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Abstract – *A compressor model has been implemented in the RELAP5-3D© code. The model is similar to that of the existing pump model, and performs the same function on a gas as the pump performs on a single-phase or two-phase fluid. The compressor component consists of an inlet junction and a control volume, and optionally, an outlet junction. This feature permits cascading compressor components in series. The equations describing the physics of the compressor are derived from first principles. These equations are used to obtain the head, the torque, and the energy dissipation. Compressor performance is specified using a map, specific to the design of the machine, in terms of the ratio of outlet-to-inlet total (or stagnation) pressure and adiabatic efficiency as functions of rotational velocity and flow rate. The input quantities are specified in terms of dimensionless variables, which are corrected to stagnation density and stagnation sound speed. A small correction was formulated for the input of efficiency to account for the error introduced by assumption of constant density when integrating the momentum equation. Comparison of the results of steady-state operation of the compressor model to those of the MIT design calculation showed excellent agreement for both pressure ratio and power.*

I. INTRODUCTION

The recent focus on the research, development, design, and exploration of next-generation nuclear energy technologies known as "Generation IV"[a] Reactors has created the need for advanced capabilities to be added to the computer codes that will be used to do analyses for these technologies. The Gas Fast Reactor design, as an example, is designed to operate in a Brayton Cycle. To support this design concept, the capability to simulate the response of a compressor in a Brayton Cycle reactor design, with coupling via a shaft to turbine, generator, or pump components, has been implemented in the RELAP5-3D©/ATHENA code.

The compressor model is similar to the pump model in RELAP5-3D©. It performs the same function on a gas as the pump performs on single-phase and two-phase fluids. It is anticipated that the compressor will usually be driven by a turbine on the same shaft, although the other capabilities presently available in the pump, i.e. the speed table, the motor torque table, and the coastdown feature, are also available in the compressor. The compressor head curve specification is formulated as pressure ratio, instead of directly as head as is done in the pump model and the compressor torque is calculated based on the compressor characteristic curves and the stage adiabatic efficiency. The compressor component consists of an inlet junction and a control volume, and optionally, an outlet junction. The outlet connection may also be an ordinary RELAP5-3D© junction component (e.g. single junction or branch component junction) or it can be another compressor component inlet junction. This feature allows for cascading compressor components into stages, if desired, to improve the accuracy of the representation.

The following sections describe the details of the compressor model. The equations necessary to calculate the head, the torque, and the energy dissipation of the compressor are derived, and details of the implementation in RELAP5-3D©, the method for specifying the input variables, and the results of verification test problems are discussed.

[a] Generation IV refers to the development and demonstration of one or more nuclear energy systems that offer advantages in the areas of economics, safety and reliability, sustainability, and could be deployed commercially by 2030.

II. THEORETICAL DEVELOPMENT

In a compressor, forces acting in the tangential direction only cause a change in angular momentum of the working fluid. Radial and axial forces are constrained by the physical design. Therefore, the total torque applied to a compressor can be thought of as being composed of two parts, one which changes the angular momentum of the fluid, and the other which results in reactive forces in the journal and thrust bearings. In principle, then, the isentropic torque can be calculated by considering an isentropic compression of the fluid and then applying an efficiency factor to get the total torque.

The compressor model begins with the conservation of angular momentum for the compressor rotor. A mass of fluid enters with an initial azimuthal velocity $v_{\theta 1}$ at a radius r_1 and exits with azimuthal velocity $v_{\theta 2}$ at a radius r_2. The torque, τ, required for this angular acceleration is [1]

$$\tau = \dot{m}\,(r_1 \cdot v_{\theta 1} - r_2 \cdot v_{\theta 2}) \tag{1}$$

The rate of energy transfer (N-m/sec or ft-lb_f/sec) is the product of the torque and the angular velocity (ω)

$$\tau \cdot \omega = \dot{m}\,(r_1 \cdot \omega \cdot v_{\theta 1} - r_2 \cdot \omega \cdot v_{\theta 2}) \tag{2}$$

We can also write the steady state, steady-flow energy equation for the control volume that contains the compressor rotor [2]:

$$\begin{aligned} &\dot{Q}_{c.v.} + \dot{m}\left(h_1 + \frac{v_1^2}{2 \cdot g_c} + z_1 \frac{g}{g_c}\right) \\ &= \dot{m}\left(h_2 + \frac{v_2^2}{2 \cdot g_c} + z_2 \frac{g}{g_c}\right) + \dot{W}_{c.v.} \end{aligned} \tag{3}$$

where

$\dot{Q}_{c.v.}$ = heat added to the fluid in the control volume
$\dot{W}_{c.v.}$ = work done by the fluid in the control volume.
h = fluid specific enthalpy
z = elevation
g = gravitational acceleration
g_c = units conversion

The stagnation, or total, enthalpy is defined as

$$h^T = h + \frac{v^2}{2 \cdot g_c} \tag{4}$$

Assume that the azimuthal angular acceleration associated with the torque in equation (1) is isentropic and neglect changes in potential energy. Combining equations (2) and (3),

$$\dot{W}_{c.v.} = \tau_s \cdot \omega = \dot{m}\left(h_1^T - h_{2'}^T\right) \tag{5}$$

where s denotes that the process is isentropic, and $h_{2'}^T$ refers to the isentropic total enthalpy at State 2. Equation (5) represents the isentropic work performed on the fluid by accelerating the fluid in the tangential direction. State 1, at the inlet of the compressor, is given by the upstream conditions. The entropy s at State 1 can be found as a function of total enthalpy and density.

$$s_1 = s\left(h_1^T, \rho_1^T\right) \tag{6}$$

The total pressure P^T at State 2 is determined from the tables[b] of pressure ratio versus rotational velocity and flow rate. Pressure ratio R_P is defined as

$$R_P = R_P(\omega, \dot{m}) = \frac{P_2^T}{P_1^T} \tag{7}$$

Therefore, P_2^T is

$$P_2^T = P_1^T \cdot R_P \tag{8}$$

Because the work between States 1 and 2 is isentropic, the conditions at State 2 are determined as follows:

$$s_2 = s_1 \tag{9}$$

$$h_{2'}^T = h\,(P_2^T, s_2) \tag{10}$$

and the torque corresponding to the isentropic work is

$$\tau_s = \frac{\dot{m}}{\omega}\left(h_{2'}^T - h_1^T\right). \tag{11}$$

Note the sign is reversed compared to equation (5), which indicates that work is done on the fluid. This is consistent with the sign convention within the RELAP5-3D© pump component. The adiabatic efficiency η_{ad} is defined as

$$\eta_{ad} = \frac{\text{Isentropic work}}{\text{Actual work}} = \frac{h_{2'}^T - h_1^T}{h_2^T - h_1^T} \tag{12}$$

from which we can obtain the total enthalpy of the actual state

[b] Performance data applicable to a specific compressor design. Figure 1 is an example of a compressor performance map.

$$h_2^T = \frac{h_{2'}^T - (1-\eta_{ad})\, h_1^T}{\eta_{ad}} . \tag{13}$$

The irreversible, or dissipative, torque is

$$\tau_d = \frac{\dot{m}}{\omega}\left(h_2^T - h_{2'}^T\right). \tag{14}$$

The compressor dissipation is the energy associated with the work due to friction,

$$\dot{W}_d = \tau_d \cdot \omega , \tag{15}$$

and is added to the energy equation in the same manner as the corresponding pump dissipation is added. The above derivation assumes isentropic compression to obtain the conditions at State 2, and requires entropy-based property table look-up calls, which are not presently in the code. However, torque can be calculated from the work input to the fluid without the isentropic condition if an assumption is made regarding the conditions at State 2. The momentum equation for the compressor can be written in the form [3]

$$\int_{P_1}^{P_2} \frac{dP}{\rho} + \frac{v_2^2 - v_1^2}{2} = g \cdot H \tag{16}$$

where P is fluid pressure, ρ is fluid density, and H is the head added to the fluid by the compressor. If a mean density is assumed that is independent of pressure, $(\rho_m = \frac{\rho_1 + \rho_2}{2})$, Equation (16) can be integrated

$$P_2 + \frac{\rho_m v_2^2}{2} = P_1 + \frac{\rho_m v_1^2}{2} + \rho_m \cdot g \cdot H \tag{17}$$

and can be written in terms of total pressure

$$P_2^T = P_1^T + \rho_m \cdot g \cdot H = P_1^T + P_1^T (R_P - 1) \tag{18}$$

where the pressure ratio is obtained as shown by equation (7). The compressor head that is added to the momentum equation is therefore

$$\Delta P = \rho \cdot g \cdot H = P_1^T (R_P - 1) = P_2^T \frac{R_P - 1}{R_P} . \tag{19}$$

By assumption, the heat into the system is zero, so the total work input to the fluid is given by

$$\begin{aligned} \dot{W}_{c.v.} &= \dot{m}\,(h_2^T - h_1^T) \\ &= \dot{m}\,(h_{2'}^T - h_1^T) + \dot{m}\,(h_2^T - h_{2'}^T) \\ &= \dot{W}_s + \dot{W}_d \end{aligned} \tag{20}$$

again using the sign convention where work on the fluid is positive. The isentropic work is calculated from the increase in potential energy in the control volume

$$\dot{W}_s = \dot{m} \cdot g \cdot H = \frac{\dot{m} \cdot \Delta P}{\rho_m} . \tag{21}$$

The torque corresponding to the isentropic work is obtained by combining Equations (11), (19), and (21):

$$\tau_s = \frac{\dot{m}}{\omega}\left(h_{2'}^T - h_1^T\right) = \frac{\dot{m}}{\omega}\,\frac{P_1^T (R_P - 1)}{\rho_m} \tag{22}$$

and the irreversible, or dissipative, torque is obtained by substituting Equation (13) into Equation (14) and combining with Equation (22)

$$\begin{aligned} \tau_d &= \frac{\dot{m}}{\omega}\,\frac{1-\eta_{ad}}{\eta_{ad}}\left(h_{2'}^T - h_1^T\right) \\ &= \frac{\dot{m}}{\omega}\,\frac{1-\eta_{ad}}{\eta_{ad}}\,\frac{P_1^T (R_P - 1)}{\rho_m} . \end{aligned} \tag{23}$$

The total torque is the sum of the isentropic torque and the dissipative torque

$$\begin{aligned} \tau_t = \tau_s + \tau_d &= \frac{\dot{m}}{\omega}\,\frac{1}{\eta_{ad}}\left(h_{2'}^T - h_1^T\right) \\ &= \frac{\dot{m}}{\omega}\,\frac{P_1^T (R_P - 1)}{\rho_m \eta_{ad}} = \frac{\dot{m}}{\omega}\,\frac{(P_2 - P_1)}{\rho_m \eta_{ad}} \end{aligned} \tag{24}$$

and can be input to the shaft rotational velocity equation.

As with the pump model, a quasi-static model for compressor performance is imposed on the RELAP5-3D© volume-junction flow path representation. The compressor is a volume-oriented component, and the head developed by the compressor is added to the junction that connects the suction of the compressor volume to the system. The compressor model is interfaced with the two-fluid hydrodynamic model by assuming the head developed by the compressor is similar to a body force. Thus, the head term appears in the mixture momentum equation; but, like the gravity body force, it does not appear in the difference momentum equation used in RELAP5-3D©. The head term is added to both the liquid and vapor phase terms of the mixture momentum equation although it is recognized that the compressor will only be operated in a system containing single-phase gas. The term that is added to the mixture momentum equation is $\rho_m gH$, where H is the total head rise of the pump (m), ρ_m is the volume fluid density (kg/m^3), and g is the acceleration due to gravity (m/s^2).

III. IMPLEMENTATION IN RELAP5-3D©

In both the semi-implicit and nearly-implicit numerical algorithms, the compressor head H is coupled implicitly to the velocities through its dependence on the volumetric flow rate, Q. The volumetric flow rate is defined as the volume mass flow rate divided by the volume density. It is assumed that the head depends on the volumetric flow rate, and can be approximated by a two-term Taylor series expansion given by

$$H^{n+1} = H^n + \left(\frac{dH}{dQ}\right)^n (Q^{n+1} - Q^n) \tag{25}$$

Thus, the numerical equivalent of the term ρgH in both algorithms is

$$\rho^n gH^n \Delta t + \rho^n g\left(\frac{dH}{dQ}\right)^n (Q^{n+1} - Q^n)\Delta t\,. \tag{26}$$

This term is added to the right side of the mixture momentum Equation (3.1-103) of Reference 4, Section 3.1.1.

The compressor energy dissipation, $\dot{Q}$, is calculated for the compressor volume in a manner similar to that derived for the RELAP5-3D© turbine, which is given by Equation 3.5-86 of Reference 4, Section 3.5.5 . Because stagnation pressures are given in Equation (7), the kinetic energy term is neglected and the equation becomes

$$\dot{Q} = \dot{m}\left(\frac{P_1 - P_2}{\rho_1}\right) - \dot{W} \tag{27}$$

Computing the rate of work from Equation (20) and accounting for the different sign convention yields

$$\begin{aligned}\dot{Q} &= \frac{\dot{m}(P_2 - P_1)}{\rho_m \eta_{ad}}\left(1 - \frac{\eta_{ad}\rho_m}{\rho_1}\right)\\ &= \tau_t \omega\left(1 - \frac{\eta_{ad}\rho_m}{\rho_1}\right)\end{aligned} \tag{28}$$

If the fluid is assumed to be incompressible, the density is constant and the compressor dissipation is consistent with that obtained for the pump in Equation (3.5-62) of Reference 4, Section 3.5.4.

The energy dissipation term is evaluated explicitly in both the semi-implicit and nearly-explicit algorithms, and it is partitioned between the liquid and vapor thermal energy equations in such a way that the rise in temperature due to dissipation is equal in each phase. (The details of the dissipation mechanism in a two-phase system are unknown, so the assumption is made that the mechanism acts in such a way that thermal equilibrium between the phases is maintained without phase change.) Thus, the energy dissipation function (DISS) terms that are added to the right sides of the liquid and vapor/gas thermal energy equations, Equations (3.1-90) and (3.1-91) of Reference 4, Section 3.1.3 for the compressor volumes are

$$\begin{aligned}DISS_f &= \tau^n \omega^n (1 - \eta_{ad}\frac{\rho_m}{\rho_1})\Delta t\\ &\cdot\left(\frac{\alpha_f^n \rho_f^n C_{pf}^n}{\alpha_f^n \rho_f^n C_{pf}^n + \alpha_g^n \rho_g^n C_{pg}^n}\right)\end{aligned} \tag{29}$$

$$\begin{aligned}DISS_g &= \tau^n \omega^n (1 - \eta_{ad}\frac{\rho_m}{\rho_1})\Delta t\\ &\cdot\left(\frac{\alpha_g^n \rho_g^n C_{pg}^n}{\alpha_f^n \rho_f^n C_{pf}^n + \alpha_g^n \rho_g^n C_{pg}^n}\right)\end{aligned} \tag{30}$$

where α denotes phasic volumetric fraction and C_p is constant pressure specific heat capacity.

The compressor head H is defined by Equation (19), and torque τ is defined by Equation (24). Values are obtained by means of an empirical compressor performance map, input to the code with the independent variables of mass flow rate $\dot{m}$ and speed N and the dependent variables of pressure ratio R_P and efficiency η.

IV. INPUT CONSIDERATIONS

IV.A. Dimensionless Variable Input

The mass flow rate and speed tables should be entered as relative corrected values. The general forms of independent dimensionless parameters are

$$\dot{m}_C = \frac{\dot{m}}{\rho_{0,in} a_{0,in} D^2} \tag{31}$$

(corrected mass flow) and

$$N_C = \frac{ND}{a_{0,in}} \tag{32}$$

(corrected speed), where subscript '0' designates stagnation and 'in' designates the inlet station. 'a' is sound speed, and 'D' represents the turbomachine size [5]. The relative corrected mass flow rate v is given as

$$v = \frac{\dot{m}}{\rho_{0,in} a_{0,in}} \bigg/ \left(\frac{\dot{m}}{\rho_{0,in} a_{0,in}} \right)_{Rated} \tag{33}$$

and the relative corrected speed α is given as

$$\alpha = \frac{N}{a_{0,in}} \bigg/ \left(\frac{N}{a_{0,in}} \right)_{Rated} . \tag{34}$$

where D cancels from the numerator and the denominator. Equations (33) and (34) should be used to correct mass flow rate and speed entries in the performance table.

IV.B. Ideal Gas Considerations

If the fluid is an ideal gas, then the equations for corrected mass flow and speed are reduced to functions of pressure and temperature:

$$\begin{aligned} \dot{m}_C &= \frac{\dot{m}}{\rho_{0,in} a_{0,in}} \\ &= \frac{RT_{0,in}}{P_{0,in}} \frac{\dot{m}}{\sqrt{\gamma RT_{0,in}}} \\ &= \frac{\dot{m}\sqrt{RT_{0,in}}}{P_{0,in}\sqrt{\gamma}} \end{aligned} \tag{35}$$

and

$$N_C = \frac{N}{a_{0,in}} = \frac{N}{\sqrt{\gamma RT_{0,in}}} . \tag{36}$$

where R is the gas constant, γ is the specific heat ratio C_p/C_v, and T_0 is fluid stagnation temperature. If the fluid species does not change, R and γ can be omitted leaving

$$\dot{m}_C = \frac{\dot{m}\sqrt{T_{0,in}}}{P_{0,in}} \tag{37}$$

and

$$N_C = \frac{N}{\sqrt{T_{0,in}}} , \tag{38}$$

respectively. It is most general to require input of rated fluid density and rated sonic speed as the input variables. Therefore, if the ideal gas form is desired, it should be converted to rated fluid density and rated sonic speed for input to the compressor model.

IV.C. Efficiency Input Correction

The dissipative torque given by Equation (23) is based upon the assumption that the enthalpy rise is given in terms of the total pressure at the compressor suction P_1^T, the pressure ration R_P, and an average (constant) density, ρ_m, as follows:

$$h_{2'}^T - h_1^T = \frac{P_1^T (R_P - 1)}{\rho_m} \tag{39}$$

It has been observed that the assumption of constant density introduces a small error, and that a correction to the efficiency input table improves the accuracy of the result. To obtain the form of the correction, rewrite the equation for total compressor torque, Equation (24), using the assumption that a corrected efficiency is necessary to satisfy the equation, and with the definition of pressure ratio from Equation (19), that is

$$\begin{aligned} \tau_t &= \frac{\dot{m}}{\omega} \frac{1}{\eta_{ad}} \left(h_{2'}^T - h_1^T \right) \\ &= \frac{\dot{m}}{\omega} \frac{P_1^T (R_P - 1)}{\rho_m \eta_{corr}} \\ &= \frac{\dot{m}}{\omega} \frac{g \cdot H}{\eta_{corr}} \end{aligned} \tag{40}$$

Solving for η_{corr} gives

$$\eta_{corr} = \eta_{ad} \frac{g \cdot H}{h_{2'}^T - h_1^T} \tag{41}$$

IV.D. Low Flow Consideration

Pressure ratio and adiabatic efficiency both decrease as flowrate decreases to near zero. If, at conditions near zero flow, the pressure ratio approaches 1 as the efficiency approaches 0, then the equation for compressor total torque,

$$\tau_T = \frac{\dot{m}}{\omega} \frac{P_1^T}{\rho_m} \frac{R_P - 1}{\eta_{ad}} , \tag{42}$$

becomes indeterminate. The limit of the ratio, as both numerator and denominator approach zero, is the same as the limit of the ratio of the derivatives (l'Hopital's rule). That is

$$\lim_{R_P \to 1, \eta_{ad} \to 0} \frac{R_P - 1}{\eta_{ad}} = \lim_{R_P \to 1, \eta_{ad} \to 0} \frac{d(R_P - 1)}{d(\eta_{ad})} . \quad (43)$$

This calculation scheme is implemented in the RELAP5-3D© code for values of $\eta_{ad} \leq 1x10^{-10}$. Therefore, the user must chose the values of R_P and η_{ad} very carefully in the region where η_{ad} is close to zero to ensure that the derivative represents the physics of the model.

IV.E. Performance Modeling

Interaction of the compressor and the fluid is described by empirically developed performance maps relating compressor pressure ratio and efficiency to the mass flow rate and compressor angular velocity. Figure 1 is an example of a compressor performance map. The performance map is an xy representation that has a mass flow parameter as the abscissa and a head-related parameter (in this case pressure ratio) as the ordinate. Lines of constant speed and lines of constant efficiency are represented parametrically on the compressor performance map.

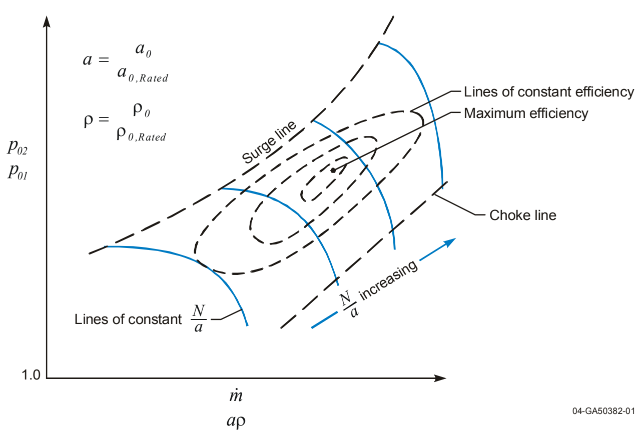

Figure 1. Compressor Performance Map.

The operating range of a compressor is the region between the surge line and the choke line. Compressor surge is characterized by some aerodynamic instability that is caused by aerodynamic stall occurring either in the impeller or in the diffuser. This results in a flow direction reversal for a short time interval, during which time the backpressure drops, and the flow resumes its proper direction. The reversal of flow also causes a reversal in the direction of forces, especially thrust forces, which can damage or destroy the compressor. The compressor model in RELAP5-3D© does not impose a limitation within the coding to prevent operation beyond (i.e. to the left and above) the surge line. However, because the user input will not generally extend past the surge line, operation of the model in this region should be precluded. Choking is a condition of maximum mass flow rate through a compressor because of sonic flow at a minimum flow area point of the compressor. The user should be aware that the actual flow areas within the compressor blade region are not represented in the model, and therefore the code has no information that would permit the prediction of choking at the choke line. As a result, the user should specify the the compressor performance map input to ensure that operation beyond the choke line does not occur.

Compressor performance maps are read in as series of tables, one for each value of angular velocity. Each table consists of number triplets of mass flow rate, pressure ratio, and efficiency. Angular velocity and mass flow rate are entered as relative values, corrected to rated stagnation sound speed and density, as described in Section IV.A. The tables are read using simple linear interpolation using the angular velocity and mass flow parameters as independent variables. The potential exists for interpolation errors when using this method. Errors in the efficiency values are especially prone to interpolation errors, because of the nonlinearity in the adiabatic efficiency curves. As shown in Figure 1, the lines of constant adiabatic efficiency are "islands" having roughly elliptical shapes on the performance map when a mass flow parameter is the variable of the abscissa and a head-related parameter is the variable of the ordinate. Because of the potential for interpolation errors, the user should supply intermediate lines of constant corrected angular velocity as necessary to obtain acceptable interpolation accuracy.

V. VERIFICATION TESTING

The compressor model was assessed by comparing results of the RELAP5-3D© compressor model to the results of design calculations performed at the Massachusetts Institute of Technology (MIT), which were done to support the development of a supercritical carbon dioxide cycle in a gas-cooled fast reactor (GFR). The supercritical carbon dioxide cycle [6] utilizes two compressors, a main compressor and a recompressing compressor. The design data presented here apply to the recompressing compressor and were obtained from Hejzlar [7].

A separate-effects model of the recompressing compressor in the GFR is illustrated in Figure 2. Component 350 represents the compressor. Components 345, 346, and 380 represent plena adjacent to the compressor in the GFR. The flow area of Component 346 was large enough to obtain stagnation conditions at the inlet to the compressor. Boundary conditions were applied using time-dependent volumes (Components 341 and 382). The pressure and temperature in Component 341 were set at rated conditions and held constant. A series of steady-state calculations was performed to investigate the performance of the compressor for a range of flows and speeds. The compressor speed and discharge pressure were held constant at their rated values until the system reached steady state. A step change was then made in the pressure of Component 382, and the pressure was held constant until a new steady state was obtained. This process was continued until the entire range of allowed compressor operation at the rated speed was covered. The compressor speed was then varied and the process repeated until the allowed range of operation was simulated for the new speed. Three compressor speeds, corresponding to 50%, 80%, and 100% of the rated speed, were simulated. Because the conditions at the inlet to the compressor were held constant at their rated values, the three speeds correspond to relative corrected speeds of 0.5, 0.8, and 1.0.

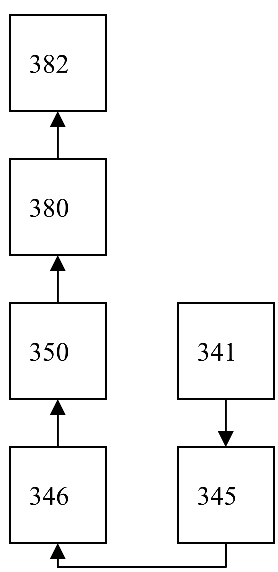

Figure 2. Model of the recompressing compressor.

Figure 3 compares the results of the steady-state RELAP5-3D© calculations to those of the MIT design calculations for the pressure ratio developed by the compressor as a function of relative corrected flow. MIT design calculations were performed for relative corrected speeds varying between 0.40 and 1.0, while, as mentioned previously, RELAP5-3D© calculations were performed for relative speeds corrected speeds of 0.5, 0.8, and 1.0. The figure also shows lines of constant efficiency, η, as well as the surge and choke lines predicted by MIT.

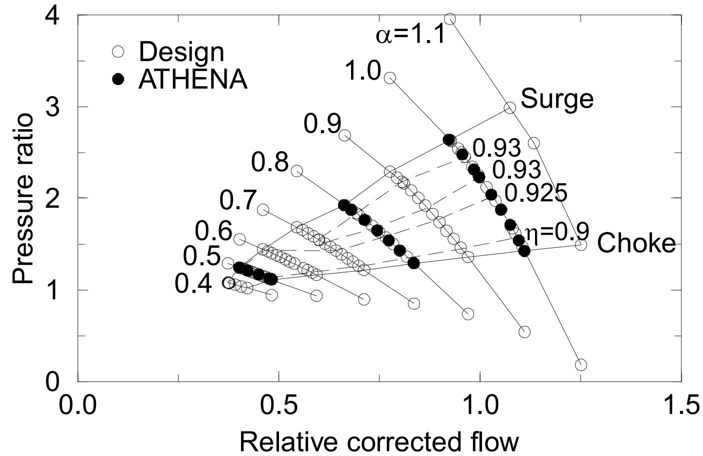

Figure 3. Pressure ratio developed by the recompressing compressor.

Compressor operation is allowed only within the relatively narrow band between the surge and choke lines. Pressure waves may be generated that might damage the compressor or other components if operation is attempted beyond the surge line. The stagnation conditions at the inlet to the compressor were held fixed at the rated conditions of 9.08 MPa and 363 K during both the design and RELAP5-3D© calculations.

The RELAP5-3D© compressor model requires that performance curves be available at relative corrected speeds and flows below and above the current operating condition so that an interpolation between adjacent curves can be performed. Consequently, the MIT results were extrapolated to a relative corrected speed of 1.1 to allow operation at the normal speed of 1.0. The performance curve at a given relative corrected speed was also extrapolated to the relative corrected flows corresponding to surge and/or choke lines of the adjacent speed curves so that the entire range of allowed operation could be simulated. The results of the calculations were monitored using control variables to determine if operation outside the surge and choke lines was attempted. The RELAP5-3D© separate-effects model of the recompressing compressor simulated the allowed range of operation at three different relative speed curves, corresponding to values of 1.0, 0.8, and 0.5, in a series of steady-state calculations.

The power consumed by the compressor for three different relative corrected speeds are shown in Figure 4 for both the design and the RELAP5-3D© calculations. As shown in Figures 3 and 4, the results calculated by RELAP5-3D© were in excellent agreement with the design calculations from MIT for both pressure ratio and power.

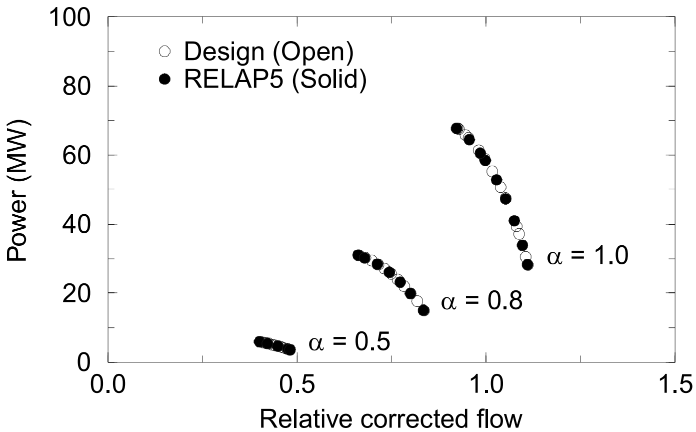

Figure 4. Power consumed by the recompressing compressor

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