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Topical Report

Analysis of Radial Compressor Options for Supercritical CO_2 Power Conversion Cycles

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Abstract

Radial compressor options and performance attributes for a 300 MWe Supercritical CO_2 Power Conversion System are reviewed in some detail. The principal focus is on the main compressor, which is unconventional in that it operates near the critical point of CO_2. A one stage version is recommended, and its projected full power 3600 rpm, plus off-normal performance, documented in the form required by the modified GAS-PASS system transient and dynamics code. For the recompressing compressor a three stage machine is proposed.

Acknowledgements

The work reported here is a deliverable under contract to Sandia as part of the projects "Qualification of the Supercritical Power Conversion System for Advanced Reactor Applications" and its extension "Supercritical CO_2 Cycle Control System Simulation and Plant Design Studies."

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Chapter 1 Introduction

1.1 Foreword

This is a required deliverable under our contract with Sandia for "Qualification of the Supercritical CO_2 Power Conversion Cycle for Advanced Reactor Applications" as called for in the amended statement of work in Sandia PR #806934 dated 3/17/2006. Its principal topic is the analysis of radial compressor options for our reference 300 MWe power conversion system (PCS).

1.2 Background

Over the past few years the MIT group has been engaged in designing and evaluating the supercritical CO_2 PCS for Gen-IV applications. A principal motivation is its ability to attain good thermodynamic efficiency (≈45%) for moderate turbine inlet temperatures (≈550°C), and its extremely compact turbomachinery. These features are summarized in Refs. [1.1], [1.2] and [1.3].

Work has evolved from overall scoping studies to progressively more detailed turbomachinery design, as documented in Refs. [1.4] through [1.8]. These analyses have focused on the use of axial turbomachinery because of its high efficiency and ability to handle high volumetric and mass flow rates.

At the same time it was recognized (and supported by the Barber Nichols industrial review team at the meeting held by DOE/Sandia with MIT/ANL on Aug. 31, 2005) that the option of using radial turbomachinery should also be evaluated, in large part because of its wider operating range and more rugged nature. This report summarizes the subsequent MIT effort along these lines.

1.3 Organization of This Report

In Chapter 2, a brief overview of the generic aspects of substituting radial for axial turbomachinery is presented.

Chapter 3 contains the heart of the matter, presenting designs for radial versions of both the main and recompressing compressors.

Chapter 4 covers off-normal performance map development, needed for future dynamic and control studies.

Chapter 5 briefly discusses a radial version of the turbine, and the reasons why it is recommended that this option be dropped.

Chapter 6 discusses hybrid radial-axial machines, which represent a compromise between pure radial and pure axial devices.

Finally, Chapter 7 summarizes our findings, including recommendations for the next steps forward.

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Chapter 2 General Considerations Relative to the Use of Radial Compressor

2.1 Introduction

Table 2.1 contrasts some generic attributes of radial and axial compressors, as abstracted from Ref. [2-1] through [2-10]. While axial machines would appear to have an overall advantage under steady-state full power conditions, it is primarily our concern with off-normal behavior which justifies serious consideration of radial compressors. The degree of desirability, or indeed necessity, for making radial compressors the reference design will not be evident until system dynamics calculations have been completed and analyzed, currently scheduled by December 2006.

In addition to the conventionally-derived assessment attributes, which are based on near-ideal-gas conditions, it must be stressed that the main compressor operates near the critical point of CO_2, and is in many ways more of a pump than a compressor. This issue will be discussed in more detail in this chapter, but it must be conceded that this regime has not been thoroughly explored by turbomachinery designers, and laboratory tests of such devices must be given a high priority.

Table 2.1 Generalizations Regarding Compressor Attributes

Radial (Centrifugal)	Axial
Approximates constant head, variable flow behavior (see Fig. 2.1)	Approximates constant flow, variable head
Best suited for lower flow rate, higher pressure ratio applications	Best suited for higher flow rate, moderate pressure ratio applications
Fewer stages, shorter	More stages, longer
Larger diameter; but can scale down to lower power (vanes and volute increase diameter—see Fig. 2.2)	Smaller diameter but large flow area; diffuser adds length
Lower efficiency by up to ≈4% in general (hence ≈ 1% lower S-CO_2 PCS overall efficiency)	Higher efficiency, especially if shrouded
Easier to maintain critical clearance	Radial clearances strongly affect efficiency; shrouding beneficial
Wider operating range between stall and surge (stall not important)	Stall and surge are important limits
Analytical/numerical modeling not as advanced; more art and empiricism involved in design	Strong (ideal gas) modeling background: e.g. via jet engine design
More rugged; can tolerate some condensation	Need to avoid droplets

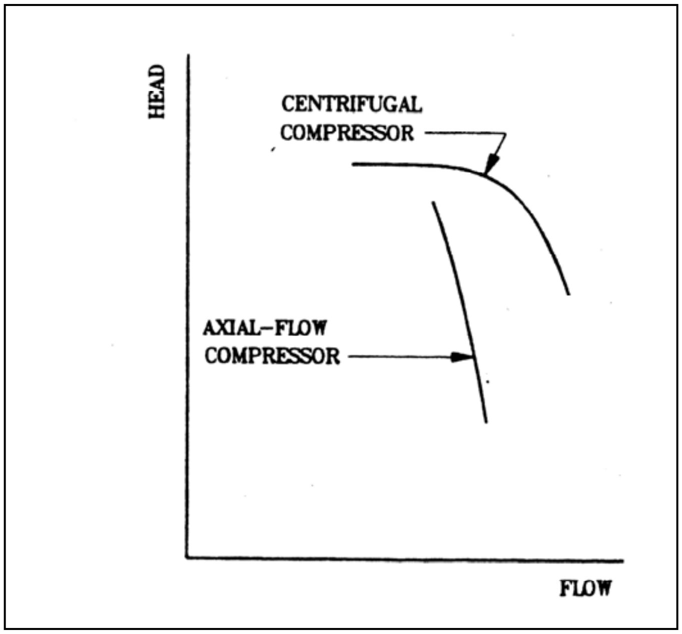

Fig. 2.1 Comparison of Compressor Types [from Ref. 2.9]

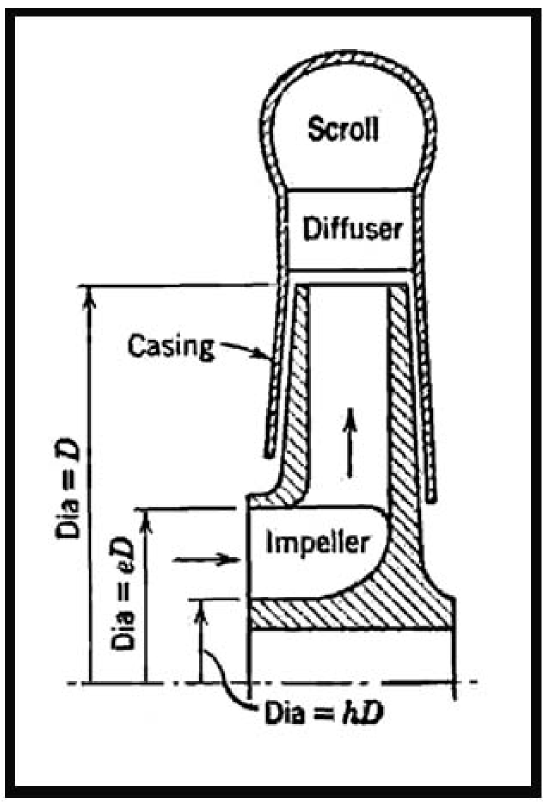

Fig. 2.2 The Effect of Adding Diffuser Vanes and a Volute on Compressor Diameter [from Ref. 2.2]

As noted in Table 2.1, one key issue standing in the way of adopting radial compressors for the reference S-CO_2 PCS design is their efficiency, which is in general inferior to that of axial compressors for applications involving high volumetric flow rates. Wilson [Ref. 2.10] gives several reasons for their inherently higher losses:

- The sharp turn in flow direction at the inlet
- Vulnerability to secondary flows in their long passages
- High outlet velocity and only moderate diffuser efficiency
- Larger wetted area than axial machines

Ref. [2.16] surveys commercial units employed in the 1960's, and reports an average efficiency of only on the order of 77%. These were, however, for non-power-cycle applications, where the incentives for ultra-high efficiency are not as paramount. The state of the art continues to improve, however (see Fig. 2.3 from Ref. 2.17), and for present purposes we have assigned a net differential penalty of 4%: i.e. 89% efficiency for axial compressors vs. 85% efficiency for radial compressors. This subject deserves priority attention in the future design and test program.

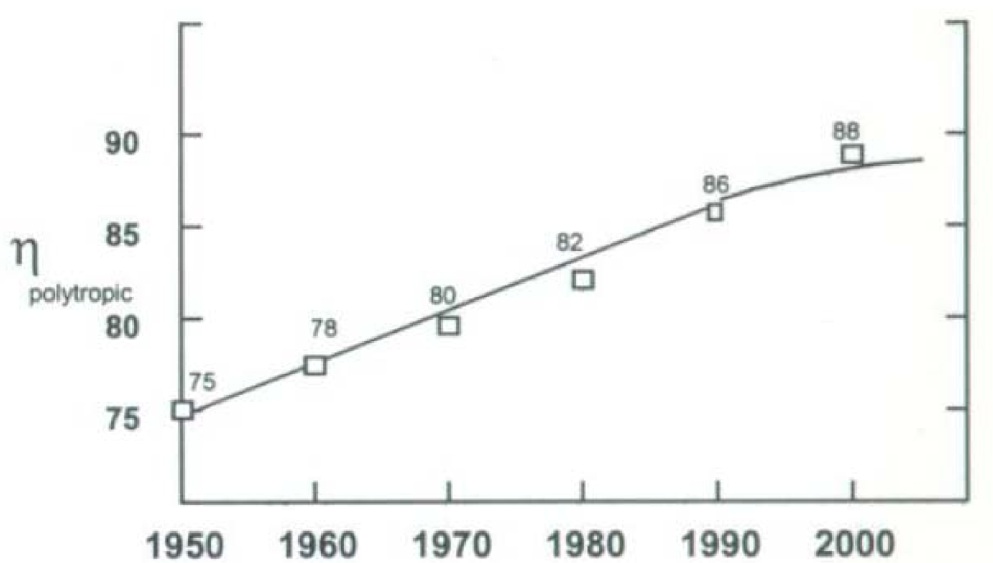

Fig. 2.3 Improvements in Radial Compressor Efficiency with Time

2.2 Some Major Design Options

Centrifugal compressors require diffusers to attain more than mediocre efficiency. The diffusers can be with or without vanes. Vaneless diffusers are in general preferred for smaller, low-throughput devices, while the larger, high mass and volume flow machines of current interest usually have diffuser vanes. They generally have a wider operating range, not being choke prone. Vaned diffuser machines are more efficient and have a smaller overall diameter.

A second option is whether to employ a shrouded impeller. Rotating shrouds add mass-hence can increase stress significantly in machines having high rotational speeds. Since our application has a fairly low pressure ratio (thus a low optimum speed) and is on the same shaft as the fixed speed (3600 rpm) generator, this is probably not a serious disadvantage. Moreover, the aerodynamic loading is comparable to, or higher than, centrifugal loading for our cases. The shroud also increases vane stiffness. Shrouds are also more common in radial pumps—hence may be beneficial for the main compressor. Our rotational speed, 3600 rpm, is well within the range of conventional usage (2500 ≈ 11,000). Since capacity is directly proportional to rpm one can in general reduce diameter by increasing speed—an option not open to us because we are synchronized to the grid unless an inverter is interposed.

Our application is within, but at the upper range of, industrial experience with respect to centrifugal compressor volumetric throughput [Ref. 2.16]. One infrequently employed accommodation is to use tandem (i.e. back-to-back) impellers to double the total flow rate accepted. This configuration has the added advantage of canceling out net axial thrust, making thrust bearing design easier.

An even more fundamental design choice is the number of stages. While the pressure ratio of 2.6 required of the S-CO_2 cycle's main and recompressing compressors is easily achieved in a single stage, using two reduces the machine's radius and/or its optimum rotational speed. As shown in Fig. 2.4 (from Ref 2.18), two or more stages would represent typical industrial practice for our situation (again for a near-ideal gas). These considerations are dealt with in considerably more detail in Chapter 3.

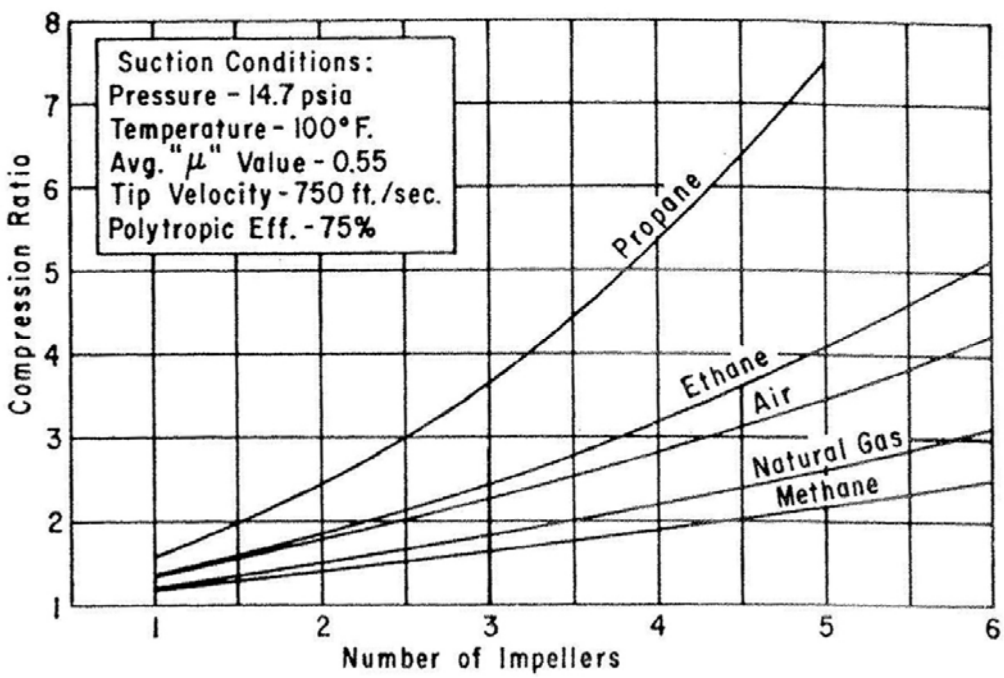

Fig. 2.4 Compression Ratio vs. Number of Impellers. Uncooled Compression (from Ref. 2.18)

There are many other details, such as the use and design of an inlet inducer, which must be considered in radial compressor design. In this respect, it is worth repeating the point made in most of the texts cited in section 2.5, namely that most refinements are closely held, unpublished, trade secrets, and hence are best left to the industrial sector.

2.3 Pump-Like Nature of Main Compressor

Most Brayton cycle turbomachinery applications deal with working fluids which are near-ideal gases. Thus our general approach has been to incorporate real gas properties into the design codes. This has proven to be a difficult endeavor for near-critical-point applications. For the main compressor another approach can be contemplated: treating it as a pump of (near)-incompressible fluid. At an analytical level, radial compressors and pumps have considerable similarity: the companion books by Japikse [Ref. 2.4], [Ref. 2.5] treat both in a consistent manner.

Isentropic compression of an ideal gas increases the density by the factor:

$$\left(\frac{\rho_2}{\rho_1}\right) = r^{\frac{1}{k}} \left(= 2.6^{0.8} = 2.15\right),$$

which in our application gives the values in parenthesis. The ratio is 1.0, by definition, for an incompressible fluid (ignoring the slight amount of thermal expansion). Figure 2.5 shows the trajectory in density space for the S-CO_2 main compressor. The density increases by a factor of 1.2: much closer to the pump (incompressible fluid) limit than to the compressor (ideal gas) ratio. If one treats this component as a pump for fluid at the average of inlet and outlet densities, the estimated work is within 2% of that computed using the AXIAL™ compressor code modified to employ NIST Ref. Prop. real gas CO_2 properties.

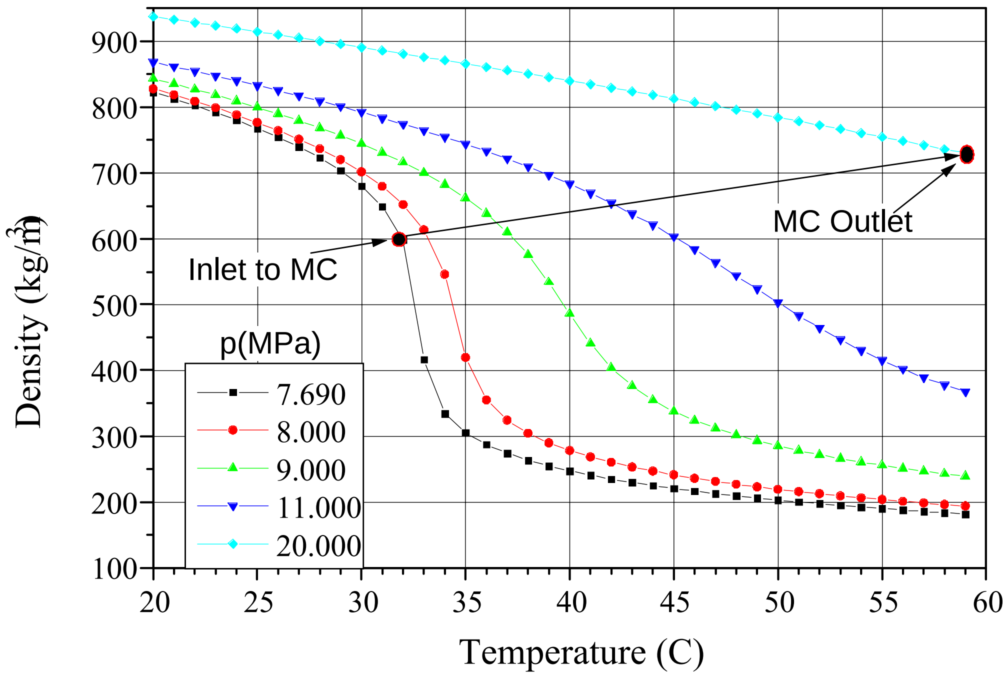

Fig. 2.5 S-CO_2 Density Versus Temperature Showing Main Compressor Trajectory

Thus one is motivated to seek out examples of real world applications of pumps operating in a parameter space encompassing our present application.

The first example is a pumped storage pump [Ref. 2.11]. Table 2.2 compares a large unit of this type to a S-CO_2 cycle compressor to show that much higher throughput is possible, at the required mean pressure ratio per stage, with high efficiency.

Table 2.2 Comparison of Pumped Storage (PS) and S-CO_2 Compressor

	PS, pump mode [Ref. 2.11]	**S-CO_2 main compressor**
Machine type	5-stage radial	8-stage axial
Diameter/length, m	2.0/3.0	0.5/1.0
Power, MW	300 (hence 60 per stage)	40
Head (ΔP), MPa	11.8	12.4
Mass flow rate, kg/s	91500	1915
	(H_2O: 1000 kg/m^3)	(CO_2: 650 kg/m^3)
Volumetric flow rate, m^3/s	91.5	4
Shaft Speed, rpm	300	3600
Efficiency	89.3%	90.5%
Pressure Ratio	118 (hence 2.6 per stage)	2.6

The second example is the F-1 (Saturn-V) rocket engine [Ref. 2.12] which has a centrifugal pump with the following specifications as compared to an axial S-CO_2 main compressor:

Table 2.3 F-1 Centrifugal Pump Compared to S-CO_2 Axial Main Compressor at 32°C

	F-1	**S-CO_2**
Mass flow rate, kg/s	2000 (LOX)	2100 (CO_2)
Fluid density, kg/m^3	1140	650
Rotational speed, rpm	5500	3600
Tip diameter, cm	51 (centrifugal)	80 (axial)
Number of Stages	1	7 (for 32°C inlet)
Discharge pressure, MPa	10	20
Pressure ratio	10	2.6
Efficiency, %	75	89

The operating regime of the two units is rather similar, but the F-1 designers have settled for lower efficiency in the interest of extreme reliability and ruggedness, as well as stability over a very wide range of parameter space. Other references suggest that centrifugal machines can be made considerably more efficient, but nevertheless appear to be about 4% less so than axial machines in the same service: this would only cost us about 1% in S-CO_2 cycle efficiency. Ref. [2.12] also describes axial turbopumps designed for such rocket motor applications.

Another example is Rankine steam cycle feed pumps, which raise pressure from ≈1 MPa to as high as 28 MPa for supercritical steam cycle applications. Pressure ratios are on the order of 2 per stage on these multistage units. The ABWR has two pumps in parallel 22 inch feedwater circuits; each provides 960 kg/s--- a mass flow rate about half that of our S-CO_2 main compressor.

Among existing radial compressors, the "world's largest" CO_2 compressor recently designed, built and tested by MHI is worthy of note [Ref. 2.13]. It has the following characteristics:

Type	radial, 7 stages
Inlet pressure	0.1 MPa
Outlet pressure	20.3 MPa
Stage pressure ratio	2.14 (inferred)
Throughput	10.3 m^3/s (inlet)
	21.6 kg/s
Power input	11.7 MW
Efficiency	85%
Discharge density	291 kg/m^3 @ 172.7°C

The proposed application is for disposal, by subterranean injection, of CO_2 from boiler exhaust gas, and builds on their experience with similar compressors used in fertilizer (urea) production plants. Because the machine ingests its gas at 1 atm, it is more appropriate to compare volumetric throughput: our S-CO_2 compressor has a volumetric rate of about 4 m^3/s: of the same order as the MHI machine. Hence its final two stages have some relevance to S-CO_2 PCS applications.

2.4 Performance Attributes

The use of radial compressors was initially ruled out on the basis of their lower efficiency and throughput compared to axial compressors. Most references suggest an efficiency deficit of on the order of 4%. Work since has weakened this motivation:

(1) Cycle efficiency calculations show that 4% lower compressor efficiency reduces overall PCS efficiency by only 1%: see Fig. 2.6.

(2) Plant layout and scale-up studies have shown that PCS ratings above about 300 MWe are not attractive.

Another consideration is that the theoretical understanding and analytical modeling of radial compressors appears to have lagged that of their axial counterparts. However, there is evidence that this gap is being closed. For example, Ref. [2.14] reports that a conventional 80.3% efficient impeller was improved to 86.5% using "inverse design methodology". Moreover radial pump (i.e. incompressible fluid) efficiencies appear to be slightly higher than that of true radial compressors working in the highly compressible ideal gas region. Ref. 2.1 notes that some "some pumps have achieved very high efficiencies, frequently ranging between 85% and 93%".

An even more recent consideration relates to the effect of compressor type on PCS dynamic response and stability. The S-CO_2 cycle appears especially vulnerable in this regard because it has two compressors running in parallel. In general, radial compressors should be better, but until detailed modeling is carried out this cannot be affirmed. For example, while centrifugal pumps improve series circuit stability, they are prone to hunt, or hog the load when even ostensibly identical units are operated in true parallel circuits without significant throttling [Ref. 2.15]. This type of behavior, however, applies to machines on separately driven shafts. In the present instance a single shaft at constant speed couples the main and recompressing compressor. Furthermore, a flow split control valve is present. Hence we do not expect problems of this type. A final complication is that the recompressing compressor could well be axial. We have not found any discussion of parallel circuit instability for axial turbomachines, let alone mixed radial-axial combinations.

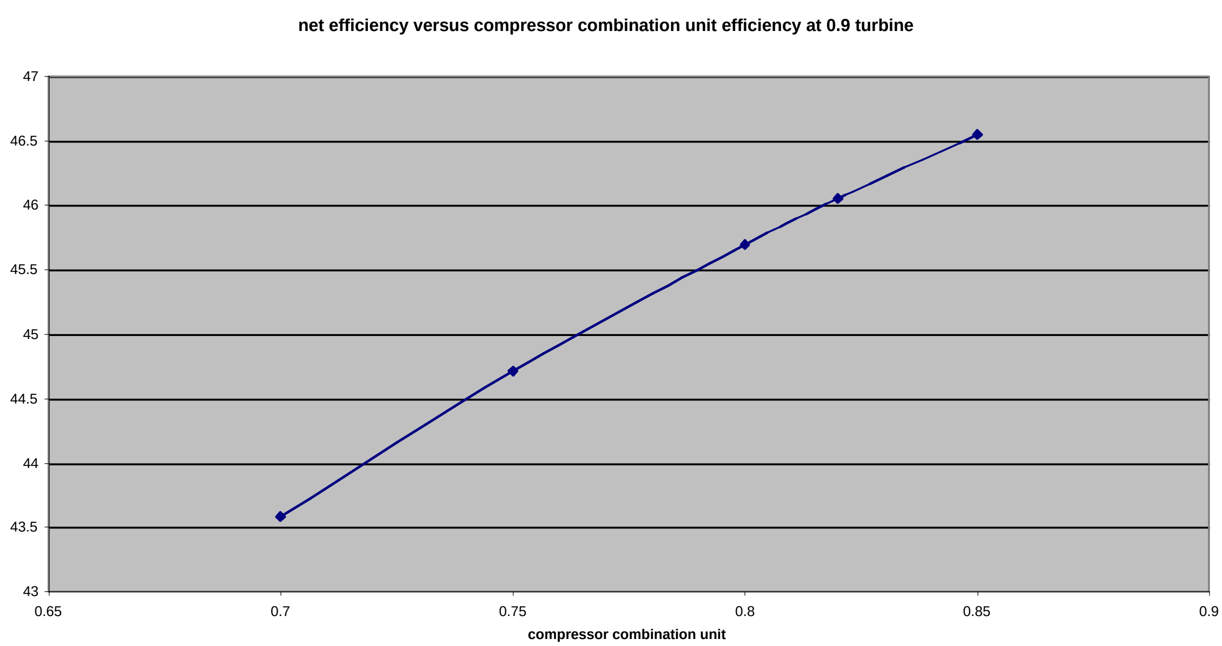

Fig. 2.6 S-CO_2 PCS Efficiency as a Function of Compressor Efficiency (Both Compressors: Main and Recompressing)

2.5 Scalability

One important pragmatic reason for preferring radial over axial machines is the ability to build a small scale radial compressor (and then proceed to a complete small scale power conversion system), with the capability of scaling up results to predict full-size behavior with a high degree of confidence. Below a few hundred kilowatts axial machine blade heights shrink to a few millimeters, precluding acquisition of meaningful data which can be extrapolated to several hundred megawatt machine characteristics.

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Chapter 3 Main Compressor and Recompression Compressor Design

3.1 Introduction

This chapter presents the steady state full-power design for a radial (centrifugal) compressor in our reference 300 MWe power conversion system. As noted earlier, this is the most challenging component in a supercritical CO_2 PCS because it operates near the critical point of CO_2, and has an exceptionally high mass and volume flow rate for a radial compressor. The design is also strongly constrained by the goal that a single shaft, constant speed (3600 rpm) without gearing, be employed. Thus one of the few major variables left for the designer is the number of stages: in effect one vs. two, since the overall pressure ratio is low (2.6) for a centrifugal machine.

The present chapter deals only with full-power, steady-state performance. Chapter 4 covers development of the off-normal performance correlations needed for PCS transient and control studies.

The chapter concludes with a brief comparison with the axial main compressor design reported earlier (3.5) and Chapter 5 discusses, in the interest of completeness, another alternative: a hybrid radial-axial machine in which the inlet stage is radial.

3.2 Design Method

The method is based on specific speed, which is commonly used by industry for radial turbomachinery during the conceptual design stage.

The method of estimating the performance and size of radial compressors and turbines is described by Balje [1980]. Two non-dimensional parameters are used to characterize the turbomachinery. These two parameters are referred to as specific speed (Ns) and specific diameter (Ds). The specific speed and specific diameter are defined as follows:

$$Ns = \frac{\Omega \sqrt{V}}{\left(g\, H_{ad}\right)^{3/4}} \qquad (3\text{-}1)$$

$$Ds = \frac{D\left(g\, H_{ad}\right)^{1/4}}{\sqrt{V}} \qquad (3\text{-}2)$$

where V is the volume flow rate [m^3/s], H_{ad} the adiabatic head [m], Ω the rotating speed [rad/s].

The exact physical meaning of Ns is not obvious. One way to look at it is that it is a comparison between the non-dimensional flow rate, or flow coefficient (ϕ) and the non-dimensional pressure rise (φ).

Non-dimensional flow coefficient is defined as

$$\phi = \frac{\dot{m}}{\rho \Omega D^2} \quad (3\text{-}3)$$

And non-dimensional pressure rise is defined as

$$\varphi = \frac{\Delta P}{\rho \Omega^2 D^2} \quad (3\text{-}4)$$

where Ω is the rotating speed, D the impeller tip diameter, ρ the density. Then Ns can be written as [Cumpsty]

$$Ns = \frac{\phi^{1/2}}{\varphi^{3/4}} \quad (3\text{-}5)$$

It is well established in the industry that radial turbomachinery efficiency can be correlated to the specific speed [Rogers, 1980]. For each given specific speed, there is a particular specific diameter, which will give that (optimal) efficiency. Fig. 3.1 and Fig. 3.2 are correlations for radial compressors.

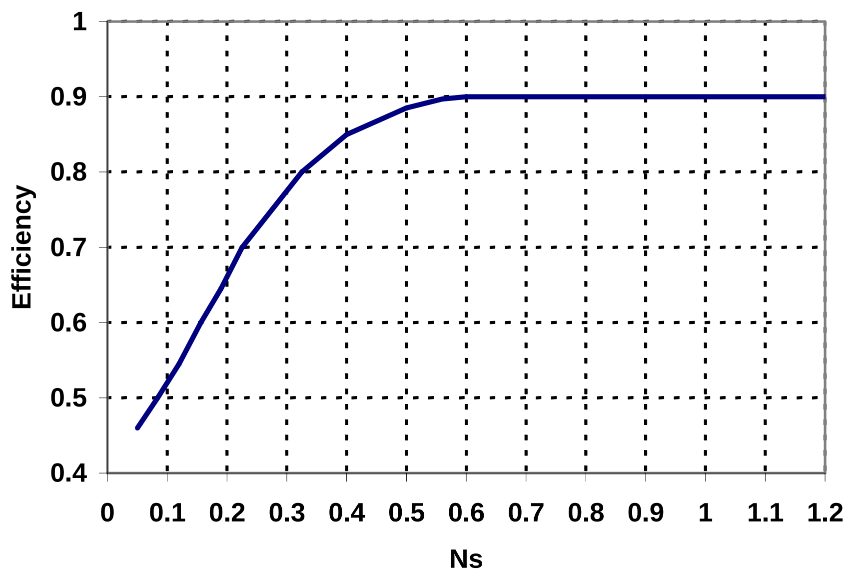

Fig. 3.1 Efficiency vs. Ns for Radial Compressors, Derived from Fig. 3.8 in Balje [1980]

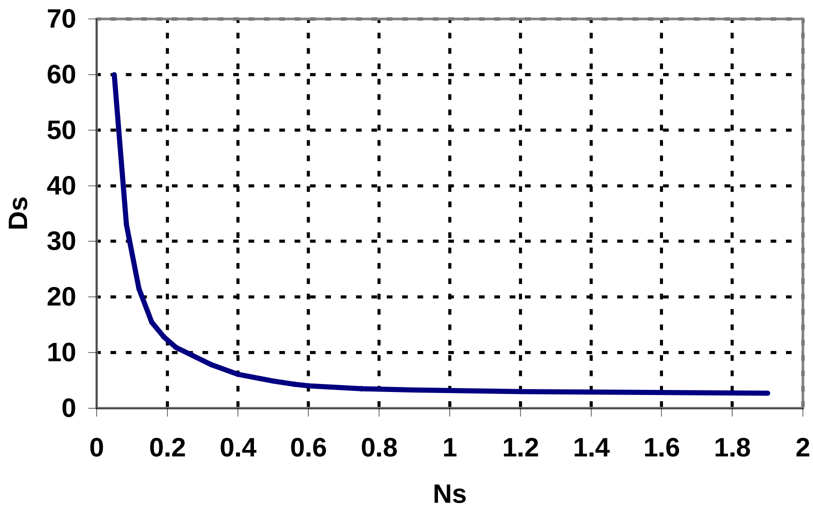

Fig. 3.2 Ns vs. Ds for Radial Compressors, Derived from Fig. 3.8 in Balje [1980]

The estimation procedure is as follows:

(1) For given cycle parameters, the exit density can be estimated. The density used by Eq 3.1 and Eq. 3.2 takes the mean value between the densities at the inlet and the exit
(2) The specific speed can be calculated using density, rotating speed, and pressure rise.
(3) The efficiency can be estimated using the curve in Fig. 3.1.
(4) Specific diameter can be obtained using the curve in Fig. 3.2
(5) From the definition of specific diameter (Ds), the impeller diameter can be computed.

3.3 Main Compressor Design for a 300MW PCS Unit

The operating conditions of the main compressor are listed in Table 3.1. A design study will be presented for single-stage, and two-stage designs.

Table 3.1 Operating Conditions of the Main Compressor for 300MW PCS unit

Mass Flow Rate (kg/s)	1915
Shaft Rotational Speed (RPM)	3600
Total-to-total Pressure Ratio	2.6
Inlet Total Temperature (^oC)	32
Inlet Total Pressure (MPa)	7.69

3.3.1 Single Stage Design

The efficiency of the radial compressor is initially estimated as 85%. Using NIST fluid property software, the inlet and exit conditions are estimated and listed in Table 3.2.

Table 3.2 Inlet and Exit Conditions of Main Compressor, 300MW PCS Unit

	Inlet	Exit
T (^oC)	32	61.545
P (MPa)	7.69	20.0
ρ (kg/m^3)	598.8	713.9

The mean density between inlet and exit is 656.4kg/m3.

Using the previously described method, the specific speed, specific diameter, efficiency and impeller diameter are listed in Table 3.3.

Table 3.3 Efficiency and Impeller Size of Main Compressor Using Single Radial Stage

300MW PCS unit main compressor.	
Ns	0.402
Ds	6.08
Efficiency	85.07%
Impeller diameter	0.887 m
Impeller tip speed	167 m/s

Both size and efficiency are in an acceptable range. Note that total diameter (impeller plus diffuser plus scroll/plenum) will be about 1.5m. Since the optimal Ns is usually around 0.7, this means that the compressor efficiency can be improved as the cycle power increases. The cycle power vs. main compressor efficiency is plotted in Fig. 3.3.

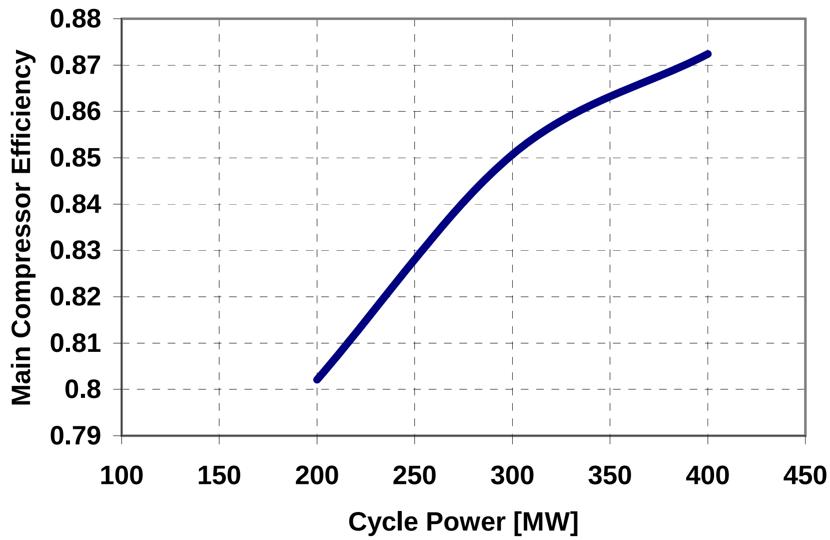

Figure 3.3 Effects of Cycle Power on the Main Compressor Efficiency, 300MW Single Radial Stage Design

3.3.2 A Two-stage Design

Another way to improve efficiency is to use two radial stages for the main compressor. A two-stage design will increase mechanical complexity but reduce the diameter of each impeller, which will reduce centrifugal loading of the impeller. Again using the procedure described in Section 3.2, with evenly distributed pressure rise in both stages, the performance and size of the impeller of each stage are listed in Table 3.4.

Table 3.4 A Design for the Main Compressor Using Two Radial Stages

300MW PCS unit main compressor, each stage of a two-stage design	
Ns	0.552
Ds	4.38
Efficiency	89.54%
Impeller diameter	0.621 m
Impeller tip speed	117 m/s

Compared to the one stage design, the efficiency is increased by 4.5%, and diameter is reduced by 30%. Therefore, a two-stage design is an attractive option since it gives higher efficiency, and also smaller radius, which means that structure loading is lower.

3.4 Recompression Compressor

Operating conditions for the recompression compressor are listed in Table 3.5.

Table 3.5 Working Conditions for the Recompression Compressor

Mass Flow Rate (kg/s)	1331
Shaft Rotational Speed (RPM)	3600
Total-to-total Pressure Ratio	2.6
Inlet Total Temperature* (^oC)	73
Inlet Total Pressure* (MPa)	8.0

If the efficiency of the radial compressor is estimated as 85%, using NIST fluid property software, the inlet and exit conditions are estimated and listed in Table 3.6.

Table 3.6 The Fluid Property at the Inlet and the Exit of the Recompression Compressor

	Inlet	Exit
T (^oC)	73	163.8
P (MPa)	8	20.8
ρ (kg/m^3)	169.3	317.0

The mean density is 243.2kg/m^3.

3.4.1 One-Stage Design

Using the procedure described in Section 3.2, the performance and size of the recompression compressor can be obtained. The key design parameters are listed in Table 3.7.

Table 3.7 Key Design Parameters and Performance for the Recompression Compressor Using a Single Radial Stage

300MW PCS unit recompression compressor.	
Ns	0.254
Ds	10.0
Efficiency	72.9%
Impeller diameter	1.546 m
Impeller tip speed	291 m/s

The result gives unacceptable efficiency with a very large impeller. The diameter of the impeller is slightly over 1.5m, which gives an overall size of the radial compressor (including impeller, diffuser, and scroll) of 3m. As the cycle power increases, the efficiency can also be improved. Fig. 3.4 shows the efficiency of the recompression compressor (using a single stage design) for different cycle powers. The results show that the efficiency will not exceed 80% unless the cycle power is much higher. Another way to improve the efficiency is to use a two or three stage design.

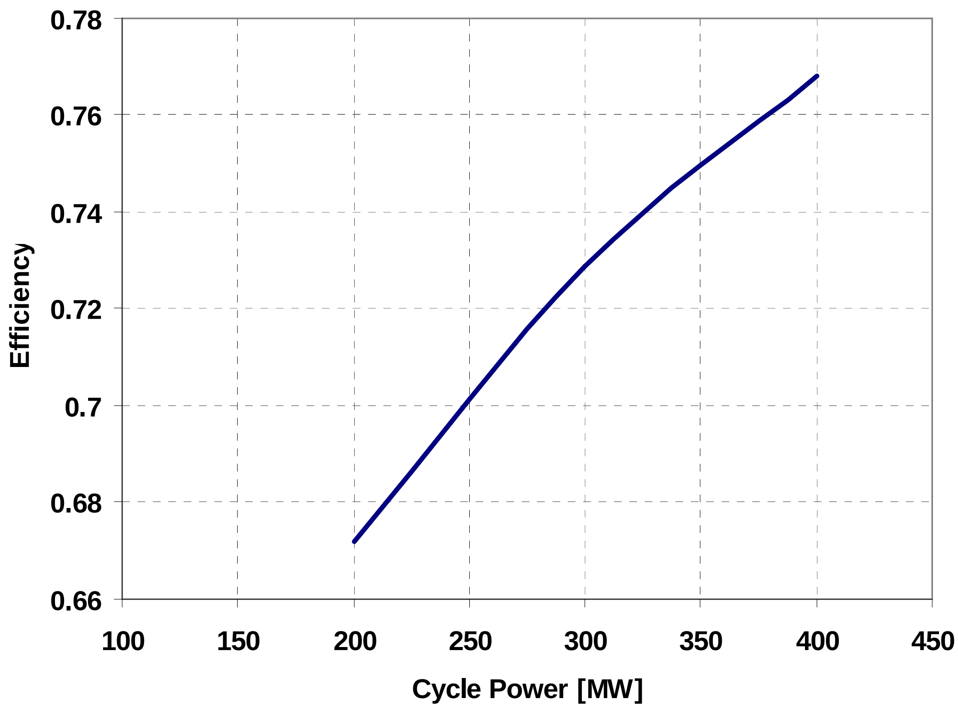

Fig. 3.4 The Impact of Cycle Power on the Efficiency of the Recompression Compressor

3.4.2 Two-Stage Design

Since the pressure rise in each stage is reduced, the specific speed increases. If the pressure rise is evenly distributed between two stages, the key design parameters and performance can be calculated; the results are listed in Table 3.8.

Table 3.8 Key Design Parameters and Performance of Recompression Compressor Using Two Radial Stages

300MW PCS unit recompression compressor, each stage of a two-stage design	
Ns	0.427
Ds	5.78
Efficiency	85.9%
Impeller diameter	1.061 m
Impeller Tip speed	200 m/s

By using a two-stage design, the efficiency increased by 13%, and impeller diameter reduced by 30%.

3.4.3 Three-Stage Design

The efficiency can be further improved if a three-stage design is employed. Table 3.9 lists key design parameters and performance data of a three-stage design of the recompression compressor.

Table 3.9 Key Design Parameters and Performance of Recompression Compressor Using Three Radial Stages

300MW PCS unit recompression compressor, each stage of a three-stage design	
Ns	0.579
Ds	4.17
Efficiency	89.8%
Impeller diameter	0.848 m
Impeller tip speed	160 m/s

The three-stage design improves the efficiency by another 4% compared to the two-stage design. Compared with the single stage design, the efficiency is improved by 17%. The size is reduced by 45% compared to the single stage design, and 20% compared to the two-stage design.

The efficiency versus number of stages is plotted in Fig. 3.5. The plot clearly shows that there will be no significant efficiency gain as more stages are used. Fig. 3.6 shows how the diameter varies with the number of stages. Again, as the number of stages increases to above 3, it has less impact on the diameter of the impeller.

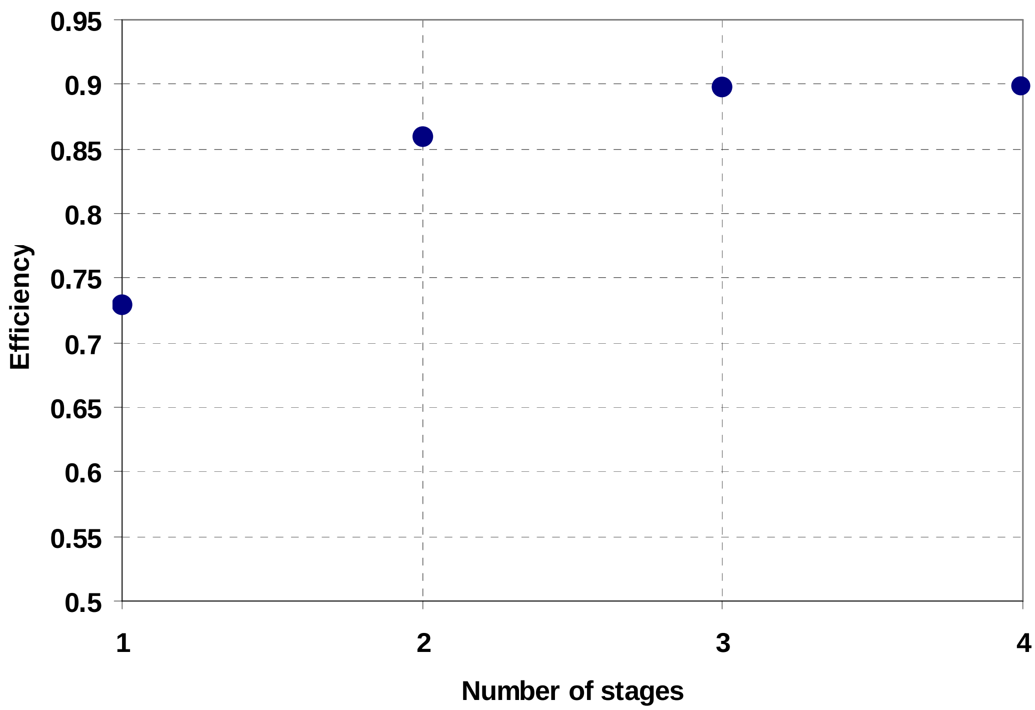

Fig. 3.5 Efficiency vs. Number of Stages for the Recompression Compressor

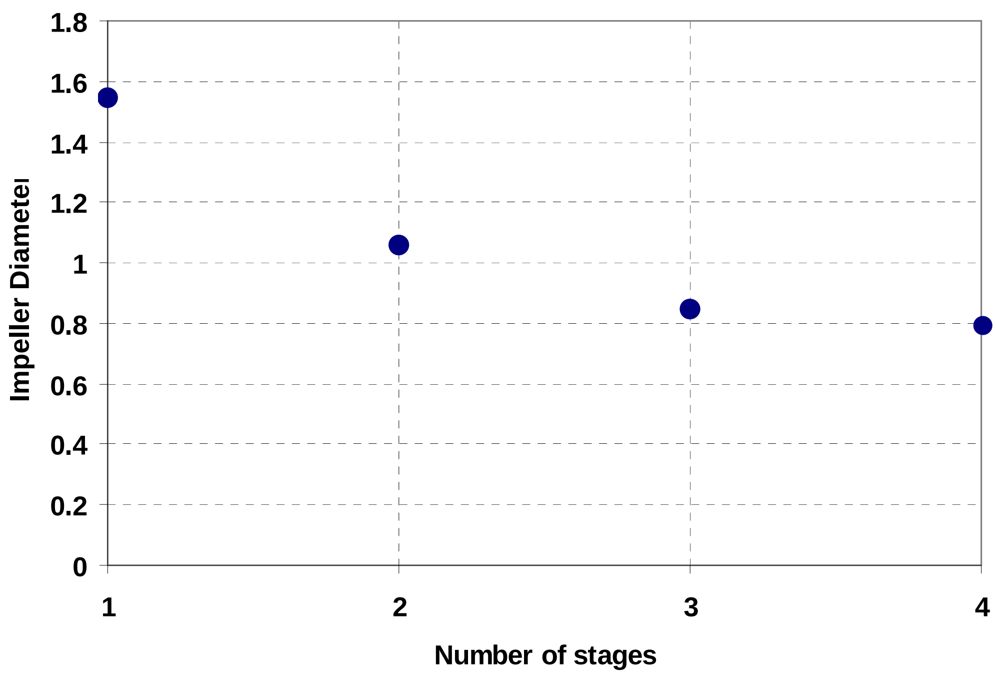

Fig. 3.6 Diameter of the Impeller vs. the Number of Stages

3.5 Summary

A design study has been performed for radial versions of the main and recompression compressors. It is found that for the main compressor, a single stage design can provide adequate efficiency with reasonable size. For the recompression compressor, a single stage design gives unacceptably low efficiency with very large diameter. This will result in low system performance and difficulties in mechanical design. It is found that a two-stage design can significantly increase the efficiency of the recompression compressor, and at the same time, significantly reduce the diameters. A three-stage design can further improve efficiency and reduce diameter.

The study also indicates that further increasing the number of stages does not lead to a significant gain in both efficiency and size. It is concluded that a single stage main compressor and three-stage recompression compressor represent good design choices. As noted elsewhere in this report (see Chapter 6), a hybrid design, which mixes axial and radial stages, may capture the benefits of both options. Therefore we suggest that a further study should be carried out for a hybrid design.

3.6 References for Chapter 3

3.1 Rodgers, C., "Efficiency of Centrifugal Compressor Impellers", Paper 22 of AGARD Conference Proceedings No 282, 1980.

3.2 Aungier, R. H., Centrifugal Compressors, A Strategy for Aerodynamic Design and Analysis, ASME Press, 2000.

3.3 Balje, O.E., Turbomachines: A Guide to Design, Selection, and Theory, John Wiley & Sons, Inc., 1981.

3.4 Concepts/NREC, User's Guide to AXIAL™, Version 7.4, 2001.

3.5 Wang Y., Guenette G.R., Hejzlar P., and Driscoll M.J., "Aerodynamic Design of Turbomachinery for 300 MWe Supercritical Carbon Dioxide Brayton Power Conversion System", MIT-GFR-022, Topical Report, Department of Nuclear Science and Engineering, MIT, March 2005.

3.6 Cumpsty, N.A., Compressor Aerodynamics, Krieger Publishing Company, 2004

Chapter 4 Off-Normal Compressor Characteristics

4.1 Introduction

The single most compelling reason for preferring use of a radial rather than an axial main compressor is the much wider operating range (between stall and surge) of the radial machine. Hence the final selection of type may come down to component/PCS performance during transients. This evaluation is a separate task in the subject project, in which the GAS-PASS PCS dynamics code is being modified to handle non-ideal, near-critical working fluids. Thus maps of radial compressor off-normal operating characteristics must be developed in the specific format needed by GAS-PASS.

Turbomachinery analysts differ in their choice and normalization of parameter groupings for such maps. The convention adopted for GAS-PASS is to employ efficiency and pressure ratio curves (both normalized by their full power values) as a function of mass flow rate (again normalized), for a range of shaft speeds (again normalized). Fig. 4.1 shows the input developed earlier in this project for the <u>axial</u> main compressor. For comparison Fig. 4.2 shows <u>radial</u> compressor (non-normalized) operating maps from Ref. [4.1] for a conventional (<u>ideal</u> gas) machine. Both are useful frames of reference for comparison later with our radial, real gas compressor maps.

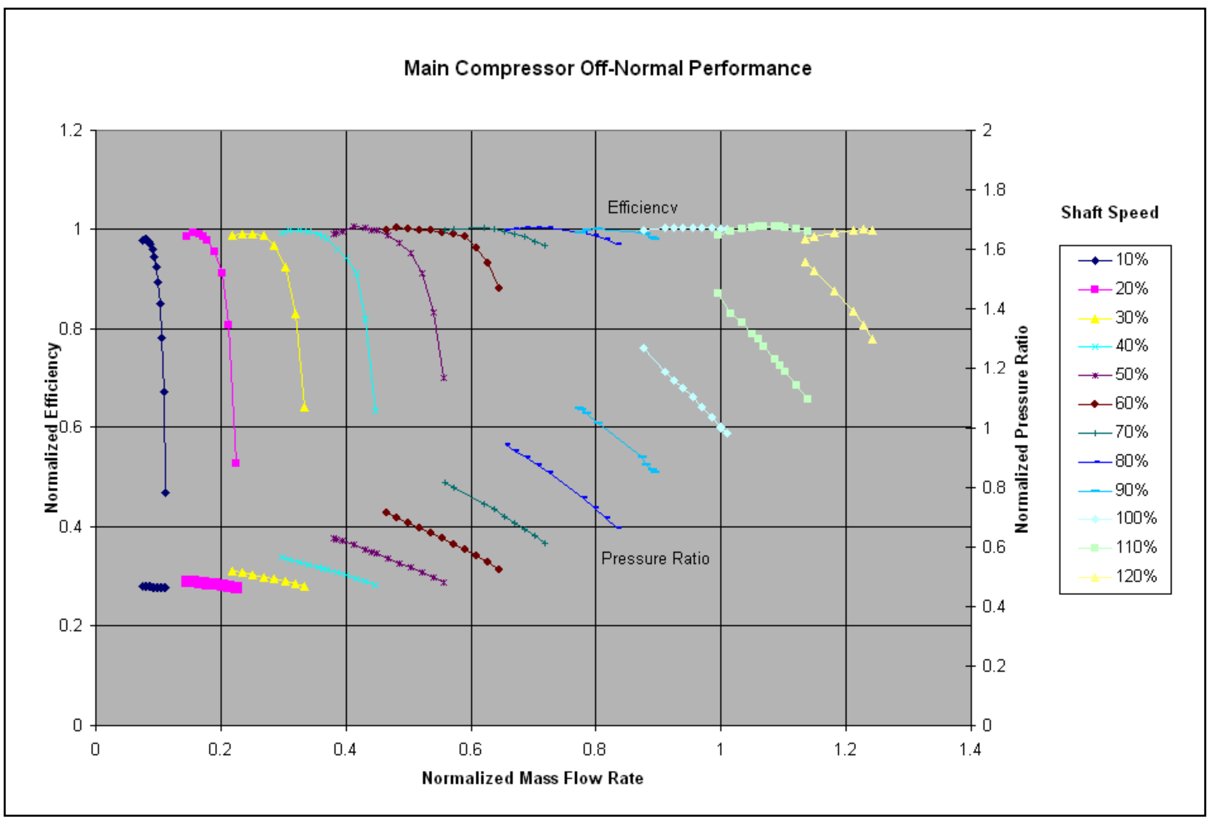

Fig. 4.1 AXIAL Main Compressor Off-Normal Performance Map

Centrifugal compressor experimental development

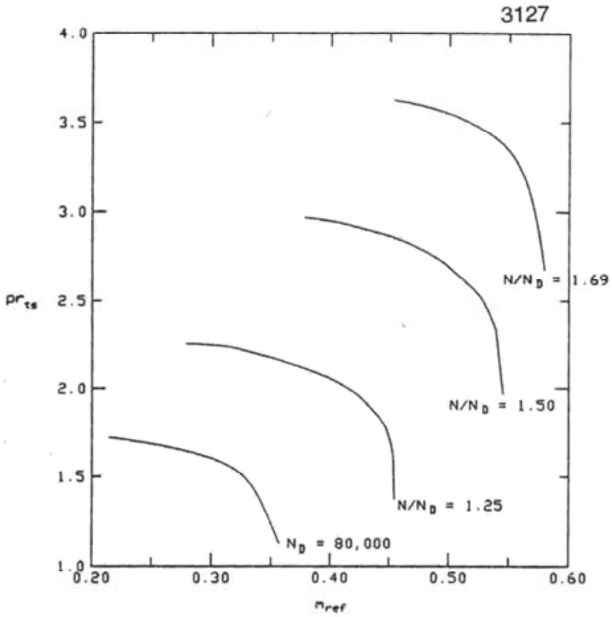

Figure 7.1a Basic operating map of pressure ratio versus flow

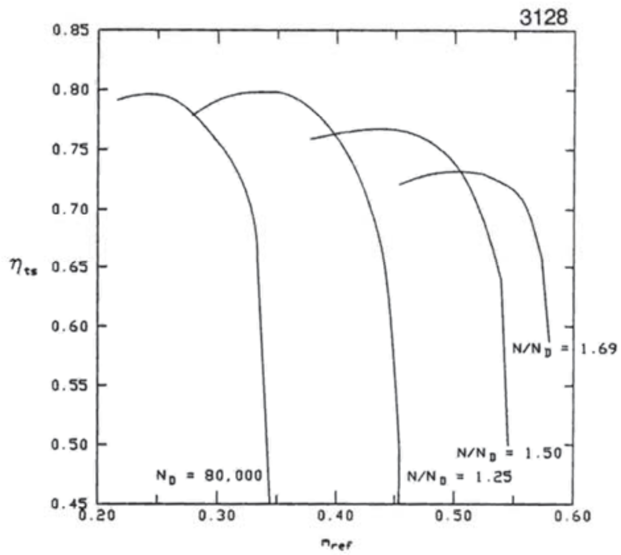

Figure 7.1b Basic operating map of efficiency versus flow

Fig. 4.2 Operating Curves for a Conventional (Ideal Gas) Centrifugal Compressor
[from Ref. 4.1]

The off-normal (or off-design) performance estimation is essential for part load operation, transient simulation, and other changes in operational conditions (for example at different plant cooling water temperatures). It should be noted that off-design performance estimation is a challenging task even for conventional turbomachinery with ideal gas properties. Thus what follows is a first order estimate of the behavior of turbomachinery under off-normal conditions. This work should continue to obtain more reliable calculations of off-normal performance. In this section, aerodynamic scaling is briefly reviewed, and some data are surveyed to give a justification of the current approach which is constructed based on low speed compressor approximations.

4.2 Approach to Off-normal Performance of Compressors for S-CO_2 Cycle

Based on aerodynamic scaling, for a given compressor (geometry) and ideal gas (constant specific heat ratio, and gas constant), the pressure ratio and efficiency are functions of corrected mass flow (which reflects the inlet Mach number) and correct rotating speed (which is proportional to the Mach number of compressor tip speed).

$$\frac{Pt,exit}{Pt,inlet} = f\left\{\frac{\dot{m}\sqrt{Tt,inlet}}{Pt,inlet}, \frac{\Omega}{\sqrt{Tt,inlet}}\right\} = f\{\dot{m}_{corr}, \Omega_{corr}\} \quad (4\text{-}1)$$

$$\eta \quad = \quad g\left\{\frac{\dot{m}\sqrt{Tt,inlet}}{Pt,inlet}, \frac{\Omega}{\sqrt{Tt,inlet}}\right\} \quad = \quad g\{\dot{m}_{corr}\ , \Omega_{corr}\} \qquad (4\text{-}2)$$

The following figure is a typical radial compressor map. If is obvious that a compressor map cannot be represented by a single curve (as is sometimes done) when it is nondimensionalized by reference mass flow and reference rotating speed even for fixed inlet conditions (constant T and Pt at inlet).

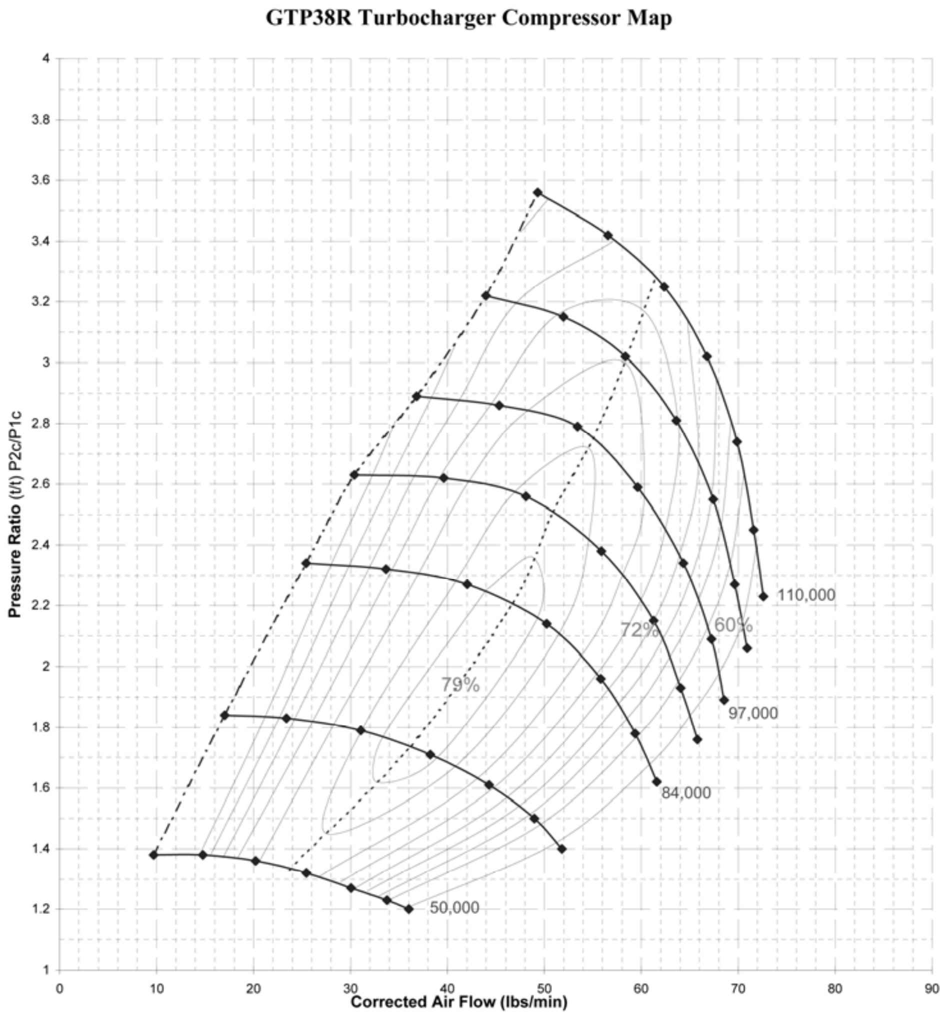

Fig. 4.3 A Performance Map for an Existing Radial Compressor

For low Mach number limits (in low pressure rise compressors and pumps in which the speed sound is very high), the pressure rise is near unity and Mach number effects can be neglected. The compressor pressure rise (normalized by dynamic head) and efficiency are functions of flow coefficient only.

$$\frac{\Delta P}{\rho U^2} \quad = \quad f(\phi) \qquad (4\text{-}3)$$

$$\eta \quad = \quad g(\phi) \tag{4-4}$$

For a radial compressor, flow coefficient is defined by Eq (3-3). For an axial compressor, flow coefficient is defined as

$$\phi \quad = \quad \frac{Vx}{U} \tag{4-5}$$

In general for high-pressure compressors, the Mach number effect becomes quite significant. For the pressure ratio (of ~ 2.6) used by the S-CO_2 cycle, the low speed representation might be sufficient. The following figure shows a compressor map plotted in a different way. The compressor has a pressure ratio of above 2, which is quite close to the pressure ratio for our S-CO_2 cycle. The example shows that Mach effects are not significant for a radial compressor with pressure ratio of 2.

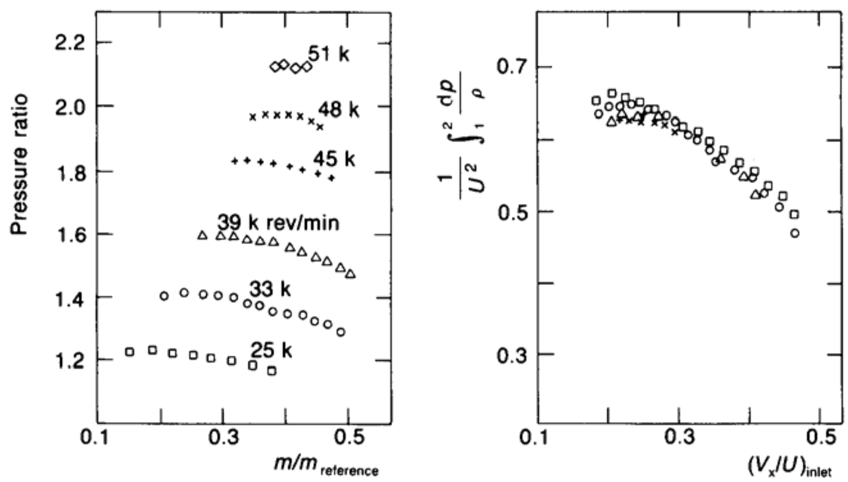

Figure 4.4 A Compressor Performance Map Represented in Two Different Ways. The Compressor Is a Radial Type with Pressure Ratio of 2.2

One implication is that for the current application, the compressor characteristics can be represented using an approach for incompressible flow (i.e. for a pump). Therefore the way in which compressor maps are represented in GAS-PASS is an acceptable approximation for the S-CO_2 cycle.

4.3 Constructing an Off-Normal Performance Map for a Radial Compressor

Based on the previous discussion, we conclude that for the pressure ratio of the S-CO_2 cycle, the low speed assumption is still a good approximation, if we assume that the behavior of our compressor is similar to a conventional compressor with ideal gas. It is still a significant challenge to develop a reliable method for prediction of a compressor map if the inlet fluid is near its critical point. Current knowledge is not sufficient for turbomachinery which is operated near the critical point of the working fluid. At this stage of the program, the best (most reliable) way to construct a compressor map is using methods developed for incompressible turbomachinery. For an incompressible fluid compressor, the performance (pressure rise and efficiency) is a function of flow coefficient, as shown in Eq (4-3), and Eq (4-4).

Based on these ideas, the compressor map is constructed using the following assumptions:

(1) pressure rise is scaled with $U^2\rho$
(2) the mass flow is scaled with $U\rho$

where U is the wheel speed, which is proportional to rotational speed, and ρ is fluid density.

The entire map can be constructed using a nominal pressure rise characteristic. The entire compressor map can be expressed as

$$\Delta P = \Delta P_{ref}\, f_p(\varphi) * (U/U_{ref})^2 * (\rho/\rho_{ref}) \tag{4-6}$$

$$\eta = \eta_{ref}\, f_\eta(\varphi) \tag{4-7}$$

$$m = m_{ref} \frac{U}{U_{ref}} \frac{\rho}{\rho_{ref}} \tag{4-8}$$

The reference values can be the design conditions estimated in the previous sections. Sample compressor maps are plotted in Figure 4.5 and Figure 4.6. The normalized pressure is defined as $\Delta P/\Delta P_{ref}$, and normalized mass flow m/m_{ref} .

It can be observed that compressor efficiency does not change[1] when speed and mass flow rate are reduced proportionally. However, lower speed results in a smaller pressure rise, affecting cycle efficiency. Therefore, for lower speeds, cycle net efficiency will be reduced.

[1] This is an approximation. In reality, the peak efficiency changes slightly with rotating speed (see Fig. 4.2).

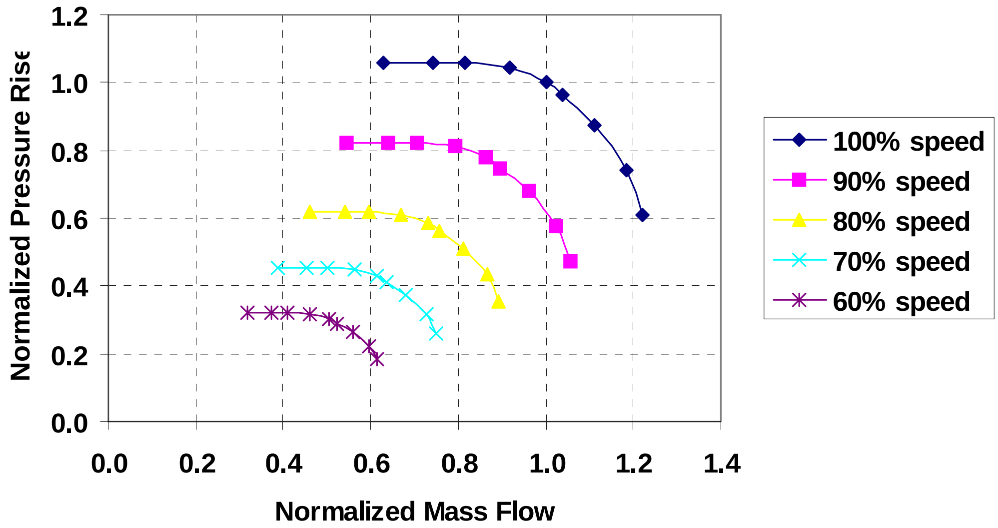

Fig. 4.5 Sample compressor Pressure Rise Characteristics

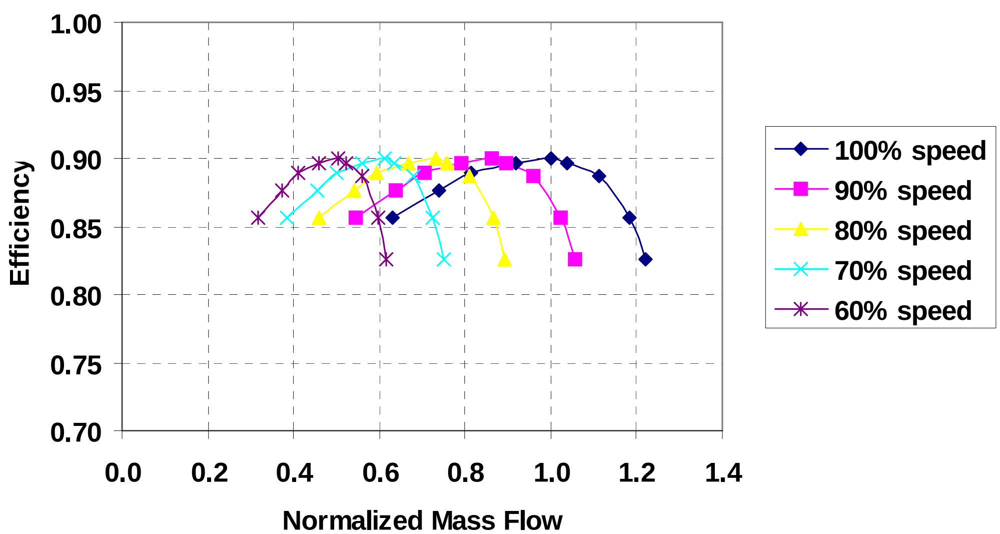

Fig. 4.6 Sample Compressor Efficiency Characteristics

A final note is in order on the difference between axial and radial compressors. It is generally accepted that radial compressors have a wider operating range than axial compressors. The statement is certainly consistent with many existing machines. The reason is that a radial machine can be operated at low mass flow, since the pressure rise at the impeller is due to centrifugal force. At the inlet, the relative Mach number is usually much smaller than the axial machine with a similar pressure rise. Therefore it has a large section of its performance curve which is almost flat at low mass flow. The difference can be illustrated in the following two cases. Fig. 4.7 is a performance map for a single stage fan (axial machine). At a pressure ratio of 1.8, the fan tip speed is already above Mach 1. The speed curves are quite steep.

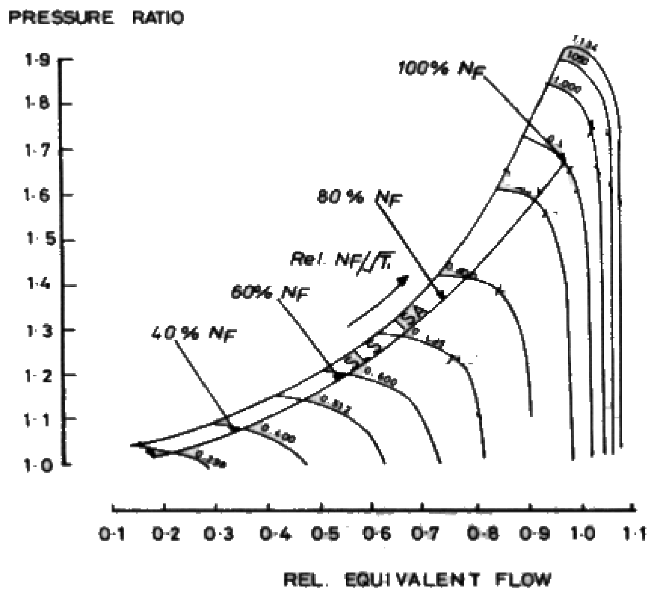

Fig. 4.7 A Performance Map for an Existing Axial Fan

Fig. 4.8, which is the lower pressure ratio portion of Fig. 4.3, shows a quite different picture. At a pressure ratio around 2, the curve is still very flat, since the relative Mach number at the inlet is still low. The argument here again explains why the off-normal compressor map for a radial compressor can be constructed using the low speed approximation.

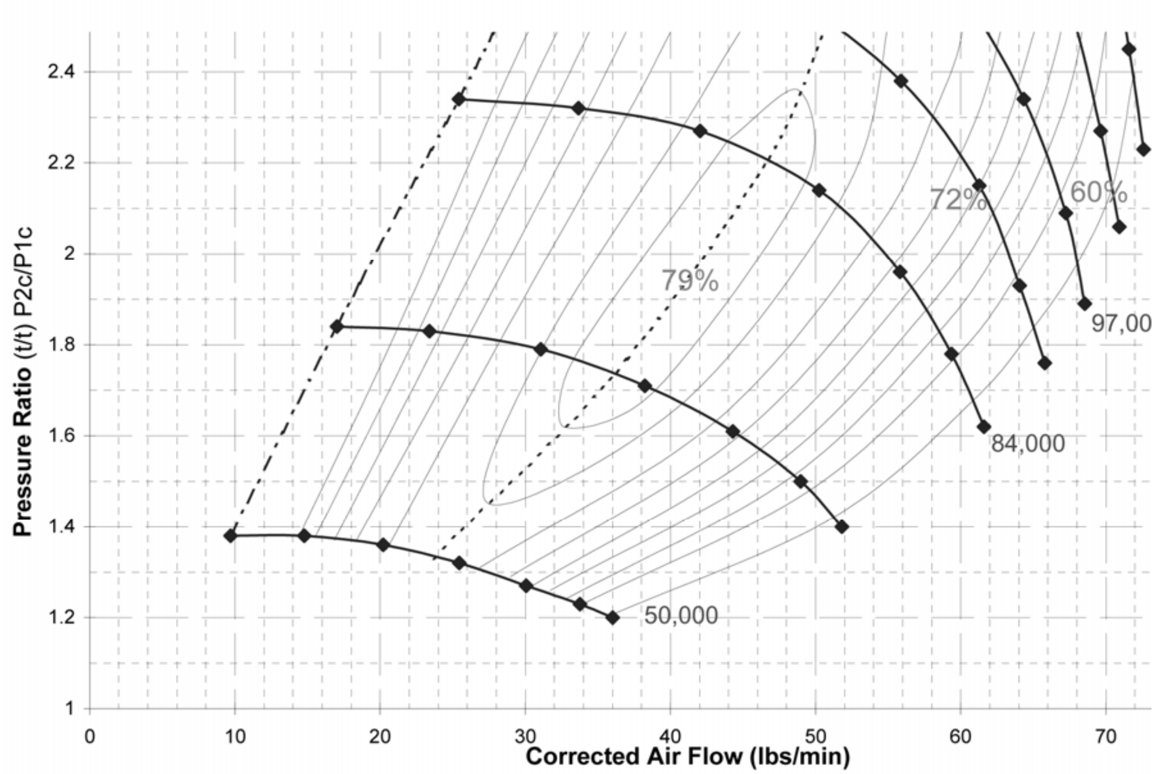

Fig. 4.8 A Compressor Map for a Typical Radial Compressor

4.4 Summary

Figs. 4.5 and 4.6 are the best currently available off-normal performance maps for our radial compressors in advanced of actual experimental data on machines constructed for, and tested in, a supercritical CO_2 loop. The maps are normalized to reference state performance and can, at this stage, be applied to both the main and recompressing compressors. They are in the form required for input into the revised GAS-PASS cycle dynamics program under development by MIT/ANL.

4.5 References for Chapter 4

4.1 Wang Y., Guenette G.R., Hejzlar P., and Driscoll M.J., "Aerodynamic Design of Turbomachinery for 300 MWe Supercritical Carbon Dioxide Brayton Power Conversion System", MIT-GFR-022, Topical Report, Department of Nuclear Science and Engineering, MIT, March 2005.

4.2 Cumpsty, N.A., Compressor Aerodynamics, Krieger Publishing Company, 2004.

4.3 Fink, D. A., Surge Dynamics and Unsteady Flow Phenomena in Centrifugal Compressors, GTL Report 193, Gas Turbine Laboratory, Massachusetts Institute of Technology, 1988.

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Chapter 5 Radial Turbine Assessment

5.1 Introduction

Although there appear to be considerably fewer obvious incentives for substituting a radial turbine for our reference axial design, a brief evaluation was carried out. One motivating factor was that, if radial machines are specified for the compressors, a radial turbine might be more compatible if (as in the reference choice) a single shaft layout is specified.

5.2 Radial Turbine Characteristics

The efficiency of a radial turbine has been estimated. For a single stage radial turbine, the efficiency is estimated as 0.85, due to the relatively low specific speed (which is 0.32). A two-stage design gives an efficiency of 0.9. Both of these values are significantly less than the efficiency of the current axial design. The key characteristics of the radial turbines are as follows.

Table 5.1 Design of Radial Turbines and Comparison to the Axial Turbine

	Efficiency	Rotor tip diameter [m]	Specific speed
1 radial stage	0.85	2.2**	0.32
2 radial stages	0.90	1.4**	0.54
4 axial stages	0.95*	1.25	Not Relevant

* With a diffuser at its exit. The pressure recovery is 0.7.
** The diameter of a stage is about 2 times the diameter of the rotor, to allow for a radial diffuser and volute.

Our results are consistent with a Barber Nichols preliminary design value of 1.9 m rotor diameter for a single stage turbine cited at the previously noted 8/31/05 industrial review meeting. The large diameter and low efficiency of the one-stage radial turbine rule out its use in the present application: for example, a 5% reduction in turbine efficiency results in about 2% lower overall cycle efficiency. The preference for an axial machine is clear.

5.3 Conclusions and Recommendations

Unless future cycle transient studies, which will include mixing radial compressors with an axial turbine, contraindicate, we recommend dropping the radial turbine option. Since the turbine operates in a near-ideal-gas range of pressures and temperatures, it is not likely that our scoping studies and intercomparisons are in any appreciable error.

Chapter 6 Hybrid Radial-Axial Turbomachines

6.1 Introduction

To improve efficiency, size and operating range, designs combining axial and radial machines have been examined. The most likely design option is to use a radial compressor stage followed by several axial stages for the main compressor, and to use axial compressor stages followed by a radial compressor stage for the recompression compressor.

There are several advantages for these configurations.

(1) For the main compressor, putting the radial compressor at the front relaxes the constraint on the maximum wheel speed. The ability to use higher wheel speed can increase the pressure ratio of each stage, therefore reducing the number of stages. The fluid at the main compressor inlet is very close to the critical point. Using a radial type machine can also enhance the robustness of the main compressor.

(2) For the recompression compressor, using a radial stage as the last stage of the compressor can increase the compressor operability range significantly, since the surge margin is set by the rear stages for most multi-stage axial compressors. This type of design choice is common for small conventional gas turbine engines, as shown in Figure 6.1.

(3) For the current design, the radial stage provides about 1/3 of the total pressure rise. Therefore the size (diameter) is much less than for a single radial stage design.

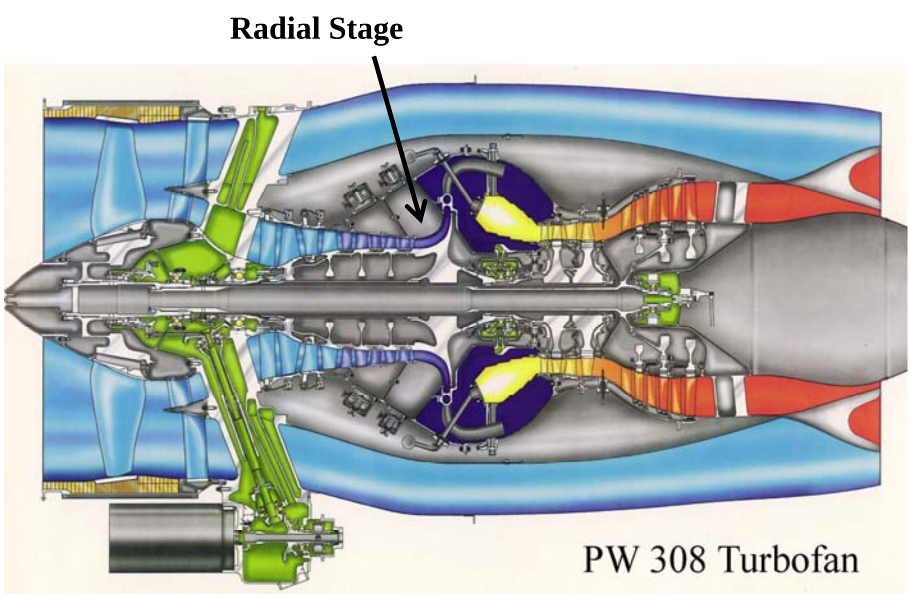

Fig. 6.1 A Radial Stage is Used as the Last Stage of the Compressor in a Small Aircraft Engine

6.2 Configurations and Analyses

The characteristic of the radial stage and compressor configuration are shown in Table 6.1.

Table 6.1 Radial Stage and Compressor Configuration Characteristics

	Configuration	Specific Speed of the radial stage	Efficiency of the radial stage
Main Compressor	1 radial stage + 4 axial stages	0.9	90%
Recompression Compressor	5 axial stages + 1 radial stage	0.6	90%

The layouts of the main compressor and recompression compressor are illustrated in the following two figures, Figs. 6.2 and 6.3.

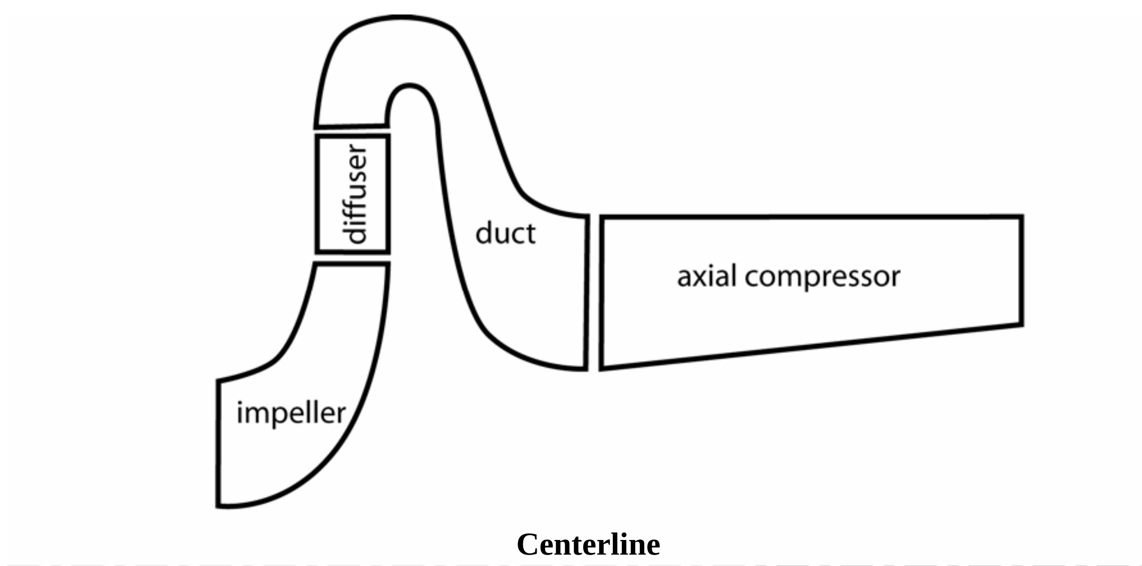

Fig. 6.2 A Proposed Layout of the Main Compressor for a 300MW PCS

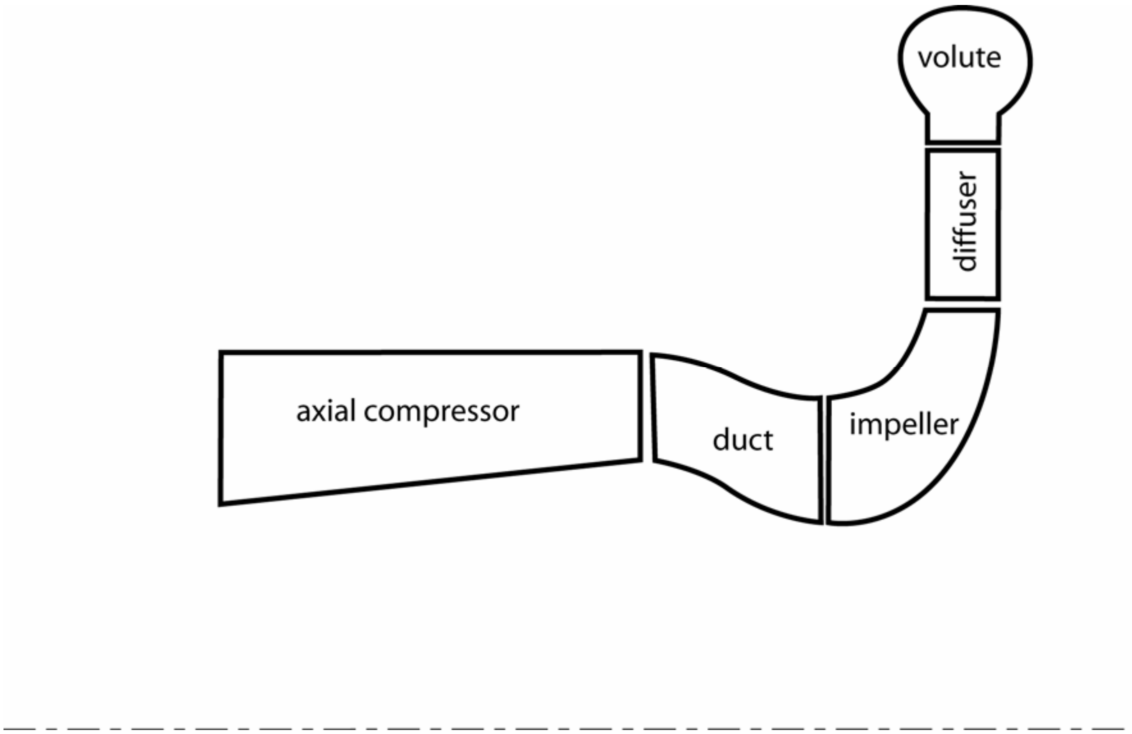

Fig. 6.3   A Proposed Layout of the Recompression Compressor of a 300MW PCS

For the main compressor, the radial stage is located at the front of the compressor. The radial stage key characteristics are listed in Table 6.2.

Table 6.2 Design of the Radial Stage for the Main Compressor

Mass flow [kg/s]	Pressure Ratio	Efficiency	Specific Speed	Rotor Tip Diameter [m]
1915	1.77	0.9	0.7	0.665

Three axial stages follow the radial stage. The geometries of the three stages are the same as stages 5, 6, and 7 in the 7-stage axial design (6.1). The overall efficiency is 0.918. The overall efficiency for the current mixed radial and axial design is the same as the all-axial design. The reason that the efficiency does not decrease is that the efficiency of the front stages for the axial design is relatively low at about 0.9.

The configuration of the recompression compressor is proposed to be axial stages followed by a radial stage. The arrangement is the same as for most other compressors which mix axial and radial stages. The arrangement is four axial stages followed by one radial stage. The key characteristics of the radial stage are shown in Table 6.3.

Table 6.3 Design of the Radial stage for the Recompression Compressor

Mass Flow [kg/s]	Pressure Ratio	Efficiency	Specific Speed	Rotor tip diameter [m]
1331	1.43	0.87	0.47	0.88

The geometries of the four axial stages are the same as the front half of the 8-stage axial design. The overall compressor efficiency is 0.908.

6.3 Conclusions and Recommendations

The use of compressors having both radial and axial stages is an interesting option, which is worth future reconsideration depending on the outcome of cycle control and dynamics studies, and the results of small scale test facility operations. For example, hybrid machines may be more compatible with purely axial turbines and/or axial recompressing compressors under off-design conditions. While machine efficiencies close to those of all-axial compressors can be attained, the overall power cycle efficiency is less sensitive to compressor efficiency than to turbine efficiency. The compressor diameter is smaller than that of single-stage radial machines, but less significantly smaller than two-stage all-radial compressors. Hence, at present, hybrids rank behind their more-rugged, wider-operating range, all-radial counterparts.

6.4 References for Chapter 6

6.1 Y. Wang, G.R. Guenette, P. Hejzlar, M.J. Driscoll, "Supercritical CO2 Turbine and Compressor Design," MIT-GFR-015, June 2004.

Chapter 7 Summary, Conclusions and Recommendations

7.1 Summary and Conclusions

The main conclusions based on the work documented in this report are as follows:

(1) Radial compressors should be adopted as the reference design choice for the 300 MWe S-CO_2 PCS under evaluation at MIT

(2) Specifically, a one stage main compressor and a 3 stage recompressing compressor are recommended. Tables 3.3 and 3.9 are repeated here, summarizing the key parameters of each.

(3) Priority should be given to obtaining experimental data on a scalable main compressor. As noted in Ref. (7.2), "it is standard practice to scale a map of an existing compressor" rather than relying only on analytic/numerical methods. This is in preference to attempting methodology refinements such as those summarized in Ref. (7.2). Since fluid properties have important second order effects (7.3), tests should be carried out using supercritical CO_2 as the working fluid.

(4) The choice of turbomachinery should be revisited after experimental data is available, and after PCS plant transient studies have been completed. At that point evaluation of hybrid radial/axial machines may be justifiable.

Repeat of Table 3.3 Efficiency and Impeller Size of Main Compressor Using Single Radial Stage

300MW PCS unit main compressor.	
Ns	0.402
Ds	6.08
Efficiency	85.07%
Impeller diameter	0.887 m
Impeller tip speed	167 m/s

Repeat of Table 3.9 Key Design Parameters and Performance of Recompression Compressor Using Three Radial Stages

300MW PCS unit recompression compressor, each stage of a three-stage design	
Ns	0.579
Ds	4.17
Efficiency	89.8%
Impeller diameter	0.848 m
Impeller tip speed	160 m/s

7.2 Recommendations

The principal recommendation for future work is that a radial compressor of about 100 kW rating be procured from an experienced commercial vendor, and tested to obtain performance maps in the near-critical region for CO_2. The goal at this scale should be to meet or beat 80% total-to-total efficiency. As noted earlier, the state-of-the-art, as embodied in published methods, does not yet permit high confidence in a strategy which is based only on analytical/ numerical methods. Considerable expertise is still closely held as commercially proprietary information. It should also be recognized that a certain amount of trial-and-error may be involved, such that the device should readily accommodate changing-out impellers.

It is also recommended that, in anticipation of the development of further supporting information, the reference design main compressor for the 300 MWe PCS be a one-stage radial machine. While not expected to be required, substitution of an axial machine should be possible even at a very late date. For example, Frutschi [Ref. 7.1] notes that the second generation of Escher Wyss air-working-fluid Brayton units were, in the early 1960's, originally designed and built using radial compressors because of their reduced size and cost. However, because of lower-than-planned efficiency, their lead-plant Coburg and Haus Aden 6MW units were, after initial tests, changed over to axial units. While radial compressor design and efficiency still lags that of axial machines, the progress over the past 45 years should be sufficient to preclude surprises of this sort: see Fig. 2.3 of Chapter 2.

Finally, more work is needed on dynamic simulation of the power conversion system under transient and accident conditions. For this, realistic off-normal performance maps of high quality are needed for all turbomachinery components. This in general requires acquisition of experimental data rather than reliance on only analytical/numerical studies.

7.3 References for Chapter 7

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7.2 J. Kurzke, C. Riegler, "A New Compressor Map Scaling Procedure for Preliminary Conceptual Design of Gas Turbines," Proc. of ASME IGTI TurboExpo 2000, Munich, May 2000: ASME 2000 GT-0006.

7.3 S.K. Roberts, S.A. Sjolander, "Effect of the Specific Heat Ratio on the Aerodynamic Performance of Turbomachinery," ASME Journal of Engineering for Gas Turbine and Power, Vol. 127, p. 773, Oct. 2005.