



EXECUTIVE SUMMARY

**FINAL REPORT ARTI-21CR/605-10010-01
THE EFFICIENCY LIMITS OF WATER VAPOR COMPRESSORS**

A scoping study has been completed by Concepts ETI, Inc. (CETI) for the Air Conditioning and Refrigeration Technology Institute (ARTI) as part of their 21-CR project, in order to determine which compressor configuration might be “best” suited for water vapor compressor applications. The primary focus of this work was to determine the attainable efficiency and power levels, as well as approximate geometries of single and multistage axial and centrifugal compressors for the selected refrigeration cycle.

The design point used for this study was an inlet pressure (P_{00}) of 0.13163 psia, an inlet total temperature (T_{00}) of 501.67°R (saturated vapor conditions), an inlet mass flow (m) of 3.116 lbm/sec, and a total-to-static pressure ratio (pr_{ts}) of 6.60, with the fluid being water vapor. The design point mass flow results in 1,000 tons of refrigeration at the 502°R evaporator discharge temperature.

The first part of this project consisted of a literature and technology survey on water vapor compression. Relevant books on compressor technology in general, and published literature on water as a refrigerant, ice slurry generation, and currently used compressor types for water vapor compression were reviewed. Lists of organizations working in this field, along with relevant upcoming conferences and Internet sites, were included along with a reference list of over 150 publications.

A scoping study was completed for both single- and two-stage centrifugal compressors. A set of reasonable design and modeling parameters was selected, and the design point rotational speed was varied to

investigate a specific speed range from 60 to 130 (US units= $\frac{rpm \sqrt{ft^3/sec}}{\left(\frac{ft * lbf}{lbm}\right)^{3/4}}$).

For the single-stage compressor, the geometric configuration consisted of a three-dimensional impeller, channel diffuser, and volute. Two impeller backsweep angles, 0° and -30° were considered. As can be seen from the table for the highest efficiency designs for both backsweep angles, the -30° backswept impeller has about three points higher efficiency with a 4% reduction in required power.

TABLE E1

BEST EFFICIENCY SINGLE STAGE CENTRIFUGAL COMPRESSORS		
	0° BACKSWEEP	-30° BACKSWEEP
Rotational Speed (N, rpm)	4805	5491
Specific Speed (NSS)	70	80
Stage Total-to-Static Efficiency	74.7%	77.6%
Power (hp)	823.4	792.4
Impeller OD (in)	104.8	98.2
Tip Speed (U2, ft/sec)	2197.5	2351.7
Machine OD* (in)	239.3	231.1

*Diameter to outside of compressor volute

Both of these designs are unattractive for a number of reasons, the primary ones being large power requirements (>670 hp), and extremely high impeller tip speeds ($U_2 > 2200$ ft/sec). The impeller tip speeds for these designs will lead to bore stresses in excess of yield for most common impeller materials, including titanium. There are a number of design options available to reach these tip speeds, including flexible blades mounted on extremely thin impeller hubs, or fabricating the impeller out of a composite material, but these manufacturing methods may involve substantial development time and cost.

For the two-stage compressor, three different approaches to determining the pressure ratio split between the two stages were evaluated. These were equal pressure ratio split, equal specific speed split, and equal work (ΔH) split. As with the single-stage compressor, the geometric configuration consisted of a 3D impeller and channel diffuser. The exit element for the 1st stage compressor was a return channel that connects the two stages, while the exit element for the 2nd stage compressor was a volute. Depending on the final intercooler arrangement selected by ARTI, the 1st stage exit element for a two-stage centrifugal compressor may be a volute, rather than a return channel. Since the total pressure loss in a typical volute is slightly less than that of a typical return channel, the results presented in this report for two-stage compressors may be slightly pessimistic. Three different intercooler configurations were considered, no intercooler (NIC), an intercooler which brought the fluid temperature to saturation without mass addition (IC), and a flash intercooler (FC) which brought the fluid temperature back to saturation using water injection (mass addition between Stages 1 and 2).

In all of the two-stage cases, the peak efficiency levels for the -30° backsweep designs exceeded those of the 0° backsweep designs. The following results are for the -30° backsweep, two-stage designs:

TABLE E2

BEST EFFICIENCY TWO STAGE CENTRIFUGAL COMPRESSOR DESIGNS									
	Equal PR Split			Equal NSS Split			Equal Work (ΔH) Split		
	NIC	IC	FC	NIC	IC	FC	NIC	IC	FC
Rotational Speed	3719	3377	3377	3552	2952	3881	3751	3826	3522
Overall Efficiency	79.3%	78.7%	78.8%	78.6%	78.9%	78.8%	79.5%	79.0%	78.9%
Power (hp)	791.5	701.4	745.7	799.7	711.7	729.1	789.5	699.2	721.9
Stage 1 Impeller OD (in)	96.5	105.6	105.6	120.7	107.8	108.4	102.2	95.7	104.2
Stage 2 Impeller OD (in)	106.6	104.4	104.7	83.7	83.7	74.8	99.4	91.9	98.1
Stage 1 U2 (ft/sec)	1567	1556	1556	1870	1859	1836	1673	1598	1602
Stage 2 U2 (ft/sec)	1730	1539	1542	1297	1237	1267	1627	1534	1508
Machine OD* (in)	248	231	231	237	222	223	223	221	228

*Diameter to outside of compressor volute

For all of the above two-stage designs, the impeller tip speeds range from a low of 1,237 ft/sec to a high of 1,870 ft/sec. For the lowest power design (equal ΔH , intercooled), the tip speeds range from 1,534 ft/sec - 1,597 ft/sec, which puts this design safely in the range of high strength aluminum. Alternately, the backsweep angle for this design can be further increased (up to the material strength limit) to lower the power required even further, though the impeller ODs are already quite large and will be quite costly to fabricate.

Low solidity airfoil (LSA) diffusers were considered as a substitute to the channel diffusers, but the total pressure losses were almost always higher than those of the channel diffuser and the pressure recovery was usually lower. The LSA diffuser should not be completely abandoned, however, because LSAs can offer greater flow range than the channel diffuser and may be better suited to the overall refrigeration cycle, even if they do come with a slight design point efficiency penalty. They also require less radial space than a channel diffuser, and could lower the machine OD by 20% to 30%, which may be important for very compact applications.

The effect of interstage pressure drop (2% to 6%) on performance was considered. The impeller and machine size were affected very little, as was the overall efficiency. The primary effect was an increase in power. For the minimum power design, the power increased about 4.96 hp for each percent pressure drop increase.

Finally, the effect of Stage 1 inlet temperature on performance (for a fixed design) was analyzed. Again, the overall efficiency changed by an insignificant amount. However, for a 15° increase in temperature, the power increased by about 2.5%.

In summary, none of the one-stage or two-stage centrifugal compressor designs investigated thus far meet the maximum power required goal of 670 hp. The lowest power design is still 4% above this goal. The one-stage designs will probably be eliminated because of extremely high impeller tip speeds and large power requirements. The most attractive centrifugal compressor designs are the two-stage designs with some form of intercooling and either equal work or equal specific speed stages. These designs have manageable impeller tip speeds and power requirements between 4% and 9% above the design goal. With increased impeller backsweep and further design refinement, it is likely that the power goal of 670 hp can be met, though the resultant machines will be quite large and perhaps expensive to manufacture.

A scoping study was also carried out for a multistage axial compressor configuration. The rotational speeds covered varied from 5,000 rpm to 6,900 rpm, with the number of stages limited to six or seven. All calculations were done without intercooling.

The speeds below 6,000 rpm were quickly eliminated. The speeds were too low to generate the necessary stage pressure ratio with only six or seven stages, with reasonable levels of blade/vane camber angles. Hub loading was excessively high in the first stage, as well. At 6,900 rpm and above, a shock loss is added to the already high profile losses and secondary losses due to the low Reynolds number operation. This makes these high speed cases unattractive, also.

From the 6,000 rpm to 6,600 rpm cases run, three feasible compressor configurations were identified. They are:

- Configuration 1: 7 stages, 6,000 rpm
- Configuration 2: 6 stages, 6,600 rpm
- Configuration 3: 7 stages, 6,600 rpm

TABLE E3

MOST PROMISING AXIAL COMPRESSOR CASES (with $C_p = 0.60$ Diffuser)			
	Configuration 1	Configuration 2	Configuration 3
Total-to-Static Pressure Ratio	6.88	6.66	6.70
Total-to-Static Efficiency	81.7%	82.3%	82.6%
Power (hp) (rematched to $pr=6.6$)	729.0	724.3	721.1
Tip Speed (ft/sec)	1361	1440	1440
Axial Length (in)	127.0	116.5	127.2
Maximum OD* (in)	53	51	51

*Diameter to outside of largest rotor

The most promising of these designs is Configuration 3. The increase in performance over the other configurations is small. However, the overall design is much better balanced. The rotor and stator diffusion factors and the blade/vane solidity distributions are improved over the other two configurations. The overall 82.6% efficiency is about three points above that achieved by the best two-stage centrifugal design, and the power (when compared to the no-intercooling centrifugal cases) 10% lower. If the use of intercooling allows the axial compressor to achieve the same 8.5% to 11% decrease in required power as for the centrifugal compressor, the power requirement would be within the 670 hp power goal.

When investigating the effect of inlet temperature on performance, an increase in inlet temperature of 465.5°R caused Configuration 3 to choke, so that an increase of 15° could not be achieved. However, increasing the rotational speed from 6,600 rpm to 6,720 rpm alleviated the choking and allowed the flow rate goal to be reached. While the efficiency was reduced by only 0.2 points, the required design power increased by 3.3%.

The advantages of the two-stage centrifugal machine is a lower number of parts, shorter axial length, and possibly lower manufacturing costs. The disadvantages are lower efficiency, higher power, and a larger overall diameter. The centrifugal compressor may require a more complex bearing design and rotor dynamic analysis, depending upon the selected bearing arrangements. The advantages of an axial machine are higher efficiency, lower tip speeds and smaller overall diameter. The key technical challenges of the axial machine may be minimizing the inlet losses to the first stage due to the high inlet relative Mach number, and controlling the peak blade stress in the first stage and disk stresses in the back stages. In addition, variable inlet guide vanes or an interstage bleed/jet flap arrangement may be needed for start-up and part power operation, possibly with variable geometry stators, adding additional parts and complexity.

The preferred configuration, to carry forward for detailed design in the later phases of this project, will depend not only upon performance, but also on ARTI weighted evaluation criteria such as manufacturing cost, development cost/risk, and overall footprint preferences. These criteria can be studied in more detail in the later phases of the project as the specific application of each machine is determined. In fact, a centrifugal machine may be more appropriate in some applications, while an axial machine will be better suited to others. This type of decision will likely be applied on a case-by-case basis.