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Comparison on Sizing and Performance of Air-based and Refrigerant-based Centrifugal Compressors

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1. Introduction

Centrifugal compressors have widely applications such as in turboshaft engines, refrigeration systems, fuel cell electric vehicles[1]. The design and off-design performance can be estimated from map-based calculations or physics-based one-dimensional models [2, 3]. In the physics-based one-dimensional models, individual losses are computed using analytical equations and empirical correlations which relate loss levels to velocity diagram characteristics and overall geometry. The calculated losses are then used to estimate overall changes in enthalpy loses which in turn used to estimate the outlet pressure and temperature rise. The complete discussion of the loss calculations can be found in [2]. One-dimensional models provide good guidance on preliminary design of centrifugal compressors. However, it is found that the predicted total pressure ratio and efficiency were higher than the experimental measurements, when the compressor mass flow is limited by the impeller[2]. Computational Fluid Dynamics (CFD) [4-7] have been used to understand the complex flow phenomena in centrifugal compressors. Krivitzky et al. [4] addressed some key aerodynamic design challenges in wide-operability, single-stage turbocharger compressors. Sundstrom et al.[5] numerically investigated the flow in a ported-shroud centrifugal compressor under various mass flow rates and speed lines. They studied the flow instability in the compressor flow near surge. The unsteady features of the flow field are quantified by means of Fourier transformation analysis, proper orthogonal decomposition and dynamic mode decomposition. Hung et al.[6] studied off-design performance for a centrifugal refrigerant compressor. Using ANSYS CFX, Sausse et al.[7] conducted CFD simulation on centrifugal compressors with R134a as the fluid medium. Theoretical isentropic efficiency and theoretical surge line were compared to experimental data. It is also worth noting that Kalitzin et al.[8] simulated unsteady flow in screw compressor, with the focuses on compactional aspects such as hybrid structured/unstructured moving grids.

In the previous studies [9], a dynamic vapor cycle modeling toolset ATTMO was developed to address two-phase flow systems. The performance of the centrifugal compressors was modelled using first-principle enthalpy based calculations. In the present study, CFD simulations are used to understand the flow characteristics in the centrifugal compressors, and to further improve first-principle modeling. Using ANSYS CFX to understand flow phenomena and predict the off-design performance of centrifugal compressors. We first start with air as the fluid medium, then extend to refrigerants. In our study, we begin with fundamental input parameters and geometry data, generate a preliminary design of the impeller using Vista Centrifugal Compressor Design (CCD). Then the resulting geometry is passed to BladeGen to generate impeller blades. The obtained centrifugal compressor is then meshed and CFD simulations are conducted using ANSYS CFX. In this work, we conduct both 1D modeling and CFD simulations to study the sizing and performance of air-based and refrigerant-based centrifugal compressors.

2. Baseline centrifugal compressor

We start with a baseline centrifugal compressor, in which the inlet total pressure is 100 kPa, the inlet temperature is 303K, the design flow rate is 1.0kg/s; the design rotational speed is 58,000 rpm, the desired pressure ratio is 6.25, and the number of the impeller blades is 15. No splitters are used and the fluid working medium is air in the baseline case.

2.1 Initial sizing

Vista CCD is a preliminary design tool integrated into ANSYS for centrifugal compressors, it was developed by PCA Engineers Limited. Vista CCD determines the approximate skeletal geometry and the flow parameter for the design, based on the well-established correlations [10]. Initial sizing study is conducted using Vista CCD, and the obtained results are compared with those from ATTMO [11] and NASA models [2]. The compressor design requirements such as the overall pressure ratio, mass flow rate, design rotational speed, total pressure and temperature are listed in Table 1, in which the gas properties are also summarized for air as an ideal gas, such as the gas constant, specific heat ratio, and the inlet dynamic viscosity. The main blades number is 15, and no splitter is used to make a comparison with ATTMO and NASA compressors.

In Table 2, we listed some of the sizing results from ATTMO, NASA models, and Vista CCD. It can be seen that the sizing results from the three models/tools are reasonably close. For example, D_{t1}, the diameter of impeller blade at the inlet, and D_2, the diameter of impeller at the exit, the ATTMO/NASA/Vista CCD show very similar results on these key parameters. The blade angle β, and blade thickness t are also shown good comparisons.

Table 1 Input parameters used in Vista CCD

Duty	Overall pressure ratio	6.25
	Mass flow	1.0 kg/s
	Rotational speed	58,000 rpm
Inlet Stagnation conditions	Temperature	303.15 K
	Pressure	100 KPa
Gas properties	Gas properties model	Ideal gas
	Fluid medium	Air
	Gas constant R	287.048 J/kg K
	Specific heat ratio γ	1.4
	Viscosity μ	1.789e-5 Pa s
Geometry	Main vanes	15
	Inter vanes	0
	Backsweep angle	45°

Table 2 Sizing results

Parameters	ATTMO	NASA	Vista CCD
D_{h1}(cm)	5.08	7.11	6.0
D_{t1}(cm)	10.2	11.5	10.9
t_1 (cm)	0.08	0.06	0.1
β_1 (°)	55.1°	53.9°	55°
D_2 (cm)	21.0	20.4	20.2
t_2 (cm)	0.35	0.22	0.2
β_2 (°)	45°	45°	45°
b_2 (mm)	4.5	5	6.43

2.2 Model generation

More details of geometrical parameters of the centrifugal compressor model generated by Vista CCD and performance at the operating point are summarized in Table 3. Several efficiency correlations are provided in Vista CCD, here the Casey-Robinson correlations[12] is used, and the obtained pressure ratio variation with respect to the mass flow at 100%, 90%, 80%, 70%, 60%, and 50% are presented in Fig. 1.

Table 3 Output parameters in Vista CCD

Impeller leading edge	Hub diameter (cm)	6.0
	RMS diameter (cm)	8.917
	Shroud diameter (cm)	10.992
	Inclination angle at hub	3.0°
	RMS Inclination angle	1.5°
	Inclination angle at shroud	0°
Impeller exit	T_0 (K)	594.8
	P_0 (kPa)	716.71
	P (kPa)	371.10
Overall performance	ϕ	0.035
	η_{poly}	0.777

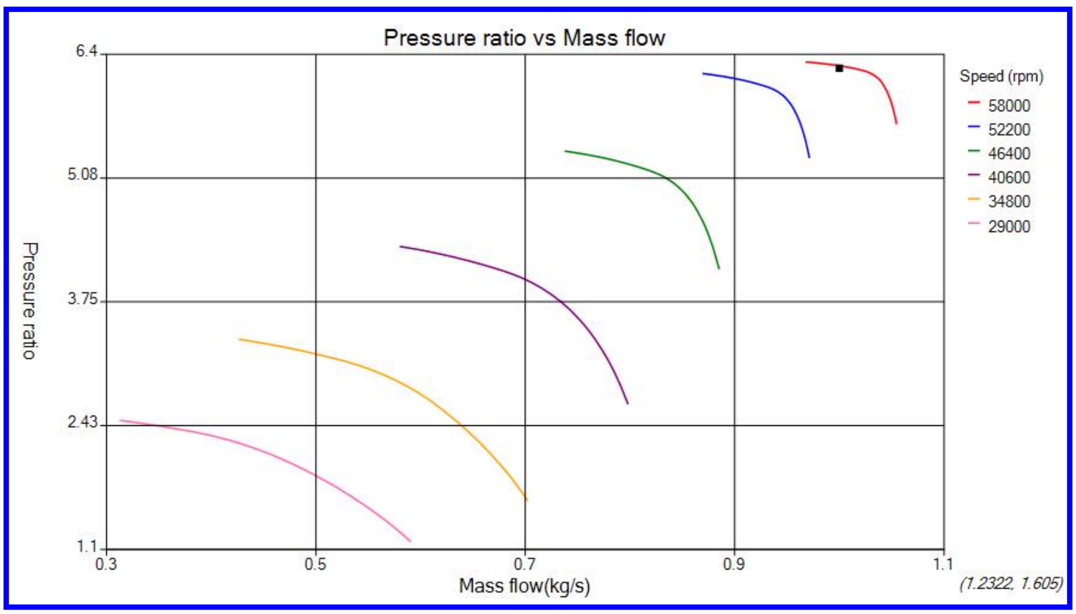
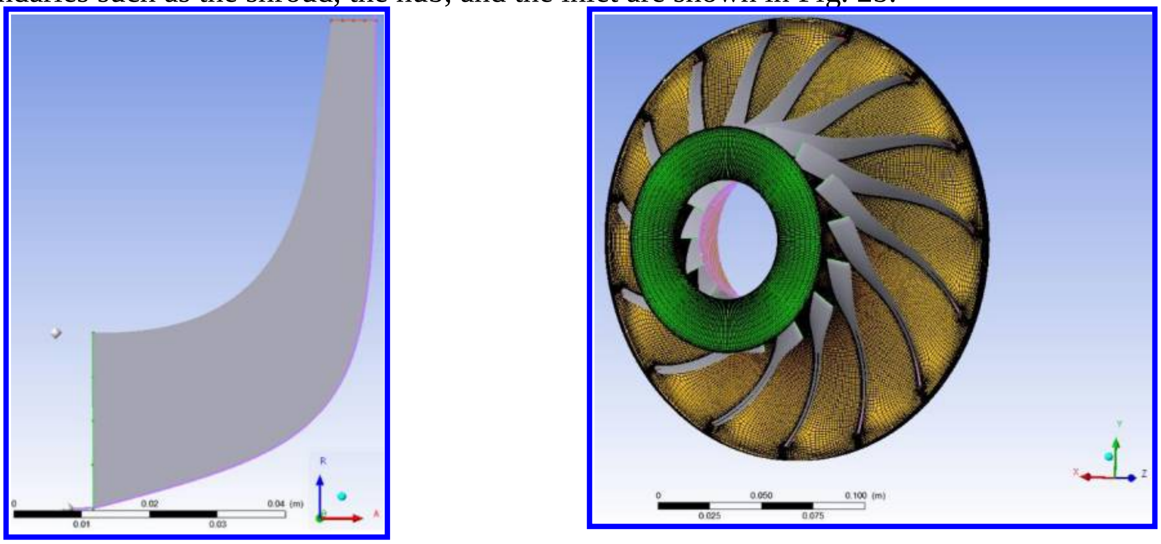


Fig. 1 Pressure ratio ~ mass flow

2.3 Meshing

Vista CCD creates an optimized 1D design, and the resulting geometry can be passed to BladeGen, which is a geometry creation tool that is specialized for turbomachinery blades. Once we get the geometry of the compressor, meshing is conducted using TurboGrid, as shown in Fig. 2. The flow passage is presented only along the mid-span of the blade (Fig. 2a). Meshes on the boundaries such as the shroud, the hub, and the inlet are shown in Fig. 2b.

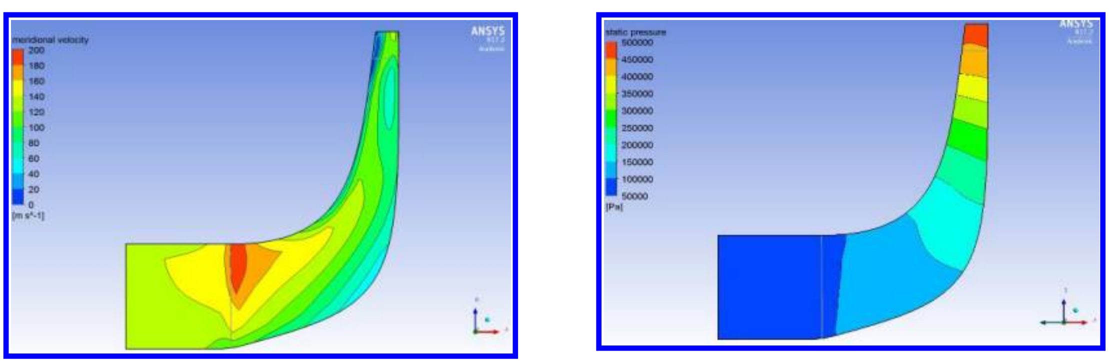


(a) (b)

Fig. 2 (a) Meshing in the flow passage, presented in the mid-span only; (b) Meshing on boundaries

2.4 CFD simulations

CFD simulations are used to understand flow phenomena and predict the off-design performance of centrifugal compressors. In this study, ANSYS CFX is used for the settings of boundary conditions, simulations, and post processing. In Fig. 3, the simulation results such as the velocity magnitude and the static pressure in the meridional surface of the impeller are shown.



(a) (b)

Fig. 3 Velocity magnitude (a) and static pressure (b) on the meridional surface

3. Air-based compressor with splitter

In this session, we use both the main blade and the splitter in the impellers. Both the number of main blades and the number of splitters are 15. Similar to the previous session, the Vista CCD is used to generate the impeller model using the input parameters listed in Table 1.

3.1 Model generation

The sizing results and output parameters are listed in Table 4 and Table 5, respectively. With the splitter, the impeller diameter at the both the inlet and outlet are smaller than those in the compressor without splitters. The width of the blade at the impeller exit is about 20% thinner than the case without splitters.

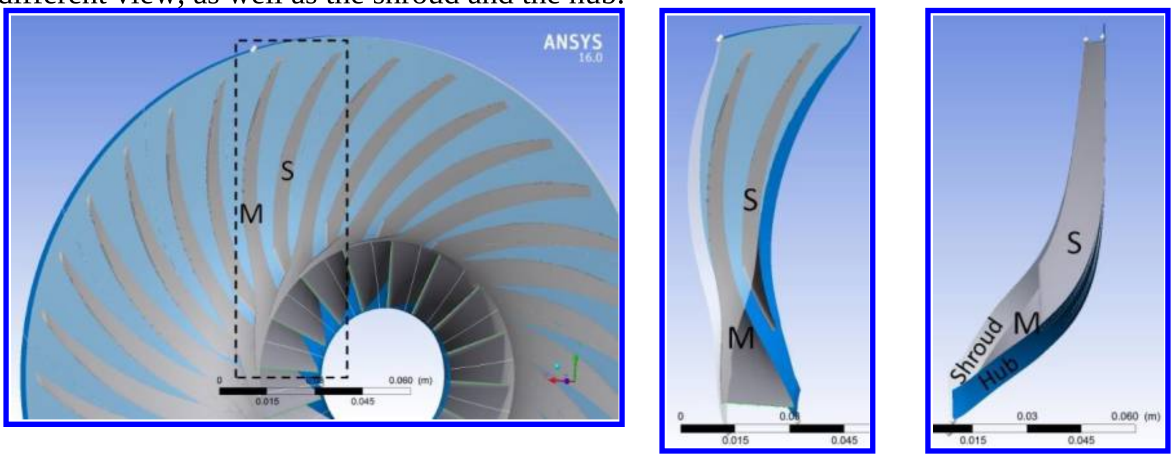
Table 4 Sizing results of air centrifugal compressors without and with splitters.

Parameters	Without splitter	With splitter
D_{h1}(cm)	6.0	6.0
D_{t1}(cm)	10.9	9.83
t_1 (cm)	0.1	0.1
β_1 (°)	55°	36.5°
D_2 (cm)	20.2	20.1
t_2 (cm)	0.2	0.2
β_2 (°)	45°	45°
b_2 (mm)	6.43	5.19

Table 5 Output parameters in Vista CCD for centrifugal compressors with splitters.

Impeller leading edge	Hub diameter (cm)	6.0
	RMS diameter (cm)	82.63
	Shroud diameter (cm)	100.28
	Inclination angle at hub	10
	RMS Inclination angle	4
	Inclination angle at shroud	0°
Impeller exit	T_0 (K)	594.19
	P_0 (kPa)	716.51
	P (kPa)	361.68
Overall performance	ϕ	0.035
	η_{poly}	0.784

The generated model of impeller with both main and splitter blades is shown in Fig. 4. (a) shows the created impeller, in which the main blade (M) and splitter (S) outlined in the dash rectangle is further presented in Fig. 2(b). Fig.2(c) shows the main blade and the splitter in a different view, as well as the shroud and the hub.

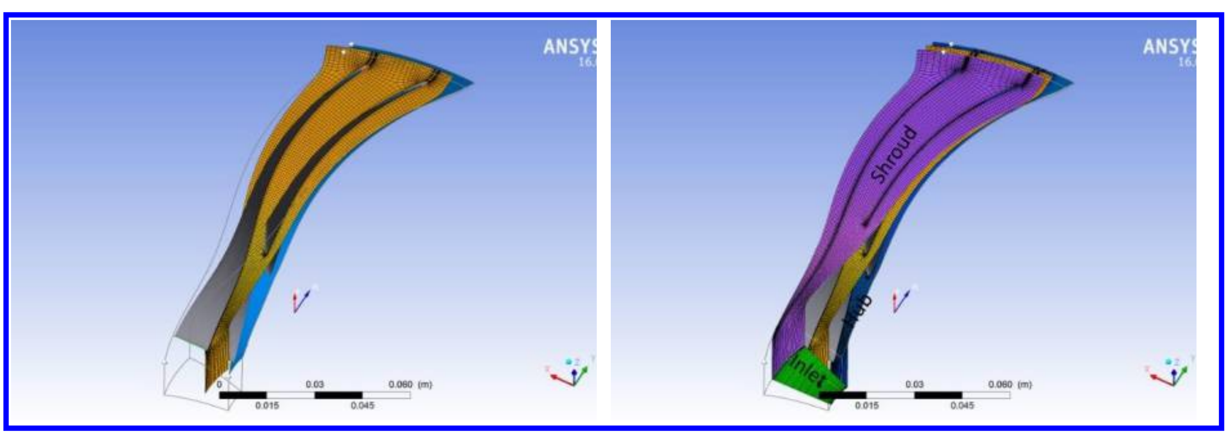


(a) (b) (c)

Fig. 4 Impeller blades generated by BladeGen. 'M' is the main blade, and 'S' is the splitter.

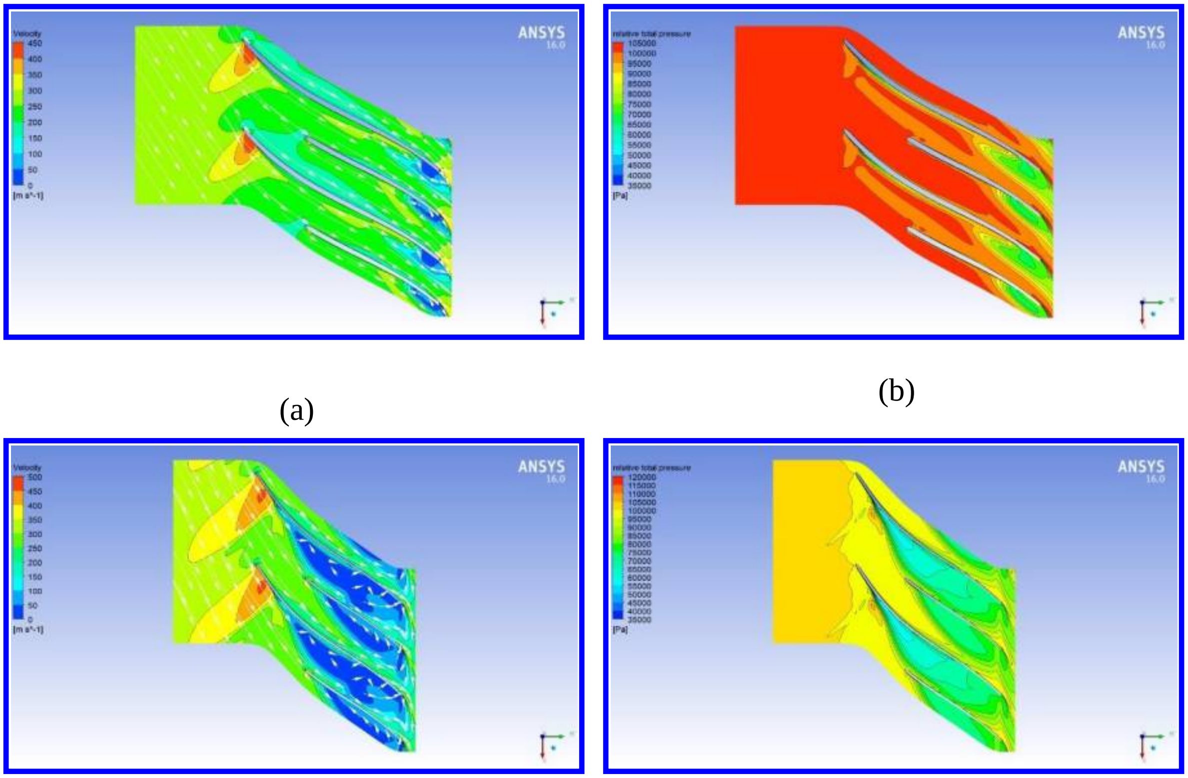
3.2 Meshing

Meshing is further conducted using TurboGrid, as shown in Fig. 5. The meshing in the flow passage is presented only along the mid-span of the blade Fig. 5a. Meshes on the boundaries such as the shroud, the hub, and the inlet are shown in Fig. 5b.



(a) (b)

Fig. 5 (a) Meshing in the flow passage, presented in the mid-span only; (b) Meshing on boundaries



(a) (b)

(c) (d)

Fig. 6 Velocity and relative total pressure on blades (a) Velocity vectors at 50% span location (b) Relative total pressure at 50% span location (c) Velocity vector at 90% span location (d) Relative total pressure at 90% span location

Fig. 6 shows the velocity vector and the relative total pressure on the impeller blades. The span locations are at 50% and 90%, respectively.

4. Refrigerant-based centrifugal compressors

In this session, we take refrigerant R134a as the working fluid medium, and the number of the main blade is 15. No splitter is used. The design parameters such as the inlet pressure, design flow rate, rotational speed, and desired pressure ratio are listed in Table 1. In Vista CCD and Ansys CFX, the thermodynamics properties of the refrigerant R134a are described by Aungier-Redlich-Kwong equation of state[13].

It can be seen from Table 6 that size of the R134a centrifugal compressor is smaller than that of the air centrifugal compressor. For example, the impeller diameter at the inlet of the R134a centrifugal compressor is 6.28cm, which is about 57.6% of the impeller inlet diameter of air centrifugal compressor. The impeller diameter at the outlet of the R134a centrifugal compressor is 10.2cm, accounting for 50.5% of the impeller outlet diameter of air-based counterpart. Also, the blade width at the impeller exit is 3.58 mm in R134a compressor, much thinner as compared to 6.43 mm in air centrifugal compressor. Table 7 presents details about the impeller inlet and outlet geometries, and the obtained over performance. It can be seen that total temperature is around 378.9K, which is much smaller than that in the air centrifugal compressor. Also, the polytropic efficiency η_{poly} in R134a compressor is 4.4% higher than that from air centrifugal compressor.

Table 6 Sizing results of R-134a centrifugal compressor, and comparison with air centrifugal compressor.

Parameters	Air	R134a
D_{h1}(cm)	6.0	3.0
D_{t1}(cm)	10.9	6.28
t_1 (cm)	0.1	0.1
β_1 (°)	55°	29.7°
D_2 (cm)	20.2	10.2
t_2 (cm)	0.2	0.2
β_2 (°)	45°	45°
b_2 (mm)	6.43	3.58

Table 7 Output parameters in Vista CCD for R134a centrifugal compressor

Impeller leading edge	Hub diameter (cm)	3.0
	RMS diameter (cm)	5.052
	Shroud diameter (cm)	64.84
	Inclination angle at hub	10.2
	RMS Inclination angle	4
	Inclination angle at shroud	0°
Impeller exit	T_0 (K)	378.88
	P_0 (kPa)	734.97
	P (kPa)	337.82
Overall performance	ϕ	0.076
	η_{poly}	0.813

5. Summary

In this study, both 1D modeling and CFD simulations are used to study the sizing and performance of air-based and refrigerant-based centrifugal compressors. Using Vista CCD, and the fundamental input parameters and geometry data, we first generate a preliminary design of the impeller. The resulting geometry is then passed to BladeGen to generate impeller blades. The obtained centrifugal compressor is then meshed, and CFD simulations are conducted using ANSYS CFX. Using the air-based centrifugal compressor without splitters as the reference, we also studied the air-based centrifugal compressor with splitters, and the R134a-based centrifugal compressor without splitters. With splitters added, the sizes of centrifugal compressor can be slightly reduced. Under the same overall pressure ratio, mass flow rate, and rotational speed, with R134a as the working fluid medium, smaller and high efficiency centrifugal compressor is obtained as compared to the air-based centrifugal compressor.

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