Piping Design Considerations in a Solar-Rankine Power Plant

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Two of the main parameters in sizing the piping of a solar power plant are presented. These two parameters are the working pressure of the vapor leaving the solar collectors, and the type of working fluid used. Numerical examples for each case are given using the graphical Moody friction charts and the analytical Darcy-Weisbach equation. Different working pressures of steam vapor in the solar collector-turbine pipe connection indicate their major role in the design. The size variation was found not to be in linear proportion to vapor density variations. On the other hand, high molecular weight organic fluids such as R-11 and R-113, when compared with water, have shown insignificant changes in piping sizes.

I. Introduction

The major factors associated with the proper design of the piping system in a solar-driven power plant are the selection of the working fluid itself, allowable pressure losses, and the temperature levels in the cycle. The type of working fluid in the thermodynamic cycle could be either water or, for reasons to be explained later, an organic fluid similar to that used in refrigeration machines. Examples of these organic fluids are refrigerant R-113, as used in a prototype described in Ref. 1, or refrigerant R-11, which is used in Ref. 2. The allowable pressure losses in the lines are usually limited by the designer to keep the pumping power to a minimum value. The upper temperature limit in the power cycle depends on the type of solar collector used. It varies from 90°C for flat plate collectors to more than 160°C for medium concentration types. The lower temperature limit in the power cycle varies with the ambient temperature and the type of cooling medium used in the condenser.

Rather than the technical consideration of the thermodynamic cycle and its performance characteristics, this study is related to the size parameters encountered in the piping design only. The economic feasibility of building a solar-driven power plant depends on the installation cost, which in turn depends on the size of the various plant components and their connecting piping, which constitutes a large percentage of the installation cost. Therefore, a thorough investigation is needed to study the relative importance of piping in the plant cost analysis.

The study is divided into two main sections. The first section is a comparative piping design between a solar-powered power plant, with its low-pressure and low-temperature capability of producing steam, and the opponent case, the fuel-powered, high-pressure and high-temperature conventional power plant. The working fluid will be water in both cases. The second section is a comparison of the piping design when water is used versus an organic substance at the same operating conditions of a given power plant.

II. Friction in Pipes

The selection of a pipe size to transmit a fluid from one end to another, as shown in Fig. 1, depends on the following parameters:

(1) Type of the flowing fluid.

(2) Initial operating pressure.

(3) The allowable pressure drop or the allowable friction rate. The friction rate is defined as the pressure drop per unit equivalent pipe length.

(4) Total equivalent pipe length in the longest run, adding the effects of valves, bends, fittings, etc.

(5) The mass flow rate through the pipe.

Many charts and tables are available in the literature to help the designer select the proper pipe diameter most suitable for the job, with specific information on the above five parameters. The data given in Refs. 3 and 4 were used throughout this work.

The basic formula used to construct a pipe selection chart is the Darcy-Weisbach formula, expressed as

$$\frac{\Delta P}{L} = \frac{f}{d} \cdot \frac{U^2}{2} \cdot \rho \tag{1}$$

where

P = the end-to-end pressure drop along the pipe, N/m^2

L = the equivalent pipe length, m

d = the inner pipe diameter, m

U = the mean flow velocity, m/s

ρ = the flow density, kg/m^3

f = the friction factor, dimensionless

Equation (1) applies for either laminar or turbulent flow. The value of (f) depends on whether the flow is laminar or turbulent, and it is in general a function of the Reynolds number (R_e) and the relative roughness. The friction factor (f) is usually plotted graphically in what is called "Moody Friction Charts". For convenience, an analytical expression has been made to accurately calculate the friction factor for the special case of smooth walled circular tubes, known as the Karman-Nikuradse equation, as follows:

$$\begin{aligned} f &= \frac{64}{R_e} && R_e \leqslant 2000 \text{ (laminar)} \\ f &= 0.316\, R_e^{-0.25} && 10^5 \geqslant R_e > 4000 \text{ (turbulent)} \\ f &= 0.184\, R_e^{-0.20} && R_e > 10^5 \text{ (turbulent)} \end{aligned} \tag{2}$$

where $R_e = \rho\, Ud/\mu$ (dimensionless) and μ is the dynamic viscosity of the fluid in kg/(m · s). Equation (2) is used throughout this article with occasional checking with the graphical solutions given in Refs. 3 and 4 when the pipe roughness is considered.

According to the continuity equation, the mass flow rate $\dot{m}$ is given by

$$\dot{m} = \frac{\pi}{4} d^2 \cdot U \cdot \rho \text{ Kg/s} \tag{3}$$

The pump power needed to overcome the friction losses in the pipe, Fig. (1), can be written as

$$\text{friction power} = \dot{m} \left(\frac{\Delta P}{\rho}\right) \cdot \frac{1}{\eta_p}, \text{ watts} \tag{4}$$

where η_p is the pump efficiency.

III. Piping Design

This part is divided into two sections: the first is a study of the effect on the piping size of varying the initial steam pressure in the steam power plant; the second deals with the effect of varying the type of working fluid on the piping size. Each section is handled separately with its different assumptions.

A. Effect of Varying the Initial Pressure on Sizing a Steam Pipe

The necessity to work with steam in solar-powered engines at pressures close to atmospheric is due to the limited capability of low-concentration ratio collectors to obtain high-saturation temperatures. Since the specific volume of dry saturated vapors is inversely proportional to the temperature or the pressure, the effect of operation at low pressure or low temperature on the piping design has to be studied for its size impact.

Let us visualize a power plant whose boiler section is located at some distance (L) from the turbine-condenser, as shown in Fig. 2, and is capable of producing steam in a dry and saturated condition, at the desired pressure. The mass flow rate ($\dot{m}$) through the lines is kept unchanged, independent of the boiler pressure. Also, since the condenser temperature and pressure are dependent only on the cooling medium temperature, which is kept constant, the feed water line need not be resized when the boiler pressure is changed. The only piece of piping that is dependent on the boiler pressure is that connection from the boiler to the turbine. This steam line always needs a heavy thermal insulation and, because of its large size, it is the most expensive piece of piping in the power plant. This piping cost is proportional to its diameter, and constitutes a major part of the installation cost.

The following numerical values were used to construct Table 1 as an example for the study:

steam pressure range	10 to 148 N/cm^2 absolute
condition of steam at boiler exit	dry and saturated
mass flow rate $\dot{m}$	2268 kg/h (5000 lb/h)
maximum allowable friction rate, ($\Delta P/L$)	4.5 N/cm^2 per 100 m (2 psi/100 ft)
condensation temperature	40°C

Selection of the above friction rate ($\Delta P/L$) was based on the practice of designing steam pipes and is kept in the range of, but not exceeding, 4.5 N/cm^2 per 100 m for a first design trial. The following remarks can be abstracted from Table 1:

(1) The largest pipe diameter required to transmit the flow at different pressures is proportional to the specific volume, but not in linear form. For instance, a 12.7-cm (5-in.) nominal diameter pipe is sufficient to carry the steam at atmospheric pressure (10.13 N/cm^2) while a 7.62-cm (3-in.) nominal diameter pipe is required to carry the same flow rate at the same maximum allowable friction rate at a pressure of ~ 148 N/cm^2 (~ 14 atmospheres). Although the specific volume decreased by a factor of 12.4 to 1 from its value at atmospheric pressure, the necessary pipe diameter has decreased by a factor of 5/3 or 1.67 to 1 only. In Fig. 3, the selected pipe diameter is plotted versus the initial steam pressure. The dotted line does not represent any curve fitting to the points abstracted from Table 1, but rather shows how steep the change is at the low pressure range from 10 N/cm^2 to 45 N/cm^2. The main difference between the rate of decrease of the specific volume and the rate of decrease of the pipe diameter is due to the steam velocity that has acquired another rate of decrease at higher pressures to keep the pressure lost in friction below the maximum permissible rate listed.

(2) The shaft power needed to support the flow against the pipe friction, as given by Eq. 4, indicates that the larger the specific volume, the larger the friction power will be for the same mass flow rate and pressure drop. This is evidenced in Table 1, where the friction power is equal to 40.5 kW/100 m for 10.13 N/cm^2 (atmospheric pressure) (No. 1 in Table 1) versus 3.3 kW/100 m for 148 N/cm^2 (14 atmospheres) (No. 4 in Table 1). On the other hand, higher steam pressures imply a higher enthalpy potential difference in the turbine nozzles, and more work output from the power plant. For example, for dry and saturated (d.s.) steam at 10 N/cm^2 pressure, a 75 percent turbine isentropic efficiency, and a condensation temperature of 40°C, the example plant can produce 180 kW compared with 377 kW when d.s. steam at 148 N/cm^2 pressure is used.

It appears from the above that if a hypothetical plant is constructed with an equivalent pipe length of 100 m between the boiler section and the turbine, the friction horsepower in the first case of Table 1 would represent 22 percent of the turbine output, but only 1 percent of the turbine output for the 4th case in Table 1. The piping designer might then search for another criterion to compare the pipe sizes since this comparison based on the same friction rate (or same pressure drop) might prove satisfactory for steam heating purposes, but not suitable for power plants.

(3) An alternative procedure that might be considered for comparing the sizes of steam pipes in steam power

plants is to keep the ratio of friction power to turbine power within a certain percentage value. For example, if the ratio in case 4, Table 1, proved to be an acceptable friction power ratio, i.e., ~ 1 percent of the turbine power for every 100-m equivalent length, then case 1, Table 1, has to change its friction power from 40.5 kW/100 m to 1.8 kW/100 m. This last step necessitates an increase in the nominal pipe diameter from 12.7 cm (5 in.) to 25.4 cm (10 in.), as found from the friction chart, Ref. 3. The friction rate is then dropped from 2.8 N/cm^2 per 100 m to 0.09 N/cm^2 per 100 m, and the friction power drops from 40.5 kW/100 m to 1.3 kW/100 m and the ratio of the pipe friction power to the turbine power becomes 1.3/180 or 0.7 percent per 100 m.

A summary of the above findings follows: If for two steam power plants the piping connecting the boiler to the turbine is compared where one plant is working with dry and saturated (d.s.) steam at 10 N/cm^2 (atmospheric pressure) versus 148 N/cm^2 (14 atmospheres) for the other, then:

(1) The specific volume of steam in the first plant is 12.4 times the specific volume of steam in the second.

(2) The largest nominal diameter yielding the same maximum allowable friction rate ($\Delta P/L$) is 12.7 cm (5 in.) for the first and 7.62 cm (3 in.) for the second, a diameter ratio of 1.67 to 1.

(3) The largest nominal diameter yielding the same maximum allowable ratio of friction horsepower to turbine power is 25.4 cm (10 in.) for the first and 7.62 cm (3 in.) for the second; a diameter ratio of 3.33 to 1. In both cases, the diameter ratio of (2) and (3) is not the same as the specific volume ratio.

B. Effect of working fluids other than water

The idea of using Rankine power cycles with working fluids other than water was initially put under investigation to benefit the automotive industry, and only recently has attention been directed to coupling them with solar energy. A number of studies (for example, Ref. 5) have shown that the use of high-molecular-weight fluids, ranging from 60 to 300, leads to compact turbomachinary for low-power levels below 500 kW. Theoretically, a fluid vapor that has a high molecular weight has 3 properties: (1) a small specific volume, (2) a small isentropic expansion enthalpy drop, and (3) a small sonic velocity. The second property requires a large mass flow increase for a given power output, which may be offset by the first property. Also, the second property creates low nozzle velocities, low blade-tip speeds, and thus low rotational speed, a characteristic that is desirable for small units. The third property may make the flow in the blade passage supersonic, which is complicated. Whether high-molecular-weight fluids are superior in performance or not, it is felt that more work is needed specifically for solar-powered devices.

In this section, a comparison is made between two power plants working on the same thermodynamic cycle, which is chosen to be the Rankine cycle, the same temperature limits, and having the same net power output, but employing two different working fluids. In one case, water is used and in another case either one of the organic fluids, refrigerant R-11 or refrigerant R-113 is used. These two organic fluids were claimed by some researchers (Refs. 1 and 2) to be superior to water, both in performance and economics, for operation in low-temperature power cycles. The reason behind making this comparison is to justify the large or small differences, if they were found, between any of these two organic fluids versus water in a solar-Rankine power cycle. The following parameters were kept unchanged in a numerical example:

(1) Thermodynamic cycle: simple Rankine with no superheat.

(2) Working fluids: water, R-11, R-113.

(3) Condition of fluid entering the turbine: saturated vapor.

(4) Condition of fluid entering the pump: saturated liquid.

(5) Net power plant output: 100 kW_e.

(6) Electric generator efficiency: 90 percent.

(7) Evaporation temperature: 100°C (212°F).

(8) Condensation temperature: 40°C (104°F).

(9) Isentropic efficiency of compression or expansion processes: 75 percent.

The results of the thermodynamics of the cycle are given in Table 2. The following remarks are made from Table 2:

(1) The thermal efficiency of the Rankine cycle appears to have a weak relationship with the type of working fluid or the pressure range in the cycle. It is also evident from Table 2 that working with water produced a 10 percent increase in the thermal efficiency compared with refrigerants R-11 and R-113. This result is simply due to the large latent heat of vaporization that water possesses over any other fluid. The heat added in the Rankine cycle consists of two parts: a sensible heat

part at constant pressure, and a latent heat part at constant temperature and pressure. Therefore, the isothermal portion of the total heat addition is larger for water than for any other fluid. The rankine cycle will approach the idealizations set by Carnot's principle as more and more heat is added or rejected in reversible isothermal processes.

(2) For a specific power output, such as 100 kW_e output in the example, the mass flow rate in the case of water is 1/13.74 times that required for R-11, or 1/14.88 times that required for R-113. Again, this small water flow is due to the extremely large latent heat of vaporization for water compared to R-11 or R-113. At first glance, it may appear to the designer that water piping can be smaller in size than the comparative organic fluids. On the other hand, water vapor has a very large specific volume compared to R-11 or R-113. For example, the specific volume of dry and saturated steam at the turbine inlet is 71.7 times that for R-11 vapor, or 50.1 times that for R-113 vapor. However, the designer will find that the water vapor discharge (mass flow × specific volume product) is only 5.23 times that for R-11 vapor, or 3.37 times that for R-113 vapor. The only key left in designing the piping for each fluid is the allowable friction loss as described below.

(3) The design of steam piping connecting the boiler to the turbine could first be made using the data in Table 2 and the friction charts of Ref. 3 as follows. Taking the mass flow rate of 1388 kg/h (3060 lb/h) of dry and saturated steam at atmospheric pressure from Table 2, and an arbitrary maximum friction rate of 0.34 N/cm^2 per 100 m (0.15 psia/100 ft), the largest nominal pipe diameter is found to be 20.32 cm (8 in.). This corresponds to an inner diameter of 20.272 cm (7.981 in.), a flow velocity of 19.985 m/s (3934 ft/min), a Reynolds number of 202,740 and a friction factor (f) for smooth pipes of 0.016. The result of this selection is a friction rate of 0.10 N/cm^2 per 100 m, and a friction power rate of 0.86 kW/100 m for a smooth pipe or 1.2 kW/100 m if 40 percent allowance is made due to pipe roughness, resulting in a friction power on the order of 1 percent of the turbine power output for every 100 m of equivalent pipe length.

(4) Tables 3 and 4 were constructed for refrigerants R-11 and R-113, respectively, using the data in Table 2 and Eqs. 1, 2, 3 and 4. The following findings are made:

   (a) When the maximum permissible friction rate ($\Delta P/L$) was the same for all the fluids concerned, the size of the pipe connecting the boiler to the turbine was found the same and equaled 20.32 cm (8 in.) for the 100 kW_e power plant example.

   (b) If, on the other hand, the design was based on an equal ratio of friction power to total plant power (or equal friction power in the case of comparing two plants of the same power output), the design pipe diameter was found equal to 15.24 cm (6 in.) for R-11 vapor versus 20.32 cm (8 in.) for both water vapor and R-113 vapor. This corresponds to a friction power rate of less than or equal to ~1 percent of the total plant power per 100-m equivalent length

IV. Conclusions

Comparisons made were based on the assumption that the most expensive piece of piping in the power plant was that piece connecting the solar collectors (or boiler) to the turbine section. The following conclusions can be reached:

(1) Pipe size differences due to different working pressures can be a major factor in the design, while pipe size differences due to different working fluids are minor ones.

(2) The points in favor of using water as the working fluid in power plants over the selected organic fluids are numerous. From the point of view of thermodynamic behavior in the cycle, cost, nonflammability, and nontoxicity, water is the superior fluid.

References

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Table 1. Sizing a steam pipe at different initial pressures with constant flow and friction rates

Case no.	Initial pressure, N/cm^2 (psi) absolute	Specific volume ν, m^3/kg	Largest[a] nominal diameter, cm (in.)	Inner[b] pipe diameter, cm	Flow velocity[c] U, m/sec	Viscosity μ, kg/m · h	Friction[d] factor f (smooth pipes)	$(\Delta P/L)$ calculated for smooth pipes, N/cm^2/100 m	$(\Delta P/L)$[a] working value, N/cm^2/100 m	Friction power,[a] kW/100 m	Total plant power with the same flow rate,[e] kW	Friction power to total plant power ratio per 100 m
1	10.13 (14.7)	1.673	12.7 (5)	12.819	81.64	0.043	0.0132	2.0	2.8	40.5	180	0.22
2	20.48 (29.7)	0.858	10.16 (4)	10.226	65.82	0.0457	0.0128	3.0	4.2	30.8	233	0.13
3	44.61 (64.7)	0.416	8.89 (3 1/2)	9.012	41.08	0.0493	0.0127	3.0	4.2	14.9	289	0.05
4	148.03 (214.7)	0.135	7.62 (3)	7.793	17.83	0.0553	0.0126	2.0	2.8	3.3	377	0.01

[a]Taken from steam pipe charts, Ref. (3).

[b]Taken from pipe standards Tables, Ref. (3).

[c]Using Eq. 3 this article.

[d]Using Eq. 2 this article.

[e]Based on condenser temperature of 40°C and turbine isentropic efficiency of 75 percent.

Table 2. Comparison between water and organic fluids R-11 and R-113 in the performance of a solar-Rankine engine

Type of working fluid	Water	R-11	R-113
Molecular weight	18.0	137.4	187.4
Heat addition, Wh/kg	696.5	55.8	51.5
Net work output, Wh/kg	80.04	5.83	5.38
Rankine thermal efficiency, percent	11.49	10.45	10.45
Carnot's thermal efficiency, percent between 40°C and 100°C	16.08	16.08	16.08
Isentropic enthalpy drop in the turbine, Wh/kg	106.77	7.99	7.29
Solar-boiler pressure, N/cm^2	10.13	81.82	44.28
Ratio of boiler pressure to condenser pressure	13.735:1	4.718:1	5.658:1
Mass flow rate ($\dot{m}$) to produce 100 kW_e, kg/h (lb/h)	1,388 (3060)	19,069 (42,040)	20,660 (45,545)
Specific volume (ν) of d.s. vapor at turbine inlet m^3/kg (ft^3/lb)	1.673 (26.801)	0.0233 (0.3735)	0.0334 (0.5347)
Discharge (area × velocity product) at turbine-boiler pipe, m^3/s	0.6452	0.1234	0.1917

Table 3. An example of piping design for refrigerant R-11 vapor

Nominal pipe diameter,[a] cm (in.)	Inner diameter, cm	Cross sectional area, cm^2	Velocity[b] U, m/s	Reynolds[c] number, R_e $\times 10^{-6}$	f (smooth pipes)	$\Delta P/L$ (smooth pipes), $N/cm^2/100$ m	Friction horsepower, kW/100 m
10.16 (4)	10.226	82.13	15.027	4.777	0.0085	4.02	6.711
12.70 (5)	12.819	129.07	9.562	3.811	0.0089	1.36	2.274
15.24 (6)	15.405	186.39	6.622	3.171	0.0092	0.56	0.932
20.32 (8)	20.272	322.76	3.824	2.410	0.0097	0.15	0.254

[a]Steel pipe Schedule 40.

[b]At a flow rate of 19,069 kg/h and a specific volume of 0.0233 kg/m^3.

[c]At a viscosity μ of 0.0497 kg/m • h.

Table 4. An example of piping design for refrigerant R-113 vapor

Nominal pipe diameter,[a] cm (in.)	Inner diameter, cm	Cross sectional area, cm^2	Velocity[b] U, m/s	Reynolds Number[c] R_e, $\times 10^{-6}$	f (smooth pipes)	$\Delta P/L$ (smooth pipes), $N/cm^2/100$ m	Friction horsepower, kW/100 m
12.70 (5)	12.819	129.07	14.851	4.685	0.0085	2.19	5.67
15.24 (6)	15.405	186.39	10.284	3.898	0.0088	0.90	2.34
20.32 (8)	20.272	322.76	5.939	2.963	0.0093	0.25	0.63

[a]Steel pipe Schedule 40.

[b]At flow rate ($\dot{m}$) of 20660 kg/h and a specific volume of 0.0334 m^3/kg.

[c]At a viscosity μ of 0.0438 kg/m • h.

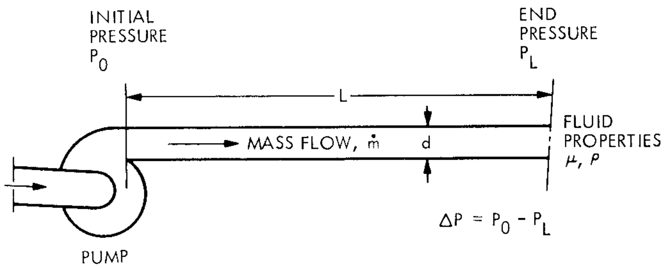

Fig. 1. Pipe schematic

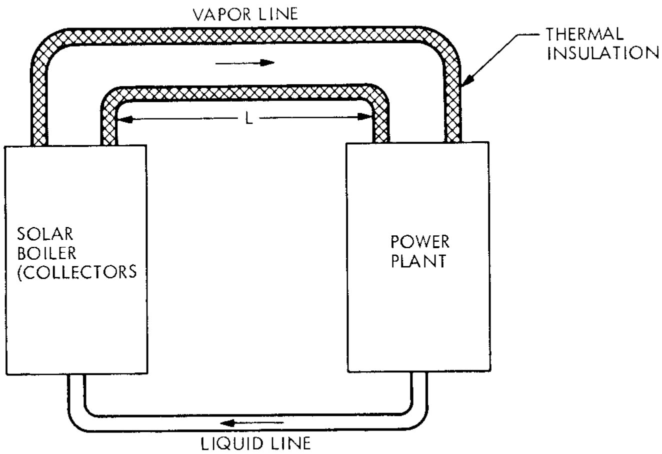

Fig. 2. A solar boiler, working at different pressures, is located at a distance *L* from the power plant

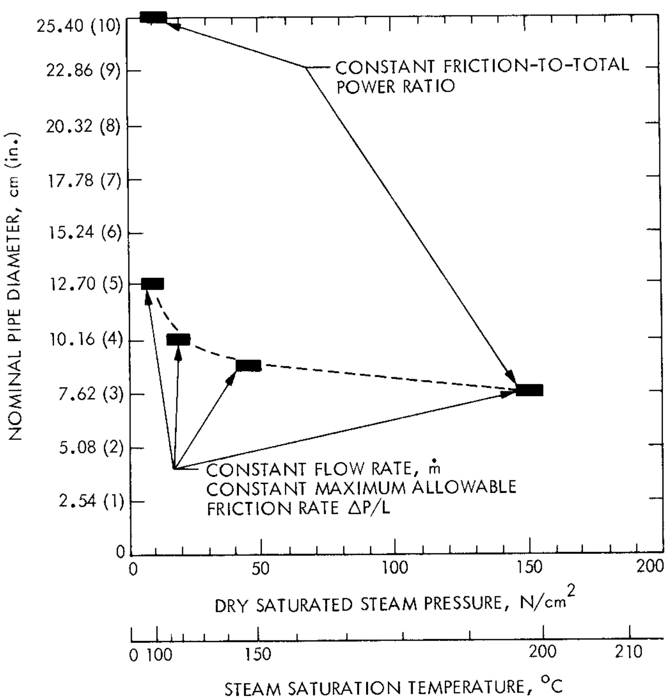

Fig. 3. The effect of varying the initial steam pressure on the pipe design with (1) constant friction rate, and (2) constant friction-to-total power ratio