



ARTICLE

A One-Dimensional Design Model for High-Speed Centrifugal Compressors with Wet-Outlet Assessment

Baris Burak Kanbur* 

Section of Mechanical Engineering, Institute of Mechanical and Electrical Engineering, University of Southern Denmark (SDU), Odense, Denmark

*Corresponding Author: Baris Burak Kanbur. Email: babk@sdu.dk

Received: 16 August 2026; Accepted: 13 October 2026; Published: 27 October 2026

ABSTRACT: High-speed centrifugal compressors are promising candidates for compact and efficient compression in high-temperature thermal-fluid processes, but their feasibility depends strongly on the interaction between refrigerant thermodynamic properties, operating conditions, impeller aerodynamics, and outlet phase behavior. Conventional preliminary assessments often focus on cycle-level performance or simplified compressor sizing, while the transition between dry and wet outlet operation is not explicitly treated. This study develops a one-dimensional design model for high-speed centrifugal compressors with real-fluid thermodynamic properties and wet-outlet assessment. The objective is to provide an early-stage framework that can distinguish between intrinsic thermodynamic limitations and design-dependent aerodynamic constraints. The model combines real-fluid property evaluation, compressor state calculation, phase detection, velocity-triangle analysis, slip-factor-based blade-angle estimation, inlet and outlet geometry calculation, and design-space classification. The model is demonstrated for a brewery hot-water process using three hydrocarbon working fluids: R-601/n-pentane, R-601a/isopentane, and R-600/n-butane. For each refrigerant, 3300 combinations of inlet flow coefficient, outlet flow coefficient, work coefficient, and blade number are evaluated. The results show that all three refrigerants provide broad preliminary aerodynamic design spaces. The acceptable design fractions are 0.818 for R-601, 0.776 for R-601a, and 0.790 for R-600. A common representative design region is identified at an inlet flow coefficient of 0.16, outlet flow coefficient of 0.07, work coefficient of 0.85, and 20 blades. The corresponding impeller outlet diameters are 0.338 m, 0.330 m, and 0.266 m for R-601, R-601a, and R-600, respectively. The inlet relative Mach numbers remain below unity for all representative designs. However, R-601 and R-601a enter wet-outlet operation, with vapor qualities of 0.877 and 0.852, respectively, whereas R-600 remains single-phase at the outlet. The study shows that compressor feasibility cannot be assessed from thermodynamic performance or geometry alone. Wet-outlet operation is identified as an intrinsic thermodynamic constraint for the hydrocarbons within the present fixed thermodynamic-state model and under the specified boundary conditions, while inlet Mach number, inlet diameter ratio, blade angle, and blade-height ratio are design-dependent aerodynamic quantities. The proposed model therefore provides a structured preliminary tool for connecting real-fluid thermodynamics, compressor geometry, and wet-outlet risk before detailed loss modelling, diffuser design, and full compressor-unit co-design.

KEYWORDS: One-dimensional model; homogeneous equilibrium model; compressor; high temperature; process heat; industrial decarbonization; two phase flow; hydrocarbons; thermal energy

1 Introduction

High-temperature heat represents one of the major challenges in energy-system decarbonization because a large fraction of final energy demand is used to provide process heat over a wide range of

temperature levels. Several bottom-up studies have shown that high-temperature heat demand is strongly distributed across sectors, end uses, and temperature intervals, ranging from low-temperature food and paper processes to high-temperature chemical, mineral, and metal industries [1–3]. This heterogeneity makes the replacement of fossil-fuel-based heat supply difficult, since a single technology cannot address all process conditions. Nevertheless, high-temperature thermal processes have emerged as one of the most promising routes for electrifying industrial heat, especially when waste heat is available at a suitable temperature level and the required heat sink temperature is within the achievable range of vapor-compression systems [4–7]. Recent studies have therefore focused on advanced cycle configurations, low-global-warming-potential working fluids, and improved component design to extend their operation toward higher temperature lifts and higher supply temperatures [8–12]. In this framework, the compressor remains a central component because it directly controls the pressure ratio, discharge temperature, power consumption, operating range, and overall system efficiency [13].

Centrifugal compressors are attractive for high-temperature thermal systems because they can provide high efficiency, compactness, large capacity, and potentially oil-free operation. These features are relevant for broader high-temperature process applications where clean compression, reduced maintenance, and high-speed operation are desirable. Early studies on centrifugal compressors in advanced thermal cycles demonstrated that compressor performance is strongly linked to working-fluid properties, rotational speed, operating range, and off-design behavior [14]. Schiffmann and Favrat [15,16] experimentally demonstrated compact, direct-driven radial compressors and showed that small-scale oil-free turbo compressors can reach high rotational speeds and promising efficiencies when carefully designed. Kus and Neksa [17] later developed a one-dimensional preliminary design and performance model for oil-free carbon dioxide turbo-compressors, emphasizing that fast early-stage models are useful for identifying weak points before detailed design. More recent studies have further demonstrated the feasibility of centrifugal compressor at larger capacities and elevated supply temperatures, including advanced two-cycle systems, low-GWP refrigerants, and carbon dioxide cycle compressors with high-speed motors and magnetic bearings [18–20]. These works prove the technical relevance of high-speed centrifugal compressors, but they also show that compressor design cannot be separated from fluid selection and operating conditions.

An extensive amount of literature study has moved from isolated compressor or cycle models toward coupled cycle-compressor design. Meroni et al. [21] developed a mean-line centrifugal compressor model and coupled it with a high-temperature cycle, showing that working-fluid choice affects both cycle performance and compressor feasibility. Uusitalo et al. [22] investigated large-scale centrifugal-compressor-driven cascade solutions and showed that different working-fluid combinations lead to different compressor geometries, rotational speeds, efficiencies, and losses. Following studies extended this direction to hydrocarbon working fluids and high-temperature thermal processes, highlighting the strong interaction between temperature lift, compressor sizing, Mach number, axial force, and cycle configuration [23–25]. Jaatinen-Värri et al. [24] also showed that high Mach numbers can become a major obstacle in centrifugal compressor designs for high-temperature lifts. More recently, full compressor-unit co-design frameworks have included not only aerodynamic design, but also high-speed electric machines, active magnetic bearings, rotor dynamics, bearing control, and motor cooling [26]. These developments are important because they move the field closer to practical compressor units. However, they also increase model complexity, whereas early-stage screening still requires transparent tools that can reveal the origin of design constraints before committing to detailed aerodynamic, electromagnetic, and rotordynamic co-design.

One issue that has received increasing attention is the possible transition from dry compression to wet or near-saturation operation. Many working fluids considered for high-temperature applications

exhibit strong real-fluid behavior near the saturation boundary, and some dry fluids may approach or enter the two-phase region during compression at large temperature lifts [8,27]. Mateu-Royo et al. [8] discussed the importance of cycle configuration and internal heat exchange to ensure dry compression in advanced high-temperature thermal processes using low-GWP refrigerants. Yan et al. [27] proposed a holistic centrifugal compressor design framework that explicitly considered wet compression prevention, real-fluid properties, and global optimization, arguing that wet compression can introduce mechanical and efficiency penalties and should be considered during compressor design. While wet-compression prevention is a rational strategy for many systems, early-stage compressor assessment can also benefit from identifying whether a wet-outlet condition is intrinsic to the imposed thermodynamic operating point or whether it can be mitigated through aerodynamic design choices. This distinction is particularly important for hydrocarbon working fluids, where favorable thermodynamic performance may coexist with phase-boundary constraints, while alternative fluids may provide drier outlet states but weaker cycle-level performance or smaller critical-temperature margins.

Following the above-mentioned studies, the present study develops a one-dimensional design model for high-speed centrifugal compressors with wet-outlet assessment. The model uses real-fluid thermodynamic properties, detects the compressor outlet phase, and applies a homogeneous equilibrium model continuation when the outlet state enters the two-phase region. The model combines compressor state calculation, velocity-triangle analysis, slip-factor-based blade-angle estimation, inlet and outlet geometry calculation, and preliminary design-space classification. Unlike studies that only rank working fluids by thermodynamic cycle performance, or studies that proceed directly to detailed compressor optimization, the proposed model was designed as an intermediate diagnostic tool. It evaluates whether apparent compressor limitations arise from the thermodynamic state itself or from design-dependent aerodynamic choices such as inlet flow coefficient, outlet flow coefficient, work coefficient, and blade number. The framework was demonstrated using three hydrocarbon working fluids, which are (i) R-601/n-pentane, (ii) R-601a/isopentane, and (iii) R-600/n-butane, under a representative high-temperature process-heating condition. The specific contribution of the study is therefore not only the calculation of preliminary compressor geometry, but the separation of intrinsic thermodynamic constraints from aerodynamic risks that can be mitigated during early-stage design.

2 System Description

The present study focuses on the preliminary design assessment of high-speed centrifugal compressors for high-temperature thermal processes. Although the model is expected to be applicable to different refrigerant-based process-heating systems, a representative industrial hot-water production case was selected to define the thermodynamic compressor boundary conditions. The selected case was the brewery hot-water system reported by Andersen et al. [28], where a hydrocarbon high-temperature thermal process was integrated with an existing hot-water system and heat recovery tank. In that reference case, the process delivered 2.4 MW of hot water at 145°C using R-601a/isopentane in a single-stage cycle with an internal heat exchanger. The reported heat source was a stratified heat recovery tank supplying water from 90°C to 80°C, and the reference process was evaluated using a compressor isentropic efficiency of 70% and a heat exchanger pinch temperature difference of 5 K [28]. Fig. 1a illustrates the above-mentioned brewery case.

It is worth noting that the brewery case was not re-optimized at the plant level. Instead, it was used as a representative high-temperature process-heating boundary condition for compressor design analysis. This distinction is important because the objective of the study was not to reproduce the complete plant integration, control strategy, or economic performance of the reference brewery system. The reference case was used to define a realistic temperature level, heating capacity, and hydrocarbon operating range for

evaluating whether high-speed centrifugal compressors can provide feasible preliminary geometries and whether wet-outlet operation appears under the imposed thermodynamic state. Therefore, a simplified vapor-compression representation was adopted upstream of the compressor design model. The system was represented by four main components: an evaporator/source heat exchanger, a compressor, a condenser/sink heat exchanger, and an expansion valve, as shown in Fig. 1b. The compressor was the central component of interest, while the surrounding cycle was used only to generate the inlet and outlet pressure levels, enthalpy rise, refrigerant mass flow rate, and phase state required by the one-dimensional design model. The internal heat exchanger included in the reference brewery concept was not modelled explicitly as a separate heat exchanger in the present study. Instead, its main role was represented through the imposed compressor inlet superheat condition. This representation was used only to define a reproducible single-phase compressor inlet state and does not represent detailed internal heat exchanger sizing, effectiveness, pressure loss, or heat-transfer performance.

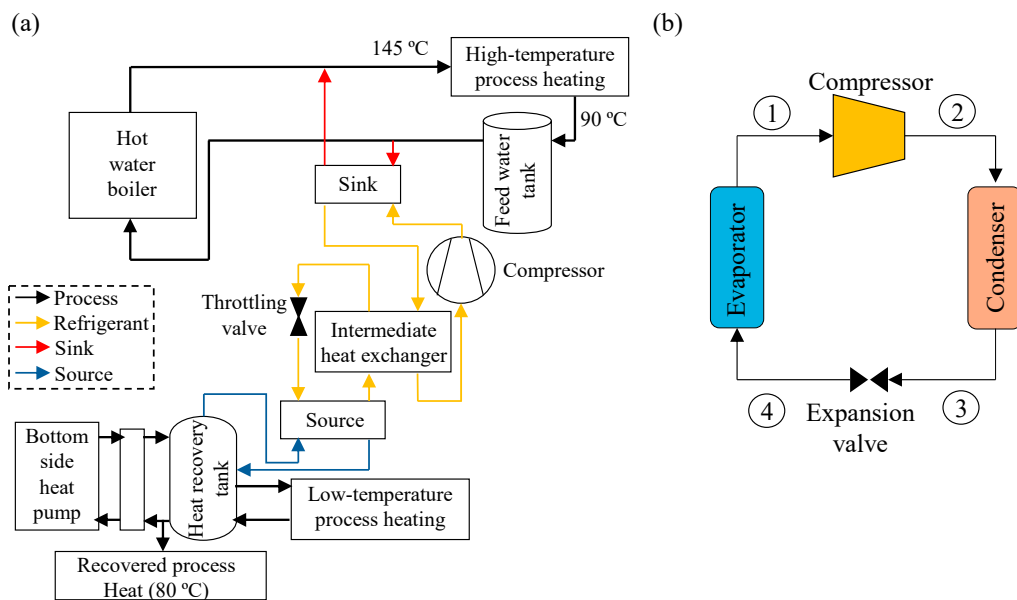


Figure 1: High-temperature thermal process for the current work; (a) redrawn and simplified schematic of the work by Andersen et al. [28], and (b) basic vapor compression cycle schematic for one-dimensional compressor model.

This simplification was selected to keep the focus on compressor-side design feasibility rather than detailed heat-pump cycle optimization. The source and sink temperature levels were selected to remain consistent with the reference brewery application. The source-side outlet temperature was used to define the evaporating level, while the hot-water delivery temperature was used to define the condensing level. A constant heat exchanger pinch temperature difference of 5 K was applied, following the same assumption by Andersen et al. [28]. As one of the most significant design criteria, the compressor inlet was required to remain in the single-phase vapor region by applying a specified superheat. The condenser outlet was represented using a small subcooling degree. These assumptions produced a consistent set of thermodynamic states for the compressor without introducing additional degrees of freedom associated with detailed heat exchanger design.

The study considered three hydrocarbon working fluids: R-601/n-pentane, R-601a/isopentane, and R-600/n-butane. These fluids were selected because they represent reasonable hydrocarbon candidates for high-temperature process-heating applications [29] and provide different combinations of critical temperature,

pressure level, density, volumetric flow rate, and outlet phase behavior. It is worth mentioning that other refrigerants, such as water (R-718), have also been discussed as promising candidates for high-temperature processes, but they introduce substantially different compressor-design challenges due to their very high volumetric-flow requirements [30]. The final refrigerant set therefore allowed the model to compare two C5 hydrocarbons with higher critical-temperature margins against a C4 hydrocarbon that provides a more compact compressor but is thermodynamically more constrained at high condensing temperatures. The compressor design inputs generated from the simplified cycle calculation are summarized in Table 1. These values were used as fixed thermodynamic boundary conditions in the subsequent design-space exploration. The same inlet temperature, heat exchanger assumptions, and initial compressor efficiency were applied to all refrigerants to ensure that differences in compressor geometry and outlet phase behavior resulted from refrigerant properties rather than from inconsistent modelling assumptions.

Table 1: Compressor design inputs based on the brewery hot water production case by Andersen et al. [28].

Parameter	R-601/n-Pentane	R-601a/Isopentane	R-600/n-Butane
Process case	Brewery hot-water production	Brewery hot-water production	Brewery hot-water production
Heating capacity	2.4 MW	2.4 MW	2.4 MW
Compressor inlet temperature	80°C	80°C	80°C
Evaporating temperature	75°C	75°C	75°C
Condensing temperature	150°C	150°C	150°C
Compressor inlet pressure	3.2378 bar	4.0457 bar	9.0616 bar
Compressor outlet pressure	15.9200 bar	18.6800 bar	36.7460 bar
Pressure ratio	4.9169	4.6173	4.0551
Refrigerant mass flow rate	11.843 kg/s	13.773 kg/s	17.361 kg/s
Isentropic enthalpy rise	53.671 kJ/kg	49.576 kJ/kg	50.742 kJ/kg
Initial compressor isentropic efficiency	0.70	0.70	0.70

The rotational speed was treated as a fixed refrigerant-specific input in the present design-space exploration. The values were obtained from the preceding preliminary compressor-sizing step for each refrigerant and were kept constant while the nondimensional aerodynamic design variables were swept. The selected speeds were 16,983 rpm for R-601/n-pentane, 16,694 rpm for R-601a/isopentane, and 20,981 rpm for R-600/n-butane. Thus, the present design-space loop does not optimize rotational speed; it evaluates the effect of ϕ_1 , ϕ_2 , ψ , and Z at fixed thermodynamic boundary conditions and fixed rotational speed for each refrigerant. For fixed enthalpy rise and work coefficient, the impeller tip speed is determined by the velocity triangle, while the physical outlet diameter scales inversely with rotational speed through $D_2 = 2U_2/\omega$. Therefore, the absolute compressor diameter is sensitive to the selected rotational speed. In contrast, the outlet phase state is determined by the thermodynamic state at p_2 and h_2 , and is therefore not affected by rotational speed. The reported diameters should consequently be interpreted as representative preliminary dimensions for the selected speeds, not as optimized final compressor sizes.

The resulting compressor operating points are not meant to represent final optimized compressor designs. They are preliminary design inputs used to evaluate the influence of refrigerant properties on impeller size, inlet geometry, blade height, velocity triangles, Mach-number indicators, and outlet phase state. This structure allowed the analysis to isolate the compressor-design implications of the selected

high-temperature process condition. In particular, it allowed the study to investigate whether the main limitations were imposed by the thermodynamic operating point itself, such as wet-outlet formation, or whether they could be mitigated through aerodynamic design variables such as inlet flow coefficient, outlet flow coefficient, work coefficient, and blade number. In addition to the system details, operating conditions, and basic assumptions above, other system-level assumptions used before entering the compressor design model are listed below:

- A 2 K condenser outlet subcooling was applied.
- Pressure drops in heat exchangers and piping were neglected.
- Heat losses to the environment were neglected.
- The expansion valve was assumed isenthalpic.
- The analysis was steady-state and design-point based.
- Motor, bearing, rotordynamic, sealing, and mechanical losses were not included in the system description stage.
- The purpose of the simplified cycle calculation was not to predict final coefficient of performance (COP), but to provide consistent compressor boundary conditions for the one-dimensional design model.

This system definition provides the interface between the industrial process-heating case and the compressor design model. The following section describes how these boundary conditions were converted into compressor velocity triangles, geometry parameters, phase-state indicators, and preliminary feasibility classifications.

3 Methodology

The overall methodology of the present study is explained in Fig. 2. The model is created in the MATLAB environment together with the CoolProp [31] integration.

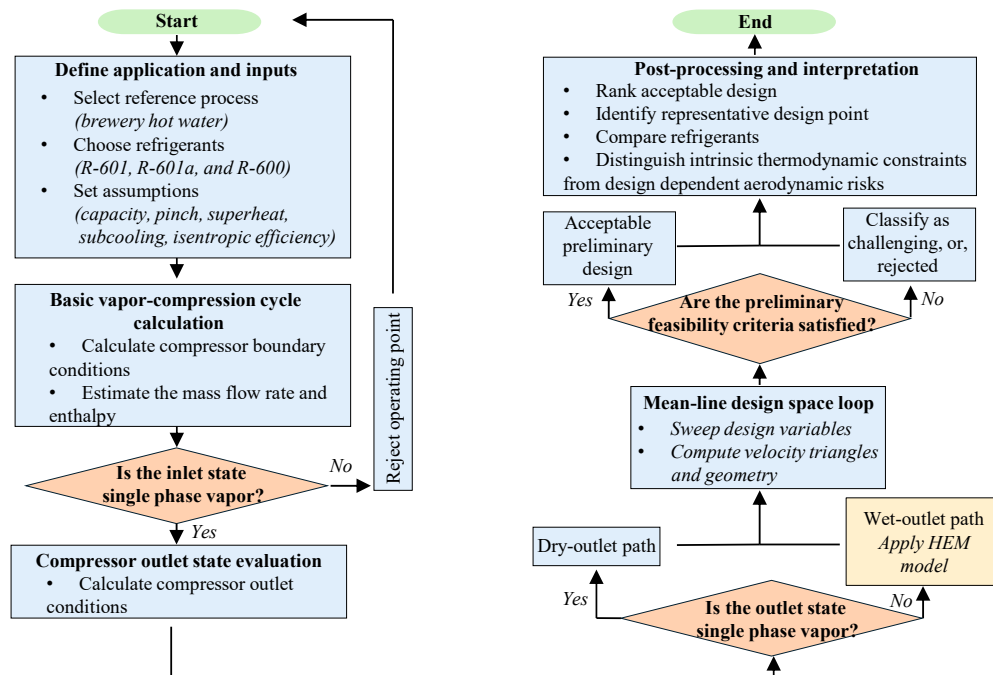


Figure 2: Methodology flowchart for the one-dimensional compressor model.

Overall, the methodology can be explained as follows; the compressor inlet and outlet thermodynamic states are defined from the cycle calculation. Based on the cycle data, a basic vapor-compression cycle is modeled. If the inlet phase is not a single-phase vapor, the model goes back to the start point to check and review the cycle conditions. That is, the model does not work in two-phase inlet conditions. With the single phase vapor inlet conditions are defined, the outlet phase condition is checked. If the compressor outlet was in the two-phase region, a homogeneous-equilibrium-model-based continuation was applied together with relevant two-phase speed of sound calculations. Otherwise, the dry compressor outlet mode is preferred. After that, the impeller inlet and outlet velocity triangles were calculated. Finally, geometric and aerodynamic feasibility indicators were evaluated over a range of design variables. To better explain these steps, the whole method is explained in five subsections below.

3.1 Thermodynamic States of Compressor

The compressor inlet state was defined by the inlet pressure and temperature as shown in Eqs. (1) and (2), respectively,

$$p_1 = p_{\text{evap}} \quad (1)$$

$$T_1 = T_{\text{evap}} + \Delta T_{\text{SH}} \quad (2)$$

where p_1 is the compressor inlet pressure, p_{evap} is the evaporating pressure, T_1 is the compressor inlet temperature, T_{evap} is the evaporating temperature, and ΔT_{SH} is the degree of superheating. The compressor outlet pressure is given in Eq. (3),

$$p_2 = p_{\text{cond}} \quad (3)$$

where p_2 is the compressor outlet pressure and p_{cond} is the condensing pressure. The compressor pressure ratio is then defined in Eq. (4),

$$\Pi = p_2/p_1 \quad (4)$$

where Π is the compressor pressure ratio. The inlet enthalpy and entropy are obtained from real-gas thermodynamic properties using CoolProp [31] as defined in Eqs. (5) and (6),

$$h_1 = h(T_1, p_1) \quad (5)$$

$$s_1 = s(T_1, p_1) \quad (6)$$

where h_1 and s_1 are the compressor inlet specific enthalpy and specific entropy, respectively. The isentropic outlet state is determined from Eqs. (7) and (8),

$$s_{2s} = s_1 \quad (7)$$

$$h_{2s} = h(p_2, s_1) \quad (8)$$

where s_{2s} is the entropy at the isentropic compressor outlet and h_{2s} is the corresponding isentropic outlet enthalpy. Following this definition, the isentropic enthalpy difference is calculated in Eq. (9),

$$\Delta h_s = h_{2s} - h_1 \quad (9)$$

where Δh_s is the isentropic specific enthalpy difference across the compressor. For a given initial compressor isentropic efficiency, the actual enthalpy difference is shown in Eq. (10),

$$\Delta h = \Delta h_s / \eta_{is} \quad (10)$$

where Δh is the actual specific enthalpy difference and η_{is} is the compressor isentropic efficiency. Therefore, the actual compressor outlet enthalpy is given in Eq. (11),

$$h_2 = h_1 + \Delta h \quad (11)$$

where h_2 is the actual compressor outlet enthalpy. The corresponding outlet temperature and density are obtained from Eqs. (12) and (13), respectively,

$$T_2 = T(p_2, h_2) \quad (12)$$

$$\rho_2 = \rho(p_2, h_2) \quad (13)$$

where T_2 is the actual compressor outlet temperature and ρ_2 is the outlet density. The compressor inlet density and speed of sound are calculated in Eqs. (14) and (15),

$$\rho_1 = \rho(T_1, p_1) \quad (14)$$

$$a_1 = a(T_1, p_1) \quad (15)$$

where ρ_1 is the inlet density and a_1 is the single-phase speed of sound at the compressor inlet. The compressor inlet is required to remain single-phase vapor. If the inlet state entered the two-phase region, the design point is rejected as shown in Fig. 2.

3.2 Outlet Phase Check and Homogeneous Equilibrium Model Conditions

The compressor outlet phase condition is determined from the vapor quality as shown in Eq. (16),

$$x_2 = x(p_2, h_2) \quad (16)$$

where x_2 is the compressor outlet vapor quality. For single-phase states, the property returns a non-physical quality value, and the outlet is treated as single-phase. For two-phase states, $0 \leq x_2 \leq 1$, the model continued using a homogeneous-equilibrium model (HEM) approximation. Thus, for a two-phase outlet state, the homogeneous mixture specific volume is represented in Eq. (17);

$$v_m = x_2 v_v + (1 - x_2) v_l \quad (17)$$

where v_m is the mixture specific volume, v_v is the saturated vapor specific volume, and v_l is the saturated liquid specific volume at p_2 . The corresponding mixture density is shown in Eq. (18),

$$\rho_m = 1/v_m \quad (18)$$

where ρ_m is the homogeneous-equilibrium mixture density. In the implementation, the mixture density is obtained directly from the real-fluid property calculation at (p_2, h_2) . The void fraction was calculated in Eq. (19),

$$\alpha = (x_2/\rho_v)/(x_2/\rho_v + (1 - x_2)/\rho_l) \quad (19)$$

where α is the void fraction, ρ_v is the saturated vapor density, and ρ_l is the saturated liquid density at the compressor outlet pressure. A Wood homogeneous-equilibrium speed of sound [32] is used to define wet-region Mach indicators in Eq. (20),

$$1/\rho_m a_{\text{HEM}}^2 = \alpha/\rho_v a_v^2 + (1 - \alpha)/\rho_l a_l^2 \quad (20)$$

where a_{HEM} is the homogeneous-equilibrium speed of sound, a_v is the saturated vapor speed of sound, and a_l is the saturated liquid speed of sound. It is worth mentioning that this speed of sound is not interpreted as a validated two-phase compressor acoustic model. It is used only to define first-order wet-region Mach indicators. The Wood-type relation is consistent with the homogeneous-equilibrium interpretation because it treats the two-phase outlet as a homogeneous mixture with shared pressure, shared phase velocity, and equilibrium thermodynamic properties. The resulting speed of sound corresponds to an equilibrium mixture compressibility estimate. It does not account for phase slip, finite-rate phase change, relaxation effects, droplet dynamics, liquid-film motion, or frequency-dependent acoustic propagation. Therefore, the resulting Mach numbers are not interpreted as validated two-phase compressor Mach numbers.

3.3 Impeller Outlet Velocity Triangle

The impeller outlet tip speed is calculated from the work coefficient as shown in Eq. (21),

$$U_2 = \sqrt{\Delta h/\psi} \quad (21)$$

where U_2 is the impeller outlet tip speed and ψ is the compressor work coefficient. The impeller outlet diameter is then obtained from Eq. (22),

$$D_2 = 2U_2/\omega \quad (22)$$

where D_2 is the impeller outlet diameter and ω is the angular velocity. The angular velocity is calculated from the rotational speed as shown in Eq. (23),

$$\omega = 2\pi n/60 \quad (23)$$

where n is the rotational speed in rpm. The outlet meridional velocity is defined by the outlet flow coefficient as given in Eq. (24),

$$C_{m2} = \phi_2 U_2 \quad (24)$$

where C_{m2} is the outlet meridional velocity and ϕ_2 is the outlet flow coefficient. Assuming zero inlet pre-whirl, the Euler compressor equation is given in Eq. (25),

$$\Delta h = U_2 C_{\theta 2} \quad (25)$$

where $C_{\theta 2}$ is the actual tangential component of the absolute velocity at the impeller outlet. Therefore, it can be defined as shown in Eq. (26).

$$C_{\theta 2} = \Delta h / U_2 \quad (26)$$

Following this equation, the absolute velocity at the impeller outlet is calculated as shown in Eq. (27),

$$C_2 = \sqrt{C_{m2}^2 + C_{\theta 2}^2} \quad (27)$$

where C_2 is the absolute velocity magnitude at the impeller outlet. The relative tangential component is defined in Eq. (28),

$$W_{\theta 2} = U_2 - C_{\theta 2} \quad (28)$$

where $W_{\theta 2}$ is the tangential component of the relative outlet velocity. The relative outlet velocity magnitude is then,

$$W_2 = \sqrt{C_{m2}^2 + W_{\theta 2}^2} \quad (29)$$

where W_2 is the relative velocity at the impeller outlet. The absolute and relative outlet flow angles are calculated as shown in Eqs. (30) and (31), respectively,

$$\alpha_2 = \tan^{-1} \left(\frac{C_{m2}}{C_{\theta 2}} \right) \quad (30)$$

$$\beta_{2,\text{flow}} = \tan^{-1} \left(\frac{C_{m2}}{W_{\theta 2}} \right) \quad (31)$$

where α_2 is the absolute outlet flow angle and $\beta_{2,\text{flow}}$ is the relative outlet flow angle. A preliminary slip factor is included to estimate the blade-exit metal angle. The slip factor is calculated as given in Eq. (32),

$$\sigma = 1 - \frac{\sqrt{\cos \beta_{2,\text{ref}}}}{Z^{0.7}} \quad (32)$$

where σ is the slip factor, Z is the number of impeller blades, and $\beta_{2,\text{ref}}$ is a reference backswept blade angle used in the preliminary slip estimation. In this work, the slip factor is not used to alter the Euler work directly. Instead, it was used to infer the ideal blade-exit tangential component required before slip as defined in Eq. (33),

$$C_{\theta 2,\text{blade}} = C_{\theta 2} / \sigma \quad (33)$$

where $C_{\theta 2,\text{blade}}$ is the ideal tangential component associated with the blade-exit direction. The corresponding blade-relative tangential component is given in Eq. (34),

$$W_{\theta 2, \text{blade}} = U_2 - C_{\theta 2, \text{blade}} \quad (34)$$

where $W_{\theta 2, \text{blade}}$ is the blade-relative tangential component. The estimated outlet blade metal angle is shown in Eq. (35),

$$\beta_{2, \text{metal}} = \tan^{-1} \left(\frac{C_{m2}}{W_{\theta 2, \text{blade}}} \right) \quad (35)$$

where $\beta_{2, \text{metal}}$ is the estimated outlet blade metal angle. The impeller outlet blade height is obtained from continuity equation as shown in Eq. (36),

$$b_2 = \dot{m} / (\rho_2 \pi D_2 C_{m2}) \quad (36)$$

where b_2 is the impeller outlet blade height and $\dot{m}$ is the refrigerant mass flow rate. For two-phase outlet states, ρ_2 is replaced by the homogeneous-equilibrium mixture density.

3.4 Impeller Inlet Geometry and Mach Number

The inlet meridional velocity is defined independently from the outlet meridional velocity as shown in Eq. (37),

$$C_{m1} = \phi_1 U_2 \quad (37)$$

where C_{m1} is the inlet meridional velocity and ϕ_1 is the inlet flow coefficient. This allowed the inlet geometry to be treated as a design variable rather than forcing $C_{m1} = C_{m2}$. The required inlet flow area is calculated as shown in Eq. (38). Eq. (39) is defined for a specified inlet hub-to-tip diameter ratio,

$$A_1 = \dot{m} / (\rho_1 C_{m1}) \quad (38)$$

$$\nu = D_{1h} / D_{1t} \quad (39)$$

where A_1 is the inlet annulus area, ν is the hub-to-tip ratio, D_{1h} is the inlet hub diameter, and D_{1t} is the inlet tip diameter. The inlet tip diameter is obtained from Eq. (40), and then the inlet hub-diameter can be defined as Eq. (41). Following this, the inlet mean diameter, D_{1m} , is defined in Eq. (42). Then, the inlet blade speed at the mean diameter is given in Eq. (43),

$$D_{1t} = \sqrt{4A_1 / (\pi(1 - \nu^2))} \quad (40)$$

$$D_{1h} = \nu D_{1t} \quad (41)$$

$$D_{1m} = (D_{1t} + D_{1h}) / 2 \quad (42)$$

$$U_{1m} = \omega D_{1m} / 2 \quad (43)$$

where U_{1m} is the inlet mean blade speed. With zero inlet pre-whirl ($C_{\theta 1} = 0$), where $C_{\theta 1}$ is the inlet tangential component of the absolute velocity. The inlet absolute velocity can be defined in Eq. (44). Then, the inlet relative velocity can be calculated as shown in Eq. (45),

$$C_1 = \sqrt{C_{m1}^2 + C_{\theta 1}^2} \quad (44)$$

$$W_1 = \sqrt{C_{m1}^2 + (U_{1m} - C_{\theta 1})^2} \quad (45)$$

where C_1 and W_1 are the inlet absolute and relative velocities, respectively. Following this, the inlet relative flow angle is calculated as given in Eq. (46),

$$\beta_1 = \tan^{-1} \left(\frac{C_{m1}}{U_{1m} - C_{\theta 1}} \right) \quad (46)$$

where β_1 is the inlet relative flow angle. The inlet absolute and relative Mach numbers are defined in Eqs. (47) and (48), respectively,

$$M_{\text{abs},1} = C_1/a_1 \quad (47)$$

$$M_{\text{rel},1} = W_1/a_1 \quad (48)$$

where $M_{\text{abs},1}$ and $M_{\text{rel},1}$ are the inlet absolute and relative Mach numbers. For single-phase outlet states, the outlet Mach numbers are calculated in Eqs. (49) and (50), respectively,

$$M_{\text{abs},2} = C_2/a_2 \quad (49)$$

$$M_{\text{rel},2} = W_2/a_2 \quad (50)$$

where a_2 is the single-phase speed of sound at the compressor outlet. For two-phase outlet states, HEM-based wet-region indicators can be calculated by using Eqs. (51) and (52), respectively,

$$M_{\text{abs},2,\text{HEM}} = C_2/a_{\text{HEM}} \quad (51)$$

$$M_{\text{rel},2,\text{HEM}} = W_2/a_{\text{HEM}} \quad (52)$$

where $M_{\text{abs},2,\text{HEM}}$ and $M_{\text{rel},2,\text{HEM}}$ are the HEM-based absolute and relative wet-region Mach indicators. These quantities were used only to compare the relative severity of wet-outlet operation among candidate refrigerants and design points.

3.5 Design-Space Exploration and Feasibility Criteria

The inlet hub-to-tip diameter ratio was fixed at 0.50 for all refrigerants and design points. The reference backswept blade angle used in the slip-factor estimate was 60° . The design-space sweep used the following sampled values: ϕ_1 from 0.08 to 0.18 with the interval of 0.01, ϕ_2 from 0.05 to 0.14 with the interval of 0.01,

ψ from 0.60 to 0.85 with the interval of 0.05, and Z from 12 to 20 with the interval of 2. This gives $11 \times 10 \times 6 \times 5 = 3300$ design combinations for each refrigerant. The preliminary mean-line model is evaluated over the following design space given in Eq. (53),

$$\begin{aligned} 0.08 &\leq \phi_1 \leq 0.18 \\ 0.05 &\leq \phi_2 \leq 0.14 \\ 0.60 &\leq \psi \leq 0.85 \\ 12 &\leq Z \leq 20 \end{aligned} \quad (53)$$

where ϕ_1 , ϕ_2 , ψ , and Z are the inlet flow coefficient, outlet flow coefficient, work coefficient, and blade number, respectively. Each refrigerant is evaluated for 3300 combinations. A design point is classified as preliminarily acceptable if it satisfies the following geometric and aerodynamic constraints defined in Eq. (54),

$$\begin{aligned} M_{\text{rel},1} &< 1.0 \\ \frac{D_{1t}}{D_2} &< 0.85 \\ 0.005 &< b_2/D_2 < 0.08 \\ 5^\circ &< \beta_1 < 35^\circ \\ 5^\circ &< \beta_{2,\text{flow}} < 45^\circ \\ 10^\circ &< \beta_{2,\text{metal}} < 70^\circ \end{aligned} \quad (54)$$

For two-phase outlet cases, the additional wet-region indicator constraints are shown in Eq. (55), while the corresponding outlet relative Mach constraint for single-phase outlet cases is shown in Eq. (56),

$$\begin{aligned} M_{\text{abs},2,\text{HEM}} &< 2.0 \\ M_{\text{rel},2,\text{HEM}} &< 1.0 \end{aligned} \quad (55)$$

$$M_{\text{rel},2} < 1.0 \quad (56)$$

Because of the uncertainty associated with the two-phase acoustic approximation, the HEM-based Mach quantities were used as diagnostic wet-region severity indicators rather than firm compressor acceptance limits. They were retained to compare the relative severity of wet-outlet operation between refrigerants and design points, but the final interpretation of a wet-outlet case should rely on detailed two-phase modelling and, ultimately, experimental data. The reported acceptable design fractions should therefore be interpreted as preliminary screening indicators within the selected assumptions, not as definitive compressor feasibility maps.

The above criteria were selected as preliminary screening limits rather than final compressor design rules. Their purpose was to remove clearly impractical or high-risk one-dimensional designs before more detailed blade design, diffuser design, loss modelling, or computational fluid dynamics analysis. The inlet relative Mach number limit was used to avoid transonic inducer operation at the preliminary stage. The inlet-to-outlet diameter ratio limit was imposed to prevent excessive inlet diameter relative to the impeller exit, which would lead to unfavorable inlet geometry. The outlet blade-height ratio limits were used to exclude unrealistically narrow passages, where clearance and manufacturing effects would become dominant, as well as excessively tall passages, which would indicate impractical outlet geometry. The blade-angle limits were used to avoid extreme velocity-triangle solutions and unrealistic backswept blade-

metal angles. For wet-outlet cases, the HEM-based Mach limits were used only as severity indicators for preliminary screening and should not be interpreted as validated two-phase compressor design limits.

It must be noted that these criteria are used only for preliminary design-space classification. They are not intended to replace detailed blade design, diffuser design, or loss-based performance prediction. The present model should therefore be interpreted as a preliminary compressor-design screening tool rather than a validated performance-prediction model. No detailed impeller, diffuser, leakage, incidence, clearance, windage, bearing, motor, or mechanical loss correlations were included. The initial compressor isentropic efficiency was used only to define a consistent thermodynamic outlet state and enthalpy rise for each refrigerant. The subsequent one-dimensional calculation evaluates velocity triangles, geometry ratios, Mach-number indicators, outlet phase behavior, and preliminary feasibility trends. Consequently, the model cannot predict final compressor efficiency or operating maps. Experimental validation and loss-based mean-line modelling are required before the approach can be used for final compressor design.

4 Results and Discussion

The multi-fluid mean-line analysis was conducted for the brewery hot-water case using R-601/n-pentane, R-601a/isopentane, and R-600/n-butane. The objective was not only to generate a preliminary compressor geometry, but also to determine whether the limitations observed in the initial thermodynamic screening were intrinsic to the operating point or could be mitigated through aerodynamic design choices. The results are discussed in four parts. First, the outlet phase behavior is analyzed because it defines whether the compressor operates in a single-phase or wet-outlet regime. Second, the inlet design maps are used to assess whether the inlet Mach number and inlet-to-outlet diameter ratio impose a strong geometric limitation. Third, the outlet Mach indicators and blade-angle maps are interpreted. Finally, the refrigerants are compared using representative preliminary design points.

4.1 Outlet Phase Behavior and Wet-Compression Constraint

The outlet phase condition provided the first important distinction between the refrigerants. For R-601/n-pentane, the actual compressor outlet state entered the two-phase region with an outlet vapor quality of $x_2 = 0.877$; for R-601a/isopentane, the outlet quality was slightly lower, $x_2 = 0.852$. Thus, both C5 hydrocarbons produced wet-outlet operation under the investigated boundary conditions. The predicted wet-outlet states should be interpreted as equilibrium thermodynamic outlet states, not as direct evidence that fully developed equilibrium wet compression occurs throughout the impeller. In the present model, the outlet phase is determined only from the calculated thermodynamic state at p_2 and h_2 . Therefore, the model identifies whether the final equilibrium outlet state lies inside the two-phase region, but it does not resolve the local thermodynamic path inside the impeller. It does not determine where condensation begins, whether sufficient residence time exists for equilibrium condensation to be established, or how liquid droplets, liquid films, phase slip, blade-surface interactions, and two-phase losses develop. This distinction is particularly important because the predicted outlet vapor qualities correspond to non-negligible liquid mass fractions. The values $x_2 = 0.877$ for R-601 and $x_2 = 0.852$ for R-601a denote liquid mass fractions of approximately 12.3% and 14.8%, respectively, at the equilibrium outlet state. These values should therefore be treated as a strong warning flag for detailed two-phase compressor analysis rather than as a resolved prediction of the internal two-phase flow structure. In this sense, the term wet-outlet operation in the present study refers to the equilibrium outlet state predicted by the real-fluid thermodynamic calculation, whereas confirmed wet compression inside the impeller would require additional non-equilibrium phase-change modelling, two-phase loss modelling, and experimental validation.

The corresponding void fractions were high, indicating that the outlet flow volume was still vapor-dominated. However, the liquid mass fraction was not negligible. This distinction is important: the flow may be volumetrically dominated by vapor, but the presence of liquid mass can still affect losses, blade-surface interaction, and wet-compression reliability. In contrast, R-600/n-butane remained single-phase at the compressor outlet. This result is particularly interesting because n-butane had been less attractive in the preliminary thermodynamic screening due to its smaller critical-temperature margin and lower COP. The mean-line model therefore revealed a trade-off: the C5 hydrocarbons provided larger critical-temperature margins under the selected condensing level, but they introduced wet-outlet operation; n-butane avoided the wet-outlet constraint, but it remained more constrained with respect to the high condensing temperature. Since the simplified cycle calculation in the present study was used only to define compressor boundary conditions, no final heat-pump COP ranking is made here.

This result demonstrates why thermodynamic boundary-condition screening alone is insufficient for compressor selection. A refrigerant that appears suitable from pressure-ratio and critical-temperature considerations may still lead to an outlet state that complicates compressor design. Conversely, a refrigerant with a cleaner compressor outlet state may be more constrained by its proximity to the critical temperature at the required condensing level. It is also worth mentioning that the HEM-based quantities should be interpreted carefully. They do not represent validated two-phase compressor Mach numbers and do not account for non-equilibrium phase change, droplet formation, liquid-film effects, phase slip, blade-surface interaction, or two-phase loss mechanisms. Their role in the present work is limited to providing first-order wet-region severity indicators that allow different refrigerants and design points to be compared consistently at the preliminary stage. Therefore, the wet-outlet HEM indicators are used to flag a potential design risk, not to predict the detailed aerodynamic performance of a two-phase compressor.

4.2 Inlet Design Maps

Fig. 3 compares the inlet-side design space for the three refrigerants by presenting the inlet relative Mach number as a function of the inlet-to-outlet diameter ratio. The three panels show very similar trends, which indicates that the inlet-side behavior was governed primarily by the selected inlet flow coefficient rather than by the refrigerant identity alone. As ϕ_1 increased, the inlet meridional velocity increased and the required inlet annulus area decreased. Consequently, the inlet tip diameter became smaller relative to the impeller outlet diameter, moving the designs toward lower D_{1t}/D_2 . This trend is important because an excessively large inlet diameter would imply an unfavorable impeller geometry and could also increase the inlet blade speed.

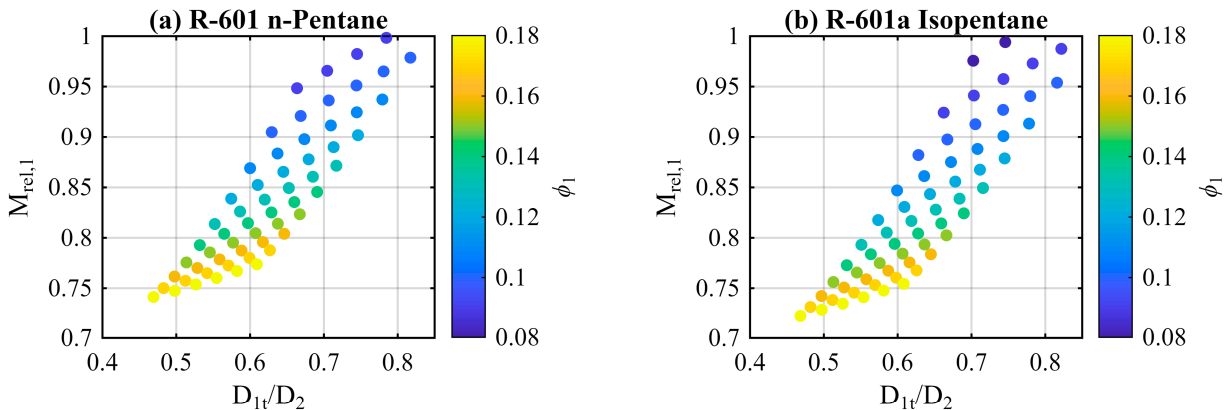


Figure 3: Cont.

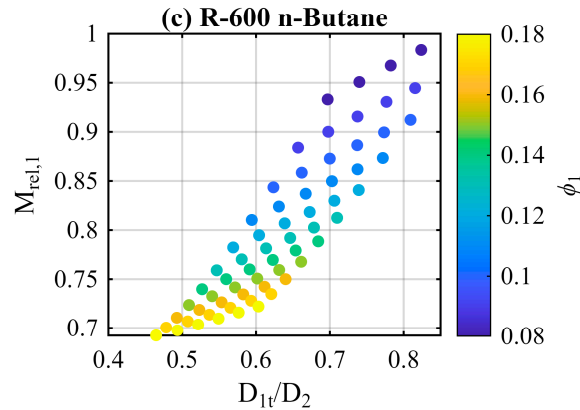


Figure 3: Mach number and diameter ratio for compressor inlet; (a) R-601, (b) R-601a, and (c) R-600.

The figure also shows that the inlet relative Mach number remained below unity for a broad portion of the acceptable design space. The representative designs gave $M_{rel,1} = 0.804$, 0.784 , and 0.750 for R-601, R-601a, and R-600, respectively, while D_{1t}/D_2 remained close to 0.64 – 0.65 . These values suggest that the inlet could be designed within a reasonable preliminary range for all three refrigerants. R-600 gave the lowest inlet relative Mach number, mainly because its higher density and smaller required impeller diameter led to a more compact inlet arrangement.

The key implication of Fig. 3 is that the inlet limitation was design dependent. In the initial single-point calculation, a high inlet relative Mach number appeared as a possible constraint. However, once ϕ_1 was treated as an independent design variable, the inlet Mach number and inlet diameter ratio could be reduced simultaneously. Therefore, the inlet-side risk was not an unavoidable consequence of the thermodynamic operating point. Instead, it was a meanline-design issue that could be mitigated by appropriate selection of inlet meridional velocity and inlet geometry.

The inlet Mach numbers shown in Fig. 3 are evaluated at the inlet mean diameter. This is useful for mean-line comparison, but it does not prove that the full inlet span remains subsonic. Under the zero-prewhirl assumption, the blade speed and relative velocity increase toward the inlet tip. Therefore, an additional inlet-tip relative Mach check was performed for the representative designs. The estimated inlet-tip relative Mach numbers were approximately 1.05 for R-601, 1.02 for R-601a, and 0.98 for R-600. These values show that the inlet constraint is mitigated at the mean-line level, but the tip region may approach or slightly exceed transonic conditions, especially for the C5 hydrocarbons. Therefore, the present model should not be interpreted as proving choke-free inlet operation. A detailed inducer and throat analysis is required in the next design stage.

4.3 Outlet Mach Indicators

Fig. 4 compares the outlet compressibility behavior of the three refrigerants. For R-601 and R-601a, the compressor outlet state was in the two-phase region, and the plotted quantities therefore represent HEM-based wet-region Mach indicators. For R-600, the outlet remained single-phase, and conventional outlet Mach numbers were calculated. This distinction is essential: the indicators for R-601 and R-601a should not be interpreted as validated two-phase compressor Mach numbers, but as first-order measures of wet-outlet compressibility severity under the homogeneous-equilibrium approximation. A consistent trend is visible in all three panels. Increasing the work coefficient ψ shifts the points toward lower relative outlet Mach values. This follows directly from the relation $U_2 = \sqrt{\Delta h/\psi}$: for a fixed enthalpy increase, a higher

work coefficient reduces the required tip speed and changes the outlet velocity triangle. The representative designs selected $\psi = 0.85$, which corresponded to relatively low relative outlet indicators. The representative values were $M_{\text{rel},2,\text{HEM}} = 0.329$ for R-601, $M_{\text{rel},2,\text{HEM}} = 0.339$ for R-601a, and $M_{\text{rel},2} = 0.341$ for R-600. Thus, the impeller-relative outlet flow did not appear to be the most restrictive aerodynamic feature.

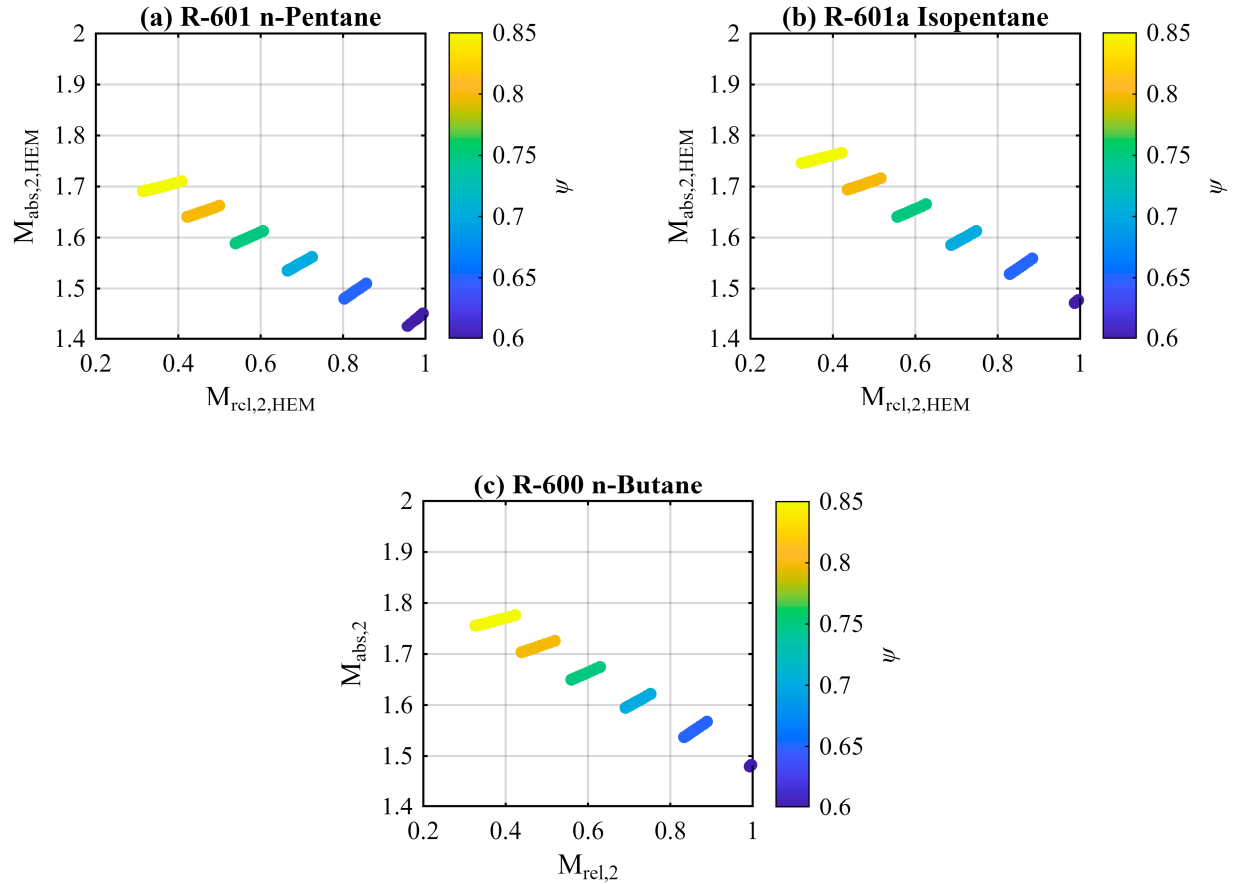


Figure 4: Outlet compressibility performance for (a) R-601, (b) R-601a, and (c) R-600.

In contrast, the absolute outlet indicators remained high. The representative values were $M_{\text{abs},2,\text{HEM}} = 1.693$ for R-601, $M_{\text{abs},2,\text{HEM}} = 1.749$ for R-601a, and $M_{\text{abs},2} = 1.758$ for R-600. This result suggests that the diffuser-entry condition may be more critical than the impeller-relative outlet flow. Even though R-600 avoided wet-outlet operation, its absolute outlet Mach number was comparable to the HEM-based indicators of the C5 hydrocarbons. Therefore, avoiding wet outlet did not remove the need for careful diffuser design. Fig. 4 thus separates two issues: wetness was refrigerant- and thermodynamic-state dependent, whereas the high absolute outlet velocity was a broader compressor-design challenge.

4.4 Blade Number and Outlet Metal Angle

Fig. 5 shows the estimated outlet blade metal angle as a function of blade number for the three refrigerants. The distributions are nearly identical across R-601, R-601a, and R-600, which indicates that this part of the preliminary design space was controlled mainly by the slip-factor formulation and the selected aerodynamic design variables, rather than by the refrigerant thermodynamic state. This is expected because $\beta_{2,\text{metal}}$ was determined from the outlet velocity triangle, the slip factor, the outlet flow coefficient, the work

coefficient, and the blade number. The figure shows that blade number has a clear effect on the estimated blade metal angle. For a fixed aerodynamic operating point, increasing Z increases the slip factor and reduces the difference between the ideal blade-exit tangential component and the actual post-slip tangential component. As a result, the estimated metal angle shifts toward more moderate values. Low blade numbers can lead to large metal angles for some combinations of ϕ_2 and ψ , while higher blade numbers provide a wider region of practically acceptable blade angles. The selected representative design used $Z = 20$, giving $\beta_{2,\text{metal}} = 45.35^\circ$ for all three refrigerants. The corresponding relative outlet flow angle was $\beta_{2,\text{flow}} = 25.02^\circ$. The difference between these two angles reflects the role of slip: the blade geometry must generate a stronger ideal tangential component than the final post-slip flow carries. This separation between flow angle and metal angle is useful in a preliminary meanline model because it allows the velocity triangle and the blade geometry requirement to be examined independently. The main conclusion from Fig. 5 is that the outlet blade-angle constraint was design-dependent. Unlike the wet-outlet behavior of R-601 and R-601a, which persisted across the aerodynamic design space, the blade-angle distribution could be adjusted by changing Z , ϕ_2 , and ψ . Therefore, blade metal angle should be treated as an aerodynamic design variable, not as an intrinsic refrigerant limitation.

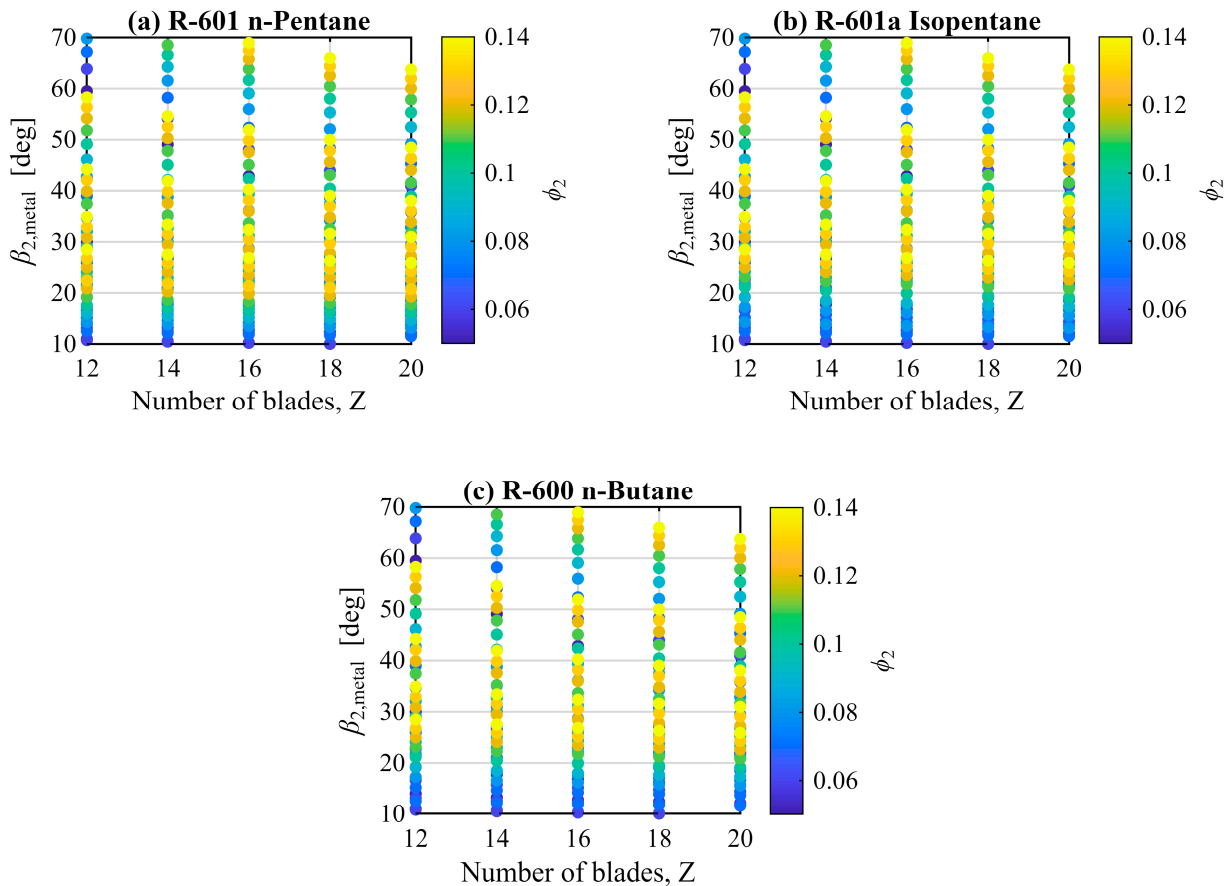


Figure 5: Relation between blade angle and blade number for (a) R-601, (b) R-601a, and (c) R-600.

4.5 Design-Space Fraction and Representative Preliminary Design Comparison

Fig. 6a compares the acceptable preliminary design-space fraction for the three refrigerants. Out of 3300 evaluated design combinations, the acceptable numbers were (i) 2699 for R-601/n-pentane, (ii) 2562

for R-601a/isopentane, and (iii) 2607 for R-600/n-butane. The corresponding acceptable fractions were approximately 0.818, 0.776, and 0.790. These results show that all three refrigerants provided broad acceptable preliminary design spaces according to the selected geometric and aerodynamic criteria. This finding is significant because it indicates that none of the three refrigerants was eliminated by the mean-line geometry constraints alone. The compressor design space was sufficiently broad to identify acceptable combinations for each fluid. Therefore, the final refrigerant assessment cannot be based only on whether a feasible mean-line geometry exists. Instead, the outlet phase condition, thermodynamic performance, critical-temperature margin, and diffuser-entry indicators must be considered together. The acceptable design fractions should be interpreted only within the sampled design space and the selected screening limits. They are not general refrigerant suitability metrics. The classification depends on the chosen ranges of ϕ_1 , ϕ_2 , ψ , and Z , the sampling increments, and the geometric and aerodynamic thresholds. In the present sweep, rejected points were associated with one or more violations of the inlet relative Mach limit, inlet-to-outlet diameter-ratio limit, outlet blade-height ratio, blade-angle range, or wet-region diagnostic limits. Since several criteria can be violated simultaneously, the rejection categories are not mutually exclusive. The representative design point was selected using a ranking criterion that favored low inlet relative Mach number, moderate inlet-to-outlet diameter ratio, moderate outlet blade-height ratio, and an outlet blade metal angle close to a practical value. The selected point lies at the upper bound of the investigated work coefficient and blade-number ranges, which indicates that the current ranking favors higher ψ and higher Z within the present preliminary constraints. Therefore, the representative point should be interpreted as a useful comparison point, not as a globally optimized compressor design.

In addition, Fig. 6b compares the representative preliminary design points selected for the three refrigerants. The same design-variable combination was selected for all three fluids as follows; $\phi_1 = 0.16$, $\phi_2 = 0.07$, $\psi = 0.85$, and $Z = 20$. This consistency is useful because it allows the refrigerant effects to be compared under a similar aerodynamic design strategy. The impeller outlet diameter, D_2 , decreased from C5 hydrocarbons to n-butane. The representative values were found to be (i) 0.338 m for R-601, (ii) 0.330 for R-601a, and (iii) 0.266 for R-600. The smaller diameter for n-butane is consistent with its higher pressure level and lower volumetric flow requirement. This makes n-butane attractive from a compactness perspective. The inlet-to-outlet diameter ratios were very similar, lying between 0.640 and 0.646. The outlet blade-height ratios were also close, ranging from approximately 0.029 to 0.032. These similarities suggest that, once the design variables are selected appropriately, the basic impeller geometry can be brought into a comparable preliminary range for all three refrigerants. However, this geometric similarity should not obscure the important thermodynamic distinction: R-601 and R-601a remained wet at the outlet, while R-600 stayed single-phase. Thus, the representative design comparison confirms that geometric feasibility and outlet phase feasibility are separate design dimensions.

The compressor dimensions obtained here are also consistent with the order of magnitude discussed in previous centrifugal-compressor studies for refrigerant-based and high-temperature heat-pump applications. Prior coupled cycle-compressor studies have shown that working-fluid selection can significantly affect compressor rotational speed, impeller geometry, efficiency, Mach number, and loss distribution [21–24]. The rotational speeds in the present work, approximately 16.7–21.0 krpm, and the outlet diameters, approximately 0.266–0.338 m, therefore appear reasonable for a preliminary large-scale refrigerant compressor screening exercise. This comparison does not constitute validation of the present model, because no detailed loss prediction or experimental compressor map is included. However, it indicates that the obtained preliminary geometries are not outside the expected range of published centrifugal-compressor designs for comparable thermal applications.

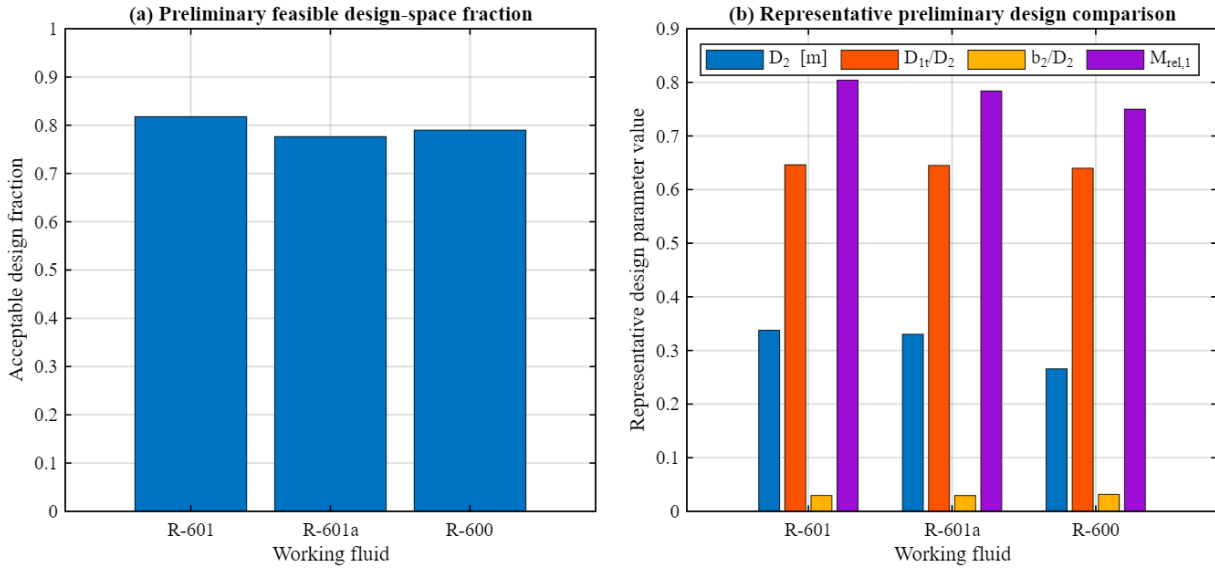


Figure 6: Preliminary refrigerant performances according to (a) feasible design-space fraction and (b) representative design comparison.

4.6 Outlet Condition and Absolute Mach Comparison

Fig. 7 summarizes the outlet condition and the absolute outlet Mach or HEM-based absolute indicator. For R-601 and R-601a, the outlet quality was below unity, confirming wet-outlet operation. For R-600, the outlet was single-phase and was represented by a dry-vapor indicator of unity. The absolute outlet indicators were similar in magnitude for all three refrigerants. R-601 and R-601a gave HEM-based indicators of 1.693 and 1.749, respectively. R-600 gave a single-phase absolute outlet Mach number of 1.758. This result has two implications. First, the wet-outlet issue for R-601 and R-601a is not simply a consequence of excessive velocity. It is tied to the thermodynamic outlet state. Second, even when the outlet remains single-phase, as for R-600, the absolute outlet Mach level can still be high. Therefore, diffuser design remains important for all refrigerants. The comparison also clarifies the difference between intrinsic and design-dependent risks. The wet-outlet condition for R-601 and R-601a persisted across the investigated aerodynamic design space, because the outlet quality was determined by the thermodynamic state. By contrast, the inlet Mach number, inlet diameter ratio, blade metal angle, and blade-height ratio varied significantly with ϕ_1 , ϕ_2 , ψ , and Z . These were therefore design-dependent risks that could be mitigated by appropriate mean-line choices.

It should be emphasized that the term intrinsic is used here only within the present fixed-state modelling framework. In the design-space sweep, the aerodynamic variables change the velocity triangles, geometry ratios, and Mach-number indicators, but they do not change the thermodynamic outlet state because p_2 and h_2 are fixed before the mean-line loop. Therefore, the wet-outlet condition is intrinsic to the specified boundary conditions and thermodynamic assumptions of the present model. This statement should not be interpreted as proof that wetness is independent of compressor geometry in a real machine. In practice, geometry-dependent losses, local static pressure and temperature distributions, residence time, non-equilibrium condensation, and diffuser behavior may affect the actual onset and development of condensation.

The wet-outlet result is also sensitive to assumptions that change the thermodynamic outlet state, particularly inlet superheat, compressor isentropic efficiency, and condensing temperature. Increasing the

inlet superheat or reducing the assumed isentropic efficiency would increase the actual outlet enthalpy and could move the outlet state toward the superheated vapor region. On the contrary, changing the condensing level would shift the outlet pressure and saturation boundary. Therefore, the wet-outlet conclusion should be interpreted for the specified brewery boundary condition, 5 K inlet superheat, 2 K subcooling, and 0.70 initial isentropic efficiency, rather than as a universal statement for all operating conditions.

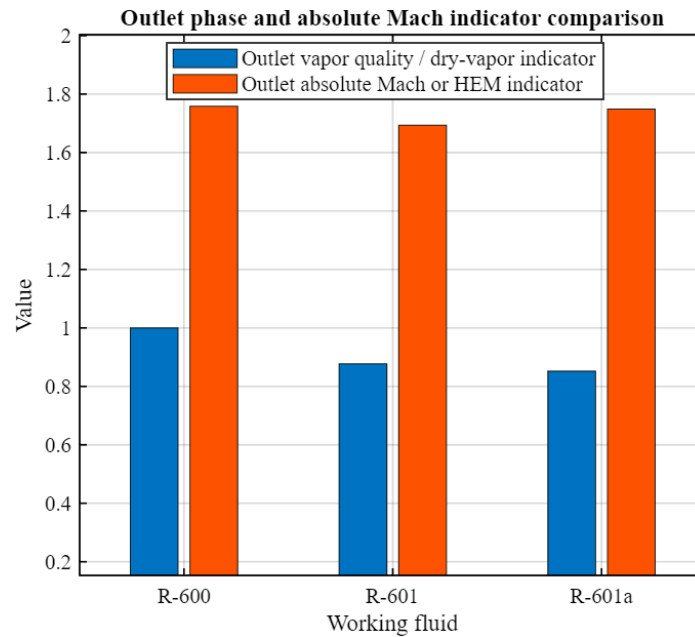


Figure 7: Outlet phase and absolute Mach indicator comparison for the representative preliminary compressor designs. Note: the “value” represents a dimensionless value.

Overall, the results demonstrate that the proposed mean-line framework can separate different types of compressor-design limitations. In the preliminary thermodynamic screening, n-pentane and isopentane appeared attractive because of their favorable cycle-level behavior. However, the mean-line model showed that both refrigerants led to wet-outlet operation for the selected brewery boundary condition. This finding does not necessarily eliminate these refrigerants, but it changes the design interpretation. They should not be treated simply as dry centrifugal compressor candidates; instead, their use would require explicit consideration of wet-outlet compression, condensation behavior, and diffuser-entry flow. n-butane presented the opposite trade-off. It avoided wet-outlet operation and produced the most compact representative impeller. However, its earlier thermodynamic disadvantages, including lower COP and very small critical-temperature margin, remain important. Therefore, n-butane cannot be selected solely because it provides a dry outlet. Its narrow thermal margin makes it sensitive to operating-condition variation and limits its robustness for high-temperature process heating. The main originality of the model lies in this separation of constraints. A conventional thermodynamic screening would primarily rank refrigerants by COP, pressure ratio, discharge temperature, or critical-temperature margin. A conventional preliminary compressor sizing would mainly report diameter, speed, and Mach numbers.

The present framework connects these two views and adds a diagnostic layer: it identifies which risks are imposed by the thermodynamic state and which can be addressed through aerodynamic design. For the brewery case, the wet-outlet behavior of R-601 and R-601a was intrinsic to the selected thermodynamic operating point. The inlet Mach and geometry limitations were not intrinsic; they were reduced by

treating the inlet flow coefficient as an independent design variable. The blade-angle constraint was also design-dependent, responding to blade number and outlet flow coefficient. The diffuser-entry challenge remained relevant for all refrigerants because the absolute outlet Mach or HEM-based absolute indicator remained high. These findings suggest that early-stage compressor design for high-temperature thermal processes should not rely on a single performance metric. Refrigerant selection, outlet phase behavior, inlet geometry, impeller velocity triangles, and diffuser-entry indicators must be evaluated together. The proposed framework provides a practical way to perform this assessment before moving to detailed mean-line loss modelling, blade design, and full compressor-unit co-design.

4.7 Sensitivity of Wet-Outlet Interpretation

Since the wet-outlet conclusion is central in the present study, its interpretation should be linked directly to the thermodynamic assumptions used to define the compressor outlet state. The outlet phase is determined by the calculated state at p_2 and h_2 , rather than by the mean-line geometry itself. Therefore, changes in inlet superheat, compressor isentropic efficiency, or condensing pressure can shift the outlet state relative to the saturation boundary. A higher inlet superheat or a lower compressor efficiency increases the actual outlet enthalpy and may move the state toward or further into the superheated vapor region, whereas a higher efficiency can keep the actual outlet state closer to the isentropic path and may increase the risk of entering the two-phase region for dry fluids with an overhanging saturation curve. Within the investigated brewery boundary condition, the important result is that the wet-outlet behavior of R-601 and R-601a is imposed by the thermodynamic outlet state and not by the selected mean-line geometry. Varying ϕ_1 , ϕ_2 , ψ , or Z changes the velocity triangles, Mach-number indicators, blade angles, and geometry ratios, but it does not change p_2 , h_2 , or the corresponding outlet quality. Therefore, the wet-outlet condition for the C5 hydrocarbons should be interpreted as an intrinsic thermodynamic constraint under the specified boundary conditions and assumptions. It should not be generalized to all operating conditions without reassessing the outlet state for the selected superheat, efficiency, and condensation level.

5 Conclusion

This study developed a real-gas, one-dimensional preliminary mean-line framework for evaluating high-speed centrifugal compressors in hydrocarbon-based high-temperature thermal processes. The framework was applied to a reference case, which was a brewery hot-water case using R-601/n-pentane, R-601a/isopentane, and R-600/n-butane. The purpose was not only to estimate preliminary compressor geometry, but also to identify whether the main design limitations were intrinsic to the thermodynamic operating point or dependent on aerodynamic design choices. The model combined thermodynamic state calculation, phase detection, homogeneous-equilibrium continuation for wet-outlet operation, velocity-triangle analysis, slip-factor-based blade-angle estimation, inlet geometry calculation, and preliminary feasibility classification. A total of 3300 design combinations were evaluated for each refrigerant by varying the inlet flow coefficient, outlet flow coefficient, work coefficient, and blade number.

The results showed that all three refrigerants provided broad preliminary aerodynamic design spaces. The acceptable design fractions were 0.818 for R-601, 0.776 for R-601a, and 0.790 for R-600. A common representative design region was found at $\phi_1 = 0.16$, $\phi_2 = 0.07$, $\psi = 0.85$, and $Z = 20$. For this design point, the impeller outlet diameters were 0.338 m for R-601, 0.330 m for R-601a, and 0.266 m for R-600. The corresponding inlet relative Mach numbers were 0.804, 0.784, and 0.750, respectively, indicating that the inlet aerodynamic constraint could be managed through appropriate mean-line design. The most important distinction appeared at the compressor outlet. R-601 and R-601a entered the two-phase region,

with outlet vapor qualities of 0.877 and 0.852, respectively. In contrast, R-600 remained single-phase at the outlet. Therefore, under the fixed thermodynamic boundary conditions used in the present model, the wet-outlet behavior of R-601 and R-601a is a thermodynamic-state constraint rather than a consequence of the swept mean-line design variables. However, R-600's dry outlet must be balanced against its weaker thermodynamic performance and limited critical-temperature margin.

A limitation of the present study is that the one-dimensional model has not yet been validated against experimental compressor data and does not include detailed loss correlations. The results should therefore be interpreted as preliminary design-space and phase-behavior indicators. Future work should couple the present framework with validated impeller and diffuser loss models, off-design compressor maps, and experimental data for hydrocarbon high-speed centrifugal compressors.

The study demonstrated that thermodynamic refrigerant screening and compressor mean-line feasibility can lead to different rankings. C5 hydrocarbons were attractive thermodynamically but introduced wet-outlet operation, while n-butane provided a compact and dry compressor outlet but remained thermodynamically marginal. The proposed framework therefore offered a structured way to separate thermodynamic constraints from design-dependent aerodynamic risks. Future work should extend the model by including detailed loss correlations, diffuser modelling, incidence and blade-loading losses, clearance and disk-friction effects, and iterative coupling between cycle performance and compressor efficiency. For wet-outlet candidates, a more advanced non-equilibrium two-phase compressor model should also be developed to assess condensation dynamics, phase slip, droplet effects, and wet-compression losses beyond the homogeneous-equilibrium approximation used in this preliminary framework.

Acknowledgement: The author gratefully acknowledges the support of the Section of Mechanical Engineering at the University of Southern Denmark. The author thanks Paride Gullo for valuable discussions on the brewery hot-water case considered in this case study. The author also thanks his partner, Begum Tanis, for her patience, understanding, and encouragement during the preparation of this work, much of which was carried out while visiting her in the Netherlands. He also acknowledges FlixBus for providing a suitable environment for reading and reviewing the literature during his frequent journeys between Denmark and the Netherlands. Finally, he thanks Miso the cat for allowing him to work from home with inner peace.

Funding Statement: The author received no specific funding for this study.

Availability of Data and Materials: The data that support the findings of this study are available from the author, upon reasonable request.

Ethics Approval: Not applicable.

Conflicts of Interest: The author declares no conflicts of interest.

Abbreviations

The following abbreviations are used in this manuscript:

COP	Coefficient of Performance
HEM	Homogeneous Equilibrium Model

References

1. Rehfeldt M, Fleiter T, Toro F. A bottom-up estimation of the heating and cooling demand in European industry. *Energy Effic.* 2018;11(5):1057–82. [[CrossRef](#)].

2. Naegler T, Simon S, Klein M, Gils HC. Quantification of the European industrial heat demand by branch and temperature level: quantification of European industrial heat demand. *Int J Energy Res.* 2015;39(15):2019–30. [[CrossRef](#)].
3. Kosmadakis G. Estimating the potential of industrial (high-temperature) heat pumps for exploiting waste heat in EU industries. *Appl Therm Eng.* 2019;156:287–98. [[CrossRef](#)].
4. Chua KJ, Chou SK, Yang WM. Advances in heat pump systems: a review. *Appl Energy.* 2010;87(12):3611–24. [[CrossRef](#)].
5. Zhang X, Keramati H, Arie M, Singer F, Tiwari R, Shooshtari A, et al. Recent developments in high temperature heat exchangers: a review. *Front Heat Mass Transf.* 2018;11:1–14. [[CrossRef](#)].
6. Bamigbetan O, Eikevik TM, Neksa P, Bantle M. Review of vapour compression heat pumps for high temperature heating using natural working fluids. *Int J Refrig.* 2017;80:197–211. [[CrossRef](#)].
7. Zhang J, Zhang HH, He YL, Tao WQ. A comprehensive review on advances and applications of industrial heat pumps based on the practices in China. *Appl Energy.* 2016;178:800–25. [[CrossRef](#)].
8. Mateu-Royo C, Arpagaus C, Mota-Babiloni A, Navarro-Esbrí J, Bertsch SS. Advanced high temperature heat pump configurations using low GWP refrigerants for industrial waste heat recovery: a comprehensive study. *Energy Convers Manag.* 2021;229:113752. [[CrossRef](#)].
9. Frate GF, Ferrari L, Desideri U. Analysis of suitability ranges of high temperature heat pump working fluids. *Appl Therm Eng.* 2019;150:628–40. [[CrossRef](#)].
10. Chen Z, Wang Y, Wu H, Zhao W, Yan J, Peng L, et al. Experimental investigation on heating performance and frosting behavior of an integrated R290 secondary loop heat pump. *Front Heat Mass Transf.* 2026;24(2):4. [[CrossRef](#)].
11. Bamigbetan O, Eikevik TM, Neksa P, Bantle M, Schlemminger C. Experimental investigation of a prototype R-600 compressor for high temperature heat pump. *Energy.* 2019;169:730–8. [[CrossRef](#)].
12. Jiang J, Hu B, Wang RZ, Ge T, Liu H, Zhang Z, et al. Experiments of advanced centrifugal heat pump with supply temperature up to 100°C using low-GWP refrigerant R1233zd(E). *Energy.* 2023;263:126033. [[CrossRef](#)].
13. Yang W, Liu Y, Gui C, Shi Y, Ge M. Operation performance of scroll compressor under different pressure ratio conditions. *Front Heat Mass Transf.* 2025;23(6):2051–72. [[CrossRef](#)].
14. Turunen-Saaresti T, Røytta P, Honkatukia J, Backman J. Predicting off-design range and performance of refrigeration cycle with two-stage centrifugal compressor and flash intercooler. *Int J Refrig.* 2010;33(6):1152–60. [[CrossRef](#)].
15. Schiffmann J, Favrat D. Experimental investigation of a direct driven radial compressor for domestic heat pumps. *Int J Refrig.* 2009;32(8):1918–28. [[CrossRef](#)].
16. Schiffmann J, Favrat D. Design, experimental investigation and multi-objective optimization of a small-scale radial compressor for heat pump applications. *Energy.* 2010;35(1):436–50. [[CrossRef](#)].
17. Kus B, Neksa P. Development of one-dimensional model for initial design and evaluation of oil-free CO₂ turbo-compressor. *Int J Refrig.* 2013;36(8):2079–90. [[CrossRef](#)].
18. Liu H, Zhao B, Zhang Z, Li H, Hu B, Wang RZ. Experimental validation of an advanced heat pump system with high-efficiency centrifugal compressor. *Energy.* 2020;213:118968. [[CrossRef](#)].
19. Grauberger A, Young D, Bandhauer T. Experimental validation of an organic Rankine-vapor compression cooling cycle using low GWP refrigerant R1234ze(E). *Appl Energy.* 2022;307:118242. [[CrossRef](#)].
20. Jiang P, Tian Y, Wang B, Guo C. Design and analysis of centrifugal compressor in carbon dioxide heat pump system. *Sci Rep.* 2024;14(1):5286. [[CrossRef](#)].
21. Meroni A, Zühlsdorf B, Elmegaard B, Haglind F. Design of centrifugal compressors for heat pump systems. *Appl Energy.* 2018;232:139–56. [[CrossRef](#)].
22. Uusitalo A, Turunen-Saaresti T, Honkatukia J, Tiainen J, Jaatinen-Värri A. Numerical analysis of working fluids for large scale centrifugal compressor driven cascade heat pumps upgrading waste heat. *Appl Energy.* 2020;269:115056. [[CrossRef](#)].
23. Uusitalo A, Jaatinen-Värri A, Turunen-Saaresti T, Honkatukia J, Tiainen J. Centrifugal compressor design and cycle analysis of large-scale high temperature heat pumps using hydrocarbons. *Appl Therm Eng.* 2024;247:123035. [[CrossRef](#)].

24. Jaatinen-Värri A, Honkatukia J, Uusitalo A, Turunen-Saaresti T. Centrifugal compressor design for high-temperature heat pumps. *Appl Therm Eng.* 2024;239:122087. [[CrossRef](#)].
25. Uusitalo A, Jaatinen-Värri A, Turunen-Saaresti T. Centrifugal compressor design analysis for large-scale trans-critical carbon dioxide heat pumps. *Appl Therm Eng.* 2024;257:124355. [[CrossRef](#)].
26. Uusitalo A, Narsakka J, Nerg J, Abubakar I, Nevaranta N, Honkatukia J, et al. Co-design of high-speed centrifugal compressors for heat pumps including aerodynamic-, electromagnetic-, rotordynamic design, and bearing control. *Results Eng.* 2026;31:111466. [[CrossRef](#)].
27. Yan Y, Yang F, Zhang H, Wang N, Xu Y, Cheng J, et al. A holistic design framework of centrifugal compressors for high temperature heat pumps considering wet compression prevention and global optimization. *Energy Convers Manag.* 2025;342:120142. [[CrossRef](#)].
28. Andersen MP, Poulsen JL, Elmegaard B, Zühlsdorf B. Developing a high-temperature heat pump technology concept using natural refrigerants. *HPT Mag.* 2023;41(1):21–4.
29. Wang R, Xu Z, Palm B, Kim M, Dong Y, Feng J. High-temperature heat pumps for industrial decarbonisation. 61st technical brief on refrigeration technologies. Paris, France: International Institute of Refrigeration; 2026. 31 p. [[CrossRef](#)].
30. Elwardany M, Reddy YSK, Nassif N. Water (R-718) as a refrigerant for high-temperature heat pumps: opportunities, challenges, and future directions. *Appl Therm Eng.* 2026;304:132756. [[CrossRef](#)].
31. Bell IH, Wronski J, Quoilin S, Lemort V. Pure and pseudo-pure fluid thermophysical property evaluation and the open-source thermophysical property library CoolProp. *Ind Eng Chem Res.* 2014;53(6):2498–508. [[CrossRef](#)].
32. Wood AB. A textbook of sound: being an account of the physics of vibrations with special reference to recent theoretical and technical developments. New York, NY, USA: Macmillan; 1930.