Vessel heat stress technology trial program – sheep 2019: static dehumidification trial report

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Contents

Nomenclature

Symbol	Definition
C	Proportionality constant relating T_{WB} to the internal energy rise ($^{O}C/(kJ/kg)$)
E	Energy (kW)
H	Metabolic heat rate (W/kg)
h	Specific Enthalpy (kJ/kg)
M	Liveweight in the ventilation zone (kg/m^2)
Q	Heat Load (kW)
T	Temperature (^{O}C)
V	Volumetric Flowrate (m^3/s or m^3/hr)

Greek Symbols	Definition
Δ	Changes (or Difference)
ρ	Density (kg/m^3)
ω	Absolute Humidity, mass ratio between water vapour and dry air ($kg_{w.v}/kg_{d.a}$)

Subscripts	Definition
Amb.	Ambient
d.a.	Dry air
DB	Dry Bulb
env	Environment
Fans	Supply fans
g	Saturated vapour
in	Input
L	Latent
out	Output
Pen	Pen
Space	Space referring to deck
S	Sensible
T	Total
V/A	Volumetric to Pen Area
vel	Equally distributed fresh air velocity on pen floor
w.v.	Water vapour
WB	Wet Bulb

Abbreviations	Definition
AEH	Air Exchanges per Hour (1/hr)
DHM	Dehumidifier
HCU	Humidity Control Unit
HSRA	Heat Stress Risk Assessment
LFM	Linear Feet per Minute (ft/min)
PAT	Pen Air Turnover (m/hr)
RDCs	Research and Development Corporations
RH	Relative Humidity (%)
S&E	Supply and Exhaust

INTRODUCTION

About LiveCorp

The Australian Livestock Export Corporation Limited (LiveCorp) is a not-for-profit industry body, funded through statutory levies collected on the live export of sheep, goats and beef cattle, and a voluntary levy collected on live dairy cattle exports. LiveCorp is one of the 15 Australian rural Research and Development Corporations (RDCs).

LiveCorp is the only RDC focused solely on the livestock export industry and works hard to continuously improve performance in animal health and welfare, supply chain efficiency and market access. LiveCorp delivers this by investing in research, development and extension and providing technical and marketing services and support to enhance the productivity, sustainability and competitiveness of the livestock export industry.

LiveCorp works across several program areas, often in close consultation with other industry stakeholders, including the Australian Government, but does not engage in agri-political activity.

LiveCorp works closely with other RDCs, research providers and industry stakeholders to achieve strategic outcomes for the industry and leverage higher returns for investments that demonstrate value for money for livestock exporters. This includes engaging with the technology startup and innovation ecosystem to solve major industry challenges and exploit opportunities.

LiveCorp would like to thank and acknowledge the contribution made by the importing company / vessel owner, the University of Sydney, Herd Health, Beanstalk AgTech and the dehumidification technology company. This project was funded by the Australian Government.

Overview of the Project

Challenge Statement: ***to find a proven and scalable solution for managing the on-board vessel environment to mitigate heat stress risk, optimise stocking rates and maintain license to operate.***

Australia exports sheep to the Middle East and North Africa via sea. These exports require the shipment of livestock through and into areas where there is a risk that wet bulb temperatures (heat and humidity) will reach high levels. Such climatic conditions can present a risk of heat stress for livestock, which requires mitigation.

At present, the livestock export industry uses a heat stress risk assessment (HSRA) framework developed by LiveCorp and Meat and Livestock Australia. The HSRA uses a range of weather, animal and ship related data – applied within a framework of logical assumptions, estimations and accepted risk thresholds – to determine the risks from a particular voyage. If the HSRA predicts that the risk thresholds will be exceeded, exporters are required to decrease the numbers of animals exported or are prevented from exporting at all.

In 2018, LiveCorp undertook a challenge-led Open Innovation project to explore technologies that could mitigate wet bulb temperatures from reaching levels that exceed the heat stress thresholds of sheep, but which also provide an environment that supports acclimation to destination country conditions.

From the initial technology scouting and assessment process, several technologies were identified that – either individually or in combination – could potentially reduce the risks or improve mitigation of heat stress.

These included:

- dehumidification
- route optimisation
- environmental monitoring data loggers, and
- data network connectivity.

It is worth noting that the scout process and subsequent media attention of livestock export has seen a continued organic stream of interest from technology providers with potential solutions. This continued attention from the innovation ecosystem is welcomed and LiveCorp continues to review any credible solutions presented. This broader technology interest will continue to be important as the next stages of research activities are considered.

Trial Program Overview

A technology trial program was developed to assess the effectiveness, viability and commercial scalability of each of the identified technologies that reduce the risk of heat stress.

A key aim of this approach is to de-risk the trial program by having five separate trials over four phases with distinct go/no-go decision making points (Table 1) – nominally between each, but particularly between Phase 2 and Phase 3.

The trial program was designed to be agile and adaptive to maximise the project's capacity to pivot as required in light of new information and data obtained. This approach provides the most efficient pathway and likely chance of identifying a successful technology solution that is cost effective, timely and scalable. Importantly, this approach also ensures LiveCorp has the line of sight to determine whether to persevere with the project, change direction, try a different approach, or cease it.

Table 1: Trial Program Structure

Phase	**PHASE 1 Pre-trial**	**PHASE 2 Static Trials**			**PHASE 3 Dynamic Trials**		**PHASE 4 Reporting**
Trial	**Pre-trial**	**Trial 1**	**Trial 2**	**Trial 3**	**Trial 4**	**Trial 5**	**Post-trial**
Details	Pre-trial Planning	Route Planning	Network Trials	**Static Dehumidification Trial**	Dynamic Chamber Trials	Dynamic Sea Trials	Final Report

Static Dehumidification Trial

Through the previously mentioned Open Innovation technology scout and consultation with industry, a specific type of dehumidification technology, known as DryCool, was identified as a technology with the potential to meet the objectives of the project. DryCool dehumidification removes moisture from air passing through the machine by using a desiccant wheel to absorb the water vapor and condense it into liquid water. However, unlike most other dehumidification technology, DryCool also conducts a small element of cooling to the air, which results in the outlet treated air having a lower relative humidity level and temperature than the ambient air when it entered the unit.

The dehumidification technology company conducted a desktop analysis using its proprietary performance simulation software to estimate the likely output of treated air based on ambient input conditions. This software was used to approximate the expected temperature and humidity given ambient conditions and, based on these results, it was identified that DryCool technology would be well suited to the proposed live export application. As such, in 2019 LiveCorp entered Phase 2 of vessel heat stress technology trial program and performed the *'Static Dehumidification Trial'.*

A Trial Working Group was established with experts in HVAC thermodynamics, scientific research trial methodology, statistical modelling, animal physiology / veterinary science, epidemiology and engineering. The Trial Working Group developed a trial methodology and protocol to scientifically test the performance of dehumidification on an empty livestock vessel with the objective to:

1. Develop a proof of concept of deck dehumidification and establish predictive effects on Wet Bulb Temperature (T_{WB}), and ventilation (Air Exchanges per Hour [AEH] / Pen Air Turnover [PAT]).
2. Collect baseline automated environmental data (temperature and humidity) during a commercial sheep voyage to the Middle East (May 2019).
3. Provide a proof of concept for livestock vessel dehumidification for heat stress mitigation (without livestock).
4. Provide clear information, including identifying barriers, risks etc., to guide a decision on whether, and how, to proceed with live animal trials (i.e. stop/go).

Metrics for determining the success of the experimental trial were developed and defined as:

1. Dehumidified air mixed with ambient air at reduced airflow rates provides an effective option to reduce the risk, magnitude and duration of wet bulb temperatures on a commercial vessel (without livestock).
2. Data and information are obtained that allows the effect of dehumidification on pen wet bulb temperature to be quantified.
3. Data and information are obtained that allows the combined effects of ventilation and dehumidification on pen wet bulb temperature on a commercial vessel (without livestock) to be quantified.
4. Environmental data collection technologies operate effectively on-board a livestock vessel and withstand the relevant challenges (e.g. water, dust, livestock), while effectively collecting relevant information.
5. Sufficient data and insights are obtained to enable preparation of a trial report and outline for how to proceed to the potential Phase 3 animal trial program for commercial validation.

METHODOLOGY

Trial Setup

Vessel

The trial was conducted aboard a vessel provided with the grateful assistance of a Middle Eastern importer and shipping company. It was a closed deck vessel, with 12 decks (10 of which could hold animals). The lower decks were separated into forward and aft, with a water-tight bulkhead between the holds. The two lowest decks on the vessel were used for the trial; 2 Forward (2F) and 3 Forward (3F). These were the smallest deck spaces available on the vessel and were the most suitable for the trial for several reasons, including:

- The smaller space increased the rate at which air turnover occurred and, as a result, increased the rapid change to deck conditions and faster equilibration following change to either fans or dehumidification settings.
- There were fewer spaces through which passive air flow could occur (leakage) – for example, cooler air sinking and hotter air rising – and the space was easier to seal, thereby reducing the risk of these variables impacting on testing.

Prior to the trial commencing, the vessel was de-stocked, cleaned and water troughs emptied.

Location

The trial was conducted in Dubai, with the vessel dockside in the water.

Several days were required to establish arrangements at the dock and aboard the vessel in preparation for the trial. The trial subsequently occurred between the dates of 24 and 28 June 2019.

During the trial, the vessel was orientated in a West-North-Westerly direction, with the starboard side in the full sun for the majority of the day.

Expertise

To ensure the trial was scientifically feasible within the constraints of a working vessel and dockyards, engineers from the University of Sydney advised, supported and observed the development of the trial methodologies and their delivery.

Equipment and operational lay-out

DryCool units

The trial used two HCU3000 and two HCU6000 DryCool dehumidifier units. The four Humidity Control Units (HCU) were located on the west side of the ship on the dock-side (Figure 1) while the vessel was moored. These units were placed at an angle offset from each other, with plastic sheeting installed between each exhaust and the adjacent unit's inlet to prevent re-ingestion of heated exhaust air. Each deck was ducted from one HCU6000 (10,000m^3/hr output) and one HCU3000 (5,000m^3/hr output) with a combined theoretical flow rate of 15,000m^3/hr of treated air.

Figure 1: Four HCU units

Ducting

Using polyethylene lay flat tube, treated air was ducted directly into the vessel via the deck 7 loading opening, with no loading ramp deployed due to site constraints. This arrangement required the treated air to travel at an acute inclination from the dockside to the vessel. From entry into the ship, the ducts separated, with those treating deck 3F running to the loading ramp (Figure 2), making a 90-degree bend (Figure 3) and running down the ramps to the constructed timber sealing between decks 4F and 3F (Figure 4). For deck 2F, the ducts were directed to an access hatch, making a 90-degree bend and running vertically down to the deck 2F (Figures 5 and Figure 6). The ducting entered deck 2F with a different orientation from 3F, entering vertically (Figure 7).

Figure 2: Ducting set up

Figure 3: Ducting set up

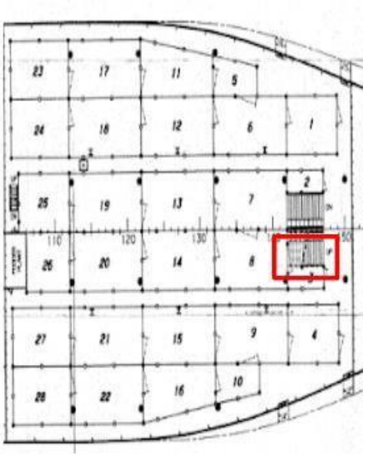


Figure 4: Ducting set up

Figure 5: Ducting set up

Figure 6: Ducting set up

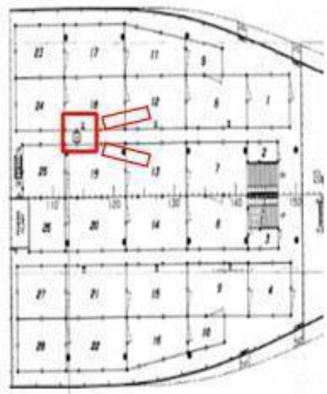


Figure 7: Ducting set up

Ducting flow release caps

It was recognised that the DryCool units have an approximate 10-minute lag time between being switched on and reaching full output. To avoid this lag impacting the quality of the trial data, each duct was fitted with a 'Y' shaped diverter join with a removable cap, fabricated from sheet metal (Figure 8). The cap was removed to divert the air from the ducting while still maintaining full operational capacity of the DryCool units and then replaced to instantly supply treated air down the ducting to the decks.

The diverter joins for the HCU3000 units were both located on the dock, while the diverter joins for the HCU6000 units were located on deck 7.

Figure 8: Y shaped diverter in the ducting

Sealing of the Decks

The decks were sealed at the loading ramps between decks 3F and 4F, as well as between decks 2F and 3F. Air cannot move between decks 2F and 1F as there are no livestock pens on 1F, hence it was already sealed off from 2F. The sealing of the decks isolated the two test zones to prevent the cold air from sinking and accumulating on the lower deck and the hot air rising to the higher deck (given the different environments between the other deck spaces, where no treatment was occurring).

Each of the test zones had plywood boarding that was then sealed with silicone around the open ramps and entry points (Figure 9). The ducting entered through cut outs and was sealed with duct tape (Figure 10).

In addition to sealing the loading ramps, the stairways between 4F and 3F and between 2F and 3F were sealed in a similar manner. All supply and exhaust fans were left unsealed for all trials.

Figure 9: Plywood boarding around ducts

Figure 10: Sealed ducts

Fans

The vessel had a total of 8 supply and 8 exhaust fans for decks 2F and 3F. The vessel owner provided the pen air turnovers, air exchange and air flow figures for the decks, which were used by the engineer advising the project to calculate the sequencing of the supply and exhaust fans for each of the trials. Further air flow and exchange measurements were taken to underpin the calculations and inform the analysis.

Data Logging Equipment

Kestrel D3FW Drop

The Kestrel D3FW Drop was the primary data logger used for capturing the temperature and humidity for the trial (Figure 11).

The D3FW is a small, portable, battery powered logger that measures dry bulb temperature (T_{DB}), relative humidity (RH), and wet bulb temperature (T_{WB}). The D3FW Drop has Bluetooth connectivity, which was used for data transfer.

Figure 11: Primary data logger

Kestrel 5400 Cattle Heat stress Tracker

A handheld Vane Prop Anemometer was used for air velocity mapping on the test decks.

Tinytag

The Tinytag Ultra 2 was used to log indoor temperature and relative humidity data.

Vaisala Temperature and Humidity

The Vaisala HM42 handheld with HM46PROBE (calibrated 3 May 2019) was used to measure the temperature and humidity inside the ducting.

Vaisala Air Velocity Measurement

The Testo 435-4 with hot wire probe, article No. 0560 4354, and Hot wire Probe, article No. 0633-1025 (calibrated 2 May 2019), was used to measure the air velocity inside the ducting.

FLIR Thermal Imaging

A handheld thermal imaging camera (Model: FLIR C2) was used to capture supporting data on the impact of radiant solar heat on the vessel.

Data Logger Layout

29 data loggers were arranged in a linear array across the two trial decks (Figure 12 and Figure 13). All loggers that made up the linear array were suspended on the sheep pens at a height of approximately 93cm (Figure 14).

Individual loggers were mounted directly at the opening outlet of all four ducts where the air entered the trial decks (Figure 12, 13 and 15).

One logger was installed in the Stevenson screen outside on the upper bridge deck (alongside the vessel's wet bulb thermometer- Figure 16) and another was installed at the air intake of one of the DryCool units at the dock side (Figure 17).

A further five remaining loggers were placed at varying locations consistent with the linear array to provide more detailed mapping of the movement of dehumidified air.

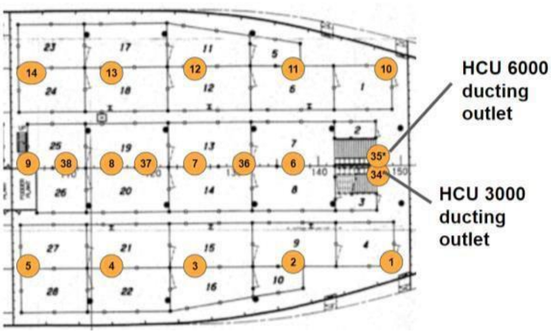


Figure 12: Deck 3F logger locations

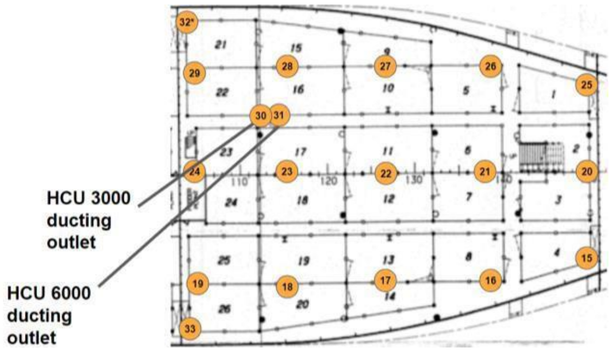


Figure 13: Deck 2F logger locations

Figure 14: Sheep pen logger locations

Figure 15: Duct logger locations

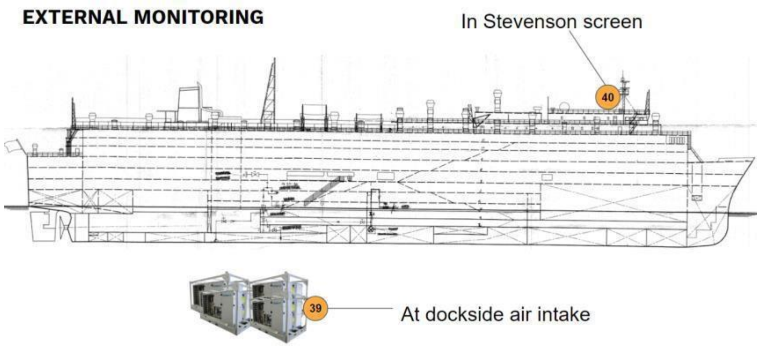


Figure 16: Stevenson screen logger

Figure 17: Dockside logger location

Pre- trial logger and equipment testing

Before the trial commenced, a validation exercise of the Kestrel loggers was undertaken to identify their range and patterns of operation and confirm their fitness for use on the vessel. This information subsequently informed the analysis of the trial data.

The validation process included the bundling of the devices and keeping them within a cool, dry environment for a period before moving them to a hot, humid environment. The results identified that there was variation of potentially +/- 1 degree in the dry bulb temperatures of the loggers as they adjusted to the changes, but that overtime this narrowed to a smaller range. Similarly, relative humidity recordings revealed a range of around 7% between individual loggers. The trends of the loggers in operation were consistent. It was also identified that – particularly when moved between extremes of temperature – the dry bulb temperature recorded adjusted more slowly than the humidity readings, which were very responsive.

One logger (#35) started to record erratic data on the second day of the Trial, and hence this was removed from the subsequent analysis of the trials.

Velocity mapping and thermal camera recordings

Velocity Mapping Recording Methodology	
The purpose of this test was to verify the air velocity from the ventilation system on deck 3F. The Kestrel 5400 unit was used to perform the mapping. Each air velocity reading was taken at a position near to each Kestrel data logger (positioned at pen height – approximately 93cm high). The air velocity, T_{DB} and RH were recorded. In addition to taking readings in the pen area, the side walls and steel bulkhead were also included. The following outlines the methodology applied to this process.	
(i) Objective:	To perform mapping of velocity and temperature profile with only supply and exhaust fans in operation. This trial will serve as a base validation case.

(ii) Setup:	- Deck 3F only - All supply and exhaust fans switched on - DHM running at 0% - Sealing between deck 2/3 and 3/4 - Safety review
(iii) Treatments:	Stage 1: Base (validation) 1: 0% DHM + 100% Supply and Exhaust Fans (20min) 2: Mapping velocity profile on the 15 grid points (30min)
(iv) Replicates:	3 x 20min cycles + 3 x 30min (mapping)= 150min (2.5hr)
(v) Recording:	- Date, Time of trial - Airflow from duct (LFM, Linear Feet/Minute corrected to m^3/hr) - Datalogging: WBT, RH, ^{O}C - Hot wire anemometer: Air velocity and T_{DB} - Infrared Thermometer: T_{DB} - Infrared Camera: Temperature contour for ground (centreline leaning to bow) and walls (centre of the deck) - Stable Temperature = movement $0^{O}C$ over 5min

Thermal Camera Imaging Methodology

Thermal images were taken at the centre of deck 3F, including the bow and stern direction steel bulkheads and the two side walls. To capture ground and ceiling thermal images, the thermal image camera was positioned at the steel bulkhead (from stern direction facing bow direction). The following summarises the methodology applied for the thermal image captures.

(i) Objective:	To perform mapping of velocity and temperature profile with supply and exhaust fans and DHM units running. This trial will serve as a base validation case.
(ii) Setup:	- Deck 3F only - All supply and exhaust fans switched on - DHM running at 100% - Sealing between deck 2/3 and 3/4 - Safety review
(iii) Treatments:	Stage 1: Base (validation) 1: 100% DHM + 100% Supply and Exhaust Fans (20min) 2: Mapping velocity profile on the 15 grid points (30min)
(iv) Replicates:	3 x 20min cycles + 3 x 30min (mapping)= 150min (2.5hr)
(v) Recording:	- Date, Time of trial - Airflow from duct (LFM, Linear Feet/Minute corrected to m^3/hr) - Datalogging: WBT, RH, ^{O}C - Hot wire anemometer: Air velocity and T_{DB} - Infrared Thermometer: T_{DB} - Infrared Camera: Temperature contour for ground (centreline leaning to bow) and walls (centre of the deck) - Stable Temperature = movement $0^{O}C$ over 5min

Trial Design

The trial series aimed to develop a proof of concept of deck dehumidification and establish predictive effects on Wet Bulb Temperature (T_{WB}), Dry Bulb Temperature (T_{DB}) and Relative Humidity (RH) with no livestock on-board the vessel.

For the purpose of the trials, it must be noted that latent heat load is the heat required to turn a solid into a liquid or gas, or a liquid into a gas, without change of temperature. The energy released by the thermodynamic system can affect the state or phase of the air but not the temperature e.g. removes the amount of moisture in the air, but not change the temperature. Sensible heat is heat exchanged by a thermodynamic system that changes the temperature of the system (air) e.g. air being cooled by cooling coils or heated by heating coils.

The following sections outline the trial methodologies that were developed and applied during the project. It is noted that all times identified in the following methodologies are pre-trial estimates of the time required for a rate response to be observed. Where rate responses varied during the practical trial, the actual times were adjusted accordingly.

Trial 3A - Sequential ambient baseline dehumidification rate response

(i) Objective

The aim of this trial was to establish an ambient baseline for sequential dehumidification rate response under sealed conditions to support the modelling of predictive impact on T_{WB}, RH and T_{DB}.

(ii) (ii) Setup

Decks 2F and 3F were sealed and Supply and Exhaust (S&E) fans were controlled via the fan control room as per the required sequence. The highest DHM rate (15,000 m^3/hr) was selected as the first treatment in order to determine the impact of full dehumidification.

(iii) Treatments

All S&E Fans were switched on for 15-20min between treatments in order to achieve ambient conditions.

Treatment	DHM Capacity	Units	Time
1	15,000m^3	1xHCU3000 + 1xHCU6000	30min
2	10,000m^3	1xHCU6000	30min
3	5,000m^3	1xHCU3000	30min

(iv)) Replicates

3 x 40min cycle = 120min (2hr).

(iv)) Recording

- Rate Response and Time taken to achieve Stable T_{WB} and RH
- Rate Response and Time taken to achieve Ambient T_{WB} and RH
- Data loggers (T_{WB}, RH, T_{DB}): 40 x Kestrel D3FW, 2 x Vaisala Probe Units

Trial 3B - Cumulative dehumidification rate response

(i) Objective

The objective of this trial was to establish a cumulative dehumidification rate response under sealed deck conditions to support modelling of the predictive impact on T_{WB}, RH and T_{DB}.

This trial contained similar treatments as outlined in Trial 3A, the key difference being that S&E fans were not switched on between treatments so that each deck received an accumulated dehumidification load starting from 5,000m^3/hr, building to 10,000m^3/hr and then 15,000m^3/hr.

Following the completion of each trial, the time taken for each deck space to return to ambient was recorded.

(ii) (ii) Setup

Decks 2F and 3F were sealed and S&E fans were controlled via the fan control room, as per the required sequence.

(iii) Treatments

Treatment	DHM Capacity	Units	Time
1	5,000 m^3/hr	1xHCU3000	30min
2	10,000 m^3/hr	1xHCU6000	30min
3	15,000 m^3/hr	1xHCU3000 + 1xHCU6000	30min

(iv)) Replicates

3 x 80min cycle = 240min (4hr).

(iv) Recording

- Rate Response and Time taken to achieve Stable T_{WB} and RH
- Rate Response and Time taken to achieve Ambient T_{WB} and RH
- Data loggers (T_{WB}, RH, T_{DB}): 40 x Kestrel D3FW, 2 x Vaisala Probe Units

Trial 3C - Ambient airflow dehumidification dilution

(i) Objective

The objective of this trial was to establish the dilution impact on dehumidified air of introducing sequenced ambient ventilation fans under sealed deck conditions.

(ii) (ii) Setup

Decks 2F and 3F were sealed and dehumidification units were operated at 100% for this trial (ie. 15,000 m^3/hr for each deck). S&E fans were controlled via the fan control room as per the required sequence.

(iii) Treatments

Treatment	DHM Capacity	Fan Capacity	Time
1	100%	25%	20min
2	100%	50%	20min
3	100%	100%	20min

(iv)) Replicates

3 x 40min cycles = 120min (2hr)

(iv)) Recording

- Rate Response and Time taken to achieve Stable T_{WB} and RH
- Rate Response and Time taken to achieve Ambient T_{WB} and RH
- Airflow mapping (LFM, Linear Ft/Min correlated to m^3/hr) at set locations
- Thermal Camera
- Data loggers (T_{WB}, RH, T_{DB}): 40 x Kestrel D3FW, 2 x Vaisala Probe Units

Trial 3D - Ambient airflow rate response (no dehumidification)

(i) Objective

The objective of this trial was to establish an ambient airflow rate response baseline under sealed deck conditions to support modelling of the predictive impacts on T_{WB}, RH and T_{DB}. This would assist comparisons to Trial 3C.

(ii) (ii) Setup

Decks 2F and 3F were sealed and S&E fans were controlled via the fan control room, as per the required sequence. No dehumidification (DHM) was operating.

(iii) Pre-trial

- All S&E fans were turned ON for ambient conditions.
- Note that treatments in Trial 3D were reversed from Trial 3C as the vessel engineer requested less generator load on the sequenced fan start-up. In this way, all fans could be initiated progressively prior to the start of the trial and then turned off as required without impact to the cut-out switches.

(iii) Treatments

Treatment	DHM Capacity	Fan Capacity	Time
1	0%	100%	20min
2	0%	50%	20min
3	0%	25%	20min

(iv)) Replicates

3 x 75min cycle = 180min (3hr).

(iv)) Recording

- Rate Response and Time taken to achieve Stable T_{WB} and RH
- Rate Response and Time taken to achieve Ambient T_{WB} and RH
- Airflow mapping (LFM, Linear Ft/Min correlated to m^3/hr) at set locations
- Thermal Camera
- Data loggers (T_{WB}, RH, T_{DB}): 40 x Kestrel D3FW, 2 x Vaisala Probe Units

Trial 3E - Cumulative exhaust fan rate response

(i) Objective

The objective of this trial was to establish the effect of cumulative exhaust arrangements on dehumidification conditions under sealed deck conditions.

(ii) (ii) Setup

The vessel electrical engineer switched the polarity of selected supply fans in order to reverse the fans to exhaust. Three exhaust fan sequences (i.e. treatments) were then selected in order to increase air velocity across the deck.

The position of the DHM outlets varied between 3F (supply from ramp) and 2F (supply from the access hatch). It was acknowledged that 3F was a more representative deck setup as the DHM outlets were located at the opposite end of the deck from the exhaust fans.

(iii) Treatments

DHM air (100%, 15,000m^3/hr) was introduced to decks 2F and 3F and allowed to stabilise at the start of the trial. Switching of fan sequences occurred immediately between treatments without altering DHM airflow.

Treatment	**DHM Capacity**	**Fan Capacity**	**Time**
1	100%	2	20min
2	100%	4 (2 to 4)	20min
3	100%	8 (4 to 8)	20min

(iv)) Replicates

2 x 80min cycle = 160min (2:40hr).

(iv)) Recording

- Rate Response and Time taken to achieve Stable T_{WB} and RH
- Rate Response and Time taken to achieve Ambient T_{WB} and RH
- Airflow mapping (LFM, Linear Ft/Min correlated to m^3/hr) at set locations
- Thermal Camera
- Data loggers (T_{WB}, RH, T_{DB}): 40 x Kestrel D3FW, 2 x Vaisala Probe Units

Trial 3F - Sequential exhaust fan rate response

(i) Objective

The objective of this trial was to establish the effect of sequential exhaust arrangements on dehumidification conditions under sealed deck conditions.

(ii) (ii) Setup

The vessel electrical engineer switched the polarity on selected supply fans in order to reverse the fans to exhaust. Three exhaust fan sequences (i.e treatments) were selected in order to increase air velocity across the deck.

The position of the DHM outlets varied between 3F (supply from ramp) and 2F (supply from the access hatch). It was acknowledged that 3F was a more representative deck setup as the DHM outlets were located at the opposite end of the deck from the exhaust fans.

(iii) Treatments

After each treatment was completed, all the fans were switched on and the decks were allowed to stabilise near to ambient conditions.

Treatment	DHM Capacity	Fan Capacity	Time
1	100%	2 (0 to 2)	20min
2	100%	4 (0 to 4)	20min
3	100%	8 (0 to 8)	20min

(iv)) Replicates

2x80min = 160min (2.40hr).

(iv) Recording

- Rate Response and Time taken to achieve Stable T_{WB} and RH
- Rate Response and Time taken to achieve Ambient T_{WB} and RH
- Airflow mapping (LFM, Linear Ft/Min correlated to m^3/hr) at set locations
- Thermal Camera
- Data loggers (T_{WB}, RH, T_{DB}): 40 x Kestrel D3FW, 2 x Vaisala Probe Units

Preparation of data for analysis

The average T_{DB}, RH and T_{WB} are presented for each deck in the following graphs. The average properties (T_{DB}, RH and T_{WB}) of each deck are reported, rather than the individual data streams from each of the 17 Kestrel data loggers on each deck for three primary reasons:

1) Plotting all 21 curves for each deck (17 space + 2 ambient + 2 dehumidifier duct outlet) on a single graph would be very dense and almost impossible to interpret.
2) The measurements from each data logger contains a combination of noise and systematic bias. By averaging over several loggers (17), the effects of noise and bias are minimised.

3) There is expected to be small variations in the space while at steady state. The primary quantities of interest are the average properties at steady state in the deck.

Where available, the data captured by the two data loggers at each of the dehumidifier duct outlets on each deck were not averaged. As expected, there were variations in the performance of each dehumidifier, and capturing this difference was important for interpretation of the results.

Likewise, the two ambient temperature measurements from data logger 40 (LC040 – located on the vessel's bridge) and data logger 39 (LC039 - located next to the dehumidifier intake grill) have not been averaged. A small difference between the vessel bridge temperature and the temperature at ground level (where the dehumidifier units were located) is expected due to the difference in elevation and radiant temperatures between the locations. The bridge temperature is most likely indicative of the fresh air supply conditions. The dehumidifier inlet T_{DB} and RH needed to be determined as accurately as possible for the dehumidifier sensible and latent load calculations.

Cleaning the Data

When all the data was collated from the loggers at the end of the trial, some variation in the timestamp of the data was identified, although this was resolved. In terms of logger failure rates, data logger No. 35 (LC035) started to show some problems on the afternoon of 24 June 2019. After analysing the data from that logger, it was concluded that it had failed to record data correctly and was completely removed from the trial. On 27 June 2019, data logger No.32 (a supplementary logger that was proving unnecessary in its current location) was taken from deck 2F to replace data logger No. 35.

RESULTS

Introduction

Assessing DHM unit performance and potential influences on duct work

To determine the operating performance of the dehumidification units and any possible heat transfer or air ingress from the delivery duct work;

- a data logger was placed at the ambient inlet of each dehumidifier unit
- a probe data logger measured the output conditions directly at the outlet of the dehumidifiers, and
- a data logger was placed at the exit of each of the ducts supplying dehumidified air to the deck.

The air flow rates measured at the outlet of each dehumidification unit with a probe were within, at most, ±20% of the rated flow rate for the unit, which is within expected range of operation. However, the data collected did not fully correlate the T_{DB} and the RH measured from the units, with the conditions measured in the trial deck space. This suggested that there was potentially some heat transfer or air ingress occurring from the delivery duct work (i.e. the dehumidified air was being influenced during its movement from the unit to the deck space, for example by radiant heat). This also indicates the results from the trial reflect a worst-case scenario in terms of the efficiencies and performance of the dehumidification units, as compared to a situation where they could be actually fitted on a vessel (e.g. with less ducting loss).

There were some difficulties due to time stamp variations between some recordings – particularly with regard to when the flow rate measurements were taken. However, there were several cases where the measurement of temperature and relative humidity at the exit of the dehumidifier could be considered reliable and permitted the cooling load of the dehumidifier and heat load calculations along the supply ductwork to be obtained (although without simultaneous measurement of the flow rate).

The total heat load, together with the sensible and latent heat loads for both the dehumidifiers and the supply ductworks across various days, are computed and tabulated in Table 2. The sensible and latent heat load of the dehumidifiers and ductworks are calculated based on comparing the conditions of the air at the inlet and outlet of the dehumidification units, and the air conditions at the inlet (dehumidifier outlet) and outlet of the ductwork respectively. Due to the failure of data logger #35, the ductwork heat load could not be computed for the duct connecting from HCU 3000 to deck 3F.

Notes relating to the data reported in Table 2.

1. *There is a 4-minute delay turning off the fans.*
2. *The HCU6000 unit cap was removed, and the Y-branch was placed side by side with the other duct, and an air short circuit is suspected.*
3. *Duct issue solved at 11:41. This reading is obtained when the 15,000m^3/hr step in 4 minutes prior. Thermal equilibrium may not be reached at this state.*
4. *From Trial 3C, both RH and T_{WB}, after 17:21 are still reducing in value before the fans are increased to 100% at 17:35. Thermal equilibrium is not reached for this case.*
5. *The high load in the ductwork may be attributed to the trial just starting 2 minutes prior. It takes more than 5 minutes to cool down the duct outlet. Thermal equilibrium not reached for this case.*
6. *The HCU3000 unit cap was removed.*
7. *The HCU6000 unit cap was removed, and the Y-branch was placed side by side with the other duct, and an air short circuit is suspected.*
8. *The logger #34 is placed between the duct outlets and is further down the ramp. Hence, the higher RH value is expected.*

Table 2: Sensible, latent, and total heat load calculations for the dehumidifier and supply ductwork. Only data where partial (only dehumidifier) or complete (dehumidifier and supply ductwork) heat load calculations can be made are shown.

Deck & Unit	Time (Dubai) 2019	DHM intake (LC039)			DHM direct output			Duct outlet to space			Rated Flowrate	DHM Heat Loads (kW)			Ductwork Heat Loads (kW)			Trial (Replicate)	Trial Period (time 24hr)	Note
		T_{DB} (°C)	RH (%)	T_{WB} (°C)	T_{DB} (°C)	RH	T_{WB} (°C)	T_{DB} (°C)	RH (%)	T_{WB} (°C)	$V\ (m^3/h)$	Sensible	Latent	Total	Sensible	Latent	Total			
2F 3000	25/06/2019 19:29	35.9	65.2	30.0	35.7	45.80	25.8	36.4	44.0	25.9	5000	0.32	30.60	30.92	1.12	-0.11	1.01	3B (R1)	1915 - 1935	5000 m³/hr - Cumulative DHM rate response
2F 3000	25/06/2019 21:09	35.9	67.5	30.4	36.5	45.70	26.5	35.8	45.4	25.7	5000	-0.96	31.13	30.17	-1.12	-3.15	-4.26	3B (R2)	2101 - 2121	5000 m³/hr - Cumulative DHM rate response
2F 3000	26/06/2019 11:29	36.2	58.0	28.8	36.3	42.30	25.6	36.1	44.5	25.8	5000	-0.16	24.04	23.88	-0.32	2.66	2.34	3B (R3)	1118 - 1138	5000 m³/hr - Cumulative DHM rate response
2F 3000	26/06/2019 12:13	35.9	63.9	29.7	36.8	41.20	25.7	35.1	41.2	24.2	5000	-1.44	31.56	30.13	-2.70	-5.82	-8.53	3B (R3)	1158 - 1226	15000 m³/hr - Cumulative DHM rate response
2F 3000	26/06/2019 17:26	37.8	51.1	28.6	35.3	41.70	24.6	34.5	38.2	23.0	5000	4.01	25.05	29.06	-1.28	-7.54	-8.82	3C (R3)	1735 - 1810	15000 m³/hr + 50% fan
2F 3000	25/06/2019 20:01	36.3	64.6	30.2	36.2	44.00	25.9	35.4	44.0	25.1	5000	0.16	32.72	32.88	-1.28	-2.92	-4.19	3B (R1)	1955 - 2015	15000 m³/hr - Cumulative DHM rate response
2F 3000	27/06/2019 18:39	36.5	54.8	28.4	35.5	45.50	25.6	36.2	42.3	25.3	5000	1.60	18.56	20.16	1.12	-2.29	-1.17	3F (R2)	1840 - 1901	15000 m³/hr + 2 exhaust fans (9.1 & 9.4)
2F 3000	25/06/2019 15:33	37.0	58.5	29.6	38.0	42.00	26.9	36.0	45.7	25.9	5000	-1.59	22.92	21.33	-3.18	-1.73	-4.90	3A (R1)	1510 - 1540	5000 m³/hr
2F 3000	25/06/2019 16:27	39.9	54.6	31.3	37.2	40.80	25.9	36.0	42.6	25.2	5000	4.30	36.88	41.18	-1.91	-1.47	-3.38	3A (R2)	1620 - 1652	5000 m³/hr
2F 6000	25/06/2019 17:45	36.5	60.8	29.6	35.4	44.60	25.4	37.6	44.2	26.9	10000	3.52	59.78	63.31	7.07	15.57 [#2]	22.64	3A (R3) [#1]	1732 - 1753	5000 m³/hr
2F 6000	25/06/2019 19:31	36.4	67.3	30.8	35.3	45.20	25.4	36.5	45.7	26.4	10000	3.53	78.54	82.07	3.86	10.65 [#2]	14.50	3B (R1)	1915 - 1935	5000 m³/hr - Cumulative DHM rate response
2F 6000	25/06/2019 21:10	35.9	67.8	30.5	35.6	44.80	25.6	36.7	45.8	26.6	10000	0.96	73.11	74.07	3.53	11.52 [#2]	15.05	3B (R2)	2101 - 2121	5000 m³/hr - Cumulative DHM rate response
2F 6000	26/06/2019 12:02	35.4	65.3	29.5	34.7	41.50	24.1	36.8	43.4	26.1	10000	2.25	76.05	78.30	6.77	20.57 [#3]	27.34	3B (R3)	1158 - 1226	15000 m³/hr - Cumulative DHM rate response
2F 6000	26/06/2019 17:21	38.8	50.3	29.3	34.5	40.00	23.6	36.6	42.1	25.6	10000	13.81	67.38	81.19	6.78	20.47 [#4]	27.24	3C (R3)	1715 - 1735	15000 m³/hr + 50% fan
2F 6000	25/06/2019 20:02	37.4	65.2	31.4	35.2	44.10	25.1	36.4	44.0	25.9	10000	7.05	87.66	94.72	0.64	1.13	1.78	3B (R1)	1955 - 2015	15000 m³/hr - Cumulative DHM rate response
2F 6000	27/06/2019 18:42	38.2	54.9	29.8	34.7	41.20	24.0	36.8	44.0	26.2	10000	11.24	73.00	84.24	6.78	23.35 [#5]	30.13	3F (R2)	1840 - 1901	15000 m³/hr + 2 exhaust fans (9.1 & 9.4)

3F 3000	25/06/2019 19:39	35.5	65.6	29.6	31.6	60.50	25.4	The logger #34 is initially placed on the duct outlet of the HCU 3000. It is later swapped with logger #35 (originally placed in the outlet of HCU 6000 in deck 3) due to malfunction of logger # 35. No data available due to removal of logger #35 on 25th June 2019.			5000	6.33	25.82	32.14	The ductwork loads cannot be determined as there is no information at the duct outlet to space due to the malfunction of logger #35.			3B (R1) [#6]	1935 - 1955	10000 m^3/hr - Cumulative DHM rate response
3F 3000	25/06/2019 21:07	36.2	67.6	30.7	32.0	57.80	25.2				5000	6.81	34.65	41.46				3B (R2)	2101 - 2121	5000 m^3/hr - Cumulative DHM rate response
3F 3000	26/06/2019 11:43	35.5	62.8	29.1	31.4	59.10	24.9				5000	6.66	24.06	30.72				3B (R3) [#6]	1138 - 1158	10000 m^3/hr - Cumulative DHM rate response
3F 3000	26/06/2019 12:04	35.4	65.7	29.6	31.6	58.30	24.9				5000	6.17	28.11	34.27				3B (R3)	1158 - 1226	15000 m^3/hr - Cumulative DHM rate response
3F 3000	26/06/2019 12:31	35.5	61.4	28.9	31.4	58.70	24.9				5000	6.66	22.39	29.05				3B (R3)	1158 - 1226	15000 m^3/hr - Cumulative DHM rate response
3F 3000	26/06/2019 17:16	38.2	51.4	29.1	29.7	58.80	23.4				5000	13.87	26.22	40.10				3C (R3)	1715 - 1735	15000 m^3/hr + 50% fan
3F 3000	25/06/2019 20:06	36.4	65.6	30.5	32.0	58.40	25.3				5000	7.13	31.85	38.98				3B (R1)	1955 - 2015	15000 m^3/hr - Cumulative DHM rate response
3F 3000	27/06/2019 18:51	37.3	56.8	29.5	30.5	59.10	24.1				5000	11.07	27.57	38.64				3F (R2)	1840 - 1901	15000 m^3/hr + 2 exhaust fans (9.1 & 9.4)
3F 3000	25/06/2019 15:35	36.8	59.2	29.5	33.2	55.50	25.7				5000	5.81	22.32	28.13				3A (R1)	1510 - 1540	5000 m^3/hr
3F 3000	25/06/2019 16:24	37.6	54.7	29.3	32.8	52.00	24.8				5000	7.76	25.17	32.93				3A (R2)	1620 - 1652	5000 m^3/hr
3F 6000	25/06/2019 17:45	36.5	60.8	29.6	34.7	56.80	27.3	35.8	54.8	27.8	10000	5.79	29.81	35.60	3.54	4.10 [#7]	7.64	3A (R3)	1732 - 1753	5000 m^3/hr
3F 6000	25/06/2019 19:33	35.8	64.8	29.8	36.7	45.30	26.5	36.2	48.7	26.8	10000	-2.87	52.43	49.55	-1.60	6.62 [#7]	5.02	3B (R1)	1915 - 1935	5000 m^3/hr
3F 6000	25/06/2019 21:06	36.1	67.1	30.5	36.9	46.80	27.1	35.8	50.6	26.8	10000	-2.55	56.48	53.92	-3.51	2.70 [#7]	-0.81	3B (R2)	2101 - 2121	5000 m^3/hr
3F 6000	26/06/2019 12:07	35.7	65.8	29.9	36.6	42.90	25.9	36.4	43.3	25.7	10000	-2.87	62.84	59.97	-0.64	-0.22	-0.86	3B (R3)	1158 - 1226	15000 m^3/hr
3F 6000	26/06/2019 17:18	38.7	50.9	29.4	35.7	42.40	25.1	35.2	48.5	26	10000	9.60	53.22	62.82	-1.61	14.38 [#4, #8]	12.77	3C (R3)	1715 - 1735	15000 m^3/hr + 50% fan
3F 6000	25/06/2019 20:04	35.9	64.8	29.9	36.6	45.20	26.5	36.5	46.4	26.6	10000	-2.24	54.66	52.42	-0.32	2.99	2.67	3B (R1)	1955 - 2015	15000 m^3/hr
3F 6000	27/06/2019 18:46	37.2	56.9	29.4	36.7	41.00	25.5	36.6	42.5	25.7	10000	1.60	55.86	57.46	-0.32	4.01	3.69	3F (R2)	1840 - 1901	15000 m^3/hr + 2 exhaust fans (9.1 & 9.4)

Formatting of figures and diagrams

The following sections contain graphical representations and descriptions of the findings from the different trials. The information presented in the figures for each trial reflect the most representative replicates that were conducted.

In terms of formatting the figures (graphs / diagrams), they have all been given a consistent ordinate axis increment for the respective variable (T_{DB}, RH, T_{WB}) to aid in a consistent comparison between trials and replicates. The following summarises the details for each of the variables:

- The T_{DB} figures in the figures are all scaled to have a range of $\Delta T = 10^{\circ}C$. Most of the T_{DB} figures have limits from $30^{\circ}C$ to $40^{\circ}C$, with only a few ranging between $32^{\circ}C$ to $42^{\circ}C$.
- The RH figures are all scaled to have a range of $\Delta RH = 40\%$. Most of the RH figures range between 30 to 70%, with only a few ranging between 40 to 80%.
- The T_{WB} figures are all scaled to have a range of $\Delta T_{WB} = 15^{\circ}C$. All of the T_{WB} figures have a range between $20^{\circ}C$ to $35^{\circ}C$.

To aid in the interpretation of the figures, a consistent colour scheme has also been used for all figures, which is outlined below.

References to ‘space’ in the following sections refers to the trial decks (e.g. the space average, is the trial deck average). As the space average gets nearer to the DHM outlet measurements, it suggests a greater influence from the conditioned air provided from the DHM. Where the space average is closer to the ambient or intake data recordings, it shows that the space is nearer to an unconditioned state. Within the graphs, a closeness between these figures could suggest that the influence of the DHM conditioned air is lower or show the periods within the trials where the fans were used to return the deck spaces to the baseline ambient conditions between trials. The following results and figures presented demonstrate the most representative replicates completed for the various trials.

Plot Interpretation

Deck 3F plots (top figure of each page)

Logger	Marker	Properties
Logger 34 at deck 3 (LC034)		T_{DB}, RH, T_{WB} of the dehumidifier air at the supply duct outlet in the space
Logger 32 (replacement for logger LC035) at deck 3 (LC032)		T_{DB}, RH, T_{WB} of the dehumidifier air at the supply duct outlet in the space
Logger 40 (LC040)		T_{DB}, RH, T_{WB} of the bridge (outside air)
Logger 39 (LC039)		T_{DB}, RH, T_{WB} of air intake for a dehumidifier (outside air)
Average properties of the space		T_{DB}, RH, T_{WB}

Deck 2F plots (bottom figure of each page)

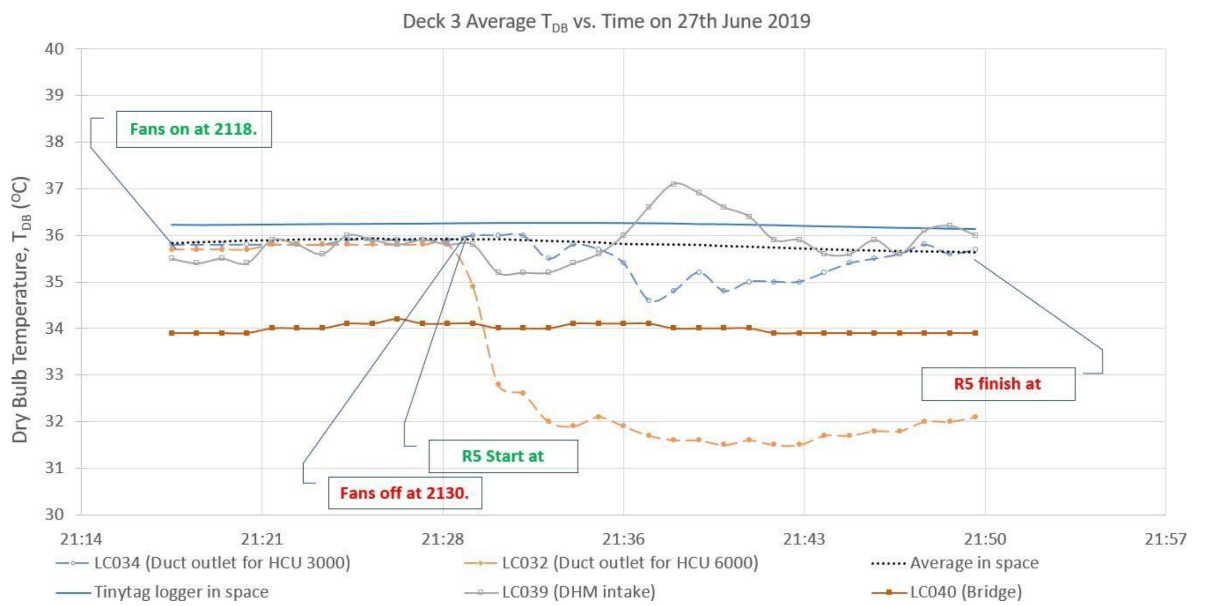
Logger	Marker	Properties
Logger 30 at deck 2 (LC030)		T_{DB}, RH, T_{WB} of the dehumidifier air at the supply duct outlet in the space
Logger 31 at deck 2 (LC031)		T_{DB}, RH, T_{WB} of the dehumidifier air at the supply duct outlet in the space
Logger 40 (LC040)		T_{DB}, RH, T_{WB} of the bridge (outside air)
Logger 39 (LC039)		T_{DB}, RH, T_{WB} of air intake for a dehumidifier (outside air)
Average properties of the space		T_{DB}, RH or T_{WB}

Trial 3A - Sequential ambient baseline dehumidification rate response

The aim of this trial was to establish an ambient baseline for sequential dehumidification rate response under sealed conditions to support modelling of predictive impact on T_{WB}, RH and T_{DB}. Decks 2F and 3F were sealed and all supply and exhaust fans were switched on between treatments to achieve ambient conditions before application of DHM.

Three treatments were applied: 15,000m^3/hr; 10,000m^3/hr and 5,000m^3/hr – with the highest DHM rate (15,000m^3/hr) selected as the first treatment in order to determine the impact of full dehumidification. Figures 18, 19 and 20 provide the results from the 15,000m^3/hr treatment.

Dry bulb temperature results



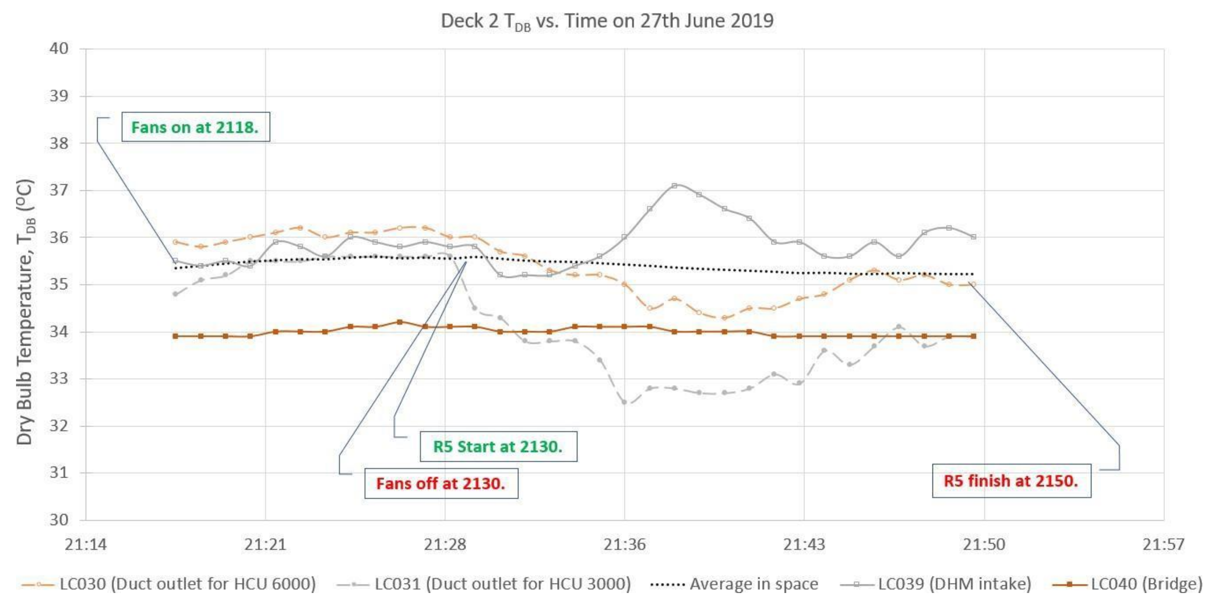
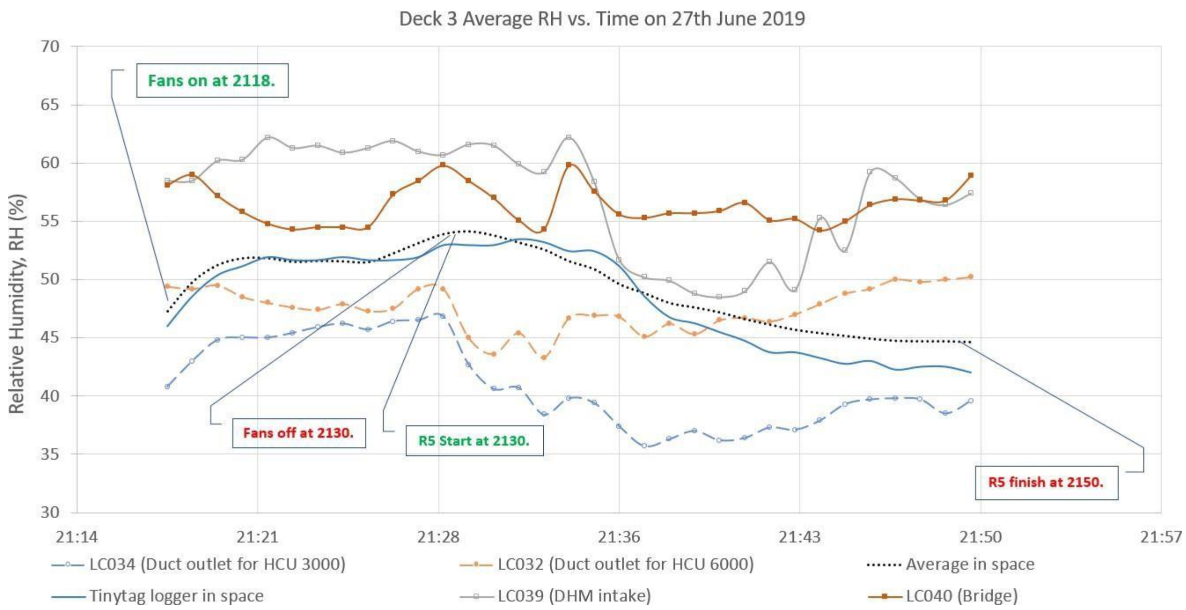


Figure 18: The Dry Bulb Temperature (T_{DB}) rate response for dehumidifier units delivering 15,000 m^3/hr of dehumidified air to deck 3 (top) and deck 2 (bottom)

- During this replicate, when the dehumidifier units were switched on at 21:30, there is a decrease of 0.3°C to 0.5°C of the average T_{DB} in both decks for 20min.
- The variation between the Kestrel loggers' average space $T_{DB,}$ and the Tinytag data logger T_{DB} was approximately 0.5°C for deck 3.
- Although the Bridge (LC040) T_{DB} is lower than the space average for both decks, the T_{DB} of the dehumidifier intake (LC039) is very close to the average space T_{DB} across all decks.
- The duct outlets (LC030, LC031, LC032 and LC034) registered a drop in T_{DB} differently.

<u>Relative humidity results</u>



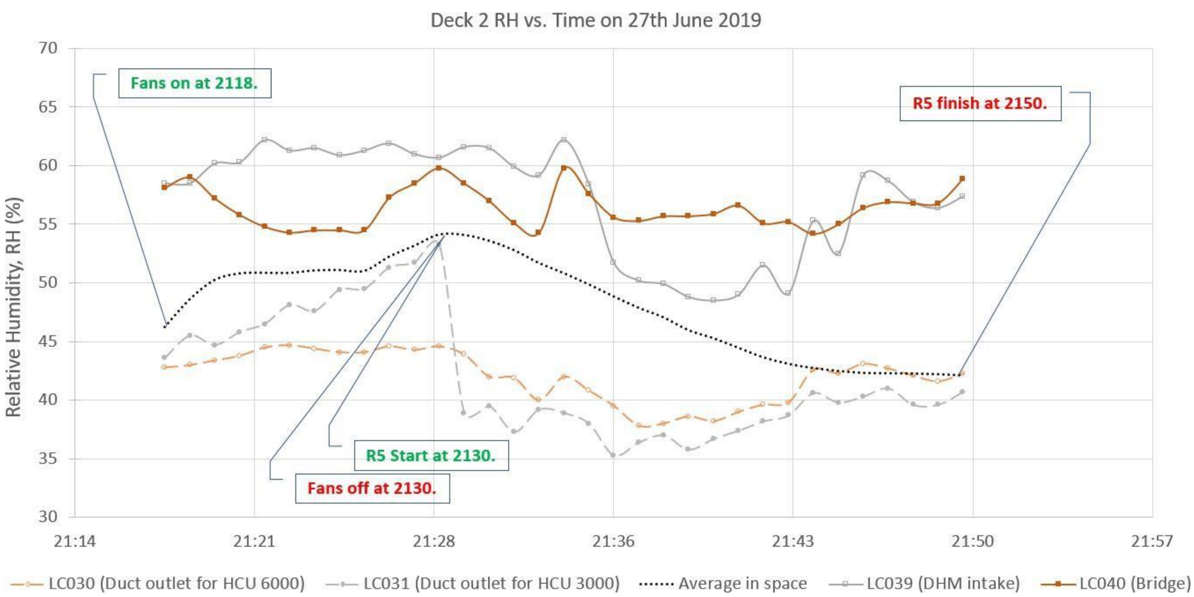
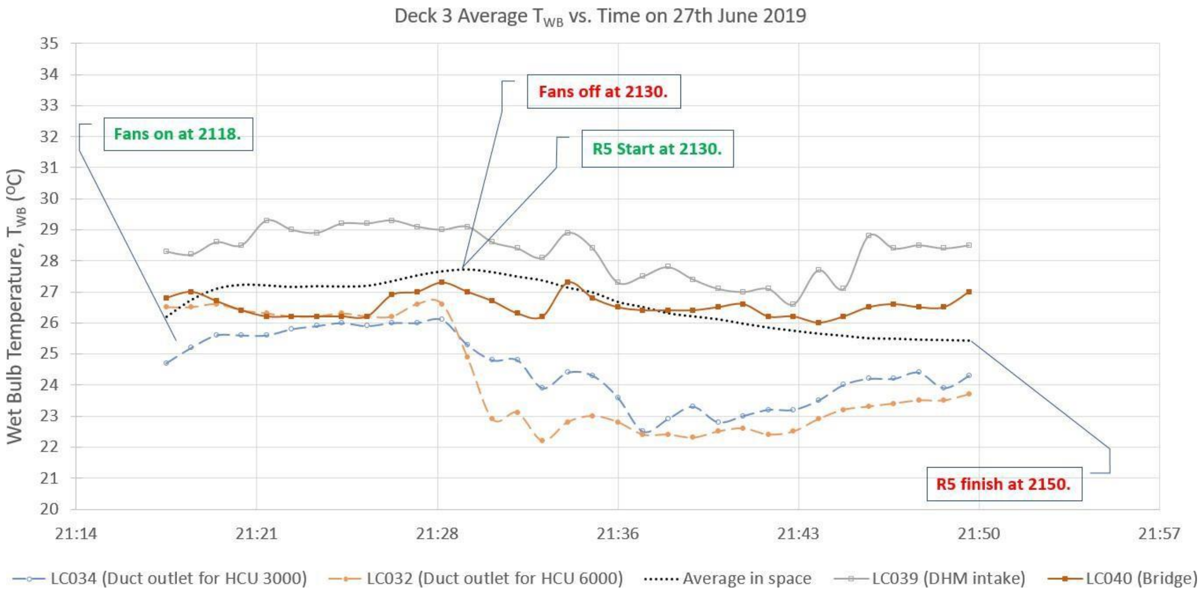


Figure 19: The Relative Humidity (RH) rate response for dehumidifier units delivering 15,000 m^3/hr of dehumidified air to deck 3 (top) and deck 2 (bottom)

- The RH for both the dehumidifier intake (LC039) and the Bridge (LC040) have a variation of no more than 5% for both decks.
- There is a reduction of average space RH of 9% and 12% for deck 3 and 2, respectively, with DHM applied.
- The Tinytag logger data showed consistency with the Kestrel data on deck 3.
- The deck 3 average space RH eventually settled between the RH of the two duct outlets.
- The average space RH on deck 2 converged nicely with the RH of one of the duct outlets.
- When stabilised (after 21:43), the RH of the two duct outlets in deck 2 are only 2% different.

Wet bulb temperature results



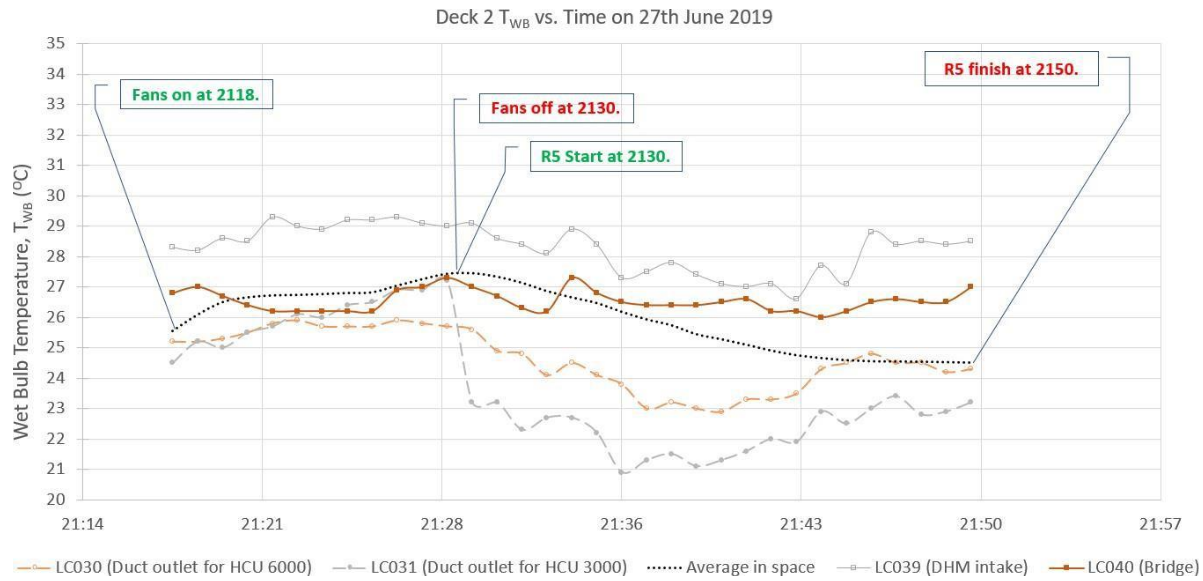


Figure 20: The Wet Bulb Temperature (TWB) rate response for dehumidifier units delivering 15,000 m^3/hr of dehumidified air to deck 3 (top) and deck 2 (bottom)

- After the DHM units are switched on, there is a 3.0^{O}C recorded reduction of T_{WB} on deck 2 within 20min.
- The space average T_{WB} on deck 2 converged with one of the duct outlets towards the end of the 20min period.
- The T_{WB} differences between duct outlets in deck 2 ranged from 1.0 to 1.5^{O}C.
- The deck 3 T_{WB} decreased 2.0^{O}C over the 20min period. Most of the duct outlets' T_{WB} also dropped accordingly.
- In the final phase of the trial, the space average T_{WB} in deck 3 showed a difference between
  1.0 to 2.5^{O}C with respect to the two duct outlets (LC034 and LC032).
- There will be no comparison between Kestrel data loggers (space average) and Tinytag data logger on deck 3 because the Tinytag data logger only records T_{DB}, RH, and T_{DP} (Dew Point Temperature).

Trial 3B - Cumulative dehumidification rate response

The objective of this trial was to establish a cumulative dehumidification rate response under sealed deck conditions to support modelling of the predictive impact on T_{WB}, RH and T_{DB}.

This trial contained similar treatments as outlined in Trial 3A, the key difference being that fans were not switched on between treatments so that each deck received an accumulated dehumidification load starting from 5,000m^3/hr, building to 10,000m^3/hr and then 15,000m^3/hr.

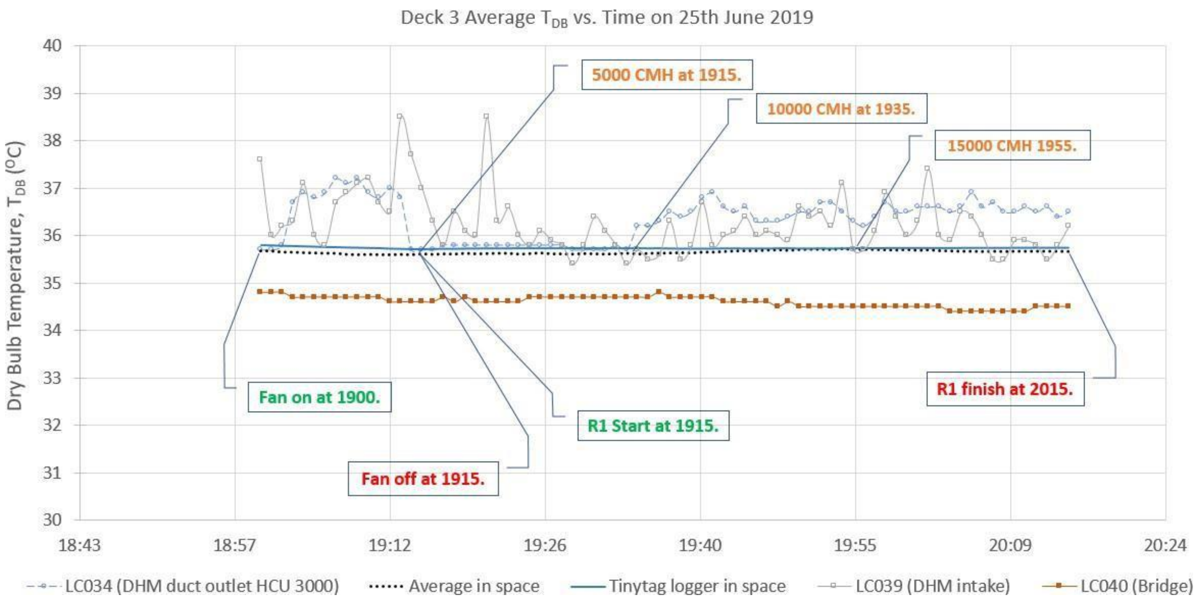
Instead of immediately switching on the full flowrate capacity (15,000 m^3/hr) of the DHM units at the initiation of each replicate (as in Trial 3A), the flowrate was increased by sequential 5,000 m^3/hr increments every 20 minutes.

To deliver these sequenced incremental increases, "Y" branches and capping were used to either enable dehumidified air from a unit to flow to the deck, or to divert that air onto the dockside area or deck 7 space (where the ductwork entered the vessel). It was advised by the technology provider that the DHM units may require 5 to 10 minutes to reach steady operating condition once turned on from a shutdown state. The diversion / capping arrangement was implemented to overcome this limitation.

At the commencement of the trial, one DHM unit (HCU3000) was used to supply 5,000 m^3/hr of dehumidified air to each deck for 20 minutes. The air from each HCU3000 unit was then diverted using the "Y" branch in the ductwork. The DHM units continued to run, but the dehumidified air was expelled into other spaces (dockside for the HCU3000 units or onto deck 7 for the HCU6000 units). As this occurred, the diversion cap on the "Y" branch in the ductwork for the DHM unit HCU6000 was removed, supplying 10,000 m^3/hr to the deck.

After another 20 minutes passed, the diversion cap for HCU3000 unit was removed so that both it and the HCU6000 were concurrently delivering a total of 15,000 m^3/hr or dehumidified air to the deck. This procedure was the same for the units used across both decks.

Dry bulb temperature results



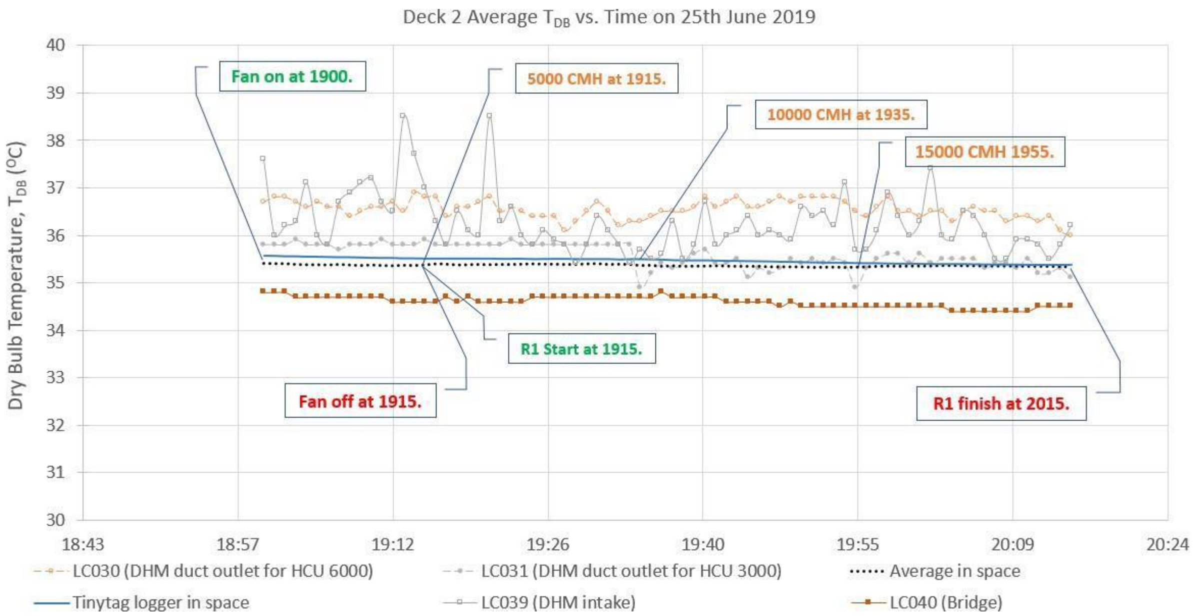
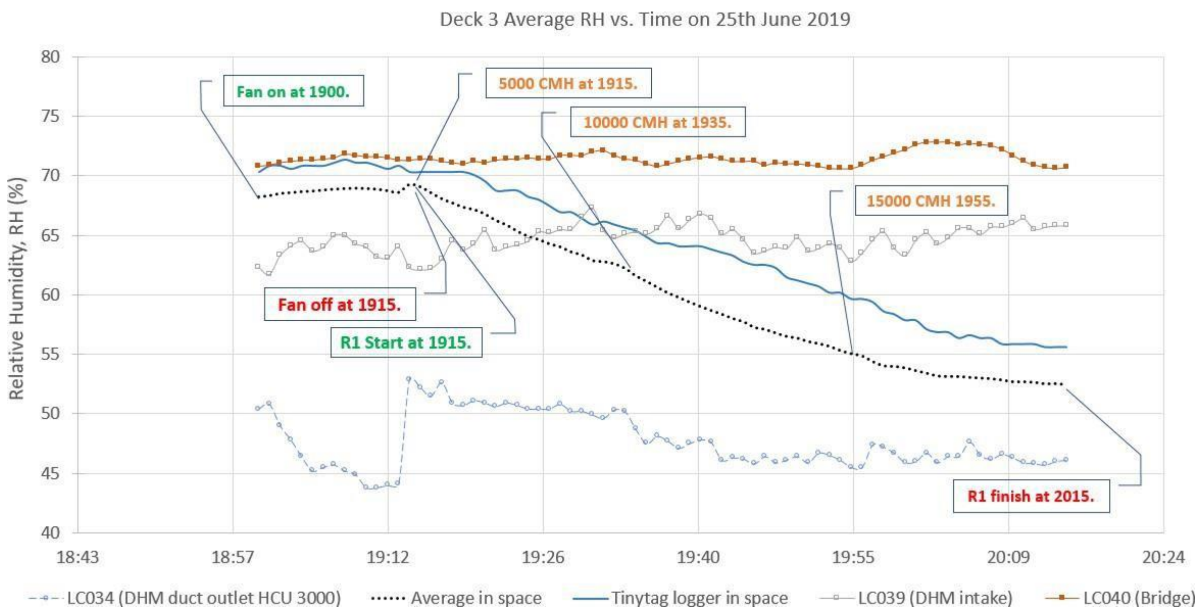


Figure 21: The Dry Bulb Temperature (T_{DB}) cumulative rate response for dehumidifier units delivering 15,000 m^3/hr of dehumidified air to deck 3 (top) and deck 2 (bottom)

- With increasing DHM flowrate, the T_{DB} for both decks had minimal changes throughout the replicate.

Relative humidity results



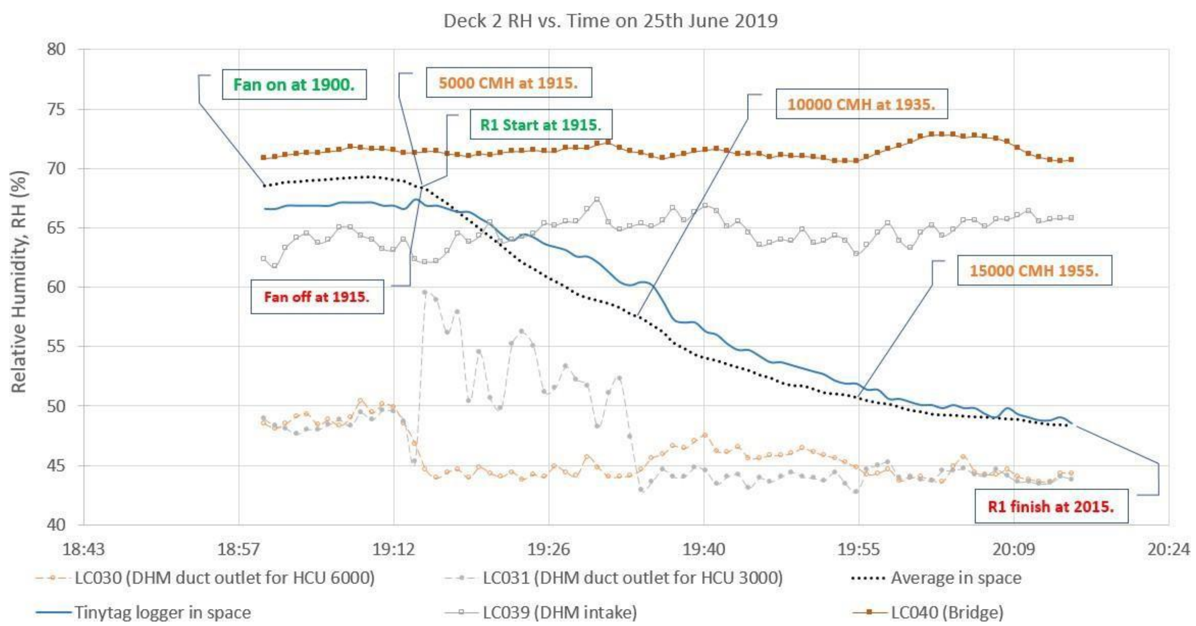
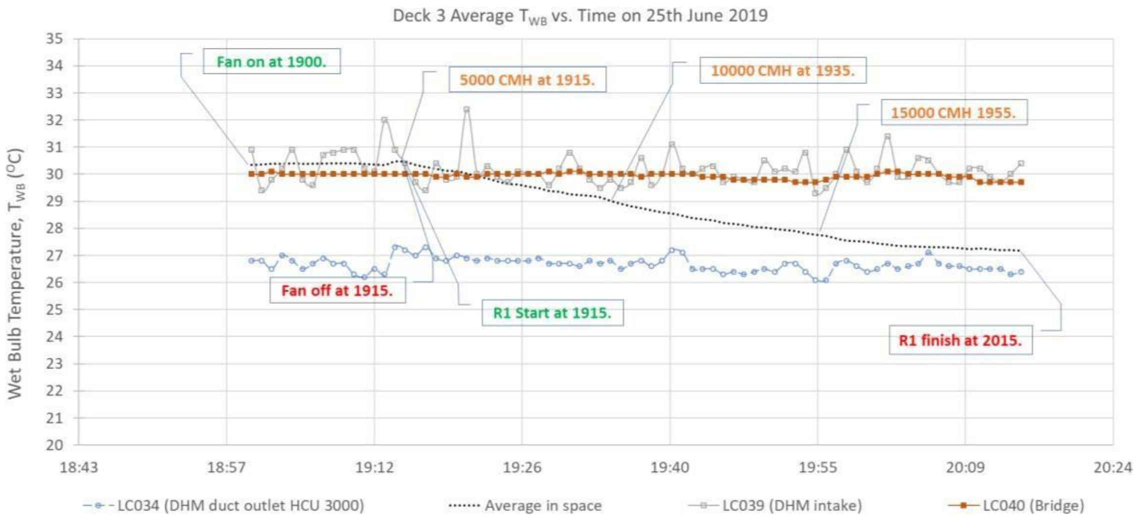


Figure 22: The Relative Humidity (RH) rate response for dehumidifier units delivering 15,000 m^3/hr of dehumidified air to deck 3 (top) and deck 2 (bottom)

- Throughout the replicate, deck 3 recorded a 17% decrease in RH and deck 2 recorded a 20% RH decrease.
- The space average RH recorded in both decks was consistent with readings from the TinyTag data loggers.
- The space average RH in deck 2 was 5% higher than the duct outlets at the end of the trial. For deck 3, the RH difference between space average and duct outlet was 7%.
- When the cumulative step-up to 15,000m^3/hr of dehumidified air being delivered into the decks occurred, the decrease in RH was not as much compared to the application of the lower flowrates (5,000m^3/hr and 10,000m^3/hr).

Wet bulb temperature results



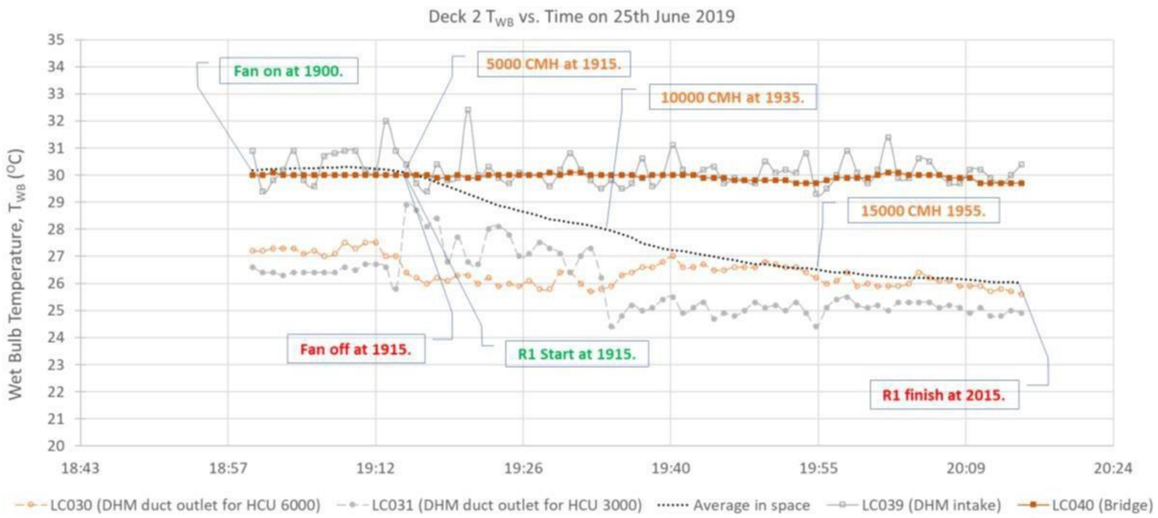


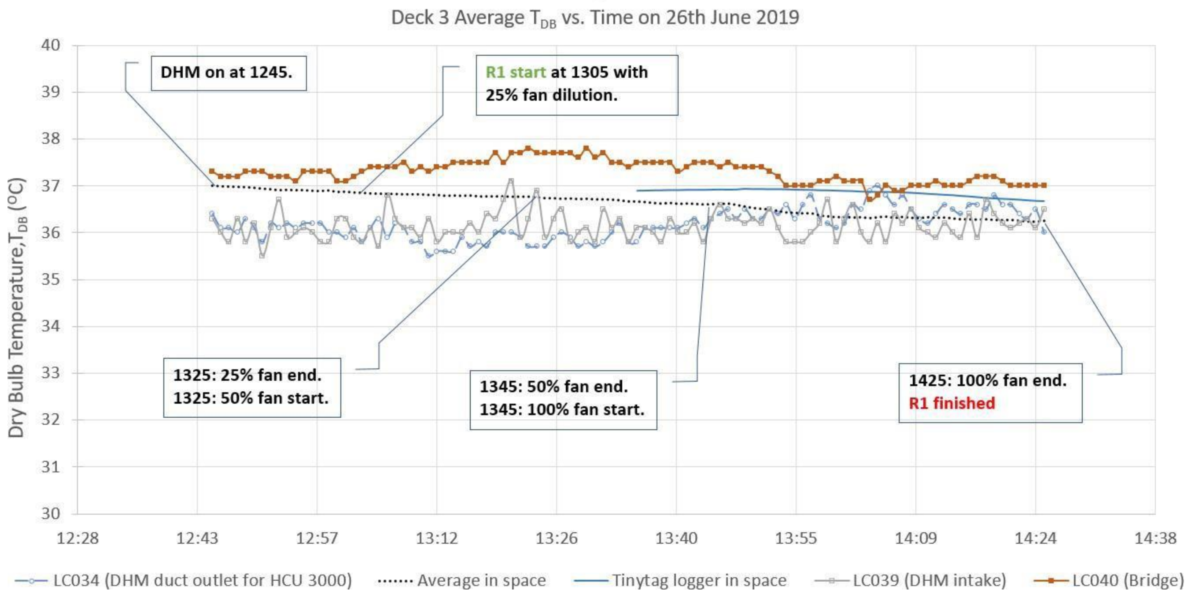
Figure 23: The Wet Bulb Temperature (T_{WB}) rate response for dehumidifier units delivering 15,000 m^3/hr of dehumidified air to deck 3 (top) and deck 2 (bottom)

- The decrease in the space average T_{WB} throughout the trial in deck 3 was 3.0^{O}C and 4.0^{O}C in deck 2.
- At the end of the trial, the space average T_{WB} was 1.0^{O}C higher than the duct outlet for HCU 3000 unit for both decks.
- In deck 2, the HCU6000 unit duct outlet T_{WB} was overlapping with the space average T_{WB} during the middle of the 10,000 m^3/hr session.
- As observed from the RH plot, when 15,000 m^3/hr of dehumidified air was supplied to the decks, the decrease in T_{WB} was not as great when the decks were supplied with 5,000 m^3/hr and 10,000 m^3/hr of dehumidified air.
- The DHM intake and Bridge recorded T_{WB} were consistent.

Trial 3C - Ambient baseline dehumidification dilution

The objective of this trial was to establish the dilution impact on dehumidified air of introducing sequenced ambient ventilation fans under sealed deck conditions. The DHM units were operated at 100% for this trial (ie. 15,000m^3/hr for each deck). Three treatments of fan capacity were applied (25%, 50% and 100%), with sequencing of specific supply and exhaust fans calculated prior to the trial.

Dry bulb temperature results



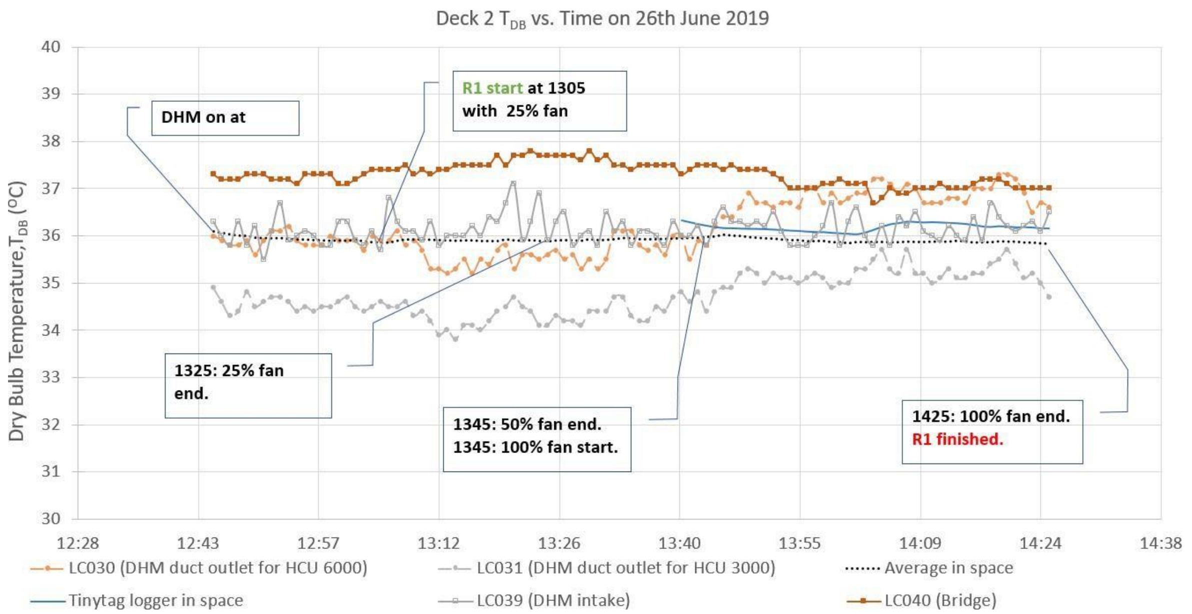
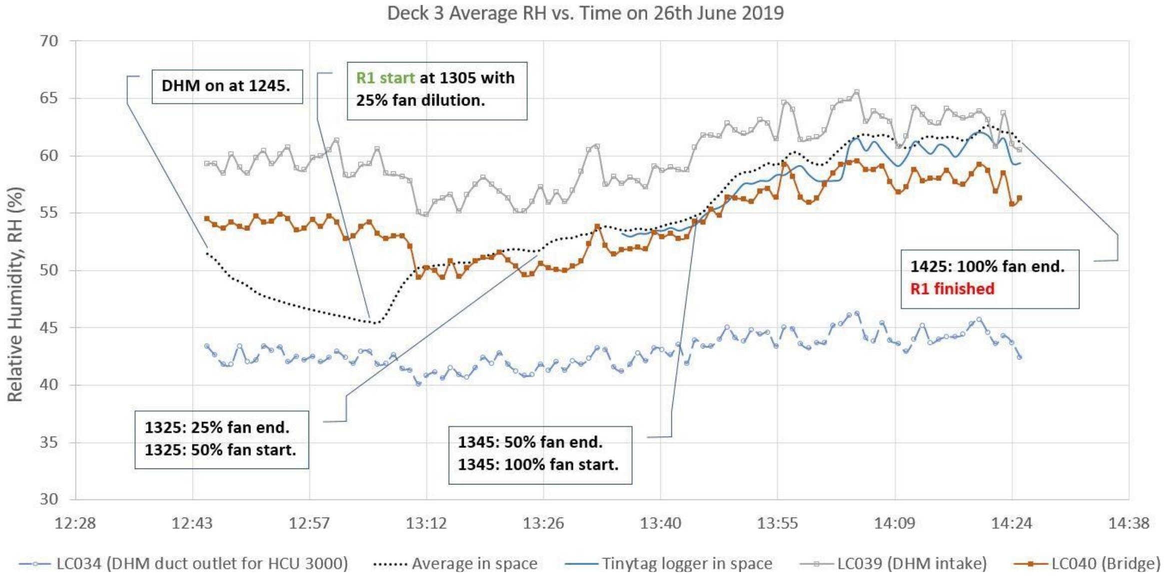
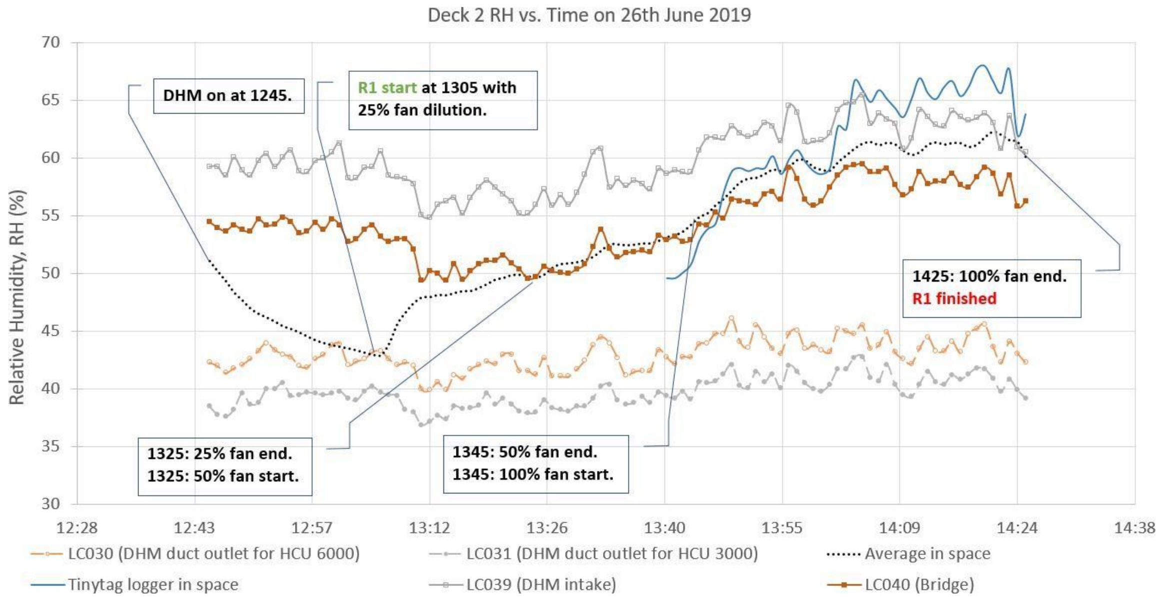


Figure 24: The Dry Bulb Temperature (T_{DB}) rate response for ambient baseline dehumidifier dilution in deck 3 (top) and deck 2 (bottom)

- The changes of T_{DB} in deck 3 were more than in deck 2, which was 0.7^{O}C compared with 0.3^{O}C.
- Differences between average space T_{DB} and Tinytag data logger T_{DB} varied from 0.5^{O}C to 0.7^{O}C in deck 3 and from 0.2^{O}C to 1.2^{O}C in deck 2.

Relative humidity results



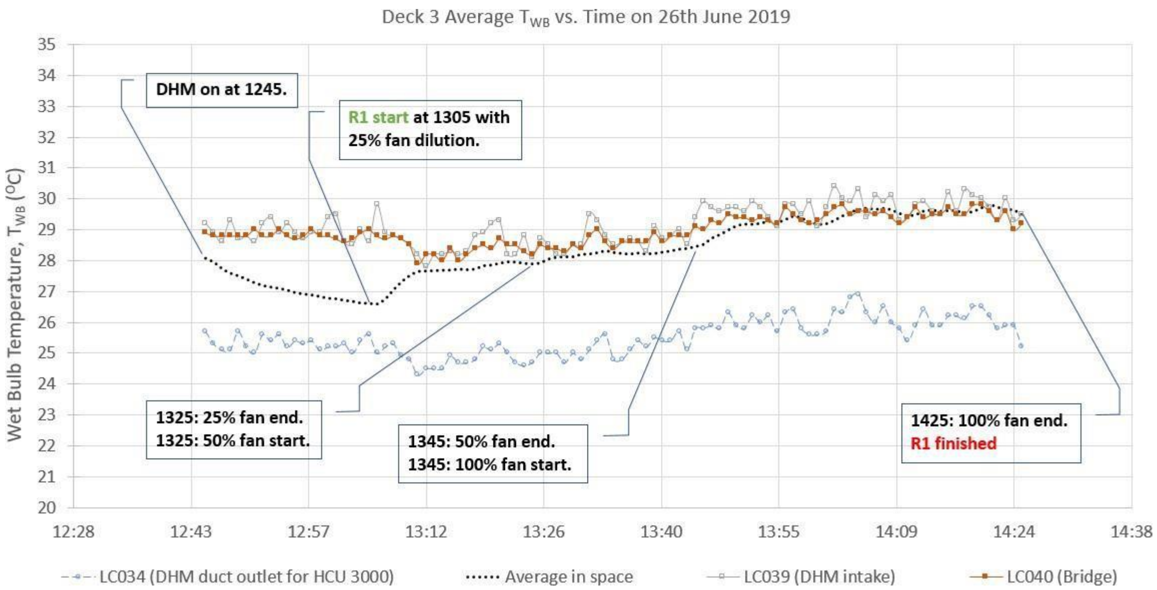


Figue 25: The Relative Humidity (RH) rate response for ambient baseline dehumidifier dilution in deck 3 (top) and deck 2 (bottom)

- The space average RH in both decks decreased once the dehumidifier units were switched on at 12:45. The dehumidifier units were supplying 15,000 m^3/hr (PAT_{DHM} = 18) of dehumidified air to each deck, with no fans operating. The RH in deck 2 decreased from 52% to 43%, while in deck 3 the RH decreased from 52% to 45%. The rates of the RH decrease for both decks were steep initially (12:45 – 12:50) and thereafter, declined less steeply (12:50 – 13:05).
- The space average RH in deck 2 increased (7%) when 25% (equivalent to 61,000 m^3/hr; $PAT_{V/A}$ = 72.3) of supply (31,000 m^3/hr; $PAT_{V/A}$ = 36.8) and exhaust (30,000 m^3/hr; $PAT_{V/A}$ = 35.6) fans was introduced to the space.
- The same RH increase (7%) trend appeared on deck 3 with the introduction of 25% (equivalent to 60,000 m^3/hr or $PAT_{V/A}$ = 71.2) of supply (32,000 m^3/hr; $PAT_{V/A}$ = 37.9) and exhaust (28,000 m^3/hr; $PAT_{V/A}$ = 33.2) fans.

- The RH increased rapidly on both decks once 25% fan capacity was introduced and then increased at a less steep rate. The time frame for the rapidly increased rate of RH was about 5min.
- It can be observed that with just 25% of fans introduced, the RH in the space is quickly increased to a level equivalent to that which occurred before the space was dehumidified.
- The introduction of 50% (equivalent to 108,000 m^3/hr; PAT$_{V/A}$ = 128.1) supply (62,000 m^3/hr; PAT$_{V/A}$ = 73.5) and exhaust (46,000 m^3/hr; PAT$_{V/A}$ = 54.6) fans further increased the space average RH in both decks by about 3% to 4%. When 100% fans (230,000 m^3/hr; PAT$_{V/A}$ = 272.8) were introduced, the RH level in both decks increased to more than 60%.
- The space average RH in both decks sits between the dehumidifier intake (LC039) and the Bridge (LC040). Before 25% of fans were introduced to the space in deck 2, the average space RH reduced to the same level as one of the duct outlets. As in deck 3, lowest point for the average space RH was still 2 to 3% different from the duct outlet.

Wet bulb temperature results



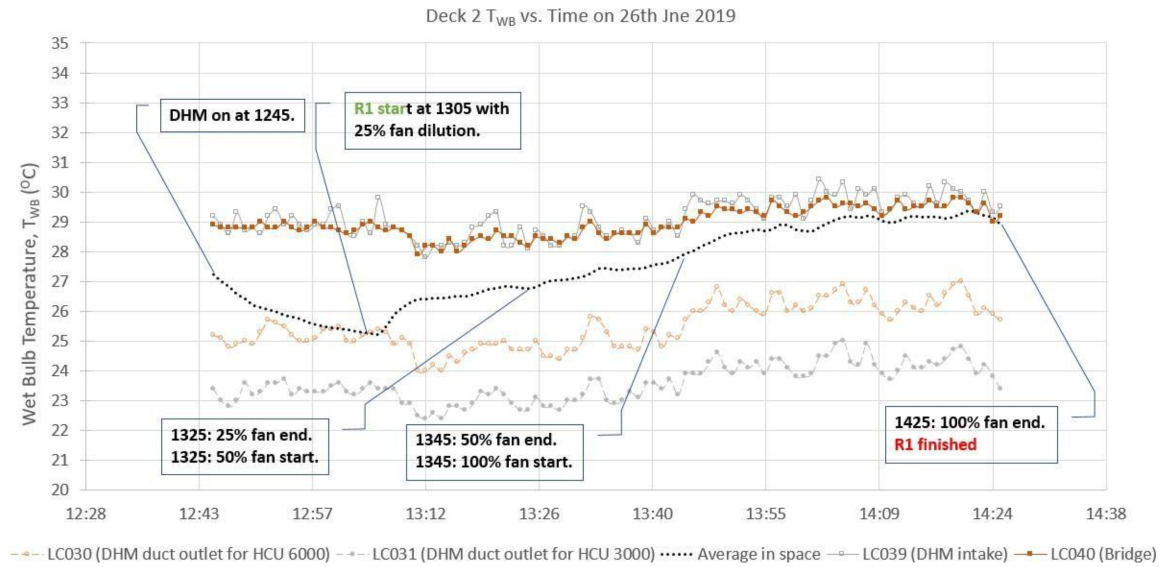


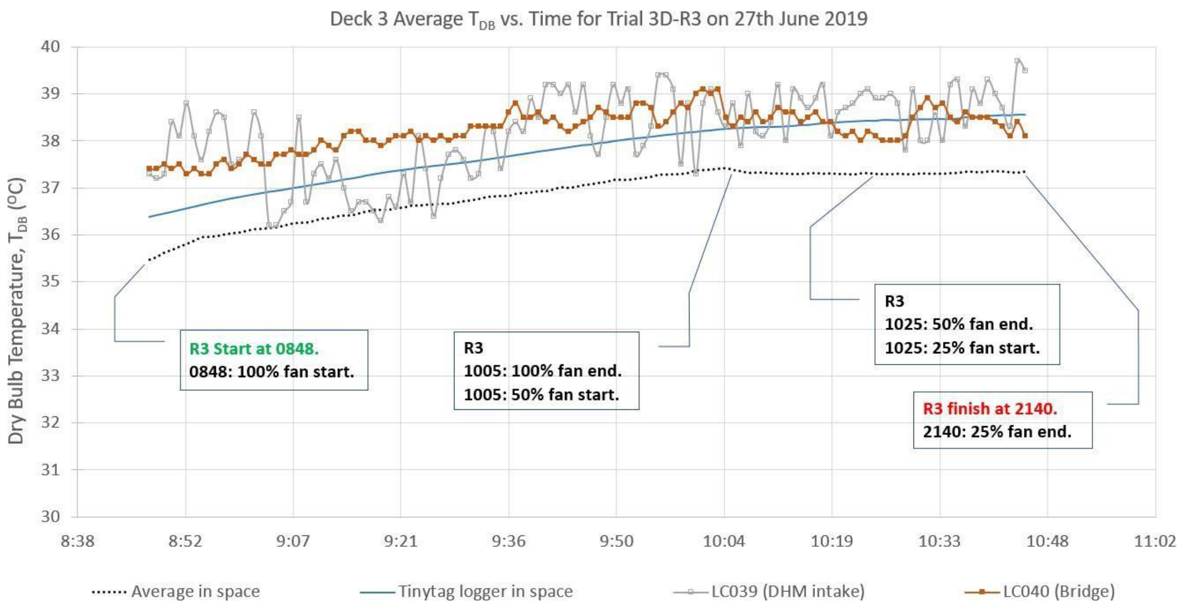
Figure 26: The T$_{WB}$ rate response for ambient baseline dehumidifier dilution in deck 3 (top) and deck 2 (bottom)

- The space average T_{WB} for both decks 2 and 3 possessed a similar trend as those shown in the previous RH plots. When the dehumidifier was initially switched on (before the addition of fans), the average space T_{WB} dropped 2.0^{O}C in deck 2 and 1.5^{O}C in deck 3.
- Once the supply and exhaust fans were introduced (61,000 m^3/hr at 25%; $PAT_{V/A}$ = 72.3), the space average T_{WB} for both decks quickly rose back to the level before the space was being dehumidified.
- The rapid decrease and increase in the space average T_{WB} when dehumidifiers were switched on, and the 25% fans introduced respectively echoed those in the RH plots.
- With more supply and exhaust fans switched on, the T_{WB} increased accordingly. The 100% fans rose the T_{WB} in both decks to be the same as those in the dehumidifier intake (LC039) and Bridge (LC040).

Trial 3D - Ambient Airflow Rate Response (no dehumidification)

The objective of this trial was to establish an ambient airflow rate response baseline under sealed deck conditions to support modelling of the predictive impact on T_{WB}, RH and T_{DB}. No dehumidification was used in the trial, and only the supply and exhaust fan sequences used in Trial 3C were applied in a staged approach (at levels of 100% and then decreasing to 50% and 25%).

Dry bulb temperature results



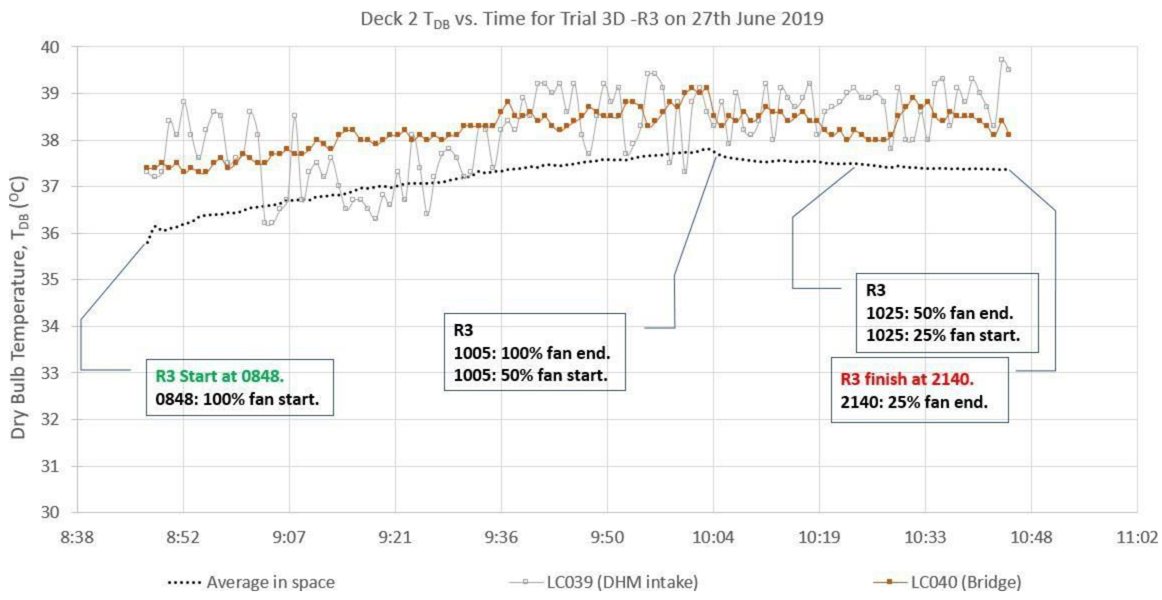
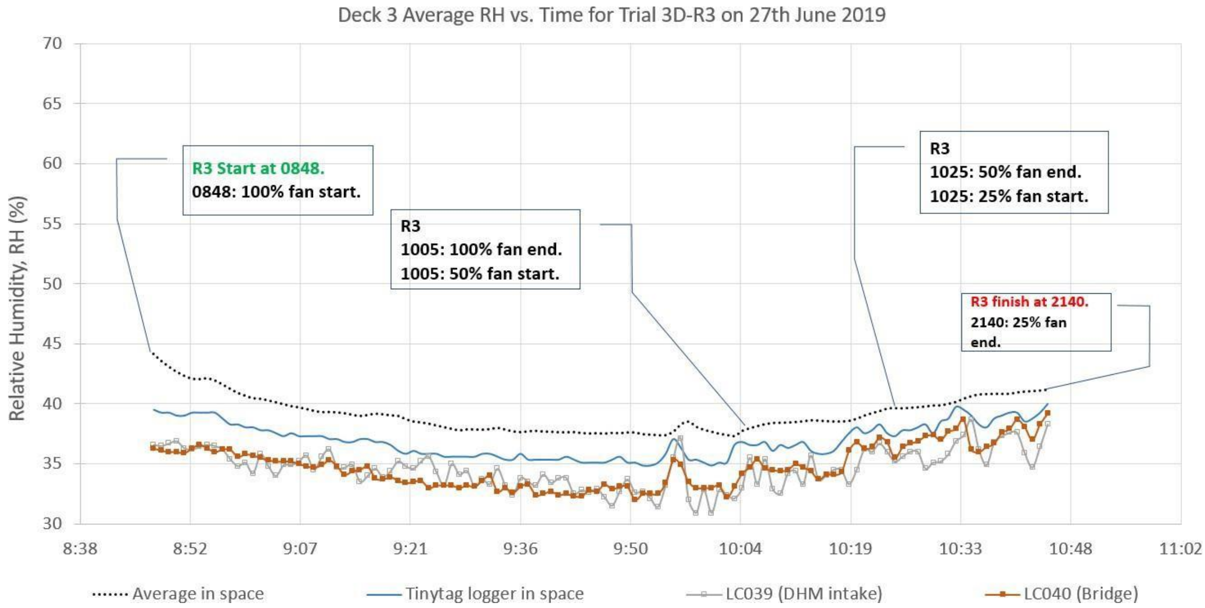


Figure 27: The Dry Bulb Temperature (T_{DB}) rate response for ambient airflow baseline in deck 3 (top) and deck 2 (bottom)

- No dehumidifier units were involved in this trial, and hence no relevant duct outlet temperatures are shown.
- The trial started with 100% supply and exhaust fans (total of 230,000 m^3/hr; $PAT_{V/A}$ = 272.8).
- The space average T_{DB} increased steadily in both decks. From 08:48 to 10:05, an increase of 2.0^{O}C was recorded for both decks.

- When the supply and exhaust fan flow rate was reduced to 50% at 10:05, the average space T_{DB} on both decks decreased very little (0.1^{O}C to 0.2^{O}C).
- The further reduction of supply and exhaust fans to 25% also observed very minimal changes in T_{DB} for both decks.
- The Tinytag data logger and the Kestrel data loggers had the same trend throughout the replicate, but levels differed between 0.9^{O}C to 1.3^{O}C. There is also consistency in the initial rising trend of ambient temperatures (dehumidifier intake and Bridge) with average space T_{DB}.

<u>Relative humidity results</u>



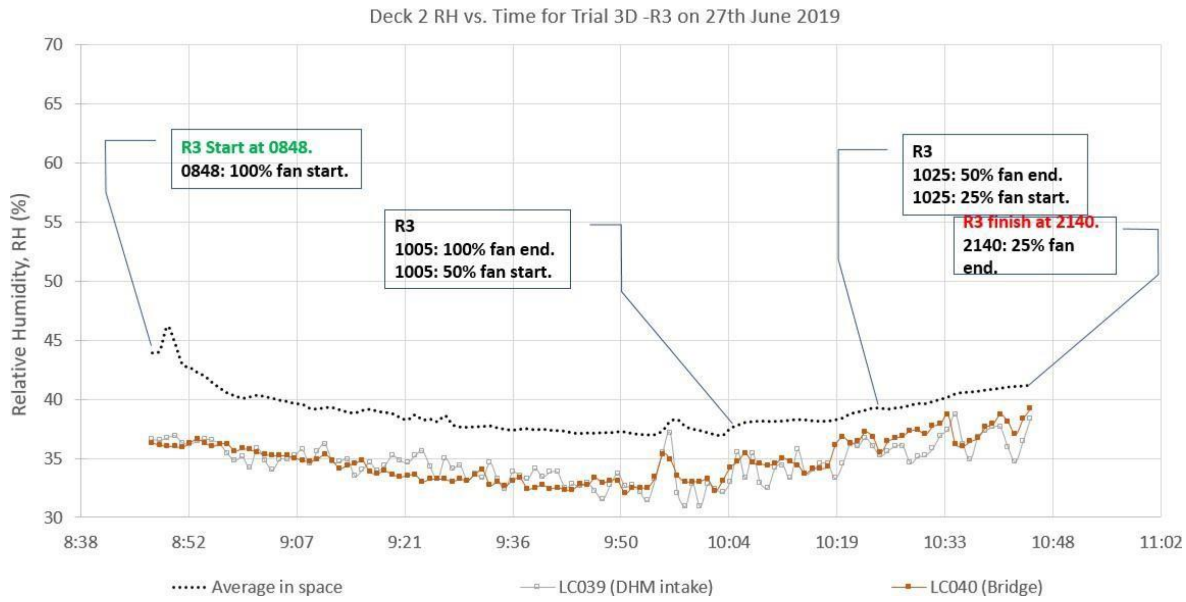
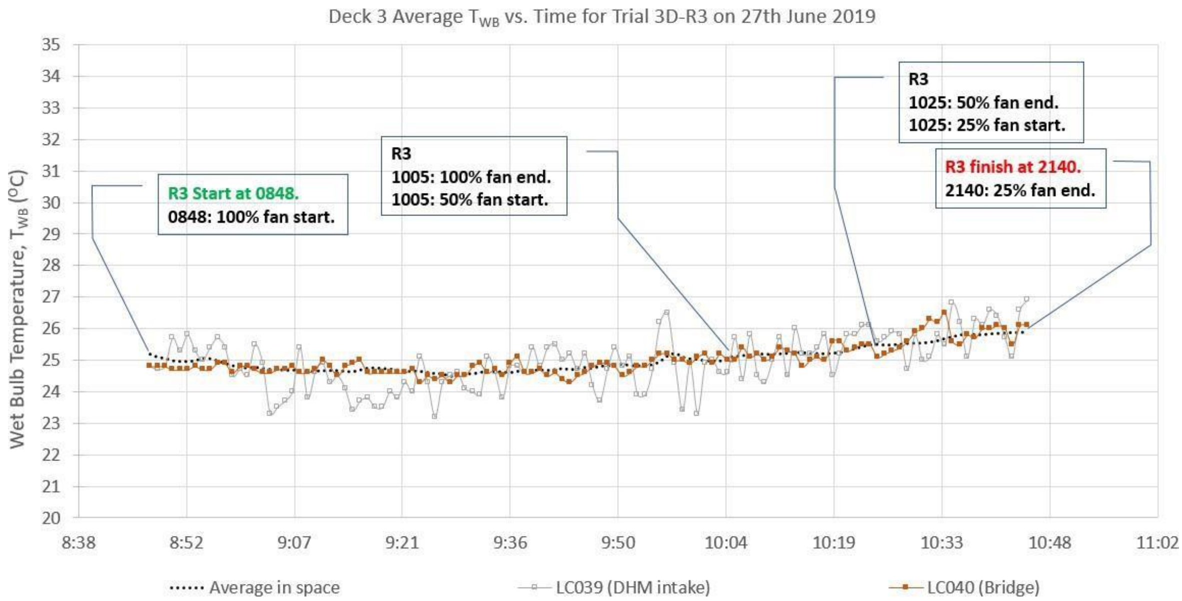


Figure 28: The Relative Humidity (RH) rate response for ambient airflow baseline in deck 3 (top) and deck 2 (bottom)

- The RH in both decks decreased with the introduction of 100% fans.
- The RH on both decks dropped with deck 3 recording 7%, and deck 2 recording 9% decreases.

- When the flow rate capacity of the fans was halved (50%), both decks recorded a moderate rise of 5% RH. When the flow rate capacity was then halved again (25%), both decks recorded a small rise of 2% RH.

Wet bulb temperature results



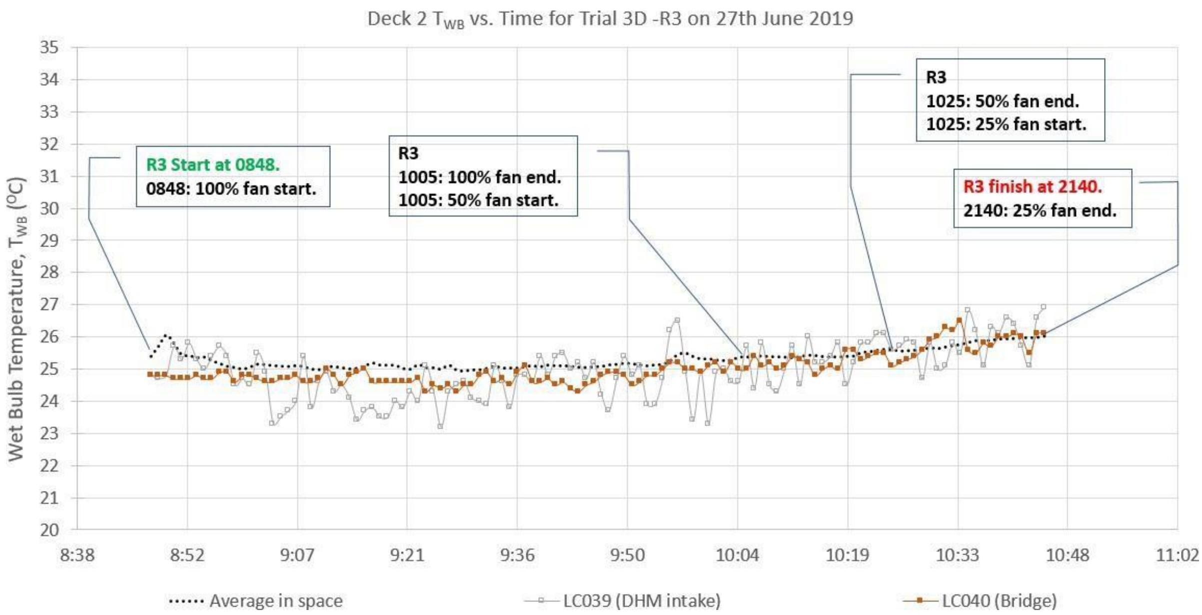


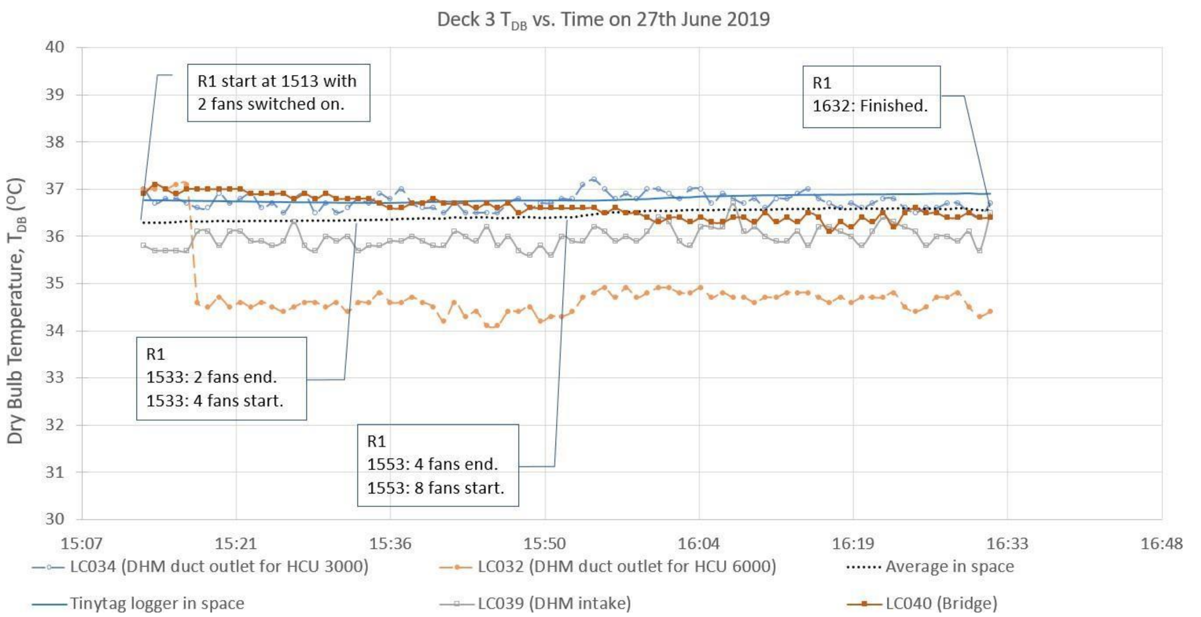
Figure 29: The Wet Bulb Temperature (T_{WB}) rate response for ambient airflow baseline in deck 3 (top) and deck 2 (bottom)

- The T_{WB} for the average space in both decks, dehumidifier intake and Bridge correlated in high degree throughout the replicate.

Trial 3E - Cumulative exhaust fan rate response

The objective of this trial was to establish the effect of cumulative exhaust arrangements on dehumidification under sealed deck conditions. Three exhaust fan sequences (i.e. treatments) were selected to increase air velocity across the deck, including through the reversing of the polarity of some supply fans. At the start of the trial, dehumidified air was introduced to decks 2F and 3F (100%, 15,000m^3/hr; PAT_{DHM} = 17.8) and conditions allowed to stabilise. The application of the fan sequences (2, 4 and 8 fans) then occurred immediately between treatments without altering DHM airflow.

Dry bulb temperature results



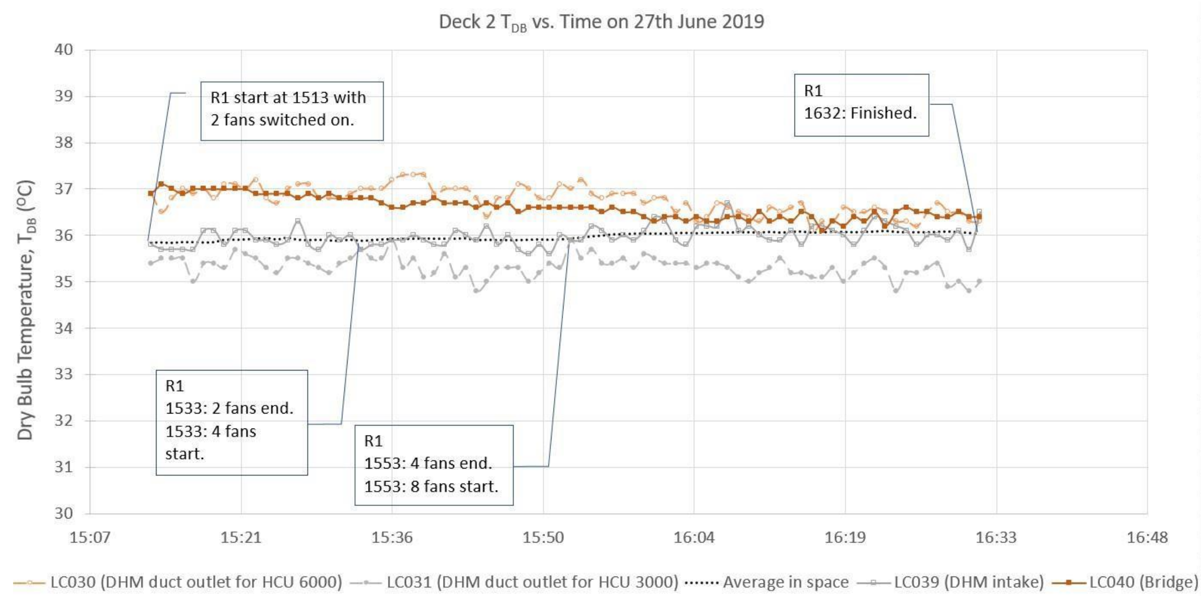
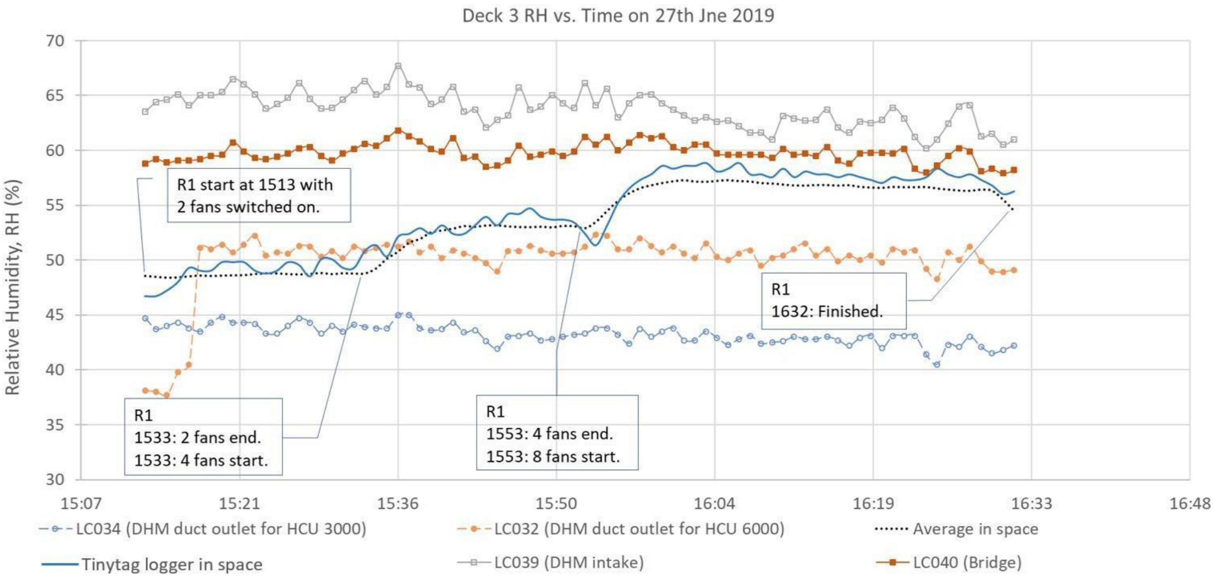


Figure 30: The Dry Bulb Temperature (T_{DB}) rate response for ambient baseline dehumidifier dilution in deck 3 (top) and deck 2 (bottom)

- The application of increasing exhaust fans had minimal effect on the T_{DB} throughout the trial.

Relative humidity results



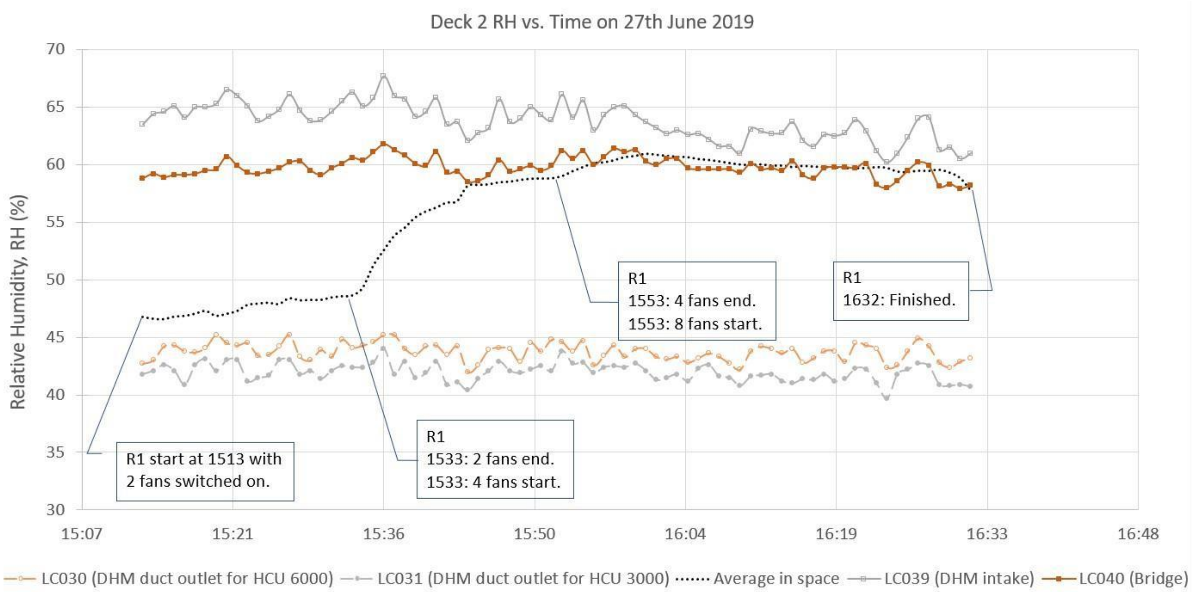
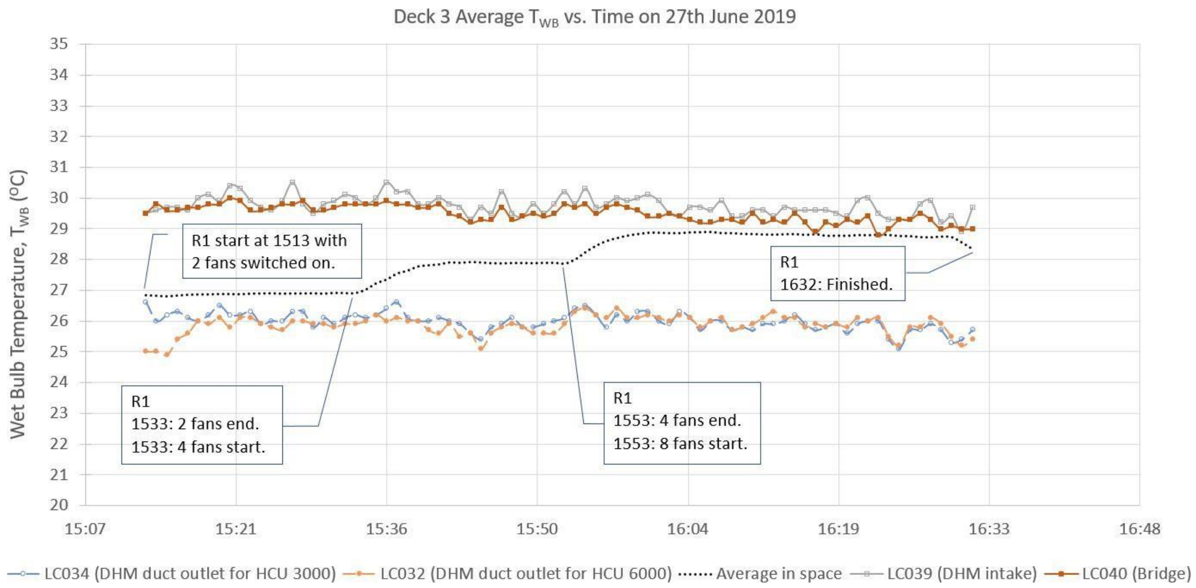


Figure 31: The Relative Humidity (RH) rate response for ambient baseline dehumidifier dilution in deck 3 (top) and deck 2 (bottom)

- The DHM units were switched on initially to dehumidify the decks and remained on for the trial. A step shape curve was then observed in deck 3, with the increasing number of exhaust fans switched on in sequence. Each step change increased the space average RH by 5%.
- When all eight exhaust fans were switched on, the space average RH in deck 3 was very close to the Bridge RH. The space average RH sat between both DHM duct outlets readings.
- In deck 2, the step change was not as prominent compared to deck 3. As all eight exhaust fans were switched on, the space average RH in deck 2 sat between the Bridge and DHM intake.
- The space average RH in both decks was higher than the two DHM duct outlets.

Wet bulb temperature results



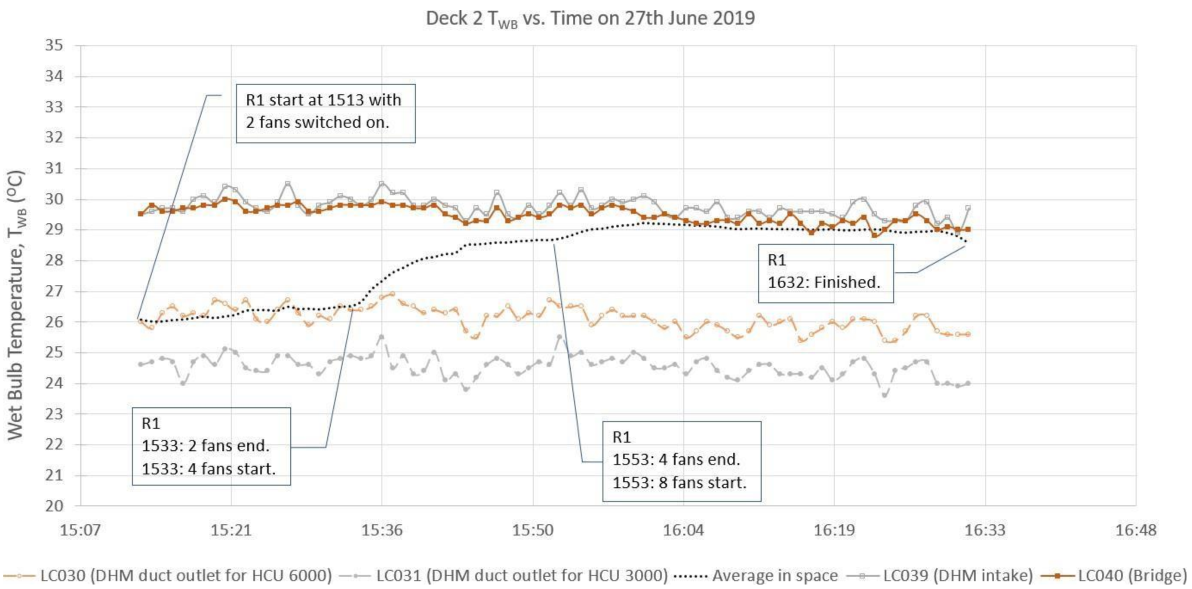


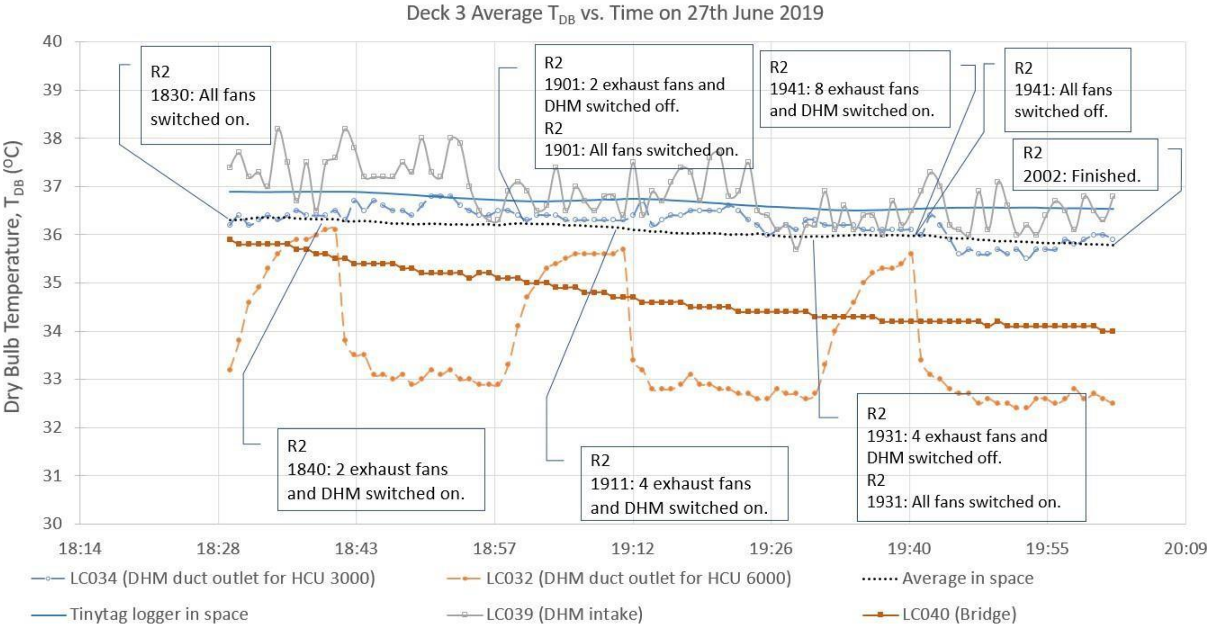
Figure 32: The Wet Bulb Temperature (T_{WB}) rate response for ambient baseline dehumidifier dilution in deck 3 (top) and deck 2 (bottom)

- The space average T_{WB} in deck 3 followed the same trend of step curve as those in the RH plots.
- Each step change in exhaust fans increased the T_{WB} by 1.0^{O}C on deck 3.
- The deck 2 T_{WB} did not have a distinct step change curve as in deck 3.
- The space average T_{WB} in deck 2 demonstrated the same trend as in the RHplot.

Trial 3F - Ambient sequential exhaust rate response

The objective of this trial was to establish the effect of sequential exhaust arrangements on dehumidification conditions under sealed deck conditions. Three exhaust fan sequences (i.e. treatments) were selected to increase air velocity across the deck, including through the reversing of the polarity of some supply fans. Fans were turned off at the start of each treatment in the trial. The application of the fan sequences (2, 4 and 8 fans) in conjunction with 100% dehumidified air (15,000m^3/hr) then occurred as separate treatments.

Dry bulb temperature results



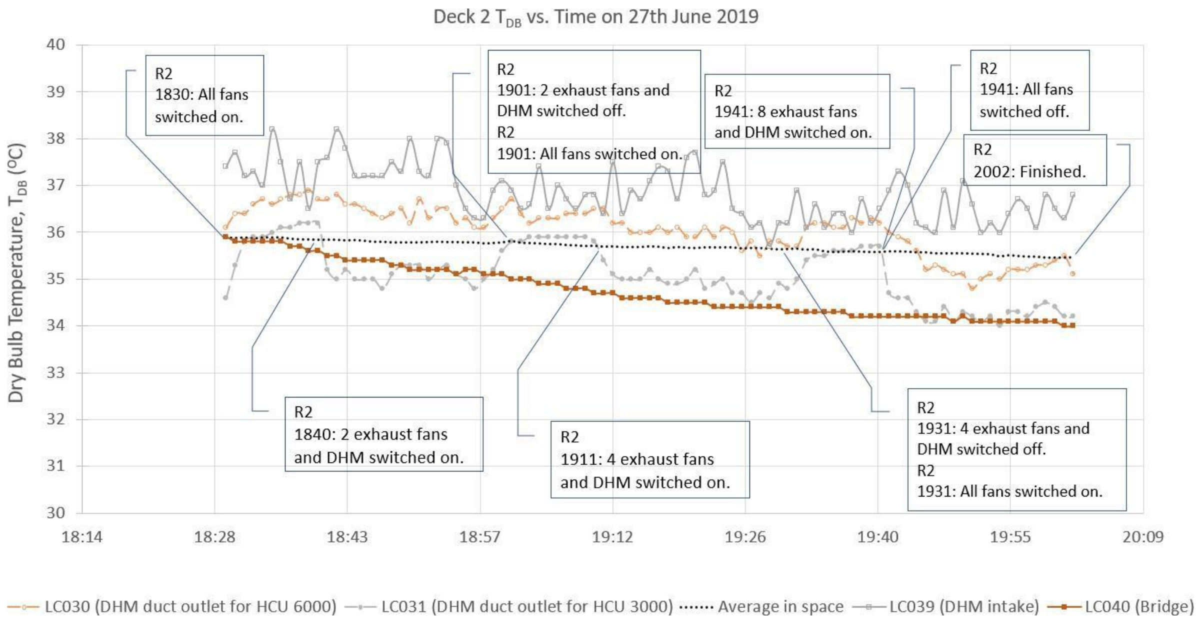
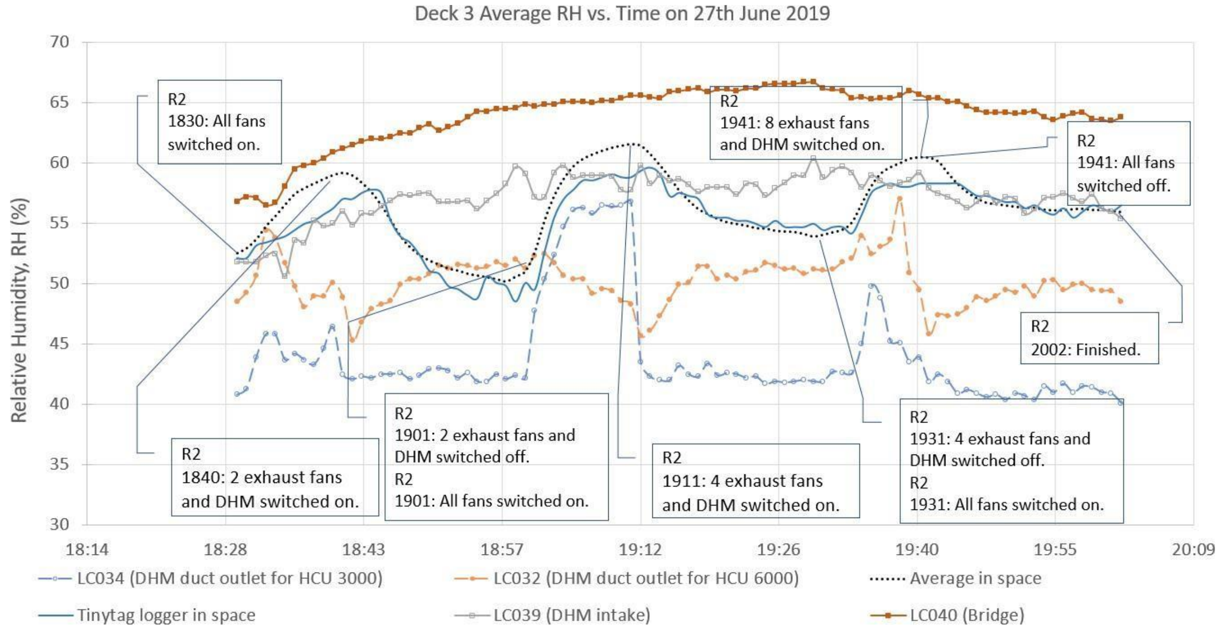


Figure 33: The Dry Bulb Temperature (T_{DB}) rate response for ambient airflow baseline in deck 3 (top) and deck 2 (bottom)

- The T_{DB} in both decks had minimal change throughout the trial.

Relative humidity results



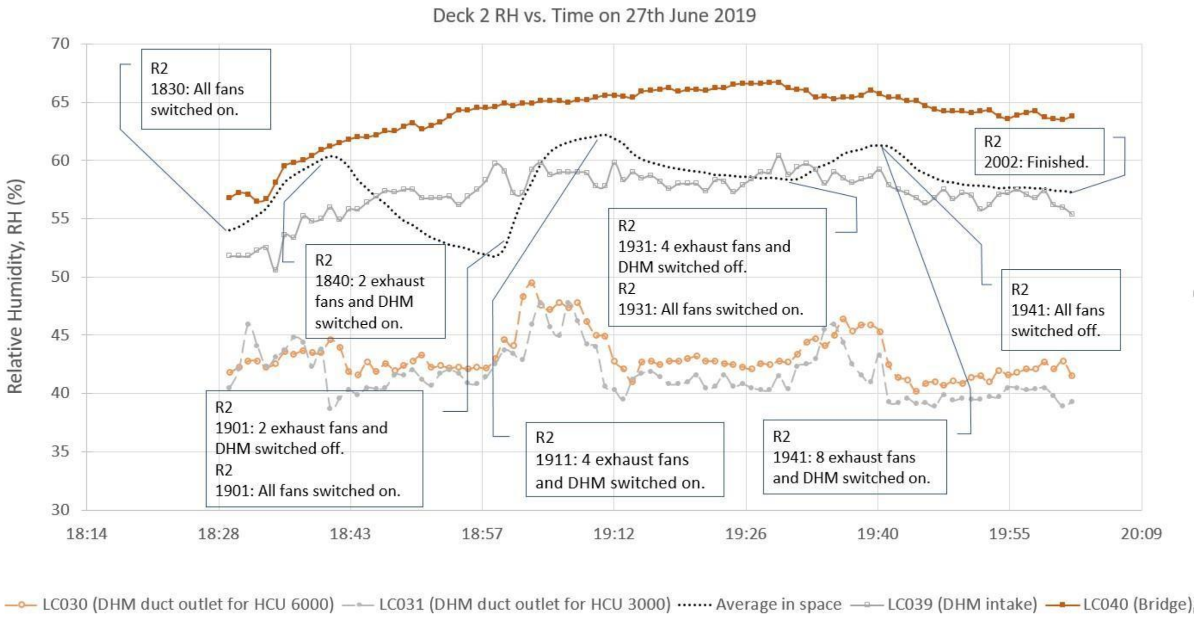


Figure 34: The Relative Humidity (RH) rate response for ambient airflow baseline in deck 3 (top) and deck 2 (bottom)

- At the commencement of the trial, all supply and exhaust fans were switched on to equate the deck conditions with the outside conditions.
- Once outside ambient conditions were reached in both decks, all supply fans and exhaust fans were switched off (at 18:40). At 18:40, two selected exhaust fans were switched on, the DHM was introduced and the effects were monitored. The space average RH in both decks reduced by about 9%.
- All supply and exhaust fans were switched on at 19:01. Once the conditions in the deck returned to the baseline ambient conditions, four selected exhaust fans were switched on and the DHM introduced. It was observed that the space average RH in deck 3 dropped 7%.

- The process was repeated with eight selected exhaust fans and the DHM turned on. A 5% RH reduction was recorded in deck 3.
- In deck 2, the four exhaust fans and DHM treatment recorded only a 4% decrease in space average RH. This drop in RH was the same as in the eight exhaust fans average space RH.
- For deck 3, the space average RH during the eight exhaust fans treatment reached the same level as the outside air condition (DHM intake). As for deck 2, space average RH reaches the outside air RH during the four and eight exhaust fans treatments.

Wet bulb temperature results

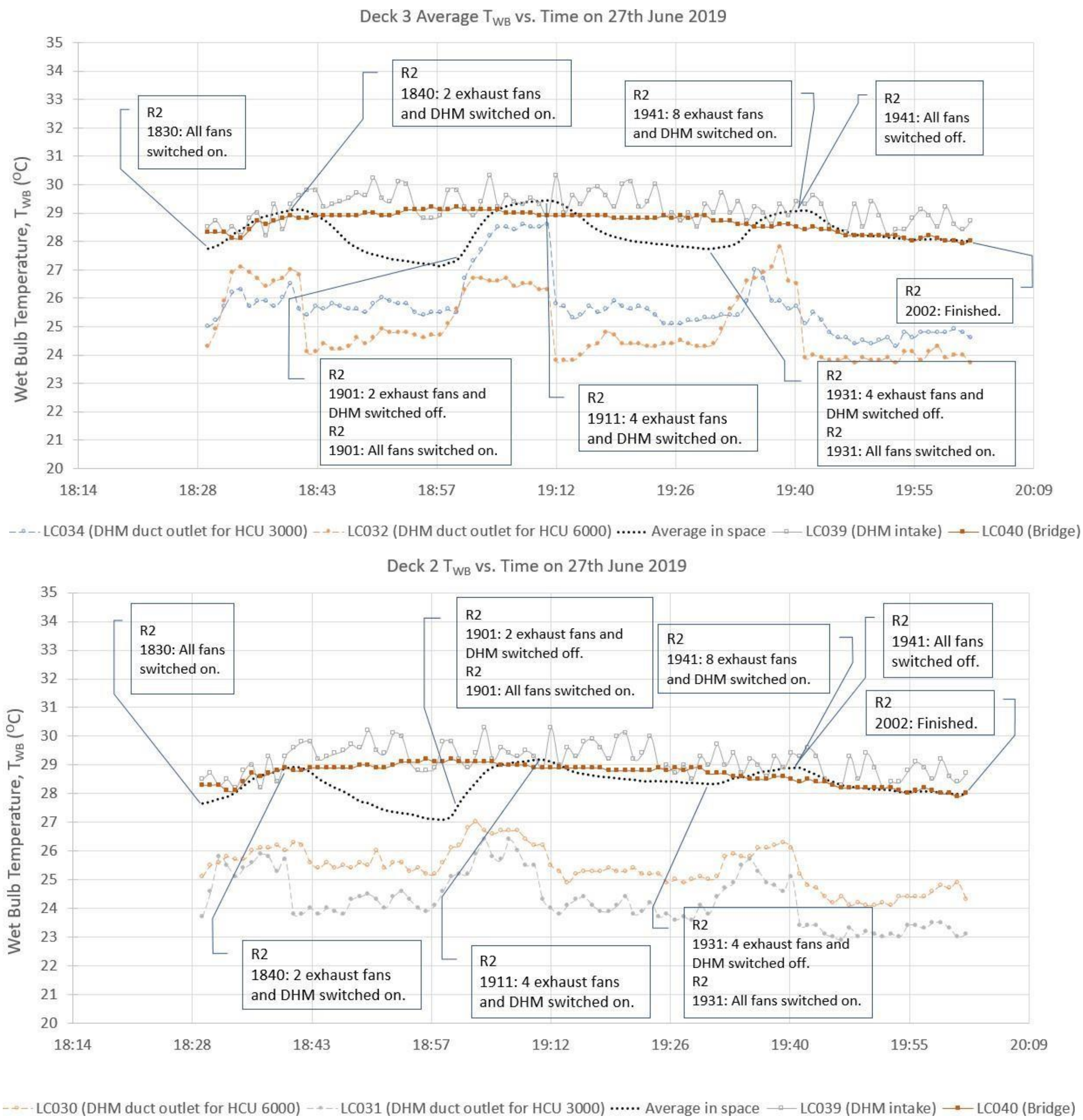


Figure 35: The Wet Bulb Temperature (T_{WB}) rate response for ambient airflow baseline in deck 3 (top) and deck 2 (bottom)

- The space average T_{WB} in deck 3 reduced with a distinctive pattern, with two exhaust fans reducing the average space T_{WB} by 2.0^{O}C, four exhaust fans reducing the average space T_{WB} by 1.5^{O}C and eight exhaust fans reducing the average space T_{WB} by 1.0^{O}C. Similar with the RH plot, the average space T_{WB} during the eight exhaust fan treatment reached the same outside vessel conditions.
- The deck 2 space average T_{WB} reduced 2.0^{O}C during the two exhaust fan treatment and the subsequent treatments (four and eight exhaust fans) only reduced it by 1.0^{O}C. The four and eight exhaust fans treatment recorded the average space T_{WB} close to and same with outside vessel conditions.

Performance of dehumidification units

The dehumidifiers used in this trial were optimised for moisture removal, as opposed to dry bulb temperature reduction. Irrespective of the dehumidifier performance in terms of total cooling load, measurements across all dehumidifiers indicated a larger latent heat removal fraction, with a small sensible cooling rate fraction. The larger fraction of the total cooling load from the dehumidifiers being latent heat removal confirms that moisture removal was made a priority for all dehumidifiers in all cases.

Within each deck, all trials that involved dehumidifiers demonstrated a reduction in RH and T_{WB}, whereas the T_{DB} in all trials did not change greatly.

The dehumidification trial project showed a successful proof of concept for dehumidifying the decks of a livestock export vessel (without livestock) with the RH being reduced by an average of 15% and the T_{WB} being reduced around 4.0^{O}C. The deck space average properties for T_{DB} and RH obtained with the Kestrel data loggers closely correlated with the available data from the Tinytag data loggers during the trial.

Deck 2F – HCU 3000 dehumidification unit

The results from the deck 2F HCU3000 dehumidifier unit and duct work heat loads were used as the primary reference case for all the detailed analysis and modelling where animals are considered. The performance of the deck 2F HCU3000 dehumidifier unit is tabulated in Table 2, with the rated flow rate being 5,000m^3/hr. The fluctuation (-8kW to 1kW) of the duct work heat loads (sensible and latent) for this unit is acceptable and can be attributed to heat transfer, air infiltration (leakage) and measurement uncertainty. As a consistency check, for all cases, there are no large (>1 kW) heat loads indicating heat loss (sensible or latent) from the duct. Such large positive heat loads would not be considered realistic considering the ambient air is always at a dry bulb temperature higher than the dehumidified air flowing through the duct work.

The deck 2F HCU 3000 dehumidifier unit total heat load varies from 20 to 41 kW, with the variation due to different ambient RH and T_{DB} conditions. It was noted that the dehumidifiers' performance increased when higher wet bulbs were experienced, with moisture removal easier at higher humidity

levels. This pattern was common to all the dehumidifier units used during the trials. As such, an expected trend in the performance of the dehumidifiers is that it will increase with increasing wet bulb temperatures.

Deck 2F – HCU 6000 dehumidification unit

For the deck 2 HCU6000 dehumidifier, the rated flowrate is 10,000m^3/hr and the total heat load ranged from 63 to 94 kW, as shown in Table 2. The duct work heat load was calculated to be very high. This is attributed to the 'cap off' procedure used during the trial, where dehumidified air was diverted into open deck space in deck 7. The cap off procedure (Figure 8) was applied to this HCU6000 unit when the trial needed only 5000m^3/hr of treated air to be supplied the deck. This cap-off procedure allowed the unit to be continuously run, without power interruption and the need to reheat the desiccant wheel. This cap-off procedure led to an air short circuit, whereby there was a small volume of air that was still flowing down the ducting to the decks when the cap was off.

In addition, because of the diversion, the duct work for the unit will not reach thermal equilibrium within the 5 to 7 minutes available when the trial begins. Due to the duct work heat load being significant, irrespective of the source, the deck 2F HCU6000 dehumidifier cannot be used as an ideal example for subsequent further analysis. The effect of ductwork non-equilibrium is tabulated in Table 3.

<u>Deck 3F – HCU 3000 dehumidification unit</u>

For the deck 3F HCU3000 dehumidification unit, the total heat load ranged from 28kW to 41kW; similar to the deck 2F HCU3000 unit. The deck 3F HCU3000 dehumidifier unit removed almost the same amount of moisture compared to the deck 2F HCU3000 unit, as shown in Figure 37. The T_{DB} recorded in the DHM output was lower than the deck 2F unit. The time history data taken over the period of the trial indicates that the dehumidification unit may not have reached a steady performance state within 5 to 7 minutes into its operation, potentially causing a reduced moisture removal rate from the space during the trial. Due to the unavailability of a data logger in the duct outlet for this unit, no duct work heat load can be calculated.

<u>Deck 3F – HCU 6000 dehumidification unit</u>

For the deck 3F HCU6000 dehumidification unit, the total heat load ranged from 35 to 62kW and the dehumidifier RH reduction ranged from 15% to 22%. In comparison with the deck 2F equivalent unit, the deck 3F unit underperformed in terms of total heat load (63kW - 94kW compared to 35kW -62kW). The deck 3F HCU6000 dehumidification unit's latent heat load is also lower when directly compared to the deck 2F HCU6000 unit. This is supported by the results outlined in Figures 36 and 37.

The deck 3F HC6000 dehumidification unit had its refrigerant recharged in the afternoon of 25 June 2019, which improved the performance. However, it still did not perform as well as the same unit on deck 2F. The ductwork total heat loads were better than that of the same unit on deck 2F. There is only one very high ductwork heat load (case 12.77kW), which is attributed to the data logger at the duct outlet initially being placed further downstream from the outlet than for other duct outlets. The logger was placed back at the direct duct outlet when the shortage of data loggers was solved.

Further information concerning the dehumidification unit performance

Figures 36 and 37 represent the plot of Total Heat Load (Q_T) and Latent Heat Load (Q_L) for each of the dehumidification units during trial period. For the smaller DHM units (HCU3000), the Q_T and Q_L are quite similar on both decks and most of the time the plots gather in the same region. However, for the larger units (HCU6000) the deck 2F unit had a higher Q_T and Q_L. The differences of Q_T and Q_L between deck 2F and 3F ranged from 29% to 50% and 19% to 50% respectively. This is a major difference and hence the HCU6000 unit on deck 3F is considered to have underperformed.

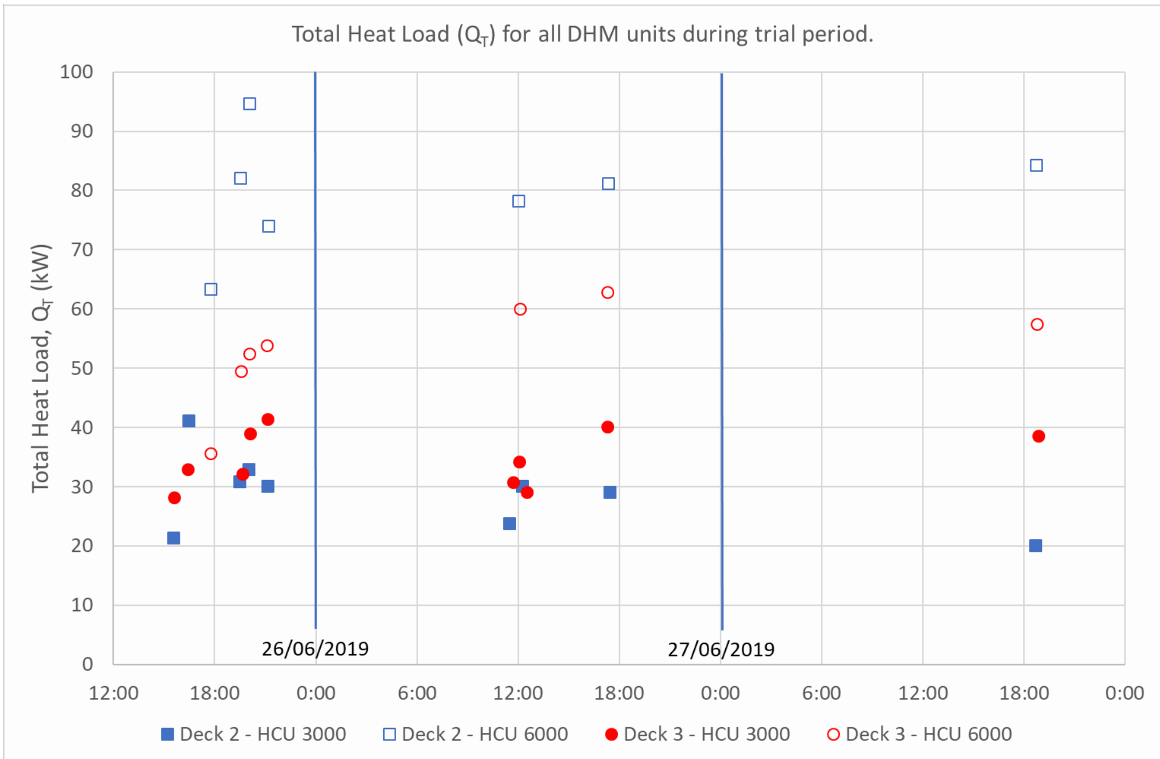


Figure 36: Total Heat Load from DHM units for decks 2F and 3F during the trial period

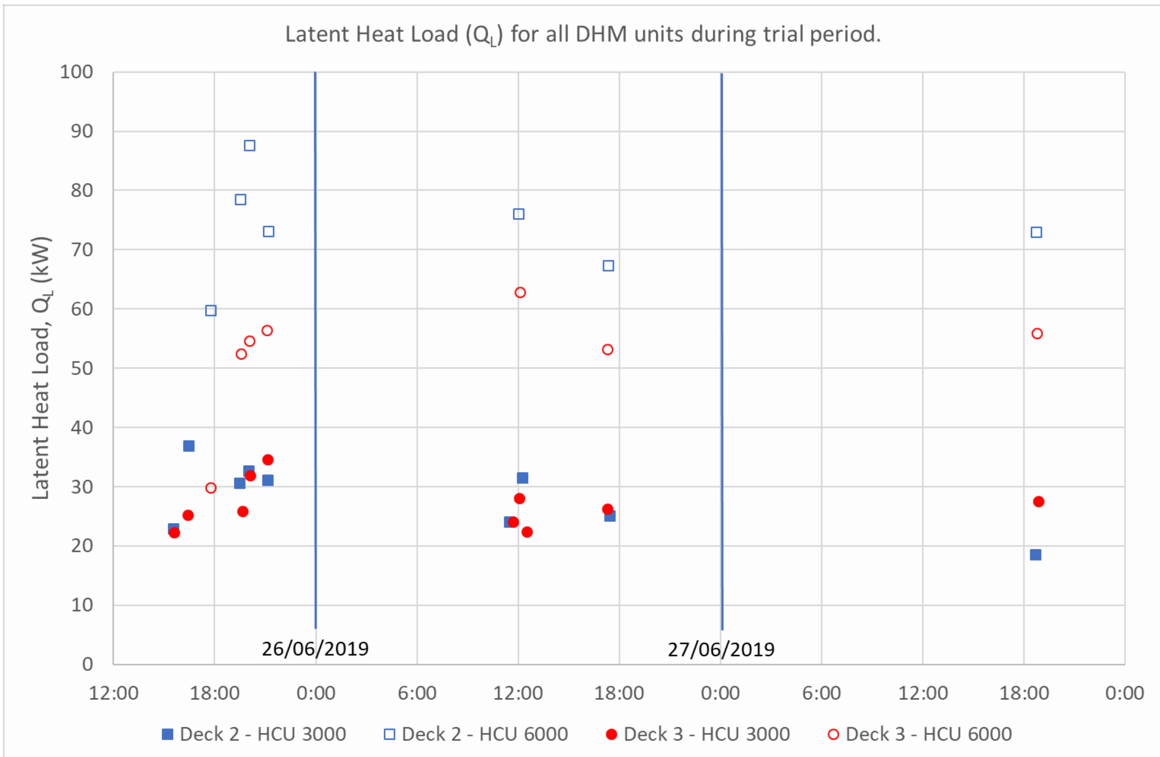


Figure 37: Latent Heat Load from DHM units for decks 2F and 3F during the trial period

Table 3: The analysis of change in Q_S and Q_L is based on 10,000 m^3/hr flowrate and a base T_{DB} of 35 ^{o}C and RH 44% for ductwork heat load.

	T_{DB} (^{o}C)													
	35.0		35.5		36.0		36.5		37.0		37.5		38.0	
RH (%)	Q_L (kW)	Q_S (kW)	Q_L (kW)	Q_S (kW)	Q_L (kW)	Q_S (kW)	Q_L (kW)	Q_S (kW)	Q_L (kW)	Q_S (kW)	Q_L (kW)	Q_S (kW)	Q_L (kW)	Q_S (kW)
49.0	14.49	0.00	18.48	1.61	22.57	3.22	26.77	4.84	31.07	6.46	35.49	8.08	40.02	9.71
48.5	13.04	0.00	16.98	1.61	21.03	3.22	25.18	4.84	29.44	6.46	33.81	8.08	38.29	9.71
48.0	11.59	0.00	15.49	1.61	19.49	3.22	23.60	4.84	27.81	6.46	32.13	8.08	36.56	9.70
47.5	10.13	0.00	13.99	1.61	17.95	3.22	22.02	4.84	26.18	6.45	30.45	8.07	34.83	9.70
47.0	8.68	0.00	12.50	1.61	16.42	3.22	20.43	4.83	24.55	6.45	28.78	8.07	33.11	9.70
46.5	7.23	0.00	11.01	1.61	14.88	3.22	18.85	4.83	22.93	6.45	27.10	8.07	31.38	9.69
46.0	5.79	0.00	9.52	1.61	13.35	3.22	17.27	4.83	21.30	6.45	25.43	8.07	29.66	9.69
45.5	4.34	0.00	8.03	1.61	11.81	3.22	15.69	4.83	19.67	6.44	23.75	8.06	27.94	9.68
45.0	2.89	0.00	6.54	1.61	10.28	3.22	14.11	4.83	18.05	6.44	22.08	8.06	26.22	9.68
44.5	1.45	0.00	5.05	1.61	8.75	3.21	12.54	4.83	16.42	6.44	20.41	8.06	24.50	9.68
44.0	0.00	0.00	3.56	1.61	7.21	3.21	10.96	4.82	14.80	6.44	18.74	8.05	22.78	9.67

Table 3 demonstrates that when the ductwork is in non-equilibrium condition, the T_{DB} and RH obtained may have a large impact on the calculated heat load, be it sensible or latent. A base condition of 10,000 m^3/hr flowrate with 35 ^{O}C and 44% RH was used to simulate the changing heat load conditions. An increase or change of 0.5 ^{O}C in T_{DB} is enough to generate 1.6 kW of sensible heat. As for the RH, an increment of 0.5% will increase the latent heat to about 1.5 kW. For the 5,000 m^3/hr flowrate, the Sensible Health Load (Q_S) and Q_L is halved. The magnitude of the change is the same when the base conditions are changed to a different T_{DB} and RH.

DISCUSSION

Technical Discussion of Results

Trial 3A - Ambient baseline dehumidification rate response

Trial 3A examined the rate response of dehumidification when both decks 2 and 3 were initially flooded with outside untreated air using the fans prior to the treatments commencing.

During the trial, the changes in T_{DB} observed were very minimal (< 0.5^{O}C) as shown in Figure 18. This in turn meant that there was a very small sensible heat load during the trial. Given the main function of the dehumidifiers was to remove moisture, this result is not unexpected.

The average RH in both decks recorded a reduction across a period of 20 minutes, as shown in Figure 18. The decrease in the average space RH in deck 2 was larger than in deck 3.

The HCU6000 unit used for deck 3 (Figure 19) did not perform as well as expected, as the duct outlet (LC032) reduction and stable RH were not as low as compared to the other duct outlet (LC034) on the same deck. While differences in performance are expected, these should be not be as large as 10%.

The smaller capacity dehumidification unit (HCU3000) on deck 2 showed a drop in RH when it was switched on. The RH of the two duct outlets in deck 2 also demonstrated consistency when the dehumidification units were both switched on. This proves that the two units in deck 2 performed consistently and reliably. This is also supported by the heat load figures from Table 1 and the Q_T and Q_L plots in Figures 36 and 37. Comparison of the decaying of space average RH between decks 2 and 3 shows that the deck 2 space average RH was within the duct outlets' RH levels. As with the deck 3 case, comparing only the LC034 duct outlet for HCU3000, there is still a 5% RH difference. With a longer running time (> 30 minutes), it could be expected that both space average RH and duct outlet average would converge.

As seen in Figure 20, a larger decrease of T_{WB} is recorded in deck 2 (3.0^{O}C) than in deck 3 (2.5^{O}C). The space average T_{WB} in deck 3 does not reduce to a level as low as the T_{WB} of the duct outlets. The deck volume for deck 3 is 2505 m^3 and 2393 m^3 for deck 2. With only a 4% increase of deck volume and the same dehumidification unit type supplying treated air to both decks, the deck 3 space average T_{WB} is higher than deck 2. It is expected that this could be attributed to the duct layout, which were as follows:

- For deck 3, the outlet ducts were placed along the ramp next to the bow steel bulkhead. The concept was that the treated air from the duct outlets would travel downstream, hit the bow steel bulkhead and deflect towards the stern steel bulkhead direction. It is possible that some of the treated air may have been trapped at the corners of the bulkhead and that this could have an influence over the dehumidification of the rest of the deck accordingly.
- For deck 2, the ducts passed down from deck 7 to deck 2 through an access hatch located near the centre of the deck. The treated air then travelled radially and would therefore be expected to have an equal RH and T_{WB} decrease in all directions.

Trial 3B - Cumulative dehumidification rate response

The purpose of this trial was to examine the cumulative rate response for the DHM units. As such, instead of immediately switching on the full flowrate capacity (15,000 m^3/hr) of the DHM units at the initiation of each replicate (as in Trial 3A), the flowrate was increased by sequential 5,000 m^3/hr increments every 20 minutes.

As observed in Figure 21, there were minimal changes in the T_{DB} during the trial, which implied that there was small Q_S present. This is clearly shown in Table 2, with all small Q_S in normal operating condition (no diversion / capping and thermal equilibrium condition) in deck 2. For the deck 3 case, the Q_S is large due to non-thermal equilibrium condition.

The RH reduction trend for both decks was similar to those recorded in Trial 3A, except that a larger reduction was recorded in 3B. In this trial, the RH in deck 3 reduced a total of 17% and in deck 2 reduced by 20%, while in Trial 3A, deck 3 recorded a RH reduction of 9% and deck 2 a reduction in RH of 12%. The reduction rate of the RH was almost similar for both trials in the first 20 minutes. In Trial 3B, as the treatment switched from 5,000 m^3/hr to 10,000 m^3/hr, deck 3 and deck 2 recorded 8% and 11% reductions in RH, respectively. The RH reduced further when allowed to run for a longer period and with increased DHM flowrate.

In deck 3, the T_{WB} dropped 3.3^{O}C at the highest treatment rate and in deck 2, the T_{WB} dropped 4.0^{O}C. In Trial 3A, the comparative figures for deck 3 was a reduction of 2.3^{O}C and for deck 2 a reduction of 3.0^{O}C.

Trial 3C - Ambient baseline dehumidification dilution

The purpose of Trial 3C was to study the effect of DHM treated air diluted with untreated air (outside air delivered via fans on the vessel). In general, the plots in RH and T_{WB} showed a serious dilution effect even with only 25% of fans switched on. With 25% of supply fans and exhaust fans switched on, the dehumidified spaces were quickly brought back to the pre-dehumidified conditions.

At the commencement of the replicates, the dehumidification units were switched on to dehumidify the space. The RH and T_{WB} were close to (deck 3) or reached (deck 2) the space average condition before the application of the fans. This dehumidification process started with rapid decreases in RH and T_{WB} in the first five minutes, followed by less rapid decreases in the remaining 15 minutes (Figure 25 and 26). This observed reduction trend was the same as those reported in Figure 19 and 20, which proves consistency between Trial 3A and 3C when the dehumidifiers were in operation.

After the stable dehumidified condition was achieved, exhaust and supply fans equivalent to 25% of the total capacity were switched on. The introduction of outside air quickly increased the space average RH and T_{WB} back to the level of pre-dehumidified condition.

With the addition of supply and exhaust fans to capacities of 50% and 100% in subsequent treatments, the space average condition RH and T_{WB} were reported to be the same as those outside the vessel. However, this field trial was conducted in an empty vessel where there was no within pen sensible and latent heat load presence. In practice, when livestock are loaded into the vessel it can contribute some sensible and very large latent heat loads. This means that the RH and T_{WB} in the space (or pen) will further increase, and the impact of nature of the dilution effect may shift. Further detailed analysis is outlined in the Practical Discussion and Next Steps section to model the effects of adding livestock to the scenario.

The layout of the supply fans and exhaust fans for the 25% and 50% capacity cases used for decks 2 and 3 are shown below (Figure 38 and 39). Selection of fan combinations were made based on avoiding air short circuits and balancing the flowrate between supply fans and exhaust fans. No fans were selected in the middle of the deck to try and allow the treated air to maintain and dehumidify it.

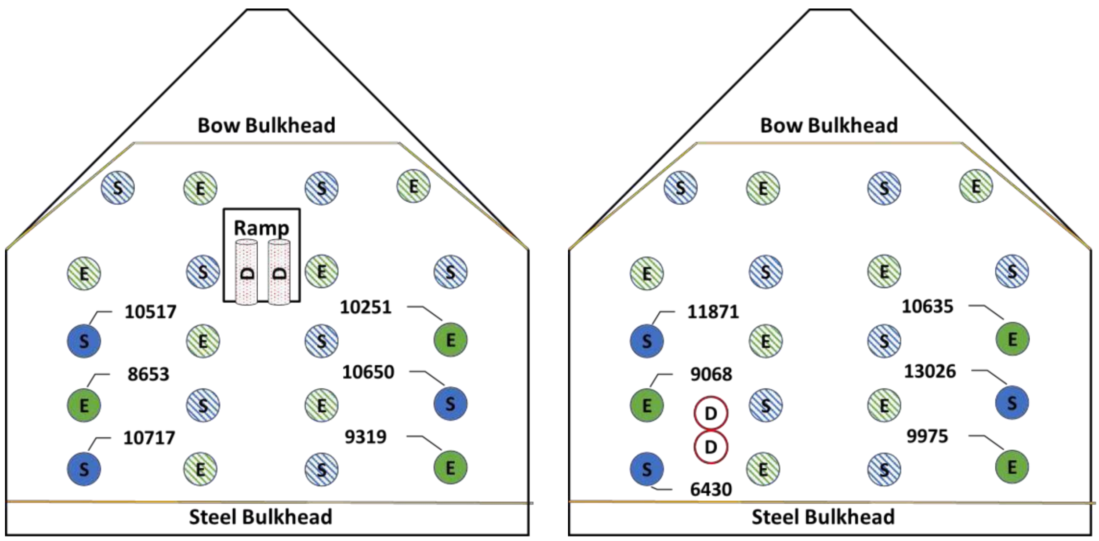


Figure 38: Supply fans (S), Exhaust fans (E) and duct outlet layout in deck 3 (Left) and deck 2 (Right). Solid circle represents switched on fans during 25% total capacity and diagonal line circle represent fans not switched on. The numbers labelled to the switched-on fans are respective flowrate in m^3/hr.

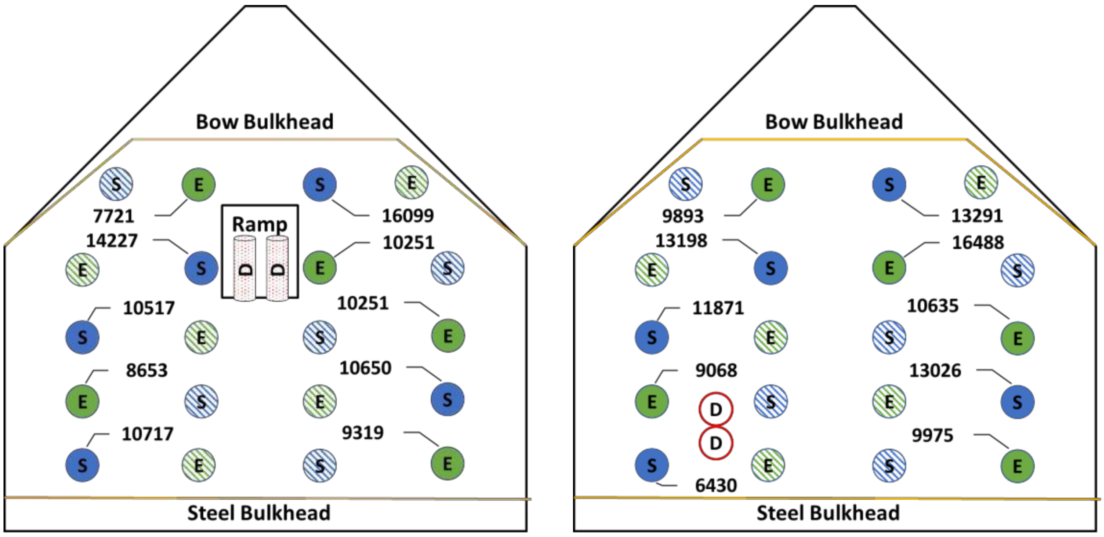


Figure 39: Supply fans (S), Exhaust fans (E) and duct outlet (D) layout in deck 3 (Left) and deck 2 (Right). Solid circle represents switched on fans during 50% total capacity and diagonal line circle represent fans not switched on. The numbers labelled to the switched-on fans are respective flowrate in m^3/hr.

Trial 3D - Ambient airflow baseline (sealed & no dehumidification)

Trial 3D did not involve the use of the DHM units and only focused on the results the supply and exhaust fans being switched on. The purpose of this trial was to examine the actual condition inside the deck without DHM intervention, and only relying on drawing in fresh air from outside the vessel to cool both decks.

With the absence of treated and the presence of untreated fresh air, there was a high correlation between the space average wet bulb temperature and the DHM intake and bridge wet bulb temperatures. However, the space average T_{DB} was below that of the bridge and DHM intake measurements and the space average RH was recorded to be higher that the RH outside the vessel (DHM intake and bridge condition).

Trial 3E - Cumulative exhaust fan dehumidification rate response

The purpose of this trial was to create a pulling phenomenon to determine its influence on regulating the deck condition. As previously indicated, on deck 3 the dehumidified air was delivered through duct outlets positioned in the ramp. During the trial, all fans in row 5, 6 and 7 were switched off, and only fans (normal and modified) in row 8 and 9 were switched on in a cumulative manner.

It was observed that the space average RH and T_{WB} in deck 3 possessed a distinctive step curve shape due to increases in exhaust fan numbers. This distinctive step shape curve is not clearly shown in deck 2 and will be discussed further below.

Several supply fans in rows 8 and 9 on the vessel for decks 2 and 3 were reversed to act as exhaust fans through simple electrical circuit modification in the switch board room. However, this modification does not guarantee the same flowrate is achieved (when the supply fan is converted to exhaust fan, there is a reduction of flowrate). The positions of all fans for both decks are shown below (Figure 40). These fans are arranged in a zig-zag form for supply and exhaust.

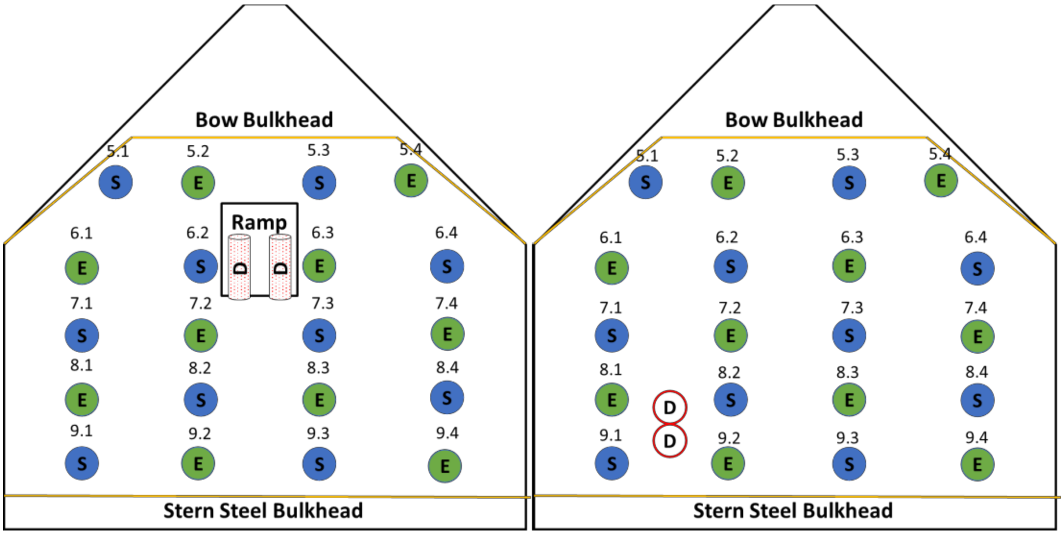


Figure 40: Schematic layout of supply fans (denoted S and blue colour), exhaust fans (denoted E and green colour), and duct outlets (denoted D) for deck 3 (Left) and deck 2 (Right). The numbers on top of fans are the coordinate number respective fan.

As noted earlier, on deck 3 the dehumidified air was released from the duct outlets and then reflected by the bow steel bulkhead. The dehumidified air was then dispersed throughout the deck and stabilised. Two exhaust fans (9.1 and 9.4) were then switched on near the stern steel bulkhead. The dehumidified air was subsequently pulled towards the stern steel bulkhead due to the running of the exhaust fans there. Four exhaust fans in row 9 and later eight exhaust fans in row 8 and 9 were then switched on. The subsequent Figures 41 (two exhaust fans), 42 (four exhaust fans) and 43 (eight exhaust fans) show the air flow pattern in both decks.

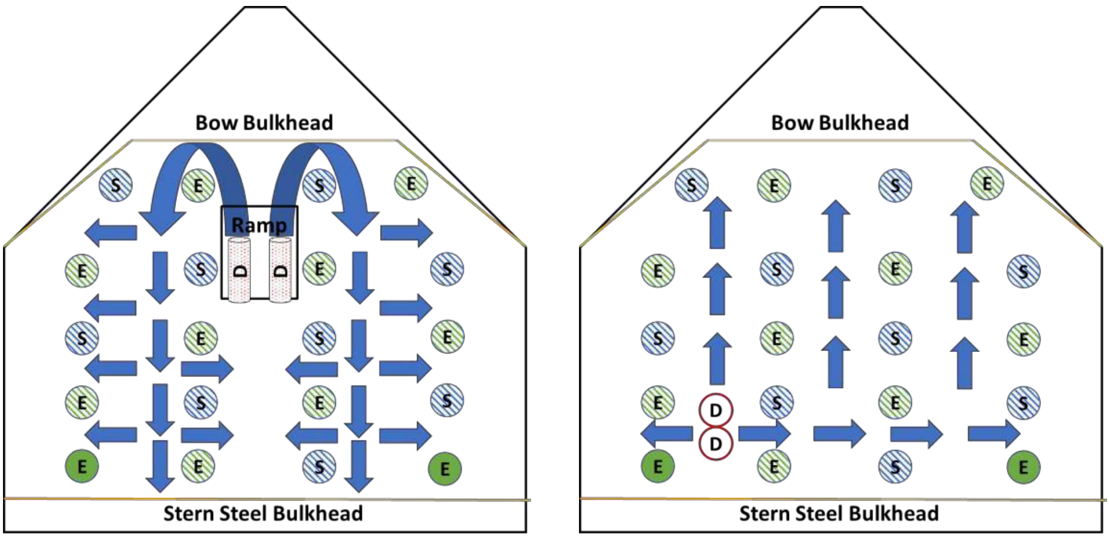


Figure 41: Dehumidified air flow when exhaust fans 9.1 and 9.4 are running for deck 3 (Left) and deck 2 (Right). Fans with diagonal lines are switched off.

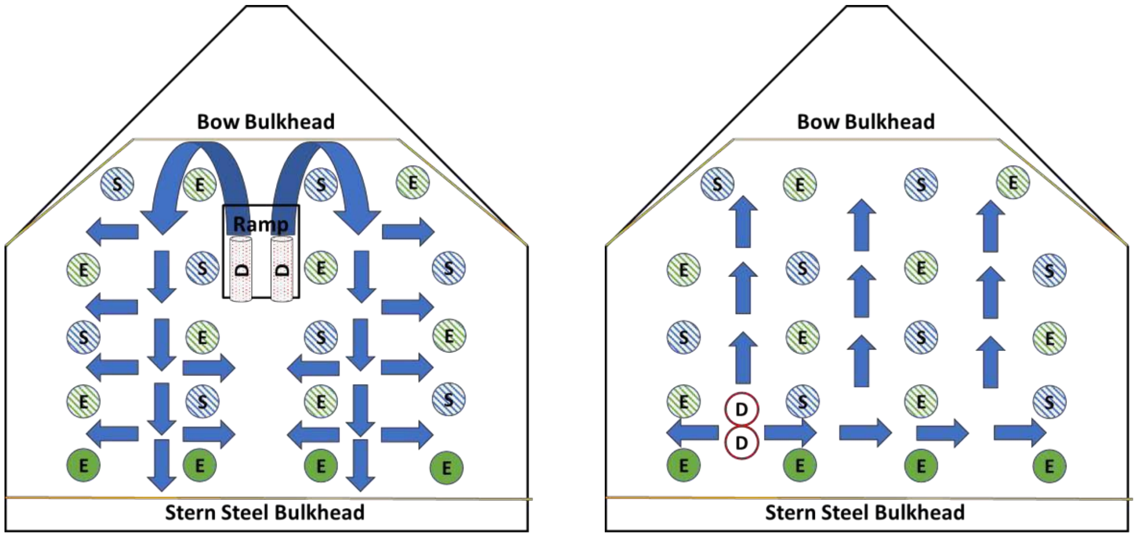


Figure 42: Dehumidified air flow when exhaust fans 9.1, 9.2, 9.3 and 9.4 are running for deck 3 (Left) and deck 2 (Right). Fans with diagonal lines are switched off.

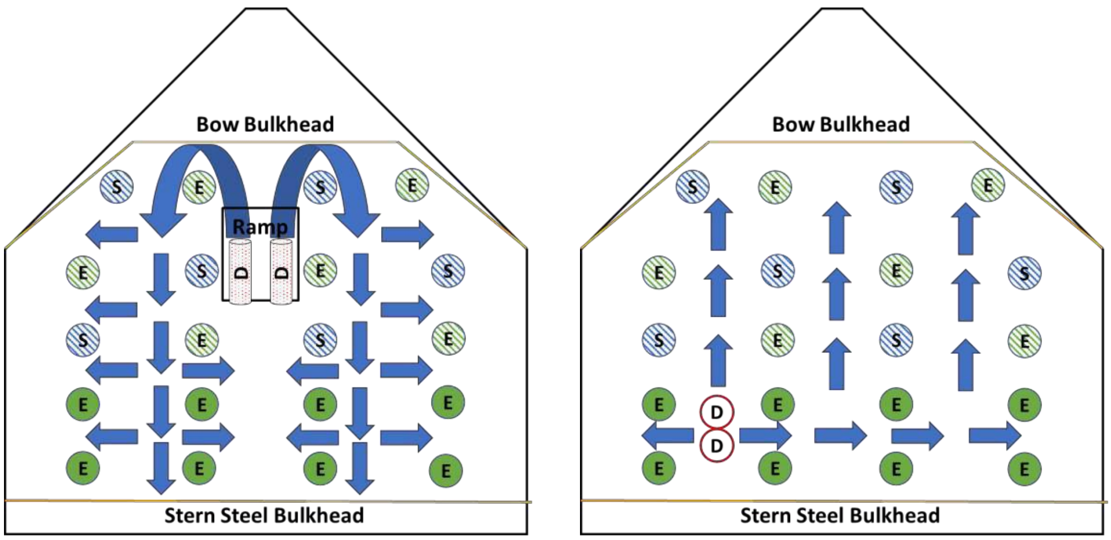


Figure 43: Dehumidified air flow when exhaust fans in row 8 and 9 are running for deck 3 (Left) and deck 2 (Right). Fans with diagonal lines are switched off.

With the dehumidified air supplied in one end of the deck and the exhaust at the other end of the deck 3, it is not surprising to see the step shape curve shown in Figures 26 and 27. The dehumidified air was not directly extracted by the exhaust fans as it flowed across the space in deck 3. Even with eight exhaust fans and DHM running, both the RH and T_{WB} were still lower than the outside condition.

In deck 2, the dehumidified air was directly extracted by the exhaust fans when four and then eight exhaust fans were switched on because of their proximity to the DHM unit outlets. In this regard, it is noted that the entrance for the duct work to deck 2 was via an access hatch (different to deck 3) and that it was not possible to apply different fan configurations on a deck basis (e.g. one built in fan and duct work supply or exhaust all of the decks via a series of outlets, but these cannot be adjusted or turned on or off at an individual deck level). Deck 3 was used as the basis for determining the overriding fan configuration for both decks for this trial and Trial

3F. As such, Figures 26 and 27 showed a less distinctive step curve shape for deck 2. When the four-exhaust fan treatment reached stable condition, the RH and T_{WB} were close to or similar to outside condition.

Trial 3F - Ambient Sequential Exhaust Rate Response

This trial was similar to Trial 3E, with DHM running and exhaust fans switched on in a sequenced manner to expel air from the decks out of the vessel. Outside air was initially introduced to the decks by switching on all fans until the condition in the decks were equal to the outside conditions. Once this was achieved, the DHM units and two exhaust fans were switched on for 20 minutes to examine the reduction of the RH and T_{WB} on both decks. This cycle was repeated for four and eight exhaust fans. The air flow patterns of treated air from the DHM units across the decks are shown in Figures 41 to 43.

In Figures 29 (RH) and 30 (T_{WB}), the shape of the curve resembled a distinctive damping waveform in deck 3. There was also a damping waveform in deck 2 but in a lesser degree compared to deck 3. The Figures 29 and 30 for deck 3 showed that even with up to four (39,963 m^3/hr; $PAT_{V/A}$ = 47.4) and eight (86,311 m^3/hr; $PAT_{V/A}$ = 102.4) exhaust fans running, the space average RH and T_{WB} were still lower than the outside condition. As in Trial 3E, when four exhaust fans were introduced on deck 2, the space average RH and T_{WB} were the same as the outside condition. As mentioned in the earlier Trial 3E discussion, the deck 2 outcomes reflected the sub-optimal layout of the fans and DHM outlets (due to practical limitations) and showed that the treated air was being exhausted before reaching the centre of the deck. The differences of the RH and T_{WB} results between the decks is solely due to the location of the duct outlets relative to the exhaust fans.

Practical Discussion and Next Steps

Interpretation of trial results and development of an environmental model

The trial in Dubai provided invaluable data describing how fans and dehumidifiers operate in a range of ambient wet bulb temperature conditions. However, there were several factors that could vary in real world situations including:

- the dehumidifiers and their outputs
- the fans and their outputs
- the ambient wet bulb temperature, dry bulb temperature and humidity, and
- the presence of live sheep.

Recognising that the proof of concept for the dehumidification of an empty livestock vessel had been achieved, the analysis focused on extrapolating that data to model how the variables and their influence on the pen environment would change in a wider set of circumstances.

Therefore, a simple thermodynamic analysis was performed to estimate the space properties (T_{DB}, RH and T_{WB}) with the influence of sheep heat load generation, DHM treated air supply and supply fans supplied fresh air. The scenario can be compared directly to the Psychrometric simple mixing analysis. In the current scenario, there are two inflows into the deck (space), and the mixing result will be those in the space. The results from the trial are provided as an input condition for DHM and ambient air.

The conservation of Energy (1) is defined as total energy of mass entering the control volume (in our case, referred to deck 3) subtract total energy of mass leaving the control volume equal to net change in energy in the control volume. Assuming steady flow (2), the ΔE_{cv} term is equal to zero and the equation is simplified shown in equation (3). The dot above the "E"

denotes rate of change with respect to time and the equation is in rate form (2 and 3) and usually carries the unit of J/s or kJ/s.

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{cv} \quad (1)$$

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{cv} = 0 \quad (2)$$

$$\dot{E}_{in} = \dot{E}_{out} \quad (3)$$

The $\dot{E}_{in}$ and $\dot{E}_{out}$ terms are then expanded (4) to include input and output for work ( ) and heat ($\dot{Q}$), energy flow ($\dot{m}\,h$), kinetic ($\frac{V^2}{2}$) and potential (gz) energies.

$$\dot{W}_{in} + \dot{Q}_{in} + \sum_{in} \dot{m}\left(h + \frac{V^2}{2} + gz\right) = \dot{W}_{out} + \dot{Q}_{out} + \sum_{out} \dot{m}\left(h + \frac{V^2}{2} + gz\right) \quad (4)$$

Not all terms are suitable to apply in the analysis, therefore, the equation is further simplified (5) to equate a few terms to zero, which are $\dot{W}_{in}$, $\dot{W}_{out}$, $\dot{Q}_{out}$, $\frac{V^2}{2}$ and gz. The E_{in} (Energy input) term include sheep heat load ($\dot{Q}_{in}$), DHM treated air energy flow and fresh air intake energy flow. The E_{out} term will consist only energy flow in space (deck 3).

$$\dot{Q}_{in} + \dot{m}_{DHM,d.a.}h_{DHM,d.a.} + \dot{m}_{air,d.a.}h_{air,d.a.} = \dot{m}_{space,d.a.}h_{space,d.a.} \quad (5)$$

The mass flow rate, $\dot{m}$ in (6) is the product of density multiply with volume flow rate ($\dot{V}$). The mass flow rate carries the unit of kg/s. The density, ρ (kg/m³) is dependent on atmospheric pressure, P_{atm} (usually 101.325 kPa), saturated vapour pressure, P_g (kPa), relative humidity, RH (%) and dry bulb temperature, T_{WB} (°C). The mass flow rate for both DHM and fresh air are calculated with respective density, RH and T_{DB}.

$$\dot{m} = \rho\dot{V} = \left[\frac{P_{atm} - P_g\left(\frac{RH}{100}\right)}{0.287(T_{DB} + 273.15)}\right]\dot{V} \quad (6)$$

The mass flowrate at the space, $\dot{m}_{space,d.a.}$ is determined by conservation of mass equation, for dry air component. The inlets and the outlet of the control volume is considered and is showed below:

$$\dot{m}_{DHM,d.a.} + \dot{m}_{air,d.a.} = \dot{m}_{space,d.a.} \quad (6)$$

The specific enthalpy for space, $h_{space,d.a.}$ contained both the sensible (c_pT_{DB}) and latent (ωh_g) components shown in equation (7). Specific enthalpy carries the unit of kJ/kg. The constant specific heat is denoted by c_p (kJ/kg°K) and the absolute humidity is represented by ω ($kg_{w.v.}/kg_{d.a.}$)

$$h_{space,d.a} = c_pT_{DB} + \omega_{space,d.a.}h_g \quad (7)$$

The saturated vapour specific enthalpy, h_g is a summation of saturated vapour specific enthalpy, $h_{g@0\ deg\ C}$ and the sensible heat, c_pT_{DB}, where $h_{g@0\ deg\ C}$ has the value of 2500.9 kJ/kg at 0°C and c_p has the value of 1.82kJ/kg°K.

$$h = h_{g@0^oC} + 1.82T_{DB} \quad (8)$$

The absolute humidity in space, $\omega_{space,d.a.}$ can be solved by assuming the water vapour mass is conserve in the control volume (deck) shown in equation (9). The absolute humidity is the ratio of water vapour mass and dry air mass.

$$m_{sheep} + m_{DH} + m_{air} = m_{space} \quad (9)$$

$$\omega_{sheep,d.a.}.m_{sheep,d.a.} + \omega_{DH,d.a.}.m_{DH,d.a.} + \omega_{air,d.a.}.m_{air,d.a.} = \omega_{space,d.a.}.m_{space,d.a.} \quad (10)$$

$$\frac{Q_{sheep}}{h_g} + \omega_{DH,d.a.}.m_{DH,d.a.} + \omega_{air,d.a.}.m_{air,d.a.} = \omega_{space,d.a.}.m_{space,d.a.} \quad (11)$$

$$\omega_{space,d.a.} = \frac{\frac{Q_{sheep}}{h_g} + \omega_{DH,d.a.}.m_{DH,d.a.} + \omega_{air,d.a.}.m_{air,d.a.}}{m_{space,d.a.}} \quad (12)$$

The h_g and $h_{space,d.a.}$ terms are then substitute into equation (5) and becomes equation (13). The equation is then rearranged to obtain T_{DB} (14).

$$Q_{t} + m_{DH,d.a.}.h_{DH,d.a.} + m_{air,d.a.}.h_{air,d.a.} = m_{space,d.a.}[c_pT_{DB} + \omega_{space,d.a.}(h_{g@0^oC} + 1.82T_{DB})] \quad (13)$$

$$T_{DB} = \frac{\frac{Q_{t} + m_{DH,d.a.}.h_{DH,d.a.} + m_{air,d.a.}.h_{air,d.a.}}{m_{space,d.a.}} - \omega_{space,d.a.}(h_{g@0^oC})}{c_p + 1.82\omega_{space,d.a.}} \quad (14)$$

The saturated vapour pressure (in kPa) is then determined by equation (15) where T_{DB} is main variable.

$$P_g = 0.61121e^{\left[\left(18.678-\frac{T_{DB}}{234.5}\right)\left(\frac{T_{DB}}{257.14+T_{DB}}\right)\right]} \quad (15)$$

The Relative Humidity, RH (%) is then computed from equation (16) through substituting the P_g from equation (15).

$$\frac{RH}{100} = \frac{\omega_{space,d.a.}.P_{atm}}{P_g(0.622 + \omega_{space,d.a.})} \quad (16)$$

The T_{WB} is estimated by a novel method (Lee & Wang 2018[1]) which gives better than 0.1^{O}C accuracy. After computing relevant terms of $e_s(T_{DB})$ and $f(T_{WB})$ represented by equations (17) and (18) respectively, the T_{WB} can then be calculated as shown in equation (19).

$$e_s(T_{DB}) = 2.796413\times10^{-8}T_{DB}^5 + 2.671942\times10^{-6}T_{DB}^4 + 2.73199\times10^{-4}T_{DB}^3 + 1.41951\times10^{-2}T_{DB}^2 + 0.444226T_{DB} + 6.1078 \quad (17)$$

$$�(T_{��}) = 0.01�H��(T_{��}) + 10\gamma�_{a��}T_{��} \quad (18)$$

[1] Chien Lee and Yu-Jen Wang. "A novel method to derive formulas for computing the wet-bulb temperature from relative humidity and air temperature." Measurement, 128 (2018), 271-275

$$T_{??} = 7.438995 \times 10^{-10}[?(T_{??})]^5 - 4.063282 \times 10^{-7}[?(T_{??})]^4 + 9.16616 \times 10^{-5}[?(T_{??})]^3 - 1.15133 \times 10^{-2}[?(T_{??})]^2 + 1.02533\, ?(T_{??}) - 5.85331 \quad (19)$$

The calculated T_{WB} is assumed to be the pen T_{WB}. The difference of T_{WB} between ambient fresh air and the pen, the ΔT_{WB} is then computed.

Within the wider combined model, consideration was also given to how the ventilation provided from the fans and the dehumidifiers was represented.

The industry used terminology (from the HSRA) of Pen Air Turnover (PAT) was considered for the model. PAT is the ratio between volumetric flowrate (m^3/s or m^3/hr) and pen area (m^2). The result of such a ratio equates to velocity (m/s or m/hr). The relationship between PAT and dimensions of the livestock housing can be explained as if the fresh air is introduced equally through the pen floor and extracted evenly through the ceiling above it. This vertical air velocity can be defined as the PAT. Previous industry research relating to HSRA model suggests that the range for PAT for vessels is commonly between 100 and 300 m/hr[3].

The use of Air Exchanges per Hour (AEH) was also considered as a means of expressing ventilation of the space in the model. AEH is the ratio between volumetric flowrate (m^3/s or m^3/hr) and deck volume (m^3). Given the same stocking density and heat load but a space with twice the deck height, would require doubling the volumetric flowrate to equate with the PAT's "air changes".

It was concluded to use a refined definition of PAT in the model. As such, the calculation of the ventilation in the following model simulations defined 'PAT' as a ratio between total air supply volume flowrate (DHM treated air and/or fresh untreated air from fans) and deck 3 pen area. This is different to the current PAT term used in the HSRA model, which carries the unit of m/hr. To differentiate between these different PAT forms, the PAT term used within this document will take the form of $PAT_{V/A}$ and the HSRA aligned PAT will take the form of PAT_{vel}. For the DHM PAT, it is denoted as PAT_{DHM}.

Modelling the impact of animals

The following section outlines the development of the animal-sub model that integrated into the environmental model and was used for more detailed simulations.

From the information gather during the trial on the performance of the dehumidifiers and fans, a thermodynamic model of vessel pen space was constructed. Within this overarching model, a sheep animal sub-model was developed to predict the combined effect of heat generation by sheep, ventilation and dehumidification on pen conditions.

The sheep heat production sub-model was developed using information on the heat generation characteristics of sheep taken from literature[3,4] and expert opinion.
The animal heat generation model has two key concepts:

1. Sheep metabolic heat production – estimated at 3.2W/kg liveweight (i.e. 3.2 Joules per kg of liveweight per second), and
2. heat loss or dispersion fractions as sensible and latent heat.

[3] LIVE112,Salmonellosis control and best-practice in live sheep export feedlots (2003), LiveCorp / Meat and Livestock Australia (MLA).

[4] Ames et al. J. An. Sci. 32(4) 784-8; LiveCorp/MLA research reports – LIVE116, Development of a Heat Stress Risk Assessment Model (2004); SBMR.002, Investigation of the Ventilation Efficacy on Livestock Vessels (2001).

In summary, animals can lose surplus metabolic heat via radiation, conduction, convection and evaporation. In general, as ambient temperatures increase, evaporative cooling becomes the predominant method for expelling surplus heat in livestock – especially in sheep due to their wool cover. The proportion of heat loss via evaporative cooling (i.e. panting) in relation to ambient temperature used in the animal sub-model is presented in Table 4.

Table 4: Percentage evaporative heat loss by sheep at various ambient temperatures

Ambient Temperature (T_{DB})	**Percentage Evaporative Heat Loss**
20	40
25	55
30	70
35	95
40	110*

* When ambient temperatures are above body temperature, animals must expel surplus metabolic heat and any extra heat transferred to them from the environment in order to maintain body temperature within normal range. For this reason, evaporative heat loss percentages are above 100%.

The industry heat stress risk assessment model (HSRA) is a thermodynamic model that combines voyage variables, such as class of livestock, ambient wet bulb temperature conditions and ventilation fan output to calculate a recommended stocking density and risk of heat stress for voyages travelling to or through the Middle East.

The calculation in the HSRA model is based upon estimating the elevation in deck wet bulb temperature (T_{WB}) above ambient wet bulb temperature (i.e. the delta T_{WB}), due to the presence of livestock and after adjusting for the application of interventions (i.e. ventilation fans).

Typical estimated ΔT_{WB} are understood to range between 1.0 – 2.2°C. This elevation in pen T_{WB} is combined with estimates of expected ambient T_{WB} to estimate the risk of a heat stress event for the proposed voyage.

The equation used within the HSRA model for calculating the ΔT_{WB} is[5]:

$T_{WB} = 3.6 \times C \times M \times h / (\rho \times PAT)$ where:

- T_{WB} is the wet bulb temperature increase (˚C);
- C is the 'constant' of proportionality relating T_{WB} to the internal energy rise, taken as 0.230C/(kJ/kg);
- M is the liveweight in the particular ventilation zone (kg/m^2) (M = beast weight ∣area per head) (275kg/m^2 for cattle, 180kg/m^2 for large sheep, etc.);
- H is the 'per mass' rate of metabolic heat. This is variable, however here we will take 2W/kg for Bos indicus cattle, 2.4W/kg for Bos taurus cattle and 3.2W/kg for sheep;
- ρ is the density of air (1.2kg/m^3) PAT, the pen air turnover in m/hr, is the ratio of the fresh air flowrate (Q) in m /hr to the pen area (A) in m^2;
- The factor 3.6 at the front corrects units from W to kW and hours to seconds.

[5]Note, within this version of the HSRA equation H is equivalent to h.

The information from HSRA was used to help verify the animal thermodynamic sub-model developed for this project within the overarching thermodynamic model. The animal thermodynamic sub-model's role is to predict the combined effects of livestock, ventilation (fans) and dehumidifier performance on deck T_{WB} (i.e. ΔT_{WB}).

Verification of the sub-model was undertaken by estimating the required fan performance (as a proportion of maximum) in the absence of dehumidification that would be required to limit the increase in pen T_{WB} to 2.2 degrees (i.e. an assumed typical estimate for ΔT_{WB} arising from the presence of livestock).

The results of this verification activity are presented in Table 2 and show that the fans on this vessel must operate at or above 75% maximum capacity in order to limit the T_{WB} elevation to 2.2 degrees. This verification is reflective of reality and suggests that the animal sub-model developed for use here is also reflective of the animal model used within the (proven) HSRA model.

This verification offers confidence that the animal sub-model, when included into the environmental thermodynamic model, provides accurate predictions of deck T_{WB} conditions from across the expected range of combinations of livestock class and type, stocking density, ambient conditions, fans and dehumidifier use.

It should also be noted that this sub-model reflects, but is not the same, as the industry HSRA. One difference is that a different metric for ventilation is used. This is referred to as $PAT_{V/A}$, or volumetric Pen Air Turnover. The figures presented therefore are not the same as the Pen Air Turnover figures used in the HSRA model.

Modelling the effect of dehumidification on a vessel stocked with sheep

The combined animal-environmental thermodynamic model was subsequently used to simulate the effect of dehumidification on vessels stocked with sheep (to the level required under the Australian Standards for the Export of Livestock version 2.3).

Deck 3F of the vessel was used for the simulation and the key information used in the calculation in relation to this space was as followed:

sheep density (sheep/m^2)	**3.45**
Sheep heat load (W/kg)	**3.2**
Sheep weight (kg)	**40**
Latent Percentage (%)	**84.557**
Pen area (m^2)	**843.25**
Deck Area (m^2)	**1237**
Deck volume (m^3)	**2505**

The baseline simulation – effectively being the trial scenario with sheep added – allowed the effect of dehumidification on deck wet bulb temperature to be defined across varying ambient wet bulb temperatures and fan ventilation rates (fans operating at 100% capacity provided $PAT_{V/A}$ of 162.5 [equivalent to 54.7 air exchanges per hour] in this case). The results of this simulation are summarised in Table 5. They suggest that the addition of dehumidification to fan ventilation would reduce deck wet bulb temperatures by between 0.5–0.6°C, when compared to the use of fan ventilation alone from across the range of expected ambient temperatures. The actual observed differences from specifically modelled combinations are presented in Figure 44.

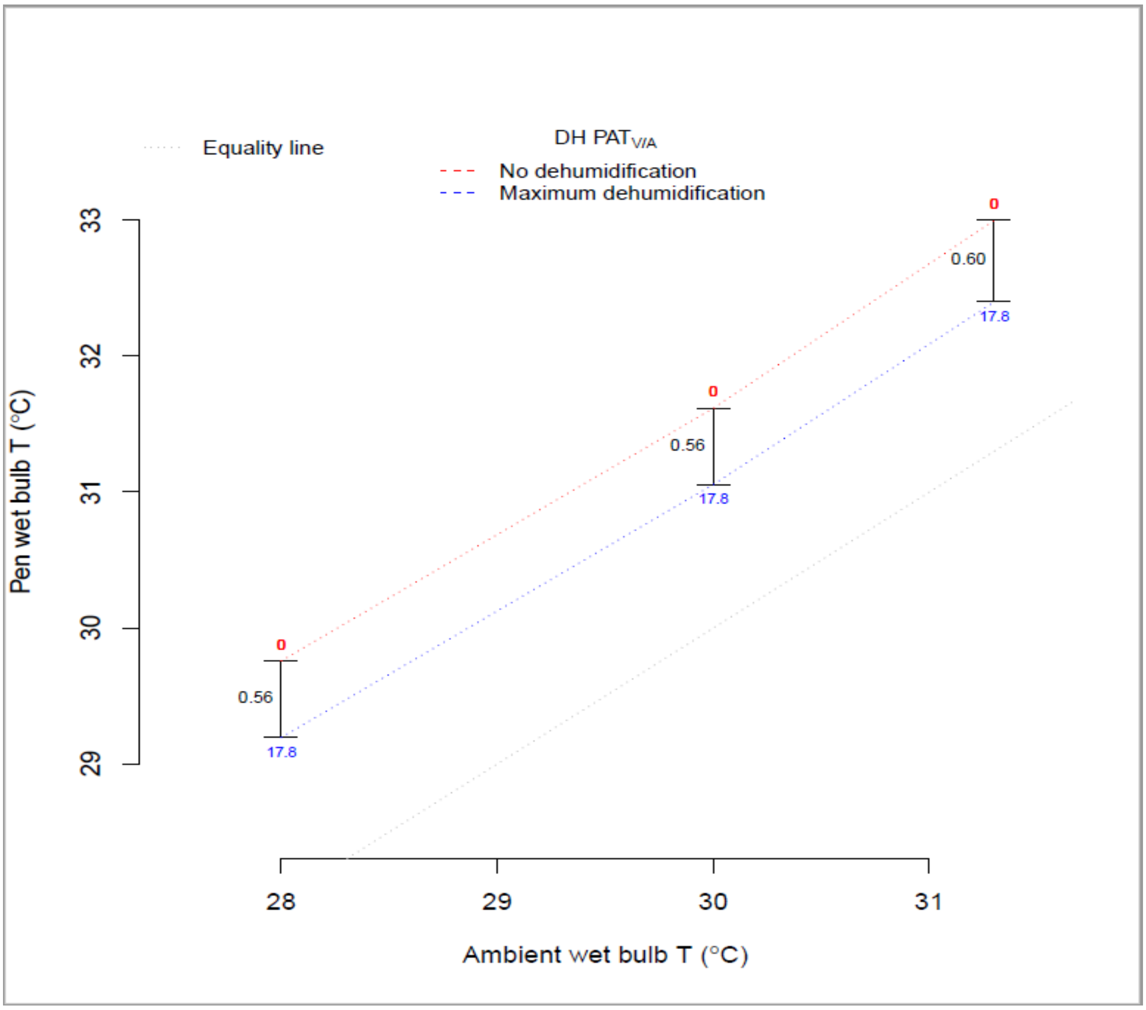


Figure 44: Pen wet bulb temperature by ambient temperature and dehumidifier (DH) $PAT_{V/A}$ (at 100% fan $PAT_{V/A}$ - 54.7) from animal thermodynamic modelling

Using the model, further simulations were able to be completed that extended the DHM maximum $PAT_{V/A}$ beyond the trial limits (e.g. based on $PAT_{V/A}$ 17.8 for DHM). This included doubling and tripling the volumetric flowrates to $PAT_{V/A}$ = 35.6 and $PAT_{V/A}$ = 54.7. It was also possible to look at how the increases in DHM $PAT_{V/A}$ beyond the trialled rates would be influenced by fan rates beyond those trialled. The results of these simulations are shown in Tables 5 to 10, with comments provided on each. Figures 45 to 47 plot the change in T_{WB} against PAT_{DHM} in different ambient conditions.

Table 5 sets out the baseline effects of fans and dehumidifiers as tested in the trials (i.e. 100% of fan $PAT_{V/A}$ = 162.5; and 100% of DHM $PAT_{V/A}$ = 17.8) against various ambient conditions, with the modelled addition of live sheep. In summary, the recorded ΔT_{WB} for the baseline application of fans (Fans 100% and DHM 0%) in the selected ambient conditions ranged from 1.62 to 2.04 ^{O}C. The ΔT_{WB} for the dilution case, with maximum flow capacity for fans and DHM applied (Fans 100% and DHM 100%) for the selected ambient conditions, was estimated to be 1.06 to 1.47 ^{O}C. The ΔT_{WB} with only two DHM units supplying 15,000 m^3/hr (PAT_{DHM} = 17.8) to deck 3, and no fans, ranged from 8.42 to 11.73 ^{O}C, with the RH in the pen exceeding 100%, across the ambient conditions modelled.

Table 5: Baseline fans and two DHM units (supplying 15,000 m^3/hr). The T_{WB} difference between pen and outside condition is estimated for four different outside conditions running with two DHM units (one HCU3000 and one HCU6000) operating in deck 3 (equivalent of PAT_{DHM} = 17.8).

Operating Configurations	$T_{DB\ Amb.}$ (°C)	$RH_{Amb.}$ (Const.)	$T_{WB\ Amb.}$ (°C)	AEH_{Fans} (1/hr)	AEH_{DHM} (1/hr)	$PAT_{V/A}$ (m/hr)	PAT_{DHM} (m/hr)	$T_{DB\ Pen}$ (°C)	RH_{Pen} (Const.)	$T_{WB\ Pen}$ (°C)	$\Delta T_{WB\ (Pen - Amb.)}$ (°C)
Fans 100%, DH 0%	36.0	0.547	28.0	54.7	0.0	162.5	0.0	36.14	62.90	29.76	1.76
Fans 100%, DH 100%	36.0	0.547	28.0	54.7	6.0	162.5	17.8	36.00	60.61	29.19	1.19
Fans 0%, DH 100%	36.0	0.547	28.0	0.0	6.0	0.0	17.8	36.01	111.97	37.76	9.76
Fans 79%, DH 0% (+2.2 T_{WB})	36.0	0.547	28.0	43.3	0.0	128.6	0.0	36.16	65.06	30.20	2.20
Fans 48%, DH 100% (+2.2 T_{WB})	36.0	0.547	28.0	26.2	6.0	77.9	17.8	36.01	65.79	30.20	2.20
Fans 100%, DH 0%	35.9	0.652	30.0	54.7	0.0	162.5	0.0	36.06	73.21	31.62	1.62
Fans 100%, DH 100%	35.9	0.652	30.0	54.7	6.0	162.5	17.8	35.98	70.51	31.06	1.06
Fans 0%, DH 100%	35.9	0.652	30.0	0.0	6.0	0.0	17.8	36.20	118.20	38.82	8.82
Fans 73%, DH 0% (+2.2 T_{WB})	35.9	0.652	30.0	39.8	0.0	118.1	0.0	36.10	76.24	32.20	2.20
Fans 41%, DH 100% (+2.2 T_{WB})	35.9	0.652	30.0	22.4	6.0	66.5	17.8	36.04	76.59	32.20	2.20
Fans 100%, DH 0%	35.0	0.400	24.0	54.7	0.0	162.5	0.0	35.34	48.10	26.04	2.04
Fans 100%, DH 100%	35.0	0.400	24.0	54.7	6.0	162.5	17.8	34.99	46.78	25.47	1.47
Fans 0%, DH 100%	35.0	0.400	24.0	0.0	6.0	0.0	17.8	34.98	105.12	35.73	11.73
Fans 93%, DH 0% (+2.2 T_{WB})	35.0	0.400	24.0	50.8	0.0	150.8	0.0	35.36	48.72	26.19	2.19
Fans 62%, DH 100% (+2.2 T_{WB})	35.0	0.400	24.0	34.2	6.0	101.6	17.8	34.99	50.20	26.21	2.21
Fans 100%, DH 0%	39.9	0.546	31.3	54.7	0.0	162.5	0.0	39.90	62.43	33.00	1.70
Fans 100%, DH 100%	39.9	0.546	31.3	54.7	6.0	162.5	17.8	39.66	60.59	32.40	1.10
Fans 0%, DH 100%	39.9	0.546	31.3	0.0	6.0	0.0	17.8	37.50	114.90	39.72	8.42
Fans 12%, DH 50%	39.9	0.546	31.3	6.6	3.0	19.6	8.9	39.14	95.52	38.42	7.12
Fans 77%, DH 0% (+2.2 T_{WB})	39.9	0.546	31.3	41.9	0.0	124.5	0.0	39.90	64.80	33.49	2.19
Fans 42%, DH 100% (+2.2 T_{WB})	39.9	0.546	31.3	23.2	6.0	68.9	17.8	39.41	67.02	33.50	2.20

Table 6: Baseline fans and four DHM units (supplying 30,000 m³/hr). ***The T_{WB} difference between pen and outside condition is estimated for four different outside conditions running with four DHM units in deck 3 (equivalent of PAT_{DHM} = 35.6).***

Operating Configurations	$T_{DB\ Amb.}$ (°C)	$RH_{Amb.}$ (Const.)	$T_{WB\ Amb.}$ (°C)	AEH_{Fans} (1/hr)	AEH_{DHM} (1/hr)	$PAT_{V/A}$ (m/hr)	PAT_{DHM} (m/hr)	$T_{DB\ Pen}$ (°C)	RH_{Pen} (Const.)	$T_{WB\ Pen}$ (°C)	$\Delta T_{WB\ (Pen - Amb.)}$ (°C)
Fans 100%, DH 0%	36.0	0.547	28.0	54.7	0.0	162.5	0.0	36.14	62.90	29.76	1.76
Fans 100%, DH 100%	36.0	0.547	28.0	54.7	12.0	162.5	35.6	35.89	58.73	28.72	0.72
Fans 0%, DH 100%	36.0	0.547	28.0	0.0	12.0	0.0	35.6	35.66	75.86	31.72	3.72
Fans 79%, DH 0% (+2.2 T_{WB})	36.0	0.547	28.0	43.3	0.0	128.6	0.0	36.16	65.08	30.20	2.20
Fans 17%, DH 100% (+ 2.2 T_{WB})	36.0	0.547	28.0	9.3	12.0	27.6	35.6	35.75	67.04	30.21	2.21
Fans 100%, DH 0%	35.9	0.652	30.0	54.7	0.0	162.5	0.0	36.06	73.21	31.62	1.62
Fans 100%, DH 100%	35.9	0.652	30.0	54.7	12.0	162.5	35.6	35.91	68.29	30.59	0.59
Fans 0%, DH 100%	35.9	0.652	30.0	0.0	12.0	0.0	35.6	35.94	82.16	33.07	3.07
Fans 73%, DH 0% (+2.2 T_{WB})	35.9	0.652	30.0	39.8	0.0	118.1	0.0	36.10	76.19	32.19	2.19
Fans 9.4%, DH 100% (+ 2.2 T_{WB})	35.9	0.652	30.0	5.1	12.0	15.2	35.6	35.93	77.13	32.20	2.20
Fans 100%, DH 0%	35.0	0.400	24.0	54.7	0.0	162.5	0.0	35.34	48.10	26.04	2.04
Fans 100%, DH 100%	35.0	0.400	24.0	54.7	12.0	162.5	35.6	34.73	45.58	25.00	1.00
Fans 0%, DH 100%	35.0	0.400	24.0	0.0	12.0	0.0	35.6	34.32	68.23	29.13	5.13
Fans 93%, DH 0% (+2.2 T_{WB})	35.0	0.400	24.0	50.9	0.0	151.1	0.0	35.36	48.71	26.19	2.19
Fans 33%, DH 100% (+ 2.2 T_{WB})	35.0	0.400	24.0	18.2	12.0	54.1	35.6	34.55	51.97	26.20	2.20
Fans 100%, DH 0%	39.9	0.546	31.3	54.7	0.0	162.5	0.0	39.90	62.43	33.00	1.70
Fans 100%, DH 100%	39.9	0.546	31.3	54.7	12.0	162.5	35.6	39.47	59.04	31.90	0.60
Fans 0%, DH 100%	39.9	0.546	31.3	0.0	12.0	0.0	35.6	37.50	79.11	34.02	2.72
Fans 12%, DH 50%	39.9	0.546	31.3	6.6	6.0	19.6	17.8	38.75	83.46	35.98	4.68
Fans 76%, DH 0% (+2.2 T_{WB})	39.9	0.546	31.3	41.9	0.0	124.5	0.0	39.90	64.86	33.50	2.20
Fans 7.0%, DH 100% (+ 2.2 T_{WB})	39.9	0.546	31.3	23.2	12.0	68.9	35.6	38.08	73.22	33.49	2.19

In the simulation shown in table 6, four DHM units (equivalent to 2 HCU 3000 and 2 HCU 6000 units) supplying 30,000m^3/hr (PAT_{DHM} = 35.6) are incorporated into the model and the effect assessed in concert with fans, ambient conditions and livestock heat generation.

The ΔT_{WB} for the dilution cases (fans 100% and DHM 100%) are considerably lower compared to those in the PAT_{DHM} = 18 cases (in table 5). The ΔT_{WB} ranged from 0.59 ^{O}C – 1.00 ^{O}C.

With 100% DH (PAT_{DHM} = 35.6), none of the cases have a pen RH higher than 100% (unlike the pen RH in the PAT_{DHM} = 18 cases). The ΔT_{WB} ranged from 2.72^{O}C – 5.13 ^{O}C. To reduce the ΔT_{WB} to 2.20 ^{O}C would require 100% DH to be combined with a reduced fan flow capacity.

For reference, the recorded ΔT_{WB} for the baseline application of existing fans (Fans 100% and DHM 0%) in the selected ambient conditions ranged from 1.62 to 2.04 ^{O}C.

Table 7: Baseline fans and six DHM units (supplying 45,000 m³/hr). The T_{WB} difference between pen and outside condition is estimated for four different outside conditions running with six DHM units in deck 3 (equivalent of PAT_{DHM} = 53.5).

Operating Configurations	$T_{DB\ Amb.}$ (^oC)	$RH_{Amb.}$ (Const.)	$T_{WB\ Amb.}$ (^oC)	AEH_{Fans} (1/hr)	AEH_{DHM} (1/hr)	$PAT_{V/A}$ (m/hr)	PAT_{DHM} (m/hr)	$T_{DB\ Pen}$ (^oC)	RH_{Pen} (Const.)	$T_{WB\ Pen}$ (^oC)	$\Delta T_{WB\ (Pen - Amb.)}$ (^oC)
Fans 100%, DH 0%	36.0	0.547	28.0	54.7	0.0	162.5	0.0	36.14	62.90	29.76	1.76
Fans 100%, DH 100%	36.0	0.547	28.0	54.7	18.0	162.5	53.5	35.80	57.15	28.32	0.32
Fans 0%, DH 100%	36.0	0.547	28.0	0.0	18.0	0.0	53.5	35.44	63.56	29.27	1.27
Fans 79%, DH 0% (+2.2 T_{WB})	36.0	0.547	28.0	43.3	0.0	128.6	0.0	36.16	65.09	30.21	2.21
Fans 0.04%, DH 100% (+ 2.2 T_{WB})	36.0	0.547	28.0	0.0	18.0	0.1	53.5	35.44	63.55	29.27	1.27
Fans 100%, DH 0%	35.9	0.652	30.0	54.7	0.0	162.5	0.0	36.06	73.21	31.62	1.62
Fans 100%, DH 100%	35.9	0.652	30.0	54.7	18.0	162.5	53.5	35.86	66.43	30.19	0.19
Fans 0%, DH 100%	35.9	0.652	30.0	0.0	18.0	0.0	53.5	35.78	69.88	30.76	0.76
Fans 73%, DH 0% (+2.2 T_{WB})	35.9	0.652	30.0	39.8	0.0	118.1	0.0	36.10	76.17	32.18	2.18
Fans 0.04%, DH 100% (+ 2.2 T_{WB})	35.9	0.652	30.0	0.0	18.0	0.1	53.5	35.78	69.87	30.76	0.76
Fans 100%, DH 0%	35.0	0.400	24.0	54.7	0.0	162.5	0.0	35.34	48.10	26.04	2.04
Fans 100%, DH 100%	35.0	0.400	24.0	54.7	18.0	162.5	53.5	34.50	44.58	24.60	0.60
Fans 0%, DH 100%	35.0	0.400	24.0	0.0	18.0	0.0	53.5	33.85	55.92	26.40	2.40
Fans 93%, DH 0% (+2.2 T_{WB})	35.0	0.400	24.0	50.9	0.0	151.1	0.0	35.36	48.71	26.19	2.19
Fans 3.1%, DH 100% (+ 2.2 T_{WB})	35.0	0.400	24.0	1.7	18.0	5.0	53.5	33.89	54.73	26.21	2.21
Fans 100%, DH 0%	39.9	0.546	31.3	54.7	0.0	162.5	0.0	39.90	62.43	33.00	1.70
Fans 100%, DH 100%	39.9	0.546	31.3	54.7	18.0	162.5	53.5	39.30	57.72	31.47	0.17
Fans 0%, DH 100%	39.9	0.546	31.3	0.0	18.0	0.0	53.5	37.50	66.78	31.73	0.43
Fans 12%, DH 50%	39.9	0.546	31.3	6.6	9.0	19.6	26.7	38.51	75.80	34.37	3.07
Fans 77%, DH 0% (+2.2 T_{WB})	39.9	0.546	31.3	41.9	0.0	124.5	0.0	39.90	64.80	33.49	2.19
Fans 0.04%, DH 100% (+ 2.2 T_{WB})	39.9	0.546	31.3	0.0	12.0	0.1	35.6	37.50	66.76	31.73	0.43

A further reduction of overall ΔT_{WB} compared to PAT_{DHM} = 18 and PAT_{DHM} = 36. In some cases, the ΔT_{WB} = 2.20 ^{O}C cannot be achieved. In this simulation, the ΔT_{WB} for the dilution cases (fans 100% and DHM 100%) range from 0.17 ^{O}C – 0.60 ^{O}C.

With 100% DH (PAT_{DHM} = 53.5), the ΔT_{WB} is from 0.43 ^{O}C – 2.40^{O}C and again the recorded ΔT_{WB} for the baseline application of existing fans (Fans 100% and DHM 0%) in the selected ambient conditions ranged from 1.62 to 2.04 ^{O}C.

Table 8: Additional fan supply and two DHM units (supplying 15,000 m³/hr). The T_{WB} difference between pen and outside condition is estimated for different fan and ambient conditions with two DHM units running in deck 3.

Operating Configurations	$T_{DB\ Amb.}$ (°C)	$RH_{Amb.}$ (Const.)	$T_{WB\ Amb.}$ (°C)	AEH_{Fans} (1/hr)	AEH_{DHM} (1/hr)	$PAT_{V/A}$ (m/hr)	PAT_{DHM} (m/hr)	PAT_{Total} (m/hr)	$T_{DB\ Pen}$ (°C)	RH_{Pen} (Const.)	$T_{WB\ Pen}$ (°C)	$\Delta T_{WB\ (Pen\ -\ Amb.)}$ (°C)
Fans 100%, DHM 0%	35.0	0.400	24.0	54.7	0.0	162.5	0.0	162.5	35.34	48.10	26.04	2.04
Fans 100%, DHM 100%	35.0	0.400	24.0	54.7	6.0	162.5	17.8	180.3	34.99	46.78	25.47	1.47
Fans **140%**, DHM 0%	35.0	0.400	24.0	76.6	0.0	227.7	0.0	227.7	35.26	45.77	25.47	1.47
Fans 100%, DHM 0%	36.0	0.547	28.0	54.7	0.0	162.5	0.0	162.5	36.14	62.90	29.76	1.76
Fans 100%, DHM 100%	36.0	0.547	28.0	54.7	6.0	162.5	17.8	180.3	36.00	60.61	29.19	1.19
Fans **149%**, DHM 0%	36.0	0.547	28.0	81.4	0.0	241.9	0.0	241.9	36.10	60.20	29.19	1.19
Fans 100%, DHM 0%	35.9	0.652	30.0	54.7	0.0	162.5	0.0	162.5	36.06	73.21	31.62	1.62
Fans 100%, DHM 100%	35.9	0.652	30.0	54.7	6.0	162.5	17.8	180.3	35.98	70.51	31.06	1.06
Fans **155%**, DHM 0%	35.9	0.652	30.0	85.0	0.0	252.6	0.0	252.6	36.01	70.34	31.05	1.05
Fans 100%, DHM 0%	39.9	0.546	31.3	54.7	0.0	162.5	0.0	162.5	39.90	62.43	33.00	1.70
Fans 100%, DHM 100%	39.9	0.546	31.3	54.7	6.0	162.5	17.8	180.3	39.66	60.59	32.40	1.10
Fans **158%**, DHM 0%	39.9	0.546	31.3	86.2	0.0	256.2	0.0	256.2	39.90	59.58	32.39	1.09
Fans 0%, DHM **456%**	**39.9**	**0.546**	**31.3**	**0.0**	**27.3**	**0.0**	**81.1**	**81.1**	**37.5**	**58.22**	**30.01**	**-1.29**

For an outdoor ambient wet-bulb temperature, $T_{WB,Amb}$ = 24.0°C (T_{db} = 35.0°C, RH = 40.0%), combining 100% fresh air fan flowrates (137,000 m^3/hr, $PAT_{V/A}$=162.5) with 100% dehumidifier flowrates (15,000 m^3/hr as used in the trial for one deck), a pen wet-bulb temperature $T_{WB,pen}$ = 25.5°C is predicted. To maintain a pen wet-bulb temperature of $T_{WB,pen}$ = 25.5°C at an outdoor wet-bulb temperature of $T_{WB,Amb}$ = 24.0°C without the dehumidifier operating, the fan flowrate would need to be increased above the 100% fan flowrate value by 40% to 191,800 m^3/hr ($PAT_{V/A}$ = 227.7).

For ambient T_{WB} of 28.0^{O}C, combining 100% fans (137,000 m^3/hr, $PAT_{V/A}$ = 162.5) with 100% DHM (15,000 m^3/hr; PAT_{DHM} = 17.8) results in a pen T_{WB} of 29.19^{O}C. With only fans running, an increase of 49% of flowrate capacity (from 137,000 to 204,130 m^3/hr; or $PAT_{V/A}$ = 162.5 to $PAT_{V/A}$ = 241.9 respectively) is required to achieve the same outcome (i.e. a ΔT_{WB} = 1.19^{O}C).

The next case is for $T_{WB, AMb}$ = 30^{O}C. With 100% fans and 100% DHM, the pen T_{WB} = 31.06^{O}C. To achieve the same ΔT_{WB} = 1.06^{O}C with fans alone would require a fan flowrate of 212,350 m^3/hr (155% or $PAT_{V/A}$ = 252.6). For $T_{WB,Amb}$ = 31.3°C (T_{DB} = 39.9°C, RH = 54.6%), with 100% fans and 100% DHM to 100% the $T_{WB,pen}$ = 32.40°C.

To achieve the same outcome with fans alone would require increasing the fan flowrate by 58% to 216,460 m^3/hr ($PAT_{V/A}$ = 256.2). However, at $T_{WB,Amb}$ = 31.3°C using only a dehumidifier flowrate of 68,400 m^3/hr ($PAT_{V/A}$ = 81.1), it is possible to reduce the pen wet-bulb temperature to 30.0°C. This is a value that would be impossible to achieve with an increase in fresh air fan flowrate. The reduction of the pen wet bulb temperature to 30.0°C with the dehumidifier indicates that using a dehumidifier could have the potential to deliver pen conditions that would otherwise be impossible using conventional fresh air-based ventilation strategies (noting the increased outputs that would be needed).

Table 9: Additional fans and four DHM units (supplying 30,000 m³/hr). The T_{WB} difference between pen and outside condition is estimated for different fan and ambient conditions with four DHM units running in deck 3.

Operating Configurations	$T_{DB\ Amb.}$ (^{O}C)	$RH_{Amb.}$ (Const.)	$T_{WB\ Amb.}$ (^{O}C)	AEH_{Fans} (1/hr)	AEH_{DHM} (1/hr)	$PAT_{V/A}$ (m/hr)	PAT_{DHM} (m/hr)	PAT_{Total} (m/hr)	$T_{DB\ Pen}$ (^{O}C)	RH_{Pen} (Const.)	$T_{WB\ Pen}$ (^{O}C)	$\Delta T_{WB\ (Pen - Amb.)}$ (^{O}C)
Fans 100%, DHM 0%	35.0	0.400	24.0	54.7	0.0	162.5	0.0	162.5	35.34	48.10	26.04	2.04
Fans 100%, DHM 200%	35.0	0.400	24.0	54.7	12.0	162.5	35.6	198.0	34.73	45.58	25.00	1.00
Fans **206%**, DHM 0%	35.0	0.400	24.0	112.6	0.0	334.4	0.0	334.4	35.18	43.92	25.00	1.00
Fans 100%, DHM 0%	36.0	0.547	28.0	54.7	0.0	162.5	0.0	162.5	36.14	62.90	29.76	1.76
Fans 100%, DHM 200%	36.0	0.547	28.0	54.7	12.0	162.5	35.6	198.1	35.89	58.73	28.72	0.72
Fans **250%**, DHM 0%	36.0	0.547	28.0	136.5	0.0	405.6	0.0	405.6	36.06	57.98	28.72	0.72
Fans 100%, DHM 0%	35.9	0.652	30.0	54.7	0.0	162.5	0.0	162.5	36.06	73.21	31.62	1.62
Fans 100%, DHM 200%	35.9	0.652	30.0	54.7	12.0	162.5	35.6	198.0	35.91	68.29	30.59	0.59
Fans **280%**, DHM 0%	35.9	0.652	30.0	153.3	0.0	455.4	0.0	455.4	35.96	68.04	30.59	0.59
Fans 100%, DHM 0%	39.9	0.546	31.3	54.7	0.0	162.5	0.0	162.5	39.90	62.43	33.00	1.70
Fans 100%, DHM 200%	39.9	0.546	31.3	54.7	12.0	162.5	35.6	198.0	39.47	59.04	31.90	0.60
Fans **289%**, DHM 0%	39.9	0.546	31.3	158.1	0.0	469.6	0.0	469.6	39.90	57.32	31.90	0.60

Compared to the previous simulation, the above table shows the results of modelling an increase in the DHM PAT from 17.8 to 35.6 (15,000m^3/hr to 30,000 m^3/hr) for different ambient conditions.

For $T_{WB,Amb.}$ = 24.0°C (T_{db} = 35.0°C, RH = 40.0%), combining 100% fan flowrates (137,000 m^3/hr, $PAT_{V/A}$= 162.5) with 200% dehumidifier flowrates (30,000 m^3/hr or $PAT_{V/A}$ = 35.6) a $T_{WB,pen}$ = 25.0°C is predicted. To maintain $T_{WB,pen}$ = 25.0°C with outdoor wet-bulb temperature of $T_{WB,Amb}$ = 24.0°C, without the dehumidifier operating, the fan flowrate would need to be 206% to 282,220 m^3/hr ($PAT_{V/A}$ = 334.4).

With increasing ambient air properties (T_{DB} = 36.0^{O}C, RH = 54.7% and T_{WB} = 28.0^{O}C), the predicted ΔT_{WB} is 0.72^{O}C between pen area and ambient with an operating configuration of 100% fan and 200% DHM. To achieve the same ΔT_{WB} without the DHM operating, the fresh air fan flowrate capacity would need to be 250% (from 137,000 m^3/hr to 342,500 m^3/hr; or $PAT_{V/A}$ = 162.5 to 405.6)).

At T_{WB} = 30.0^{O}C (T_{DB} = 35.9^{O}C, RH = 65.2%) the predicted pen T_{WB} = 30.59^{O}C with a 100% fan and 200% DHM operating configuration. The ΔT_{WB} = 0.59^{O}C. To achieve the same ΔT_{WB} without the use of DHM, the fan flowrate would need to be increased to 280% to 383,600 m^3/hr ($PAT_{V/A}$ = 455.4).

For T_{WB} = 31.3^{O}C (T_{DB} = 39.9^{O}C, RH = 54.6%) the predicted pen T_{WB} = 31.9^{O}C with a 100% fan and 200% DHM operating configuration. To achieve the same ΔT_{WB} (ΔT_{WB} = 0.6^{O}C) without DHM, the fan flowrate would need to be increased to 289% (395,930 m^3/hr; $PAT_{V/A}$ = 469.6).

Table 10: Additional fans and six DHM units (supplying 45,000 m³/hr). The T_{WB} difference between pen and outside condition is estimated for different fan and ambient conditions with six DHM units running in deck 3.

Operating Configurations	$T_{DB\ Amb.}$ (°C)	$RH_{Amb.}$ (Const.)	$T_{WB\ Amb.}$ (°C)	AEH_{Fans} (1/hr)	AEH_{DHM} (1/hr)	$PAT_{V/A}$ (m/hr)	PAT_{DHM} (m/hr)	PAT_{Total} (m/hr)	$T_{DB\ Pen}$ (°C)	RH_{Pen} (Const.)	$T_{WB\ Pen}$ (°C)	$\Delta T_{WB\ (Pen\ -\ Amb.)}$ (°C)
Fans 100%, DHM 0%	35.0	0.400	24.0	54.7	0.0	162.5	0.0	162.5	35.34	48.10	26.04	2.04
Fans 100%, DHM 300%	35.0	0.400	24.0	54.7	18.0	162.5	53.4	215.8	34.50	44.58	24.60	0.60
Fans **337%**, DHM 0%	35.0	0.400	24.0	184.4	0.0	547.9	0.0	547.9	35.12	42.39	24.60	0.60
Fans 100%, DHM 0%	36.0	0.547	28.0	54.7	0.0	162.5	0.0	162.5	36.14	62.90	29.76	1.76
Fans 100%, DHM 300%	36.0	0.547	28.0	54.7	18.0	162.5	53.4	215.9	35.80	57.15	28.32	0.32
Fans **565%**, DHM 0%	36.0	0.547	28.0	309.0	0.0	917.9	0.0	917.9	36.03	56.15	28.32	0.32
Fans 100%, DHM 0%	35.9	0.652	30.0	54.7	0.0	162.5	0.0	162.5	36.06	73.21	31.62	1.62
Fans 100%, DHM 300%	35.9	0.652	30.0	54.7	18.0	162.5	53.4	215.8	35.86	66.43	30.19	0.19
Fans **832%**, DHM 0%	35.9	0.652	30.0	455.1	0.0	1351.9	0.0	1351.9	35.92	66.16	30.19	0.19
Fans 100%, DHM 0%	39.9	0.546	31.3	54.7	0.0	162.5	0.0	162.5	39.90	62.43	33.00	1.70
Fans 100%, DHM 300%	39.9	0.546	31.3	54.7	18.0	162.5	53.4	215.8	39.30	57.72	31.47	0.17
Fans **1007%**, DHM 0%	39.9	0.546	31.3	550.9	0.0	1636.5	0.0	1636.5	39.90	55.38	31.47	0.17

This simulation studies an increase in DHM PAT (from 17.8 to 53.4 or from 15,000m^3/hr to 45,000 m^3/hr), with the table above summarising the changes in T_{WB} for different ambient conditions.

At $T_{WB,Amb.}$ = 24.0°C (T_{db} = 35.0°C, RH = 40.0%), combining 100% fresh air fan flowrates (137,000 m^3/hr, $PAT_{V/A}$=163) with 300% dehumidifier flowrates (45,000 m^3/hr or $PAT_{V/A}$ = 53.4) will result in a predicted $T_{WB,pen}$ of 24.6°C. To maintain a pen wet-bulb temperature of $T_{WB,pen}$ = 24.6°C, without the dehumidifier operating, the fan flowrate would need to be increased above the 100% fan flowrate value by 237% to 461,690 m^3/hr ($PAT_{V/A}$ =547.9).

At T_{WB} = 28.0^{O}C (T_{DB} = 36.0^{O}C, RH = 54.7%), the predicted ΔT_{WB} is 0.32^{O}C between the pen area and ambient air with an operating configuration of 100% fans and 300% DHM. To achieve the same ΔT_{WB} without the DHM operating, the fresh air fan flowrate capacity would need to increase to 565% (from 137,000 m^3/hr to 774,050 m^3/hr; $PAT_{V/A}$ = 162.5 to 917.9).

At T_{WB} = 30.0^{O}C (T_{DB} = 35.9^{O}C, RH = 65.2%), the predicted pen T_{WB} = 30.19^{O}C with 100% fan and 300% DHM operating. The ΔT_{WB} between pen and ambient is 0.19^{O}C. To achieve the same ΔT_{WB} without using the DHM, the fan flowrate would need to be increased above the 100% fan flowrate value by 732% to 1,139,840 m^3/hr ($PAT_{V/A}$ = 1351.9).

At T_{WB} = 31.3^{O}C (T_{DB} = 39.9^{O}C, RH = 54.6%), the predicted pen T_{WB} = 31.47^{O}C with a 100% fan and 300% DHM operating configuration. To achieve the same ΔT_{WB} = 0.17^{O}C without DHM, the fresh air fan would need to operate in the capacity of 1007% (1,379,590 m^3/hr; $PAT_{V/A}$ = 1636.5).

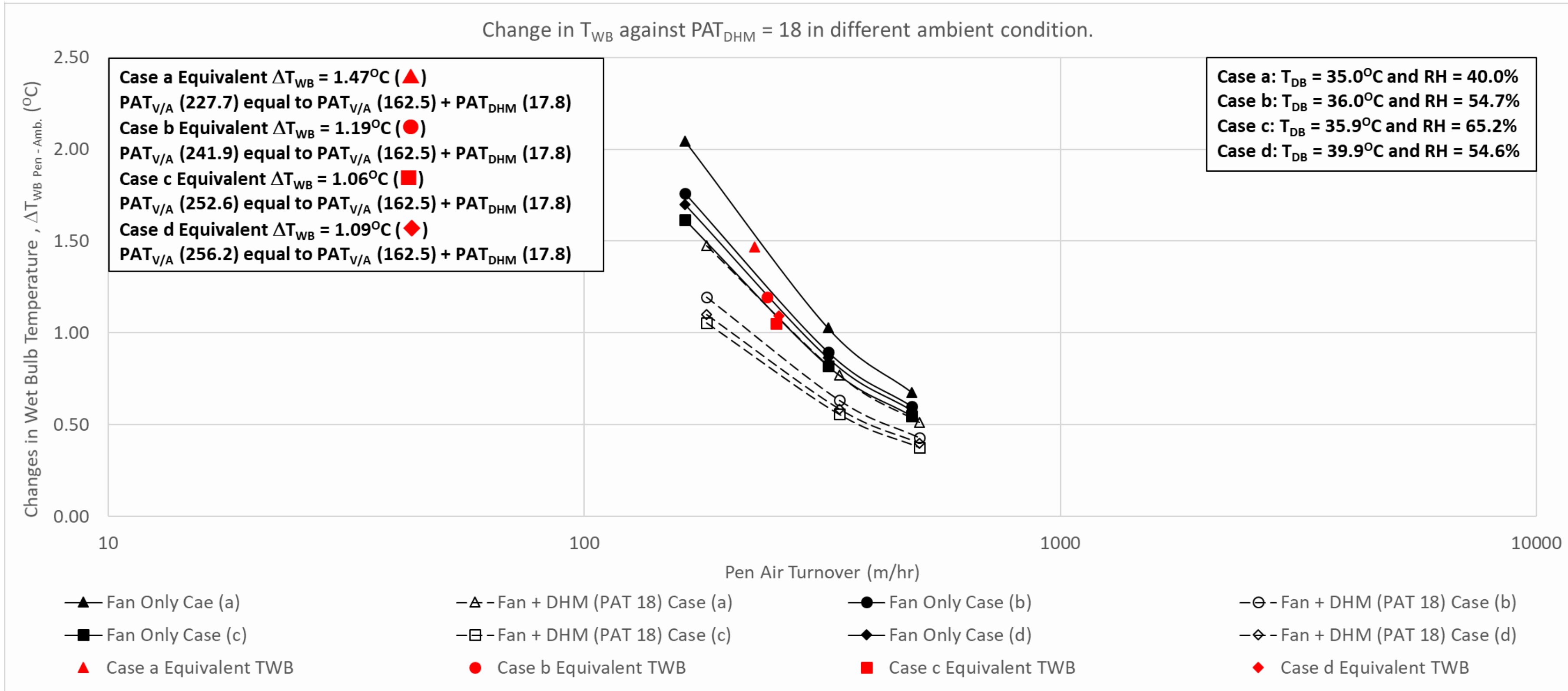


Figure 45: Change in T_{WB} against PATDHM = 18 in different ambient conditions

The simulation exercise has considered four different ambient conditions. These are shown in the above plot on the top right corner.

This above plot shows the change in T_{WB} (ΔT_{WB}) – essentially representing the difference between pen T_{WB} and ambient air T_{WB} – for a PAT_{DHM} = 18. The larger the ΔT_{WB}, the larger the difference between the pen area T_{WB} and the ambient air T_{WB}.

With increasing fresh air PAT (black solid lines), the ΔT_{WB} reduced accordingly. This implied that with increasing fresh air PAT, the pen T_{WB} is closer to ambient T_{WB}. When DHM units are operating alongside the fresh air fans (PAT_{DHM} + $PAT_{V/A}$), the ΔT_{WB} are lower in all cases (black dashed lines) compared to those with only fresh air fans operating.

The red symbols shown are the equivalent ΔT_{WB} with 100% fans and 100% DHM in all four cases.

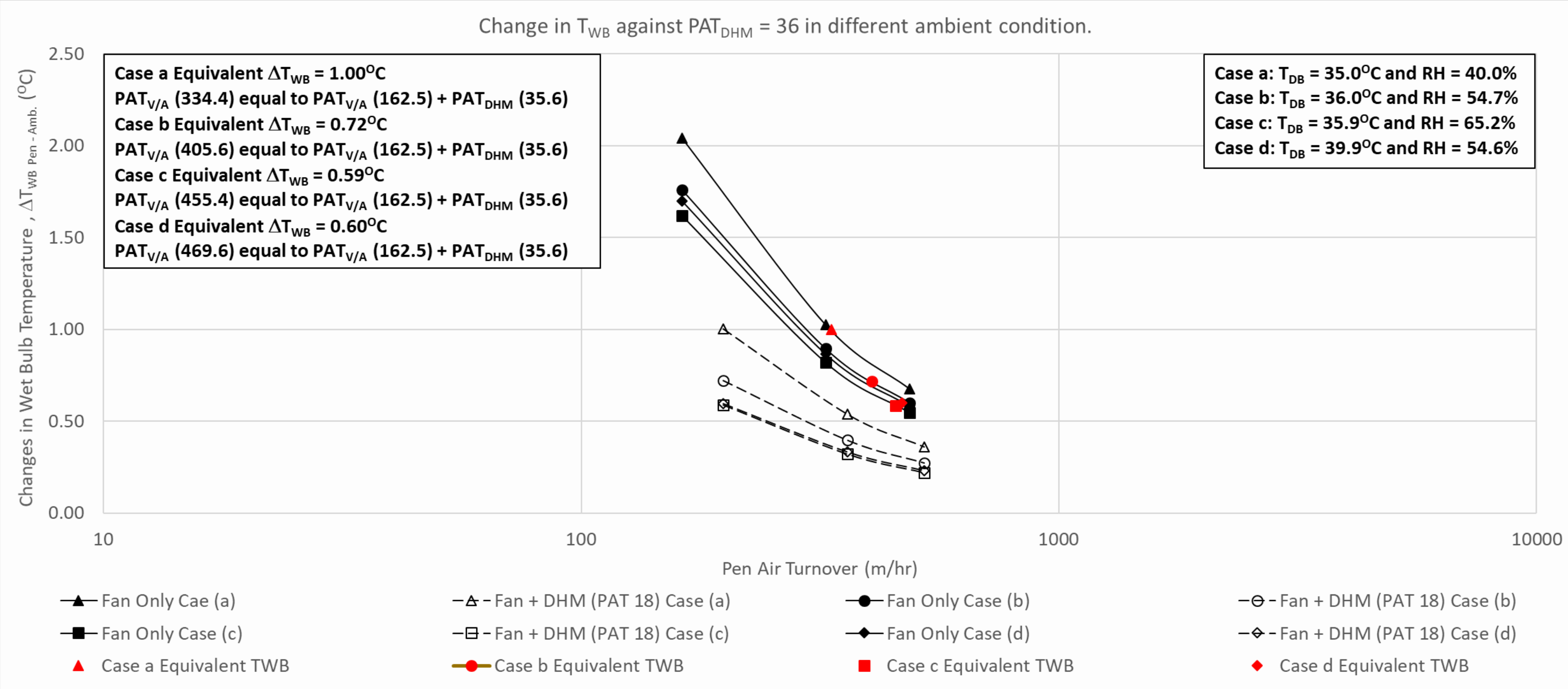


Figure 46: Change in TWB against PATDHM = 36 in different ambient conditions

The above plot is a similar representation as in figure 14, however the PAT_{DHM} in this case has been doubled to equal 36.

The plot shows that when DHM units are operating side by side with fresh air fans (PAT_{DHM} + $PAT_{V/A}$), the ΔT_{WB} are lower in all cases (black dashed lines) compared to those with only fresh air fans operating (black solid lines). They are also lower than the all cases shown in the PAT_{DHM} = 18 scenario. The red symbols are the equivalent ΔT_{WB} with 100% fans and 200% DHM (PAT_{DHM} = 36) in all four cases. The PAT required by fresh air fans only increases to match with the same ΔT_{WB} by 100% fans and 200% DHM in all four cases. Noting that those red symbols shifted to right accordingly compared to the previous plot (PAT_{DHM} = 18).

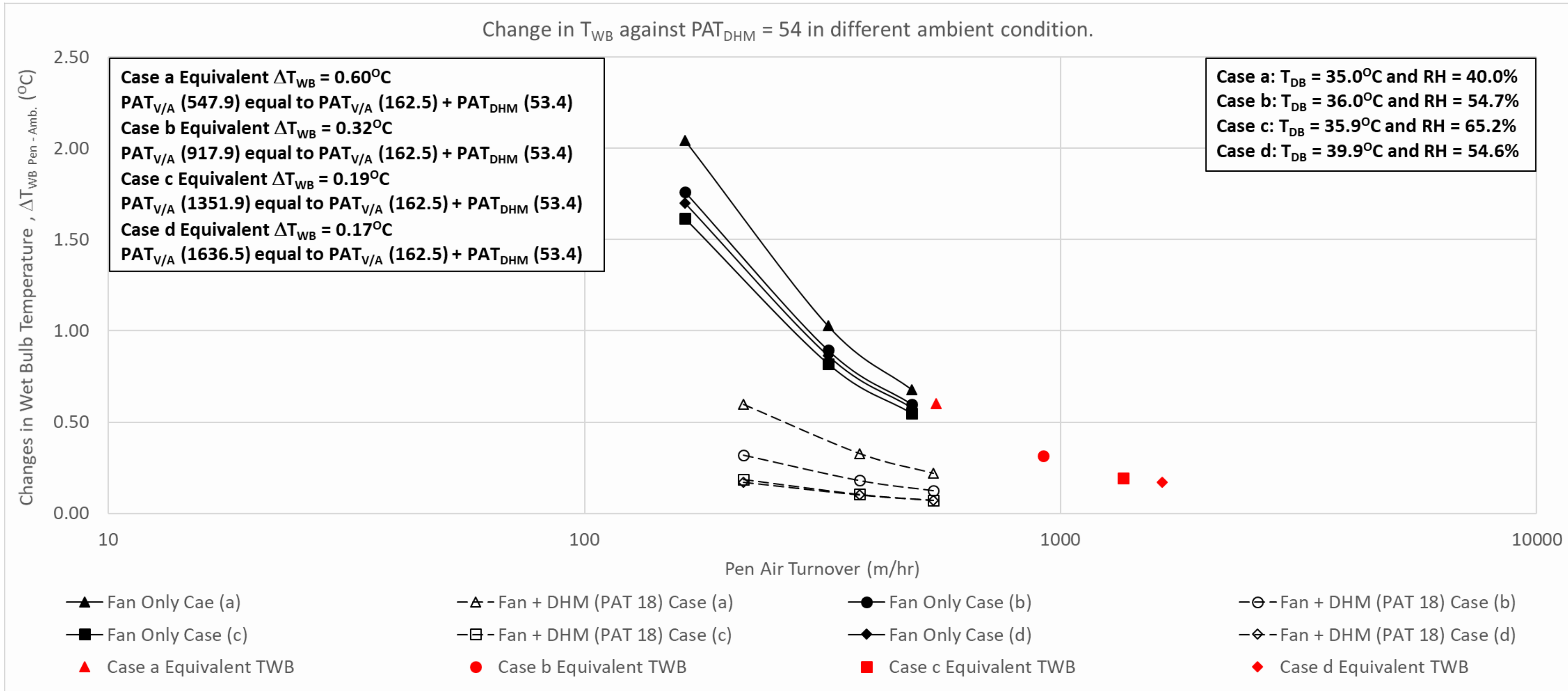


Figure 47: Change in T_{WB} against $PAT_{DHM\ V/A}$ = 54 in different ambient conditions

This plot presents the same representation as the above figures, except with an increase in the DHM PAT to 54 (a 300% increase).

The increase in the DHM PAT to 54 results in a further reduction in the ΔT_{WB} for all four cases compared to those in the PAT_{DHM} = 36 scenario. To match the equivalent change in T_{WB} between pen and fresh air ambient (ΔT_{WB}), the $PAT_{V/A}$ for fan only increases to a range from 547.9 to 1636.5.

Summary of simulation outcomes

Theoretical modelling of the thermal conditions in the pen was undertaken to assess the impact that the dehumidifiers employed in the trial would potentially have on sheep comfort and stress levels. The thermal modelling of the pen conditions included the latent and sensible heat load from the sheep and explored a range of outdoor environmental wet bulb and dry bulb temperatures, as well as a range of fresh air and dehumidifier flowrates. The modelling approach combined a model of the dehumidifier (developed as a result of the trial) with an animal heat load model developed from existing data and literature.

For an outdoor ambient wet-bulb temperature, $T_{WB,Amb}$ = 24.0°C (T_{DB} = 35.0°C, RH = 40.0%), combining 100% fresh air fan flowrates (137,000 m^3/hr, $PAT_{V/A}$=162.5) with 100% dehumidifier flowrates (15,000 m^3/hr as used in the trial for one deck), a pen wet-bulb temperature $T_{WB,pen}$ = 25.5°C is predicted. To maintain a pen wet-bulb temperature of $T_{WB,pen}$ = 25.5°C and an outdoor wet-bulb temperature of $T_{WB,Amb}$ = 24.0°C without the dehumidifier operating, the fan flowrate would need to be increased above the 100% fan flowrate value by 40% to 191,800 m^3/hr ($PAT_{V/A}$ =227.5).

At an outdoor ambient wet-bulb temperature of $T_{WB,Amb}$ = 31.3°C (T_{DB} = 39.9°C, RH = 54.6%), the impact of adding 100% dehumidifier flowrate to 100% fan flowrate is the same as increasing the fresh air fan flowrate by 58% to 216,460 m^3/hr ($PAT_{V/A}$ = 256.7). However, at $T_{WB,Amb}$ = 31.3°C, with 100% fan flowrate and 100% dehumidifier flowrate, the pen wet-bulb temperature is $T_{WB,pen}$ = 32.4°C. At an outdoor ambient wet-bulb temperature of $T_{WB,Amb}$ = 31.3°. Using a dehumidifier flowrate of 68,400 m^3/hr ($PAT_{V/A}$ = 81.1) it is possible to reduce the pen wet-bulb temperature to 30.0°C – a value that would be impossible to achieve with an increase in fresh air fan flowrate. The reduction of the pen wet bulb temperature to 30.0°C with the dehumidifier indicates that the use of the dehumidifier has potentially of delivering improved animal comfort in conditions that would otherwise be impossible using conventional fresh air-based ventilation strategies.

These findings also captured the changing relative value of dehumidification compared to fans on the wet bulb temperature outcomes in different ambient conditions. It was shown that the use of fans had a stronger relative value in reducing wet bulb temperature increases when ambient conditions were lower, that dehumidification had a stronger effect at higher wet bulb temperatures, and that a combination of the two was most effective within an intermediate range.

In the process of the thermal modelling of the pen, it was also established that the heat load for sheep as reported in literature ranged from 1.6 to 3.2 W/kg. The upper value of 3.2 W/kg was used for all the analysis reported. To model the latent heat load fraction from the sheep, a model from literature was used where the latent heat load fraction is solely a function of the dry-bulb temperature. Although the latent heat load fraction is expected to be a function of the dry *and* wet bulb temperatures, no data in literature is available to readily construct such a relationship. Present thermal stress models for sheep rely primarily on the wet-bulb temperature, however drawing a parallel to human studies where significantly more data exists, thermal stress should depend both on the wet bulb *and* dry temperature.

In order to enhance future predictive thermal modelling of pen conditions, improved and well-validated thermal models of the sheep sensible and latent loads, as well as thermal stress level prediction, will be required. In the field trial and the theoretical pen thermal modelling, cooling was provided with a dehumidifier that created large latent cooling with a small sensible cooling load. Preliminary exploratory modelling indicates that for a given condition and given electrical power input, there exists an optimal ratio of sensible and latent cooling will deliver optimal animal comfort; such optimal conditions are not solely using dehumidification, but a combination of dehumidification and regular sensible cooling. The optimisation of the dehumidification and cooling system is subject to improvements in the knowledge of the animal thermal load and thermal stress modelling.

Statistical regression modelling

Following from the simulations described above, a further statistical regression model was developed to quantify and predict deck wet bulb temperature for combinations of ambient wet bulb temperature and controls (fan ventilation and dehumidifiers). This regression model allowed prediction of deck conditions for combinations that were not specifically trialled or modelled, thereby allowing the impact of dehumidifiers from across the range of expected voyage conditions to be explored.

It is important to note that this regression analysis only looked at wet bulb temperature, which is a simplification from the simulations and the range of variables they considered. However, predicting wet bulb temperature was supported because of the common use of the measure within the livestock export industry, and because it enabled use of the HSRA Voluntary Observing Ship data (where wet bulb temperatures were recorded) to allow the subsequent estimate of voyage risk.

The statistical model obtained from observed values of dehumidification and fan identified key predictors and quantified the impact on deck wet bulb mathematically. The statistical model is described below as:

Pen T_{WB} = Intercept + Ambient T_{WB} + Fan PAT + Dehumidifier PAT

Variable	Estimate	Std. Error	t value	Pr(>|t|)
Intercept	13.826	2.247	6.152	< 0.001 ***
Ambient T_{WB}	0.819	0.071	11.568	< 0.001 ***
Fans PAT	-0.046	0.004	-10.271	<0.001 ***
Dehumidifier PAT	-0.045	0.029	-1.546	0.131

The statistical regression model found that ambient wet bulb temperature and ventilation fan rate were significant predictors of deck wet bulb temperature. While dehumidification was not found to be a significant predictor of deck wet bulb temperature, this is most likely due to the small operating range of the dehumidifiers (maximum dehumidifier $PAT_{V/A}$ was only 17.8). For the per unit effect of $PAT_{V/A}$ under this configuration, dehumidification had a similar effect on reducing deck wet bulb temperature to ventilation (-0.045°C versus -0.046°C for every extra unit of $PAT_{V/A}$ for dehumidifier and fans respectively). Only linear relationships between predictor and outcome variables were significant and no interaction terms were significant, which supports the use of the statistical model to predict deck conditions from outside the range of individual variable values (e.g. ambient wet bulb temperature) studied with confidence.

The statistical model predicted an average reduction in deck wet bulb temperature of 0.8°C[6] from operating the dehumidifier at the maximum $PAT_{V/a}$ used in the trial of 17.8. The predicted performance across a range of combinations of ventilation fans, dehumidifiers and ambient temperatures are presented in Figure 47 and Table 11.

[6] 100% Dehumidifier PAT is 17.8, so regression modelling predicts 17.8 x -0.045 = 0.80°C reduction from the use of dehumidification.

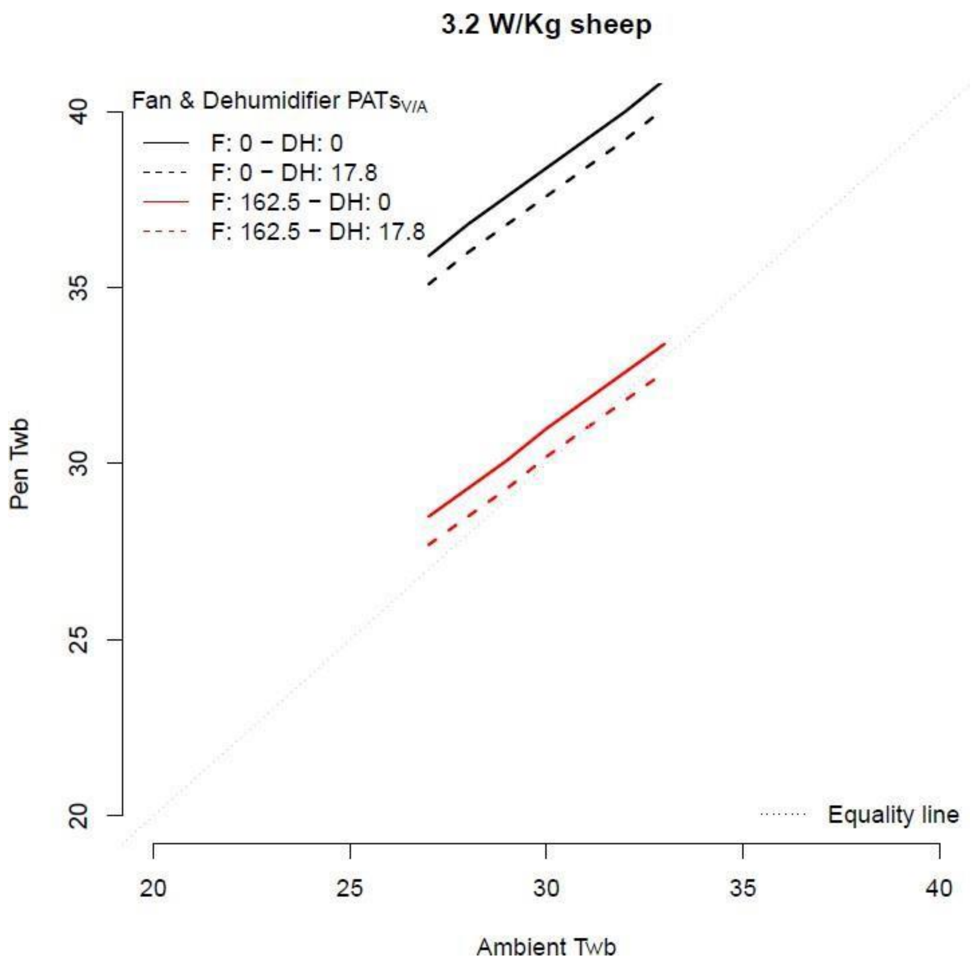


Figure 47: Predicted pen T_{WB} reduction compared to ambient T_{WB} for a pen stocked to ASEL v2.3 with 40kg sheep and under maximum dehumidification ($PAT_{V/A}$ = 17.8/Hr is 100%) and under varying fan ventilation rates ($PAT_{V/A}$ = 162.5/Hr is 100%) and ambient wet bulb temperature. Note that the grey dotted line in the chart represents the equality line where the ambient T_{WB} = deck T_{WB}.

Table 11: Predicted deck T_{WB} by ambient T_{WB} and fan and dehumidifier operating rates (from regression model)

T_{WB} Ambient	Fans AEH	Dehumidifier AEH	Fans $PAT_{V/A}$	Dehumidifier $PAT_{V/A}$	T_{WB} Pen	T_{WB} Pen Delta
27	0.0	0.0	0.0	0.0	35.9	8.9
28	0.0	0.0	0.0	0.0	36.8	8.8
29	0.0	0.0	0.0	0.0	37.6	8.6
30	0.0	0.0	0.0	0.0	38.4	8.4
31	0.0	0.0	0.0	0.0	39.2	8.2
32	0.0	0.0	0.0	0.0	40	8
33	0.0	0.0	0.0	0.0	40.9	7.9
27	27.3	0.0	81.3	0.0	32.2	5.2
28	27.3	0.0	81.3	0.0	33	5

29	27.3	0.0	81.3	0.0	33.9	4.9
30	27.3	0.0	81.3	0.0	34.7	4.7
31	27.3	0.0	81.3	0.0	35.5	4.5
32	27.3	0.0	81.3	0.0	36.3	4.3
33	27.3	0.0	81.3	0.0	37.1	4.1
27	54.7	0.0	162.5	0.0	28.5	1.5
28	54.7	0.0	162.5	0.0	29.3	1.3
29	54.7	0.0	162.5	0.0	30.1	1.1
30	54.7	0.0	162.5	0.0	31	1
31	54.7	0.0	162.5	0.0	31.8	0.8
32	54.7	0.0	162.5	0.0	32.6	0.6
33	54.7	0.0	162.5	0.0	33.4	0.4
27	0.0	3.0	0.0	8.9	35.5	8.5
28	0.0	3.0	0.0	8.9	36.4	8.4
29	0.0	3.0	0.0	8.9	37.2	8.2
30	0.0	3.0	0.0	8.9	38	8
31	0.0	3.0	0.0	8.9	38.8	7.8
32	0.0	3.0	0.0	8.9	39.6	7.6
33	0.0	3.0	0.0	8.9	40.5	7.5
27	27.3	3.0	81.3	8.9	31.8	4.8
28	27.3	3.0	81.3	8.9	32.6	4.6
29	27.3	3.0	81.3	8.9	33.5	4.5
30	27.3	3.0	81.3	8.9	34.3	4.3
31	27.3	3.0	81.3	8.9	35.1	4.1
32	27.3	3.0	81.3	8.9	35.9	3.9
33	27.3	3.0	81.3	8.9	36.7	3.7
27	54.7	3.0	162.5	8.9	28.1	1.1
28	54.7	3.0	162.5	8.9	28.9	0.9
29	54.7	3.0	162.5	8.9	29.7	0.7
30	54.7	3.0	162.5	8.9	30.6	0.6
31	54.7	3.0	162.5	8.9	31.4	0.4
32	54.7	3.0	162.5	8.9	32.2	0.2
33	54.7	3.0	162.5	8.9	33	0
27	0.0	6.0	0.0	17.8	35.1	8.1
28	0.0	6.0	0.0	17.8	36	8
29	0.0	6.0	0.0	17.8	36.8	7.8
30	0.0	6.0	0.0	17.8	37.6	7.6
31	0.0	6.0	0.0	17.8	38.4	7.4
32	0.0	6.0	0.0	17.8	39.2	7.2
33	0.0	6.0	0.0	17.8	40.1	7.1
27	27.3	6.0	81.3	17.8	31.4	4.4
28	27.3	6.0	81.3	17.8	32.2	4.2
29	27.3	6.0	81.3	17.8	33.1	4.1
30	27.3	6.0	81.3	17.8	33.9	3.9

31	27.3	6.0	81.3	17.8	34.7	3.7
32	27.3	6.0	81.3	17.8	35.5	3.5
33	27.3	6.0	81.3	17.8	36.3	3.3
27	54.7	6.0	162.5	17.8	27.7	0.7
28	54.7	6.0	162.5	17.8	28.5	0.5
29	54.7	6.0	162.5	17.8	29.3	0.3
30	54.7	6.0	162.5	17.8	30.2	0.2
31	54.7	6.0	162.5	17.8	31	0
32	54.7	6.0	162.5	17.8	31.8	-0.2
33	54.7	6.0	162.5	17.8	32.6	-0.4

Table 11 shows that the combined effect of ventilation fans and dehumidification on deck T_{WB} reduction compared to ambient can be summarised as:

1. Modelling predicts the elevation in deck wet bulb temperature for a theoretically stocked deck, without forced fan ventilation or dehumidification operating at ambient T_{WB} of 28.0°C to be 8.8°C (i.e. when the ambient T_{WB} = 28.0°C, deck T_{WB} = 36.8°C). The addition of:

    a. Fans operating at 100% capacity ($PAT_{V/A}$ = 162.5) without dehumidification will reduce the elevation in deck T_{WB} to 29.3°C; a relative increase of 1.3°C above ambient T_{WB} (a reduction of 7.5°C in the elevation due solely to the use of fans alone).

    b. Fans operating at 100% ($PAT_{V/A}$ = 162.5) and dehumidifiers operating at 100%($PAT_{V/A}$ = 17.8) will reduce the deck T_{WB} to 28.5°C ; a relative increase of 0.5°C above ambient T_{WB} (a further reduction of 0.8°C in the elevation compared to the reduction achieved through the use of fans alone).

2. Modelling predicts the elevation in deck wet bulb temperature for a theoretically stocked deck without forced fan ventilation or dehumidification operating at ambient T_{WB} of 30.0°C to be 8.4°C (i.e. when the ambient T_{WB} = 30.0°C, deck T_{WB} = 38.4°C). The addition of:

    a. Fans operating at 100% capacity ($PAT_{V/A}$ = 162.5) without dehumidification will reduce the elevation in deck T_{WB} to 31.0°C; a relative increase of 1.0°C above ambient T_{WB} (a reduction in the elevation of 7.4°C from the use of fans alone).

    b. Fans operating at 100% ($PAT_{V/A}$ = 162.5) and dehumidifiers operating at 100% ($PAT_{V/A}$ = 17.8) will reduce the deck T_{WB} to 30.2°C; a relative increase of 0.2°C above ambient T_{WB}. That is, the use of dehumidification reduces the deck wet bulb temperature by 0.8°C compared to the use of fans alone.

Analysis of risk

An assessment of the impact that the use of dehumidification could have on risk was conducted to apply the simulated outcomes and statistical analysis in a potentially more accessible form.

Within this assessment, risk was quantified as the probability that the daily maximum deck T_{WB} would exceed 28°C and 30°C in voyages in the Northern Hemisphere along the livestock export route to the Middle East for each month of the year. This was calculated by analysing the Voluntary Observing Ship (VOS) dataset, which is also used within the HSRA model.

The VOS-observed deck T_{WB} was adjusted for the inclusion of sheep by the addition of the deck ΔT_{WB} obtained from the statistical regression model above. Arbitrary heat stress cut points of T_{WB} = 28°C and T_{WB} = 30°C (daily maximum pen conditions) were used. The probability of deck condition exceeding these cut-points on any given Middle Eastern voyage through northern waters was calculated. The differences between the probability of exceeding these cut-points between various control combinations (e.g. with or without dehumidification) provides a measure of effectiveness of individual controls on actual heat stress risk reduction.

Risk-analysis results are presented in Table 12 and Figure 48 for 28+°C risk and in Table 13 and Figure 49 for 30+°C risk.

Table 12: Daily risk of wet bulb temperature of 28+°C for ambient conditions, stocked decks with a ventilation fan $PAT_{V/A}$ = (100%), and stocked decks with a ventilation fan $PAT_{V/A}$ =162.5 (100%) and dehumidification $PAT_{V/A}$ = 17.8 (100%) for vessels in the northern hemisphere along the sheep live export routes to the Middle East by month.

Calendar Month	Ambient risk 28+°C	Deck risk 28+°C	Deck risk 28+°C with Fan PAT 162.5	Deck risk 28+°C with Fan PAT 162.5 + DHM. PAT 17.8	Absolute risk reduction from addition of DHM
1	0.8%	76.8%	2.6%	1.0%	1.6%
2	1.4%	84.2%	4.1%	1.8%	2.3%
3	3.3%	95.9%	8.1%	4.3%	3.9%
4	8.7%	99.5%	21.3%	10.9%	10.4%
5	22.5%	99.9%	49.8%	30.7%	19.1%
6	26.8%	99.9%	50.5%	32.5%	18.1%
7	21.7%	100.0%	33.5%	24.3%	9.2%
8	18.4%	99.9%	27.1%	20.2%	6.9%
9	16.4%	100.0%	25.2%	18.4%	6.9%
10	11.3%	100.0%	21.9%	13.5%	8.4%
11	2.3%	97.8%	7.6%	3.4%	4.2%
12	1.6%	83.4%	4.1%	2.3%	1.8%

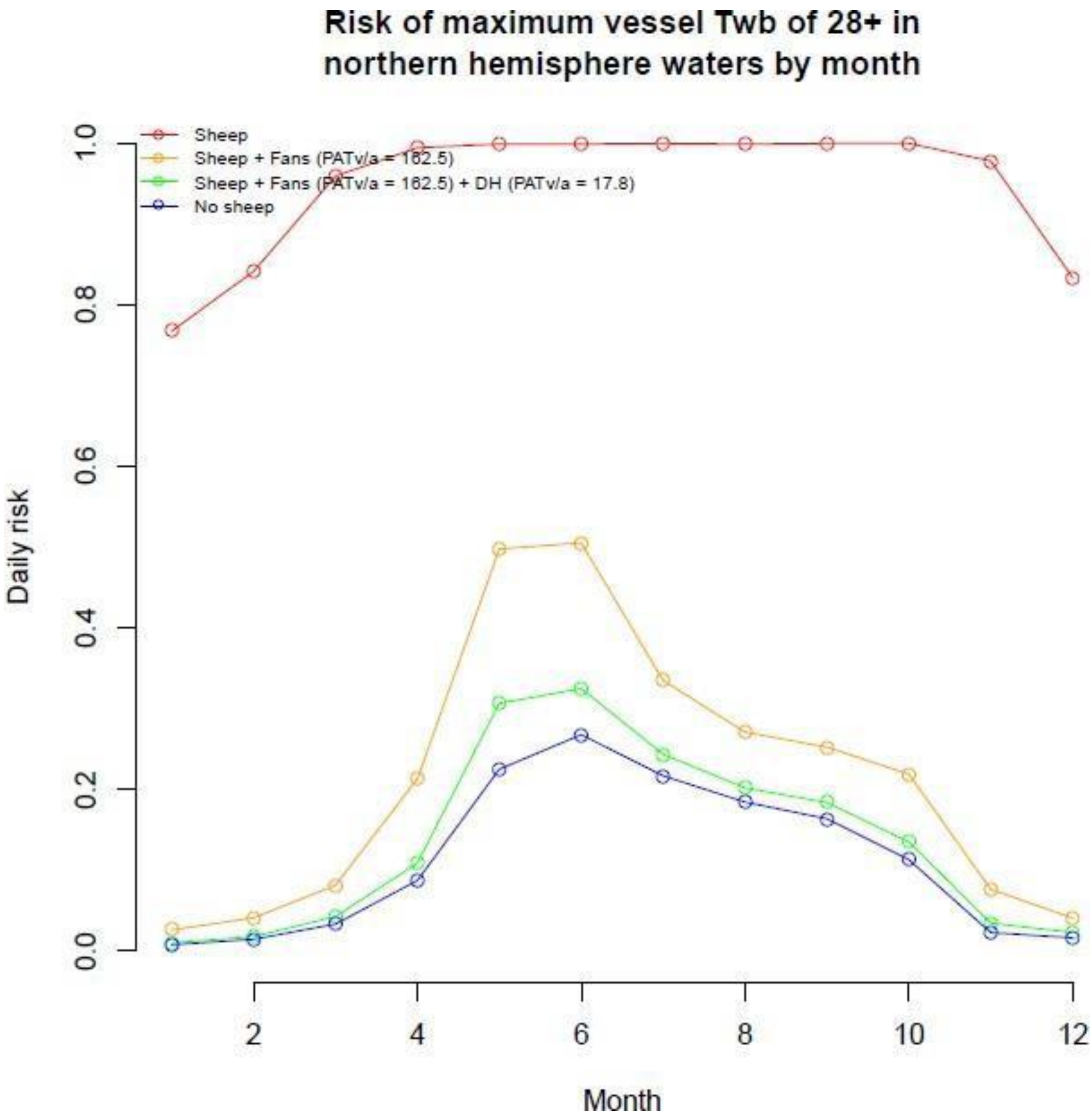


Figure 48: Daily risk of maximum deck wet-bulb temperatures of 28+°C for ambient conditions, unstocked decks, stocked unventilated decks, stocked ventilated decks with $PAT_{V/A}$ = 162.5 (100%) and stocked ventilated decks with $PAT_{V/A}$ = 162.5 (100%) and dehumidification $PAT_{V/A}$ = 17.8 (100%) for vessels in the northern hemisphere on sheep export routes to the Middle East by month.

Table 13: Daily risk of wet bulb temperature of 30+°C for ambient conditions, stocked decks with a ventilation fan $PAT_{V/A}$ = 162.5, stocked decks with ventilation fan $PAT_{V/A}$ = 162.5 and dehumidification $PAT_{V/A}$ = 17.8 for vessels in the northern hemisphere on sheep export routes to the Middle East by month.

Calendar Month	Ambient risk 30+°C	Deck risk 28+°C	Deck risk 30+°C with Fan PATv/a = 162.5	Deck risk 30+°C with Fan PATv/a = 162.5 + DHM. PATv/a = 17.8	Absolute risk reduction from addition of DHM
1	0.0%	58.8%	0.2%	0.0%	0.2%
2	0.1%	68.0%	0.4%	0.2%	0.2%
3	0.8%	84.9%	1.9%	0.9%	1.0%
4	2.2%	97.0%	4.0%	2.5%	1.4%
5	4.1%	99.9%	10.9%	5.6%	5.3%
6	8.1%	99.9%	15.4%	9.6%	5.8%
7	11.4%	99.8%	17.0%	13.1%	3.9%
8	11.0%	99.4%	13.8%	11.8%	2.0%
9	5.5%	99.8%	10.1%	7.0%	3.1%
10	3.2%	99.2%	6.2%	3.8%	2.5%
11	0.3%	91.3%	1.0%	0.6%	0.4%
12	0.2%	67.4%	0.7%	0.2%	0.5%

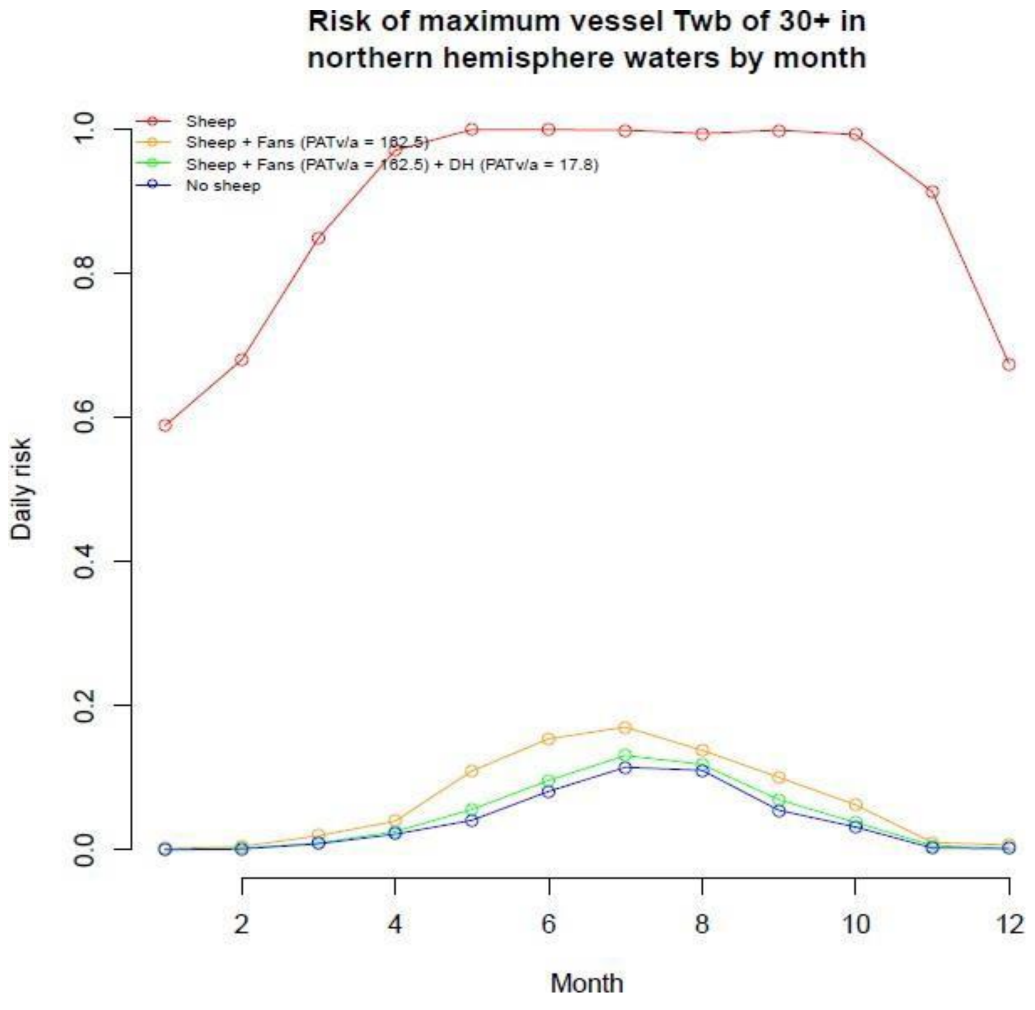


Figure 49: Daily risk of maximum pen wet-bulb temperatures of 30+°C for ambient conditions, unstocked pens, stocked unventilated pens, stocked ventilated decks with $PAT_{V/A}$= 162.5 (100%) and stocked ventilated pens with $PAT_{V/A}$ = 162.5 (100%) and dehumidification with $PAT_{V/A}$ = 17.8 (100%) for vessels in the northern hemisphere on sheep export routes to the Middle East by month.

Extrapolative modelling

Theoretical impacts of extra dehumidification on deck conditions were explored using the combined animal-environmental thermodynamic model developed in this project. This model was extended by increasing (theoretical) dehumidifier performance beyond the maximum capacity measured during the static trial (i.e. beyond a maximum $PAT_{V/A}$ = 17.8). This 'what if' simulation was undertaken by applying the assumption that greater output and/or more dehumidification units per deck may be feasible. The animal-environmental model examined dehumidifier performance increases of 200% ($PAT_{V/A}$ = 35.6) and 300% ($PAT_{V/A}$ = 53.4) over the current trial maximum.

The statistical predictive model was rebuilt to explore the impact of extra dehumidification on deck conditions (in order to re-assess non-linearity and interaction). In this model, dehumidification and interaction terms between ambient wet bulb and ventilation fan $PAT_{V/A}$ and between ventilation fan $PAT_{V/A}$ and dehumidifier $PAT_{V/A}$ became statistically significant. The coefficients for significant interaction terms were small, suggesting only minor non-linearity was present and that, within reason, reliable predictions of performance can be made. The final model is outlined below, and the results are shown in Figure 50.

Deck T_{WB} = Intercept + Ambient T_{WB} + Fan $PAT_{V/A}$ + Dehumidifier $PAT_{V/A}$ + Ambient T_{WB}:Fan $PAT_{V/A}$ + Fan $PAT_{V/A}$:Dehumidifier $PAT_{V/A}$

Variable	Estimate	Std. Error	t value	Pr(>|t|)	
Intercept	21.950	2.740	8.012	<0.001	***
Ambient T_{WB}	0.587	0.090	6.591	<0.001	***
Fans $PAT_{V/A}$	-0.116	0.022	-5.359	<0.001	***
Dehumidifier $PAT_{V/A}$	-0.177	0.016	-11.239	<0.001	***
Ambient T_{WB}:Fan $PAT_{V/A}$	0.002	0.0007	2.894	0.005	**
Fan $PAT_{V/A}$:Dehumidifier $PAT_{V/A}$	0.001	0.001	8.135	<0.001	***

This was used to predict deck conditions across the range of ambient wet bulb temperatures likely on northern summer sheep trade routes to the Middle East.

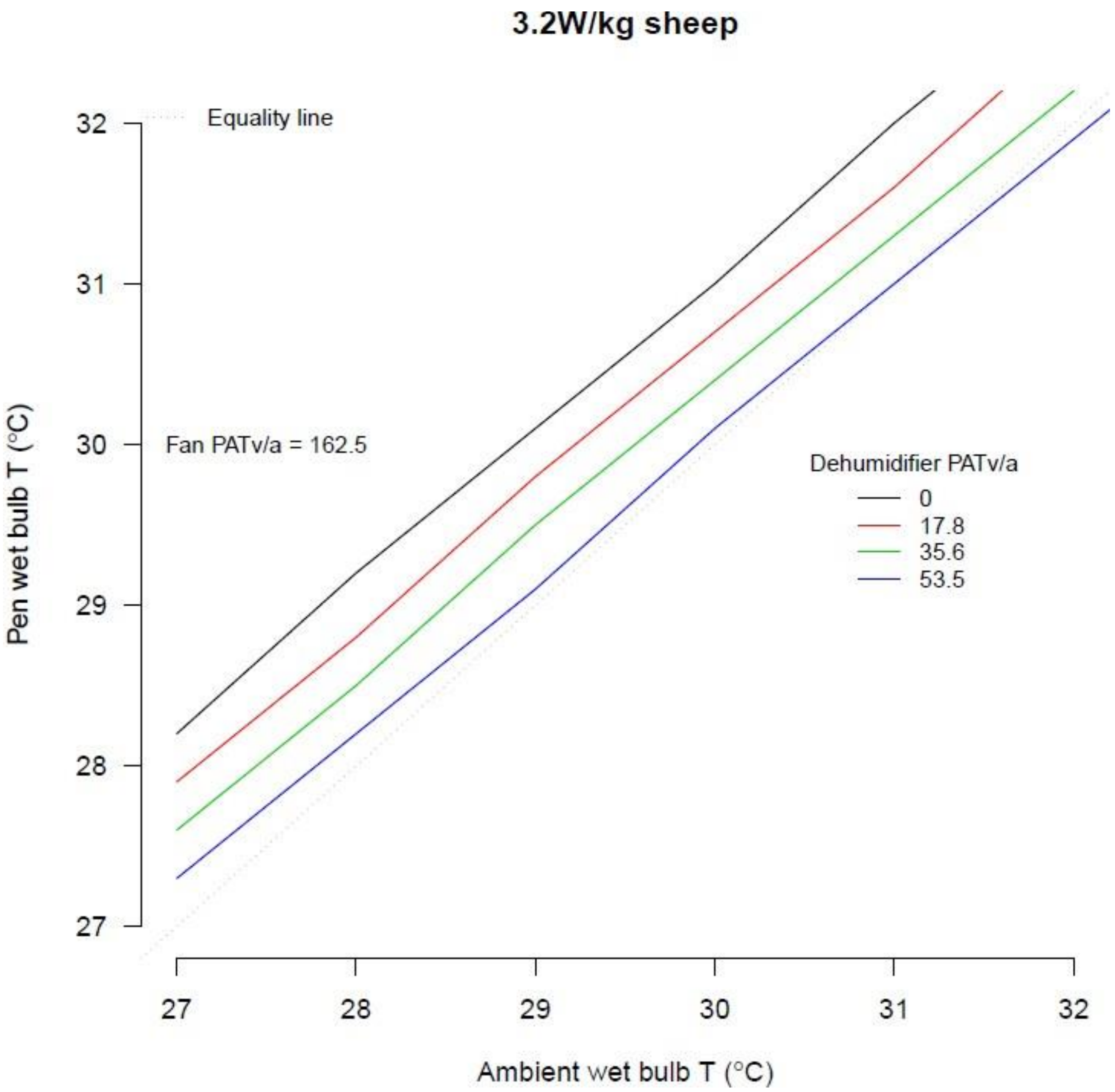


Figure 50: Predicted deck T_{WB} reduction compared to ambient T_{WB} for a deck stocked to ASEL v2.3 with 40kg sheep, with maximum fan ventilation ($PAT_{V/A}$ = 162.5/Hr is 100%) and under varying ambient wet bulb temperatures and dehumidification rates (note: trial $PAT_{V/A}$ = 17.8/Hr is 100%).

This was used to predict deck conditions across the range of ambient wet bulb temperatures likely experienced on northern summer sheep trade routes to the Middle East.

This extrapolative model suggests that dehumidification at levels equivalent to a $PAT_{V/A}$ of 53.5 can provide for deck wet bulb conditions that mirror ambient wet bulb conditions across the range of ambient wet bulb temperatures expected for sheep export voyages to the Middle East throughout the year. In other words, a dehumidifier $PAT_{V/A}$ = 53.5 would practically eliminate the increase in deck wet bulb temperature above ambient.

Controlling deck ambient wet bulb temperatures to match ambient wet bulb temperatures would result in an important reduction in the risk of heat stress for a voyage. The risk would resemble the blue lines within Figure 48 (28+°C risk) and Figure 49 (30+°C risk); being the no sheep (empty pen) line. This represents approximately a 20% absolute reduction in the risk of experiencing deck conditions of 28+°C and a 6% absolute reduction in risk of experiencing deck conditions of 30+°C, compared to non-dehumidified decks during the Northern Hemisphere summer.

Expected impact on animal welfare

Animal modelling suggests that the levels of dehumidification applied in the trial, when used in conjunction with full ventilation, will produce a modest reduction in deck wet-bulb temperature below the ambient wet-bulb temperature. The absolute risk reduction for 28+°C days, as a result of the addition of dehumidification (as per the trial rates) to ventilation fans, ranges approximately between 10–20% over the northern summer months of May, June and July (single-day heat events). The absolute risk reduction for 30+°C days, as a result of the addition of dehumidification (as per trial rates) to fans, ranges approximately between 2–6% over the northern summer months of May, June and July (single-day heat events).

However, if it were possible to add extra dehumidification capacity it would be expected to provide further valuable reduction in heat stress risk. For example, a trebling of the dehumidification capacity beyond that used in this trial would mostly eliminate increases in deck wet bulb temperatures above ambient conditions. Increased dehumidification capacity at this level or beyond would be expected to provide for deck wet bulb temperatures below ambient temperatures with concomitant heat stress risk reduction.

Controlling deck ambient wet bulb temperatures to match ambient wet bulb temperatures would result in an important reduction in the risk of heat stress for a voyage. The risk would resemble the blue lines within Figure 46 (28+°C risk) and Figure 47 (30+°C risk); being the no-sheep (empty pen) line. This represents approximately a 40% reduction in the risk of experiencing deck conditions of 28+°C and a 15% reduction in the of experiencing deck conditions of 30+°C, compared to non-dehumidified decks during the North Hemisphere summer.

Commercial and practical considerations

This field study and subsequent analysis and modelling has advanced the knowledge and tools available to understand the scope and performance required from potential technology solutions in the latent and sensible cooling space. It has also highlighted that the management of the environmental conditions on a vessel is complex and multi-faceted, and this study has helped to identify where the performance of various interventions is greatest and where their application in concert or combination may have an impact.

However, the results also show that while the potential exists for dehumidification in some form to contribute to on-board environmental management, performance improvements are needed before any technology solution could be considered practically or commercially implementable – for example, through increased outputs per unit; greater efficiencies in units; or by targeting both latent (dehumidification) and sensible (air conditioning) heat removal combinations.

As the next stages are considered, there are a range of practical and commercial considerations that will need to be better understood and applied to the framing and assessment of potential approaches. Several of these are listed below:

- Space and weight constraints – Vessels have limited space to place equipment, particularly when in terms of retro-fitting. Further, there are structural and stowage considerations that must be considered (e.g. strength of roofing or flooring to hold large equipment; effect on the stability of the vessel). Noting that the current trial used two units (one HCU3000 and one HCU6000) for one hold of a deck, the multiplied effect across a vessel or from increasing the number of units per hold is currently unrealistic for implementation without improvements in the per unit efficiencies and outputs.
- Energy use – Vessels operate from a contained energy source and the application of new technologies will need to consider the capacity of the ships to supply the needs of machinery.
- Regulatory requirements – There are various measures of ventilation with which livestock vessels must comply to ensure that heat and other pollutants are removed during shipments. The minimum thresholds applied in regulation may not align with the most efficient combination of technologies identified in the future or capture the potential of delivering heat load management in a different manner (e.g. via DHM rather than fan flow rates). However, to change regulations will require a better knowledge of how much of the value of ventilation provided contributes towards heat load management, and how much is toward the removal of pollutants.
- Maintenance – Equipment such as dehumidifiers are likely to require routine maintenance by technicians, particularly in the on-board environment. If such devices were applied, there would also need to be expertise on the vessel to fix any issues that arise.
- Redundancy or contingency arrangements – If a reliance on devices such as dehumidifiers were to expand, there would be a need to establish contingency or redundancy infrastructure and procedures. The most obvious example would be the ability to revert back to a fans-based ventilation system in the case of failure.
- Installation into vessels – Similarly to the space and weight constraints, there are also practical considerations about how conditioned air would be delivered to the decks

from the units themselves. This may be more easily provided for through built-in solutions, rather retro-fitting.

- Effect on vessel performance (speed, fuel consumption etc.) – The effect on the performance of a vessel from implementing a new air treatment approach could be significant – depending on the scale of the energy draw, the weight etc.
- Customisation – Livestock vessels have many differences in how they are built, ventilated and the types of challenges that they face in providing animal comfort (e.g. different routes, different livestock). It is unlikely that there will be a one-size-fits-all model of any technology, and customisation to the needs to different vessels may make options less, or potentially, more viable.
- Wider cost / benefit considerations – Current industry and regulatory arrangements for sheep exports to the Middle East prevent shipments between June and the later parts of September. Outside of these times (and prior to the moratorium of the summer shipments), the industry HSRA must be complied with – although its future settings / applications are under review. The return on investment and cost / benefit consideration of significant infrastructure changes to vessels will no doubt be considered against this framework (noting that the use of DHM etc. could potentially provide significant benefits by enabling shipments or stocking densities that would not otherwise be allowed).

Some of these factors are essentially unknown for new or bespoke technology solutions that may arise. In the current space, however, these are real and inhibiting factors and particularly when considered in the context of the potential application of off-the-shelf products.

As greater clarity is achieved through refined modelling and further technology exploration, there will be opportunities and needs to consider other aspects, such as whether there are configurations of the technologies or their use that can reduce the impact of the costs / constraints and deliver the benefits. For example, by understanding whether dehumidification would be required on as ongoing basis, as an emergency intervention, in a phased deck by deck approach etc.

Potential next trials

There are several potential studies that are suggested from this research project. These are:

1. Review ways to increase dehumidification performance (increase PAT, reduce humidity, reduce temperature, improve efficiencies/delivery etc.). This may allow more effective use of dehumidifiers on vessels. The questions would potentially include: can dehumidifiers operate at PATs in excess of what was trialled and what is the best combination of dehumidification and cooling of air supplied out of the dehumidifiers relative to the animal.

   If a suitable dehumidifier unit / technology (PAT, dehumidification and cooling combination) and configuration can be identified that optimises animal comfort and performance on vessels within economic and physical constraints, a series of controlled climate (chamber), pen studies with sheep could then be recommended with confidence. The objectives of such controlled sheep studies would be to:

   a. Confirm dehumidification pen performance

b. Identify impact on sheep welfare

c. Evaluate effectiveness of strategic (non-continuous) dehumidification during periods of prolonged heat

2. Better refine the science underpinning the sheep heat generation estimates. The current range in estimates – which are integral in modeling animal heat at the aggregated loaded level
– flows through to a potential difference in dehumidification output requirements of as much as double, possibly having a significant effect / application. In this regard, there is some indication that the 3.2 W/kg used is an upper estimate based on the outcomes from the use of the industry HSRA model.

3. Improve the understanding of how each of the elements within the wet bulb temperature (e.g. T_{DB} and RH) interact and what their individual and combined influences are on animal outcomes. This would assist in describing or scoping how technologies (such as DHM, or air conditioning etc.) could be best designed to match animal comfort outcomes. For example, this might consider what the most effective balance in terms of latent and sensible heat removal may be across the range of likely voyage conditions.

4. Build a stronger and more detailed understanding of how ventilation / PAT provides for both replenishment of oxygen and removal of pollutants (ammonia and carbon dioxide in particular) and the removal of heat to provide for animal comfort (i.e. of the total PAT values required what is the proportion required to provide for each of the above outcomes). This information will assist in clarifying and scoping technologies (e.g. minimum / maximum PATs) and help to identify how best to apply DHM to achieve animal comfort outcomes, while also ensuring gas exchange and pollutant removal. The development of a pollutant production and expulsion sub-model to include within the animal model may be possible and of potential value.

5. Analyse, quantify and potentially model the commercial and logistical challenges that need to be addressed to help guide technological development, which will help clarify the economic and animal welfare benefits of incorporating dehumidification of sheep vessels.

CONCLUSION

In 2018, LiveCorp undertook a challenge-led Open Innovation project to explore technologies that could mitigate wet bulb temperatures from reaching levels that exceed the heat stress thresholds of sheep, but which also provide an environment that supports acclimatisation to destination country conditions. A technology trial program was developed to assess the effectiveness, viability and commercial scalability of each of the identified technologies. Through consultation with industry, a specific type of dehumidification technology, known as DryCool, was identified as having the potential to achieve the objectives of the project.

A field trial was conducted in Dubai between June 24 – 28 2019 on-board an empty livestock vessel, with further detailed modelling conducted to use those results to predict scenarios outside of the conditions and variables studied (including the addition of animals and extra fans or dehumidification units).

The conclusions from the completion of this project are as follows:

- This trial represented the first field study conducted of its kind, where an evaluation under realistic operating conditions was undertaken of the performance of a commercial dehumidification unit on the deck environment of a livestock ship. The results captured the effect, complexities and interactions of delivering dehumidified air onto livestock vessels under a range of conditions. Such outcomes were achieved through the involvement of a cross-disciplinary team of experts (HVAC engineers and veterinarians), technology providers and practical support (the owner, Captain, officers and crew of the vessel etc).

- The structure of the trial enabled certain conditions to be defined and quantified as a rate response to a deck space (as the combined impact of all inputs such as DH, fans, livestock and ambient conditions). The results showed, as expected, that the commercial dehumidifier was able to reduce the humidity within the space but did not deliver a significant change in the dry bulb temperature (a somewhat intentional outcome based on the selection of the dehumidifier and its optimisation for moisture removal). In effect, this outcome showed proof of the basic concept of applying a commercial dehumidifier to reduce humidity and wet bulb temperature on an empty livestock vessel.

- This study also identified significant variation in the performance of the dehumidifiers under different conditions, with changes across the day correlating with shifting ambient conditions. Establishing this performance map of these dehumidifiers across a range of conditions has provided important information that was not readily available in the public domain.

- Using the interactions and results from the field trial stages, it was possible to establish an environmental model that connected dehumidifier PATs, fan PATs and ambient temperatures together in a verified framework that allowed for the assessment of different combinations and wider simulations.

- In addition, due to the cross-disciplinary skills and expertise of the team, HVAC engineers were able to work with a veterinary expert to model the addition of sheep to deck space under study. This model uncovered various assumptions and challenges around estimating the thermal load of sheep. Some of these assumptions are significant, such as the wide variation in literature on estimated heat load from sheep (1.6 – 3.2 k/W).

- Refinements to the assumptions used would improve the ability to, and accuracy of, the modelling of animal comfort and in turn the predictions that can be derived from applying different combinations of variables (fans, dehumidification, ambient temperature, stocking density, class etc.). However, while further refinements would be beneficial, it was possible by using the information derived from the trial and currently available science, to establish and verify a thermal model of the deck space – including simulating the presence of sheep. This significant outcome allowed the study to evaluate the performance of the dehumidifiers and conduct this trial without the need for live sheep.

- With the modelling of sheep into the environmental model of the deck space, this project was able to capture the changing relative value of dehumidification compared to fans on the wet bulb temperature outcomes in different ambient conditions. This showed, generally speaking, that the use of fans had a stronger relative value in reducing pen wet bulb temperature when ambient wet bulb conditions were lower, that dehumidification had a stronger effect at higher wet bulb conditions, and a combination of the two interventions had the most significant effect within an intermediate range.

- The model also provided the opportunity to simulate a range of 'what-if' scenarios for a sheep vessel. This modelling provided the researchers with the ability to predict with reasonable confidence the results in extreme conditions for the application of different combinations of treated and untreated air, including at levels beyond what could be studied (e.g. three times as much dehumidification PAT).

- This analysis enabled an assessment to be made of what rates would be required using the studied commercial dehumidification unit to achieve meaningful reductions in wet bulb temperature. The outcomes from this modelling showed that by combining off-the-shelf dehumidification products (albeit at levels that have obvious significant commercial and logistical constraints), it was possible to lower wet bulb temperatures in situations where such a decrease could not be achieved using fans alone. While the significant commercial and logistical constraints are very real challenges, this finding still provides valuable information to guide the design or identification of technologies that may deliver it – potentially in more bespoke or combined solutions.

- It is also important to recognise that the development of the model used in this project will provide enduring value for the livestock export industry when looking to future research and in helping to determine next steps. In particular, it provides the means to scope the thresholds of success for technology developments, and to model and predict the animal welfare outcomes of live animal trials before progressing to them.

- Finally, the project identified several steps that could further refine and improve the model to support its ability to produce accurate predictions and guide technology decisions and further industry exploration in this area. These included possible next steps of:

- Reviewing and seeking out technologies or opportunities to increase dehumidification / air conditioning performance (increase PAT, reduce humidity, reduce dry bulb temperature etc.) and, if a suitable technology could be identified, considering moving to a controlled climate pen study.
- Better refining the science underpinning the sheep heat generation estimates.
- Improving the understanding of how each of the elements within the wet bulb temperature (e.g. TDB and RH) interact and what their individual and combined influences are on animal outcomes.
- Building a stronger and more detailed understanding of how ventilation / PAT provides for both replenishment of oxygen and removal of pollutants (ammonia and carbon dioxide in particular) and the removal of heat to provide for animal comfort (i.e. of the total PAT values required what is the proportion required to provide for each of the above outcomes).
- Analysing and quantifying the commercial and logistical challenges that need to be addressed and guide technological development.

It is important to also recognise that this trial would not have been possible without the support of the ship owner / importing company and its executive, employees, Captains, officers and crew. LiveCorp expresses its gratitude for their significant advice, support and commitment.