PART 3

System Identification and Description, Component and System Design, and Simulation

CHAPTER 8

System Identification and Description and Component Design

CHAPTER OUTLINE

In this chapter we define several systems that a thermal engineer normally deals with. The designs of some components in the systems and a few systems are presented later in this chapter and in Chapter 9. The systems are described in some detail to bring out the thermodynamic, fluid mechanical, and heat transfer aspects. At the conclusion of the design of a component or system, the engineer normally determines its performance under conditions different from those chosen for its design. The methods for estimating the performance under off-design conditions, an aspect of simulation, are presented in Chapter 10. ■

8.1 System Identification and Description

We use the term "thermal engineer" to denote a person with expertise in the fields of thermodynamics, fluid mechanics, and heat transfer and thermal systems. Some of the systems that a thermal engineer is likely to encounter are:

1. Power plant: The prime movers for a power plant are:
   a. Steam turbines with steam from a boiler burning coal, oil, or gas
   b. Steam turbines with steam generated in a nuclear reactor
   c. Diesel engines
   d. Gas turbines
   e. Other systems: Systems using solar energy (concentrating collectors, photovoltaic cells), wind energy, and geothermal energy
2. Refrigeration systems for cold storage or air conditioning, including cryogenics
3. Heating, ventilating, air conditioning
4. Pump-pipe networks
5. Thermal systems in transport vehicles

A few of the systems are described in detail; the salient features of the others are briefly touched upon.

8.2 Power Plants

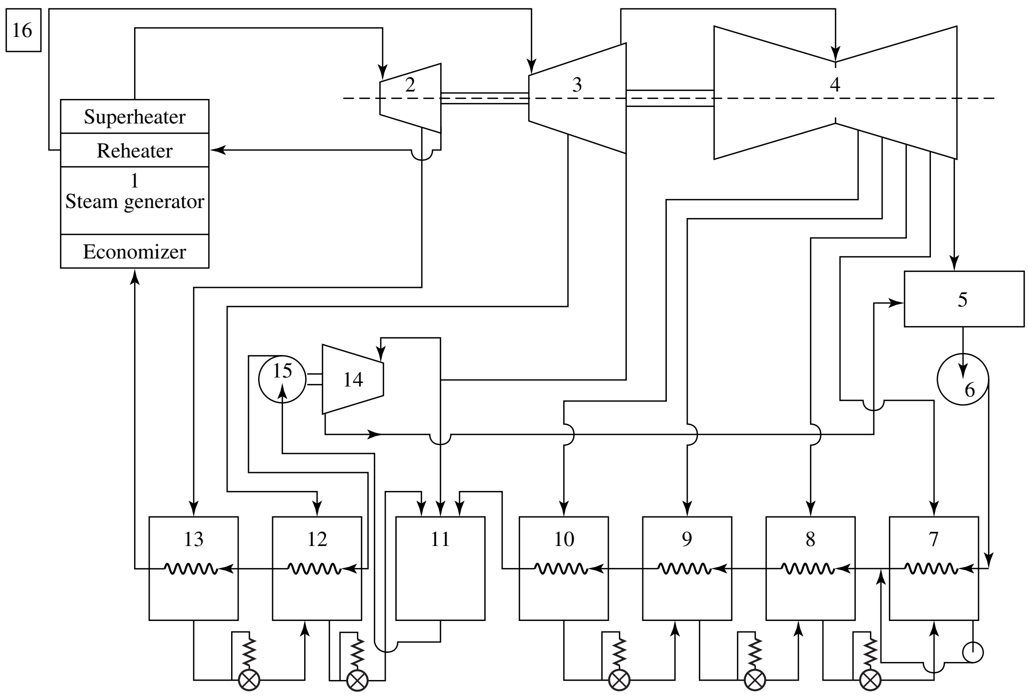
8.2.1 Steam Turbine Power Plant with Steam from a Boiler Burning Coal, Oil, or Gas

Steam power plants, based on the Rankine cycle, are widely used to generate large amounts of power using coal, gas, or oil. The efficiencies of steam plants are quite high, more than 40 percent. Because of the large number of steam power plants and the accumulated experience and knowledge, their reliability is high and repair facilities are widely available. With improved combustion and technology, pollution, one of the concerns of using fossil fuels, has been significantly reduced, although at some cost. In some cases where steam is required for processing, power is a secondary consideration.

The basic components in a steam power plant burning a fossil fuel are:

- Boiler to produce steam
- Turbine driving the generator
- Condenser to condense the steam from the turbine
- Feedwater pump to pump the condensate to the boiler

Although the four components are sufficient to produce power, to increase the efficiency of the power plant several other items are added. Some of them are shown



1	Boiler	5	Main condenser	12, 13	Closed feedwater heaters
2	High-pressure turbine	6	Condensate pump	14	Feedwater pump turbine
3	Intermediate-pressure turbine	7, 8, 9, 10	Closed feedwater heaters	15	Feedwater pump
4	Low-pressure turbine	11	Open feedwater heater	16	Air preheater

Figure 8.1 Schematic of steam power plant

in Figure 8.1. Brief descriptions of the various components follow.

a. *Boiler* (1): Steam is generated in the boiler by burning oil, gas, or coal. Of the three fossil fuels, gas is preferred as it is clean burning. Coal is abundant but air pollution, particularly with high levels of sulfur in the fuel, is a concern. Solid particles in the form of fly ash can be collected and used for lightweight building blocks. The boiler pressure may be as high as 200 atmospheres (20 MPa) or more. Currently the maximum steam temperature is around 600 °C (1100 °F).

b. *High-pressure turbine* (2): Usually the high-pressure turbine is of the impulse type. The major part of the exhaust steam from the HP turbine goes through the reheater to the intermediate pressure turbine. The remaining part goes to feedwater heater 13.

c. *Reheater:* In the reheater the exhaust steam from the HP turbine is usually heated to the maximum temperature for which the plant is designed.

d. *Intermediate Turbine* (3): After partial expansion in the turbine, a part of the steam goes to feedwater heater 12. After full expansion the major part drives the low pressure turbine 4. A fraction is used to drive the feedwater pump turbine 14 and to heat the feedwater in feedwater heater 11.

e. *Low-pressure turbine* (4): The LP turbines are of the reaction type. The major part of the work output is in this turbine. Steam is extracted at several stages, which heats the feedwater in 7, 8, 9, and 10. In large power plants, more than one low-pressure turbine is used.

f. *Main condenser* (5): The exhaust steam from the low-pressure turbines is condensed in the condenser. Besides the main condenser, there may be other condensers.

g. *Condensate pump* (6): The pressure of the feedwater is raised to the boiler pressure in two stages. In the condensate pump the pressure is raised upto the open feedwater heater pressure 11 in pump 6. In the second stage the pressure is raised to the boiler pressure in pump 15.

h. *Closed feedwater heaters* (7 through 10): In the feedwater (FW) heaters, the temperature of the feedwater is raised in stages, with steam extracted after partial expansion in the turbines. The purpose of the feedwater heaters is to reduce the irreversibility of heating the feedwater from the condensate temperature to the saturation temperature corresponding to the boiler pressure. The greater the number of FW heaters, the closer is the heating to a reversible process. In the closed FW heater, the feedwater and the bleed steam (steam extracted from the turbines) do not come into contact. The condensate from one FW heater is piped to the next lower pressure FW heater through a throttling valve.

i. *Open feedwater heater* (11): This heater also functions as a deaerator. Air leakage in those parts where the pressure is less than the atmospheric pressure (low-pressure turbine and condenser), even in small quantities, results in significant degradation of the performance of the plant. It is, therefore, necessary to continuously remove such leakage air. The removal of the leakage air is accomplished in the deaerator–open feedwater heater 11. In the open feedwater heater the bleed steam and the feedwater heater mix.

j. *Closed feedwater heaters* (12 and 13)

k. *Feedwater pump turbine* (14)

l. *Feedwater pump* (15)

m. *Economizer (not shown):* The temperature of the feedwater is raised by the flue gases in the economizer which is a part of the boiler.

n. *Air preheater* (16): The primary air used for combustion of the fuel is heated by the flue gases in the preheater.

The following information on a large coal-fired steam power plant is given to provide an appreciation of how big a power plant is:

Plant output	1350 MW
Boiler height	74 m (240 ft—approximately the height of a 24-floor building)
Boiler width	20 m (70 ft)
Main condenser dimensions	20 m long, 9 m wide, 12 m high
Steam output	445 t/h
Steam state at boiler exit	26 MPa at 568 °C
Feedwater pump turbine	47 000 kW (63 000 hp)
Forced draft fan	Three, each 6700 kW (9000 hp)
Pulverized coal consumption	590 t/h

There are other components such as cooling towers, fuel processing equipment (pulverizers in case of coal, heaters for heavy fuel oil, and so on), and stacks for flue gases (some of them are as high as 300 m, or 1000 ft). A large steam power plant is a very complex system with many components, miles of pipe lines, hundreds of valves and controllers, computer-operated controls, and so on.

8.2.2 Steam Turbine Power Plant with Steam Generated in a Nuclear Reactor

The only difference between a fossil fuel–fired plant and a nuclear power plant is that the reactor replaces the fuel–fired boiler. For details on nuclear reactors refer to Culp (1991).

8.2.3 Diesel Engine Power Plant

Diesel engines are used for small power plants either as the main unit or as a standby unit in facilities requiring substantial power such as hospitals. Diesel engine plants have the advantages of flexibility. By a proper design, the number of units in operation can be easily adjusted to the load. This is particularly advantageous if the load varies; the number of engines can be adjusted such that they operate at near their maximum efficiency. Other advantages are that the units can be brought on line in a very short time and they have high efficiencies of 40 percent or more. The utilization of the fuel can be increased further by incorporating an exhaust gas boiler to make use of the waste heat in the exhaust gases.

Diesel engines may be of the low-speed types (around 100 rpm), medium speed (300 to 900 rpm) or high speed (1800 rpm or greater). Low-speed engines have the advantage of greater reliability, higher efficiency, and longer life. They can burn low-cost heavy fuel oil. But they are bulky, very heavy, and expensive. Medium-speed engines are lighter, are also capable of burning heavy fuel oil but are not as reliable as low-speed engines. High-speed engines are much lighter, have much smaller footprints, but require lighter (more expensive) fuel.

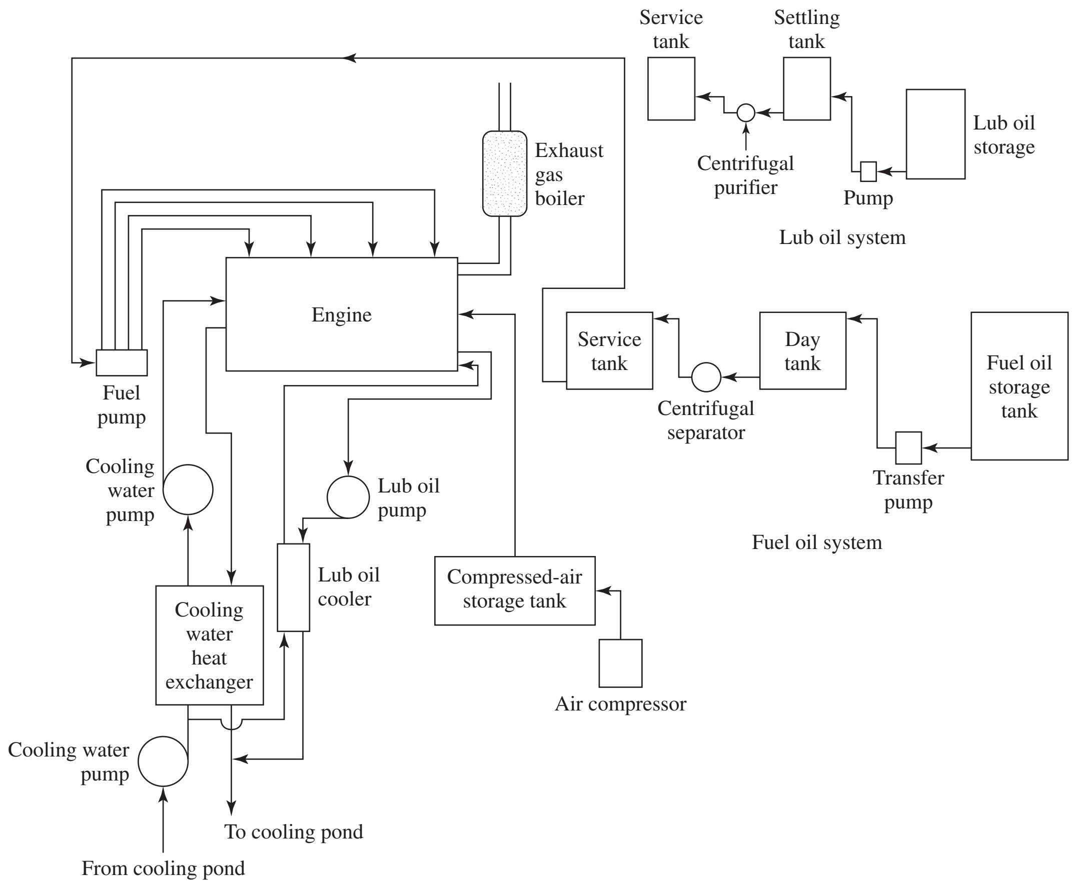


Figure 8.2 Schematic of a diesel engine power plant

The basic components of a diesel power plant are (Figure 8.2):

1. Diesel engine
2. Cooling water system
3. Lubricating oil system
4. Fuel oil system
5. Starting system
6. Exhaust gas boiler
7. Other items

1. *Diesel engine:* Low-speed engines are available with outputs of 100 000 kW/engine. Outputs of medium-speed engines and high-speed engines are much lower. If heavy fuel oil is to be used (low-speed and medium-speed engines), fuel oil treatment (with centrifugal separators, heaters, tanks) should be provided for. High-speed engines that use diesel fuel do not need such equipment.

2. *Cooling water system:* The engines are cooled with water, which, in turn, is cooled in a heat exchanger. The secondary coolant is usually water. The secondary coolant can be from a lake or from a cooling pond. If a cooling pond is used, the water is cooled in a cooling tower. Pumps, usually of the centrifugal type, are used to circulate the primary water through the engines and to circulate the water from the pond. The number of heat exchangers and pumps depends on the capacity of the plant.

3. *Lubricating oil system:* In low-speed engines, two systems, one for cylinder lubrication and another for bearing lubrication, are needed. The lubricant for the cylinder walls is different from the oil for the bearings. Cylinder lubricant is consumed in the cylinders but the bearing lubricant is reused by cooling it. Bearing oil is circulated by a pump, and cooled in a heat exchanger with water from the same source as the engine cooling water. In some cases, lub oil centrifugal separators may be used.

4. *Fuel oil system:* The components of the system are dependent on the fuel used. If diesel fuel is used, no purification or heating may be needed. Heavy fuel needs purification and heating systems.

5. *Starting system:* The engines are usually started with compressed air requiring a compressor and a reservoir. The pressure of the air in the reservoir is about 3 MPa. The minimum pressure for starting large engines is about 1 MPa. The capacity of the compressor and reservoir depends on the number of units and the number of required consecutive starts.

6. *Exhaust gas boiler:* Part of the energy in the exhaust gases is recovered in the exhaust gas boiler to produce steam. The steam may be used to produce power, for space and oil heating.

7. *Other items:* Tanks for fuel oil storage and lubricating oil storage are to be provided. In addition to the compressors to start the engines, a small diesel powered generator to drive the compressor motor may be included in the system. If the plant is used as a power generation unit for consumers, a separate engine-generator system for supplying auxiliary power to run the plant may be provided.

8.2.4 Gas Turbine Power Plant

Gas turbines are used in many applications such as aircraft propulsion, to drive pumps and compressors, power plants, and so on. They are generally not used as base units in large power plants. Gas turbine units are built with outputs of upto 35 MW per unit with an efficiency of 30.5 percent. One such unit is 38 m (122 ft) long, 11 m (36 ft) high, and 8 m (26 ft) wide [El-Wakil (1984)]. The turbine inlet temperature in a gas turbine unit is around 1100 °C; efforts to increase the temperature to 1260 °C to 1320 °C by using ceramic coatings are under way [Sorensen (1983)]. As electric power producing units, they are often used as "peaking" plants to meet the peak demand that may exist for short periods of time and as emergency units. They have the advantage that they can be started quickly so that when the demand exceeds the output of the main plant they can be brought on line. Compared with steam and diesel

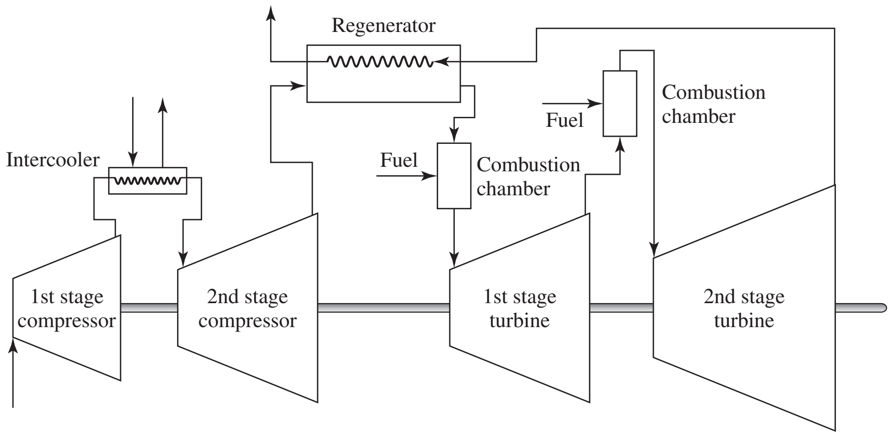


Figure 8.3 Schematic of a gas turbine plant

plants, they are light and less expensive. They are also smooth running. Their main disadvantages are that they are generally not as efficient as either steam plants or diesel engines and they are incompatible with solid fuels. The efficiency of the gas turbine system can be increased by the addition of regenerators to utilize the "waste heat" from the exhaust gases to heat the combustion air and by providing multiple-stage compressors with the addition of intercoolers to cool the compressed air. A schematic of a gas turbine power plant is shown in Figure 8.3. Gas turbines operate on the Brayton cycle, which consists of compressing the atmospheric air, a constant pressure energy addition (by the combustion of fuel), and near adiabatic expansion in the turbine. In gas turbine systems, a significant part of the turbine output is used to drive the compressor. The main components of a gas turbine system are briefly described below.

1. *Compressor:* The compression ratio of air is in the range of 4 to 15, the larger ratio being more common for stationary plants. With a high pressure ratio, intercoolers are employed to cool the air. Although cooling the air increases the fuel consumption to reach a given maximum temperature, the reduction in the compressor power input more than offsets the increased fuel consumption.
2. *Intercooler* to reduce the power input to the compressor: The function of the intercooler is to cool the air after partial compression to a temperature close to the ambient temperature. Usually water is the coolant.
3. *Regenerator:* The purpose of the regenerator is to reduce the fuel consumption per kW. The temperature of the exhaust gas at the turbine outlet is quite high, higher than the temperature of the compressed air at exit of the compressor. By employing a counterflow heat exchanger, the temperature of the compressed air can be raised to almost the exit temperature of the gas from the turbine. Such an increase in the compressed air temperature at inlet to the combustion chamber reduces the fuel consumption and increases the efficiency of the unit.

4. *Combustion chamber:* The temperature of the compressed air raised in the combustion chamber by burning fuel. The common fuel is natural gas for stationary plants but liquid fuel is used in aircraft propulsion units.
5. *Turbines:* There may be two turbines with a secondary combustion chamber. With two turbines, all the compressors, turbines, and generator may be connected to the same shaft or one turbine may be used to drive the compressors and the second one to drive the generator.

8.2.5 Other Systems

There are many other systems for producing power. Some of them are:

1. Hydroelectric units in which the potential energy of water in water falls is used.
2. Systems using solar energy either with a concentrating collector to produce steam for use in steam turbines or direct conversion in photovoltaic cells.
3. Geothermal energy, where the high-temperature earth at some depths is used to produce steam.
4. Wind energy: Turbines driven by wind.
5. Other types of power plants, such as binary vapor plants employing two different fluids operating on the conventional steam cycle; combined units with a gas turbine and a steam power plant, where the exhaust gases from the gas turbine are used to generate steam to drive the steam turbine; and so on. For more information on such units refer to Culp (1991), El-Wakil (1984), Sorensen (1983), and Weston (1992).

8.3 Refrigeration Plant

There are two types of refrigeration plants: compression refrigeration units and absorption units. Brief descriptions of the two types are given below.

8.3.1 Compression Refrigeration System

The basic compression refrigeration system (Figure 8.4) consists of:

1. A compressor where the refrigerant is compressed.
2. A condenser to liquefy the compressed refrigerant.
3. An expansion valve across which the liquid refrigerant pressure is reduced (by throttling) to a low pressure. At the lower pressure the refrigerant is in a two-phase state (vapor-liquid) at the saturation temperature. In small units, such as refrigerators, the expansion valve is replaced by a capillary tube.
4. An evaporator in which heat transfer occurs from the conditioned space or secondary refrigerant.

A number of additional components are incorporated in the basic system to improve the performance of the plant. A plant employing some of those components is shown in Figure 8.4. A brief description of the components follows.

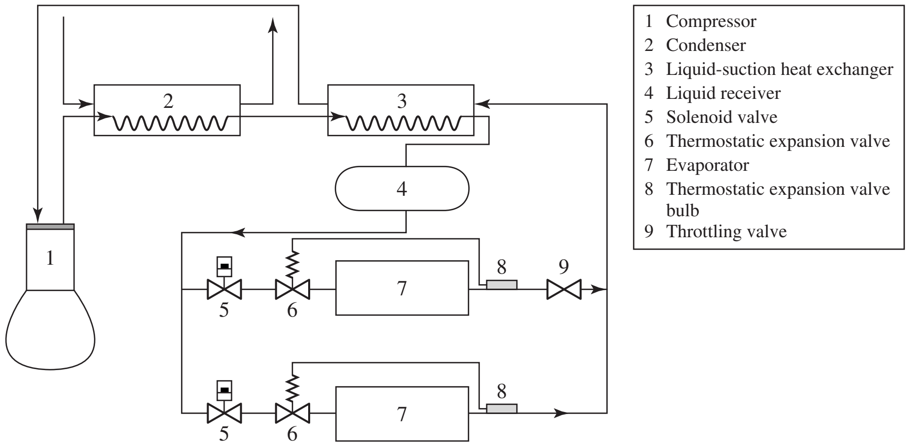


Figure 8.4 Schematic of a compression refrigeration plant

1. *Compressor:* The cool vapor is compressed in the compressor. In a reciprocating compressor, valves may get damaged if the inlet vapor contains liquid drops. Hence, it is important that the inlet vapor is superheated to a small degree. Such superheating is achieved by the thermostatic expansion valve at the inlet of the evaporator and by incorporating a liquid-suction heat exchanger where the vapor is heated by the condensate from the condenser.
2. *Condenser:* The high-pressure gas at exit of the compressor is condensed in the condenser. The cooling medium may be either water or ambient air. The pressure in the condenser depends on the temperature of the cooling medium and the heat transfer surface area. At the exit of the condenser the condensate temperature is a little below its saturation temperature.
3. *Liquid-suction heat exchanger:* The function of the heat exchanger is to cool the condensate significantly below its saturation temperature. The cool vapor from the exit of the evaporator cools the condensate. The consequent increase in the temperature of the vapor increases the work input to the compressor but the increase in the work input is more than offset by the increased cooling capacity. Another advantage is that the liquid drops in the vapor at the inlet to the compressor are eliminated.
4. *Liquid receiver:* The receiver collects the condensate. It is sized in such a way that the entire refrigerant charge of the system can be stored in it when it is necessary to shut the system for repairs or for replacing a component.
5. *Solenoid valve:* The solenoid valve controls the temperature of the conditioned space. When the temperature of the conditioned space increases above a preset point, the solenoid valve opens and the refrigerant flows into the evaporator. When the conditioned space temperature falls below another preset point, the solenoid valve closes.

6. *Thermostatic expansion valve:* Whereas the function of the solenoid valve is to control the conditioned space temperature, the function of the thermostatic expansion valve is to throttle the condensate to a lower pressure and to control the refrigerant flow rate into the evaporator such that at the exit the vapor is slightly superheated.
7. *Evaporator:* The refrigerant at low temperature in the evaporator cools the conditioned space.
8. *Thermostatic expansion valve bulb:* The bulb senses the temperature of the vapor at the exit of the evaporator and transmits a proportionate signal to the expansion valve which then controls the flow rate. If the temperature of the vapor is too low, the expansion valve opening is reduced. If the temperature of the exit vapor is too high, the opening of the expansion valve is increased.
9. *Throttle valve:* With multiple evaporators at different pressures, the refrigerant at exit from those evaporators operating at higher temperatures and, therefore, higher pressures are throttled to the lowest pressure in the system (compressor inlet).

In a large refrigeration system there are many more controls. Even the type of controls may be different. For example, in a small domestic refrigerator, the thermostatic expansion valve is replaced by a capillary tube. The system may also be different, with multiple compressors, a different arrangement of the evaporators and expansion valves, and so on.

8.3.2 Absorption Refrigeration System

The thermodynamics of an absorption system is quite different from that of a vapor compression system. The main difference between the two is that a vapor compression system requires a much larger amount of work input to drive the compressor, whereas the absorption system is essentially thermally driven with very little work input. The only work transfer is to the solution pump; as the pump handles only liquid, the work input is much less than for a vapor compressor for equal mass flow rates and pressure rise. With some modification, even the pump can be eliminated. Therefore, the absorption system is thermodynamically attractive as the major part of the energy input is in the form of thermal energy, but its initial higher cost compared with the vapor compression system has come in the way of its being more widely used. The input energy is usually from burning fossil fuel. Even the fossil fuel can be eliminated by the use of waste heat in industrial plants or by the use of solar energy.

The system operates with two fluids, a refrigerant and a carrier, between two pressures. The most common combinations are ammonia-water with ammonia as the refrigerant, and water–lithium bromide with water as the refrigerant. In the latter combination, with water as the refrigerant, the minimum attainable temperature is about 5 °C, limiting its use to air conditioning and some special applications. With ammonia, much lower temperatures can be attained. The thermodynamic analysis of the system requires a knowledge of binary mixtures (solution of two fluids), and the interested reader may refer to Herold et al. (1996).

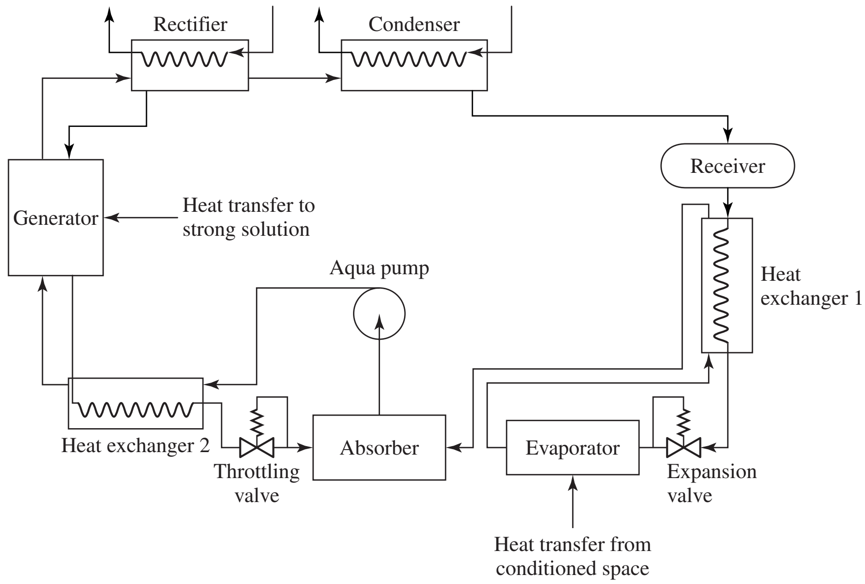


Figure 8.5 Schematic of an absorption refrigeration system

The main components of an absorption system are shown in Figure 8.5. The essential elements of the system are the generator, condenser, evaporator, expansion valve, and absorber. To improve the performance of the unit, other components such as rectifiers and heat exchangers are added. We explain the operation of an ammonia-water system but the processes are the same for a water–lithium bromide system. A strong solution of ammonia water at high pressure is heated in the generator, separating ammonia. The ammonia is condensed in the condenser. The high-pressure ammonia liquid is throttled in an expansion valve, reducing its temperature. The two-phase liquid ammonia–vapor mixture enters the evaporator, where heat transfer to the mixture maintains the conditioned space at a low temperature. The vapor from the evaporator enters the absorber. At the same time, with ammonia separating from the strong solution in the generator, the weak ammonia-water solution is throttled to the low pressure and enters the absorber where it dissolves the ammonia vapor. The strong solution is pumped to the generator. One part of the system consists of the condenser, receiver, heat exchanger 1, and expansion valve. The other part consists of the absorber, heat exchanger 2, generator, and rectifier. The evaporator has the same function as the corresponding components in the vapor compression system. A brief description of the functions of the different components shown in Figure 8.5 follows.

1. *Generator:* In the generator, the strong ammonia-water solution at a high pressure is heated. With an increase in the temperature, the solubility of ammonia in water decreases and a good part of the ammonia separates as gas leaving a weak solution in the generator. The ammonia enters the rectifier.

2. *Rectifier:* The gaseous ammonia from the generator contains some water drops and vapor. The condensed water vapor along with a small quantity of ammonia in solution goes back to the generator. The partially cooled dry ammonia enters the condenser. The ammonia in the rectifier is usually cooled by water.
(In water–lithium bromide systems, only water vapor exits the generator and, therefore, they do not require rectifiers.)
3. *Condenser:* In the condenser the high-pressure, high-temperature ammonia vapor is condensed and the condensate enters the receiver.
4. *Receiver:* The receiver is sized to hold the entire charge of ammonia.
5. *Heat exchanger 1:* In the heat exchanger, the condensate is cooled by the vapor from the evaporator so that the refrigerant capacity per unit mass of the refrigerant is increased.
6. *Expansion valve:* The function of the expansion valve is to throttle the liquid ammonia to a low pressure. At the exit of the valve the ammonia exists as two-phase fluid with a small amount of vapor, at a low pressure and temperature. The mixture enters the evaporator.
7. *Evaporator:* The liquid-vapor mixture evaporates in the evaporator with heat transfer from the conditioned space. Ammonia exits the evaporator in a slightly superheated state and enters the heat exchanger 1. In that heat exchanger the vapor cools the condensate from the receiver, increasing the superheat of the vapor. The vapor then enters the absorber.
8. *Absorber:* The weak solution from the generator enters the absorber, after being cooled in heat exchanger 2, and being throttled to the low pressure. In the absorber the ammonia from the evaporator is absorbed by the weak solution. The strong solution is pumped by the pump into heat exchanger 2.
9. *Heat exchanger 2:* The strong solution at the low temperature cools the weak solution from the generator. At the same time the strong solution is heated by the weak solution from the generator and enters the generator, thus completing the cycle.

8.3.3 Cryogenics

Cryogenics is a special application of refrigeration involving much lower temperatures than what is conventionally associated with refrigeration. An example of a cryogenic application is liquefaction of gases such as nitrogen, hydrogen, oxygen, and helium. Cryogenic temperatures are employed in fields such as medicine, space vehicles, and superconductivity.

The lowest temperature that can be attained with a vapor compression system (with cascading) is about -160 °C. Applications involving temperatures below that value are generally referred to as cryogenics. To achieve such low temperatures, different phenomena are employed—gaseous expansion (Joule-Thomson effect) and adiabatic demagnetization of a paramagnetic salt.

8.4 Heating, Ventilating, and Air-Conditioning (HVAC) Systems

A significant amount of energy is required to maintain the required temperature and humidity in living spaces—residential and commercial. Air-conditioning systems employ refrigeration equipment with the evaporator cooling the air. Heating of the conditioned spaces is generally accomplished by either a fossil fuel–fired furnace heating the air directly, or heating water that is circulated in the conditioned spaces. Electric heating is another possibility with individual heaters in the rooms or a central unit that heats the air circulated in the buildings. In large buildings, steam boilers (steam pressures of 100 kPa to 200 kPa), and water boilers (at about 120 °C, 1000 kPa) are used.

The components of an HVAC system are:

1. *Heating:* Heating of buildings is accomplished by a high-temperature source. The most common source is the combustion gases of a fossil fuel (gas, oil, or coal). A source using fossil fuel may be supplemented or replaced by a system employing solar energy or a heat pump (reverse refrigeration system).
2. *Cooling:* A vapor compression refrigeration system is generally used for cooling. Absorption systems are attractive in combination with solar energy or waste heat.
3. *Dehumidification:* When the relative humidity of the ambient air is high or when a significant amount of moisture is added inside conditioned spaces, dehumidification is necessary. One method of dehumidification is to cool the humid air below its dew point to remove a part of the moisture and then to heat it. Another is to use a desiccant.
4. *Associated ducts and fans*.

8.5 Pump-Pipe Network

The term *pump-pipe network* is used not only for liquids but also for gases where the system consists of fan(s) and ducts or compressor(s) and pipes. Pump-pipe networks are used in water supply and chemical plants. Fan-duct systems are used in ventilation.

Compressed air is used in a variety of applications—driving different types of portable tools, sand blasting, etc. Most laboratories are equipped with compressed air supply.

The solution of flow rates in such systems is complicated by the nonlinear relation between flow rates and pressure drops. Many numerical algorithms have been developed for the solution of such systems.

8.6 Thermal System in Transport Vehicles

The power plant of a modern transport vehicle includes the prime mover, usually an internal combustion engine (using gas or diesel fuel), cooling system, lubricating system, and air-conditioning (both cooling and heating) system.

There are also fluid mechanical systems to control braking, steering, and in other hydraulic systems for special applications such as dump trucks.

Component Design

We described several systems in the previous sections. We now start with the design of components to demonstrate the application of the methodology detailed in Chapter 2 and the synthesis of the different disciplines in thermal sciences. The application of the principles of design are demonstrated in the design of three components in this chapter—a condenser, a space heater, and a wind tunnel.

8.7 Condenser

Condensers are used in a large number of systems; a condenser is an essential component of steam power plants and refrigeration plants. Condensers are also used in chemical plants and other situations.

Having identified a condenser for a component design, we design a condenser for a small power plant. The condenser is to be designed for a steam turbine. The information about the turbine is available in Table 8.1.

8.7.1 Develop Preliminary Specifications and Constraints

There are no explicit constraints. In the absence of any information, we assume that the required amount of cooling water is available if we choose a water-cooled condenser. We will require that the condenser should

- Be easy to maintain and repair
- Conform to applicable practices
- Be such that the temperature rise of the cooling water should not exceed about 7 °C to conform to current practice.

8.7.2 Develop Detailed Specifications and Concept

1. Assume feedwater leaves the condenser at a temperature slightly below the temperature corresponding to the saturation temperature at the exit pressure of 10 kPa.

Table 8.1 Steam turbine

Turbine output	1000 kW
Steam pressure at inlet to turbine	2000 kPa
Steam temperature at inlet	400 °C
Exit pressure	10 kPa
Efficiency of turbine	0.8
Temperature of available cooling water	25 °C
Temperature of ambient air	40 °C

Table 8.2 Comparative merits of water-cooled and air-cooled condensers

Feature	Air-cooled	Water-cooled
Cost	−	+
Weight	−	+
Volume	−	+
Noise	−	+
Source—air or water	Available	Available
Repair	−	+
Reliability	+	+

2. The two types of condensers to be considered are the water-cooled condenser and the air-cooled condenser. Water-cooled condensers are generally of the shell-and-tube type, either one-tube pass or two-tube pass. Air-cooled condensers are generally of the cross-flow type with plates attached to the tubes acting as extended surfaces.
3. The current practice is to limit the water velocity in tubes to a maximum of about 2.5 m/s and air velocity for air-cooled condensers to about 10 m/s.

Consider the two possible designs. Evaluate the advantages and disadvantages of each type, as in Table 8.2. Instead of a numerical rating, indicate the type that is more desirable with a + and the less desirable one with a −. Then look at the overall picture to define the more desirable type of condenser. Considering the relative merits, a water-cooled condenser is the desirable choice. However, if a suitable source of water is not available, an air-cooled condenser would be chosen.

Two types of shell-and-tube condensers are the single-tube pass and the two-tube pass. In the single-pass condenser water enters the condenser at one end and leaves it at the other end; in the two-tube pass type, water enters the condenser at one end, but exits at the same end; i.e., the water flows from one end to the other twice. The two-tube pass condenser is slightly more complicated, as both the water inlet and exit are at the same end. But it is easier to maintain and repair. It is likely to lead to a smaller length. Choose the two-tube pass condenser (Figure 8.6).

8.7.3 Detailed Design

Further specifications are as follows:

- The condenser may be either rectangular or circular (including oval). The choice is somewhat arbitrary. Choose a rectangular condenser.
- Length should be between 1 to 3 times the width.
- As already indicated, water velocity is limited to a maximum of 2.5 m/s. Preferred velocity is 2 m/s.

Steps involved in the design are:

1. Estimate the heat transfer rate.
2. Determine the cooling water flow rate.
3. For different velocities of the cooling water and diameter of tubes, find the length of the tubes.

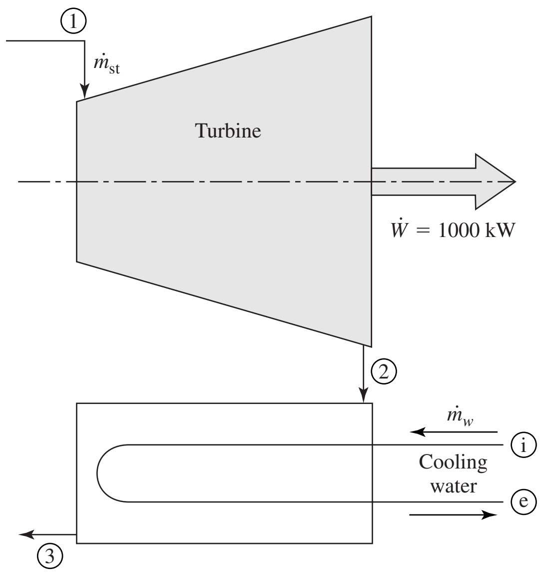


Figure 8.6 Schematic of turbine-condenser

4. Compute the pumping power in each case.
5. Choose the most desirable combination of the sizes of the condenser and pumping power.

Refer to Figure 8.6.

1. Estimate the Heat Transfer Rate With the normal nomenclature,

$$q = \dot{m}_{st}(i_2 - i_3) \tag{8.1}$$

where q is the heat transfer rate from the steam to the cooling water, $\dot{m}_{st}$ is computed from the following set of equations.

$$\eta = \frac{i_1 - i_2}{i_1 - i_{2s}} \qquad i_1 = i_1(p_1, T_1)$$

$$i_{2s} = i_{2s}(p_2, s_{2s} = s_1) \qquad \dot{W} = \dot{m}_{st}(i_1 - i_2)$$

2. Determine the Cooling Water Flow Rate Assuming a reasonable value of T_3, between the exit temperature T_e of the cooling water and the saturation temperature T_2, the heat transfer rate is found.

The mass rate of flow of the cooling water is obtained from the relation:

$$\dot{m}_w = \frac{q}{c_{pw}(T_e - T_i)}$$

Assuming a reasonable cooling water exit temperature (not to exceed 7 °C above the inlet temperature) the cooling water mass flow rate is determined.

3. For Different Velocities of the Cooling Water and Diameter of Tubes, Find the Length of the Tubes The condenser design is defined by the diameter, length, and number of tubes in the condenser, the arrangement (in-line or staggered) and spacing of the tubes, and water velocity. There are a number of combinations of the design that will satisfy the heat transfer requirement. A parametric study will be performed so that one or more appropriate designs can be identified. The steps in the design are:

1. Assume a tube diameter.
2. Assume a water velocity between, say, 1.5 m/s and 2.5 m/s.
3. Determine the number of tubes in each pass to yield the required mass rate of flow of water.
4. Determine the water-side convective heat transfer coefficient, h_i.
5. Determine the steam-side convective heat transfer coefficient, h_o. The condensation heat transfer coefficient requires the tube surface temperature, which is not known. It is determined by iteration.
6. Determine the overall heat transfer coefficient, U.
7. Determine the surface area, and length of the tubes from Equation 6.130 (replacing T_H by T_{sat})

$$\ln\left(\frac{T_{\text{sat}} - T_e}{T_{\text{sat}} - T_i}\right) = -\frac{UA}{\dot{m}_w c_{pw}} = -\frac{U\pi\, d L\, 2N}{\dot{m}_w c_{pw}}$$

where L is the length of the tubes (in each pass), N is the number of tubes per pass, and $2N$ is the total number of tubes in the condenser.

One set of calculations follows. The results for the other sets are given in Table 8.5. The EES program (Appendix B) is used to find all the property values and to perform the computations.

Determine the heat transfer rate:

$$i_1(200 \text{ kPa}, 400\,^\circ\text{C}) = 3247.2 \text{ kJ/kg}$$

$$s_1(200 \text{ kPa}, 400\,^\circ\text{C}) = s_{2s} = 7.126 \text{ kJ/kg}\cdot\text{K}$$

$$x_{2s} = 0.8637$$

$$i_{2s} = 2258.8 \text{ kJ/kg}$$

$$\eta_T = \frac{i_1 - i_2}{i_1 - i_{2s}} \Rightarrow 0.8 = \frac{3247.2 - i_2}{3247.2 - 2258.8} \Rightarrow i_2 = 2456.5 \text{ kJ/kg}$$

Corresponding to $i_2 = 2456.5$ kJ/kg and $p = 10$ kPa, $\rho = 0.07204$ kg/m^3 and x (quality) $= 0.9403$.

$$\dot{m}_{\text{st}}(i_1 - i_2) = \dot{W} \Rightarrow \dot{m}_{\text{st}}(3247.2 - 2456.5) = 1000 \Rightarrow \dot{m}_{\text{st}} = 1.265 \text{ kg/s}$$

Note that the condensate temperature should be between T_{sat} and the average cooling water bulk temperature. The maximum cooling water temperature rise is 7 °C.

Assuming an actual temperature rise of 5 °C, with the cooling water inlet temperature at 25 °C the exit temperature is 30 °C. The exit condensate is between $T_{sat} = 45.85$ °C and the average cooling water temperature of 27.5 °C. Therefore, to find the heat transfer rate to the cooling water assume that the temperature of the condensate is 40 °C. If the actual temperature of the condensate is slightly different from the assumed temperature, the computed heat transfer rate changes only by a small amount as the major part of the heat transfer is due to condensation and the effect of subcooling on the heat transfer rate is small. Thus, with $T_3 = 40$ °C, $i_3 = 167.4$ kJ/kg.

$$q = \dot{m}_{st}(i_2 - i_3) = 1.265(2456.5 - 167.4) = 2895\,\text{kW}$$

The specific heat of the cooling water at the mean temperature of 27.5 °C is 4.176 kJ/kg · K. The mass rate of cooling water is then given by

$$\dot{m}_w c_{pw}(T_e - T_i) = q \Rightarrow \dot{m}_w \times 4.176(30 - 25) = 2895 \Rightarrow \dot{m}_w = 138.7\,\text{kg/s}$$

Starting with $d = 18$ mm, find the number of tubes required in each pass, N, and the water-side convective heat transfer coefficient h_i for fully developed flow for different water velocities ranging from 1.5 m/s to 2.5 m/s. The water-side convective heat transfer coefficient is found from Equation 6.48, reproduced below:

$$\text{Nu}_d = \frac{(f/8)(\text{Re}_d - 1000)\,\text{Pr}}{1 + 12.7(f/8)^{1/2}(\text{Pr}^{2/3} - 1)}$$

$$f = (0.79 \ln \text{Re}_d - 1.64)^{-2}$$

In using Equation 6.48, it is assumed that the flow is turbulent and that the tube length is significantly greater than 10d so that fully developed conditions exist in almost the entire length of the tubes. The computations are illustrated in Table 8.3 for 18-mm-diameter tubes, employing EES.

Now determine h_o from Equation 6.87.

$$\frac{h_o d}{k} = 1.2 \times 0.729 \left[\frac{\rho(\rho - \rho_v) g i'_{fg} d^3}{(T_{sat} - T_s)\mu k} \right]^{1/4}$$

In using Equation 6.87 in the form given, the convective heat transfer coefficient has been increased by 20 percent to account for the interfacial waves. Even at a condensate Reynolds number less than 7.5, because of the dripping of the condensate on the lower tubes, condensate film is disturbed and, therefore, it is appropriate to consider the condensate as wavy for all Re_d. The surface temperature T_s is not known to begin with. The surface temperature and h_o are determined by iteration. The

Table 8.3 Number of tubes and water-side convective heat transfer coefficients

N	V, m/s	Re_d	$f = c_M$	Nu_d	h_i, W/m^2 · °C
360	1.519	32 374	0.023 21	212.3	7 220
300	1.822	38 848	0.022 22	248.5	8 450
260	2.103	44 825	0.021 49	281.1	9 557
200	2.733	58 273	0.020 24	352.1	11 972

iterative scheme adopted is:

a. Assume T_s between 30 °C and 45 °C.

b. Find h_o.

c. Find T_s from the relation $h_o(T_{\text{sat}} - T_s) = h_i(T_s - T_{w,\text{mean}})$. $T_{w,\text{mean}}$ is the mean of the cooling water temperatures at inlet and exit. In equating the heat fluxes, it is assumed that the conductive resistance is negligible. This is a reasonable assumption, as in condensers thin copper tubes (high thermal conductivity) are employed. This assumption will be applied when determining the overall heat transfer coefficient. Further, in the equation, the value of h_o is modified to account for the number of tubes in a column. Details are given in Step **d**.

d. If the computed value of T_s is different from the assumed value, repeat Steps **b** and **c** with the new computed value. The sequence of values in Table 8.4 is obtained starting with $h_i = 9557\ \text{W/m}^2 \cdot {}^\circ\text{C}$ corresponding to $V = 2.103$ m/s and $N = 260$. With a staggered arrangement with 20 tubes per row and 13 rows in each half of the condenser, there are 13 tubes in a column (note that the number of tubes per column is 13, not 26, because of the staggered arrangement). Therefore, the computed value of h_o is modified to $h_{oN} = h_o/13^{1/4}$. In Table 8.4, T_{sN} represents the new value of the surface temperature computed from the equation in Step **c** and used for the subsequent iteration.

e. From Table 8.4 we find that the value of $h_{oN} = 6787\ \text{W/m}^2 \cdot {}^\circ\text{C}$. With the assumption of negligible conductive resistance, the overall heat transfer coefficient is given by

f. $$U = \frac{1}{1/h_i + 1/h_{oN}} = \frac{1}{1/9557 + 1/6787} = 3969\ \text{W/m}^2 \cdot {}^\circ\text{C}$$

g. Finally we determine the surface area and the length of the tubes from the relation

$$\ln\left(\frac{T_{\text{sat}} - T_e}{T_{\text{sat}} - T_i}\right) = -\frac{UA}{\dot{m}_w c_{pw}} = -\frac{U\pi\, dL\, 2N}{\dot{m}_w c_{pw}}$$

Employing the values of the variables in the equation, we obtain $L = 1.361$ m.

4. Compute the Pumping Power Pumping power is the power required to pump the cooling water through the condenser and it is the power to overcome the

Table 8.4 Computation of T_s and h_o ($N = 260$, $N_c = 13$, $h_i = 9557\ \text{W/m}^2 \cdot {}^\circ\text{C}$)

T_s, °C	h_o, W/m² · °C	h_{oN}, W/m² · °C	T_{sN}, °C
40.00	15 196	8 003	35.86
35.86	13 146	6 923	35.21
35.21	12 917	6 803	35.13
35.13	12 890	6 789	35.12
35.12	12 887	6 787	35.12

frictional pressure drop. Denoting the pressures of the cooling water at inlet and exit by p_i and p_e respectively, the pumping power (Equation 5.41) is given by:

$$\dot{W}_{\text{rev}} = \frac{\dot{m}_w}{\rho}(p_i - p_e)$$

For the present case, the pressure difference is a combination of the pressure drop in the tubes and in the water boxes. Neglecting changes in kinetic and potential energies, the pressure drop in the tubes (Equation 5.34) is given by:

$$h_f = c_M \frac{V^2}{2g}\frac{L}{d} \qquad p_i - p_e = h_f \rho g \qquad c_M = (0.79 \ln \text{Re}_d - 1.64)^{-2}$$

The value of $c_M = 0.02149$. For the friction head in the pipe of a total length of 2.722 m (twice the length of the pipe from one end to the other) with a velocity of 2.103 m/s and density of 997.1 kg/m^3, we have

$$h_f = 0.021\,49 \times \frac{2.103 \times 2.103 \times 2.722}{2 \times 9.807 \times 0.018} = 0.733\,\text{m}$$

To this we add an estimated 1 m to account for the losses in the water boxes, valves, and screens at inlet (to prevent debris) so that the total head loss h_{ft} is 1.733 m. The result is:

$$\dot{W}_{\text{rev}} = h_{ft} \times g \times \dot{m}_w = 1.733 \times 9.807 \times 138.7 = 2357\,\text{W}$$

Assuming a pump efficiency of 0.8, the pumping power is

$$\dot{W}_p = \dot{W}_{\text{rev}}/\eta_p = 2946\,\text{W}$$

With the procedure outlined, define the design of the condenser for different diameters, velocities, and arrangement of tubes. The results are given in Table 8.5. Note that the number of tubes per column equals half the number of rows.

Table 8.5 Condenser design

d, m	V, m/s	N (number of tubes per pass)	Number of rows	Number of columns (number of tubes/row)	h_i, W/m$^2\cdot$°C	h_{oN}, W/m$^2\cdot$°C	U, W/m$^2\cdot$°C	L, m	Pump power, W	Surface area, m^2
0.015	1.574	500	40	25	7 635	6524	3518	0.9579	2360	45.14
	1.749	450	30	30	8 364	6991	3808	0.9832	2515	41.70
	2.249	350	35	20	10 388	6451	3980	1.210	3262	39.91
	2.624	300	30	20	11 861	6683	4275	1.314	3927	37.15
0.018	1.519	360	40	18	7 220	6242	3348	1.165	2301	47.43
	1.822	300	30	20	8 450	6626	3714	1.260	2596	42.75
	2.103	260	26	20	9 557	6787	3969	1.361	2946	40.02
	2.733	200	20	20	11 972	7105	4459	1.574	3993	35.60
0.022	1.464	250	20	25	6 805	7264	3514	1.308	2194	45.20
	1.830	200	20	20	8 245	7050	3801	1.511	2545	41.77
	2.033	180	18	20	9 026	7172	3997	1.597	2776	39.74
	2.440	150	20	15	10 555	6819	4143	1.849	3422	38.34

An interesting feature of the design is that with three different diameters and four different water velocities, the surface areas of the different designs are surprisingly close, except for the lowest velocities. We may decide on the design, depending on our criteria for the final configuration. If the surface area, which also indicates the total mass (assuming the same thickness of tubes) is required to be a minimum, the best design would be with 15-mm diameter and a water velocity of 2.624 m/s. However, with that design the pumping power is also very high. Considering pumping power and surface area, possible choices are with a water velocity of about 1.8 m/s. With 22-mm-diameter tubes, the number of tubes is the lowest with 200 tubes per pass. As the number of tubes is directly related to manufacturing costs, we choose that design. It should be noted that the condenser should include water boxes at either end (Figure 6.26) and a suitable plate at the top to prevent the impingement of the high-speed exhaust steam (with water droplets) on the tubes. These will make the overall dimensions significantly higher than the tube bank dimensions.

Diameter of Tube at Inlet to the Condenser The diameter of the tube depends on the velocity of the steam at inlet. A steam inlet velocity of 30 to 60 m/s is suggested in Fraas (1989). With steam at 10 kPa and a quality of 0.9403, the density of the steam is 0.07204 kg/m^3. For a steam velocity of 40 m/s, the diameter of the tube at inlet of the condenser is

$$\dot{m}_{\mathrm{st}} = \frac{\pi \times d^2 \times \rho \times V}{4} \qquad 1.265 = \frac{\pi \times d^2 \times 0.072\,04 \times 40}{4}$$

$$d = 74.8\,\mathrm{cm} \approx 75\,\mathrm{cm}$$

Diameter of Cooling Water Pipes at Inlet to the Condenser The density of water at 25 °C, 100 kPa is 997.9 kg/m^3. Velocity of water is generally around 2 m/s for such applications. Assuming a velocity of 2 m/s, for a mass flow rate of water of 138.7 kg/s, the cooling water tube diameter is

$$\dot{m}_W = \frac{\pi \times d^2 \times \rho \times V}{4} \qquad 138.7 = \frac{\pi \times d^2 \times 997.9 \times 2}{4}$$

$$d = 29.75\,\mathrm{cm} \approx 30\,\mathrm{cm}$$

The final design is given in Table 8.6.

Table 8.6 Design of condenser

Diameter of tubes	22 mm	Number of columns	20
Length of tubes	1.51 m	Pitch between rows	88 mm
Number of tubes per pass	200	Pitch between columns	88 mm
Surface area (total)	41.77 m^2	Width of tube banks	1.94 m
Cooling water velocity in condenser tubes	1.83 m/s	Height of tube banks	1.94 m
		Diameter of steam tube	75 cm
Pumping power	2545 W	Diameter of cooling water pipes to condenser	30 cm
Tube arrangement	Staggered		
Number of rows	20		

In the final design we find that the ratio of the length to the width of the tube banks is 0.78. In the further specifications in Section 8.7.3, the suggested length-to-width ratio is between 1 and 3. However, adding a water box on either side of the tube banks in the suggested design increases the condenser length, bringing the ratio closer to 1.

8.8 Electric Space Heater

In many residential buildings, during winters the heating of some of the rooms may not be adequate for comfort. Because of unbalance in the supply of the heating medium (heated air or water), a room may not attain the desired temperature, or some persons may feel more comfortable at a higher room temperature. In such cases, additional heating is provided by an auxiliary, stand-alone unit. One popular type of such a unit is an electric space heater.

8.8.1 Preliminary Specifications

An electric space heater is to be designed to raise the temperature of a large room by 10 °C. The temperature of the room is to be controlled by a manually set thermostat. The space heater must be functional and satisfy safety features that will allow it to be successfully marketed and sold to the general public. It is to be designed to supplement the existing central heating system.

8.8.2 Develop More Detailed Specifications

A typical house would have several rooms, not all of the same size, with partitions and doors in between. Maximum heating load would be for a room with maximum exposure to outside air with a reasonable number of windows of conventional sizes. The list below reflects these ideas and will be used as the basis for selecting the heating load.

- Space to be heated is a room measuring 6 m × 6 m × 2.5 m (20 ft × 20 ft × 8 ft).
- Room has two interior and two exterior walls. Each exterior wall is fitted with a 1.5 m × 1.5 m (5 ft × 5 ft) double pane window. One of the interior walls has a 0.9 m × 2.1 m × 0.05 m (3 ft × 7 ft × 2 in) hardwood door of normal construction.
- The room has a basement below and a second floor above.
- Exterior walls consist of a dry wall (12.7 mm), fiberglass insulation (90 mm), studs (38 mm × 90 mm), plywood (12.7 mm), foam insulating sheathing (13 mm), and wood bevel lapped siding (13 mm × 200 mm). The ceiling and the floor are of traditional construction, consisting of drywall, studs, and hardwood floor.
- All the studs are placed 400 mm (~16 in) apart (on center).

8.8.3 Develop Concepts

The information in this project is based on two major sources: Kreider and Rabl (1994), and ASHRAE (2001) SI edition.

The space heater is to be designed to raise the temperature of the space by 10 °C. The time required for the heating is not specified. Once the heater is turned on, it will stay on until the temperature reaches the desired temperature. If the unit is equipped with an on-off thermostat, the heater is switched off when the room temperature is slightly above the desired room temperature and switched on when the temperature falls below a set temperature (slightly below the desired temperature).

The total heating power required must be in excess of the total heat load. Energy storage in the room furniture and structure will extend the period of heating required.

The major components of the heating load are the heat loss to the exteriors across the walls and windows, and the energy required to heat the cold air infiltrating through the various cracks and openings in windows and doors.

For a given heat transmission coefficient (overall heat transfer coefficient) U_k, the total heat flow through a shell made of layers of different materials is

$$q_c = \sum_k (U_k A_k)(T_i - T_{ok}) \tag{8.2}$$

Here T_i is the interior temperature and T_{ok} is the temperature of the ambient air in the neighboring spaces. In most buildings, the shell consists of a number of different parts. As a simplification, we consider the effective values for the following groups: glazing, exterior walls, interior walls, ceiling, and floor, which are identified by subscripts g, ew, iw, c, and f, respectively.

$$\sum (U_k A_k) = U_g A_g + U_{ew} A_{ew} + U_{iw} A_{iw} + U_c A_c + U_f A_f \tag{8.3}$$

The heating load due to the infiltration air is

$$q_{ak} = \sum_k \rho c_p Q_k (T_i - T_{ok}) \tag{8.4}$$

where Q_k is the infiltration rate in m^3/s to the room from the neighboring spaces at temperature T_{ok}. The total heating load is then given by,

$$q_t = \sum_k [(U_k A_k) + \rho c_p Q_k](T_i - T_{ok}) \tag{8.5}$$

We assume that the only variable in this equation is the internal temperature T_i. Thermal properties of the enclosure and air are assumed to be independent of temperature. Neighboring zones remain at their respective temperatures T_{ok}. At the beginning of the heating cycle τ_1, the heat loss is

$$q_{\tau_1} = \sum_k [(U_k A_k) + \rho c_p Q_k](T_{i,\text{initial}} - T_{ok}) \tag{8.6}$$

The space is maintained at $T_{i,\text{initial}}$ by the central heating source. The power required to maintain the space at the final desired temperature of $T_{i,\text{final}}$ at the end of the heating cycle at time τ_2 is

$$q_{\tau_2} = \sum_k [(U_k A_k) + \rho c_p Q_k](T_{i,\text{final}} - T_{ok}) \tag{8.7}$$

From the specifications, $T_{i,\text{final}} - T_{i,\text{initial}} = 10$ °C. For a fixed external temperature T_{ok}, Equations 8.6 and 8.7 will define two rates of heat loss for initial and final room temperatures of $T_{i,\text{initial}}$ and $T_{i,\text{final}}$ respectively. The difference between the two is the required heating capacity q_h.

$$q_h = (T_{i,\text{final}} - T_{i,\text{initial}}) \sum_k [(U_k A_k) + \rho c_p Q_k] \tag{8.8}$$

8.8.4 Detailed Design

Estimate Heating Capacity q_h

1. Estimation of Air Leakage The air leakage is determined by employing the Lawrence Berkeley National Laboratory (LBNL) model. This model is based on Equation 8.9 [ASHRAE (2001)]:

$$Q = A_{\text{leak}} \sqrt{A \Delta T + B V^2} \tag{8.9}$$

where

A_{leak} = total effective leakage area, cm^2
A = stack coefficient, (L/s)^2/(cm^4 · K)
ΔT = temperature difference, K
B = wind coefficient, (L/s)^2/[cm^4 · (m/s)^2]
V = average wind speed, m/s
Q = air flow rate, L/s

Stack coefficient A and wind coefficient B are given in Tables 8.7 and 8.8, respectively [ASHRAE (2001)]. The definition of shielding class in Table 8.8 is:

1. No obstruction or local shielding
2. Light local shielding; few obstructions, few trees, or small shed

Table 8.7 Stack coefficient A

	Number of floors		
	One	Two	Three
Stack coefficient	0.000 145	0.000 290	0.000 435

Table 8.8 Wind coefficient B

	House height (floors)		
Shielding class	One	Two	Three
1	0.000 319	0.000 420	0.000 494
2	0.000 246	0.000 325	0.000 382
3	0.000 174	0.000 231	0.000 271
4	0.000 104	0.000 137	0.000 161
5	0.000 032	0.000 042	0.000 049

Table 8.9 Space design parameters

Component	Area (m^2) or perimeter (m)
Windows	4.5 m^2 or 12 m
Door	1.89 m^2
Walls at sill, caulked	24 m
Ceiling wall	24 m
Electric outlets/switches	8 units total

3. Moderate local shielding; some obstructions within two house heights, thick hedge, solid fence, or one neighboring house
4. Heavy shielding; obstructions around most of perimeter, building or trees within 10 m in most directions; typical suburban shielding
5. Very heavy shielding; large obstructions surrounding perimeter within two house heights; typical downtown shielding

Table 8.9 gives the space design parameters for the room.

The calculation of air leakage rate, Equation 8.9, requires the difference between the room and ambient temperatures, and the wind speed. Assuming an outdoor winter temperature of −10 °C, a reasonable value for the midwest, and an indoor air temperature of 20 °C, a temperature difference between the conditioned space and the ambient air of about 30 °C would yield an upper limit. For winters, a design wind speed of 6.7 m/s (15 mi/h) is recommended by ASHRAE. For the floor and ceiling, temperature difference would not exceed 10 °C, since all the floors are considered occupied. Therefore, the upper estimate for the stack coefficient and the wind coefficient from Table 8.7 and Table 8.8 are 0.000 435 $(L/s)^2/(cm^4 \cdot K)$ and 0.000 494 $(L/s)^2/[cm^4\ (m/s)^2]$.

Table 8.10 shows effective leakage areas for a variety of residential building components. The values in the table present leakage areas per item, per unit surface area, or per unit length of crack (lmc), whichever is appropriate [ASHRAE (2001)].

For maximum estimates, the total leakage area and air infiltration are given by Equations 8.10 and 8.11, respectively. Only half the values for walls, ceiling, and electric outlets in Table 8.10 are used, as the other half of the space is exposed to adjacent living areas.

$$A_{\text{leakage}} = 12 \times 1.9 + 12 \times 1.2 + 12 \times 2.5 + 4 \times 3.5 = 81.2\,\text{cm}^2 \tag{8.10}$$

$$Q = 81.2\sqrt{0.000\,435 \times 30 + 0.000\,494 \times 6.7^2} = 15.240\,\text{L/s} \tag{8.11}$$

Table 8.10 Effective leakage areas (low-rise residential applications)

Component	Units	Best estimate	Minimum	Maximum
Windows, double hung, weather stripped	cm^2/lmc	0.65	0.2	1.9
Doors, single door, not weather stripped	cm^2 each	21	12	53
Joints, floor-wall, caulked	cm^2/lmc	0.8	0.075	1.2
Joints, ceiling-wall	cm^2/lmc	1.5	0.16	2.5
Electric outlets/switches with gaskets	cm^2 each	0.15	0.08	3.5

As noted only the effective leakage areas to external air are taken into consideration. Other data have been provided for completeness.

2. Space Heat Loss To find the value of $\sum U_k A_k$ in Equation 8.8, the thermal properties of the different materials as given in ASHRAE (2001) are used, and are given in Table 8.11.

Referring to Figure 8.7, with studs 400 mm apart, $l_{oc} = 400$ mm, $l_s = 38$ mm, and $l_i = 362$ mm. The effective areas of the insulation and stud are $A_i = l_i \times H$, $A_s = l_s \times H$, with a total area $A = A_i + A_s$. Area fractions are $A_s/A = 0.095$, $A_i/A = 0.905$. The actual construction differs from the model used in finding the fractions. ASHRAE recommendations for these fractions are 0.75 for insulation cavity; 0.21 for studs, plates, and sills; and 0.04 for headers. These recommendations contain an allowance for multiple studs, plates, sills, extra framing around windows, headers, and band joints. The latter values are used in the analysis presented here.

The average overall *R*-values and *U*-factors are calculated by assuming parallel heat flow paths (Figure 8.8).

For the outside walls, one path is the dry wall, stud, foam insulation sheathing, and siding, and the other parallel path is dry wall, fiber insulation, foam insulation sheathing, and siding. For the area A_s,

$$R''_s = R''_{c,i} + R''_1 + R''_s + R''_2 + R''_3 + R''_{c,o} = \frac{1}{U_s} \tag{8.12}$$

Table 8.11 Typical thermal properties for common building and insulating materials-design values

Description	Resistance R'', $K \cdot m^2/W$
Outside walls	
Outside surface	0.030
Wood bevel lapped siding, 13 mm × 200 mm	0.140
Rigid foam insulating sheathing, 13 mm	0.700
Mineral fiber insulation, 90 mm	2.290
Wood stud, 38 mm × 90 mm	0.630
Gypsum wall board, 12.7 mm	0.079
Inside walls	
Inside surface, still air (both sides)	0.120
Wood stud, 38 mm × 90 mm	0.630
Air space, 90 mm	0.190
Gypsum wall board, 12.7 mm (both sides)	0.079
Ceiling (floor is the same in reverse order)	
Inside surface, still air (both sides)	0.110
Gypsum wall board, 12.7 mm	0.079
Wood stud, 38 mm by 140 mm	0.980
Air space, 140 mm	0.140*
Plywood (douglas fir) 12.7 mm	0.110
Carpet and fibrous pad	0.370

*90-mm data are used
Surface emittance of ordinary building materials is considered to be 0.90.

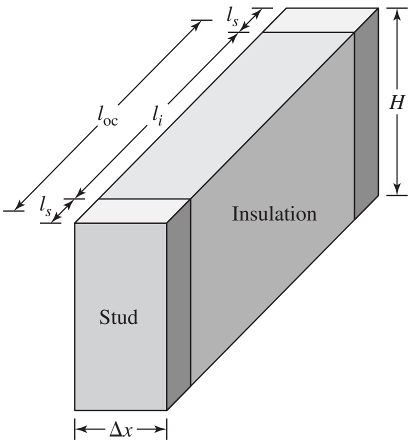


Figure 8.7 Insulation and wood stud arrangement

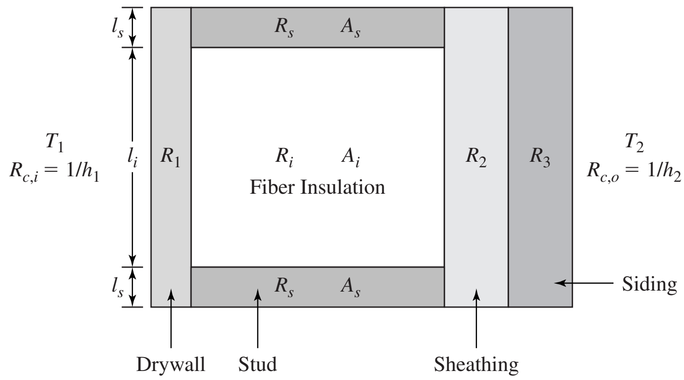


Figure 8.8 Thermal resistances for outside wall

and for A_i,

$$R_i'' = R_{c,i}'' + R_1'' + R_i'' + R_2'' + R_3'' + R_{c,o}'' = \frac{1}{U_i} \tag{8.13}$$

With assumption of parallel paths,

$$U_{\text{av}} = \frac{A_s}{A}U_s + \frac{A_i}{A}U_i \tag{8.14}$$

Therefore, for the outside walls we have:

$$R_s'' = 0.120 + 0.079 + 0.630 + 0.700 + 0.140 + 0.030 = 1.699\,\text{K}\cdot\text{m}^2/\text{W}$$
$$R_i'' = 0.120 + 0.079 + 2.290 + 0.700 + 0.140 + 0.030 = 3.359\,\text{K}\cdot\text{m}^2/\text{W}$$
$$U_s = 0.5886\,\text{W/m}^2\cdot\text{K}$$
$$U_i = 0.2977\,\text{W/m}^2\cdot\text{K}$$
$$U_{ow} = 0.25 \times 0.5886 + 0.75 \times 0.2977 = 0.370\ \text{W/m}^2\cdot\text{K}$$

With similar procedures for calculations for different configurations using the data from Table 8.11, we obtain the overall heat transfer coefficients given in Table 8.12.

Table 8.12 Overall heat transfer coefficients ($\text{W/m}^2\cdot\text{K}$)

Outside walls (U_{ow})	0.370
Inside walls (U_{iw})	1.519
Ceiling (U_c)	0.933
Floor (U_f)	0.933

Table 8.13 Values of *UA*

	Description	A, m^2	U, W/m$^2 \cdot$ K	UA, W/K
Exterior walls	2 walls 6 m × 2.5 m less windows	25.5	0.370	9.435
	2 windows 1.5 m × 1.5 m	4.50	2.63	11.84
Interior walls	2 walls 6 m × 2.5 m less door	28.11	1.519	42.7
	Door 0.9 m × 2.1 m	1.89	1.18	2.23
Ceiling	6 m × 6 m	36.00	0.933	33.59
Floor	6 m × 6 m	36.00	0.933	33.59
Total				133.39

For windows, we assume glass with a pyrolytic (hard) coating with an emittance of 0.40. For a double glazing with 12.7-mm argon-filled space, wood-vinyl frame type, and horizontal slider, the overall coefficient of heat transmission [ASHRAE (2001)] $U_w = 2.630\,\text{W/m}^2 \cdot \text{K}$.

For an interior door of 50 mm total thickness with two 11-mm panels, one on each side, and still air,

$$R'' = 0.11 + \frac{0.011}{0.16} + 0.49 + \frac{0.011}{0.16} + 0.11 = 0.8475\,\text{K} \cdot \text{m}^2/\text{W}$$

and

$$U_{\text{door}} = \frac{1}{R''} = 1.18\,\text{W/m}^2 \cdot \text{K}$$

To calculate (UA), areas of respective components are to be incorporated. Values UA for each component are given in Table 8.13.

Employing Equation 8.8, considering heat losses through the shell and the energy required to heat the leakage air, gives

$$q_h = 10(133.39 + 1.177 \times 1007 \times 15.240 \times 10^{-3}) = 1531\,\text{W} \tag{8.15}$$

For smaller rooms or those that require less power, we will consider multiple heating elements to operate at part capacities. Most residential buildings are equipped with 20-A circuit breakers so that the total power available is more than 2000 W. The heater, even at full power, will permit the operation of units such as a computer, lights, and so on.

3. Electric Heater Type Selection Having determined the heater capacity, we proceed to the design of the heater. Some of the available options are:

- Forced or natural convection
- Use of extended surfaces
- User-controlled variable power setting
- Circular or rectangular cross-section configuration
- Air passage configuration

Forced convection systems require an additional fan. But the higher heat transfer coefficient results in a more compact heater. Natural convection heaters are bulky but are silent and, without a fan, they are more reliable. Natural convection oil-filled systems are low-temperature applications. They are safe but their thermal inertia is large and, therefore, they are slow to react to changes. They are also more expensive.

Extended surfaces are used mostly for natural convection electric heaters, and will be considered if the final selection warrants it.

The option of variable power setting lends itself to manual control of the power input. With a thermostat to control the room temperature, electric heaters operate as on-off appliances. If the power required is much less than the rated power, the unit switches on for short periods, which is not desirable. With two or three power settings, the user can choose a lower power setting, and the frequency of turning the power on and off can be reduced significantly. Although the additional feature adds to the cost, it is more convenient and also improves public acceptance.

A rectangular cross-section heater provides a larger usable air space compared with a circular heater with the same maximum dimensions (foot print).

The air inlet to the heater can be at the bottom, sides, or top. Bottom feed is risky since it may draw light objects such as fiber or lint from the floor; their accumulation may cause blockage to the fan and may even be a fire hazard. Top feed is also risky since accidental spills of liquids or small objects are common occurrences in houses, and the associated risks should be avoided. The logical choice is, therefore, side feed.

On the basis of these discussions, two types, forced and natural convection heaters, are considered. The two types of heater are now evaluated for different desirable features, with a numerical value between 5 (the most desirable) and 1 (least desirable). The design with the highest cumulative score is chosen.

Option/objective	Simplicity	Reliability	Compactness	Safety	Noise	Economy	Total points
Natural convection	5	5	1	3	5	2	21
Forced convection	3	3	5	3	3	4	21

These ratings are somewhat subjective but informative. Neither of these design options is a clear choice. Market analysis may show that both heaters are equally acceptable and well received. From the view of compactness we choose an electric heater with the following specifications:

- Multiple heating cores
- Fan-driven forced-air circulation (forced-convection heater)
- A manually set power control and a thermostat
- Rectangular cross section with air inlet and discharge on opposite sides

4. Heater Element Design To design the heating element from an energy balance we have

$$q_h = q_{\text{rad}} + q_{\text{conv}} \tag{8.16}$$

$$q_{\text{conv}} = h(\pi d_w L_w)(T_w - T_{\text{air,in}}) \tag{8.17}$$

$$q_{\text{rad}} = \varepsilon\sigma \sum_i A_w F_{w,j}\left(T_w^4 - T_j^4\right) \tag{8.18}$$

where

$q_{\text{rad}}, q_{\text{conv}}$ = radiative and convective heat transfer rates
T_w = wire temperature
ε = emissivity
$T_{\text{air,in}}, T_{\text{air,out}}$ = air inlet and outlet temperatures
ρ = air density
V_{air} = air velocity
H, W = heater air box height and width
d_w, L_w = heater wire diameter and length
A_w = the combined area of the heater elements that interacts radiatively with external surfaces each at a temperature T_j in the heater assembly with a view factor $F_{w,j}$ between these areas.

The summation of A_w is over number of surfaces j from which radiative heat transfer must be taken into account.

The estimation of radiative heat transfer term is complex. We, therefore, make the following simplifications. The heater type recommended is one of helical shape to make it compact, where the wire diameter and total length are d_w and L_w respectively. The heater elements are totally shrouded within the enclosure by a metallic shield assumed to be at a uniform temperature T_e. Therefore, Equation 8.18 may be expressed in a simpler form,

$$q_{\text{rad}} = \varepsilon\sigma(\pi d_w L_w)\left(T_w^4 - T_e^4\right) \tag{8.19}$$

The estimation of T_e requires detailed radiation shield analysis with convective terms taken into consideration. It is dependent on the wire temperature T_w and the geometry and the dimensions of the enclosure. In the analysis, T_e/T_w is assumed to be a design variable named T_{ratio}, the effect of which will be discussed later. Emissivity of the heater wire is assumed to be 0.9.

The convective heat transfer coefficient h should be based on correlations that involve fluid motion normal to the axis of a heater coil of helical type. There is no correlation for this configuration. The closest we have is that of a cylinder in a cross-flow (Equation 6.42, reproduced as Equation 8.20).

$$\text{Nu}_d = \frac{h d_w}{k_{\text{air}}} = 0.3 + \frac{0.62\,\text{Re}_d^{1/2}\,\text{Pr}^{1/3}}{\left[1 + \left(\dfrac{0.4}{\text{Pr}}\right)^{2/3}\right]^{1/4}} F(\text{Re}_d) \tag{8.20}$$

The convection coefficient resulting from Equation 8.20 is likely to overestimate the heat transfer rate, since only part of the heating wire is in cross-flow with fluid motion. It is then corrected by a factor $h_{\text{ratio}} < 1.0$, to account for this effect. The main variables are then the wire diameter and the wire total length (Equations 8.17 and 8.19).

Air exit temperature is calculated from Equation 8.21.

$$q_h = \rho V_{\text{air}}(HW)c_{pa}(T_{\text{air,out}} - T_{\text{air,in}}) \tag{8.21}$$

Another variable is the heater wire type. Design may suggest multiple heater coils, N, each of a length l and a helix diameter d. Electric wire resistance R is found from the following relation:

$$q_h = E^2/R \tag{8.22}$$

where q_h is the power in watts and E is 110 volts. R is the resistance in ohms of the resistance wire which can be expressed in terms of resistance per unit length R' in Ω/m,

$$R = R'L_w \tag{8.23}$$

The front view of the heater is shown in Figure 8.9. Note that there is no solid surface inside the coil.

The heating element is a helical coil with an inner diameter d, with α being the angle of helix. The length of one coil is $\pi d/\cos\alpha$ (Figure 8.10). The pitch, l_p, is the axial distance between corresponding points on two successive turns,

$$l_p = \pi d \tan\alpha \tag{8.24}$$

A heating core of length l will have $n = l/(\pi d \tan\alpha)$ coils with a total wire length of

$$L_w = \frac{Nl}{(\pi d)\tan\alpha}\frac{(\pi d)}{\cos\alpha} = \frac{Nl}{\sin\alpha} \tag{8.25}$$

N = 2

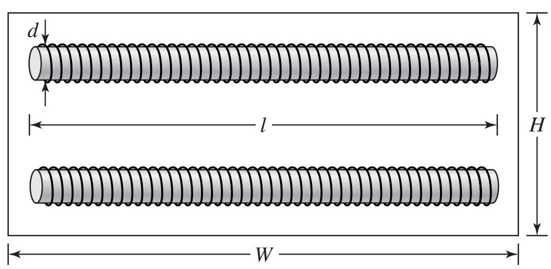


Figure 8.9 Electric heater geometry

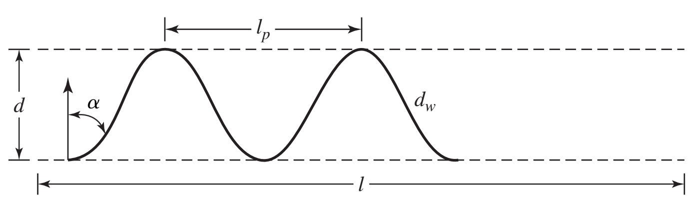


Figure 8.10 Heater core geometry

N is the number of heating elements recommended. The helix angle α is arbitrary. However, if one considers a minimum pitch $l_p = 2 \times d_w$, the helix angle α is governed by Equation 8.26.

$$\tan\alpha = \frac{2 \times d_w}{\pi \times d} \tag{8.26}$$

which will ensure sufficient distance between the two consecutive windings to prevent short circuiting. A higher pitch may be employed to reduce the risk of short circuit or to achieve a better optimization between the total wire length L_w and the length of the heating element l.

5. Parametric Study of the Heater Element The only design specification is the power input. The design variables are:

- Air inlet velocity V_{air}
- Wire temperature T_w
- Wire diameter (gauge) d_w

Heater overall dimensions W and H are dependent on heater element length l, helix diameter d, and the number of coils, N. The temperature of the enclosure, T_e, around the heater elements, and the convective heat transfer coefficient correction factor are yet to be chosen.

Air velocity V_{air} should be between 1 and 2 m/s to minimize draft and noise. For a heater with acceptably low noise level, a fixed air velocity of 1.25 m/s is chosen. With the design temperature rise of 10 °C and a maximum room air temperature of 20 °C, the air inlet temperature varies from 10 °C to 20 °C.

Nichrome is widely used for resistance wires. The resistances of per unit length of different diameters nichrome wires are given in Table 8.14 [CRC Handbook of Chemistry and Physics (2000)].

High wire temperature T_w results in small lengths and a compact arrangement, but requires careful attention to insulation and shield features. A lower range, around 200 °C, is safer but increases the size of the unit.

Air exit temperature depends on the air flow rate. Since the velocity is selected, flow rate is determined by the box dimensions W and H. The height H depends on the number of coils, N. For a coil bank, the center-to-center distance of coils is

Table 8.14 Resistance of nichrome wire

Gauge	Diameter d_w, mm	Resistance R', Ω/m
10	2.588	0.204
12	2.053	0.325
14	1.628	0.516
16	1.291	0.821
18	1.024	1.300
20	0.8118	2.080

chosen to be twice the coil diameter. We provide an allowance of 3 cm between the heating bank and the enclosure to accommodate the radiation shield and necessary wiring. Accordingly, we have the following equations for the dimensions in centimeters:

$$W = l + 6 \tag{8.27}$$

$$H = (2 \times N - 1) \times d + 6 \tag{8.28}$$

EES software is used for the computations. The procedure is as follows.

1. Select wire gauge and, hence, d_w.
2. Calculate the wire length L_w from Equations 8.22 and 8.23.
3. Calculate heater wire temperature T_w, using Equations 8.16, 8.17, 8.19, and 8.20, with appropriate correction factors both for shield temperature and convective heat transfer coefficient.
4. Determine air box dimensions W and H (Equations 8.27 and 8.28).
5. Solve for air exit temperature $T_{\text{air,out}}$ (Equation 8.21).

Temperature and convection coefficient correction factors T_{ratio} and h_{ratio} were both set at 0.8 for the initial analysis.

Since multiple power setting configuration is selected, it would be desirable to observe the impact of various heater banks at different power ratings. Current practice is to have two or three power levels. Two levels would require two heaters, each of 750 W, allowing operation at either 750 W or 1500 W. With one heating element at 500 W and a second one at 1000 W or three heating elements each at 500 W, the unit can be operated at three settings of 500 W, 1000 W, and 1500 W. Table 8.15 is a partial summary of an analysis performed with only one heater bank at a total power

Table 8.15 Heater wire gauge selection

Case	Power, W	Gauge	Dia., mm	R', W/m	N	d, mm	L_w, m	l, cm	W, cm	H, cm	T_w, °C	$T_{\text{air, out}}$, °C
1	500	20	0.812	2.08	4	8	11.63	18.8	24.8	11.6	175.7	31.5
2	500	20	0.812	2.08	4	10	11.63	15.0	21.0	13.0	175.7	32.1
3	500	20	0.812	2.08	4	12	11.63	12.5	18.5	14.4	175.7	32.4
4	750	18	1.024	1.3	4	8	12.41	25.2	31.2	11.6	216.6	33.7
5	750	18	1.024	1.3	4	10	12.41	20.2	26.2	13.0	216.6	34.5
6	750	18	1.024	1.3	4	12	12.41	16.8	22.8	14.4	216.6	35.1
7	750	20	0.812	2.08	4	8	7.76	12.5	18.5	11.6	368.9	43.1
8	750	20	0.812	2.08	4	10	7.76	10.0	16.0	13.0	368.9	43.8
9	750	20	0.812	2.08	4	12	7.76	8.4	14.4	14.4	368.9	44.0
10	1000	16	1.291	0.821	4	8	14.74	37.7	43.7	11.6	218.1	33.0
11	1000	16	1.291	0.821	4	10	14.74	30.2	36.2	13.0	218.1	34.0
12	1000	16	1.291	0.821	4	12	14.74	25.2	31.2	14.4	218.1	34.7
13	1000	18	1.024	1.3	4	8	9.31	18.9	24.9	11.6	368.7	42.9
14	1000	18	1.024	1.3	4	10	9.31	15.1	21.1	13.0	368.7	44.0
15	1000	18	1.024	1.3	4	12	9.31	12.6	18.6	14.4	368.7	44.6

rating indicated under "Power" column. The study provides a possible range of temperatures during operation and a list of acceptable choices.

From Table 8.15 it is clear that a heater with 500 W and 1000 W heating elements is not desirable as they require wires of different gauges to be consistent with heater width W. There is also considerable discrepancy in the heater coil temperature. This may not be a crucial issue, however. There are several acceptable choices for two or three heating bank designs such as cases from 1 to 9. Cases 1, 2, and 3 yield lower wire temperatures, reasonable height-to-width ratio, and permits a lower air outlet temperature, both resulting in greater occupant comfort. We should keep in mind that the last column in Table 8.15 indicates the outlet temperature for the power input specified in column 2. Since the increment is about 12 °C at 500 W, we would expect air outlet temperatures 24 °C and 36 °C higher for power ratings of 1000 W and 1500 W respectively. That would suggest that minimum and maximum temperatures of 22 °C and 46 °C for an indoor temperature of 10 °C, and 32 °C to 56 °C for an indoor temperature of 20 °C. We choose an electric space heater with three heater banks, each at 500 W.

The validity of some of the assumptions for h_{ratio} and T_{ratio} used in the analysis are now verified. Table 8.16 shows the results for case 6, with different values of these two variables where $W = 22.8$ cm, $H = 14.4$ cm, and $T_{\text{air,out}} = 35.1$ °C. It is clear from these results that the correct form of heat transfer correlation, such as Equation 8.20, is more critical for design then the exact value of radiation shield temperature enclosing the heating elements. It should be noted, however, that there will be no net change in the air temperatures leaving the unit.

It is also important to note that the helix angle to yield the wire pitch of $l_p = 2d_w$ seems to be quite appropriate for the design, and no further study is performed to establish a more suitable ratio.

Final design values are listed in Table 8.17. For the set of values in the table, heating wire temperature T_w is around 200 °C and air exit temperature $T_{\text{air,out}}$ is

Table 8.16 Effect of T_{ratio} and h_{ratio}

T_{ratio}	h_{ratio}	T_w, °C
0.8	0.6	281.7
0.8	0.7	244.6
0.8	0.8	216.6
0.8	0.9	194.9
0.8	1.0	177.4
0.6	0.8	216.4
0.7	0.8	216.5
0.8	0.8	216.6
0.9	0.8	216.8
1.0	0.8	217.0

Table 8.17 Design values for the heater

Rated power	1500 W
Number of heating elements	3 staggered
Power per element	500 W
Heater wire gauge	20 Nichrome
Total length of wire required	11.63 m/bank
Number of heater coils	4/bank
Coil diameter of heating element	12 mm
Length of heater coil	12.5 cm
Pitch of the coil	1.624 mm
Overall width of the heater box (W)	18.5 cm
Heater total width	20.5 cm
Overall height of the heater box (H)	14.4 cm
Heater total height	16.5 cm
Overall length of the heater box	25 cm
Air velocity	1.25 m/s

between 24 °C and 56 °C as discussed earlier, depending on the air inlet state and the power setting.

Fan Selection Fan selection is based on the flow rate and relevant pressure drop in the flow passages. Although this is not a complicated design, there is no well-established methodology to predict the pressure drop across helically wound wire heaters such as used here. The main assumption is that pressure drop across the heaters will be modeled as "pressure drop in tube banks" (Sec. 5.7.2). The other two losses considered are minor losses one for the inlet and one for the exit. Both exit and inlet sections of the heater open directly to the room through a safety screen with a loss coefficient of 1.0. Friction loss in the main flow passage inside the heater is neglected because of its very short distance and low air velocities. Overall power requirement will be increased by 100 percent to account for additional blockage for wiring and control modules.

Pressure drop over tube banks is calculated from Equation 5.30 and Figure A.4:

$$p_{\text{in}} - p_{\text{exit}} = n\chi \frac{\rho V_{\max}^2}{2} f \tag{8.29}$$

where a three-bank tube arrangement ($n = 3$) is considered. The values of χ and f are presented graphically (Figure A.4). $V_{\max}$ is the air maximum velocity over the tube bank where the area is smallest. However density variations due to high temperature in the region of the banks must be taken into account.

$$\rho_b V_{\max}(H \times W - 4 \times d \times l) = \rho_\infty V_\infty (H \times W) \tag{8.30}$$

ρ_∞ and ρ_b are densities at inlet and bank region, respectively. Maximum velocity is estimated at 2.0 m/s with a corresponding value of Re = 1000. The values of χ and f are 1.5 and 1.0, respectively. Pressure drop across the heater bank is

$$p_{\text{in}} - p_{\text{exit}} = 3 \times 1.5 \times \frac{1.0 \times 2.0^2}{2} \times 1.0 = 9.0\,\text{Pa} \tag{8.31}$$

Additional assumptions are $a = b = 2$. Since there is no clear procedure to calculate pressure drop across the screens, we assume it to be equal to 2 times the pressure drop across the tube banks both for inlet and exit. Total pressure drop with 100 percent safety factor is therefore about 54 Pa. Air flow rate across the fan is

$$Q = V_\infty(H_h \times W_h) = 1.25 \times (0.164 \times 0.205) = 0.042\,\text{m}^3/\text{s} \tag{8.32}$$

and the fan power required with an assumed fan efficiency of 65 percent is

$$P = \frac{\Delta p_{\text{total}} \times Q}{0.65} = 3.5\,\text{W} \tag{8.33}$$

However, considering the unaccounted pressure losses, we propose a 10-W fan.

The final design is shown in Figures 8.11 and 8.12.

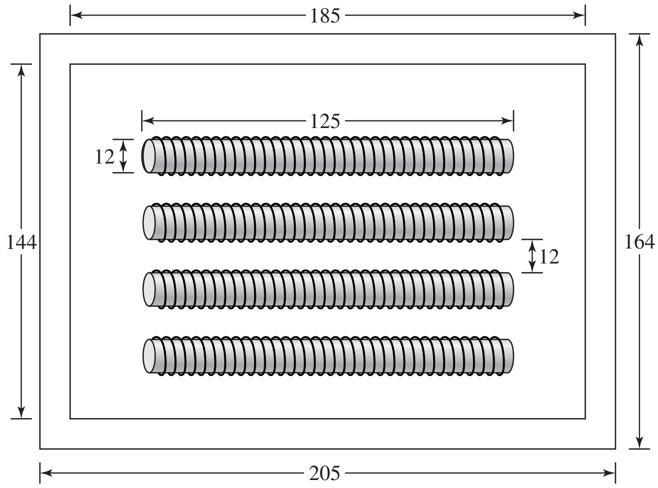


Figure 8.11 Front view (dimensions in millimeters)

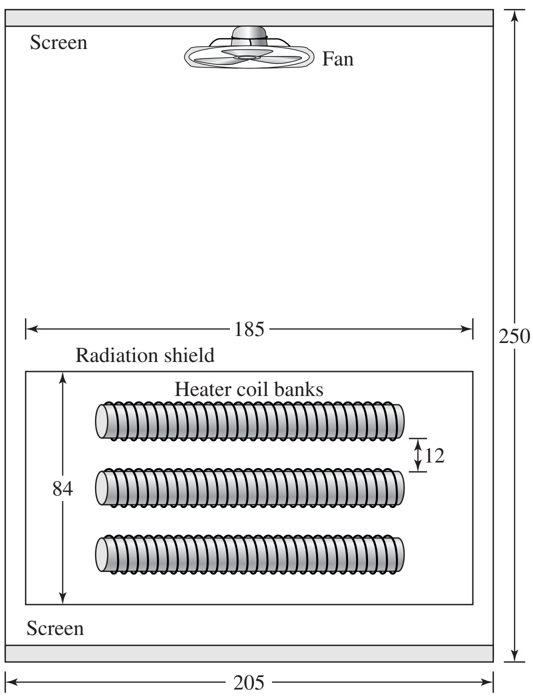


Figure 8.12 Top view (dimensions in millimeters)

8.9 Wind Tunnel

Wind tunnels are widely used in research labs, universities, and industry to study fluid flow and heat transfer phenomena. They provide valuable data on scale models in finding drag/lift forces on objects, surface pressure distributions, and enable visualization of fluid flow phenomena (boundary layer separation, flow over cylinders, etc.). Wind tunnels may be designed for research, for finding quantitative values to be used in actual design or production, or for instructional purposes. A wind tunnel for instructional purposes may not have all the sophisticated features that one may find in other wind tunnels.

Basic classifications of wind tunnels can be made on the basis of their architecture (open-circuit or closed-circuit) and prevailing air speed (subsonic, transonic, etc.). They can be built to operate at atmospheric pressure or at different pressures, in which case the entire tunnel is enclosed and the pressure controlled in the chamber. The essential components of a wind tunnel are a fan to achieve the desired flow and a duct with a test section. Other components such as a converging section and flow straighteners are incorporated to obtain uniform velocities in the test section at low turbulence level. A typical open-circuit wind tunnel is shown in Figure 8.13. In an open-loop wind tunnel, the air is drawn from the atmosphere at one end and discharged to the atmosphere at the other. In a closed-loop wind tunnel the gas is recirculated. The fan may be placed either at the inlet or exit of the test section. Placing the fan at exit has the advantage that turbulence caused by the fan blades is not propagated into the test section. Flow straighteners, generally a large number of tubes in conjunction with a converging section, are placed at the inlet end to reduce turbulence and to obtain a more uniform axial velocity.

8.9.1 Preliminary Specifications

An open-circuit bench top wind tunnel to be used in an undergraduate laboratory to demonstrate fluid flow phenomena, for flow visualization, and to determine drag coefficients is to be designed. It should be easy and economical to fabricate, functional, and should satisfy safety features.

8.9.2 Develop More Detailed Specifications

The unit should satisfy the following requirements.

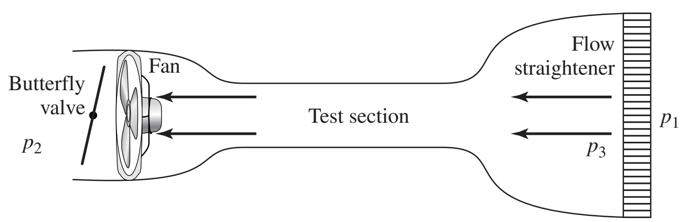


Figure 8.13 Schematic of a wind tunnel

- A typical bench-type unit has a length of about 2 m. Allowing for the converging inlet and the exit section to house the fan, a length of 2 m can accommodate a test section length of about 60 cm.
- Many tests require simulation of external flow. With an object in the test section, the flow can be considered as external flow and good results obtained if the blockage (defined as the ratio of the frontal area of the object to the cross-sectional area of the duct) due to the object is about 10 percent, and acceptable results with blockage as high as 20 percent. For a bench type wind tunnel, with a square test section with 30 cm on the side, a 10-cm-diameter sphere will result in a blockage of less than 9 percent. If 20 percent blockage is acceptable, the test piece diameter can be as high as 15 cm.
- The experiments to be conducted are, but not limited to, analyzing boundary layer and turbulent flow, determining forced convection heat transfer coefficient and friction drag over various geometric shapes, etc.

8.9.3 Develop Concept

Boundary layer flows over flat plates are considered to be laminar up to Reynolds numbers of about 500 000 and turbulent for higher Reynolds number. The drag coefficient for a relatively blunt object, such as a cylinder or a sphere, actually decreases when the boundary layer becomes turbulent. This behavior is observed for a range of Reynolds numbers (with the diameter as the characteristic length) between 1000 and about 600 000. The wind tunnel should be capable of achieving high enough Reynolds numbers to permit the students to observe such phenomena.

For air at atmospheric pressure and a typical laboratory temperature of 20 °C, $\nu = \mu/\rho = 1.5 \times 10^{-5}\ \text{m}^2/\text{s}$, and with a 10-cm-diameter object, an air velocity of 30 m/s in the test section yields a Reynolds number of 200 000; with a 15-cm-diameter object the Reynolds number is 300 000. To attain high Reynolds numbers in the range of 600 000 appears to be impossible for the type of wind tunnel in mind. The transition to turbulent flow about a flat plate generally takes place around $10^5 \leq \text{Re}_L \leq 10^6$. A test section 60 cm long can accommodate a flat plate about 50 cm in length. A free stream velocity of 30 m/s in the test section yields a maximum Reynolds number of 10^6.

For compressible flow effects to be negligible, the Mach number (ratio of the flow velocity to the sonic velocity) should be less than about 0.3. At a room temperature of 20 °C, from Equation 5.42 the sonic velocity a is

$$a = (kRT)^{0.5} = (1.4 \times 287 \times 293)^{0.5} = 343.1\ \text{m/s}$$

For the test section at an air velocity of 30.0 m/s, the Mach number is 0.09 and the flow can be treated as incompressible.

Even with a uniform velocity at the entrance to the test section, because of the boundary layers adjacent to the side plates of the test section (Figure 8.14), the core region where the velocity is uniform is less than the actual cross-sectional dimensions of the duct. The maximum boundary layer thickness occurs at the trailing end

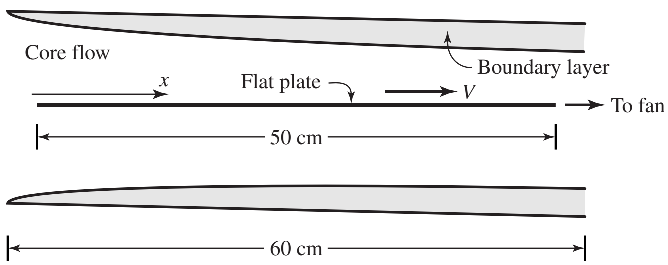


Figure 8.14 Boundary layer growth in the test section

of the test section. The flow outside the boundary layer is considered to be inviscid flow with a uniform velocity profile.

The boundary layer thickness, δ, is the distance from the plate at which the fluid velocity is within some arbitrary value of the upstream velocity, typically $0.99V$. Because of the velocity deficit within the boundary layer, the flow rate across the boundary layer thickness is less than that if the velocity were uniform at the free stream value (frictionless flow). If the frictionless flow is displaced by an amount δ^*, the boundary layer displacement thickness, the flow rate across each section will be identical. Denoting the Reynolds number for transition from laminar to turbulent boundary layer by Re_{cr}, for boundary layers over flat plates

$$\frac{\delta}{x} \approx \frac{5.0}{\mathrm{Re}_x^{1/2}} \qquad \frac{\delta^*}{x} \approx \frac{1.75}{\mathrm{Re}_x^{1/2}} \qquad \mathrm{Re}_x < \mathrm{Re}_{cr} \tag{8.34}$$

$$\frac{\delta}{x} \approx \frac{0.16}{\mathrm{Re}_x^{1/7}} \qquad \delta^* \approx \frac{\delta}{8} \qquad \mathrm{Re}_x > \mathrm{Re}_{cr} \tag{8.35}$$

$$\mathrm{Re}_x = \frac{\rho V x}{\mu}$$

8.9.4 Detailed Design

Having determined the preliminary and detailed specifications, we can proceed with the detailed design.

8.9.5 Boundary Layer Thickness in the Test Section

Assuming uniform velocity at the entrance to the test section, the maximum boundary layer thickness in the test section is at the trailing edge. Corresponding to the maximum velocity of 30 m/s, the Reynolds number is 10^6 and from Equation 8.35, the boundary layer displacement thickness is 1.4 mm. Therefore, the core region available with a 30-cm-square test section is not significantly altered. The effect of the reduced core should be taken into account in designing the scaled models to satisfy exactly the blockage ratio requirements.

8.9.6 Flow-Straightening Screen

Flow straightening is necessary to ensure uniform air flow across the test section. It can be achieved in a number of ways, with a converging section, a number of fine meshed screens, or a number of small-diameter tubes, or some combination of those. Two configurations of a combination of a converging section and tubes are shown in Figure 8.15.

The inlet section is much larger than the test section, permitting the air to settle at a smaller velocity and leading to a smooth transition into the tubes. In option A, the tubes are placed in the converging section. In option B the tubes are in an elongated part of the test section.

Square-Tube Flow Straighteners The inlet duct may be of square cross section with flow straighteners of square or circular cross section. First we consider a square converging section with square tubes. The thickness of the tubes will be assumed to be 1 mm. From conservation of mass for an incompressible fluid, referring to Figure 8.16

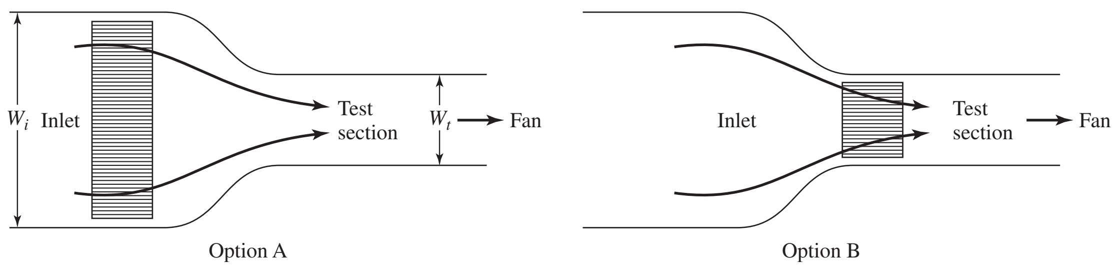


Figure 8.15 Wind tunnel configuration

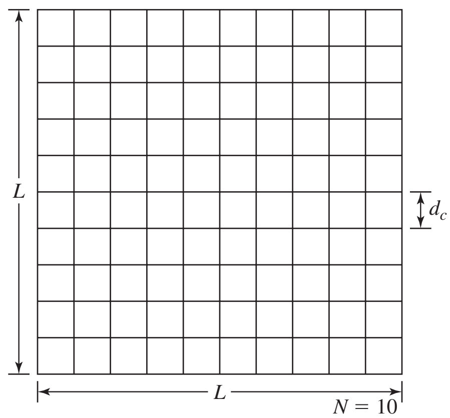


Figure 8.16 Flow straighteners screen configuration

we have,

$$\text{Net area of one cell: } a_c = \{[L - (N-1) \times t]/N\}^2$$
$$\text{Hydraulic diameter: } d_{hc} = [L - (N-1) \times t]/N$$

where

L = width of the convergent section
N = the number of cells
t = wall thickness

For the square test section with $L_{\text{test}} = 30$ cm on one side, for an incompressible fluid,

$$V_{\text{test}} \times L_{\text{test}}^2 = V_{\text{cell}} \times N^2 \times a_c \tag{8.36}$$

The entrance length L_e in internal flows is estimated as

$$\frac{L_e}{d_{hc}} \approx 0.06\,\text{Re}_d \qquad \text{Re} < \text{Re}_{\text{cr}} \tag{8.37}$$

$$\frac{L_e}{d_{hc}} \approx 4.4\,\text{Re}_d^{1/6} \qquad \text{Re} > \text{Re}_{\text{cr}} \tag{8.38}$$

The Reynolds number is based on cell hydraulic diameter d_{hc}. The accepted design value for tube-flow transition is taken to be $\text{Re}_{\text{d,crit}} \approx 2100$.

Friction head in an internal flow is calculated from Equation 5.34,

$$h_f = c_M \frac{L}{d_{hc}} \frac{V^2}{2g} \tag{8.39}$$

From Equations 5.35 and 5.37,

$$c_M = \frac{64}{\text{Re}_d} \qquad \text{Laminar flow} \tag{8.40}$$

$$c_M = \left\{0.7817 \ln\left[\frac{6.9}{\text{Re}_d} + \left(\frac{\varepsilon/d}{3.7}\right)^{1.11}\right]\right\}^{-2} \qquad \text{Turbulent flow} \tag{8.41}$$

From Equation 5.34 the friction pressure drop in the direction of flow, Δp, is given by

$$\Delta p = \rho g h_f \tag{8.42}$$

Referring to Figure 8.13, the pressures at the inlet to the duct, p_1, and exit of the fan, p_3, are atmospheric. The pressure drop $\Delta p = p_1 - p_2 = p_3 - p_2$. However, the pressure drop from Equation 8.42 does not include additional pressure losses that may be caused by the sudden contractions and expansions. The fan power is obtained from,

$$P = \Delta p Q = \rho g Q h_f \tag{8.43}$$

where Q refers to volumetric flow rate.

Table 8.18 Square-tube flow straightener

N	W_t, cm	W_i, cm	L_e, cm	Δp, Pa	P, W
10	30	30	28	1084	4508
20	30	30	13	1161	4828
30	30	30	8	1268	5272
15	30	45	28	757	3147
30	30	45	13	776	3226
45	30	45	8	803	3336
20	30	60	28	699	2905
40	30	60	13	706	2935
60	30	60	8	716	2976

There are several additional head losses h_L (minor losses) that must be accounted for. The procedure is to consider a loss coefficient K and evaluate the equivalent head loss h_m,

$$h_m = K\frac{V^2}{2g} \tag{8.44}$$

The exit section of the tunnel opens directly to the room. A loss coefficient of 1.0 is considered reasonable for that section. The loss coefficients for the conical nozzle before the test section entrance (when in use) and tunnel inlet are assumed to be 0.2 and 0.5, respectively. As the total cross-sectional area of the tubes is nearly equal to the cross-sectional area of the enclosing duct, the losses due to entrance and exit from the tubes are neglected.

Table 8.18 shows the results for flow straighteners of square cross section for a few cases, with W_i and W_t representing the widths of the inlet and test section respectively. L_e is the required entrance length to achieve fully developed flow. The last two columns are equivalent pressure drop in Pa, and the fan power P. Fan-motor efficiency of 65 percent has been used. The top three rows are for flow straightener in option B in Figure 8.15. The remaining rows are for the tubes in the converging section, option A. Several different dimensions of the converging section are considered in the computations.

Circular-Tube Flow Straighteners Another alternative for flow straighteners is cylindrical tubes as shown in Figure 8.17. As shown in the enlarged view, tubes and the space enclosed by a four-tube cluster (shaded area) present different resistances to flow; as the pressure drops in the two parts are equal, the velocities in the two sections are different.

Area and hydraulic diameters for the tube and space between the tube clusters, respectively, are given in Equations 8.45 and 8.46.

$$a_t = \pi d_i^2/4 \qquad d_H = d_i \tag{8.45}$$

$$a_c = d_o^2\left(1 - \frac{\pi}{4}\right) \qquad d_H = \left(\frac{4}{\pi} - 1\right) d_o \tag{8.46}$$

Here d_o and d_i are tube outer and inner diameters.

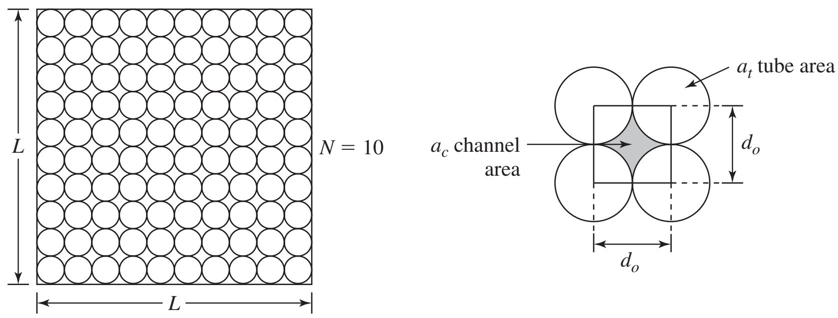


Figure 8.17 Flow straightener screen configuration

Table 8.19 Circular-tube flow straighteners

N	d_o, cm	W_t, cm	W_i, cm	L_e, cm	Δp, Pa	Power, W
20	1.5	30	30	14.0	1132	4753
30	1.0	30	30	9.0	1171	4915
60	0.5	30	30	4.0	1314	5516
30	1.5	30	45	14.0	776	3257
45	1.0	30	45	9.0	786	3299
90	0.5	30	45	4.0	854	3583
40	1.5	30	60	14.0	716	3004
60	1.0	30	60	9.0	721	3026
120	0.5	30	40	4.0	741	3108

The velocities in the two parts are calculated by iteration such that mass conservation is satisfied and the pressure drops are equal. Results of analysis similar to that for the square tubes are shown in Table 8.19. Tubes are of plastic material with fixed thickness of 1 mm as in the case of square-profile straighteners. The last row is for 5-mm-diameter straws (used for drinking soft drinks), for which the thickness was assumed to be 0.1 mm.

L_e is the entrance length to achieve fully developed flows. The last two columns are the pressure drop (Pa) and the fan power (W) respectively.

8.9.7 Conclusions

Although a more compact tunnel can be achieved with option B in Figure 8.15, the power required may be significantly higher in most cases than with option A. Another issue is the entrance length to achieve fully developed flow in the test section. Since laminar flow is no longer possible unless design dimensions are increased or test section velocity is reduced, using the recommended entry length is no longer a critical issue in the design and is ignored. The selection of the flow straightener will depend on the user and the availability of the material. Straws are easily available and are inexpensive. Therefore, a circular converging section with straighteners

Table 8.20 Final design of wind tunnel

Test section width	W_t	30 cm
Test section height	H_t	30 cm
Test section length	L_t	60 cm
Inlet section width	W_i	60 cm
Inlet section height	H_i	60 cm
Number of straightener tubes	N	14 400
Straightener tube outer diameter	d_o	5 mm
Straightener tube length	L_e	4 cm
Straightener tube thickness	t	0.5 mm
Fan power required	P	3400 W

made out of straws is recommended. Fan power was increased by 10 percent to accommodate some unforeseen minor losses.

8.9.8 Final Design

The recommended design, option A, in Figure 8.15 is given in Table 8.20. A butterfly valve (or louvres) is used at the exit of the fan (Figure 8.13) to control the flow rate so that velocity-dependent experiments may be performed.

Although constructional details are not a necessary part of the thermal design, a few suggestions are made for the fabrication of the unit. The test section should be made of 6-mm-thick, clear polycarbonate plastic so that flow visualization is easily accomplished. The converging section and fan enclosure may be made of either painted sheet steel or stainless steel. Instruments should be provided for the different experiments.

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PROBLEMS

8.1 It is required to remove moisture from air containing water droplets. First, the droplets down to a diameter of 25 μm are to be removed in a centrifugal separator before the rest of the moisture is removed by another process such as passing it through a desiccant. Propose a design for the centrifugal separator with air at 200 kPa, 20 °C flowing at a mass flow rate of 0.25 kg/s of dry air.

How would the design be different if solid particles down to 0.1 μm are to be removed from heavy fuel oil with throughput of 0.08 kg/s?

At very low Reynolds number, the viscous drag force on spherical objects is given by Stokes law as $C_D = 24/\text{Re}$, with the drag coefficient C_D is defined as

$$C_D = \frac{2F}{\rho V^2 A}$$

where

F = viscous drag
ρ = fluid density
V = fluid velocity relative to the particle
A = frontal area, πR^2 for a sphere

Stokes law is applicable for very low Reynolds number of about 1 or less. F. M. White in, *Viscous Fluid Flow,* McGraw-Hill: New York (1971), gave the following expression for a wider range of the Reynolds number:

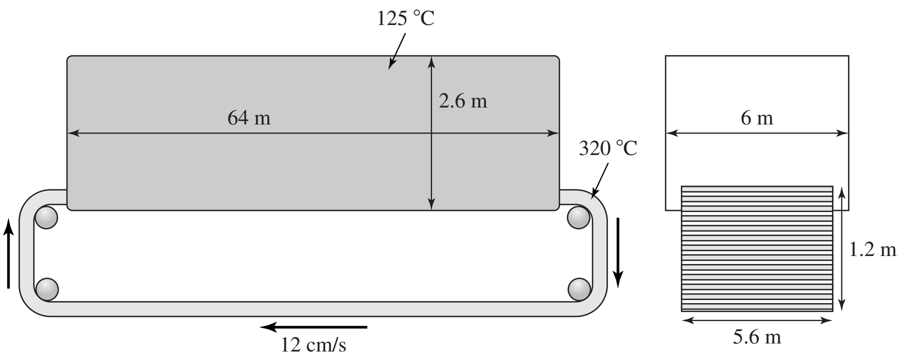
$$C_D = \frac{24}{\text{Re}} + \frac{6}{1 + \sqrt{\text{Re}}} + 0.4 \qquad 0 < \text{Re} < 2 \times 10^5$$

8.2 Design a 100-W hydraulic turbine with water available at a height of 5 m.

8.3 Design a viscometer to determine the viscosity of olive oil. Students research and study the relationship between viscosity and flow or force in different situations (force on a sphere, flow in a capillary tube, etc.) and design a meter.

8.4 Heavy fuel oil is to be pumped in a long pipe at a mass flow rate of 35 kg/s. As the viscosity of the oil is high, the required pumping power with oil from the storage tank maintained at 40 °C is also high. To reduce the pumping power (leading to a pump with a less power for the driving motor) it is proposed to heat the oil to 80 °C with saturated steam available at 500 kPa. The oil velocity is not to exceed 1.5 m/s. Design a heat exchanger to heat the oil from 35 °C to 80 °C. Properties of the oil: density = 920 kg/m^3, viscosity at 35 °C = 0.397 N · s/m^2, c_p = 885 J/kg · K.

8.5 A gas-fired pellet preheating furnace for processing iron ore is 64 m long, 6 m wide and 2.6 m high (see figure). The pellets are transported in the furnace by a 5.6-m-wide traveling grate, moving at 12 cm/s. The material of the grate is mainly stainless steel with a mass of 2600 kg/lineal meter. The mass of the pellets is much less than that of the grate. The grate exits the furnace at 320 °C and re-enters the furnace at the other end as shown in the figure. The side and top plates are at an average temperature of 125 °C and the ambient air is at 35 °C. To reduce the gas consumption and to make the work place more comfortable, suggest a method to reduce the heat transfer rate to the surroundings. Determine the reduction in the gas consumption as a result of the reduced heat transfer rate.



8.6 A test chamber, 50 cm × 50 cm × 50 cm, is to be maintained at −50 °C. Cooling is obtained by expanding high-pressure air to a lower value through a nozzle. The proposed system contains a compressor, air- (or water-) cooled heat exchanger, and a nozzle to supply the air to the test chamber. A modest velocity of the air in the test space and a pressure slightly higher than atmospheric pressure are acceptable. Design the test chamber. Determine the basic specifications and design features of the components.

8.7 Instead of a wind tunnel, design a water table to demonstrate fluid flow phenomena such as flow around bodies and drag force. The shallow, open-channel water table is to study boundary layer development on a flat plate. The water table should be capable of studying both laminar and turbulent boundary layer phenomena. What are the advantages and disadvantages of a water table compared with a wind tunnel?

8.8 Design a device where wet steam (quality of approximately 90 percent) entering the device exits as dry, saturated vapor at the same pressure. The device must be self-operating with low annual maintenance.

8.9 Choosing a site, design a flat-plate solar collector with finned tubes. Determine the number of collectors to heat 3 kg/s of water by 10 °C. Collector inlet temperature varies from 20 °C to 50 °C. The collectors should perform satisfactorily during May.

8.10 Choosing a site, design a concentrating solar collector to heat 3 kg/s of water by 40 °C. Collector inlet temperature varies from 20 °C to 80 °C. The collectors should perform satisfactorily during May.

8.11 Design a lubricating oil cooler to cool 0.2 kg/s of lubricating oil from 70 °C to 50 °C. Water is available at 15 °C and air at 30 °C. Discuss the advantages and disadvantages of the different options for the heat exchanger.

8.12 With groundwater available at 10 °C, a heat pump is to supply 10 kW of energy to heat a building. The building heating water enters the condenser of the heat pump at 40 °C and exits at 60 °C.

8.13 The exhaust gas from a 10 000-kW diesel engine is to be used to heat water at 700 kPa from 80 °C to 120 °C. The exhaust gases are available at 300 °C. Determine the flow rate of water and design the heat exchanger to accomplish the output.

8.14 The exhaust gas from a 20 000-kW net output gas turbine is to be used to produce steam for supplementing power production. The pressure ratio in the turbine is 10 and the maximum temperature in the unit is 1200 °C. Design the boiler.

8.15 Design the intercooler for a two-stage compressor with a capacity of 0.5 m^3/s at an exit pressure of 2 MPa. Determine the intermediate pressure for the least work and assume that cooling water is available at the same temperature as the ambient air.

8.16 Operating between pressures of 100 kPa and 800 kPa, a gas turbine produces 5000 kW of net energy. Design a regenerator for the gas turbine. Maximum combustion gas temperature is 1200 °C.

8.17 The electric space heater designed in Section 8.9 was based on heat transfer correlations used for fluids in cross-flow over cylinders. Another design for the heating element is a bank of ceramic rods with electric resistance wire wound around them. The rods may be considered to be at uniform temperature. Redesign the space heater for the same power rating but using correlations for tube banks.