The Brayton Cycle with Regeneration, Intercooling, & Reheating

Section 8.9-10

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ME 372 Thermodynamics

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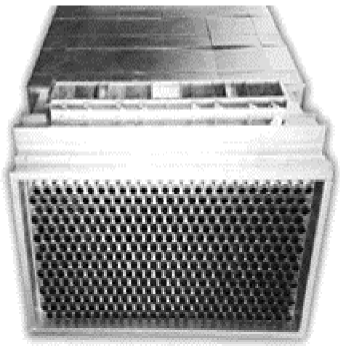
Development of Gas Turbines

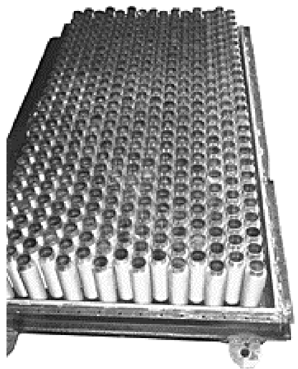
The gas turbine has experienced phenomenal progress and growth since its first successful development in the 1930's. The early gas turbines built in the 1940's and even 1950's had simple-cycle efficiencies of about 17 percent because of the low compressor and turbine efficiencies and low turbine inlet temperatures due to metallurgical limitations of those times. Therefore, gas turbines found only limited use despite their versatility and their ability to burn a variety of fuels. The efforts to improve the cycle efficiency concentrated in three areas:

1. Increasing the turbine inlet (or firing) temperatures.
2. Increasing the efficiencies of turbo-machinery components.
3. Add modifications to the basic cycle. The simple-cycle efficiencies of early gas turbines were practically doubled by incorporating intercooling, regeneration (or recuperation), and reheating. The back work ratio of a gas-turbine cycle improves as a result of intercooling and reheating. However, this does not mean that the thermal efficiency will also improve. Intercooling and reheating will always decrease the thermal efficiency unless they are accompanied by regeneration. This is because intercooling decreases the average temperature at which heat is added, and reheating increases the average temperature at which heat is rejected. Therefore, in gas-turbine power plants, intercooling and reheating are always used in conjunction with regeneration. These improvements, of course, come at the expense of increased initial and operation costs, and they cannot be justified unless the decrease in fuel costs offsets the increase in other costs. In the past, the

relatively low fuel prices, the general desire in the industry to minimize installation costs, and the tremendous increase in the simple-cycle efficiency due to the first (2) increased efficiency options to approximately 40 percent left little desire for incorporating these modifications. With continued expected rise in demand and cost of producing electricity, these options will play an important role in the future of gas- turbine power plants. The purpose of this paper is to explore this third option of increasing cycle efficiency via intercooling, regeneration, and reheating.[1]

Gas turbines installed until the mid-1970's suffered from low efficiency and poor reliability. In the past, large coal and nuclear power plants dominated the base-load electric power generation.1 Base load units are on line at full capacity or near full capacity almost all of the time. They are not easily nor quickly adjusted for varying large amounts of load because of their characteristics of operation.[2] However, there has been a historic shift toward natural gas-fired turbines because of their higher efficiencies, lower capital costs, shorter installation times, better emission characteristics, the abundance of natural gas supplies, and shorter start up times.1 Now electric utilities are using gas turbines for base-load power production as well as for peaking, making capacity at maximum load periods and in emergency situations because they are easily brought on line or off line.2 The construction costs for gas-turbine power plants are roughly half that of comparable conventional fossil fuel steam power plants, which were the primary base-load power plants until the early 1980's, but peaking units are much higher in energy output costs. A recent gas turbine manufactured by General Electric uses a turbine inlet temperature of 1425°C (2600°F) and produces up to 282 MW while achieving a thermal efficiency of 39.5 percent in the simple-cycle mode. Over half of all power plants to be installed in the foreseeable future are forecast to be gas turbine or combined gas-steam turbine types.[1]





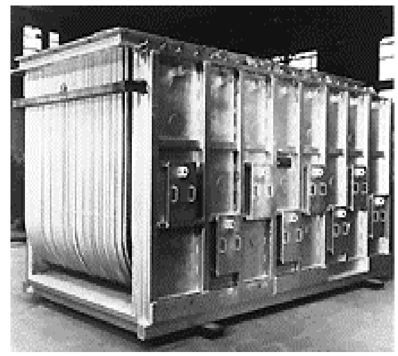


Figure 1: Vertical Recuperator Figure 2: Recuperator Inside Figure 3: Horizontal Recuperator

The Brayton Cycle with Regeneration

In gas-turbine engines, the temperature of the exhaust gas leaving the turbine is often considerably higher than the temperature of the air leaving the compressor. Therefore, the high-pressure air leaving the compressor can be heated by transferring heat to it from the hot exhaust gases in a counter-flow heat exchanger, which is also known as a regenerator or recuperator.[1] Gas turbine regenerators are usually constructed as shell-and-tube type heat exchangers using very small diameter tubes, with the high pressure air inside the tubes and low pressure exhaust gas in multiple passes outside the tubes.[3] The thermal efficiency of the Brayton cycle increases as a result of regeneration since the portion of energy of the exhaust gases that is normally rejected to the surroundings is now used to preheat the air entering the combustion chamber. This, in turn, decreases the heat input (thus fuel) requirements for the same net work output. Note, however, that the use of a regenerator is recommended only when the turbine exhaust temperature is higher than the compressor exit temperature. Otherwise, heat will flow in the reverse direction (to the exhaust gases), decreasing the efficiency. This situation is encountered in gas turbines operating at very high-pressure ratios.1

The highest temperature occurring within the regenerator is the temperature of the exhaust gases leaving the turbine and entering the regenerator.1 The gas turbine recuperator receives air from the turbine compressor at pressures ranging from 73.5 to 117 psia and temperatures from 350 to 450°F.3 Under no conditions can the air be preheated in the regenerator to a temperature above this value. In the limiting (ideal) case, the air will exit the regenerator at the inlet temperature of the exhaust gases. Air normally leaves the regenerator at a lower temperature.1 Gas turbine exhaust gas passes over the other side of the recuperator at exhaust temperatures ranging from 750 to 1000°F. Compressor air temperatures are now raised to higher temperatures up to about 750 to 900°F as it enters the combustor. Turbine exhaust gases are then reduced to between 500 and 650°F from the original 750 to 1000°F. This heat recovery contributes appreciably to the turbine fuel rate reduction and increase in efficiency.3 The regenerator is well insulated and any changes in kinetic and potential energies are neglected.1

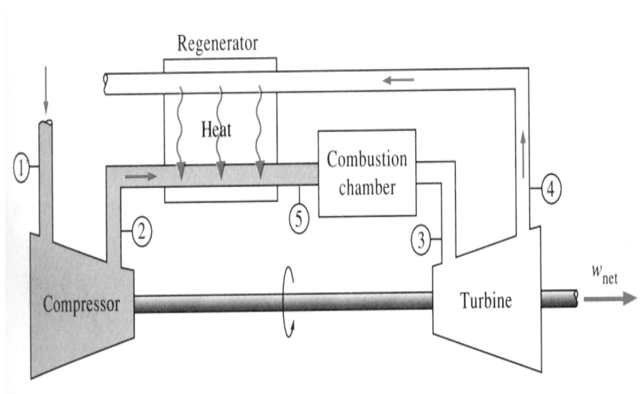


Figure 4: A gas-turbine engine with recuperator

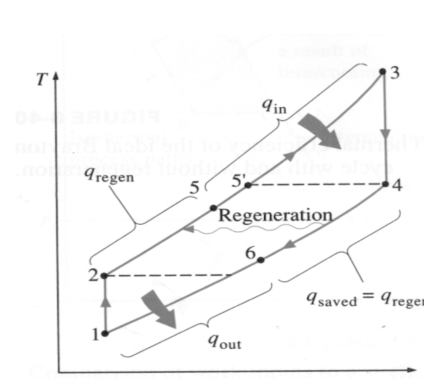
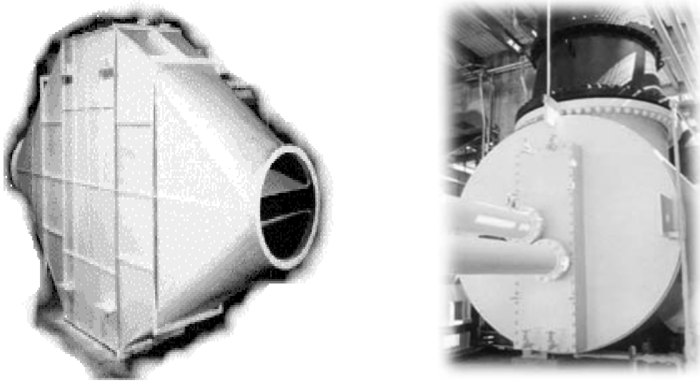


Figure 5: T-s diagram of a Brayton cycle with regeneration

A regenerator with a higher effectiveness will save a greater amount of fuel since it will preheat the air to a higher temperature prior to combustion.1 However, achieving a higher effectiveness requires the use of a larger regenerator, which carries a higher price tag and causes a larger pressure drop because shaft horsepower is reduced. Pressure drop through the regenerator or recuperator is important and should be kept as low as practical on both sides. Generally, the air pressure drop on the high-pressure side should be held below 2% of the compressor total discharge pressure. The gas pressure drop on the exhaust side (hot side) should be held below 4 in. of water.3 Therefore, the use of a regenerator with a very high effectiveness cannot be justified economically unless the savings from the fuel costs exceed the additional expenses involved. The effectiveness of most regenerators used in practice is below 0.85. The thermal efficiency of an ideal Brayton cycle with regeneration depends on the ratio of the minimum to maximum temperatures as well as the pressure ratio. Regeneration is most effective at lower pressure ratios and low minimum-to-maximum temperature ratios.1

Brayton Cycle with Intercooling, Reheating, and Regeneration

The net work of a gas-turbine cycle is the difference between the turbine work output and the compressor work input, and either decreasing the compressor work, or increasing the turbine work, or both can increase it. Carrying out the compression process in stages and cooling the gas in between the lower and higher-pressure stages will decrease the work required to compress a gas between two specified pressures. This is called multistage compression with intercooling. As the number of stages is increased, the compression process becomes nearly isothermal at the compressor inlet temperature, and the compression work decreases.1



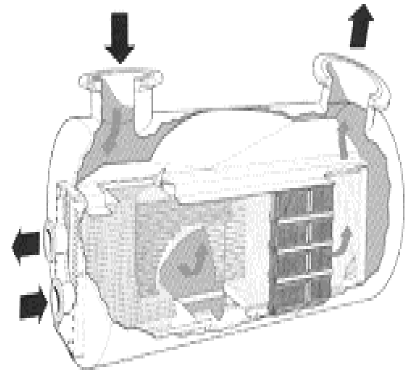


Figure 6: Intercoolers

Figure 7: Intercooler flow

Likewise, the work output of a turbine operating between two pressure levels can be increased by expanding the gas in stages and reheating it in between – that is, utilizing multistage expansion with reheating.1 This process involves dividing the turbine into two parts, a high-pressure and a low-pressure turbine. After the gas passes through the high-pressure turbine it is extracted from the turbine and admitted to a second combustor. Reheated gas flow into the low-pressure turbine, which may be on a separate shaft, or both turbines and the compressor, may be connected to a common shaft. In either case, the reheat process is thermodynamically the same.[4] This is accomplished without raising the maximum temperature in the cycle. As the number of stages is increased, the expansion process becomes nearly isothermal. This is based on a simple principle: The steady-flow compression or expansion work is proportional to the specific volume of the fluid. Therefore, the specific volume of the working fluid should be as low as possible during a compression process and as high as possible during an expansion process. This is precisely what intercooling and reheating accomplish.1

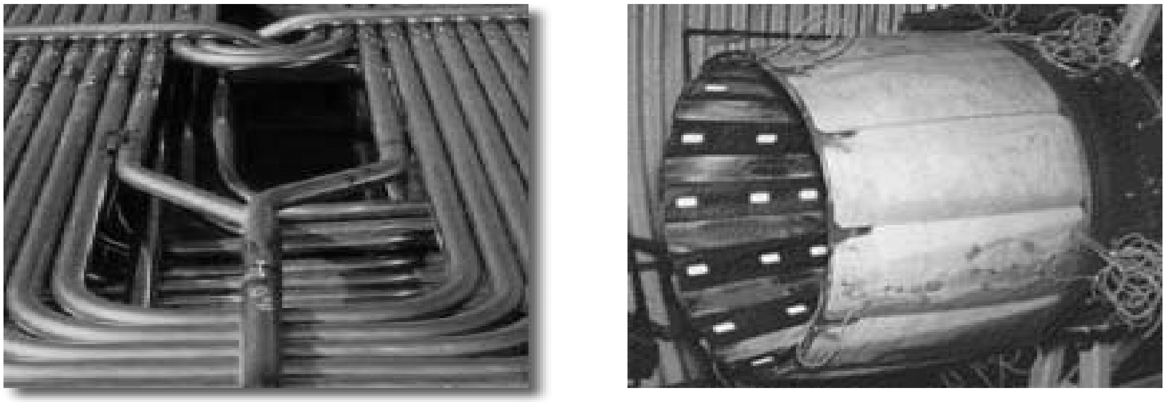


Figure 8: Reheaters

Combustion in gas turbines typically occurs at four times the amount of air needed for complete combustion to avoid excessive temperatures. Therefore, the exhaust gases are rich in oxygen, and reheating can be accomplished by simply spraying additional fuel into the exhaust gases between two expansion states.1

The working fluid leaves the compressor at a lower temperature and the turbine at a higher temperature, when intercooling and reheating are utilized. This makes regeneration more attractive since a greater potential for regeneration exits. Also, the gases leaving the compressor can be heated to a higher temperature before they enter the combustion chamber because of the higher temperature of the turbine exhaust.1

The gas enters the first stage of the compressor and is compressed isentropically to an intermediate pressure and cooled at constant pressure. It is then compressed in the second stage isentropically to the final pressure. The gas now enters the regenerator, where it is heated at a constant pressure. In an ideal regenerator, the gas will leave the regenerator at the temperature of the turbine exhaust. The gas enters the first stage of the turbine and expands isentropically where it enters the reheater. It is reheated at constant pressure, where it enters the second stage of the turbine. The gas exits the turbine and enters the regenerator, where it is cooled at a constant pressure. The cycle is completed by cooling the gas to the initial state (or purging the exhaust gases).1

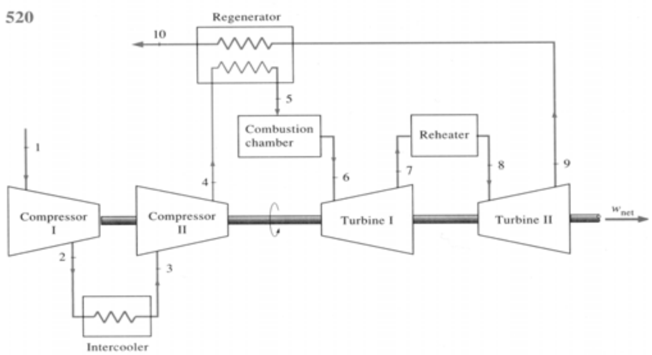


Figure 9: A gas turbine engine with two-stage compression with intercooling, two-stage expansion with reheating, and regeneration

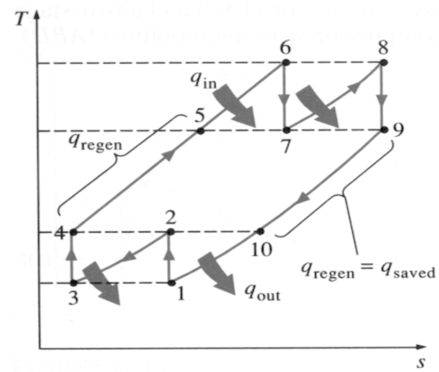


Figure 10: T-s diagram of an ideal gas-turbine with intercooling, reheating, and regeneration

In the analysis of the actual gas-turbine cycles, the irreversibilities that are present within the compressor, the turbine, and the regenerator as well as the pressure drops in the heat exchangers should be taken into consideration.1

Conclusion

If the number of compression and expansion stages is increased, the ideal gas-turbine cycle with intercooling, reheating, and regeneration will approach the Ericsson cycle and the thermal efficiency will approach the theoretical limit (the Carnot efficiency). That is, the thermal efficiency almost doubles as a result of regeneration, intercooling, and reheating. However, the contribution of each additional stage to the thermal efficiency is less and less, and the use of more than two or three stages cannot be justified economically.1

Following are two example problems. The first is an example of a simple ideal Brayton cycle without any modifications to the basic cycle. The second example shows how modifications of intercooling, reheating, and regeneration effect the basic Brayton cycle.

Example 1:

A simple ideal Brayton cycle with air as the working fluid has a pressure ratio of 11. The air enters the compressor at 300K and the turbine at 1200K. Accounting for the variation of the specific heats with temperature determine (a) the air temperature at the compressor and turbine exits, (b) the back work ratio, and (c) the thermal efficiency.

Assumptions: 1. Steady operating conditions exit. 2. The air-standard assumptions are applicable. 3. Kinetic and potential energy changes are negligible.

Analysis: (a) The air temperatures at the compressor and turbine exits are determined by applying the energy equation to the 4 processes involved in the Brayton cycle:

Process 1-2 (isentropic compression of an ideal gas):
Given: T1 = 300K
From Table A-17: h1 = 300.19 kJ/kg
Pr1 = 1.386
Pr2 = (P2/P1)8Pr1 = (11)(1.386) = 15.284
T2 = 579K (at compressor exit)
h2 = 584.96 kJ/kg
Process 3-4 (isentropic expansion of an ideal gas):
Given: T3 = 1200K
From Table A-17: h3 = 1277.79 kJ/kg
Pr3 = 238
Pr4 = (P4/P3)*Pr3 = (1/11)(238) = 21.636
T4 = 648K (at turbine exit)
H4 = 657.89 kJ/kg

(b) To find the back work ratio, we need to find the work input to the compressor and the work output of the turbine:
wcomp, in = h2 - h1 = 584.96 – 300.19 = 284.77 kJ/kg
wturb, out = h3 - h4 = 1277.79 – 657.89 = 619.9 kJ/kg
Thus, Back work ratio (brw) = $\frac{\text{Wcomp,in}}{\text{Wturb, out}}$ = 284.77/619.9 = 0.459
That is, **45.9 percent** of the turbine work output is used just to drive the compressor.

(c) The thermal efficiency of the cycle is the ratio of the net power output to the total heat input:
qin = h3 – h2 = 1277.79 – 584.96 = 692.83 kJ/kg
wnet = wout – win = 619.9 – 284.77 = 335.13 kJ/kg
Thus, ηth = $\frac{\text{wnet}}{\text{qin}}$ = 335.13/692.83 = **0.484 or 48.4%**

Example 2:

Consider an ideal gas-turbine cycle with two stages of compression and two stages of expansion. The pressure ratio across each stage of the compressor and turbine is 3. The air enters each stage of the compressor at 300K and each stage of the turbine at 1200K. Determine the back work ratio and the thermal efficiency of the cycle, assuming (a) no regenerator is used and (b) a regenerator with 75 percent effectiveness is used. Use constant specific heats at room temperature.

Note: In the previous example the pressure ratio is 11 versus 3 in this example. Gas-turbine plants without intercooling, reheating, and regeneration operate more efficiently at higher pressure ratios. Gas-turbine plants incorporating intercooling, reheating, and regeneration operate more efficiently at lower pressure ratios. Had the plant in the previous example operated at a pressure ratio of 3 as in the current example, its back work ratio would have been 33.5 % and the thermal efficiency would have been 25.4%

Assumptions: 1. Steady operating conditions exits. 2. The air-standard assumptions are applicable. 3. Kinetic and potential energy changes are negligible.

Analysis: For two-stage compression and expansion, the work input is minimized and the work output is maximized when both stages of the compressor and the turbine have the same pressure ratio. Thus,

$P2/P1 = P4/P3 = \sqrt{3} = 1.73$ and $P6/P7 = P8/P9 = \sqrt{3} = 1.73$

Air enters each stage of the compressor at the same temperature, and each stage has the same adiabatic efficiency (100 percent in this case). Therefore, the temperature (and enthalpy) of the air at the exit of each compression stage will be the same. A similar argument can be given for the turbine. Thus,

At inlets: T1 = T3, h1 = h3 and T6 = T8, h6 = h8

At exits: T2 = T4, h2 = h4 and T7 = T9, h7 = h9

Under these conditions, the work input to each stage of the compressor will be the same, and so will the work output from each stage of the turbine.

(a) In the absence of any regeneration, the back work ratio and the thermal efficiency are determined as follows:

Given: T1 = 300K
From Table A-17: h1 = 300.19 kJ/kg
Pr1 = 1.386
$Pr2 = (P2/P1)*Pr1 = (\sqrt{3})(1.386) = 2.401$
T2 = 351K
H2 = 351.39 kJ/kg
Given: T6 = 1200K
From Table A-17: h6 = 1277.79 kJ/kg
Pr6 = 238
$Pr7 = (P7/P6)*Pr6 = (1/\sqrt{3})(238) = 137.4$
T7 = 1048K
H7 = 1100.75 kJ/kg

Then
wcomp, in = 2*(wcomp, in, 1) = 2*(h2 - h1) = 2*(351.39 – 300.19) = 102.4 kJ/kg
wturb, out = 2*(wturb, out, 1) = 2*(h6 – h7) = 2*(1277.79 – 1100.75) = 354.08 kJ/kg
wnet = wturb, out – wcomp, in = 354.08 – 102.4 = 251.68 kJ/kg
qin = qprimary + qreheat = (h6 – h2) + (h6 – h7)
= (1277.79 – 351.39) + (1277.79 – 1100.75) = 1103.44 kJ/kg
Thus,

$$rbw = \frac{\text{wcomp, in}}{\text{wturb, out}} = 102.4/354.08 = \mathbf{0.289 \text{ or } 28.9\%}$$

and

$$\eta th = \frac{\text{wnet, in}}{\text{qin}} = 251.68/1103.44 = \mathbf{0.228 \text{ or } 22.8\%}$$

A comparison with the previous example from single stage compression and expansion reveals that multistage compression with intercooling and multistage expansion with reheating significantly improves the back work ratio, but also significantly hurts the thermal efficiency. Therefore, intercooling and reheating are not recommended in gas-turbine power plants unless they are accompanied by regeneration.

(b) The addition of a regenerator does not affect the compressor work and the turbine work. Therefore, the net work output and the back work ratio of an ideal gas-turbine cycle will be identical whether there is a regenerator or not. A regenerator, however, reduces the hot exhaust gases leaving the turbine. In an ideal regenerator, the compressed air is heated to the turbine exit temperature T9 before it enters the combustion chamber. Thus, under standard air assumptions, h5 = h7 = h9. Here, the regenerator is of 75 percent effective so Pr7 is reduced by 25%. Thus, the heat input and the thermal efficiency in this case are:

Pr7 = (P7/P6)*Pr6*.75 = (1/√3)(238)(.75) = 103.057
From Table A-17: T5 = 975K
H5 = 1017.32 kJ/kg
qin = qprimary + qreheat = (h6 – h5) + (h6 – h7)
= (1277.79 – 1017.32) + (1277.79 – 100.75) = 437.51 kJ/kg
and

$$\eta th = \frac{\text{wnet}}{\text{qin}} = 251.68/437.51 = \mathbf{0.575 \text{ or } 57.5\%}$$

That is, the thermal efficiency increases by approximately 10% as a result of regeneration compared the first example without intercooling, reheating, and regeneration. The power put into compression is reduced by 255.32 kJ/kg because of intercooling, reheating, and regeneration while the power output decreases by 83.45 kJ/kg because of the lower pressure ratio. If the gas flows through the cycle at 18.14 kg/s, the cycle uses 4632 kJ/s or kW less in compression and produces 1514 kW less power. Including intercooling, reheating, and regeneration is usually well worth the extra cost associated with the second stage. A power generation plant, in an ideal situation, is in production mode 24 hours a day for 365 days per year. This equates to 8,736 hours per year. Businesses pay an average cost of $0.04173/kWh. At this price, the power generation plant would realize $1,136,576 additional profits per year with the reduction of compression electricity required. Adding more stages (no matter how many) can increase the efficiency an additional 7.3 percentage at most and usually cannot be justified economically.

Bibliography

[1] *Thermodynamics: An Engineering Approach*, Cengel, Y.A. and Boles, M.A., New York, McGraw-Hill Book Co., 1998.
[2] *Guide to Electric Power Generation*, Pansini, A.J. and Smalling, K.D., Lilburn, GA, The Fairmont Press, Inc., 1991.
[3] *Practical Heat Recovery*, Boyen, J.L., New York, John Wiley & Sons Inc., 1975.
[4] *Thermodynamics Processes and Applications*, Logan, Jr., E., New York, Marcel Dekker, Inc., 1999.