SOFTWARE IN THE CLASSROOM

DEMONSTRATING A STEPWISE APPROACH TO TURBOMACHINERY DESIGN

H. Perez-Blanco, S. Fishbone, L. Moroz.

Using software tools in turbomachinery design courses in universities helps to cover more ground, particularly in solid modeling with instant thermofluid feedback during the course. Students, however, should observe caution in interpreting the results. Care and dedication are required. The below article describes how undergraduate students learned the design of centrifugal compressors using AxStream — a commercial software. When surveyed about their learning experience, the class of 45 students responded that the software allowed to visualize and make generalizations that lectures alone would not have enabled them to do.

The need for tools

The aero thermal design of centrifugal compressors can be complicated, particularly if Mach numbers (relative or absolute) are projected to exceed one at some points in the machine. In the classroom, some design aspects can be gleaned using slip factors and 1-D compressible flow relationships. These procedures allow definition of vane and diffuser angles, and enable sketching an approximate geometric configuration. However, estimates of impeller and diffuser efficiencies are required to ascertain the discharge stagnation pressures. Design optimization is then beyond the reach of this methodology.

Recognizing these difficulties, SoftInWay Inc., proprietor of AxStream, donated the software for the Turbomachinery course at the Mechanical and Nuclear Engineering Department of Penn State. Using dedicated software reduces the guesswork of selecting isentropic efficiencies and generates a viable design in a fraction of the time required by hand calculations.

However, undergraduate engineering students in their junior or senior years need to be introduced gradually to the software, as to allow the instructors to sufficiently emphasize the design steps, loss mechanisms and interpretation of results. The AxStream software offers a series of design tools of increasing complexity.

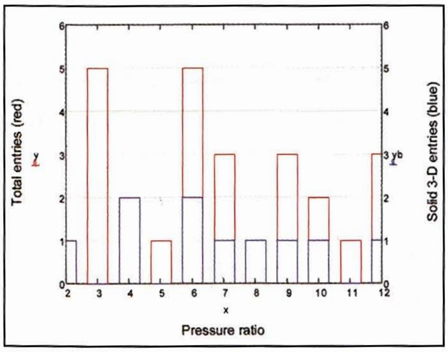
The user starts with design conditions and important variable ranges, such as rotational speeds and diameters that result in a thermofluid meanline analysis optimized for efficiency. In the second level, the user determines the design viability via a streamline curvature analysis, and proceeds to design the impeller vanes and the diffusers. In the third level, the user appraises the design by inspecting a 3-D image of the impeller and important blade and vane angles. Stress calculations, inviscid and viscous flow calculations and Campbell diagrams are a fourth level, but were not taught as part of the course.

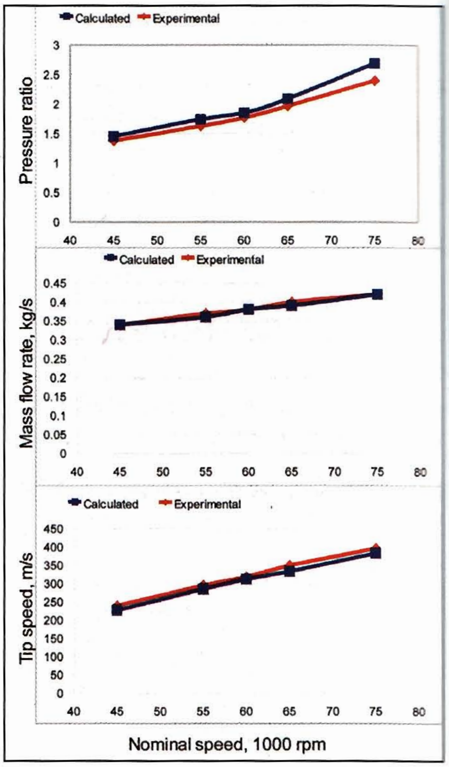
Design targets:	Mass flow rate: 5 kg/s Pressure ratio (total to total): 2 Inlet conditions: air, pressure 101 kPa, temperature 288 K									
Student	Rotor Diameter (m)	Tip velocity (m/s)	Vane exit angle (deg)	Rotational speed (rpm)	Power input (kW)	Efficiency	No. of vanes	No. of diffusers	Coefficients	
									Flow	Load
1	0.48	325	48.4	12968	370	0.92	9	23	1.24	0.7
2	0.48	357	61.9	14343	360	0.92	10	16		0.72
3	0.39	321	53.2	15726	358	0.93	8	16	0.56	0.68
4	0.21	327	62.5	29740			16	29		
5	0.28	416	37.6	28382	388	0.88	11	31		
6	0.56	326	42.8	11140	379	0.89	14	20	0.51	0.2

Table: Results of thermofluid design of centrifugal compressors

Figure 1 (below): Frequency distribution of pressure ratios

Figure 2: Compressor performance comparison





Semester activities

The students learned the software gradually via brief classroom demonstrations and increasingly complex homework. Lectures in parallel with the homework illustrated the basics of centrifugal compression, vaned and vaneless diffusion and loss mechanisms. When sufficient familiarity had been achieved, a homework required the students to specify their own optimization ranges to attain a given pressure ratio on a given flow of known inlet conditions.

Selected results from participating students have been summarized in the Table. As the students specified their ranges for the optimization search individually, some variety can be detected in the results. The maximum tip speed was 416 m/s and the minimum was 321 m/s. The power inputs were rather uniform, as were the efficiencies, clustered at around 90%.

Compared to the efficiencies adopted for hand calculation in the course, typically extracted for impellers and diffusers from existing literature data, the efficiencies computed by AxStream appeared to be high. Some students specified the number of blades and vanes; others adopted the program values. Not all the outputs contained enough information to ascertain the load and flow coefficients, and in one case the power input and efficiency were not available.

In a second activity toward the semester end, the students were given a challenge, consisting of producing the largest possible discharge pressure for the following inlet conditions: Initial pressure of 100 kPa, initial temperature of 285 K, and air mass flow rate of 50 kg/s.

In this problem, the students were asked to complete all the levels, up to producing a 3-D image of the impeller. Such a complete design needs further validation in terms of stresses and CFD computations, but those were not required for the project.

Simply stated, the desired outcome was to produce a compressor design that AxStream can draw in 3-D. The students were asked to specify their own range of values for optimization. We present the results of this effort as a frequency distribution plot (Figure 1), with the total to total pressure ratio in the horizontal axis, and the number of students attaining such ratio along the ordinate axis. Two series are presented: The one in red shows the total entries attaining a pressure ratio around the abscissa value. The one in blue shows the number of entries culminating on a 3-D solid illustration by the program.

Students optimized the pressure ratio by varying the ranges of dimensions, speed, flow coefficients and impeller exit angle before optimization. As the project proceeded, the preliminary design underwent a streamline curvature check. Later the user had to draw the vanes and diffusers in the program environment.

It was observed that errors in the optimization or in the checks for streamline curvature or blade loading resulted in AxStream producing a defective 3-D view or none at all. The results obtained by all users are given by the red series.

The blue series fulfilled two qualifications: The 3-D view of the impeller revealed a complete specification of blades and vanes, and impeller exit angles close to a common value at the tip. Clearly, the number of entries is considerably reduced when subject to those qualifications. Yet, the two top entries reached pressure ratios of 10 and 12, but the results were not always valid.

It was made clear to students that high pressure ratio entries would result in designs that cannot be realized in practice. So the class limited iteself to meanline analysis.

In addition, the loading profiles for high-pressure ratios had noticeable spikes; namely the first derivative of the pressure loading curves exhibited many large discontinuities. Such is not the case for some of the low-pressure ratio (2-4) entries: Their pressure loading is continuous with a continuous first derivative, anticipating a design with good operational prospects.

Experimental data

A third activity consisted of comparing the meanline analysis from AxStream to experimental data acquired from the SR-30, a small gas turbine with a centrifugal compressor. The particular geometry used in the compressor is hard to model in AxStream, so the results must be regarded as approximate. Nevertheless, experimental data can offer useful comparisons.

Figure 2 shows a comparison between measured and calculated values. The pressure ratio increases with rotational speed, and the trends are similar for the SR-30 compressor and the calculated ones. However, the calculated values exceed the measured ones over the whole range of speed. The maximum percentage error (using the calculated value as reference) is 11%, but the most frequent discrepancy is within 5 %. The mass flow rate trends are also quite similar in both the actual and simulated unit, but the accuracy of the prediction is much improved over that for the pressure ratio. The maximum deviation amounts to 3% of the calculated value.

The last plot in Figure 2 shows the tip speed vs. rotational speed. Whereas the trends ought to be the same for solid body rotation, it must be borne in mind that the software picks both the diameter and the rotational speed. The consistency between the tip speeds and the pressure ratios is an indication that the meanline analysis reflects experimental trends well. Regarding isentropic efficiencies, the calculated values are much higher than the measured ones. The discrepancies are due to differences in diffuser geometry, and to the experimental uncertainties associated with the measurement of the discharge temperature..

Author

Horacio Perez-Blanco is a professor of thermal systems at Penn State, Scott Fishbone is a project engineer with Conoco-Phillips, and Leonid Moroz leads SoftInWay.