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Research paper

Algebraic generation of single domain computational grid for twin screw machines Part II – Validation

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ABSTRACT

Algebraic procedures are available to generate computational grid for CFD analysis of twin screw compressors. Recently new algebraic method was formulated to generate numerical grids for CFD calculation of twin screw machines with grids generated from outer casing boundaries [16,18]. In this paper, the grids of Rotor to Casing and Casing to Rotor type are tested for performance calculation of a dry air screw compressor using ANSYS CFX solver and the results have been compared with measurements. Firstly the base-line grid of the Rotor to Casing grid type was used to obtain CFD results. A grid independent solution was obtained for this base-line grid. The size of the mesh thus obtained has been used with other grid variants for comparison. A set of successively refined Casing to Rotor grid type was tested by increasing the density of nodes on the rotor profile in the interlobe leakage region. A gradual improvement in the accuracy of flow prediction was achieved with successive refinement in the interlobe region. The third variant used for comparison is a Casing to Rotor grid type with a single rotor domain that has no interface between the rotor blocks. A significant improvement in the prediction of flow and internal pressure was achieved. These developments have also extended the capability of the deforming grids to be used with other CFD solvers like STAR-CCM+ and ANSYS FLUENT. Due to fully hexahedral cell structure and improved global grid quality, addition of physical phenomena like oil injection in the models has now become achievable.

1. Introduction

The work presented in Part I of this paper was the implementation of a procedure for generation of twin screw rotor grids with the boundary distribution governed from the non-rotating outer casing to the rotating inner rotors (CR type grids). CR grid type distribution gave the advantage of a non-rotating outer boundary. Two control functions were introduced to get fast regularisation of the initial algebraic distribution. One was an analytical sine function and the other was a background blocking procedure for guiding the initial algebraic distribution on the rotor boundary. In this Part II of the paper, the CR type grids have been tested for performance calculation of a dry air screw compressor.

The application of RC type algebraic grids was previously reported in several studies [1–4]. The early references can be found in the thesis by Kovačević, [2] and the manuscript Kovačević et al. [4]. In these works, the analysis of three vital case studies was reported that form the first instance of any full 3D transient computational activity on twin screw compressors. The solution was obtained using COMET solver. One of the case studies was with a dry air screw compressor in which two different rotor profiles were compared using CFD results for their performance. The second case study was of an oil injected screw compressor where oil was treated as a passive scalar. The distribution of oil in the compression chamber and the comparison of results with measurements were presented. This case also forms the first validation of the RC type of grids. Flow delivery and indicated power was found in a good agreement with the measurements. The deviation was about 10% and 8% respectively. The third analysis was of an oil injected ammonia refrigeration screw compressor with the real gas definition of the equation of state.

More recently, Pascu et al. [8] have reported optimisation of the discharge port of a Twin screw compressor using the RC type of algebraic grids and ANSYS CFX solver. Optimisation was based on the selection of the port geometry by relative comparison of flow fields predicted by the CFD models. Sauls and Branch [11] used the same grid type to obtain results from CFD calculations with Ansys CFX to develop an improved one-dimensional thermodynamic model for refrigeration screw compressors, by extracting calibra-

Abbreviations: RC, Distribution from Rotor boundary to the Casing boundary; CR, Distribution from Casing boundary to the Rotor boundary.

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Nomenclature

ϕ_w	wrap angle [Degree]
z_1	number of lobes on main rotor [-]
z_2	number of lobes on gate rotor [-]
α	male rotor rotation angle [Degree]
η_v	volumetric efficiency [%]
P_{si}	specific indicated power [kW/m^3/min]
T_m	Main rotor torque [N m]
ρ_0	suction density [kg/m^3]
rpm, n	rotor speed [rpm]
$\dot{m}$	mass flow rate [kg/min]
P_i	indicated power [W]
$\dot{V}$	volumetric flow rate [m^3/min]
η_m	mechanical efficiency [%]
T_g	Gate rotor torque [N m]

tion coefficients that influence the pressure variation during the discharge process. Kovačević and Rane [5] reported an analysis of a dry air twin screw expander using this type of the mesh. This report was the first validation of the RC type grids in expander applications. Flow and indicated power as well as internal pressure variation curve were compared with the measurements carried out at TU Dortmund. These analysis highlighted that prediction of indicated power from CFD models was is good agreement over the range of filling pressure and rotor speeds. But at high filling pressures and low rotor speeds there was a large deviation in the predicted gas flow through the expander. This large variation was attributed to changes in clearances during the operation [12]. In another recent study by Kovačević et al. [6], the RC type of grids were used to solve flow in dry air compressor using two different flow solvers. One solver was a coupled vertex-centre based solver ANSYS CFX and the other solver was segregated cell-centre based formulation from Simerics Pumplinx. Both were pressure based solvers. The performance indicators of pressure variation in the compressor chamber, mass flow rate, indicated power and the volumetric efficiency were used for comparison. Analysis showed a big influence of the solver and mesh treatment on the flow predictions of the two solvers. Rane et al. [9,10] extended the RC type grid generation to variable geometry rotors such as variable lead and variable profiles and reported cases on comparison of the performance of these different rotor geometries.

On the application of CR type of grid structure, Vande Voorde et al. [13] and Vande Voorde [14] were the first to implement a grid generation algorithm for block structured mesh from the solution of the Laplace equation for twin screw compressors and pumps using differential methods and then construct a CR type of rotor grid to be used for final calculations. In his thesis, Vande Voorde et al. [13] and Vande Voorde [14] presented the solution of flow in a double tooth compressor and an oil free twin screw compressor. The results were compared with experimental data over a range of discharge pressures and rotor speeds. A larger deviation in the volumetric efficiency and specific work was reported at lower rotor speeds. More recently, Papes et al. [7] presented a 3D CFD analysis of an oil injected twin screw expander using the same differential CR type grid generation approach. A real gas model for the working fluid R245fa was used and results for the different leakage flows were presented together with chamber pressure variation curves and performance analysis at different pressure ratios.

The test case presented here uses the CR type grid structure with algebraic implementation and background blocking regularisation approach described in Part I of this paper [16,18]. The study has two objectives.

- First is to verify that the new CR type algebraic grid structure implemented in Part I [18] produces cells that are valid and within quality requirements of the flow solver.
- The second objective is to compare the model results under similar set of operating conditions with measured performance and understand the influence of interlobe grid refinement and non-conformal rotor to rotor interface on the accuracy of performance predictions.

It is expected that the new CR type grids will allow for better accuracy in predictions from the flow solver because of the fully hexahedral cell structure. It is also expected that the elimination of the non-conformal interface will improve robustness of the solver.

2. Measurements on the test screw compressor

Fig. 1a shows the 3D CAD model of the test compressor and Fig. 1b shows its rotors. The rotors are designed with 'N' profile with 3 lobes on the main rotor and 5 lobes on the gate rotor. The rotor centre distance is 93.00 mm and the main rotor length to outer diameter ratio is 1.6. The compressor has an axial suction and a combined axial and radial discharge. The built in volume index is set to be 1.8 and the wrap angle on the main rotor is 285°. The design clearances of the machine are 120 to 160 micro meters in all, the rotor interlobe gaps, radial rotor to casing clearance gaps and the high pressure end clearance gap.

This oil free compressor is used for dry air compression in the range of operating speeds from 6000 to 12,000 rpm. The suction is at atmospheric conditions and the maximum discharge pressure for safe temperature limits is about 3.0 bar The compressor runs under synchronised operation and both the rotors are driven by timed two stage helical gear box integral with the compressor housing.

Test measurements of the compressor performance were carried out in the air compressor test rig at City University London. Fig. 2 shows the layout of the measurement rig its main components and measurement points. The compressor is driven by a variable speed 75 kW motor and has an internal synchronising gear box with the gear ratio 7.197:1. The speed of the motor is adjusted using a variable frequency drive. The torque meter is installed on the motor shaft while the digital encoder for the speed measurement is mounted on the male rotor shaft.

The discharge pressure is controlled via a check valve (I) in the discharge line. Gas flow rate is measured using an orifice meter (H) in the discharge line. Since the discharge gas temperature is high in the absence of oil cooling, a cross flow water heat exchanger (G) is installed to cool the gas before it enters the main discharge line in the rig. Fig. 3a shows the compressor in the test rig. Speed encoder on the main rotor shaft, suction and discharge pressure transducer and discharge temperature thermocouple are visible.

The pressure and temperature of the gas are measured at the inlet, discharge and upstream of the orifice plate. In addition, three pressure transducers are used for recording the interlobe pressures and are located in the working chamber through the compressor casing on the male rotor side. Fig. 3b shows the location of the dynamic interlobe pressure transducers in the compressor housing. Measurement data acquisition is carried out using CompactRIO (CRIO-9022) system from National Instruments and Labview software. The measurements were taken for discharge pressures up to 2.5 bar and speeds from 6000 to 8000 rpm. Table 1 presents the instrumentation and transducer details used in the test rig. BS ISO 1217:2009 has been followed in setting the rig and measurement uncertainties have been estimated for mechanical power and flow rate measurements (Bianchi and Cipolloni [15,17]) to be used for comparison of CFD results.