4 ESTIMATION OF HEAT LOADS

Heat load is defined as the amount of heat that needs to be removed from a facility either to maintain product temperatures (in storage situations) or reduce product temperatures (in chillers or freezers). The amount of heat load determines the size of refrigeration system required and largely defines the capital and operation cost of the facility. There are a variety of factors that contribute to the heat load:

- Product (in freezing and chilling)
- Fans, defrost and lights
- Other mechanical devices (e.g. forklifts, conveyors) and people
- Heat infiltration (through walls, ceiling, floor)
- Air interchange (through doors)
- Cooling of room structures, particularly concrete floors

A typical heat lead breakdown is as follows

	Coldstore/Coolstore	Chiller/Freezer
Product	0-20%	50-75%
Fans	10-25%	10-40%
Heat Infiltration	20-50%	0-15%
Air Interchange	20-50%	0-15%

Some heat loads do not occur at a constant rate, e.g. in freezers when a new batch is introduced the rate at which heat is released to the air from new units will be much larger than the heat release rate from products nearing complete freezing. Hence the average heat load is important, but the peak heat load can also matter if it adds to the overall peak load on the plant, and hence to the plant peak electricity demand.

The general methodology for calculating heat load is as follows:

a) Individual load components are found. Normally average loads are used, but peaks must be also considered.

b) The total loads (average and peak) are found by summation:

Total load $= \phi_T = \phi_p + \phi_f + \phi_l + \phi_{md} + \phi_{pe} + \phi_i + \phi_a + \phi_s + \phi_d$ (15)

where ϕ_p = product load (W)
ϕ_f = fan load (W)
ϕ_l = lighting load (W)
ϕ_{md} = mechanical devices load (W)
ϕ_{pe} = people load (W)
ϕ_i = heat infiltration load (W)
ϕ_a = air infiltration load (W)
ϕ_s = room structure load (W)
ϕ_d = defrost load

c) A load analysis is carried out by plotting the predicted loads versus time on a 24 hour scale. Each load component is plotted separately versus time, and then the summation conducted to get the load versus time profile. The refrigeration plant size must be such that **peak** heat loads can be adequately handled.

This load analysis will illustrate total peak, average and minimum loads of a facility. This load analysis can be used to change operating procedures to ensure that peak loads are minimised and occur during times for cheap electricity.

4.1 THE EQUATIONS

HEAT LOAD	SCENARIO	EQUATION	NOMENCLATURE
Product Load (ϕ_p)	Chilling Average	$\frac{m_p c_L(\theta_{in} - \theta_{out})}{t}$	θ_{out} = product out temperature (°C) θ_{in} = product in temperature (°C) m_p = mass of product processed (kg) t = time of the process (s) c_L = specific heat of unfrozen product (J/kg°C)
	Freezing Average	$\frac{m_p\left[c_L(\theta_{in} - \theta_f) + \Delta h_f + c_S(\theta_f - \theta_{out})\right]}{t}$	θ_f = product freezing temperature (°C) c_S = specific heat of frozen product (J/kg°C) Δh_f = enthalpy change in freezing (kJ/kg)
	Respiring Product Storage Average	$m_p R_R$	R_R = respiration rate (W/kg)
Fan Load (ϕ_f)	Average	$N_f W_f \eta_m$ **or** $\frac{\Delta P Q}{\eta_m \eta_{fan}}$	N_f = number of fans W_f = nominal fan power (W) ΔP = pressure drop fan is working against (Pa) Q = volumetric flow rate of air (m^3/s) η_m = fan motor efficiency η_{fan} = fan efficiency
Light Load (ϕ_l)	Peak	$N_l W_l$	N_l = number of lights W_l = light wattage (W)
Mechanical Devices (ϕ_{md}) e.g. forklifts	Peak	$N_m W_m F$	N_m = number of mechanical devices W_m = mechanical device nominal power (W) F = % operating time
Room Structural Load (ϕ_s)	Average	$\frac{m_s c_s(\theta_{st} - \theta_{fal})}{t}$	θ_{st} = structure start temperature (°C) θ_{fal} = structure final temperature (°C) m_s = mass of structure(kg) t = time of the process (s) c_s = specific heat of structure(J/kg°C)

HEAT LOAD	SCENARIO	EQUATION	NOMENCLATURE
People (ϕ_{pe})	Peak	$500N_{pe}$	N_{pe} = number of people
Infiltration Through Building (ϕ_i)	Average	$\frac{kA(\theta_o-\theta_i)}{x}E_i$	x = insulation thickness (m) A = surface area of the wall, roof etc. (m^2) E_i = insulation effectiveness factor θ_o = temperature of the outer surface of the wall, roof, floor or ceiling(°C) θ_i = temperature of the inner surface of the wall, roof, floor or ceiling (°C) k = thermal conductivity of the insulation (W/mK)
Air Infiltration (ϕ_a)	Peak	$\frac{1}{2}A_D v p_{ai}(h_o-h_i)F_D$ **where** $v=5.91\sqrt{\frac{H(1-S)}{(1+\sqrt[3]{S})^3}}(1-PF_D)$ **and** $S=\frac{\rho_{ao}}{\rho_{ai}}$	A_D = door cross-sectional area (m^2) v = mean air velocity through the open door (m/s) ρ_{ai} = density of inside air kg/m^3) ρ_{ao} = density of outside air (kg/m^3) h_o = enthalpy of outside air (kJ/kg) h_i = enthalpy of inside air (kJ/kg) F_D = Fraction of time door used during operation time PF_D = Protective factor of protective device H = height of the door (m) S = ratio of warm air to cold air density
Any Peak Load	Daily Average Load	$Peak\times\frac{x_h}{24}$	x_h = time operated per day (hrs)
Defrost Load (ϕ_d)	Average	$0.15\times\sum(all\ \ other\ \ average\ \ loads)$	

4.2 USEFUL DATA

4.2.1 Product Thermal Properties

To perform calculations involving chilling and/or freezing and to calculate product heat loads it is necessary to know the various thermal properties for the product under consideration. This subject was covered in the second year thermal properties of biological materials courses.

4.2.2 Fan Efficiency Values

η_m = fan motor efficiency (typically 0.85 – 0.95)

η_{fan} = fan efficiency (typically 0.55 – 0.70)

4.2.3 Wall/Ceiling/Floor Temperature Assumptions

Internal surface temperatures are approximately the internal operating temperature. Increments to add to ambient temperatures due to radiation effects to estimate external walls and ceiling temperature are:

	Protected (Weather Sheild)	Not Protected
Walls facing the sun	5	10
Walls not facing the sun	0	0
Roof	10	25

These apply for light coloured surfaces. Double for dark coloured surfaces and half for 24-hour averages.

4.2.4 Air Interchange Protective Devices

- Well installed air curtains reduce interchange by 85 %
- Plastic strips in good condition reduce interchange by 95%
- Forklift traffic increases the calculated interchange on doors with protection by 30% for plastic strips, and 100% for vertical air curtains, but reduces it by 20% for unprotected doors, and by 20% for horizontal air curtains.

4.2.5 Structural Thermal Properties

Thermal Conductivities

Fibreglass	0.040 – 0.050 W/mK
Corkboard	0.035 – 0.040 W/mK
Polyurethane foam	0.022 – 0.030 W/mK
Polystyrene foam	0.028 – 0.035 W/mK

Specific Heat and Density

Concrete	$c =$	600 - 900 J/kgK;	$\rho =$	2300 kg/m^3
Steel		500 – 900 J/kgK;		7800 kg/m^3
Aluminium		880 – 900 J/kgK;		2700 kg/m^3

4.2.6 Air Density

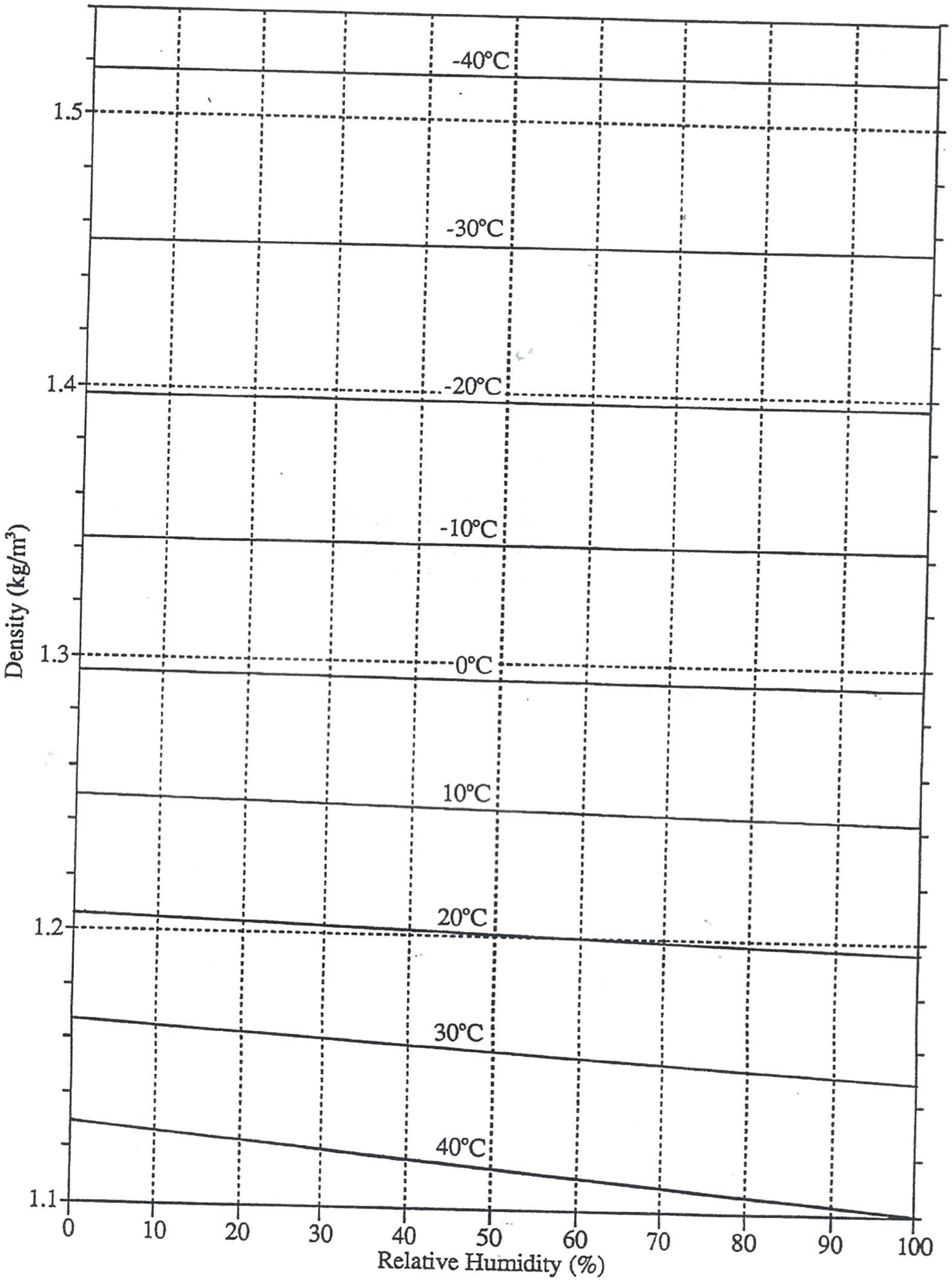


Figure 42, Plot of air density (kg/m^3) as a function of air relative humidity (%) and dry bulb temperature (^{0}C).

4.2.7 Air Enthalpy

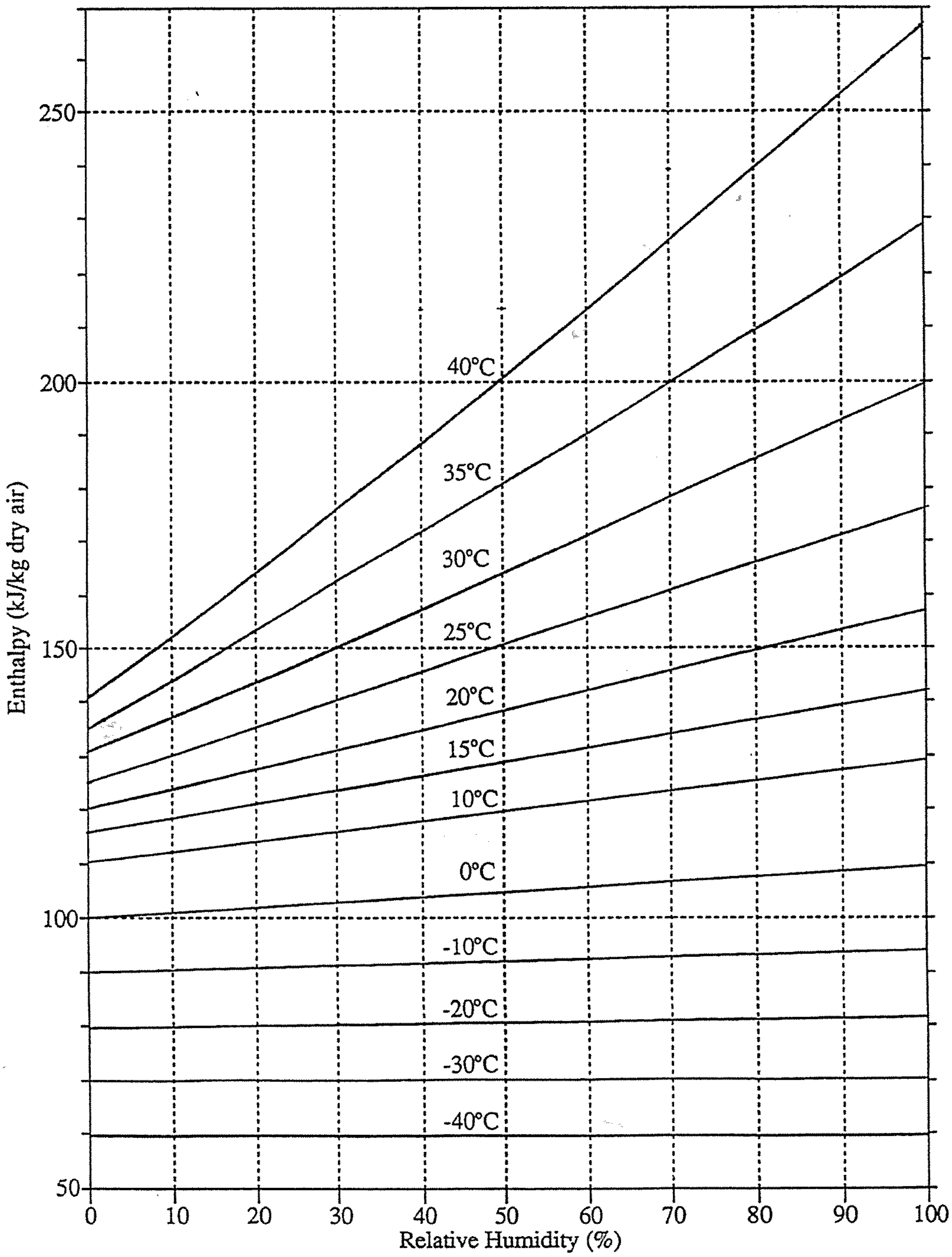


Figure 43, Enthalpy of air (J/kg) as a function of air relative humidity (%) of dry bulb temperature (^{0}C).

4.3 WORKED EXAMPLE – CALCULATION OF A COLDSTORE HEAT LOAD

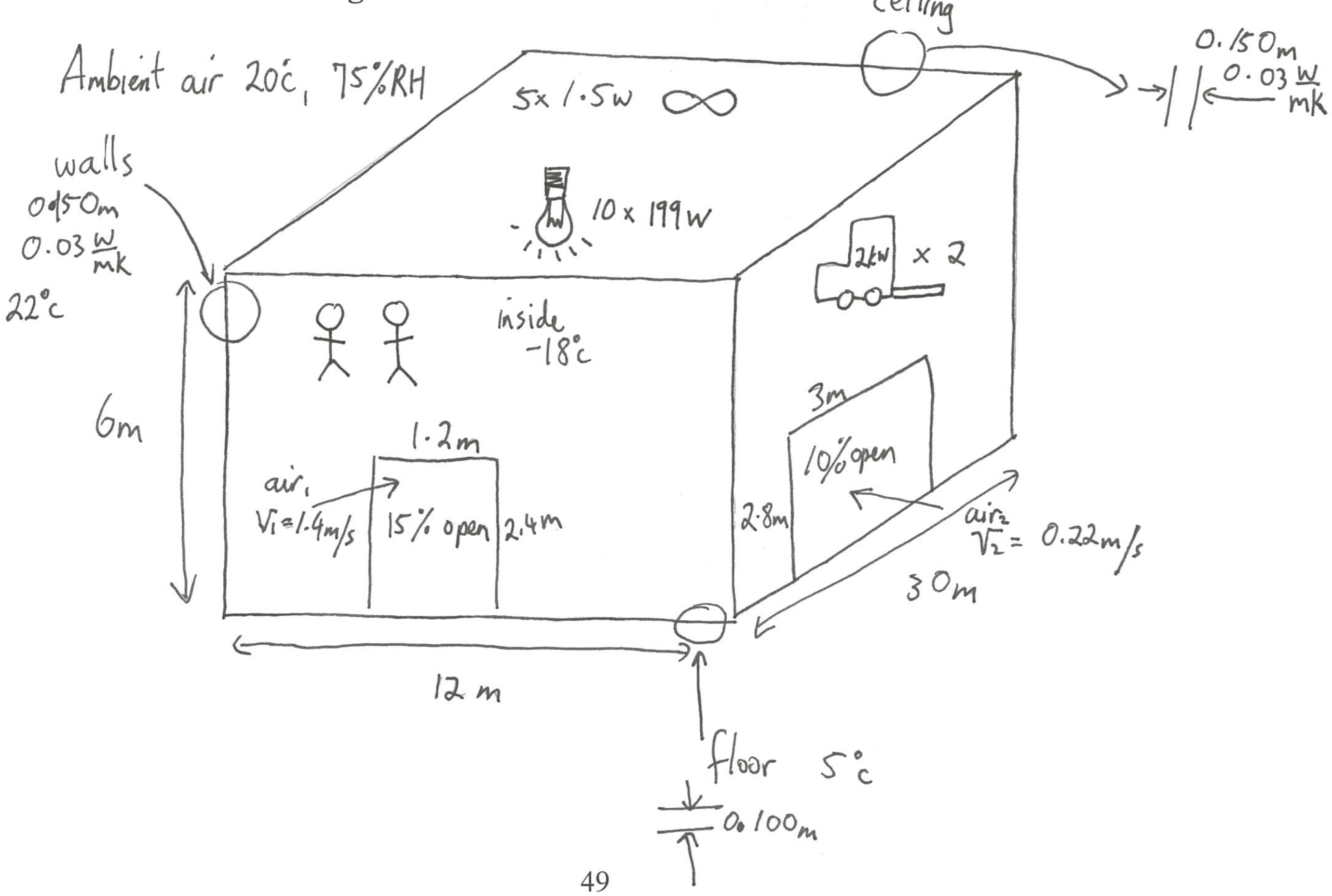
Calculate the mean heat load of a coldstore with dimensions of height 6m, width 12m length 30m. The coldstore operates at -18°C and is manned during working hours with two people on average in the store at any one time. Work starts at 8.00 am and finishes at 5.00 pm. The coldstore is lit by 10 sodium vapour fixtures with a total of 199 W per fixture. The workers use a small, unprotected, door 2.4m high and 1.2m wide to get in and out of the coldstore. This door is open about 15% of the time during working hours. Two forklifts are in use. They consume 2 kW each and are inside the coldstore 75% of the working item. They use a door, 2.8m high and 3m wide, which is protected by an air curtain and it is in use during 10% of the time during working hours. The small door has a measured air velocity of 1.4 m/s through it when open, the large door 0.22 m/s. The roof has a weather shield that keeps the roof insulation surface temperature 10^{0}C above the ambient (20°C and 75%RH). The ceiling is composed of sandwich panel constructed of polystyrene foam board 150mm thick (thermal conductivity 0.03 W/mK). The walls are insulated in an identical fashion as the roof, but he protective shield reduces the temperature to 2^{0}C above ambient. The floor has a 100mm layer of the same polystyrene foam but under floor heating keeps the ground temperature at 5^{0}C. It has been estimated that the extra heat entering by conduction through wall penetrations etc., is 30% above that calculated theoretically.

insulation effectiveness factor

There are 5 fans each with a 2.0 HP (1.5 kW) motor installed on them. Defrosting occurs during a one hour period every day.

Roof 30°c

4.3.1 Scenario Diagram



4.3.2 Product

Enters at -18°c or no products

$\phi_p = 0\ kW$

4.3.3 Fans

$\phi_f \quad N_f W_f \eta_m$

$= 5 \times 1.5 \times \frac{23}{24} = 7.19\ kW$

4.3.4 Lights

- Peak

$\phi_l = N_l W_l = 10 \times 199 = 1.99\ kw$

- Average $= peak \times \frac{9}{24} = 0.75\ kw$

4.3.5 Mechanical Devices

- Peak

$\phi_{MD} = \Sigma M_{MD} W_{MD} F$

$= 2 \times 2 \times 0.75 = 3\ kw$

- Average $\phi_{MD} = peak \times \frac{9}{24} = 1.13\ kw$

4.3.6 People

- Peak

$\phi_{ppl} = 500\ N_{ppl}$

$= 500 \times 2$

$= 1\ kw$

- Average $\phi_{ppl} \times \frac{9}{24} = 0.375\ kw$

4.3.7 Room Structures

$\phi_s = 0$ Assuming the cooling system in the room has been running for a significant period of time, hence the structures are cooled prior to entry of the product.

4.3.8 Heat Infiltration

- Ceiling

$$\phi_{ci} = \frac{k}{x} A (\theta_o - \theta_i) E_i$$

$$= \frac{0.03}{0.150} (30 \times 12)(30 - {}^-78)\ 1.3$$

$$= 4.49\ kW$$

- Floor

$$\phi_{FL} = \frac{0.03}{0.100} (30 \times 12)(5 - {}^-78)\ 1.3$$

$$= 3.23\ kW$$

- Walls

$$\phi_{wall} = \frac{0.03}{0.150} (30 \times 6 \times 2 + 12 \times 6 \times 2)(22 - {}^-78) \times 1.3$$

$$= 5.24\ kW$$

- Total Heat Infiltration

$$\phi_T = 4.49 + 3.23 + 5.24$$

$$= 12.96\ kW$$

4.3.9 Air Interchange

- Small Door Peak

air density from figure 42

enthalpy from figure 43

$$\phi = \frac{1}{2} A_D V \rho_{air} (h_o - h_i) F_D$$

$$= \frac{1}{2} (1.2 \times 2.4)(1.4)(1.39)(148 - 83) \times 0.15$$

$$= 27.3\ kW$$

- Large Door Peak

$$\phi = \frac{1}{2} (3 \times 2.8) \times 0.22 \times 1.39 (148 - 83) \times 0.10$$

$$= 8.3\ kW$$

- All Doors Peak $\phi_{Ad} = 8.3 + 27.3$

$$= 35.6 \text{ kW}$$

- All Doors Average

$$= 35.6 \times \frac{9}{24}$$

$$= 13.35 \text{ kW}$$

4.3.10 Defrost $\phi_{def} = 0.15 \times \sum (\text{all other average loads})$

$$= 0.15\ (13.35 + 0.375 + 1.13 + 0.75 + 7.19 + 12.96)$$

$$= 35.75 \text{ kW} \times 0.15 = 5.36 \text{ kW}$$

4.3.11 Summary

Heat Sources	Peak Operational Load (kW)	Night Time Load (kW)	average 24 hour Mean Load (kW)
Product	0	0	0
Lights	2	0	0.75
Fans	7.19	7.19	7.19
Forklifts	3	0	1.13
People	1	0	0.375
Structure cooling	0	0	0
Heat infiltration	12.96	12.96	12.96
Air interchange	35.6	0	13.35
Defrost	—	—	5.36
TOTAL	61.75 kW	20.15 kW	41.1 kW

Note that the cooling system will only operate 23 hours per day, but would be sized to cope with the maximum demand, and so in 23 hours at full capacity it has plenty of spare capacity to remove the total heat load coming in over 24 hours.

4.3.12 Load Analysis