Isentropic turbine efficiency calculation

 Isentropic efficiency is considered constant, at 83% for compressors and 90% for gas and STs. The exhaust temperature at the output of hrSG is maintained above the condensation temperature of combustion products. From: Exergy (Second Edition), 2013Osamah Siddiqui, Ibrahim Dinser, to Exergetic, Energy and Environmental Dimensions, 2018Inetropic turbine efficiency affects the overall performance of the system. Fig. 5 shows the effect of the isentropic efficiency of turbine 1 and turbine 2 on the overall energy of the system and the efficiency of exergy. The energy efficiency of the entire system varies from 26.6% to 34.1%, as the efficient efficiency of turbines ranges from 50% to 90%. In addition, the efficiency of exergy of the common system increases from 24.2% to 39.9%, as the efficient efficiency of turbines increases from 50% to 90%. As the turbine's efficientness increases, it is irreversible. Consequently, a turbine with higher isentropic efficiency is capable of producing more energy. This leads to an increase in the overall efficiency of the system. Figure 5. The effect of turbine isentropic efficiency on the overall system energy and efficiency of exergy.P. Ginies, ... D. Gross, at the 7th International Conference on Compressors and Their Systems 2011, 2011 Relative Isentropic Efficiency (1) gives an image of the compressor behavior over the operating card. The relative isentropic efficiency was drawn compared to the APR for a compressor with two IDV holes and for an IDV-free compressor. These comparative curves were drawn for a number of saturated condensation temperatures: 35/45/55/65 degrees Celsius. The horizontal APR axis allows you to find the evaporation temperature for each case of temperature condensation. (1) Relative isentropic efficiency is an isentropic efficiency multiplied by a factor. Charts 3, 4, 5, 6 show that the advantage in isenttropic efficiency is greater for the ASIA-Pacific region of less than 2.0, and that there is a small disadvantage for the APR greater than 3.0. The advantage with weight in terms of applications, which is more than the weight disadvantage observed for a high compression ratio, corresponding to exceptional operating conditions.A.M.Y. Razak, in modern gas turbine systems, 2013Azentropic efficiency of the compression process is defined as the ratio of ideal work done to the actual work done. Thus, the isentropic efficiency is given: 11.13c'cp (T'2'T1)cp (T2'T1), where T2 is the ideal exit temperature of the compressor. T1 is the temperature of the input compressor and CP specific heat at constant pressure -- we will consider constant constant constant heat, and the effect of changing specific heats will be discussed later in the chapter. The equation can be presented in terms of compressor pressure ratio as: where Copt-pry-1 and Prc is the compressor pressure ratio. Therefore, the temperature of the compressor output can be determined by: For the process of expanding isentropic efficiency (TT) is given: 11.16t'cp (T3'T4)cp (T3'T'4), where T4 is the ideal extender/exit temperature turbine, T3 is an extender/temperature input turbine and CP is a specific heat at constant pressure - again we will assume the constant specific heat and effect of specific heat will be discussed later. Thus, as the compressor isentropic analysis above, you can show that the expansion/temperature output of the turbine is given: where c'Prt and Prt is the expansion/pressure ratio of the turbine.A.M.Y. Razak, in industrial gas turbines, the 2007 third method determines the performance of the gas turbine using enthalpies and entropies at various points. It is considered to be the most accurate method of calculating the design point of the gas turbine. The method is much more detailed and is usually carried out with the help of a computer program developed for this purpose. However, the processes involved will be outlined. By integrating equation 2.44, which describes a change in specific heat with air temperature and combustion products, equations can be developed for entalpic and entropy. Thus: 2.65 HHA (TST0) BT2-T022-s (1T1T0) 2.66SaITT0-b (TST0)-c2 (1T2-1T02) - RlnPP0, where T and P are temperature and pressure of air or gas, respectively, and T0 and P0 are reference temperature and pressure when enthalpy and entropy are respectively considered zero when the temperature and pressure are 273 K and 1.013 Bar-A, respectively. Constants A, b and c are defined as follows: a'∑i'1nocai×mfib∑i'1nocbi×mfic'∑i'1cci×mfiai, bi and ci - are constants defined in table 2.1 for each component, and noc is the number of components in the air or fire. In the example, the pressure and temperature at the compressor entrance is 1,013 bar and 288 K. From equations 2.65 and 2.66, we calculate enthalpy and entropy at the compressor input as:H1'14.876 kJ/kgS1'053 kJ/kg K. For compressor pressure ratio 20, compressor discharge pressure, P2 20.26 Bar-A. From equation 2.66, you can determine the discharge temperature of the esanthropic compressor. This is achieved by using P2 for the term pressure in Equation 2.66 and temperature changes so far, until entropy is 0.053 kJ/kg K. Isentropic compressor temperature discharge, T2 , works up: Using this value in the equation 2.65, enthalpy on the compressor discharge, H2 due to the isentropic compression received: Isentropic efficiency Equation 2.28 for the compression process can be written in terms of enthalpies as: The compressor that corresponds: Using the value for H2 in equation 2.65, the actual discharge temperature of the compressor, T2, can be determined implicitly: compressor-specific work: Wc No H2 and H1. Thus: The ratio of fuel to air can now be calculated similarly to those discussed in Method 2. The temperature of the insertion and the rise in the temperature of the horistor in this case is 702.86 K and 697.14 K, respectively. The theoretical ratio of fuel to air, f, 0.0195. The actual fuel-to-air ratio is fa 0.0195/0.99 and 0.0197. Thermal input cin: zin-0.0197×43100-849,388 kJ/kgSy fuel is used by kerosene and can be modeled as C12H24. Knowing the fuel—air ratio and the air composition, the composition of the products of combustion can be calculated, as described by Goodger.13[2.67]CxHy+m(O2+0.78090.2095N2+0.00930.2095Ar+0.00030.2095CO2)=n1CO2+n2H2O+n3N2+n4Ar+n5O2The quantities 0.7809, 0.0093, 0.003 and 0.2095 are the volume-fractions or molar-fractions (mole-fraction) of N2, Ar, CO2 and O2 in air, respectively, and n1, n2, n3, n4 and n5 are the mole-fraction of CO2, H2O, N2, Ar and O2 in the products of combustion, respectively. The terms x and y are mole fractions of carbon and hydrogen in fuel. For kerosene, x No. 12 and 24 and the term m is excess air, which is determined by using the fuel-to-air ratio (f) as follows: fa'12.01x-1.008y (1'0.78090.209 5'0.00930 .2095-0.00030.2095) MWwhere MW is a mole-weight air and factors 12.01 and 1,008 are atomic weights of carbon and hydrogen, Accordingly. By performing moly balance with Equation 2.67, the mole-fraction of combustion products (n1, n2, n3, n4, and n5) can be defined in the same way as discussed in Chapter 6 (section 6.18.4). Since the turbine entry temperature, T3, pressure, P3, and combustion gas composition are now known, equations 2.65 and 2.66 can be used to determine enthalpy, H3 and entropy, S3 when entering the turbine. It is necessary to determine the talp at the exit because of the isentropic expansion. This is achieved by using Equation 2.66 and changing the temperature of the turbine's output, T4, as long as entropy is equal to the value determined at the turbine input, the S3. From the equation 2.65 you can identify the enthalpy, H4 at the exit of the turbine due to the isentropic expansion. A turbine of isentropic efficiency in the equation 2.30 can be presented as: where the H4 is the actual enthalpy on the way out of the turbine. Values for H3, S3 and H4' 1272,995 kJ/kg, 0.958 kJ/kg and 428,005 kJ/kg respectively. For a turbine of isentropic efficiency of 0.9, the actual enthalpy when exiting the turbine is 512,504 kJ/kg and entropy at the exit of the turbine is 1.0768 kJ/kg. Turbine Specific Work, Wt, is: Wt'H3-H4'1272.995-512.504'760.491 kJ/kgNet specific work (Wnet) from gas turbine: Wnet-Wc-Wt-760.491-44 5.3'315.191 kJ/kgGrable efficiency (th) is: ht-Edtsina 315.191849.388'0.3711.Specific at the main points 1, 2, 3 and 4, as shown in the pic. 2.29, correspond 1.0011, 1.083, 1.2193 and and Accordingly. The corresponding values of the ratio of specific heat points, γ KP/q, at the main points 1, 2, 3 and 4 1,402, 1.3607, 1.3082 and 1,345, respectively. The increase in CP due to compression is associated with an increase in temperature, as described by the equation 2.44. Similarly, there is an increase in CP at point 3 and a decrease in point 4. However, the increase in CP in point 3 is also associated with an increase in water vapor in combustion products, which is significant as seen in Table 2.3. Also, pay attention to the increase in CO2 content in combustion products, greenhouse gases and is considered responsible for global warming. Thus, gas turbines powered by fuels such as natural gas or methane, which have a higher hydrogen content, will increase specific work due to the high water vapor content in combustion products. With methane as fuel, this increase in power may be as high as 2% compared to this when using kerosene. Note that an increase in specific heats has resulted in a decrease in γ.2.29. Turbine cycle on the temperature entropy chart. Table 2.3. The composition of combustion componentsComponentGraimetric or Mass FactionN20.744O20.162Ar0.009CO20.061H2O0.025Inth of example is considered dry air. The effects of humidity can also be included in the analysis. For example, given the relative humidity of the air, a specific humidity can be calculated, as discussed in section 2.11.1, which is the mass of water vapor per unit of dry air. Thus, specific humidity can be added to the air composition, as shown in table 2.2, and the composition of air/gas is normalized to determine the gravimetric composition of moist/wet air, and then repeat the above procedure. The additional heat input required to heat the water vapor from the compressor discharge temperature, T2, to the turbine entry temperature, T3, should be calculated. This can be determined by the equation 2.68: 2.68'Hs'2.232Ts-2352.623where Hs is water/vapor enthalpy (kJ/kg) and Ts is a water vapor/steam temperature in Celsius.S.L. Dixon B. Eng. Ph.D., C.A. Hall Ph.D., in the field of liquid mechanics and thermodynamics of turbomachine (Sixth edition), 2010Inetropical efficiency, CX, compressor or hydraulic pump efficiency, qh, generally defined as 'c (or'x) 'useful (hydrodynamic) energy input into the liquid unit in the time of input into the rotorEnergy input energy into the rotor (or impeller) is always less than the power supplied when connected due to external energy loss in bearings, glands, etc. Thus, the mechanical efficiency for the complete adiabaic compression process comes from state 1 to state 2, the specific input of the work IsFigur 1.9 (b) shows the Mollier chart, on the actual compression process is represented by a change in the state of 1-2 and Perfect process at 1-2s. For an adiabatic compressor in which potential energy changes are insignificant, the most significant efficiency is total to full efficiency, which can be written as (1.46a)'ideal (minimum) input work 'h02s'h01h02-h01. If the difference between input and kinetic energy outlets is small, 12c12≅12c22, Then for the irrepressible flow, the minimum entry of work is given to WminW ̇min/m ̇ (p2'p1)/12 (c22-c12)g (z2'z1) Thus, for the pump hydraulic efficiency is defined as (1.47)'w.minW.c'H2-H1'Wc.M. C.J. Deschamps, at the 8th International Conference on Compressors and Their Systems, 2013 The positive and isentropic efficiency of scrolling compressors depends on the transmission of heat that occurs inside pockets during suction and compression processes. This paper details a numerical model designed to predict heat transfer and distribution of the temperature of the scrolls. The model was developed by the end volume method and is connected to a thermodynamic model of the compression cycle. The results showed that the discharge temperature, projected with a solution of thermal conduction through scroll wraps, was slightly lower than when the linear temperature profile was assigned. It has also been found that heat transmission occurs in metallic contact between wrap scrolls acts to produce a linear change in temperature along its length. Philip Thomas, in the simulation of industrial processes for control engineers, 1999O the definition of isentropic efficiency, the equation (17.64), is based on the ratio of isentropic specific works of actual specific work, through the full section of the compressor. However, it is also possible to determine the differential efficiency assumed by the constant over the section, known as polytropical efficiency, qp, given We can use the equation (17.58) to convert the equation (17.72) into: (17.73)'p'dhsdh'cpdTscpdT'dTsdT'dtwhere we also assumed that specific heat is constant. The form of the equation (17.73) allows us to use the mathematical treatment outlined in section 14.3 for nozzle efficiency, except that the nozzle efficiency used in this section will be replaced by reverse polytropic efficiency, 1/p. Thus, actual compression can be characterized by where the exhibitor, m, is given: As a consequence of the equation (17.74) and the characteristic gas equation (3.2), the actual ratio of temperature in the compressor section, working on polytropic efficiency, will be given (17.76)T2T0 (p2p0)(m'0 1)/mwhere, from the definition of the exhibitor, m, in the equation (17.75)Expression for the ratio of temperature in actual compression can be found by replacing in the equation (17.61) to give the actual concrete work:(17.78) We can use this new expression for actual concrete work together with the equation (17.63), which gives out a factual concrete work: (17.78) We can use this new expression for actual concrete work together with the equation (17.63), which gives out a concrete work Efficiency: (17.79)'s'1'RwT0 (p2p0) (γ-1)/γ-1) (p2p0) (γ-1 (γ γ-1)/γ)-1) and it really is. However, it is possible to estimate the equation (17.79) by the range of pressure ratios for fixed polytropic efficiency values, and thus distinguish discrepancies. Figure 17.6 shows isentropic efficiency, designed for three typical polytropic efficiency values for a number of pressure ratios. Figure 17.6. The eschee-troc efficiency relative to the polytropic efficiency ratio of 0.7, 0.8 and 0.9, with a γ and 1.4.It ratio, will show that the isentropic efficiency of the section is the same as polytropic efficiency at the level of pressure of unity, but falls off as the pressure factor increases, falling, being more noticeable at a lower polytropic efficiency. Isentropic efficiency is between 2% and 7% less than polytropic efficiency, depending on the value of the latter, for the normal range of pressures found in industrial plants, namely 2.5 to 4.5.The fact that the efficiency of isentropic varies with the ratio of pressure for the constant value of polytropic efficiency has led some to consider polytropic efficiency as the preferred basis on which to base their analysis of the compressor section. Instead of isentropical concrete work as our ideal by which to measure the actual specific work, one can develop a new ideal measure, polytropic specific work, -Wp, is defined so that its relationship to the actual specific work of polytropic efficiency, P: The combination of equations (17.78) and (17.80), polytropical specific work arises as: (17,81)-wp'p'1'RV0 (p2p0)(m-1)/m/1) or using equation (17.77)(17,82) mm1'0W (p2p0) (m-1)/m/1) Experimental definition of m can be made after measuring pressure and temperature at the entrance section, p0,T0 and socket, p2,T2 and applying the equation (17.76), which can be decided to give the formula:(17.83)m'Inp2p0Inp2p2p2p20 Polytropic be found in a similar way by solving equations (17.76) and (17, 77) :(17.84) , the experimental definition of the exhibitor, m, using the equation (17.83) (or equivalent) will introduce a small error due to the implicit inclusion of an imperfect model of compression effects. To compensate for this, manufacturers sometimes inject an additional factor, f, into the equation for polytropic-specific work, although in most cases f is so close to unity to make minor differences. Polytropic specific work is usually given the name Polytropic Head, Hp, based on the same reasoning used for the isentropic head, and so we have the final form: (17.85)Hp'fmm'1'RwT0 (p2p0) (m-1)/m-1) power absorbed by the compressor section is given (17.68). We can use the equation (17.80) to express strength in terms of polytropic specific work/ polytropic head head Polytropical Efficiency:where the final form uses the fact that Hp and th wp. It should be emphasized that polytropic head is idealization just as the isentropic head was idealized. Along with polytropic efficiency, however, it provides a way to analyze the performance of the compressor, as will be shown in the next section. Wen W. Chang, ... Tsing L. Chen, in Computer Aid Chemical Engineering, 2018For isentropic compression, isentropic efficiency is determined to calculate practical changes enthalpy. Thus, the required power is calculated using Eq. (4). (4) Eh,u,u∑yukumh, u, cFh, uRh,uθu-1θu∀u∈COMwhere, variable Eh, u is energy consumption or generated, M is a mole fraction of component C, isentropic exhibitor θu is the ratio of thermal capacity of gas flows at constant pressure and temperature, KK is a specific thermal power, and with is a set of chemical components.P. Friedman, M. Anderson, in The Basics and Application of Supercritical Carbon Dioxide (sCO2) Based on Power Cycles , 2017Energo- and energy balances for the turbine (Figure 3.6) are,Figure 3.6. Adiabactic turbine analysis. Entrance and weekend streams carry a stream of enthalpy and exergy. Specific works are left through the turbine shaft. Destroyed exergies are shown as irreversibility. The process diagram shows the actual process (1-2) and the ideal process (1-2s). The first law (isanthropic) efficiency of the turbine compares the actual work that will be implemented in a process that comes out under the same pressure as a real turbine if the process is isetropic (designated as dot 2s). The effectiveness of the second law is the attitude of the realized work to the delivered energy. (3.39)ET, II-h1-h2ψ1-ψ2-1-iTψ1-ψ2For compressor or pump (see Figure 3.7) the guiding equations are similarly defined, figure 3.7. Adiabath compressor or pump analysis. Entrance and weekend streams carry a stream of enthalpy and exergy. Specific works are delivered through the turbine shaft. Destroyed exergies are shown as irreversibility. The process diagram shows the actual process (1-2) and the ideal process (1-2s) and, (3.43)K,II-ψ2-ψ1h2-h1'1'iCh2'h1Robert T. Balmer, in modern engineering thermodynamics, 2011Next, we define the esanthropic efficiency as the ratio of actual work to isentropic work for a working device:('s) work-producingdevice-WactualWisentropic-W ̇actualW ̇isentropicor ratio of isentropic work to actual work for the work of absorbers device:()work-absorbdevice-WisentropicWactual-W ̇isentropicW ̇actualIts are similar to the work of transport energy efficiency, defined in Chapter 4 but while the thW was based on comparing the actual performance of the device with what would happen if the device was reversible, isentropic efficiency and based on comparing the actual performance of the device with what would happen if the device was adeabatic, also reversible (i.e., isentropic). Since most prime engines and pumps are insulating, we always assume that they are adebatable when their heat loss is not given. Isentropical efficiency efficiency and work devices are mathematically defined in table 13.2. 13.2. steam turbine isentropic efficiency calculation