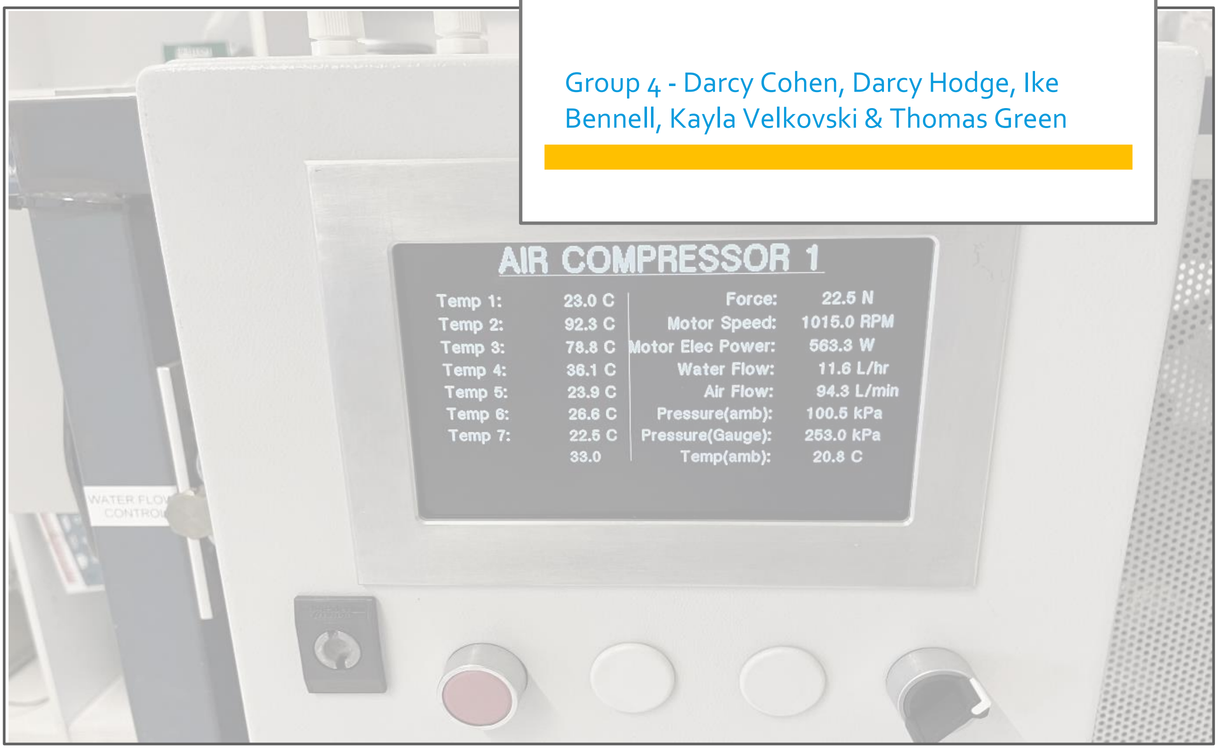
We are five UTS engineering students that are starting an internship at a large air compressor manufacturer. These air compressors are used in industrial HVAC systems. To make these systems more energy efficient to align with Australia's net-zero emissions targets, the air compressor company has been tasked to evaluate their current designs.

Air Compressor Project

43015 Thermofluids B

Group 4 - Darcy Cohen, Darcy Hodge, Ike Bennell, Kayla Velkovski & Thomas Green



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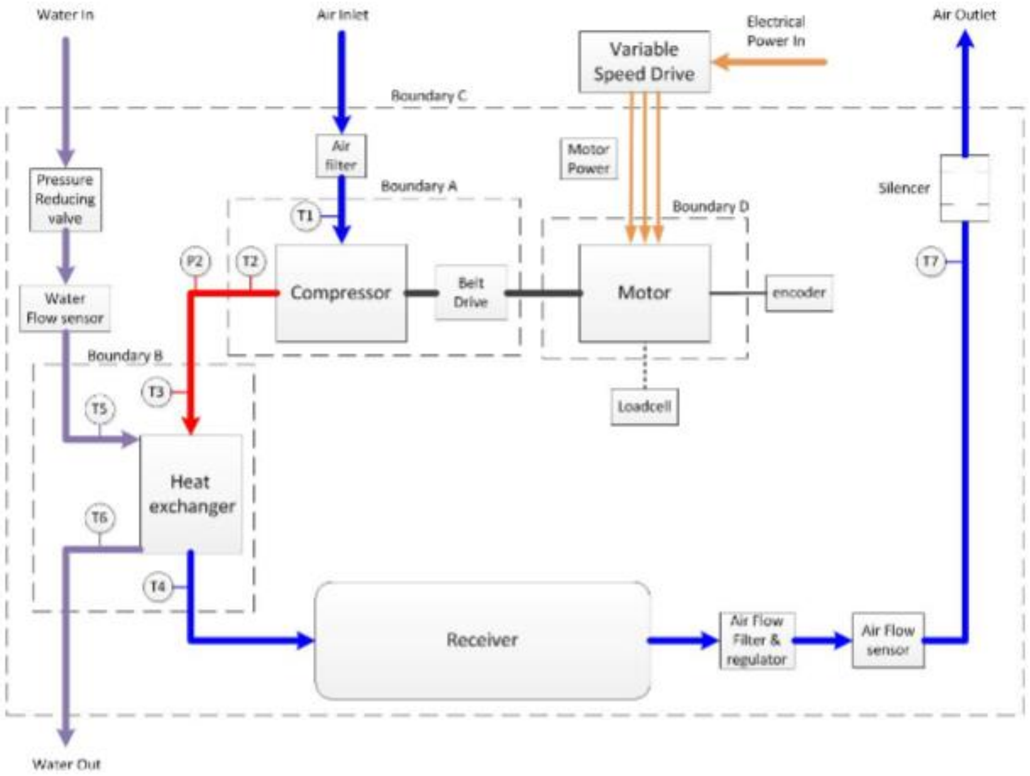
Task 1

Existing Design Evaluation



Motor (Boundary D)

Purpose

The purpose of the motor is to drive the compressor. This allows the pressure of air to be increased before it passes through the heat exchanger. The motor power is less than the actual power (power out) due to heat losses. Force generated inside motor was found from the lab to be 22.83N. The motor could be better designed, using higher quality components so that there are fewer energy losses through heat.

Theoretical Evidence

Boundary D – using Lab values

Using average value from excel sheet

$$\tau = F \times r = 22.83 \times 0.185 = 4.224\ Nm$$

$$P_{out} = \frac{2\pi \times 1016.33 \times 4.224}{60 \times 1000} = 449.56\ W$$

$$W_{out} = -|449.56| = -449.56W$$

With: $W_{in} = 575.1\ W$

$$\eta_{motor} = \frac{449.56}{575.1} = 78.17\%$$

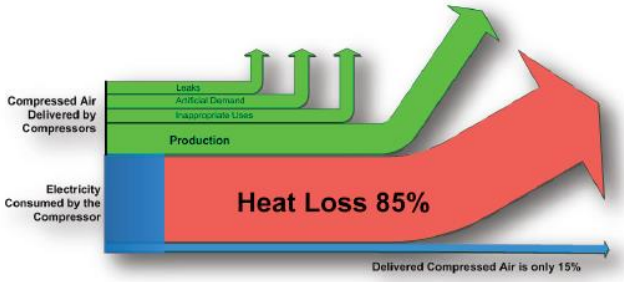
$$Q_D = W_{out} - W_{in} = -1024.66\ W$$

Compressor (Boundary A)

Purpose

Air compressors operate by putting air under extreme pressure, all while converting potential energy into kinetic energy that is stored for later usage. The purpose of a compressor is to increase pressure in a fluid system, examples of its use in everyday items include refrigerators and vehicle engines. In regard to above system, air is input into the compressor, first passing an air filter, and is then pressurized. From here, air is fed through a tube towards the heat collector. In this model, the system is powered by an additional work force produced by the motor[1].

The air compressor model does have some major drawbacks, the main being the large amount of heat loss because of inefficiency of the motor, the twin cylinder P14 Peerless compressor (producing 275L/min at "ultimate performance"). The below diagram depicts the amount of heat lost during the process. Looking at the main causes of heat loss, it is evident that heat generated by the compression process is our most prominent issue. Heat energy is concentrated as the volume of air is decreased. The compressor, to uphold optimal conditions for operation, is required to transfer the excess heat to that of a cooling device before air is released into further staged of the system. This process is the role of the heat exchanger. It is possible to adjust the system and as a result, recover almost 90% of the air lost during this process.



Overall efficiency of air compressor systems can range from as low as 10% to 50% (80% in some rare cases). In the case of the proposed system, the calculated efficiency was determined to be around 44%. There is much room for improvement with this design.

Referring to the acquired lab data, exit pressure after compression averages out to be 262L/min, almost achieving peak performance (as stated by the Peerless P14 cylinder online webpage). While the compressor itself achieves standard industry performance, the heat produces is the main problem, where exit temperature averages to around 86 (hitting a high of 89). Heat loss is extremely high in comparison to other components of this whole system, reducing this loss is a number one priority to increase the overall efficiency of the system.

Theoretical Evidence

Boundary A

Mass flow rate

$$Using\ \dot{V}_i = 95.4\frac{L}{min} = 1.59 \times 10^{-3}\frac{m^3}{s}, \qquad (Lab\ Data)$$

$$\dot{m}_1 = \rho_{air} \cdot \dot{V}_i = 1.29 \cdot 1.59 \times 10^{-3} = 2.05 \times 10^{-3}\frac{kg}{s}, \qquad (1.29\frac{kg}{m^3}\ is\ the\ density\ of\ air)$$

$$\dot{m}_1 = 2.05 \times 10^{-3}\frac{kg}{s}$$

[1] See pg2 above for further information regarding the motor.

$$m_{air} = \dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4 = \dot{m}_7$$

Velocities, Inlet and Exit

$$\bar{v}_i = \frac{\dot{V}_i}{A} = \frac{1.59 \times 10^{-3}}{2.27 \times 10^{-4}} = 7.0\frac{m}{s}\ , \qquad (A = \pi \cdot r^2 = \pi \cdot 0.0085^2 = 2.27 \times 10^{-4} m^2)$$

$$Pv = RT \rightarrow v_i = \frac{RT_1}{P_1} = \frac{0.2870 \cdot (22.5 + 273)}{100.6} = 0.8430 \ldots \frac{m^3}{kg}$$

$$v_e = \frac{RT_2}{P_2} = \frac{0.2870 \cdot (86.47 + 273)}{100.6 + 262.93} = 0.2837 \ldots \frac{m^3}{kg}\ , \qquad (P_2 = P_{ambient} + P_{gauge})$$

$$\dot{V}_e = \dot{m}_{air} \cdot v_e = 2.05 \times 10^{-3} \cdot 0.2837 \ldots = 5.82 \times 10^{-4} \frac{m^3}{s}$$

$$\bar{v}_e = \frac{\dot{V}_e}{A} = \frac{5.82 \times 10^{-4}}{2.27 \times 10^{-4}} = 2.56\frac{m}{s}$$

$$z_2 - z_1 = 1m\ , \qquad (from\ appendix)$$

$$W_{in} = 449.56\ W\ , \qquad (from\ Boundary\ D\ solution)$$

Heat energy loss in compressor

$$Q_A = \dot{m}_{air}\left[(h_e - h_i) + \frac{1}{2}\left(\bar{v}_e^2 - \bar{v}_i^2\right) + g(z_2 - z_1)\right] - W$$

$$Q_A = \dot{m}_{air}\left[cp(T_2 - T_1) + \frac{1}{2}\left(\bar{v}_e^2 - \bar{v}_i^2\right) + g(z_2 - z_1)\right] - W$$

$$Q_A = 2.05 \times 10^{-3}\left[1.005(86.47 - 22.5) + \frac{1}{2}(2.56^2 - 7.0^2) + 9.81(1)\right] - 449.56$$

$$Q_A = -449.45\frac{J}{s} = -449.45W$$

Efficiency

$$\eta_{Compressor} = \frac{Q_A}{Q_D} = \frac{-449.45}{-1024.66} = 43.86\%$$

Heat Exchanger (Boundary B)

Purpose

The heat exchanger's primary role is to cool the air flowing through it. It does so by passing the hot air and cold water through a Shell and Tube heat exchanger. This allows for some of the internal energy possessed by the air to be transferred to the colder water, primarily through conduction through the heat exchanger walls. The Heat exchanger is not adiabatically insulated, and as such, energy from the air is not wholly transferred to the water. Energy is lost through conduction with pipes, the body of the heat exchanger and the larger system. Thus, the heat energy the water gains is not equal to the heat energy lost by the air, as demonstrated below, for a total discrepancy of 57 watts.

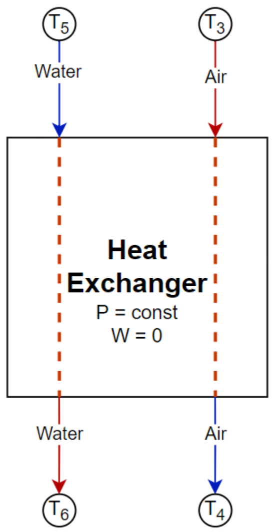
Efficiency calculations show that the air gives energy at an efficiency of 85% and that the water receives energy at a rate of 17%. This is due to the aforementioned inefficiencies but also because the thermal conductivity of water is far higher than that of air, meaning it is easier for the air to conduct heat into the water than the water into the air.

Theoretical Evidence

Boundary B

With: $\dot{V}_{water} = 0.0000034083\ \frac{m^3}{s}$ (Lab Data)

$\dot{m}_{air} = 0.00205\ \frac{kg}{s}$ (Boundary A)



Energy Change of Air: Using Table A-17

$$T_3 = 345K \ \rightarrow \ h_3 = 345.46\ \frac{kJ}{kg}$$

$$T_4 = 303K \ \rightarrow \ h_4 = 303.21\ \frac{kJ}{kg}$$

$$q_{air} = h_4 - h_3 = -42.25\frac{kJ}{kg}$$

$$\therefore \dot{Q}_{air} = \dot{m}_{air} \times q_{air} = 0.00205 \times (-42.25) = -0.0866\ \frac{kJ}{s}$$

Energy Change of Water

With $\dot{Q}_{water} = \dot{m} \times c_p \times (T_6 - T_5) = \ \dot{m} \times q_{water}$

$$\dot{m} = \rho_{water} \times \dot{V}_{water} = 997 \times 0.0000034083 = 0.0034\frac{kg}{s},\ c_p = 4.18\ \frac{kJ}{kg \times K}\ \text{(Table A-3)}$$

$$q_{water} = 4.18 \times (297.17 - 295.1) = 8.653\ \frac{kJ}{kg}$$

$$\dot{Q}_{water} = 0.0034 \times 8.653 = 0.0294\ \frac{kJ}{s}$$

Energy Loss

$$\Delta\dot{Q} = |\dot{Q}_{air}| - |\dot{Q}_{water}| = 0.0572\frac{kJ}{s} = 57.2\ W$$

Efficiency

Using efficiency: $\epsilon = \left|\frac{q}{q_{max}}\right|$ [2]

And, $q_{max} = C_{min} \times (T_{hot,in} - T_{cold,in})$ where $C_{min} = \min(c_{p,air}, c_{p,water})$

$$q_{max} = 1.005 \times (345 - 295.1) = 50.15\ \frac{kJ}{kg}$$

For Air, $q = q_{air} = -42.25\frac{kJ}{kg}$

$$\therefore \epsilon = \left|\frac{-42.25}{50.15}\right| = 0.84 = 84\%$$

Checked and confirmed with alternative method (Fakheri, 2014), see appendix

For Water, $q = q_{water} = 8.653\frac{kJ}{kg}$

$$\therefore \epsilon = \left|\frac{8.653}{50.15}\right| = 0.17 = 17\%$$

[2] (Washington University, n.d.)

Overall System (Boundary C)

Purpose

This Air compressor cycle is made up of three subsystems:

- The motor is responsible for taking electrical power and converting that into mechanical movement to be used by the compressor.
- The compressor is responsible for compressing air by over 3.5 times its original pressure.
- The heat exchanger is responsible for cooling the compressed air.

Energy loss refers to the reduction in the amount of usable energy available. Within this system energy refers to the input of electricity, through use of which we are able to complete the process of converting air (100.6 kpa measured) into pressurised air (262.93 kpa gauge pressure), while also ensuring that the air remains at a cool temperature (below 23 degrees).

Drawbacks:

- Net energy loss
- Energy loss summation

Possible causes for energy losses:

- Individual components

Whole system

Theoretical evidence

Energy loss summation

$$Q_D = 1024.66\ W$$

$$Q_A = 449.45W$$

$$Q_C = 57.2\ W$$

$$Q_C = Q_A + Q_B + Q_D = 1531.31\ W$$

Net losses=$Qc\,\frac{Qc}{Ambient}$ = 1531

Task 2

Polytropic Index of Compression Process

$$n = \frac{\log\left({}^{P_2}/_{P_1}\right)}{\log\left({}^{v_i}/_{v_e}\right)}$$

$$n = \frac{\log\left(\frac{100.6 + 262.93}{100.6}\right)}{\log\left(\frac{0.8430\ldots}{0.2837\ldots}\right)}$$

$$n = 1.18$$

Change in Specific Entropy

Change of Specific Entropy of Boundary A →

$$\Delta s = s_2 - s_1 = cp \cdot \ln\left({}^{T_2}/_{T_1}\right) - R \cdot \ln\left({}^{P_2}/_{P_1}\right)$$

$$\Delta s_A = 1.005 \cdot \ln\left(\frac{86.47}{22.5}\right) - 0.287 \cdot \ln\left(\frac{100.6 + 262.93}{100.6}\right)$$

$$\Delta s_A = 0.302 \frac{kJ}{kgK}$$

This doesn't violate the second law of thermodynamics (aka delta S is greater than 0).

Net Entropy Change of A →

$$S_{net} = S_{system} + S_{surrounding} = \dot{m}_2 s_2 - \dot{m}_1 s_1 - \frac{Q_C}{T_o}$$

$$S_{net} = 1531.31 - \frac{1531.31}{293.85} = 1526.1 \frac{kJ}{kgK}$$

Ambient tmp = 20.7

Power Required by Compressor

Polytropic Power

To find the power:

$$Power = \frac{P_f V_f - P_i V_i}{n - 1}$$

With the known values:

$$P_f = 363.53\ kPa\ (P2\ from\ Boundary\ A), \qquad P_i = 100.6\ kPa(P1\ from\ Boundary\ A)$$

$$n = 1.18\ (polytropic\ index)$$

However, the volume must be found for both initial and final conditions, through the ideal gas formula:

$$PV = RT$$

$$V = \frac{RT}{P}$$

$$V_f = \frac{0.287(86.47 + 273)}{363.53}$$

$$V_i = \frac{0.287(22.5 + 273)}{100.6}$$

$$V_f = 0.2838m^3,\ V_i = 0.843m^3$$

Now power can be found:

$$Power = \frac{363.53(0.2838) - 100.6(0.843)}{1.18 - 1}$$

$$\boldsymbol{Power = 102W}$$

Isentropic Power

To find the power produced by an isentropic system:

$$P_S = \dot{m}_{air}\ C_p(T_2 - T_1)$$

Where:

$\dot{m}_{air} = 2.05 \times 10^{-3} \frac{kg}{s}, C_p = 1.005 \frac{kJ}{kg \times K} = 1.005 \times 1000 \frac{J}{kg \times K}$ (From table A-2),

$T_2 = 86.47^o C = 359.62\ K$ (From testing), $T_1 = 22.5^o C = 295.65\ K$ (From testing)

Therefore, the Power produced can be calculated:

$$P_S = \left(2.05 \times 10^{-3}\right)(1.005 \times 1000)(359.62 - 295.65)$$

$$\boldsymbol{Power = 131.79W}$$

Isentropic Efficiency

Isentropic efficiency of the compressor:

$\dot{m}_{air} = 2.05 \times 10^{-3} \frac{kg}{s}$

$C_p = 1.005 \frac{kJ}{kg \times K} = 1.005 \times 1000 \frac{J}{kg \times K}$ (From table A-2)

$T_2 = 86.47^o C = 359.62\ K$ (From testing)
$T_1 = 22.5^o C = 295.65\ K$ (From testing)

$P_a = 449.56\ W$ (boundary D)

$$\eta_c = \frac{P_S}{P_a} = \frac{\dot{m}_{air}\ C_p(T_2 - T_1)}{P_a}$$

$$= \frac{(2.05 \times 10^{-3})(1.005 \times 1000)(359.62 - 295.65)}{(449.56)} \left(\frac{J \times K \times kg}{kg \times K \times s}\right)\left(\frac{1}{W}\right)$$

$$= \frac{131.79W}{449.56W}$$

$$= \mathbf{0.293 \rightarrow 29.3\%}$$

Power Comparison

The tested power, calculated through torque calculations, does not account for any energy loss, as the torque formula does not include any energy values, but only RPM and torque. This accounts for the substantial increase in comparison to both the polytropic and isentropic calculations above.

Tested Power	449.56W
Polytropic	102W
Isentropic	131.79W

The polytopic power, by definition, accounts for heat transfer, and thus heat loss through the compressor, whereas the isentropic power is calculated without this energy consideration. This explains the 29.8W differential in power between the two processes.

Task 3

Each team member has been given the freedom to design a new optimised system by adding, removing or changing the system's components, materials or dimensions.

Thomas G

Compressor Volumetric Efficiency Improvement

The travel of the piston within the compressor determines the volumetric efficiency of the compressor, due to the swept volume in relation to the Clearance Ratio – used in the volumetric efficiency formula:

$$n_{vol} = 1 - C\left(\frac{P_2}{P_1} - 1\right)$$

Where C is the Clearance Ratio, and can be found through:

$$C = \frac{Clearance\ Volume}{Swept\ Volume}$$

Given the data of the compressor, with the bore being 75mm, the stroke being 50mm, and the clearance distance being 10mm, the Clearance Ratio can be calculated:

$$C = \frac{\pi\left(\frac{75}{2}\right)^2 * 10}{\pi\left(\frac{75}{2}\right)^2 * 50}$$

$$C = \frac{44.18}{220.9}$$

$$C = 0.2 \rightarrow 20\%$$

Thus, the volumetric efficiency is therefore:

$$n_{vol} = 1 - 0.2\left(\frac{363.53}{100.6} - 1\right)$$

$$n_{vol} = 0.48 \rightarrow 48\%$$

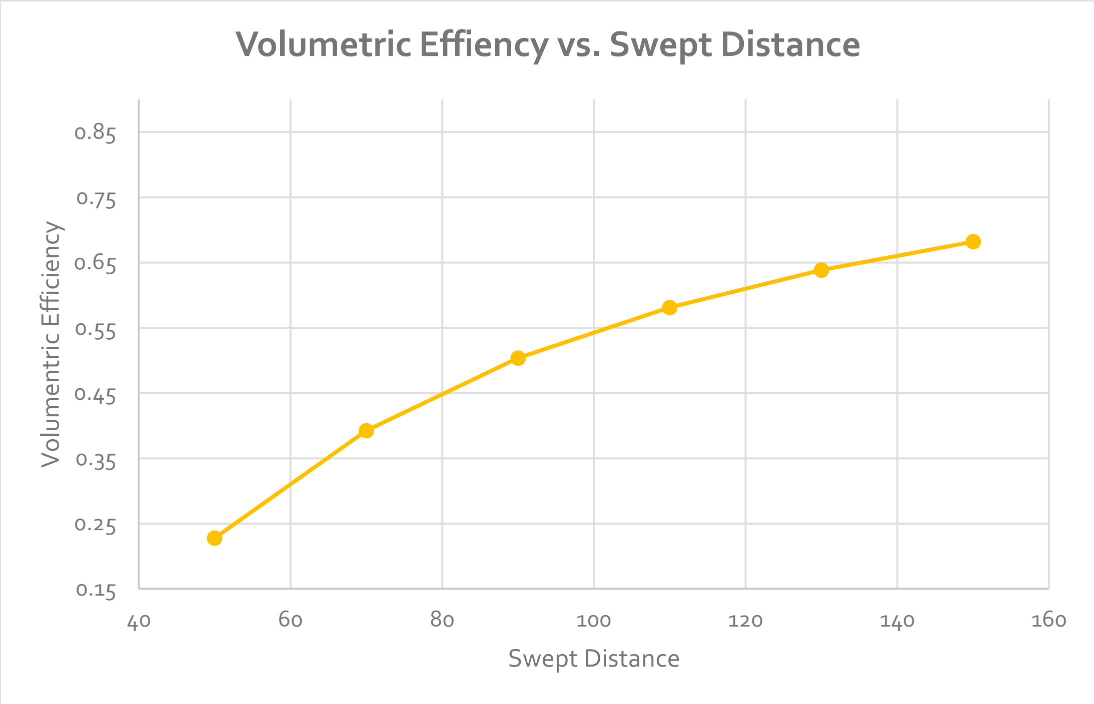
With two cyclinders, the total efficiency is then:

$$n_{vol,total} = 0.48^2 \rightarrow 23\%$$

To improve this efficiency, the swept distance can be increased, decreasing the clearance ratio, thus ultimately increasing the total efficiency of the compressor. An excel spreadsheet has been created to display the resulting efficiency increase from increasing the stroke distance:

Input Val			Output Val
Swept D	Clearance D	Clearance Ratio	Volumetric Efficiency
50	10	0.200	0.23
70	10	0.143	0.39
90	10	0.111	0.50
110	10	0.091	0.58
130	10	0.077	0.64
150	10	0.067	0.68

This presents a logarithmic increase, presented graphically here:

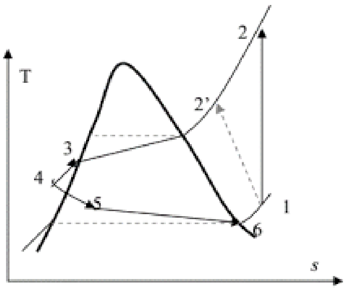


Therefore, by increasing the swept distance, the overall efficiency of the compressor can be increased, thus increasing the overall efficiency of the system.

Kayla V

Reviewing the air compressor system in Boundary A, the total efficiency of the compressor is around 43% which isn't the most ideal for the total system. See below equation.

$$\eta_{Compressor} = \frac{Q_A}{Q_D} = \frac{-449.45}{-1024.66} = 43.86\%$$



The main cause of this low efficiency (and as a result, high amounts of heat energy loss) is due to the loss of energy from heightened temperatures of compressed air. Looking at the acquired lab data from the Peerless P14 model, there is a large increase in temperature after the compression process.

$$T_1 = 22.5°C \qquad T_2 = 86.47°C$$

An ideal isothermal compression where n=1, work done on a gas during the pressurization process is as follows:

$$W = \int_{V_1}^{V_2} PdV$$

For isentropic compression:

$$Pv^n = c \quad \rightarrow \quad Pv = nRT$$

Where n is because of amount of cooling and c is a constant.

Using the above calculation for Pv…

$$w = \frac{nRT(T_2 - T_1)}{n-1} = \frac{nRT_1}{n-1}[(\frac{P_2}{P_1})^{\frac{n-1}{n}} - 1]$$

By reducing n, the value of work is also reduced (W∝$\frac{1}{n-1}$). For maximum, ideal cooling, the process would need to be isothermic and thus n=1.

$$W = RTln(\frac{P_2}{P_1})$$

For this type of compression, heat is removed extremely rapidly that the gas cannot heat during compression. Achieving an effect as close to this is the ultimate goal. Although it is Impossible to achieve such a state, as a result, polytropic compression is the best practically (while only second best in terms of reduction in work values).

Using the calculated polytropic index, the work value produces is equal to:

$$n = \frac{\log\left(\frac{100.6 + 262.93}{100.6}\right)}{\log\left(\frac{0.8430 \ldots}{0.2837 \ldots}\right)} = 1.18$$

Above is as calculated for the proposed polytropic system (almost 1 – almost ideal)

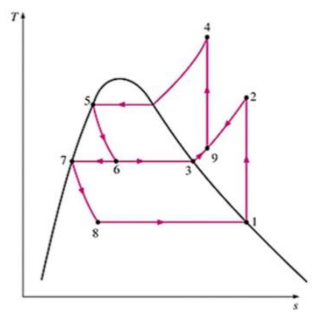
$$W = \frac{1.18 \cdot 0.287 \cdot 295.5}{1.18 - 1}\left[\left(\frac{100.6 + 262.93}{100.6}\right)^{\frac{1.18-1}{1.18}} - 1\right] = 120W$$

Work produced by the motor accounts for this need for energy efficiently.

Some small changes that can be made to help increase efficiency can be by increasing the quality of air that is input into the system. By using cooler air, the compressor does not need to exert as much power to pressurize. As well as this, using air of lower temperatures allows for higher air density, which is much more advantageous for the air compression process. It was found that for evRery 1˚C that drops in pre-cooled air, energy consumption is reduced by 0.65%. *(Ghritlahre & Chandramohan, 2018, p. 379)*.

Friction between moving parts within the machine is another factor that significantly contributes to the reduction of efficiency. Another solution would be to lubricate the compressor to allow for a reduction in friction and free movement of mechanical parts. Types of lubricants to use for small scale compressors such as the Peerless P14 include mineral and or synthetic oils. Mineral oils are more cost-effective than synthetic oils and are therefore more efficient for a compressor of this size and usage.

Using a multistage air compressor would aid in the significant reduction of heat generated by the compressor due to its design. The Peerless P14 Is a twin cylinder air compressor, however it would be much better to considering using a 3-stage compressor. The same horsepower can be used to power the compressor, as a result there will be no increase in work. Overall, multistage compressors are much more energy efficient as there isn't a significant reduction in work needed to produce the desired pressures. Looking at the T-S diagram to the right, it can be seen how multiple stages progress to the final pressure value, requiring much less heat in doing so.



Ike B

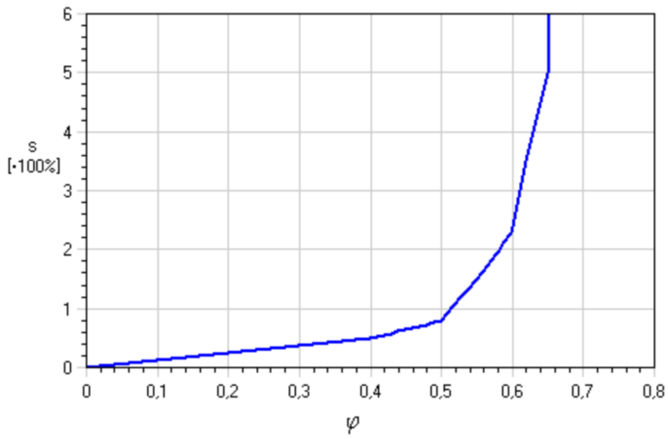
Belt drive

A commercially available air compressor will typically make use of a belt drive mechanism in their air compressors for its RPM adjustability and ease of maintenance. This however comes at a cost to the power output of the motor, resulting in a loss of efficiency.

The power loss associated with the use of belt drives can be derived from belt slippage. Belt slippage refers to the loss in efficiency that occurs when the belt connecting two shafts is not producing enough frictional force compared to the amount of torque being produced by a motor, causing it to 'slip'. This loss in efficiency in extreme cases can result in the complete loss of system functionality.

Direct drive, within the context of air compressors, refers to the direct connection between the motor shaft and the compressor pump shaft resulting in the complete lack of slippage. While this would negate almost all energy loss associated with transferring energy from the motor shaft to the compressor, it would also result in the removal of any flexibility surrounding speeds.

Built-in slip table



It is assumed that:

- The belt slip occurs on the driver pulley so the speed of all driven pulleys and idlers is influenced by the same slip.
- The belt speed change due to slip is neglected. Usual belt slip magnitude is 1% ~ 2% what results in s = 0,01 ~ 0,02

Specifications

CODE	MODEL	CYLINDERS	PUMP DISPLACEMENT	FREE AIR DELIVERY	PUMP MAX PSI	MOTOR HP	PUMP TYPE	PUMP RPM	AMPS DRAW	TANK (L)	PRICE *(INC GST)*
00257	P14	2	14 CFM	275 LPM	145	2.75	N75	852	10	55L	**$2075**

Assumptions:

- Difference in motor and compressor pump RPM is directly caused by slippage
- Slippage is the only reason for loss of efficiency I the belt drive
- Direct drive has an efficiency of 100%

$$\omega_{compressor} = 852\ RPM,\quad \omega_{Motor} = 1016.33\ RPM,\quad \eta_{Direct\ drive} = 100\%$$

$$\eta_{belt\ drive} = 1 - slippage = Speed\ ratio = \frac{\omega_{compressor}}{\omega_{Motor}} = \frac{852\ RPM}{1016.33\ RPM} = 0.8383\ \rightarrow 83.83\%$$

$$Efficiency\ increase\ (\%) = \frac{\eta_{Direct\ drive} - \eta_{belt\ drive}}{\eta_{belt\ drive}} \times 100\% = \frac{100\% - 83.83\%}{83.83\%} \times 100\%$$

$$= \mathbf{19.289}\%$$

Darcy H

Motor Optimisation

The motor power is less than the actual power (power out) due to heat loss. Force generated inside motor was found from the lab to be 22.83N. The efficiency of the motor was determined to be:

$$\eta_{motor} = \frac{W_{out}}{W_{in}} = \frac{449.56}{575.1} = 78.17\%$$

Where,

$$W_{in} = 575.1\ W$$

$$W_{out} = -|449.56| = -449.56W$$

The motor could be swapped for a more appropriate and more efficient motor. From research, the following 1.5kW, 3 phase electric motor is an ideal replacement:

Motor:	Techtop TAP "High Efficiency" Series (IP66) 1.5kW Motor - 230V Three Phase Aluminium
Efficiency (full load, 1440RPM):	85.5%

The efficiency which the motor is rated to is 85.5% (*IE3 IP66 EXD SERIES*, n.d.). The use of this new motor makes the system a more optimised design.

Compressor Optimisation

Additionally, the compressor can be considered in terms of changes to be made to the existing system to optimise the system. The current compressor nets an efficiency of:

$$\eta_{Compressor} = \frac{-449.45}{-1024.66} = 43.86\%$$

Where

$$Q_D = W_{out} - W_{in} = -1024.66\frac{kJ}{s}$$

And

$$Q_A = -449.45\frac{kJ}{s}$$

It is seen that the compressor is not very efficient (43.86%) and should be optimised.

The primary cause for this significant lack of efficiency is heat generation in the system's chosen compressor (Peerless P14). That is, when air is rapidly compressed in the piston, significant heating takes place. There are a number of methods which could increase efficiency, such as using higher quality components which produce less resistance, or changing the type of compressor. Some changes to the existing design which could produce greater efficiency include better cooling around the piston. Significantly, the existing design uses only passive cooling to limit heat transfer.

A more efficient design would use a cooling system for the piston. An addition of a low power, high efficiency fan placed on top of the motor could be very effective at cooling as it blows cooler air over the hot fins of the piston surround.

This would be a cost-effective way to make use of the current components of the system – and incur little financial penalties. This is an ideal option if limited changes must be made to the existing system to net significant improvements in efficiency.

One of the ideal, highly efficient fan choices would be the Noctua NF-A20 FLX fan (*NF-A20 FLX*, 2024). This is a 0.96W, 200mm fan typically used in computer applications. It is ideal as it is very low peak power draw so its impact on the power draw from the compressor is limited.

The rate of heat transfer can be modelled with Newton's law of cooling:

$$Q = hA(T_{fluid} - T_{Wall})$$

In this equation, h is the heat transfer coefficient, A is the surface area of the wall being cooled, T_{fluid} is the temperature of the fluid (air), and T_{wall} is the temperature of the piston surround.

Typical values for h range between 10 and 500W/(m²*K) (Malekan et al., 2021). For this case, we can assume worst case (10W) to observe worst case results.

Surface area of the piston surround (A), will be modelled as 400mm x 200mm.

T_{fluid} will be assumed to be 298K, and T_{wall} assumed to be 373K.

In this case the rate of heat transfer due to forced convection in the worst case is modelled as:

$$Q = 10 \times (0.4 \times 0.2) \times (298 - 373) \qquad [\frac{kJ}{s}]$$

$$Q = -60\ \frac{kJ}{s}$$

This would net an efficiency increase of:

$$\eta_{Compressor} = \frac{-449.45 - 60}{-1024.66 - 0.96} = 49.68\%$$

Or <u>5.82%</u> in the worst case. Actual values could be more significant.

Additionally, the Peerless P14 compressor could be replaced with a more efficient compressor.

Other compressor types include centrifugal, axial, and scroll type compressors. However centrifugal compressors are most efficient at high flow applications, and axial compressors only ideal for applications such as powerplants and aircraft. Scroll compressors are efficient at low flow rates

- "Scroll compressors are 10–15% more efficient than reciprocating compressors when run as single-speed machines." (Sarbu & Calin Sebarchievici, 2016). A scroll compressor would therefore be an ideal choice to net an increase of 10-15% efficiency in the compressor system.

However, an ideal change with limited impact on the rest of the system would be do switch the existing single-stage Peerless P14 compressor with a multistage reciprocating compressor. By design this would see an immediate increase in efficiency. Peerless does offer multistage compressors – such as the 3-stage Peerless PHP15 compressor. However this model does not contain the same motor power, so it cannot be used.

In this case, another alternative could be to use water as a fluid medium for forced convection. Water has a great heat transfer coefficient than air, h = 100W/(m^2*K) in the worst case (Malekan et al., 2021).

Substituting this into Newton's law of cooling:

$$Q = 100 \times (0.4 \times 0.2) \times (298 - 373) \qquad [\frac{kJ}{s}]$$

$$Q = -600\ \frac{kJ}{s}$$

In this case, efficiency could then be calculated as:

$$\eta_{Compressor} = \frac{-449.45 - 600}{-1024.66 - 120} = 91.69\%$$

Where 120W is the power required for water cooling (*Hot and Cold Water Cooler Electric Consumption*, 2022).

Heat Exchanger Optimisation

With a much more efficient compressor system, the system can be further optimised. Cooler air temperatures leaving the compressor means a much smaller heat exchanger can be used. This would net a significant improvement in power efficiency. The system would therefore be more efficient and most optimised.

Alternative heat exchangers include:

- Plate heat exchangers
- Scraped surface heat exchangers

The ideal choice would be a plate heat exchanger. This is because plate heat exchangers are often considered the most efficient due to turbulent flow on both sides of each plate (Currie, 2023).

However, the best choice is often the simplest. That is, the existing heat exchanger could run at lesser power to achieve the same desired cooling effect as previously. This would be the best choice as it does not require the purchase of a new heat exchanger (no financial impact), along with the time to modify the existing system (time and labour).

Summary

In summary, the following change is best for system optimisation:

1. The existing Peerless P14 single-stage compressor to be modified to accept water cooling surrounding the compression camber.
2. Heat exchanger power to be decreased as reaction to decreased air temperatures exiting compressor.
3. Motor can be substituted with TechTop motor to achieve ultimate system optimisation.

Darcy C

The Fluid Dynamics FluidEx 2332 heat exchanger that is used is a 3-pass, 345mm long (nominally) shell and tube design. There are several other heat exchangers in this range, each with differing lengths. By utilising longer heat exchangers, better heat conduction can be expected, and the minimum temperature of the air flowing through it is expected to decrease. With a fixed efficiency, heat transfer coefficient and water temperature, the following exit air temperatures are expected.

Using Constants from Boundary A Theoretical Evidence, and equations from (Fakheri, 2014)

$$\dot{Q}_{actual} = \dot{m}_{air} \times c_{p,air}(T_3 - T_4) = 0.0794$$

$$\dot{Q}_{actual} = U \times A \times \eta \times \left(T_{air,avg} - T_{water,avg}\right)$$

$$\therefore U \times A = 0.00335$$

$$\dot{Q}_{optimal} = U \times A \times \left(T_{air,avg} - T_{water,avg}\right) = 0.09334$$

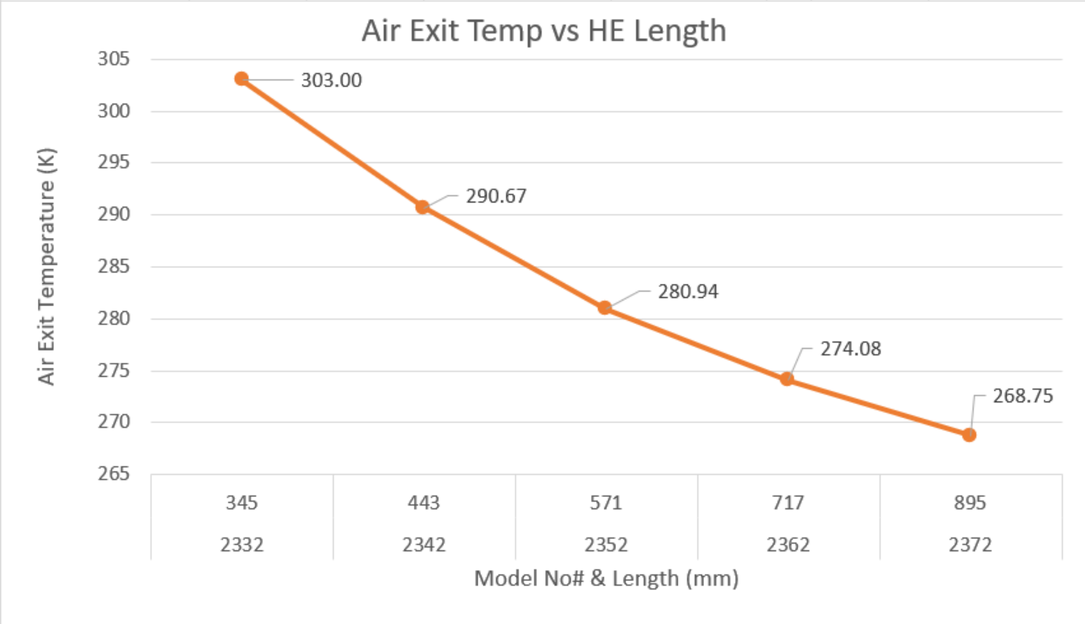
With, $A = \pi \times 0.083 \times 0.345 = 0.09$ (Values from (Fluid Dynamics, n.d.))

$$\therefore U = 0.0372$$

Where $T_{air,avg} = \frac{T_3+T_4}{2}$,

$$\therefore T_4 = \left(\frac{\dot{Q}_{optimal}}{U \times A} + T_{water,avg}\right) \times 2 - T_3$$

Heat Exchanger	L	A	UA	T_4		d	0.083
2332	345	0.08996	0.00335	302.995373		UA	0.00335
2342	443	0.11551	0.004301594	290.667864		U	0.037238977
2352	571	0.14889	0.005544493	280.939446		q_opt	0.09334
2362	717	0.18696	0.006962174	274.083464		T_wat,avg	296.135
2372	895	0.23337	0.00869058	268.75073		T_3	345



As demonstrated, the longer the heat exchanger, the further the temperature drops, meaning that more energy can be extracted from the heat with similar efficiency and heat conduction levels. It would be my recommendation that the heat exchanger is then swapped out for the Fluid Dynamics FluidEx 2372.

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Appendix

1. Lab Data

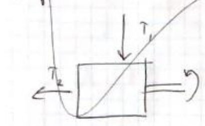
	Unit	1	2	3	Avg
load cell	N	23.2	22.7	22.6	22.83
motor speed	RPM	1018.4	1016.2	1014.4	1016.33
water flow rate	L/hr	12.4	12.3	12.1	12.27
air flow rate	L/min	96.1	95.4	94.7	95.40
motor power	Watts	561.3	555.1	555.3	557.23
P1 (Pamb)	kPa	100.6	100.6	100.6	100.60
P2 (gauge)	kPa	284.3	253.8	250.7	262.93
T1	Deg C	22.1	22.5	22.9	22.50
T2	Deg C	82.7	87.1	89.6	86.47
T3	Deg C	67.8	73.1	75.9	72.27
T4	Deg C	28	30.5	32.5	30.33
T5	Deg C	21.6	22.1	22.6	22.10
T6	Deg C	23	24.5	25	24.17
T7	Deg C	21	21.4	21.8	21.40

2. Heat Exchanger

2.1. Theoretical Evidence

Mini Project

Compressor Boundary A



RPM = 1016.33

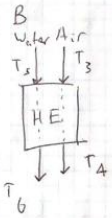
τ = 4.224 Nm

~~3.3 RPM_{moto} = RPM_{comp}~~

~~2π~~

$\therefore P_{in} =$

Boundary B



P = const

$q_{Air} = h_4 - h_3$

$q_{water} = h_6 - h_5$

$q_{total} = q_{air} + q_{water} + W$

W=0

$T_3 = 72.27$ $\quad T_4 = 30.33$

$T_5 = 22.10$ $\quad T_6 = 24.17$

$\dot{V}_{water}$ = 12.27 L/hr = 0.0000034083 m^3/s

~~$\dot{V}_{air}$ = 0.00154 m^3/s~~

~~P_2 = 262.43 kPa~~

~~Ideal gas q_{air} =~~

~~$\therefore h_3$ =~~

$\dot{m}_{air}$ = 0.00188 (Boundary A) kg/s

$\dot{m}_{water} = 1000 \times \dot{V}$ = 0.00341 kg/s

Ideal gas : Air

$T_3 = 345$ K $\rightarrow h_3 = 345.46$

$T_4 = 303$ K $\quad h_4 = 303.21$

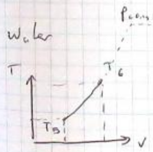
$\therefore q_{air} = -42.25$ $\quad \therefore Q_{air} = -0.07943$ kJ/s

~~$\therefore q_{water} = +42.25$~~ $\quad$ ~~$= -Q_{water}$~~

~~$\therefore Q_{water} = 0.07943$~~

$= \dot{m}_{water}(h_6 - h_5)$

Water



$\Delta h = c_p \Delta T = 8.6526$ kJ/kg

$\frac{kJ}{kg\,K}$ K

$\therefore Q_{water} = 0.0295$ kJ/s

$\therefore \Delta Q = 0.045$ kJ/s $\quad \therefore$ Air lost more heat than absorbed

Efficiency

$$\eta_{air} = \frac{T_3 - T_4}{T_3 - T_5} = 0.842$$

$$\dot{Q}_{max} = (h_3 T_3 - h_4 T_4)\,\dot{m}_{air} = 0.04308$$

$$\dot{Q}_{act} = \dot{m}_{air} c_p (T_3 - T_4) = 0.0794$$

$$\therefore \epsilon_{air} = \frac{\dot{Q}_{act}}{\dot{Q}_{max}} = 0.85$$

efficiency

or $\epsilon = \frac{q_{act}}{q_{max}}$ $\quad q_{max} = c_{min}(T_3 - T_5) = 50.25$

$q_{act_{air}} = 42.29$

$\epsilon_{air} = 0.841$

$\epsilon_{wat} = 0.172$

$$0.0794 = UA\,\overset{0.841}{\eta}\,(T_{air\,avg} - T_{wat\,avg}) \quad (= 27.865)$$

$\therefore UA = 0.00335$ $\quad$ Check using (Fakheri, 2014)

$\therefore q_{opt} = UA(T_{air_{avg}} - T_{wat_{avg}})$

$= 0.09334 \approx \dot{Q}_{max}$ ✓

~~$\eta = \frac{q}{\dot{m} c_{p_{min}} (T_{air_{avg}} - T_{wat_{avg}})}$~~

($\dot{m}$ updated post working, reflected in PDF only)

0.00188 → 0.00205

3. Polytropic Power

3.1 Theoretical Evidence

$$W = \left(\frac{P_f V_f - P_i V_i}{n-1} \right) / t \qquad P = W/t$$

$P_i = 100.6 \qquad V_i = \frac{0.287(225+273)}{100.6} = 0.843 m^3$

$P_f = 363.53 \qquad V_f = \frac{0.287(86.47+273)}{363.53} = 0.2838 m^3$

$$W = \frac{(363.53)(0.2838) - 100.6(0.843)}{1.18-1} = 102 kJ$$

$P = W/t$

3.2 Polytropic Index & Entropy

Polytropic Index of Boundary A ↓

$$n = \frac{\log\left(\frac{Pe}{Pi}\right)}{\log\left(\frac{Vi}{Ve}\right)} = \frac{\log\left(\frac{100.6+262.93}{100.6}\right)}{\log\left(\frac{0.84302683}{0.283794707}\right)} \doteq 1.18$$

Change of Specific Entropy A →

$$s_2 - s_1 = C_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right)$$

$$\Delta S_A = 1.005\ln\left(\frac{86.47}{22.5}\right) - 0.287\ln\left(\frac{100.6+262.93}{100.6}\right)$$

$= 0.302 kJ/kgK$ [doesn't violate 2nd law]

4. Compressor

4.1 Theoretical Evidence

$\dot{V}$ = volume flow rate , $\mathcal{V}$ = velocity , v = spec. volume

$\dot{V}_i = 1.59\times10^{-3}\,m^3/s$

$\dot{m} = \rho\dot{V}_i = 1.29\,kg/m^3 \cdot 1.59\times10^{-3}\,m^3/s$

$\dot{m}_1 = 2.05\times10^{-3}\,kg/s$

$\dot{m}_{air} = \dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4 = \dot{m}_7$

$\dot{V}_i = \mathcal{V}_i \times A$ [volume flow rate = velocity × Area]

$\mathcal{V}_i = 7.0\,m/s$ where $A = 2.27\times10^{-4}\,m^2$

$\dot{V}_e = \dot{m}_1 \times v_e$ where $Pv = RT$ and $v = \frac{RT}{P}$

$\dot{V}_e = 2.05\times10^{-3} \times \left(\frac{0.287\times(86.47+273)}{100.6+262.93}\right)$

$\dot{V}_e = 5.82\times10^{-4}\,m^3/s$

$\dot{V}_e = \mathcal{V}_e \times A$

$\mathcal{V}_e = 2.56\,m/s$

$g(h_e-h_i) \overset{?}{=} 1m$

$W_{in} = 449.56W$ [calc from Boundary 0]

$Q_A = \dot{m}_{air}\left[(h_e-h_i) + \frac{1}{2}(\mathcal{V}_e^2-\mathcal{V}_i^2) + g(h_e-h_i)\right] - W$

$= \dot{m}_{air}\left[c_p(T_2-T_1) + \frac{1}{2}(\mathcal{V}_e^2-\mathcal{V}_i^2) + g(h_e-h_i)\right] - W$

$= 2.05\times10^{-3}\left[1.005(86.47-22.5) + \frac{1}{2}\left[(2.56^2)-(7.0^2)\right] + 9.81(1)\right] - 449.56$

$Q_A = -449.45\,kJ/s$

4.2 Unit Conversion

$\dot{m}_{air}c_pT_2 - \dot{m}_{air}c_pT_1$

$= \frac{kg}{s}\cdot\frac{kJ}{kg\,K}\cdot K - \frac{kg}{s}\cdot\frac{kJ}{kg} = 0$

$\frac{1}{2}\dot{m}_{air}v^2 - \frac{1}{2}\dot{m}_{air}v^2$

$= \frac{1}{2}\cdot\frac{kg}{s}\cdot\frac{m^2}{s^2} - \frac{1}{2}\frac{kg}{s}\cdot\frac{m^2}{s}$

$\dot{m}_{air}\cdot g\cdot z$

$= \frac{kg}{s}\cdot\frac{m}{s^2}\cdot m = \frac{kg\cdot m^2}{s^3} = W$

$W - W$