CHE 260 First Law Lab

Quantifying Heat Loss, Work Input, and Energy Balance in a Pressurized Tank Using the First Law of Thermodynamics

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Abstract

This experiment applies the first law of thermodynamics to a rigid, pressurized acrylic tank to quantify heat loss pathways and compare the contributions of electrical heating and mechanical work from a propeller. The mass of air added to the tank ranged from (0.038 ± 0.001) kg to (0.062 ± 0.001) kg across four trials. Total heat losses were found between (19.1 ± 0.2) kJ and (97.1 ± 0.2) kJ, and plate heat losses varied from (-10.6 ± 3.81) kJ to (10.6 ± 11.1) kJ, showing non-physical values and modelling limitations. The calculated average specific heat capacity of 79.9 ± 16.8 kJ/kg·°C deviated drastically, with a percent difference of 11028% from the accepted value, demonstrating systematic error. Work done by the propeller ranged from (21.23 ± 0.07) J to (40.56 ± 0.09) J, with a temperature rate change of (7.7 ± 0.6) ºC/s to (10.7 ± 0.6)ºC/s, contributing only 0.03–0.08% of the total energy input. Thus, the lab stressed the importance of the first law for quantifying energy transfer, such as in pressurized vessels and heating systems, while highlighting errors between theoretical models and experimental analysis.

1. Introduction

The first law of thermodynamics expresses the conservation of energy in closed systems by relating heat transfer (Q), work (W), and changes in internal energy (ΔE). It is written as:

$$Q + W = \Delta E \tag{1}$$

This relationship forms the foundation for the design and analysis of energy-dependent systems. Engineers apply the first law to internal combustion engines, HVAC units, heat exchangers, and compressed-air storage systems, where the quantification of heat loss and work input is important for efficiency and safety. Pressurized systems, in particular, must be evaluated for thermal losses because temperature changes affect their limits and structural integrity [1], [2].

The objective of this experiment is to quantify heat loss and evaluate the relative significance of mechanical work versus electrical heating using the first law.

In this experiment, the measured temperature, pressure, and mass flow rate were used to determine the mass of air that had entered the tank ($m_{added\ to\ tank}$), to subsequently calculate the mass of air inside the tank ($m_{left\ tank}$) inside a rigid acrylic tank with a thermal conductivity of ~0.2 [W/mK]. This was applied in Equation (2), which uses right tank pressure (P_2) and temperature (T_2), and left tank pressure (P_1) and temperatureT_1).

$$m_{left\ tank} = m_{added\ to\ tank}\left[1 + \frac{1}{\frac{P_2T_1}{P_1T_2} - 1}\right] \tag{2}$$

The experiment also examines heat loss through the acrylic wall and the aluminum heater plates (~1000 W , ~200-250 [W/mK]). It was assumed that the total heat loss ($Q_{total\ heat\ loss}$) consisted of the heat loss from the plates (Q_{plates}) and the wall (Q_{wall}), expressed as:

$$Q_{total\ heat\ loss} = Q_{plates} + Q_{wall} \tag{3}$$

The heat loss from the cylindrical tank is modelled using Equation (4), where k is the thermal conductivity, l is the length of the tank, ΔT is the temperature difference between the inside of the tank and the room, and r_1, r_2 are the inner and outer radii of the tank, respectively.

$$\dot{Q} = 2k\pi l \frac{\Delta T}{In(\frac{r_2}{r_1})} \tag{4}$$

Finally, mechanical work from a small axial propeller is analyzed using fan similarity laws, whose relationships are used in turbomachinery design and performance scaling [3]. The projected work of the propeller on the system was used, alongside a calculated specific heat capacity, to achieve the rate of temperature change in the tank. Finally, the significance of the work done by the fan propellers and the heat added to the system was quantified and evaluated.

2. Experimental Methods

This section describes the apparatus and procedures used in the experiment.

2.1. Apparatus

The main apparatus used in this experiment consisted of the following components:

- Manometer: measures the ambient pressure of the room in psi.
- Left pressurized tank: a rigid acrylic cylindrical tank. The left pressure gauge displays the gauge pressure of the left tank.
- PID-controlled heaters: two electric aluminum heaters in the left tank. Controlled through the Hold button on LabVIEW to reach and maintain the target temperature.
- Bar Valve (BV): Located behind the left tank. It is manually opened alongside the Left Solenoid (LS) to allow for air circulation in the left tank.
- LabVIEW: controls the Solenoids (Left, Central, and Right) flow rate, and turns heaters on/off, as well as records data (i.e., pressure, temperature, and mass flow).

  Figure 1 shows top and side views of the layout, including the valves and connections.

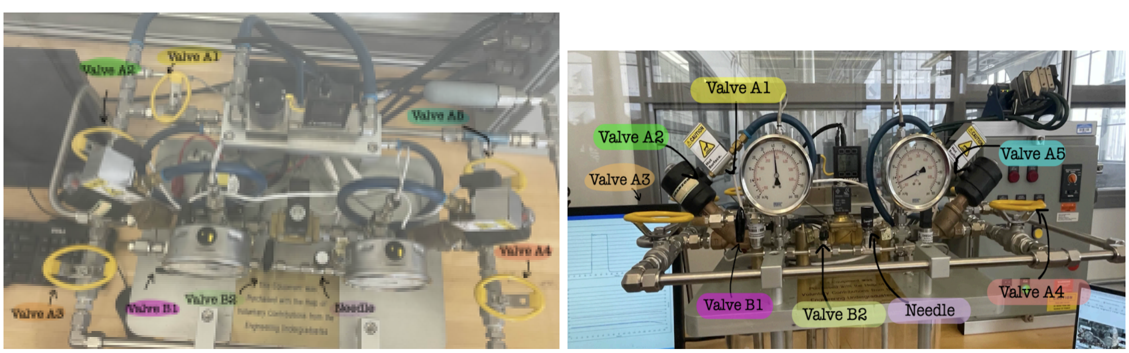

Figure 1—Top (left) and side (right) views of the apparatus setup of the pressure tanks. There are yellow valves (A1-5), black valves (B1, B2), a needle, and the connecting pipes. The bar valve is located at the bottom of the apparatus, directly under Valve B1.

2.2. Set Up and Procedure

The lab abides by all safety procedures outlined in the lab manual [4]. Room conditions were recorded as (20.3 ±0.1) ºC and (29.3 ± 0.1) Hg. Parts 1 and 2 were performed sequentially for each trial.

2.2.1 Part 1: Pressurizing Left Tank

The left tank was pressurized by opening valve A2 and the LS, and gradually increasing the flow rate to 50 g/min until reaching the desired pressures in *Table 1*.

Table 1: The desired pressures (psig) and temperatures (ºC) per trial.

Trial	Pressure (psig)	Temperature (ºC)
A	40	40
B	80	40
C	40	60
D	70	40

After the setpoint was reached, valve A2 and the LS were closed, and the flow rate reset to 0 g/min. The tank was allowed to stabilize (the values of the pressure and temperature on the graphs remained constant with time) before advancing. The tank was not emptied before moving onto the next part of the trial.

2.2.2 Part 2: Heating Left Tank

The target temperature on LabVIEW was set to the trial temperature. The heaters were left on until five minutes after the temperature in the left tank reached the desired temperature.

2.2.3 In Between Trials

After each trial, the heaters were turned off, while the LS and BV were left open until the left tank cooled to the after-trial temperature (5-10 minutes). Once the temperature stabilized (1 minute after closing the valves), the tank was evacuated by using the B1 valve.

After all trials were completed, the system was cooled down, and everything was turned off.

3. Results and Discussion

The following section describes the results of the lab and analyzes the findings.

3.1. Determining the Heat Loss in the Left Tank and Specific Heat Capacity

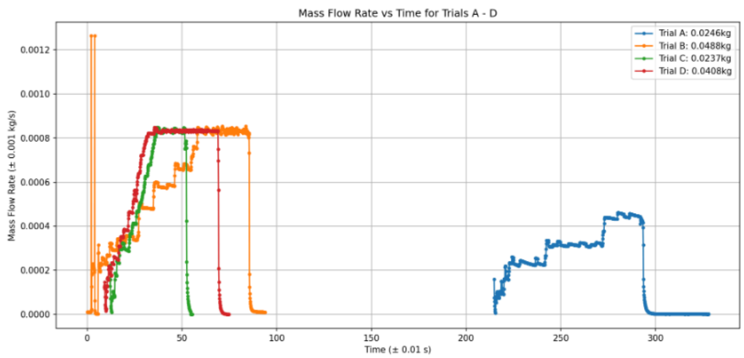

Figure 2: Graphs of Mass flow Rate with time for each trial

The total mass was found by integrating the mass flow rate curves in *Figure 2.* An uncertainty of ±0.001kg was used, based on the resolution of the data. By Equation (2), *Table 2* has the mass of the left tank.

Table 2: The mass (kg) in the left tank for each trial. P_1 and P_2 are the absolute pressures of the tank before and after the system reaches a steady state.

Trial	Mass Added $\pm\ 0.001$kg	P_1 $(\pm\ 0.02\,psia$	T_1 $(\pm\ 0.01°C$	P_2 $(\pm\ 0.02\,psia$	T_2 $(\pm\ 0.01°C)$	Mass Left Tank $(\pm\ 0.001\ kg)$
A	0.025	15.69	21.1	53.39	24.7	0.038
B	0.049	15.59	25.5	93.59	31.7	0.062
C	0.024	15.59	34.6	55.60	40.9	0.035
D	0.041	15.10	32.1	82.99	36.8	0.052

Table 3 has the heat loss (Q_{loss})of the system, using Equation (5), where $Q_{required}$ is the required energy to reach the steady state temperature, and Q_{final} is the energy added to maintain the temperature.

$$Q_{final} - Q_{required} = Q_{loss} \qquad (5)$$

Table 3: *Total Heat Loss in A System for Each Trial*

Trial	$Q_{required}$ $(\pm\ 0.1$kJ)	Q_{final} $(\pm\ 0.1$kJ)	Total Heat Loss $(\pm\ 0.2$ kJ)
A	66.5	84.4	17.9
B	48.6	83.9	35.3
C	100.0	152.8	52.8
D	108.6	205.7	97.1

Assuming that heat loss can only occur through the walls or the aluminum plates, the heat loss of the plates was found with Equation (3-4). The results are summarized in *Table 4* below.

Table 4: heat loss from the cylindrical acrylic walls and aluminum plate using Equation (5).

$\Delta T \pm 0.02°C$	$\Delta t \pm 0.02s$	$\dot{Q}_{walls}$ (J/s)	Q_{walls} (kJ)	Q_{plates} (kJ)
19.1	201.10	142.6 ± 18.1	28.7 ± 3.6	-10.8 ± 3.8
14.1	280.60	105.5 ± 13.3	29.6 ± 3.7	5.7 ± 4.0
28.7	271.80	214.7 ± 27.1	58.4 ± 7.4	-5.5 ± 7.6
29.0	398.90	216.9 ± 27.1	86.5 ± 10.9	10.6 ± 11.1

Several trials produced negative plate heat losses, which are physically impossible. A negative Q_{plates} implies there is heat flowing into the hotter cylindrical tube, which is unlikely, as heat flows downhill to a temperature gradient. This indicates missing heat-loss pathways, such as valve leakage, residual convection, heater inefficiency, and unaccounted thermal mass (e.g., clamps and internal fixtures). Similar discrepancies have been observed in heat-loss measurements of composite cylinders in literature [5].

Specific heat capacity (c_v) values were determined using the heater (Q_{heater}) values and temperature changes (ΔT), recorded in *Table 5*. The table uses the relation $Q_{heater} = mc_v\Delta T$ [5], for m the mass of air within the left tank.

Table 5: The specific heat capacity of the tank (c_v) for each trial

Trial	Q_{heater} ($\pm$ 0.1kJ)	Mass Left Tank ($\pm$ 0.001 kg)	$\Delta T \pm 0.02°C$	c_v (kJ/kgK)
A	66.5	0.038	19.1	92.8 ± 6.3
B	40.6	0.062	14.1	56.1 ± 3.8
C	100.0	0.035	28.7	98.2 ± 6.0
D	108.6	0.052	29.0	72.6 ± 3.6
Average Specific Heat Capacity: 79.9 ± 16.8 (based on standard deviation)				

Using Equation (6) calculated average specific heat (79.9 kJ/kgK) is 11028%, over two orders of magnitude larger than the actual value (0.718 kJ/kgK) [6].

$$\% \, Error = \frac{| \, Measured\,Value - Literature\,Value}{Literature\,Value} | \times 100\% \quad (6)$$

This discrepancy, alongside a non-physical negative Q_{plates} values in *Table 4* shows flaws in the experimental model, and that a simplified conduction-only model inadequately represents the system. Real tanks experience non-steady-state heating, variation in wall temperature, and air stratification, all of which reduce model accuracy [2], [5]. The model naively assumes the plates and walls are the only heat loss pathways, neglecting valve leaks, energy dissipation from the heaters, and unaccounted components like metal clamps inside the cylinder, which changes the assumption of constant wall conductivity and thickness. As well, the assumption of constant wall and plate temperatures during the steady-state period may be invalid; the apparatus might still have undergone heat inflow, causing recorded losses to be lower than actual values. A potential solution is to graph the wall and inner temperatures to find the steady temperature ranges.

3.2. Determining the work done by the Propeller in the Left Tank

Fan blade performance is calculated using pump and fan similarity equations in Equations (7-8), where Q is the volumetric flow rate, n the pump shaft speed, D the impeller diameter, P the fan power and ρ the gas density. Subscripts 1 and 2 are the manufacturer's test conditions and operating conditions, respectively [4].

$$Q_2/Q_1 = (n_2/n_1)(D_2/D_1)^3 \quad (7)$$

$$P_2/P_1 = (\rho_2/\rho_1)(n_2/n_1)^3(D_2/D_1)^5 \quad (8)$$

From the manufacturer, $n_1 = 4200\ rpm$, $P_1 = 0.001\ hp$ at standard conditions. Value ρ_1 is calculated from the ideal gas law [7], from Equation (9) with subscript i=1, for $p_1 = 101.325\ kPa,\ T_1 = 273.15\ K$ the pressure and temperature in standard conditions [8], $R_{air} = 287 J/kgK$ the specific gas constant for air [6], given in Equation (9).

$$\rho_i = \frac{p_i}{R_{air} T_i} \tag{9}$$

This gives $\rho_1 = 1.293\ kg/m^3$. Assuming $D_2 = D_1$, $n_2/n_1 = Q_2/Q_1$, so the second fan equation now reduces to Equation (10).

$$P_2 = P_1(\rho_2/\rho_1)(n_2/n_1)^3 \tag{10}$$

The values n_2 = 2000 rpm was taken from LabVIEW, and $\rho_2 = \frac{p_2}{R_{air} T_2}$ from Equation (9). Pressure and temperature values from the lab are reported in *Table 6*, with pressure values later converted to psia with $p_2 = p_{gage} + p_{atm}$.

Table 6: The pressures and temperatures in each trial of the lab, used to calculate denisty ρ_2 *from Equation (9), P2 from Equation (10), and converted to work using* $W = P\Delta t$ [9].

Trial	Pressure p_2 ($\pm\ 0.02\ psig$)	Temperature T_2 ($\pm\ 0.01°C$)	Density ρ_2 (kg/m^3)	P_2 (W)	Δt ($\pm\ 0.20s$)	Work (J)
A	42.30	40.00	4.35 ± 0.01	0.271 ± 0.001	78.40	21.23 ± 0.07
B	79.30	40.40	7.18 ± 0.01	0.447 ± 0.001	69.50	31.07 ± 0.10
C	44.60	59.20	4.26 ± 0.01	0.266 ± 0.001	108.60	28.84 ± 0.07
D	75.30	59.30	6.48 ± 0.01	0.404 ± 0.001	100.50	40.56 ± 0.09

The significance of work done on the system compared to the heat added is evaluated using Equation (12), and recorded in *Table 8*.

$$Percent\ work\ done\ on\ vs.\ work\ by\ system\ = \frac{Propeller\ work\ (W_{propeller})}{Added\ Heat\ (Q_{heater})} x100\% \tag{12}$$

Table 8: The heat from the heater Table 2, compared to the work calculated in Table 5, used to get the ratio in Equation (12).

Lab Trial	Q_{heater} ($\pm$ 0.1kJ)	$W_{propeller}$ (W)	$\frac{W_{propeller}}{Q_{heater}} x100\%$
A	66.5	21.23 ± 0.07	0.0319 ±0.0001
B	40.6	31.07 ± 0.10	0.0765 ± 0.0003
C	100.0	28.84 ± 0.07	0.0288 ± 0.0001
D	108.6	40.56 ± 0.09	0.0374 ± 0.0001

From *Table 8*, the work from the propeller was only 0.03% to 0.08% of the total heat added to the system, making it negligible in the system's energy balance in the first law (Equation (1)). This is consistent with studies showing that small axial fans have negligible bulk heating in confined volumes [3], meaning the value of the propeller is mixing, not energy input.

Several idealizations introduce error in this calculation. The fan similarity laws (Eqs. 7-8) assume idealized, incompressible flow, which is not perfectly achieved in the confined, turbulent environment of the tank. The model also neglects losses from motor inefficiency, bearing friction, and aerodynamic dissipation, meaning the actual work imparted to the air is less than calculated. Finally, the calculation of the temperature rise rate (Eq. 11) assumes an adiabatic system but is violated by concurrent heat losses (as quantified in Part 2) to the tank walls.

To calculate the rate of temperature rise, we can use $P_2 = mc_v \frac{dT}{dt}$ [5] and rearrange it for P_2 propeller power, m mass of air in tank (from part 1), c_v and the specific heat of air.

$$\frac{dT}{dt} = \frac{P_2}{mc_v} \tag{11}$$

The rate of temperature increase for the experiment is noted in *Table 7*.

Table 7: The calculated power from Table 5 and masses from Table 5, used to calculate the rate of temperature change with C_v values from Table 4 and Equation (11).

Lab Trial	$P_2\ (W)$	Mass Left Tank ($\pm\ 0.001\ kg$)	$C_v\ (kJ/kgK)$	$\frac{dT}{dt}\ (x10^{-5}\,{}^{o}C/s)$
A	0.271 ± 0.001	0.038	92.8 ± 6.3	7.7 ± 0.6
B	0.447 ± 0.001	0.062	56.1± 3.8	12.9 ± 0.9
C	0.266 ± 0.001	0.035	98.2 ± 6.0	7.7 ± 0.5
D	0.404 ± 0.001	0.052	72.6 ± 3.6	10.7 ± 0.6

The temperature rises are all within the magnitude 10^{-5}, which are extremely small, confirming that mechanical work is irrelevant in the first-law energy balance of this system.

4. Conclusion

This experiment applied the first law of thermodynamics to quantify energy interactions in a closed system. For air masses of (0.038 ± 0.001) kg to (0.062 ± 0.001) kg,

heat losses ranged from (19.1 ± 0.2) kJ to (97.1 ± 0.2) kJ with inconsistent plate losses of (-10.6 ±3.81) and (10.6±11.1) kJ. The calculated specific heat capacity, 79.9 ± 16.8 kJ/kg·°C, showed a major error of 11028%. These discrepancies show the challenges of modelling heat transfer in real systems using simplified steady-state conduction assumptions. The propeller work, between (21.23 ± 0.07) J and (40.56 ± 0.09) J, and associated heating rates of (7.7 ± 0.6) ºC/s to (10.7 ± 0.6)ºC/s, were negligible, accounting for only 0.03%-0.08% of total energy input. This confirms that mixing work is orders of magnitude smaller than direct heating in thermal systems.

Overall, the experiment demonstrates the importance of precise heat-loss modelling in engineering applications such as pressurized tanks, HVAC systems, and energy-storage devices. Future improvements should include direct measurement of wall temperature gradients and heat-loss coefficients to reduce model error.

5. References

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