Isentropic efficiency compressor refrigeration cycle

Efficiency - Working with the refrigerant circuit To measure what is happening, the first thing to do is to find temperatures and pressure at key points in the circuit. Temperature T and Press P metrics can be used to define the process. The steam compression cycle, practical circuits and P-h charts When the 1,2,3,4 P-h chart points are defined, the cycle can be plotted on a P-h chart. P-h diagrams for all common working fluids (refrigerants) are available. In these charts press and enthalpy values are shown to scale. In the old days, large diagrams were used, and refrige engineers would plot the circuit diagram on the diagram so that different parameters at each point of the circuit could be read. The pressure level before and after compression determines the position of the horizontal lines. These pressure levels define the evaporation and condensation temperature, and this can be read in the diagram by determining which temperature line is cutting the saturation curve. The measured temperatures determine the endpoints of the lines using the temperature contour lines shown in the chart. Today, refrigerant properties are readily available on your computer, which is why working with diagrams has become a thing of the past. The computer method is much faster. Information on all parameters such as evaporative temperature, condensing temperature, overheating, underrunning and COP is immediately available from the pressure and temperature sensor inputs that carry the data directly into the computer. . An explanation of the P-h chart, Practical Cooling Cycle, Evaporation and Condensation is in Understanding Cooling Pages. Efficiency or COP The conventional way to express the effectiveness of cooling is COefficient of Performance (COP). This is the relationship between heat extraction and energy supply. Both are expressed in the same units, usually kilowatts (kW). For a system where the mass flow rate is constant, the cop becomes equal to the ratio of enthalpi change in the evaporator and enthalpi increase during compression. dh1/dh2. The pH chart displays enthalpi as enthalpi per unit mass or specific enthalpi. The dh1/dh2 ratio is the speed of energy transfer in the evaporator divided by the speed of energy transfer in the compressor. In an actual system pressure drops are bound to occur in heat exchangers and connecting tubes. Slanted lines are displayed to indicate this. A downward slope indicates that the pressure is being reduced. Provided that the pressure is measured at the compressor inlet and outlet and the temperatures are measured in the right places, the cop of the system can be found. Suction line loss The wise reader will have noticed that in the case of a long connection line between the evaporator and the compressor, the condition of the refrigerant is compressor input may differ from the evaporator output. Between these two points there may be a pressure drop and an increase in enthalpi. The pressure drop would occur as a result of power loss in the line. The enthalpi increase would be due to heat gain from the refrigerant on its journey from the evaporator to the compressor. The chart shows this additional pressure drop at dP, and the increase in enthalpi is indicated by the difference between dh1 and dh2. In most systems both these effects are very small. In the diagram, they are much enlarged for clarity. For practical purposes, they only occur in systems where the evaporator or cooler is far away from the compressor. In this case, cop accuracy can be improved by measuring the temperature at the evaporator outlet and using this value to determine dh1, while using the compressor inlet temperature to determine dh2. Note that for perfection, the pressure at the evaporator outlet should also be measured, but this is not necessary in practice, because the enthalpi effect of the pressure change in the pressure, dP, is usually very small in this situation . The steam enthalpi, h is primarily temperature dependent. The compressor's efficiency Accurate measurement of pressure and temperatures at the positions shown above makes it possible to find the compressor's efficiency. How? Unload enthalpi at point 2, where the steam enters the compressor, and in point 3, where the steam leaves the compressor to find enthalpi change dh2. The increase in enthalpi occurs because the steam is being worked on to increase its pressure. The ideal (minimum) amount of work is known. This is the work to be done in the event of an ideal or reversible compression process. The enthalpi increase for such a process can be read the chart (or calculated by the computer). This is done by finding the point on the chart at the high pressure, which has the same entropy as point 2. This ideal or mimimum enthalpy difference is shown as Ideal dh2 in the chart. Entropy is the property that remains constant for reversible processes. Outlines of constant entropy appear on the chart. Compressor efficiency can be defined as the ratio between this enthalpi increase and the actual measured enthalpy increase, i.e. (Ideal dh2/dh2). It's called the compressor's tropical efficiency. An explanation of reversible process can be found on the About Concooling pages. Compressor heat loss In the above method of finding COP, the enthalpi increase dh2 is determined by measuring temperature and pressure at the end of the compression process. Measurement immediately before and after the compressor makes it possible to plot points 2 and 3 of the map so that dh2 can be read. But a compressor releases heat into the atmosphere. It's going to be hot. Heat the compressor disappears into the surrounding air, resulting in correspondingly less heat that leaves itself in the refrigerant steam flow. The consequence is that enthalpi at point 3, taken from the discharge line of temperature measurement, will tend to be too low, and DH2 is as smaller than it should be. The COP will be too high. For an accurate assessment of COP, this heat loss H must be taken into account. The heat loss of the compressor is usually expressed as a percentage of the electrical current input. The net energy supply becomes the electrical energy supply E minus the heat loss H , and this amount (E-H) corresponds to energy gain represented by dh2. For example, if it is known that 5% of the electrical power input is heat loss, then 5% should be added to the value of dh2 found from the measured temperature values. The correct dh2 and thus the correct efficiency can then be found. For most compressors, the heat loss is between 5 and 7%, and a fault in this value has only a very small effect on the final result in most cases. Sounds very complicated? The calculations might, but the computer will do all the work and it will do it almost immediately! An explanation of the P-h chart, Practical Cooling Cycle, Evaporation and Condensation is in Understanding Cooling Pages. Go to the top of the page W.G. Le Roux, J.P. Meyer, in Clean Energy for Sustainable Development, 2017 Compressor isentropic efficiency, compressor corrected mass flow rate, compressor pressure ratio and rotational speed are inextricably linked and are available from the compressor card [8.43]. Compressor and turbine maps from standard Garrett turbochargers [8] are taken into account. The compressor isentropic efficiency and axle speed are achieved with interpolation. The compressor must operate within its compressor short range, otherwise flow wave or suffocation may occur. According to Ref. [23], the efficiency of the turbine is determined by calculating the blade speed ratio [44-46] as shown in Eq. (6.28) and Eq. (6,29). The blade speed ratio is a function of inlet latehalpi, pressure ratio, turbine wheel diameter and rotational speed [23,45]. According to Guzzella and Onder [47], typical values for the maximum turbine efficiency of the automotive industry are ηt,max≈0.65−0.75. Note that the mass flow rate of the system is equal to the actual turbine mass flow rate and is calculated with Eq. (6.30), where P7 is in pounds per square inch and T7 is in Fahrenheit degrees [8], respectively. (6.28) BSR=2πN60(Dt2)[2hin(1−rt1−kk)]12(6.29)ηt=ηt,max(1−(BSR−0.60.. 6)2)(6.30)m ̇t=m ̇tCF×P7/14.7(T7+460)/519Claire Soares, in gas turbine 2008Ingested rain evaporates in a compressor. (i) the possible impact absorption area due to varying amounts of work on the liquid water, (ii) downstream gas conditions and (iii) the variation at the output; due to a change in assumptions about how many phases the liquid water absorbs work into. Conditions are as below:Blade speed (m/s)700Fast PR5Theorical ice tropical efficiency0.86Number compressor stages6WaterAirTemperature (K)293293Press (kPa)100100Mass flow (kg/s)1.0100(i) Work done on liquid water from Formula 10.9 the work of the performed in each compressor phase is DPW = 0.5 × Wwater × U2:DPW = 0.5 × 1 × 7002DPW = 245 kW per phaseSide evaporation requires some temperature increase, the work will be done in one step at least. The maximum number of stages will be the full six:DPW=245 kW minimum and 6 × 245 = 1470 kW maximum (ii) Gas conditions at compressor exit compressor temperature increase for dry air. Ignore changes in compressor performance due to intercooling. T3 - T2=T2 × T2 × (P3Q2(γ-1)/γ -1)/ETA2T3 - T2=293 × (52/7 – 1)/0.86T3=492 K = 219°C = 219°C Calibration of water ingested using Formula 10.7:Hwater=3.1566–12 × 206 – 2.9348-09 × 205 + 1.0407–06 × 204 - 0.016703-03 × 203 + 0.0120915 × 202 + 3.87675 × 20 + 0.74591Hwater=81.94 kJ/kgDriv partial steam pressure at 0.74591Hwater=81.94 kJ/kgCalc rain partial steam pressure at 0.74591Hwater=81.94 kJ/kgCalc rain partial steam pressure at 0.74591Hwater=81.94 kJ/kgCalculate partial steam pressure at 0.74591Hwater=81.94 kJ/kgE rain partially steam pressure at 0.74591Hwater=81.94 kJ/kgE rain partially compressor output Formula 10.6:Pw=P/[(0.622/SH) + 1]Pw=5/[(0.622/0.01) + 1]Pw=0.0791 barCalculation of steam using Formula 10.10 for overheated steam. Add work done on liquid water in the first compressor phase. Hsteam=2.98–04 × T2 + 183 × T 2500 – 5.1420708 × P/(T + 2 2 2 76)3 - (1.0334237 × P3 - 6.4261331 × P5)/(T + 276)14.14.114.00 787QUwater=(Hsteam – 81.94) × 1.0 – 245Now calculate change in air thapi due to heat absorbed by the water:QUair =100×1,005×(219−Tmix)Cut down to make QUair = QUwater using either built-in spreadsheet functions or manual updates. Converged solution gives Tmix = 193,5 °C.(iii) Variation in mechanical power absorber Recall performed on liquid water per step = 245 kW.Repeat over iteration with QUair = QUwater + 245:Therefore, a difference of one phase changes the mixed temperature by 2,4 K; five phases would change it at 12 K.Note: For this small water concentration the presence of the water could have been neglected, and the temperature change found by considering the air alone. At this point, it is useful to consider a case study of testing/verification of a high-performance gas turbine. In this case, the U.S. DOE is funding work that increases the efficiency of gas turbines made by OEMs in this U.S. DOE program. The main objectives arising from increased efficiency are fuel conservation and emission reduction. Pouria Ahmadi, Ibrahim Dincer, of Comprehensive Energy Systems, 2018A exergoeconomic analysis of AC is described. Further information on exergoeconomic analysis, cost balances and exergo economic factors is discussed earlier in this book and elsewhere [33-35]. To determine the cost of exergy destruction for AC, a cost balance can be utilized. balance for this component can be written as follows:(12)c1Ėx1+cwẆAC+ŻAC=c2Ėx2where c1, c2 and cw are unit costs for intake air, outlet air and work. Here, the inlet air is taken to be free, so its unit price is zero, ie,also ŻAC is the purchase cost rate of AC, which can be expressed as follows:(14)ŻAC = ZAC×CRF× (φ)N×3600where ZAC is the purchase cost of AC and CRF is the capital recovery factor, which is dependent on the interest rate and equipment lifetime n , and determined asAlso, N denotes the annual number of operating hours of the unit , and (φ) is the maintenance factor, which is often 1,06 [35]. The purchase price of ac can be approximated as follows [35]:(16)ZAC=c11ṇa1c12−ηACRACln (RAC), for which each optimization problem it is of great importance to define OF's. In this example, several EVs can be considered through multi-goal optimization. When high efficiency is desired, it is reasonable to include the exergy efficiency of the compressor as ET OF. As shown in Eq. (19) The exergy efficiency of the compressor is a function of the compressor pressure ratio and the ice tropical efficiency. Another OF may be the compressor cost, as expressed in Eq. (20), which is a function of the compressor pressure ratio, the flow rate of the air mass through the compressor and the icy efficiency of the compressor. In view of these two OPs, we can write:(19)OFI=ψAC=Ėx2−Ėx1ẆAC(20)OFII=(c11ṇa1c12−ηACRAClnRAC)CRF×(φ)N×3600E the main design parameters or decision variables for optimizing the frequency system are the compressor pressure ratio and the compressor ice-cream Thus, to perform a multi-lens optimization they are considered our decision variables. To formulate a meaningful optimization problem, there are often limitations that need to be met while optimization is being performed, often to ensure that the solutions are reasonable and realistic. Two limitations as described in Table 1 are taken into account here. Physical limitations to optimizing an air compressor (AC)ConstraintReasonRAC<22Material temperature limitηAC<0.92Corecial availability To determine the best among the optimal design parameters for an AC, a modified version of a GA developed with Matlab software was used. Fig. 4 shows the Pareto limit for multi-lens optimization of an AC, where the pressure ratio and the icy tropical efficiency of the compressor are the two main design variables. The range of values displayed is limited by the problem restrictions.Fig. 4. Pareto limit for optimization of an air compressor (AC), highlights the best trade-off among values for the objective functions (OFs). It can be seen in Figure 4 that total compressor costs increase moderately as the exergy efficiency of the compressor increases up to about 92%. Increasing overall exergy efficiency further increases costs It is seen in Figure 4 that the maximum exergy efficiency is found at design point C (94.44%), while the compressor cost speed is the largest at this point ($33.1 h−1). On the other hand, the minimum value for the compressor cost rate is done at design point A and is about $5.12 h−1. Design point C is the optimal situation when exergy efficiency is the only AF, while design point A leads the optimal design when the total cost of the product is the only OF. Point D is the ideal solution for multi-lens optimization because both GPs are at their optimal values, i.e. they are at their best value. Since this point is not located at the Pareto border, point B can be selected as one of the best solutions because it is close to the ideal solution. The variations in the compressor pressure ratio and the compressor's ice tropical efficiency are illustrated in figures 5 and 6 respectively, showing dispersion distributions for the populations in the Pareto limit for each of these design parameters. The points in these numbers were obtained from the developed Matlab code, and show how the design parameters change within their permitted ranges.Fig. 5. Scatter distribution of compressor isentropical efficiency and its permitted area with the population of pareto frontier.Fig. 6. Scatter distribution of compressor pressure ratios and its permitted area with the population of the Pareto limit. In a GA, a population (called chromosomes or genotype of the genome) that encodes candidate solutions (called individuals, creatures, or phenotypes) to an optimization problem, evolves toward better solutions. Scatter distribution of design parameters is within the range exhibiting good choices of these two parameters for optimization purposes. It should be noted in Figures 5 and 6 that the items are not only close to the boundaries, but they are spread almost randomly within their ranges (8-22 for RAC, and 0.78-0.92 for ηAC). In real optimization, the selection of decision variables is based on the dispersed distribution of decision variables, providing an effective search for the best optimal solution of OF. Ibrahim Dincer, Marc A. Rosen, in Exergy (Second Edition), 2013Air at ambient pressure (1 bar) and temperature T1 enters the compressor. The compressor outlet temperature is a function of the compressor's ice tropical efficiency ηAC, compressor pressure ratio rAC and the specific heating ratio of air γa as follows:(3.35)T2=T1×(1+1ηAC(rACγa−1γa−1)))The compressor's working speed is a function of air mass flow rate m ̇, the air-specific heat at constant pressure Cpa and the temperature difference across the compressor and can be expressed as follows: :Here, Cpa is treated as a function of the temperature as follows (Ahmadi et al. , 2011a):(3.37)Cpa(T)=1.048−(3.83T104)+(9.45T2107)−(5.49T31010)+(7.92T41014)A.M.Y. Razak, in industrial gas turbines, 2.1 Calculate the compressor input that is not dimensional flow W1R1T1/γP1.Step 2.2 Using the compressor's non-dimensional flow and pressure ratios, the compressor's non-dimensional speed N1γ1R1T1 and the compressor's icy tropical efficiency η12 are determined using the compressor's characteristics. Step 2.3 Calculate compressor discharge mass flow, pressure, temperature and gas generator speed N1 using the following:I lack of pruning: [7.14]T2=T1+T1/η12[(P2P1)γa−1γa−1][7.15]N1=γ1×R1×T1N1γ1R1T1where γa is the average ice tropical index between T1 and T2Step 2.4 Calculate compressor power is absorbed using:where cpa is the average specific heat at constant pressure between T1 and T2.J. Fadok in Advanced Power Plant Materials, Design and Technology, 2010Eiging pressure conditions and increased phase load capacity in the compressor have driven technology and materials advances to produce high efficiency gas turbine systems. A trend towards decreasing ice tropical compressor efficiency is seen when it increases the pressure ratio (at constant polytropical efficiency) (Saravanamuttoo et al., 2001, p. 61). To solve this problem, highly efficient airfoils have been developed to incorporate advanced three-dimensional aerodynamic features, mostly adapted from aircraft engines with high pressure conditions, where pressure conditions above 40:1 are successfully applied. At these high pressure conditions, the leaking of air between rotating and stationary components is more severe. This is more difficult to handle in large industrial gas turbines compared to aircraft engines due to the scale, where small holes result in large areas due to large diameters. The large casings and rotors of industrial gas turbines also have much slower thermal response to the gas track components and often result in the minimum time limits for seals and blade tips being limited by transient operation (start-up, decommissioning, etc.). Significant improvements in clearance have been made by means of the latest transient thermal mechanical analysis calibrated for engine measurements capable of accurately calculating the transient interactions between components, allowing for the optimisation of engine distances. Tiantian Zhang, ... Ruzhu Wang, in the Handbook on Energy Efficiency in Buildings, 2019Effet of varying compressor efficiency at COP of the three GSHP system configurations is presented in Figure 9 based on the results of previous studies [24.49]. The compressor isentropical efficiency varies from 65% to 100% for each of the heat pump systems, according to the area of Cengel et al. [54] for low to high-efficiency compressors. The results indicate that the COP heat pump increases almost linearly with compressor efficiency. COP is growing faster for System 3 with compressor efficiency. The differences in heat pump COP's range from as low as 3.81 to high as 5.32 (or at 1.51) for systems 1 and 2 and from 3.80 to 5.42 (or 1.62) for systems 1 and 3. GSHP in System 3 is more sensitive to variations in compressor efficiency than in Systems 1 and 2 [24.49], primarily due to the design and operation of System 3 and its two compression phases. Because the compressor efficiency varies with higher compressor efficiency, the specific enthalpi at mode 3 (compressor output) for System 1, 2 and 3 is identical.Fig. 9. Effect of varying compressor efficiency on the COP heat pump and specific enthalpi for capacitor intake to three systems. (Source: Self SJ, Reddy BV, Rosen MA. Geothermal pumps as clean energy systems: basic, applications and parametric performance analyses. In Harris AM (Ed.), Clean Energy: Resources, Production and Development. New York, NY: Nova Science Publishers, 2011. p. 87-146; Self SJ, Reddy BV, Rosen MA. Parametric performance analyses of geothermal heat pump systems. Int J Energi, Environ Econ 2012;20:563–609.) The effects of varying compressor efficiency on the compressor work requirement of System 1-3 are shown in Figure 10. As the compressor's efficiency increases, the work requirement decreases and the compressor work is lower for System 3 than for the other two heat pump systems. COP variations with compressor efficiency for the three heat pump systems are also shown in Figure 10. It is noted that the heat pump for System 3 exhibits an increasing COP as compressor efficiency increases. Details of this observation have been reported in Self et al. [24.49].Fig. 10. Effect of varying compressor efficiency on system COP and compressor work requirements for three systems. (Source: Self SJ, Reddy BV, Rosen MA. Geothermal pumps as clean energy systems: basic, applications and parametric performance analyses. In Harris AM (Ed.), Clean Energy: Resources, Production and Development. New York, NY: Nova Science Publishers, 2011. p. 87-146; Self SJ, Reddy BV, Rosen MA. Parametric performance analyses of geothermal heat pump systems. Int J Energi, Environ Econ 2012;20:563–609.) Claire Soares, in Gas Turbines (Second Edition), 2015 Compressor degradation accounts for 70-80% of GT performance loss. Therefore, the producers focused their efforts on analysing its degradation mechanism and effective means of performance recovery. For a large industrial turbine, a 1% reduction in the compressor's ice tropical efficiency results in a 1%reduction in production. A combined effect of 1% decrease in airflow and efficiency provides a 1.8% power reduction and an increase of 0.8% in GT heat speed. There are two major contributors to loss of output and efficiency:•Reduction of compressor air flow• Compressor increased aerodynamic loss operation in relation to pressure gradienting, corrosion and fouling changes the roughness and shape of the blades and affects both capacity and optimal aerodynamic behaviour. Compressor capacity is further reduced due to increased parasitic airflow caused by larger holes in seals and clearances. The capacity reduction is typically 1.6 times the decrease in compressor efficiency. It should be noted that local degradation has a chain effect. If a particular compressor phase has a higher loss, phase exit pressure decreases, temperature rises, and all other phases operate on nonconforming conditions.A.M.Y. Razak, in Industrial Gas Turbines, 2007Design point calculation of a simple cycle gas turbine will be considered using three methods. The first method described by Rogers and Mayhew,1 is where gas properties, CP and γ, are considered equal to the compression, heat-pick-up and expansion processes. The second method is similar to that discussed by Saravanamutto et al.3, using fast but γ CP and γ. Heat is determined from combustion diagrams as shown in Figure 2.17. In the third method enthalpy-entropy approach is used, as described in section 2.12. The heat supply is determined by means of the combustion diagrams in Figure 2.17. The effect of increased turbine flow rate due to the addition of fuel in the combustor is ignored, as this increased flow rate is approximately likely to be lost due to leaks and cooling effects. Design point data correspond to the following:•working media are air•compressor input temperature, T1 = 288 K•compressor intake pressure, P1 = 1,013 Bar•compressor pressure ratio, Rpc = 20•compressor istrotro-tropical efficiency, ηc = 0,87•combustible pressure loss, ΔP = 5% of the compressor's delivery pressure•combustion efficiency, combustion efficiency ηb = 0,99•turbine entry temperature, T3 = 1400 K•turbine ice tropical efficiency, ηt = 0,9•inlet and exhaust loss = 0•fuel is keroseneFor the first method, the values for cp and γ are indicated at 1,00 respectively 5 and 1.4 for compression heat addition and expansion process in the gas turbine cycle. From Equation 2.29 the compressor discharge temperature, T2, is calculated by:T2=T1+T1ηc((Rpc)γ−1γ−1)T2=288+2880.87((20)13.5−1)=736.07KThe compressor specific work input, WcWc=1.005×(736.07−288)=450.31 kJ/kgand the compressor discharge pressure, P2 equalsP2=P1×Rpc=1.013×20=20.26 Bar-AThe turbine inlet pressure, P3 is equal to:P3=P2×(1−ΔP/100)=20.26×(1−5/100)=19.247 Bar-ATherefore the turbine pressure ratio is given by:From Equation 2.31 the turbine exit temperature is given by:T4=T3−T3×ηt×(1−1Rpt)13.5T4=1400−1400×0.9×(1−(119)13.5)=683.266Kand the turbine specific work output, Wt is:Wt=1.005×(1400−683.266)=720.318 kJ/kgThe specific heat input, Qin is given by Equation 2.12:Qin=1.005×(1400−736.07)/0.99=673.99 kJ/kgThe net turbine specific work , Wnet = Wt − WcWnet=720,318−450.31=270.01 kJ/kgThe thermal is the ratio of the specific net turbine work to the heat input. The thermal efficiency, ηth, is therefore:The second method also takes into account fixed values for CP and γ but uses different values for the compression and expansion processes. These values for the compression process are the same as those used in the first method. Therefore, the compressor discharge temperature and the specific work of the compressor are the same as calculated in section 2.18.1. Thus: T2=736.07KWc=450.31 kJ/kgFly, as the compressor pressure ratio and loss of burnt pressure are the same as above, the compressor discharge pressure and turbine intake pressure will also be the same as that determined in section 2.18.1. Therefore: The turbine input pressure is equal to:Combustion temperature increase T32 = T3 − T2. For the combustor inlet temperature of 736.07 K, which corresponds to the compressor discharge temperature, and an increase in the combustor temperature of 663.93 K from Figure 2.17 is the theoretical fuel -air f 0.0195. The actual fuel -air ratio fa = f/ηb. Thus, the actual fuel-air ratio is fa = 0.0197. We have assumed that the fuel is kerosene, which has a lower heat value (LHV): Qnet is 43 100kJ / kg. Therefore, the specific heat intake is equal to:Qin=fa×Qnet=0.0197×43100=849.07 kJ/kgFor the expansion process, we assume that cp and γare 1.148 and 1,333, respectively. Turbine output temperature equals:T4=1400−1400×0.9×(1−(119)0.3331.333)=743.84KCTour-specific work is:Wt=1,148×(14000−−743.84)=753,318 kJ/kgThe net-specific work isWnet=753,318−450.31=303,008 kJ/kgThe thermal efficiency in this case is:ηth=2 303.008849.07=0.35687The third method determines the performance of the gas turbine using enthalpier and entropies at the various protruding points of the cycle. It is considered the most accurate method for calculating the construction point of a gas turbine. The method is much more detailed and is usually carried out using a computer program developed for this purpose. However, the processes involved will be outlined. Integration Equation 2.44, which describes the variation of specific heat with temperature for air and products of combustion, equations for enthalpy and entropy can be developed. Therefore:[2.65]H=a(T−T0)+bT2−T022−c(1T−1T0)[2.66]S=alnTT0+b(T−T0)−c2(1T2−1T02)−RlnPP0where T and P are air or gas pressures, respectively, and T0 and P0 are the reference temperature and pressure when enthalpi and entropy are assumed to be zero respectively, when the temperature and pressure are 273 K and 1,013 Bar-A respectively. The constants a, b and c are determined as follows:a=∑i=1nocai×mfib=∑i=1nocbi×mfic=∑i=1nocci×mfiai, bi and ci are the constants defined in Table 2.1 for each component, and noc is the number of components in air or combustion products. In the example inlet pressure and temperature are 1,013 Bar and 288 K. From equations 2.65 and 2.66, we calculate enthalpi and entropy at the compressor input as:H1=14,876 kJ/kgS1=0.053 kJ/kg K.For a compressor pressure ratio of 20, compressor discharge pressure, P2 = 20.26 Bar-A. From equation 2.66, the ice tropical compressor discharge temperature can be determined. This is achieved by using P2 for printing designation in equation 2.66 and varying the temperature until entropy equals 0.053 kJ/kg K. The icy tropical compressor discharge temperature, T2′, works out to: Using this value in Equation 2.65, enthalpi at compressor discharge, H2′ due to icy tropical compression is achieved:The icy tropical efficiency equation 2.28 for a compression process can be written in the form of enthalpier as:ηc=H2′−H1H2−H1, where H2 is the actual enthalpi when discharging the compressor, corresponding to:Using the value of H2 in equation 2.65, the actual compressor discharge temperature, T2, can be determined implicitly:Compressor-specific work: Wc = H2 − H1. Therefore: The fuel-air ratio can now be calculated in the same way as that described in Method 2. The combustor inlet temperature and combustor temperature increase for this case is 702.86 K and 697.14 K, respectively. A theoretical fuel-air ratio, f, of 0.0195 is achieved. Actual fuel-air ratio, fa = 0.0195/0.99 = 0.0197. The heat input Qin is:Qin=0.0197×43100=849,388 kJ/kgFuel is 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 concepts x and y are mole fractions of carbon and hydrogen in the fuel. For kerosene, x = 12 and y = 24 and the expression m is the excess air, determined by fuel -air conditions (fa) as follows:fa =12.01x+1.008y(1+0.78090.2095+0.00930.2095+0.00030.2095) MW, where MW is the mother weight of the air and the factors 12.01 and 1.008 are the atomic weights of carbon and hydrogen respectively. By performing a molar balance using Equation 2.67, the mole fraction of the combustion products (n1, n2, n3, n4 and n5) can be determined in the same way as that described in Chapter 6 (section 6.18.4). Since the turbine input temperature, T3, pressure, P3, and combustion gas composition is now known, Equations 2.65 and 2.66 can be used to determine enthalpi, H3 and entropy, S3 upon turbine entry. Enthalpi at the end due to ice tropical expansion This is achieved using Equation 2.66 and varying turbine output temperature, T4, until the entropy corresponds to the value determined by the turbine inlet, S3. From Equation 2.65 enthalpi, H4's at turbine exit due to the ice tropical expansion can be determined. The turbine's ice tropical efficiency in equation 2.30 can be represented as:, where H4 is the actual enthalpi at turbine output. The values for H3, S3 and H4′ respectively are 1272,995 kJ/kg, 0,958 kJ/kgK and 428,005 kJ/kg. For a turbine ice-entropic efficiency of 0.9, the actual enthalpi at the output from the turbine is 512,504 kJ/kg and the entropy at turbine output is 1.0768 kJ/kgK. Thus the turbine specific work, Wt, is:Wt=H3−H4=1272.995−512.504=760.491 kJ/kgThe net specific work (Wnet) from the gas turbine is:Wnet=Wc−Wt=760.491−445.3=315.191 kJ/kgThe thermal efficiency (ηth) is:ηth=WnetQin=315.191849.388=0.3711.The specific heats at the salient points 1, 2, 3 and 4, as shown in Fig. 2.29, correspond to 1.0011, 1.083, 1.2193 and 1.1198, respectively. The corresponding values for the ratio of specific heats, γ= cp/cv, at main points 1, 2, 3 and 4 are 1.402, 1.3607, 1.3082 and 1,345, respectively. The increase in cp due to compression is due to the increase in temperature as described in equation 2.44. Similarly, there is an increase in CP on prominent point 3 and a decrease of point 4. However, the increase in CP in point 3 is also due to the increase in water vapour in combustion products, which is significant, as shown in Table 2.3. Please also note that there is an increase in the CO2 content of combustion products, a greenhouse gas and is believed to be responsible for global warming. Therefore, gas turbines operating with fuels such as natural gas or methane, which have a higher hydrogen content, will result in increased specific work due to the high levels of water vapour in the combustion products. With methane as fuel, this increase in power output can be as high as 2% compared to the increase in the use of kerosene. Note that increases in specific heats have resulted in a decrease in γ.2.29. Turbine cycle on the temperature-entropy chart. Table 2.3. Composition of combustion productsComponentGravimmetric or mass fractionN20.744O20.162Ar0.009CO20.061H2O0.025 The above example is considered dry air. The effects of humidity can also be included in the analysis. For example, given the relative humidity, the specific humidity may be 100 000 000 000 000 000 000 000 000 000 000 The specific humidity can therefore be added to the air composition as shown in Table 2.2 and the air/gas composition is normalised to determine the sovimetric composition of moist/moist air and then repeat the above procedure. The additional heat supply required to heat water vapour from the compressor discharge temperature, T2, to turbine input temperature, T3, must be calculated. This can be determined using equation 2.68:[2.68]Hs=2,232Ts+2352.623where Hs is the water/steam enthalpy (kJ/kg) and Ts is the water steam/ steam temperature in Celsius.Table 2.4 summarizes the error due to the different methods for calculating the gas turbine design point. Error 1 in Table 2.4 is the percentage error between methods 1 and 3, and error 2 is the percentage error between methods 2 and 3. Note that the first method causes the biggest error, especially in the heat inlet. This is because the method of calculating the heat supply does not take into account the change in gas composition during combustion. The fault using method 2 is quite small, and that's because we calculate the heat supply using combustion curves and strive to adjust for the change in gas composition using different values for cp and γ during expansion. Because these values are closer to the true cp and γ average values, the errors in the design point performance calculation are small. It should be stressed that Method 2 is unsuitable for designing gas turbines and method 3 should be adopted. However, Method 2 provides a quick way to assess the design point performance of gas turbines. Error in methods of calculating the design point performance of gas turbines relative to Method 3Method→123Error 1 (%)Error 2 (%)T2 (K)736.07736.07713.1023.2213.221Wc (kJ/kg)450.631450.631445.31.1971.197Qin (kJ/kg)673.99836.14849.38820.65−1.597T4 (K)683.266743.84750.103−8.91−0.835Wt (kJ/kg)720.318753.27760.491−5.283−0.95Wnet (kJ/kg)270.01303.008315.191−14.334−3.865ηth(−)0.4010.356870.37118.057−3.835Ibrahim Dincer, Marc A. Rosen, in Exergy (Second Edition), 2013Sensitivity analyses can be used to describe the effect on the OFs of varying decision variables. The results of such sensitivity analyses are shown in Figure 24.9 for the GT power plant. 24.9. Variation of exergy efficiency with total cost rate for five design parameters of the GT power plant, for four optimized cases (A-D). a) Effect of increase in AC isentropical efficiency, (b) effect of increase in GT isentropical efficiency, (c) effect of increase in GTIT, (d) effect of increase in AC pressure ratio and (e) effect of increase in AP temperature. Variations of OPs with changes in the compressor's ice tropical efficiency are shown in Figure 24.9a. It is seen that an increase in this design parameter within the permitted range increases the exergy efficiency of the GT power plant, while reducing the overall cost rate with lower efficiency gains than exergys and increasing it for higher efficiency gains of exergy. Thus, a conflict is observed between the OF's. Moreover, as improvements are observed in both OF's, are larger than the region where conflict is observed, this design parameter must have a dispersed near the maximum values. Figure 24.8a confirms this trend. Figure 24.9b indicates that an increase in GT isentropical efficiency leads to an increase in exergy efficiency as well as a decrease in the overall cost rate. Therefore, higher values of GT ice tropical efficiency can be an advantage. Figure 24.9c shows the variation of both JVs when GTIT varies within the permitted range. An increase in this design parameter is seen to increase the exergy efficiency of the power plant, but to reduce the overall cost rate, especially when exergy efficiency is lower, and to increase the overall cost rate, especially when exergy efficiency is higher. This behaviour is due to the fact that an increase in GTIT over a reasonable range results in an increase in the cost of CC, directly affecting the total cost of the plant. The effects of increasing the compressor pressure ratio on both DPs are shown in Figure 24.9d. An increase in compressor pressure ratios increases GT exergy efficiency for all ranges, but reduces the overall cost of lower exergy efficiency gains and increases it at higher exergy efficiency. This trend is reflected for variations of GTIT on the total cost in Figure 24.9c. Figure 24.9e shows that an increase in AP temperature (T3) leads to improvements in both OPs. This explains why the optimal points in Figure 24.8e are at the higher values. Therefore, variations of this parameter do not cause a conflict between the two OF's. Ofs.

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