Cylinder bore distortion is mainly due to the difference in temperature between the two sides of the liner surface. The contribution of bore distortion to the operating clearance depends mainly on the engine operating speed and loading. A simple thermal analysis is used to calculate the bore distortion along the length of the liner. In general, low bore distortion is highly desired as it affects not only the sealing performance of piston rings but also the skirt-liner contact dynamics. Due to its significant contribution to operating clearance between the piston skirt and liner, the cylinder bore distortion is also considered as a design variable.

2.3. METAMODEL DEVELOPMENT

An optimized piston design using the seven most significant design variables of [Table 1](#) can have a significantly improved performance. One evaluation of piston design using the analytical model of section 2.1 takes 10-14 minutes of computational time on a desktop computer. Even though the analytical model is quite efficient, spending this amount of time for one design evaluation can be very time-consuming in an optimization process. In order to address this, an approximation model [18,19,20] (surrogate model or metamodel), is built using only the important design variables identified from a sensitivity analysis. Most metamodels can interpolate in the design space spanned by the available sample points, but they can not extrapolate accurately outside that domain. For this reason, appropriate bounds of the design

variables have been identified (see [Table 1](#)) and used in the optimization.

A metamodel approximates the functional relationship between the output (friction, noise and scuffing) of the simulation model and the input variables (as listed in [Table 1](#)). There are various metamodeling methods [18,19,20] such as polynomial-based response surface models, Kriging models, and radial basis function (RBF) based models, among others. In this work, all metamodels were built using the RBF method which is one of the most frequently used approaches [20] in developing a metamodel of multivariate problems. In order to approximate the real valued function $g(\mathbf{x})$ by $g^a(\mathbf{x})$, given the set of values $(g_1, \dots, g_n)$ at distinct points $\mathbf{x} = (x_1, \dots, x_n) \in \mathbb{R}^d$, the following relationship is used

$$g^a(\mathbf{x}) = \sum_{i=1}^n \lambda_i \phi(\|\mathbf{x} - \mathbf{x}_i\|) \quad (6)$$

where, $\mathbf{x} \in \mathbb{R}^d$, λ_i is a real-valued weight, $\|\mathbf{x} - \mathbf{x}_i\|$ is the Euclidean distance of prediction point $\mathbf{x}$ from sample point $\mathbf{x}_i$ and ϕ denotes the basis function. There are various basis functions available, such as the Gaussian $\phi(\xi) = \exp(-c\xi^2)$, the thin plate spline $\phi(\xi) = \xi^2 \log(\xi)$, and the multiquadric $\phi(\xi) = \sqrt{\xi^2 + c^2}$.

The metamodel accuracy is critical. It depends on the number of sample points and the relationship (linear, nonlinear, continuous or discontinuous) between the input design variables and the responses. In this study, in order to “space-fill” the entire design space, 1000 sample designs were generated for the 7 important design variables using the Optimal Latin Hypercube (OLH) sampling method. The analytical piston model was ran at each sample point to predict the value of the three performance metrics of maximum skirt pressure, friction power loss, and maximum kinetic energy loss during one engine cycle. A separate RBF-based metamodel using Gaussian basis functions, was developed for each performance metric. [Table 2](#) shows the accuracy of the developed RBF-based metamodel for all three metrics. An error of less than 4% was observed in the entire design space. Under the assumption that this error is acceptable, the metamodels were used in the deterministic and probabilistic design optimization studies of sections 4 and 5.