Automatic Simulation Method for Performance Analysis of Kerosene-Based Turbo-Pump

Xuan Jin¹,², *, Chibing Shen¹,²

¹College of Aerospace Science and Engineering, National University of Defence Technology, Changsha, China.
²Science and Technology on Scramjet Laboratory, National University of Defence Technology, Changsha, China.

*Corresponding author e-mail: 1540445629@qq.com

Abstract. Kerosene-based turbo-pump is crucial to pump-fed expansion cycle scheme of the scramjet system in aerospace applications. An automatic simulation method was developed to conduct performance analysis of kerosene-based turbo-pump at low costs. Turbulence simulation combined with a multi-species surrogate model of kerosene was employed to study objective characteristics by simulating the flow-field inside the turbo-pump at 100000 rpm. Automatic simulation process, based on Design of Experiments technique, response surface model and high-fidelity numerical simulation, was applied for similarity analysis and Pareto front analysis, respectively. Results indicate that, all the characteristic curves of pump developed head ($\Delta H_p$) and turbine shaft power ($P_t$) with respect to mass flow rate at various rotating speeds, agree well with Similarity Criterion; $\Delta H_p$ is negatively related to the efficiency of pump ($\eta_p$) in general, but the contrary is the case for $P_t$ and the efficiency of turbine ($\eta_t$).

1. Introduction
The pump-fed expansion cycle scheme of scramjet system has become a potential in solving the problems such as high combustion performance, fuel supply efficiency and long duration thermal protection, and having advantages of simple configuration and light weight [1, 2]. Through exchanging heat with the walls of cooling channels, supercritical/cracking kerosene is formed to drive the turbo-pump to pressurize the fuel (kerosene at room temperature and atmosphere pressure). Then, all the fuel flows into the scramjet combustor, and reacts with air to produce heat release and thrust.

Turbo-pump pressurization by supercritical/cracking kerosene is the key part of the expansion cycle scheme[3]. As the rotating machinery, many structural details have significant effects on turbo-pump performance[4]. Thus, the study on kerosene-based turbo-pump operating characteristics is the indispensable part for building the expansion cycle scheme of scramjet system.

The core of performance analysis in fluids mechanics lies in the database containing the results of all the computational fluids dynamic (CFD) computations. For each sample, the inputs are the geometrical parameters, fluid properties and the flow-field boundary conditions used by the 3D flow solver. The outputs are the developed head, shaft power and efficiency, which characterize the hydrodynamic performance. Nowadays, the exponential increase of computation power has allowed the development of automatic simulation methods by coupling commercial CFD softwares[5]. These
methods lead to an analysis process that relies more on a systematic methodology than on experience. Compared with experimental study and traditional approaches, they consume less time and resource.

Many efforts have been directed to corresponding researches. Shah S.R.[6] and Jaiswal N.P.[7] reviewed key technologies regarding to the centrifugal pump design and numerical simulation. Pei et al.[8] proposed an optimization method based on ANSYS 14.5-CFX, Latin Hypercube method and Response Surface Method for centrifugal pump to improve the hydraulic performance. Hu et al.[9] presented a collaborative framework in combination with a variable-complexity modeling technique for the multidisciplinary coupling analysis and design of a shrouded turbine blade. Ahmed et al.[10] carried out numerical simulation of turbine flow-field using 3D Reynolds-Averaged Navier-Stokes models with Shear Stress Transport turbulence model in ANSYS 15-CFX.

Based on the previous experimental work[11], an automatic simulation method for performance analysis of kerosene-based turbo-pump was developed. It was desired that the method should be easily modified and extended to handle more complex problems. In this paper, initial design scheme of turbo-pump was acquired according to typical design method; then, the simulation model for studying turbo-pump performance was developed and validated. Based on automatic simulation process organized by iSIGHT software, similarity analysis and Pareto front analysis of turbo-pump performance parameters were carried out.

2. Model Calculation and Analysis

2.1. Kerosene-Based Turbo-Pump Design Scheme

According to the operating Mach number (4–7) and altitude (17km–30km) of scramjet system, the operation parameters of kerosene turbo-pump at design state were designed as Table 1.

| Parameters | Nomenclature                              | Value    |
|------------|-------------------------------------------|----------|
| \(\omega\) | Rotating speed of turbo-pump (rpm)        | 100000   |
| \(q_{mp}\) | Fuel mass flow rate of pump (kg/s)        | 1        |
| \(p_{pi}\) | Fuel pressure at pump inlet (MPa)         | \(\leq 0.5\) |
| \(p_{p2}\) | Fuel pressure at pump outlet (MPa)        | 6.2      |
| \(T_p\)   | Fuel temperature at pump inlet (K)        | 300      |
| \(q_{mt}\) | Fuel mass flow rate of turbine (kg/s)     | 0.95     |
| \(p_{ti}\) | Fuel pressure at turbine inlet (MPa)      | 5.5      |
| \(p_{t2}\) | Fuel pressure at turbine outlet (MPa)      | 2.5      |
| \(T_t\)   | Supercritical/cracking fuel temperature at turbine inlet (K) | 850 |

The physical model in present paper is the structure schematic drawing of kerosene-based turbo-pump shown in Fig. 1, mainly consisting of turbo-pump shell, inducer, centrifugal pump impeller, turbine disk, shaft, and so on. The main structure parameters of turbo-pump are shown in Table 2.

2.2. Thermophysical Property Evaluation of Supercritical/Cracking Kerosene

Huang et al.[12] showed that the cracked product distribution of n-alkanes is similar to that of hydrocarbon fuel; and Xu et al.[13] verified that the thermophysical properties of n-decane are similar to kerosene. Therefore, n-decane was chosen as the surrogate of aviation kerosene for the present study. According to the experimental result of Ward et al.[14], the total reaction model for mild conversion of n-decane under pressure 3.45MPa \(\sim\) 11.38MPa was employed:

\[
\begin{align*}
C_{10}H_{22} & \rightarrow 0.151H_2 + 0.143CH_4 + 0.256C_2H_6 + 0.126C_2H_4 + 0.230C_3H_8 + 0.180C_3H_6 + 0.196C_4H_8 + 0.102C_4H_{10} + 0.171C_5H_{10} + 0.124C_5H_{12} + 0.195C_6H_{12} + 0.089C_6H_{14} + 0.169C_7H_{14} + 0.072C_7H_{16} + 0.152C_8H_{16} + 0.012C_8H_{18} + 0.053C_9H_{18} + 0.003C_{10}H_{20}
\end{align*}
\]
Table 2. Structure parameters of the kerosene turbo-pump at design state.

| Parameters | Nomenclature                          | Baseline | Range   |
|------------|---------------------------------------|----------|---------|
| \( r_2 \)  | The radius of impeller outlet (mm)    | 42.8     | [38, 50]|
| \( r_u \)  | The radius of short blade inlet (mm)  | 19.2     | [17, 22]|
| \( \theta \) | The offset angle of short blade (°)    | 45       | [40, 54]|
| \( B_1 \)  | The inlet installation angle of blade (°) | 20      | [20, 28]|
| \( B_2 \)  | The outlet installation angle of blade (°) | 30      | [20, 40]|
| \( b_1 \)  | The width of impeller inlet (mm)      | 6        | [5, 5.6]|
| \( h_{en} \) | The outlet height of turbine nozzle (mm) | 5.3     | [4.5, 6]|
| \( d_{av} \) | The intermediate diameter of turbine blade (mm) | 26      | [24, 28]|
| \( b_b \)  | The width of turbine blade (mm)       | 8        | [7, 9]  |
| \( n \)    | The number of turbine nozzles         | 3        | /       |

Fig. 1 Structure schematic drawing of kerosene-based turbo-pump.

The molar reaction rate of n-decane is:

\[
-\frac{dC_R}{dt} = A \cdot e^{-\frac{E_a}{RT}} \cdot C_R
\]

where \( A \) refers to the pre-exponential factor, \( 1.6 \times 10^{15} \text{ s}^{-1} \); \( E_a \) refers to the activation energy, 263.7 kJ/mol; \( R \) refers to the molar gas constant, 8.314 J/(mol·K).

Fig. 2 Comparison between calculated thermophysical properties and experimental data at 5 MPa.
Based on the above pyrolysis mechanism, NIST Supertrapp software\cite{15} was applied for approximately predicting the thermophysical properties of the cracked products. This property-evaluation method briefly described in this section have been used to calculate the kerosene thermophysical properties at 5 MPa. As shown in Fig. 2, comparisons were made to the published experimental data up to 650 K\cite{16-18} except for the thermal conductivity, for which the available experimental data is limited to 513 K\cite{19}. All the average errors between the calculated data and experimental one is within 13%. Good agreement has been reached between the two sets of data.

2.3. Numerical methods and mesh generation
In order to simulate the kerosene flows inside turbo-pump, the conservation equations of mass, momentum, energy, and species were numerically solved through finite volume method provided by the pressure-based solver in the CFD commercial package, ANSYS® 15- FLUENT. RNG k-\(\varepsilon\) turbulence model was adopted for treating turbulent flow, and standard wall functions were used to determine the flow variables in the near-wall region. The SIMPLEC algorithm was used to resolve the coupling between velocity and pressure, and the discretization scheme of second-order (upwind) was chosen to handle all terms in governing equations. Additionally, efficient thermodynamical properties of supercritical/cracking kerosene were computed as a function of temperature, pressure and n-decane mass fraction; then, User-defined-function (UDF) about the pyrolytic kinetics model and property-evaluation method was coded and implemented into FLUENT software.

Herein, the Solidworks software was used to generate 3D model of centrifugal pump and axial-flow turbine, which can be seen in Fig. 1. The solution domains of turbo-pump were divided into three sections, respectively; and 3D unstructured mesh with different grid density was applied to different sections of turbine and pump. The grid density was clustered at the vicinity of the wall and rotors of turbo-pump, for the required resolution precision in the boundary layer. Then, all sections of turbine and pump was jointed on the basis of the continuous joint grid technology, respectively; and no slippery wall was specified here. Non-inertial coordinate system was applied for the flow-field around pump impeller and turbine rotor-cascade\cite{20, 21}. The boundary conditions of mass flow inlet and pressure outlet were employed for pump, and the boundary conditions of pressure inlet and pressure outlet for turbine; other variables at boundary regions were extrapolated from the interior.

Prior to detailed numerical studies, the independence of turbo-pump's performance parameters from grid number was checked to ensure the numerical accuracy. The meshes shown in Fig. 3 and Fig. 4 were adopted as the grid-independent solution for steady-state calculations hereinafter; the grid sums of different sections of pump and turbine were about 3170000 and 2630000, respectively.

![Fig. 3 Mesh generation for pump flow-field.](image1)
![Fig. 4 Mesh generation for turbine flow-field.](image2)
2.4. Preliminary analysis on internal flow field of turbo-pump

The relevant boundary conditions of the fundamental calculation case for pump and turbine were specified according to the operating parameters in Table 1. The cross-sectional flow-fields of the centrifugal pump and axial-flow turbine at design state were predicted as shown in Fig. 5.

From Fig. 5 (a), it could be observed that no vortex exists in the rotational section of pump, and the streamline is smooth around the blades; the fuel pressure at the inlet of the impeller is lowest, then increases rapidly as the blades rotate to work. When the liquid flows into the diffused channel, the effect of viscous converts a small amount of kinetic energy into pressure energy; moreover, a vortex appears at the outlet of the pump. Fig. 5 (b) displays that the static pressure of the subsonic working medium undergoes a process of abrupt decrease when flowing inside the nozzle of turbine. This means that large pressure energy is translated into kinetic energy for driving turbine blades, just as the streamline direction shows. Then the kinetic energy of working medium decreases notably after doing work, which results in a vortex at the exit of turbine blades. In general, the simulation results are consistent with the basic flow-field characteristics of the turbo-pump.

Fig. 5 Detailed distributions of streamline and static pressure within turbo-pump at design state.

In this paper, shaft power ($P_t$) and efficiency ($\eta_t$) are the performance parameters for turbine, while developed head ($\Delta H_p$) and efficiency ($\eta_p$) for pump. According to the above numerical result and Eqs. (5) ~ (8), the values of $\Delta H_p$, $\eta_p$, $P_t$, and $\eta_t$ are 5.78 MPa, 50.2 %, 25.95 kW and 53.5 %, respectively.

\[
\Delta H_p = p_{p2} - p_{p1}
\]  
\[
P_t = M \cdot \frac{2\pi \cdot \omega}{60}
\]  
\[
\eta_p = \frac{q_{up} \cdot \Delta H_p}{\rho \cdot P_p}
\]  
\[
\eta_t = \frac{P}{q_{out} \cdot L_{ad}} = \frac{P}{q_{out} \cdot k \cdot \rho \cdot R \cdot T} \left[ 1 - \left( \frac{P_{\text{out}}}{P_t} \right)^{\frac{k-1}{k}} \right]^{-1}
\]
where $M$ is the torque of pump; $L_{ad}$ is the isentropic expansion work; $k$ is the specific heat ratio.

3. Results and Discussion

3.1. Automatic simulation method for kerosene-based turbo-pump

The comprehensive performance analysis of turbo-pump requires the accurate data from large amount of the experiment or high-fidelity numerical simulation, which needs huge experimental fees or calculation cost. Hereinafter, automatic simulation method was adopted to alleviate such difficulty by balancing the requirements between the calculation cost and the accuracy. Automatic simulation method synthetically adopts Design of Experiments (DOE) technique, surrogate model (such as polynomial-based response surface (PRS) model, Kriging model, and so on), and high-fidelity numerical simulation method in Section 2. The desired results of performance analysis could be acquired efficiently through the following process in Fig. 6.

![Fig. 6 Automatic simulation process organized by iSIGHT platform.](image)

Herein, DOE technique is employed to explore the given design space and obtain enough sampling points. In order to train the sampling points efficiently, iSIGHT software is used to integrate VC (Microsoft Visual C++) compiler, Solidworks software, ICEM software and Fluent software by calling corresponding batch files (Design.dat, P/T.vbs, P/T.rpl, P/T.jou). The steps (scheme design, 3D modeling, mesh generation, numerical simulation and performance calculation) were implemented subsequently, and the information exchanging between them includes parameterized data files (Configuration.dat, P/T.iges, P/T.msh and Performance.dat) and well-defined variable mapping relationship. Based on the trained sampling database, surrogate models (PRS) were obtained for corresponding performance analysis of turbo-pump.

As the most frequently applied surrogate model, PRS model of first-order or second-order (Eq. (9)) can map complex design space effectively and filter some data noise easily. The coefficients that best fits the trained database was acquired by regression process such as linear Least-Squares Analysis[22].

$$y = c_0 + \sum_{i=0}^{n} c_i x_i + \sum_{i=0}^{n} c_i x_i^2 + \sum_{i=0}^{n-1} \sum_{j=i+1}^{n} c_{ij} x_i x_j$$  (6)
where \( y \) refers to the response of PRS; \( x \) refers to \( n \)-dimensions design variables; \( c \) refers to the quadratic polynomial coefficients.

### 3.2. Similarity analysis

Similarity Criterion[23] for viscous fluid flow contains geometric similarity, kinematic similarity and dynamic similarity. Thus, for the obtained design scheme with baseline structure parameters in Table 2, the following relationships exist among \( \Delta H_p \), \( P_t \), and \( \omega \):

\[
\frac{\Delta H_p}{\Delta H_{rated}} = \left( \frac{q_{mp}}{q_{mrated}} \right)^2 = \left( \frac{\omega}{\omega_{rated}} \right)^2
\]

\[
\frac{P_t}{P_{rated}} = \left( \frac{q_{mp}}{q_{mrated}} \right)^3 = \left( \frac{\omega}{\omega_{rated}} \right)^3
\]

Based on a series of flow-field simulations at rated rotating speed of 100000 rpm, the characteristic curves of \( \Delta H_{rated} \) (second order), and \( P_{rated} \) (first order) with respect to mass flow rate \( (q_{mrated}) \) were obtained through polynomial fitting (Eq. 9). Then, the characteristic curves of \( \Delta H_p \) and \( P_t \) with respect to mass flow rate, at different rotating speed (80000, 90000, 110000, and 120000 rpm), were derived from Eqs. (12) and (13).

\[
\Delta H_p = \Delta H_{rated} \frac{\omega^2}{\omega_{rated}^2} = \left( c_{2p} q_{mrated}^2 + c_{1p} q_{mrated} + c_{0p} \right) \frac{\omega^2}{\omega_{rated}^2} = c_{2p} q_{mp}^2 + c_{1p} q_{mp} + c_{0p} \frac{\omega^2}{\omega_{rated}^2} + c_{0p} \frac{\omega^2}{\omega_{rated}^2}
\]

\[
P_t = P_{rated} \frac{\omega^3}{\omega_{rated}^3} = \left( c_{1t} q_{mrated}^3 + c_{0t} \right) \frac{\omega^3}{\omega_{rated}^3} = c_{1t} q_{mp}^3 + c_{0t} \frac{\omega^3}{\omega_{rated}^3}
\]

![Fig. 7 Similarity analysis of \( \Delta H_p \).](image1)

![Fig. 8 Similarity analysis of \( P_t \).](image2)

Fig 7 and 8 display the five characteristic curves of \( \Delta H_p \) and \( P_t \) with respect to mass flow rate at various rotating speeds of 80000, 90000, 100000, 110000, and 120000 rpm; and the spot data from automatic simulation process was used to verify corresponding characteristic curves. The maximum discrepancy between characteristic curves and spot data is below 5%; thus \( \Delta H_p \) and \( P_t \) agree well with...
Similarity Criterion[23] in general, proving the accuracy and reasonability of the turbo-pump design scheme and simulation method.

3.3. Pareto front analysis

Due to the complex structure of turbo-pump, it will be difficult to find out the variation range of performance parameters within the limits of structure parameters. According to the automatic simulation method, the given ranges of main structure parameters in Table 2 were taken as the design space; then 256 sampling points were obtained by means of Optimal Latin Hypercube method in DOE, which spread evenly and afford to capture higher-order effects. At last, second-order PRS models (Eq. (9)) of four performance parameters ($\Delta H_p$, $\eta_p$, $P_t$ and $\eta_t$) with respect to main structure parameters in Table 2 were obtained. Based on error analysis of $R^2$, the confidence levels of four second-order PRS models are all above 90%, which meets the precision requirement for prediction.

In order to comprehensively understand the relationship between the performance parameters, Pareto front depicted in Fig. 9 was also obtained on the basis of the four second-order PRS models. Fig. 9 presents 6000 sampling points from the design space shown in Table 2. These sampling points were trained by the established second-order PRS models; this method is less time-consuming than high-fidelity numerical simulation while precision guaranteed.

![Fig. 9 Pareto front for turbo-pump performance parameters.](image)

From Fig. 9 (a), $\eta_p$ rises first and falls later as $\Delta H_p$ increases. This can be concluded that $\Delta H_p$ and $\eta_p$ have their own optimal value; although the two optimal values are close to each other, they cannot reach the optimal solution at the same time. However, from Fig. 9 (b), it is clearly observed that $\eta_t$ decreases with the decrease of $P_t$. The objective value of $P_t$ should be ascertained according to the required shaft power of pump ($P_p$), while the optimal value of $\eta_t$ is fixed in the design space of turbine.

4. Conclusion

In this paper, an original investigation was carried out on the automatic simulation method for kerosene-based turbo-pump with small flow rate, which is mainly used in the fuel supply system of kerosene-based scramjet with expansion cycle.

By means of the automatic simulation method coupled with high-fidelity numerical simulation, the performance analysis process of kerosene-based turbo-pump saves time and effort compared to traditional approaches. The similarity analysis indicates that $\Delta H_p$ and $P_t$ agree well with Similarity Criterion, proving the accuracy and reasonability of the turbo-pump design scheme and simulation method; the Pareto front analysis shows that $\Delta H_p$ is negatively related to $\eta_p$ in general, but the contrary
is the case for $P_1$ and $\eta_1$. As for other aspects of applications, further investigation could be carried out to exploit the potential of the automatic simulation method.

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