

NATIONAL TECHNICAL UNIVERSITY OF ATHENS SCHOOL OF MECHANICAL ENGINEERING

THERMODYNAMIC ANALYSIS OF OPEN CYCLE EVAPORATIVE COOLING SYSTEMS WITH LIQUID DESICCANT MEDIUMS

THESIS SUBMITTED FOR THE DEGREE OF DOCTOR OF PHILOSOPHY ROSA I. CHRISTODOULAKI

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ΕΘΝΙΚΟ ΜΕΤΣΟΒΙΟ ΠΟΛΥΤΕΧΝΕΙΟ ΣΧΟΛΗ ΜΗΧΑΝΟΛΟΓΩΝ ΜΗΧΑΝΙΚΩΝ

ΘΕΡΜΟΔΥΝΑΜΙΚΗ ΑΝΑΛΥΣΗ ΣΥΣΤΗΜΑΤΩΝ ΑΝΟΙΚΤΟΥ ΕΞΑΤΜΙΣΤΙΚΟΥ ΚΥΚΛΟΥ ΜΕ ΥΓΡΑ ΠΡΟΣΡΟΦΗΤΙΚΑ ΥΛΙΚΑ

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Preface

This Thesis was carried out at the Laboratory of Applied Thermodynamics, Thermal Engineering Section, School of Mechanical Engineering at the National Technical University of Athens.

Obtaining the results in this thesis has been possible by the support from several people.

I am profoundly indebted to my supervisor Prof. Emmanuel Rogdakis for his support, encouragement and invaluable advice regarding many difficult aspects of this thesis.

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No words are able to express how grateful I am to my husband and colleague Panagiotis Tsekouras for all of his love and for his continuous encouragement, without which this work would not have been achievable. I would like also to mention my child Dimitris; with him this work was difficult to accomplish, but without him there would be no meaning in life.

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ABSTRACT

The objective of this work is to study the heat and mass transfer processes between air, desiccant solution and water during absorption and desorption processes for applications of liquid desiccant air conditioning. The study is performed both theoretically and experimentally and investigates the performance of these processes in terms of absorber efficiency and regenerator efficiency. Firstly, the theoretical background on the thermodynamic properties of the most common liquid desiccant solutions (Lithium Chloride, Lithium Bromide and Calcium Chloride) is established, followed by the development of the mathematical models for the dehumidifier and regenerator. The developed models concern three different configurations; the adiabatic device with structured packing, the nonadiabatic device with a tubular heat exchanger and the non-adiabatic device with a parallel plate heat exchanger. All models are validated with experimental data taken from the literature and the comparison indicates reasonable deviations between the theory and the experiment. The extensive parametric analysis carried out in each device indicates the most important parameters that affect the performance of liquid desiccant dehumidifiers and regenerators. The experimental procedure offered valuable information on the operation of such systems. The experimental results showed that the air dehumidification rate ranged from 6.99 to 11.90 kg water/hr and the energy consumption related to the regenerator ranged from 6.93 kWh to 26.22 kWh, depending on the operational conditions. Modest values of regeneration efficiency were achieved even under hot water temperatures as low as 46.11 °C. The proven low temperature heat required to drive liquid desiccant air conditioning systems along with the subsequent environmental and economic advantages, may facilitate the use and widespread installation of these systems.

ΠΕΡΙΛΗΨΗ

Το αντικείμενο της έρευνας αυτής είναι η μελέτη των φαινομένων μεταφοράς θερμότητας και μάζας μεταξύ του αέρα, του αφυγραντικού υλικού και του νερού που συντελούνται στα κλιματιστικά συστήματα ανοικτού εξατμιστικού κύκλου με υγρά προσροφητικά υλικά. Η έρευνα υλοποιείται τόσο σε θεωρητικό όσο και σε πειραματικό επίπεδο και εξετάζει την ενεργειακή απόδοση του αφυγραντή και του αναγεννητή. Αρχικά, αποκτάται το θεωρητικό υπόβαθρο σχετικά με τις θερμοδυναμικές ιδιότητες των πιο διαδεδομένων αφυγραντικών υδατικών διαλυμάτων (Χλωριούχο Λίθιο, Βρωμιούχο Λίθιο, Χλωριούχο Ασβέστιο). Η γνώση αυτή χρησιμοποιείται ως βάση για την ανάπτυξη των μαθηματικών μοντέλων του αφυγραντή και του αναγεννητή. Τα μοντέλα αφορούν τρεις διαφορετικές διατάξεις: την αδιαβατική συσκευή με το πληρωτικό υλικό, τη μη αδιαβατική συσκευή με το σωληνοειδή εναλλάκτη και τη μη αδιαβατική συσκευή με τον πλακοειδή εναλλάκτη. Όλα τα μοντέλα συγκρίνονται με πειραματικά δεδομένα από την ήδη υπάρχουσα βιβλιογραφία και δείχνουν αξιόπιστη συμπεριφορά, παρουσιάζοντας μικρές αποκλίσεις μεταξύ τους. Η εκτενής παραμετρική ανάλυση που διεξάγεται σε κάθε επικυρωμένο μαθηματικό μοντέλο υποδεικνύει τους σημαντικότερους παράγοντες που επηρεάζουν την ενεργειακή απόδοση του αφυγραντή και του αναγεννητή. Η πειραματική διαδικασία που διεξάγεται στα πλαίσια της εργασίας αυτής προσφέρει πολύτιμες γνώσεις για τη λειτουργία αυτών των κλιματιστικών συσκευών. Τα πειραματικά αποτελέσματα δείχνουν ότι ο ρυθμός αφύγρανσης του αέρα κυμαίνεται από 6.99 έως 11.90 kg νερού/ώρα και η ενεργειακή κατανάλωση του αναγεννητή κυμαίνεται από 6.93 kWh έως 26.22 kWh, ανάλογα με τις εκάστοτε συνθήκες λειτουργίας. Ικανοποιητική απόδοση του αναγεννητή επιτυγχάνεται ακόμη και με θερμοκρασία θερμού νερού 46.11 °C. Η χαμηλή θερμοκρασία που απαιτείται για τη λειτουργία αυτών των κλιματιστικών μονάδων ευνοεί τη χρήση ανανεώσιμων πηγών ενέργειας, αποκτώντας έτσι περιβαλλοντικά και οικονομικά πλεονεκτήματα έναντι των συμβατικών κλιματιστικών συστημάτων συμπίεσης ατμού.

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εκτενής περιλήψη

ΕΙΣΑΓΩΓΗ (ΚΕΦΑΛΑΙΟ 1)

Τα τελευταία χρόνια, ο ρυθμός της εξασθένισης του στρώματος του όζοντος έχει προκαλέσει έντονη ανησυχία στην επιστημονική κοινότητα. Η παρουσία χλωροφθορανθράκων στην ατμόσφαιρα που απελευθερώνονται από τη χρήση κλιματιστικών και ψυκτικών συσκευών είναι η βασικότερη αιτία του φαινομένου. Αυτό το παγκόσμιο περιβαλλοντικό πρόβλημα, σε συνδυασμό με τις ολοένα και αυξανόμενες απαιτήσεις για βελτίωση της ποιότητας του εσωτερικού αέρα των κτηρίων, έχουν συμβάλλει στην αλλαγή των σχεδιαστικών απαιτήσεων για τον κλιματισμό.

Η προστασία από την υπερθέρμανση είναι τώρα η βασική σχεδιαστική παράμετρος για τα νέα κτήρια, εξαιτίας της ευρείας χρήσης ηλεκτρονικών υπολογιστών, της εγκατάστασης μη αποδοτικών συστημάτων φωτισμού και της συμπεριφοράς των χρηστών του κτηρίου. Τα εναλλακτικά συστήματα κλιματισμού, όπως τα κλιματιστικά συστήματα ανοικτού κύκλου με υγρά προσροφητικά υλικά, εξασφαλίζουν τη θερμική άνεση στο εσωτερικό των κτηρίων με ένα πιο οικονομικά και περιβαλλοντικά βιώσιμο τρόπο.

Το φορτίο κλιματισμού ενός κτηρίου είναι το άθροισμα του αισθητού και του λανθάνοντος φορτίου και αντιπροσωπεύει το 20-40% της συνολικής ενέργειας που καταναλώνεται σε ένα κτήριο. Τα συμβατικά κλιματιστικά συστήματα υποψύχουν τον αέρα προκειμένου να παρέχουν αφύγρανση πέραν της ψύξης, και έτσι, απαιτείται αναθέρμανση του αέρα πριν αυτός εισαχθεί στο χώρο.

Σε αντίθεση με τα συμβατικά συστήματα, τα κλιματιστικά συστήματα ανοικτού κύκλου με υγρά προσροφητικά υλικά δε χρησιμοποιούν χλωροφθοράνθρακες ως ψυκτικά μέσα και ηλεκτρισμό για τη λειτουργία τους. Το ψυκτικό μέσο τους είναι το νερό και η βασική πηγή ενέργειας είναι η θερμότητα, η οποία μπορεί να προέρχεται από Ανανεώσιμες Πηγές Ενέργειας ή απορριπτόμενη θερμότητα. Επιπλέον, τα συστήματα αυτά λειτουργούν υπό ατμοσφαιρική πίεση, χωρίς να απαιτούν κοστοβόρα συστήματα κενού.

ΕΚΤΕΝΗΣ ΠΕΡΙΛΗΨΗ

ΑΝΤΙΚΕΙΜΕΝΟ ΕΡΕΥΝΑΣ & ΚΑΙΝΟΤΟΜΙΑ

Το αντικείμενο της έρευνας αυτής είναι η μελέτη των φαινομένων μεταφοράς θερμότητας και μάζας μεταξύ του αέρα, του αφυγραντικού υλικού και του νερού που συντελούνται στα κλιματιστικά συστήματα ανοικτού εξατμιστικού κύκλου με υγρά προσροφητικά υλικά. Η εργασία υλοποιείται τόσο σε θεωρητικό όσο και σε πειραματικό επίπεδο και εξετάζει την απόδοση του αφυγραντή και του αναγεννητή ως προς τον ρυθμό αφύγρανσης αέρα και την ενεργειακή κατανάλωση.

Ειδικότεροι στόχοι της εργασίας είναι οι εξής:

- Η απόκτηση του θεωρητικού υπόβαθρου σχετικά με τα διαθέσιμα προσροφητικά υδατικά διαλύματα, τις συσκευές αφύγρανσης και αναγέννησης που χρησιμοποιούν υγρά προσροφητικά υλικά και τις διατάξεις των κλιματιστικών συστημάτων.
- Η ανάπτυξη μαθηματικών μοντέλων για την εκτίμηση της απόδοσης των αφυγραντικών
 διατάξεων που χρησιμοποιούν υγρά προσροφητικά υλικά.
- Η ανάπτυξη μαθηματικών μοντέλων για την εκτίμηση της απόδοσης των αναγεννητικών
 διατάξεων των υγρών προσροφητικών υλικών.
- Η εξέταση, μέσω εκτενούς παραμετρικής ανάλυσης, των σημαντικότερων παραμέτρων που επηρεάζουν την ενεργειακή απόδοση του αφυγραντή και του αναγεννητή.
- Η πειραματική αξιολόγηση των κλιματιστικών συστημάτων ανοικτού τύπου με υγρά προσροφητικά μέσα.
- Η ανάδειξη μεθόδων βελτιστοποίησης των συστημάτων αυτών, βασιζόμενη στην θεωρητική μελέτη και στην πειραματική διαδικασία.

Η καινοτομία της εργασίας έγκειται στους παρακάτω τομείς έρευνας:

- Επίλυση φαινομένων μεταφοράς θερμότητας και μάζας του αφυγραντή και του αναγεννητή χρησιμοποιώντας σύστημα συνήθων διαφορικών εξισώσεων.
- Ανάπτυξη αλγόριθμου επίλυσης των συντελεστών μεταφοράς θερμότητας και μάζας σε
 - Αδιαβατικές συσκευές με πληρωτικό υλικό
 - Μη αδιαβατικές συσκευές με σωληνοειδή εναλλάκτη
 - Μη αδιαβατικές συσκευές με πλακοειδή εναλλάκτη
- Ανάπτυξη μαθηματικών μοντέλων για την προσομοίωση της λειτουργίας των
 - Αφυγραντή και αναγεννητή ως αδιαβατικές συσκευές με πληρωτικό υλικό
 - Αφυγραντή και αναγεννητή ως μη αδιαβατικές συσκευές με σωληνοειδή εναλλάκτη
 - Αφυγραντή και αναγεννητή ως μη αδιαβατικές συσκευές με πλακοειδή εναλλάκτη

- Σύγκριση και επικύρωση των μαθηματικών μοντέλων με πειραματικά δεδομένα από τη βιβλιογραφία.
- Συγκριτική ανάλυση και παραμετρικές προσομοιώσεις του αφυγραντή και του αναγεννητή με
 τις ακόλουθες εξαρτημένες μεταβλητές
 - Είδος προσροφητικού διαλύματος
 - Κατάσταση εισόδου του αέρα
 - Κατάσταση εισόδου του προσροφητικού διαλύματος
 - Κατάσταση εισόδου του νερού
- Ανάπτυξη μαθηματικού μοντέλου για την προσομοίωση των κλιματιστικών συστημάτων ανοικτού κύκλου με υγρό προσροφητικό υλικό
- Διεξαγωγή πειραμάτων αφύγρανσης και αναγέννησης με τις ακόλουθες εξαρτημένες
 μεταβλητές
 - Κατάσταση εισόδου του αέρα
 - Κατάσταση εισόδου του προσροφητικού διαλύματος
 - ο Κατάσταση εισόδου του νερού.

ΔΟΜΗ ΔΙΑΤΡΙΒΗΣ

Η διατριβή αναπτύσσεται σε οκτώ κεφάλαια. Το 1[°] κεφάλαιο εισάγει τον αναγνώστη στην τεχνολογία που πραγματεύεται η εν λόγω διατριβή. Αναφέρονται συνοπτικά οι συμβατικές τεχνολογίες κλιματισμού που εφαρμόζονται σήμερα, η ανάγκη για περισσότερο ενεργειακά αποδοτικά κλιματιστικά συστήματα και η προοπτική των κλιματιστικών συστημάτων ανοικτού εξατμιστικού κύκλου με υγρό προσροφητικό υλικό. Περιγράφονται επίσης τα πρωτότυπα στοιχεία της διατριβής.

Στο 2° κεφάλαιο παρουσιάζεται η βιβλιογραφική επισκόπηση των μεθόδων επίλυσης των δύο θεμελιωδών λειτουργιών των κλιματιστικών συστημάτων ανοικτού κύκλου με υγρά προσροφητικά υλικά: της αφύγρανσης του αέρα και της αναγέννησης του αφυγραντικού διαλύματος. Περιγράφονται αναλυτικά τα υπολογιστικά μοντέλα που έχουν αναπτυχθεί έως σήμερα από τους ερευνητές, οι βασικές υποθέσεις που έγιναν και τα συμπεράσματα που προέκυψαν.

Το 3° κεφάλαιο εισάγει τον αναγνώστη στις θερμοφυσικές ιδιότητες των υδατικών διαλυμάτων αλάτων που χρησιμοποιούνται ως αφυγραντικά μέσα στα κλιματιστικά συστήματα προσρόφησης ανοικτού κύκλου. Οι θερμοφυσικές ιδιότητες που εξετάζονται για κάθε διάλυμα είναι: όριο διαλυτότητας, τάση ατμών, επιφανειακή τάση, δυναμική συνεκτικότητα, θερμική αγωγιμότητα, ειδική θερμοχωρητικότητα και διαφορική ενθαλπία διάλυσης. Η ανάλυση αυτών των

ΕΚΤΕΝΗΣ ΠΕΡΙΛΗΨΗ

θερμοφυσικών ιδιοτήτων αποτελεί το πρώτο βήμα προς την κατανόηση του κύκλου λειτουργίας των συστημάτων αυτών και την ανάπτυξη των μοντέλων προσομοίωσης.

Το 4° κεφάλαιο περιγράφει την ανάπτυξη του υπολογιστικού μοντέλου για τον αφυγραντή και τον αναγεννητή στην απλούστερη διάταξη, την αδιαβατική. Αρχικά, προσδιορίζονται οι συντελεστές μεταφοράς θερμότητας και μάζας και ακολουθεί το υπολογιστικό μοντέλο προσομοίωσης του αφυγραντή και του αναγεννητή. Στη συνέχεια, παρουσιάζεται η επικύρωση του αναπτυχθέντος μοντέλου με πειραματικά δεδομένα από την ήδη υπάρχουσα βιβλιογραφία. Ακολουθεί η παραμετρική ανάλυση των παραγόντων εκείνων που επηρεάζουν την αφύγρανση και την αναγέννηση και συμπεραίνονται τα βασικότερα στοιχεία.

Το κεφάλαιο 5° αφορά τη μελέτη του αφυγραντή και αναγεννητή, ως μη αδιαβατική συσκευή με σωληνοειδή εναλλάκτη. Χάριν ομοιομορφίας, τα στάδια της μελέτης είναι όμοια με εκείνα του προηγούμενου κεφαλαίου. Έτσι, αρχικά υπολογίζονται οι συντελεστές μεταφοράς θερμότητας και μάζας και ακολουθεί το υπολογιστικό μοντέλο προσομοίωσης του αφυγραντή και του αναγεννητή. Η επικύρωση του αναπτυχθέντος μοντέλου επιτυγχάνεται με τη σύγκρισή του με πειραματικά δεδομένα από τη βιβλιογραφία. Στη συνέχεια, γίνεται παραμετρική ανάλυση των παραγόντων εκείνων που επηρεάζουν την αφύγρανση και την αναγέννηση και συμπεραίνονται τα βασικότερα στοιχεία.

Η πειραματική διάταξη του κλιματιστικού συστήματος ανοικτού κύκλου με υγρό προσροφητικό υλικό περιγράφεται στο κεφάλαιο 6°. Εφόσον το σύστημα αυτό αποτελείται από μη αδιαβατικό αφυγραντή και αναγεννητή με πλακοειδή εναλλάκτη, αρχικά αναπτύσσεται το θεωρητικό μέρος. Αυτό περιλαμβάνει την ανάπτυξη του υπολογιστικού μοντέλου για τον αφυγραντή ως μη αδιαβατική συσκευή με πλακοειδή εναλλάκτη. Ακολουθεί η επικύρωση του μοντέλου και η παραμετρική του ανάλυση. Το πειραματικό μέρος του κεφαλαίου ξεκινάει με την περιγραφή της πειραματικής διάταξης που έχει εγκατασταθεί στο Εργαστήριο Εφαρμοσμένης Θερμοδυναμικής ΕΜΠ. Σχολιάζεται η λειτουργία της διάταξης, παραθέτονται τα σχέδια της εγκατάστασης και του μετρητικού συστήματος και στη συνέχεια, αναπτύσσεται η υλοποίηση των πειραμάτων. Ακολουθεί η αξιολόγηση των πειραματικών μετρήσεων με εκτεταμένη ανάλυση. Ουσιώδη σημασία για την εργασία έχει η σύγκριση των πειραματικών μετρήσεων με το επικυρωμένο μαθηματικό μοντέλο. Ακολουθούν προτάσεις βελτιστοποίησης της πιλοτικής μονάδας και της πειραματικής διαδικασίας.

Στο 7° κεφάλαιο γίνεται η επισκόπηση της διατριβής και διατυπώνονται τα γενικά συμπεράσματα. Χρήσιμες πληροφορίες για προτάσεις μελλοντικής έρευνας αναφέρονται στο τέλος.

Η διατριβή ολοκληρώνεται με τα παραρτήματα στο 8° κεφάλαιο, τα οποία περιλαμβάνουν μεταξύ άλλων τις δημοσιεύσεις του συγγραφέα και τις αναφορές στις δημοσιεύεις του από άλλους ερευνητές.

Το παρακάτω διάγραμμα οπτικοποιεί τη ροή της έρευνας που διεξάγεται, από τη σύλληψη της ιδέας έως την συγγραφή της Διατριβής.



Figure 0-1 Διάγραμμα ροής της διδακτορικής διατριβής

ΒΙΒΛΙΟΓΡΑΦΙΚΗ ΕΠΙΣΚΟΠΗΣΗ ΜΕΘΟΔΩΝ ΕΠΙΛΥΣΗΣ ΑΦΥΓΡΑΝΣΗΣ ΚΑΙ ΑΝΑΓΕΝΝΗΣΗΣ (ΚΕΦΑΛΑΙΟ

2)

Τα κλιματιστικά συστήματα ανοικτού κύκλου με υγρά προσροφητικά υλικά αποτελούν μία ενεργειακά αποδοτική μέθοδο για την αφύγρανση του εσωτερικού αέρα των κτηρίων, συγκριτικά με τα συμβατικά κλιματιστικά συστήματα. Σε αντίθεση με τα συστήματα συμπίεσης ατμών, στα οποία ο ψυκτικός κύκλος χρησιμοποιεί ηλεκτρική ενέργεια και στα οποία ο αέρας υφίσταται υπόψυξη, τα κλιματιστικά συστήματα ανοικτού κύκλου με υγρά προσροφητικά υλικά είναι θερμικά κινούμενα και ο αέρας δε χρειάζεται να ψυχθεί πέραν του σημείο δρόσου. Εφόσον δε γίνεται υπόψυξη του αέρα υπό επεξεργασία, αποφεύγεται η ανα-θέρμανση του αέρα πριν αυτός εισαχθεί στο κτήριο. Κατά συνέπεια, τα κλιματιστικά συστήματα ανοικτού κάθαρότερες πηγές ενέργειας, όπως είναι η θερμική ηλιακή ενέργεια ή η απορριπτόμενη θερμότητα. Επιπλέον, η όλη διαδικασία συντελείται υπό ατμοσφαιρική πίεση, εξαλείφοντας έτσι την ανάγκη για κοστοβόρα συστήματα κενού.

Τα υγρά προσροφητικά υλικά είναι υγροσκοπικά υλικά που έχουν την ιδιότητα να απορροφούν ή να προσδίδουν υγρασία στο περιβάλλον, εξαιτίας της διαφοράς τάσης ατμών στην επιφάνειά τους και στον περιβάλλοντα αέρα. Όταν το υγρό προσροφητικό υλικό περιέχει περισσότερη υγρασία από τον περιβάλλοντα αέρα, απορροφά θερμότητα από τον αέρα, απελευθερώνει την πλεονάζουσα υγρασία του στον αέρα και προκαλεί ψύξη του περιβάλλοντος αέρα. Η διαδικασία είναι όμοια με την εξατμιστική ψύξη. Στην αντίθετη περίπτωση που το υγρό προσροφητικό υλικό περιέχει λιγότερη υγρασία από τον περιβάλλοντα αέρα, απελευθερώνει θερμότητα στον αέρα, απορροφά υγρασία από τον αέρα και προκαλεί θέρμανση του περιβάλλοντος αέρα. Το ποσό της θερμότητας που απελευθερώνει το υγρό προσροφητικό διάλυμα στον αέρα είναι ίσο με τη λανθάνουσα θερμότητα που θα απελευθέρωνε η συμπύκνωση της ανάλογης ποσότητας νερού.

Τα κλιματιστικά συστήματα ανοικτού εξατμιστικού κύκλου με υγρά προσροφητικά υλικά αποτελούνται από δύο βασικές συσκευές, τον αφυγραντή και τον αναγεννητή. Αρχικά, το ισχυρό προσροφητικό διάλυμα (υψηλή συγκέντρωση σε προσροφητικό υλικό) ψεκάζεται στο επάνω μέρος του αφυγραντή. Αέρας περιβάλλοντος, υψηλής τιμής απόλυτης υγρασίας, εισάγεται στο κάτω μέρος του αφυγραντή, μεταφέρει την υγρασία του στο προσροφητικό υλικό και απελευθερώνεται θερμότητα. Ταυτόχρονα, ψυχρό νερό, προερχόμενο από έναν εξατμιστικό ψύκτη, κυκλοφορεί σε κλειστό κύκλωμα μέσα στον αφυγραντή. Η χρησιμότητά του έγκειται στην διατήρηση της θερμοκρασίας του προσροφητικού υλικού σε χαμηλά επίπεδα και κατά συνέπεια, στη διατήρηση της τάσης ατμών του προσροφητικού υλικού σε χαμηλά επίπεδα. Ο αφυγρασμένος αέρας εξέρχεται από το επάνω μέρος του αφυγραντή και το ασθενές πλέον διάλυμα (χαμηλή συγκέντρωση σε προσροφητικό υλικό) εξέρχεται από το κάτω μέρος του αφυγραντή και οδεύει στον αναγεννητή. Η

απόδοση της αφύγρανσης μεγιστοποιείται όταν η θερμοκρασία εισόδου του προσροφητικού υλικού είναι χαμηλή και η συγκέντρωση εισόδου του προσροφητικού υλικού είναι υψηλή.

Η αναγέννηση είναι η διαδικασία κατά την οποία το προσροφητικό υλικό ανακτά την αρχική υψηλή του συγκέντρωση. Η διαδικασία αυτή έχει μεγάλη επίδραση στην ενεργειακή απόδοση των κλιματιστικών αυτών συστημάτων, καθώς απαιτεί την κατανάλωση ενέργειας. Η διάταξη του αναγεννητή είναι όμοια με τη διάταξη του αφυγραντή, όμως η διαδικασία είναι η αντίστροφη. Το ασθενές προσροφητικό διάλυμα ψεκάζεται στο επάνω μέρος του αναγεννητή και έρχεται σε άμεση επαφή με τον αέρα περιβάλλοντος. Καθώς τώρα η τάση ατμών του διαλύματος είναι μεγαλύτερη από την τάση ατμών του νερού στον αέρα, υγρασία μεταφέρεται από το διάλυμα στον αέρα που οδεύει προς απόρριψη στο περιβάλλον. Ταυτόχρονα, θερμό νερό, προερχόμενο από μία πηγή θερμότητας χαμηλής θερμοκρασιακής στάθμης, κυκλοφορεί σε κλειστό κύκλωμα μέσα στον αναγεννητή. Η χρησιμότητά του έγκειται στην αύξηση της θερμοκρασίας του προσροφητικό υλικού και κατά συνέπεια, στη αύξηση της τάσης ατμών του. Ο θερμός και υγρός αέρας απορρίπτεται στο περιβάλλον και το ισχυρό πλέον διάλυμα (υψηλή συγκέντρωση σε προσροφητικό υλικό) εξέρχεται από το κάτω μέρος του αναγεννητή.

Ιδανικές εφαρμογές για τα κλιματιστικά συστήματα ανοικτού κύκλου με υγρά προσροφητικά υλικά είναι οι περιπτώσεις όπου υπάρχουν μεγάλα φορτία λανθάνουσας θερμότητας και όπου απαιτείται ακριβή ρύθμιση της υγρασίας του εσωτερικού αέρα. Ενδεικτικά, αναφέρονται τα supermarket, τα μουσεία, τα γήπεδα παγοδρομιών, οι κλειστού τύπου κολυμβητικές δεξαμενές, χώροι φύλαξης αρχείων, νοσοκομεία, εργαστήρια, βιομηχανίες τροφίμων και φαρμακευτικές βιομηχανίες.

Το κεφάλαιο αυτό παρέχει μία εκτεταμένη βιβλιογραφική επισκόπηση των μεθόδων επίλυσης των δύο θεμελιωδών λειτουργιών των κλιματιστικών συστημάτων ανοικτού κύκλου με υγρά προσροφητικά υλικά: της αφύγρανσης του αέρα και της αναγέννησης του αφυγραντικού διαλύματος. Περιγράφονται αναλυτικά τα υπολογιστικά μοντέλα που έχουν αναπτυχθεί από τους ερευνητές έως σήμερα, οι βασικές υποθέσεις που έγιναν και τα συμπεράσματα που προέκυψαν. Υπάρχουν ποίκιλλες έρευνες σχετικά με την αφύγρανση και την αναγέννηση, με την πρώτη να αναφέρεται το 1969. Η εξέλιξη της έρευνας από τότε είναι μεγάλη: έχουν αναπτυχθεί αρκετά μοντέλα για την προσομοίωση των κλιματιστικών αυτών συστημάτων. Οι περισσότερες εργασίες επικεντρώνονται περισσότερο στην αντιρροή και λιγότερο στην ομορροή ή σταυρωτή ροή, καθώς η αντιρροή έχει αποδειχθεί ότι έχει την καλύτερη απόδοση. Υπάρχουν μερικές πειραματικές εργασίες οι οποίες συγκρίνουν τα θεωρητικά μοντέλα που έχουν αναπτυχθεί και διεξάγουν συνοπτική παραμετρική ανάλυση. Παρόλα αυτά, η επίδραση ορισμένων παραγόντων που επηρεάζουν την απόδοση τέτοιων συστημάτων παραμένει άγνωστη, όπως για παράδειγμα η βέλτιστη παροχή μάζας αέρα και προσροφητικού διαλύματος, η σχέση μεταξύ της επιφάνειας μεταφοράς μάζας και της

επιφάνειας μεταφοράς θερμότητας, η κατανομή της θερμότητας απορρόφησης μεταξύ του προσροφητικού υλικού και του αέρα. Περαιτέρω έρευνα και πειραματικά δεδομένα απαιτούνται για τη κατανόηση της λειτουργίας των συστημάτων αυτών, τη βελτίωση της συνολικής τους απόδοσης και συνεπώς, την προώθηση της ευρείας εφαρμογής τους.

ΘΕΡΜΟΦΥΣΙΚΕΣ ΙΔΙΟΤΗΤΕΣ ΠΡΟΣΡΟΦΗΤΙΚΩΝ ΔΙΑΛΥΜΑΤΩΝ (ΚΕΦΑΛΑΙΟ 3)

Τα υγρά προσροφητικά υλικά είναι υγροσκοπικά υλικά που έχουν την ιδιότητα να απορροφούν ή να προσδίδουν υγρασία στο περιβάλλον, εξαιτίας της διαφοράς τάσης ατμών στην επιφάνειά τους και στον περιβάλλοντα αέρα. Όταν το υγρό προσροφητικό υλικό περιέχει περισσότερη υγρασία από τον περιβάλλοντα αέρα, απορροφά θερμότητα από τον αέρα, απελευθερώνει την πλεονάζουσα υγρασία του στον αέρα και προκαλεί ψύξη του περιβάλλοντος αέρα. Στην αντίθετη περίπτωση που το υγρό προσροφητικό υλικό περιέχει λιγότερη υγρασία από τον περιβάλλοντα αέρα, απελευθερώνει θερμότητα στον αέρα, απορροφά υγρασία από τον περιβάλλοντα στον περιβάλλοντα αέρα.

Η επιλογή του προσροφητικού διαλύματος που χρησιμοποιείται στα κλιματιστικά συστήματα ανοικτού κύκλου είναι καθοριστική για την ενεργειακή του απόδοση και εξαρτάται από την ανύψωση του σημείου βρασμού, την ικανότητα αποθήκευσης ενέργειας, τη θερμοκρασία αναγέννησης, τη διαθεσιμότητά του στην αγορά και το κόστος. Τα συνήθη προσροφητικά διαλύματα είναι τα διαλύματα αλάτων και διαλύματα γλυκολών. Επικρατέστερα στην αγορά είναι τα διαλύματα αλάτων και διαλύματα γλυκολών. Επικρατέστερα στην αγορά είναι τα διαλύματα αλάτων και διαλύματα γλυκολών. Επικρατέστερα στην αγορά είναι τα διαλύματα αλάτων, όπως το Χλωριούχο Λίθιο (LiCl), Βρωμιούχο Λίθιο (LiBr) και Χλωριούχο Ασβέστιο (CaCl₂). Παρότι η πρώτη διάταξη που μελετήθηκε χρησιμοποιούσε τριεθυλενική γλυκόλη (TEG), σήμερα τα διαλύματα αυτά τείνουν να εκλείψουν από τις εφαρμογές.

Καθώς η σύγκριση της απόδοσης των ποικίλλων διαθέσιμων προσροφητικών διαλυμάτων δεν αποτελεί αντικείμενο της παρούσας Διατριβής, μόνο τα συνήθη προσροφητικά διαλύματα επιλέχθηκαν να μελετηθούν θερμοδυναμικά.

Έτσι, τα υδατικά διαλύματα του Χλωριούχου Λιθίου (LiCl), Βρωμιούχου Λιθίου (LiBr) και Χλωριούχου Ασβεστίου (CaCl₂) μελετήθηκαν εκτενώς στο κεφάλαιο αυτό. Επιλέχθηκαν οι θερμοδυναμικές τους ιδιότητες που είναι απαραίτητες για το σχεδιασμό κλιματιστικών συστημάτων, όπως όριο διαλυτότητας, τάση ατμών, επιφανειακή τάση, δυναμική συνεκτικότητα, θερμική αγωγιμότητα, ειδική θερμοχωρητικότητα και διαφορική ενθαλπία διάλυσης. Τα δεδομένα της μελέτης προέρχονται από εκτενή βιβλιογραφική αναφορά.

Το κεφάλαιο αυτό αποτελεί το πρώτο βήμα προς την κατανόηση του κύκλου λειτουργίας των συστημάτων αυτών και αποτελεί τη βάση για την ανάπτυξη των μαθηματικών μοντέλων προσομοίωσης.

ΕΚΤΕΝΗΣ ΠΕΡΙΛΗΨΗ

ΑΔΙΑΒΑΤΙΚΟ ΜΟΝΤΕΛΟ (ΚΕΦΑΛΑΙΟ 4)

Το κεφάλαιο αυτό παραθέτει την ανάπτυξη του υπολογιστικού μοντέλου για τον αδιαβατικό αφυγραντή και τον αναγεννητή. Ο όρος *αδιαβατικός* για τον αφυγραντή και τον αναγεννητή χρησιμοποιείται για να δείξει ότι δεν υπάρχει συναλλαγή ενέργειας μεταξύ των διατάξεων αυτών και του περιβάλλοντος. Δηλαδή, τόσο κατά την αφύγρανση όσο και κατά την αναγέννηση δεν υπάρχει θερμική αλληλεπίδραση με εξωτερική πηγή θερμότητας (π.χ ψύξη κατά την αφύγρανση και θέρμανση κατά την αναγέννηση). Στην περίπτωση αυτή, η διάταξη του αφυγραντή και του αναγεννητή περιέχει πληρωτικό υλικό που διευκολύνει την μεταφορά θερμότητας και μάζας, χωρίς να περιέχει εναλλάκτη θερμότητας. Η απόδοση των διατάξεων αυτών μελετάται ως προς τα φαινόμενα μεταφοράς θερμότητας και μάζας μεταξύ του αέρα και του προσροφητικού υλικού, όταν τα δύο αυτά μέσα βρίσκονται σε άμεση επαφή κατά την διαδικασία της αφύγρανσης και της αναγέννησης.

Αρχικά, προσδιορίζονται οι συντελεστές μεταφοράς θερμότητας και μάζας και ακολουθεί η ανάπτυξη του υπολογιστικού μοντέλου του αφυγραντή. Το μοντέλο έχει τη δυνατότητα να προσομοιώνει την απόδοση της διάταξης υπό μεταβλητές συνθήκες λειτουργίας. Στη συνέχεια, παρουσιάζεται η επικύρωση του αναπτυχθέντος μοντέλου του αφυγραντή με πειραματικά δεδομένα από την ήδη υπάρχουσα βιβλιογραφία. Από τη σύγκρισή τους, προκύπτει ότι οι προβλέψεις του μαθηματικού μοντέλου βρίσκονται σε συμφωνία με τα πειραματικά δεδομένα της βιβλιογραφίας, καθώς η μέγιστη απόκλισή τους είναι ±2.9% στην θερμοκρασία εξόδου του αροσροφητικού διαλύματος.

Στη συνέχεια, διενεργείται παραμετρική ανάλυση, που εξετάζει τους σημαντικότερους παράγοντες που επηρεάζουν την απόδοση και το ρυθμό της αφύγρανσης, όπως θερμοκρασία εισόδου αέρα, απόλυτη υγρασία εισόδου, παροχή μάζας αέρα, θερμοκρασία εισόδου προσροφητικού διαλύματος, συγκέντρωση εισόδου προσροφητικού διαλύματος, παροχή μάζας προσροφητικού διαλύματος. Βρέθηκε ότι οι παράμετροι που έχουν τη μεγαλύτερη επίδραση στην απόδοση της αφύγρανσης είναι η παροχή μάζας αέρα, η απόλυτη υγρασία εισόδου, η θερμοκρασία εισόδου προσροφητικού διαλύματος και η συγκέντρωση εισόδου προσροφητικού διαλύματος. Ο ρυθμός της αφύγρανσης αυξάνεται με την αύξηση της απόλυτης υγρασίας εισόδου, της συγκέντρωσης εισόδου προσροφητικού διαλύματος, παροχής μάζας αέρα και της παροχής μάζας προσροφητικού διαλύματος. Ο ρυθμός της αφύγρανσης μειώνεται με την αύξηση της θερμοκρασίας εισόδου προσροφητικού διαλύματος. Ο ρυθμός αφύγρανσης μεταβάλλεται ειλάχιστα με τη θερμοκρασία εισόδου αέρα. Η απόδοση της αφύγρανσης μεταβάλλεται ανάλογα με την παροχή μάζας προσροφητικού διαλύματος και τη θερμοκρασία εισόδου προσροφητικού διαλύματος,

μεταβάλλεται αντιστρόφως ανάλογα με την παροχή μάζας αέρα, ενώ μεταβάλλεται ελάχιστα με τη μεταβολή της θερμοκρασίας εισόδου αέρα, απόλυτη υγρασία εισόδου και συγκέντρωση εισόδου προσροφητικού διαλύματος. Υψηλή απόδοση στη αφύγρανση επιτυγχάνεται κάτω από υψηλές τιμές απόλυτης υγρασίας (>0.013 kg_w/kg_{dα}), γεγονός που καθιστά τα συστήματα αυτά ιδιαίτερα ελκυστικά στις τροπικές περιοχές.

Τα τρία προσροφητικά διαλύματα, LiCl, LiBr και CaCl₂, συγκρίνονται μεταξύ τους και αξιολογείται η αφυγραντική τους απόδοση κάτω από τις ίδιες αρχικές συνθήκες. Η ανάλυση δείχνει ότι η διάταξη που χρησιμοποιεί το διάλυμα LiCl παρουσιάζει υψηλότερη απόδοση από εκείνη με το LiBr και το CaCl₂, υπό ίδιες συνθήκες λειτουργίας. Παρόλο που σε περιβάλλον με υψηλές τιμές απόλυτης υγρασίας τα τρία διαλύματα παρουσιάζουν παρόμοια συμπεριφορά, η απόδοση του LiCl είναι πιο σταθερή και προβλέψιμη από ό,τι εκείνη του CaCl₂. Συμπεραίνοντας, υψηλή απόδοση στη αφύγρανση επιτυγχάνεται κάτω από μεγάλη απόλυτη υγρασία περιβάλλοντος, μικρή παροχή μάζας αέρα και διάλυμα LiCl ως προσροφητικό μέσο.

Η μελέτη της απόδοσης του αδιαβατικού αναγεννητή επιτυγχάνεται μέσω της ανάπτυξης μαθηματικού μοντέλου. Σε συμφωνία βρίσκονται τα αποτελέσματα του θεωρητικού μοντέλου και των πειραματικών δεδομένων από τη βιβλιογραφία, με την απόκλιση μεταξύ τους να βρίσκεται στο ±6.2% στη θερμοκρασία εξόδου του αέρα, ±8.1% στην απόλυτη υγρασία εξόδου του αέρα και ±1.4% στη θερμοκρασία εξόδου του διαλύματος.

Στη συνέχεια, εξετάζονται οι σημαντικές σχεδιαστικές παράμετροι για την απόδοση της αναγέννησης, όπως η επίδραση της παροχής μάζας αέρα και διαλύματος, η απόλυτη υγρασία εισόδου, θερμοκρασία εισόδου του διαλύματος και η συγκέντρωση εισόδου του διαλύματος. Εκτενή ανάλυση ευαισθησίας πραγματοποιείται με σκοπό να αναδείξει τις εξαρτημένες μεταβλητές που επηρεάζουν καθοριστικά τις τιμές εξόδου του συστήματος. Η ανάλυση δείχνει ότι οι τιμές εξόδου του αναγεννητή επηρεάζονται σημαντικά από τη συγκέντρωση εισόδου προσροφητικού διαλύματος και ελάχιστα από την απόλυτη υγρασία εισόδου. Υψηλή απόδοση της αναγέννησης συμβαίνει κάτω από υψηλές τιμές θερμοκρασίας εισόδου του διαλύματος. Επίσης, όσο αυξάνεται η παροχή μάζας προσροφητικού διαλύματος τόσο ευνοείται το φαινόμενο της μεταφοράς μάζας, γεγονός που είναι επιθυμητό στην αναγέννηση. Αυτό συμβαίνει γιατί όταν αυξάνεται η παροχή μάζας του διαλύματος, μειώνεται ο χρόνος παραμονής του διαλύματος στον αναγεννητή και έτσι, το διάλυμα δεν προλαβαίνει να ψυχθεί. Καθώς λοιπόν η θερμοκρασία του παραμένει σε υψηλά επίπεδα, και είναι μεγαλύτερη από αυτήν του αέρα, η επιφανειακή τάση ατμών του διαλύματος παραμένει μεγαλύτερη από την τάση ατμών του νερού στον αέρα. Η διαφορά αυτή, που προκαλεί τη μεταφορά μάζας, παραμένει σε υψηλά επίπεδα και συνεπώς, το φαινόμενο της μεταφοράς μάζας ευνοείται.

Η παροχή μάζας αέρα φαίνεται να επιδρά καθοριστικά στην απόδοση της αναγέννησης με αναλογική εξάρτηση, μόνο όταν πρόκειται να επιλεχθεί προσροφητικό υλικό μεταξύ του CaCl₂ και του LiBr. Χαμηλή συγκέντρωση εισόδου προσροφητικού διαλύματος είναι επιθυμητή για αποδοτική αναγέννηση, όμως το είδος του προσροφητικού διαλύματος που χρησιμοποιείται έχει μεγαλύτερη σημασία από ότι η συγκέντρωσή του. Συγκεκριμένα, η θερμότητα που απελευθερώνεται από το διάλυμα CaCl₂ χρησιμοποιείται για την εξάτμιση της πλεονάζουσας υγρασίας του, ενώ η θερμότητα που απελευθερώνεται από το διάλυμα LiCl χρησιμοποιείται για την θέρμανση του αέρα. Συνεπώς, η υψηλή τάση ατμών του CaCl₂, του προσδίδει ένα σημαντικό προβάδισμα έναντι των LiBr and LiCl. Συμπερασματικά, υψηλή απόδοση στην αναγέννηση επιτυγχάνεται με μεγάλη παροχή μάζας διαλύματος, μεγάλη παροχή μάζας αέρα, μεγάλη θερμοκρασία εισόδου του διαλύματος, μικρή συγκέντρωση εισόδου του διαλύματος και CaCl₂ ως το προσροφητικό διάλυμα.

ΜΗ ΑΔΙΑΒΑΤΙΚΟ ΜΟΝΤΕΛΟ ΜΕ ΣΩΛΗΝΟΕΙΔΗ ΕΝΑΛΛΑΚΤΗ (ΚΕΦΑΛΑΙΟ 5)

Το κεφάλαιο αυτό αφορά τη μελέτη του αφυγραντή και αναγεννητή, ως μη αδιαβατική συσκευή με σωληνοειδή εναλλάκτη. Ο όρος μη αδιαβατικός για τον αφυγραντή και τον αναγεννητή χρησιμοποιείται για να δείξει ότι υπάρχει συναλλαγή ενέργειας μεταξύ των διατάξεων αυτών και του περιβάλλοντος. Δηλαδή, τόσο κατά την αφύγρανση όσο και κατά την αναγέννηση υπάρχει θερμική αλληλεπίδραση με εξωτερική πηγή θερμότητας. Για την περίπτωση της αφύγρανσης, χρησιμοποιείται ψυχρό νερό που προέρχεται από έναν πύργο ψύξης και εισάγεται στον αφυγραντή μέσω ενός σωληνοειδούς εναλλάκτη θερμότητας. Για την περίπτωση της αναγέννησης, χρησιμοποιείται θερμό νερό που προέρχεται από έναν πύργο ψύξης και εισάγεται στον αφυγραντή μέσω ενός σωληνοειδούς εναλλάκτη θερμότητας. Για την περίπτωση της αναγέννησης, χρησιμοποιείται θερμό νερό που προέρχεται από μία πηγή θερμότητας και εισάγεται στον αναγεννητή μέσω ενός σωληνοειδούς εναλλάκτη θερμότητας. Στην περίπτωση αυτή, η διάταξη του αφυγραντή και του αναγεννητή περιέχει σωληνοειδή εναλλάκτη θερμότητας και μάζας μεταξύ του αέρα, του προσροφητικού υλικού και του νερού, με τον αέρα και το προσροφητικό υλικό να βρίσκονται σε άμεση επαφή και με το προσροφητικό υλικό και το νερό να βρίσκονται σε έμμεση επαφή. Χάριν ομοιομορφίας, τα στάδια της μελέτης είναι όμοια με εκείνα του προηγούμενου κεφαλαίου.

Αρχικά, περιγράφεται η διαδικασία της ανάπτυξης του μαθηματικού μοντέλου του αφυγραντή, το οποίο έχει τη δυνατότητα να προσομοιώνει την απόδοση της διάταξης υπό μεταβλητές συνθήκες λειτουργίας. Στη συνέχεια, παρουσιάζεται η επικύρωση του αναπτυχθέντος μοντέλου του αφυγραντή με πειραματικά δεδομένα από την ήδη υπάρχουσα βιβλιογραφία. Από τη σύγκρισή τους, προκύπτει ότι οι προβλέψεις του μαθηματικού μοντέλου βρίσκονται σε καλή συμφωνία με τα πειραματικά δεδομένα.

Ακολούθως, παρουσιάζεται η παραμετρική ανάλυση που διεξήχθη για να εντοπιστούν οι παράμετροι που επηρεάζουν την λειτουργία του αφυγραντή. Το ψυκτικό αποτέλεσμα του αέρα εξαρτάται καθοριστικά από την θερμοκρασία εισόδου του αέρα, ως εξής: καθώς αυξάνεται η θερμοκρασία εισόδου του αέρα, περισσότερη αισθητή θερμότητα μεταφέρεται στο κύκλωμα του νερού, οπότε η ψυκτική απόδοση αυξάνεται. Το γεγονός αυτό δικαιολογεί την καλύτερη απόδοση των μη αδιαβατικών αφυγραντικών συσκευών, έναντι των αδιαβατικών. Στις αδιαβατικές συσκευές, όπου δεν υπάρχει σωληνοειδής εναλλάκτης, αυξάνεται η θερμοκρασία του προσροφητικού διαλύματος με την πάροδο του χρόνου και έτσι, αυξάνεται η τάση ατμών του. Μειώνεται η κινητήριος δύναμη για μεταφορά μάζας και έτσι, μειώνεται τόσο η ψυκτική όσο και αφυγραντική απόδοση της διάταξης. Καθώς αυξάνεται η απόλυτη υγρασία εισόδου του αέρα, η αφυγραντική

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τάσης ατμών μεταξύ του αέρα και του προσροφητικού διαλύματος αυξάνεται. Συνεπώς, ενισχύεται το φαινόμενο μεταφοράς μάζας και έτσι, αυξάνεται η απόδοση της συσκευής.

Η απόδοση του μη αδιαβατικού αναγεννητή μελετάται στη συνέχεια. Τα αποτελέσματα του μαθηματικού μοντέλου που αναπτύσσεται συγκλίνουν με τα πειραματικά δεδομένα της βιβλιογραφίας. Ειδικότερα, η απόκλιση μεταξύ των προβλέψεων του μοντέλου και των πειραματικών αποτελεσμάτων δεν ξεπερνά το 10%, για όλες τις παραμέτρους που εξετάστηκαν.

Τα αποτελέσματα της ανάλυσης ευαισθησίας δείχνουν ότι οι συνθήκες εξόδου του αναγεννητή εξαρτώνται κυρίως από τη συγκέντρωση εισόδου του διαλύματος, την απόλυτη υγρασία εισόδου του αέρα, τη θερμοκρασία εισόδου του θερμού νερού, τη θερμοκρασία εισόδου του προσροφητικού διαλύματος και την παροχή μάζας του προσροφητικού διαλύματος. Ο ρυθμός ύγρανσης του αέρα, που αποτελεί ένδειξη της απόδοσης, μεταβάλλεται ανάλογα με τη θερμοκρασία εισόδου του διαλύματος και του νερού και με τη παροχή μάζας του αέρα. Μεταβάλλεται αντιστρόφως ανάλογα με την απόλυτη υγρασία εισόδου του αέρα, ενώ μεταβάλλεται ελάχιστα με την παροχή μάζας του νερού. Η απόδοση της αναγέννησης μεταβάλλεται ανάλογα με την απόλυτη υγρασία εισόδου του αέρα, του διαλύματος, αντιστρόφως ανάλογα με τη θερμοκρασία εισόδου του διαλύματος, αντιστρόφως ανάλογα με τος δου του αέρα και τη συγκέντρωση εισόδου του διαλύματος, ενώ παραμένει σχεδόν αμετάβλητη με τη θερμοκρασία εισόδου του αέρα, του νερού. Τα αποτελέσματα της μελέτης δείχνουν ότι:

- Η αύξηση της θερμοκρασίας εισόδου του αέρα προκαλεί αύξηση του ρυθμού ύγρανσης του αέρα και μικρή μείωση της απόδοσης της αναγέννησης, για όλα τα είδη διαλυμάτων και όλους τους τύπους ροών.
- Η απόδοση της αναγέννησης και ο ρυθμός ύγρανσης του αέρα μειώνεται όταν η απόλυτη υγρασία εισόδου του αέρα αυξάνεται. Αυτό συμβαίνει γιατί η αύξηση της απόλυτης υγρασίας σημαίνει αύξηση της τάσης ατμών του νερού στον αέρα και έτσι, η διαφορά της τάσης ατμών του διαλύματος με τον αέρα μειώνεται. Συνεπώς, τα φαινόμενα μεταφοράς μάζας εξασθενούν, επηρεάζοντας ανάλογα την απόδοση και τον ρυθμό ύγρανσης του αέρα.
- Υψηλές τιμές θερμοκρασίας εισόδου του προσροφητικού διαλύματος προκαλούν αύξηση στην απόδοση της αναγέννησης, για όλα τα είδη διαλυμάτων και όλους τους τύπους ροών.
- Η απόδοση και ο ρυθμός ύγρανσης του αέρα είναι αντιστρόφως ανάλογα μεγέθη με την συγκέντρωση εισόδου του διαλύματος, για όλα τα είδη διαλυμάτων. Όμως, η απόδοση και ο ρυθμός ύγρανσης του αέρα δεν εξαρτώνται από τη μεταβολή της κατεύθυνσης των ροών.
 Αυτό εξηγείται από το γεγονός ότι η τάση ατμών του νερού σε κάθε προσροφητικό διάλυμα εξαρτάται από τη συγκέντρωση. Επομένως, καθώς η συγκέντρωση εισόδου του διαλύματος

αυξάνεται, η τάση ατμών του μειώνεται. Τα φαινόμενα μεταφοράς μάζας εξασθενούν, επηρεάζοντας αρνητικά την απόδοση της αναγέννησης και τον ρυθμό ύγρανσης του αέρα.

- Ο συνδυασμός της χρήσης CaCl₂ ως το προσροφητικό διάλυμα και της αντιρροής (αέρας προς τα πάνω, διάλυμα προς τα κάτω, νερό προς τα κάτω) δίνει τα καλύτερα αποτελέσματα ως προς την απόδοση και τον ρυθμό ύγρανσης του αέρα, για όλες τις περιπτώσεις, καθώς μπορεί να αυξήσει την απόλυτη υγρασία εξόδου του αέρα κατά 166%.
- Αντίθετα, η χρήση LiCl στον αναγεννητή δίνει τα πιο αποθαρρυντικά αποτελέσματα, καθώς αυξάνει την απόλυτη υγρασία εξόδου του αέρα μόνο κατά 67% στην αντιρροή (αέρας προς τα πάνω, διάλυμα προς τα κάτω, νερό προς τα κάτω) και μόνο κατά 65% στην αντιρροή (αέρας προς τα πάνω, διάλυμα προς τα κάτω, νερό προς τα πάνω)
- Η κατεύθυνση της ροής είναι λιγότερο σημαντική από την επιλογή του είδους του προσροφητικού διαλύματος. Το LiCl έχει ξεκάθαρα μειονεκτήματα σε σχέση με τα υπόλοιπα δύο διαλύματα στην αναγέννηση.

ΠΕΙΡΑΜΑΤΙΚΗ ΔΙΑΤΑΞΗ ΚΛΙΜΑΤΙΣΤΙΚΟΥ ΣΥΣΤΗΜΑΤΟΣ ΑΝΟΙΚΤΟΥ ΚΥΚΛΟΥ ΜΕ ΥΓΡΟ ΠΡΟΣΡΟΦΗΤΙΚΟ ΥΛΙΚΟ (ΚΕΦΑΛΑΙΟ 6)

Η πειραματική διάταξη του κλιματιστικού συστήματος ανοικτού κύκλου με υγρό προσροφητικό υλικό που εγκαταστάθηκε στο Εργαστήριο Εφαρμοσμένης Θερμοδυναμικής στη Σχολή Μηχανολόγων Μηχανικών του ΕΜΠ περιγράφεται στο κεφάλαιο αυτό.

Το κεφάλαιο διαιρείται σε δύο τμήματα: το πρώτο παρέχει το θεωρητικό υπόβαθρο των φαινομένων μεταφοράς θερμότητας και μάζας που συμβαίνουν στην πειραματική διάταξη και το δεύτερο, παρέχει την περιγραφή της πειραματικής διαδικασίας και παρουσιάζει τα αποτελέσματα από τη διεξαγωγή των πειραμάτων.

Εφόσον η πειραματική διάταξη αποτελείται από μη αδιαβατικό αφυγραντή και αναγεννητή με πλακοειδή εναλλάκτη, το θεωρητικό μέρος περιλαμβάνει την ανάπτυξη του υπολογιστικού μοντέλου για τον αφυγραντή ως μη αδιαβατική συσκευή με πλακοειδή εναλλάκτη. Ο όρος μη αδιαβατικός χρησιμοποιείται για να δείξει ότι υπάρχει συναλλαγή ενέργειας μεταξύ της διάταξης αυτής και του περιβάλλοντος. Υπάρχει δηλαδή θερμική αλληλεπίδραση με ψυχρό νερό που προέρχεται από έναν πύργο ψύξης και εισάγεται στον αφυγραντή μέσω ενός πλακοειδούς εναλλάκτη θερμότητας. Στη συνέχεια, παρουσιάζεται η επικύρωση του αναπτυχθέντος μοντέλου του αφυγραντή με πειραματικά δεδομένα από την ήδη υπάρχουσα βιβλιογραφία. Από τη σύγκρισή τους, προκύπτει ότι οι προβλέψεις του μαθηματικού μοντέλου βρίσκονται σε συμφωνία με τα πειραματικά δεδομένα της βιβλιογραφίας, εντός 12%.

Η παραμετρική ανάλυση που ακολουθεί δείχνει ότι οι σημαντικότερες παράμετροι που επηρεάζουν την λειτουργία του αφυγραντή είναι η παροχή μάζας αέρα, η απόλυτη υγρασία εισόδου, η θερμοκρασία εισόδου του διαλύματος και η θερμοκρασία εισόδου του νερού. Ειδικότερα,

- Αύξηση της παροχής μάζας αέρα προκαλεί αύξηση στον ρυθμό αφύγρανσης του αέρα. Αυτό συμβαίνει γιατί ο χρόνος παραμονής του αέρα μέσα στη διάταξη μειώνεται, επιφέροντας μείωση στον συντελεστή μεταφοράς μάζας, αύξηση στην απόλυτη υγρασία εξόδου και κατά συνέπεια, μείωση στην απόδοση. Την ίδια στιγμή, αύξηση της απόλυτης υγρασία εξόδου σημαίνει μεγαλύτερη διαφορά στην τάση ατμών του διαλύματος και του αέρα, και συνεπώς, αύξηση του ρυθμού αφύγρανσης.
- Αύξηση της απόλυτης υγρασίας εισόδου αυξάνει το ρυθμό αφύγρανσης, αφού αυξάνεται η τάση ατμών του νερού στον αέρα και η διαφορά της τάσης ατμών του αέρα με το προσροφητικό διάλυμα αυξάνεται. Συνεπώς, η μεταφορά μάζας ενισχύεται και ο ρυθμός αφύγρανσης αυξάνεται.

- Η αύξηση της θερμοκρασίας εισόδου του ψυχρού νερού επιφέρει μείωση τόσο στην απόδοση της αφύγρανσης, όσο και στο ρυθμό της αφύγρανσης, για όλους τους τύπους διαλυμάτων. Αυτό συμβαίνει γιατί ο ρυθμός μεταφοράς θερμότητας από το διάλυμα στο νερό μειώνεται, οπότε η τάση ατμών του διαλύματος αυξάνεται. Έτσι, εξασθενεί το φαινόμενο μεταφοράς μάζας και κατά συνέπεια, μειώνεται ο ρυθμός αφύγρανσης και η απόδοση.
- Αυξάνοντας την θερμοκρασία εισόδου του προσροφητικού διαλύματος μειώνεται ο ρυθμός αφύγρανσης. Ομοίως με την προηγούμενη μεταβολή, η αύξηση της θερμοκρασίας του διαλύματος αυξάνει την τάση ατμών του. Έτσι, εξασθενεί το φαινόμενο μεταφοράς μάζας και η απόλυτη υγρασία εξόδου του αέρα αυξάνεται. Συνεπώς, μειώνεται ο ρυθμός αφύγρανσης και η απόδοση.
- Ο ρυθμός αφύγρανσης είναι ανάλογος με τη παροχή μάζας του διαλύματος. Καθώς αυξάνεται η παροχή μάζας, μειώνεται ο χρόνος επαφής του διαλύματος με τον αέρα και έτσι, η μεταβολή της συγκέντρωσης και της θερμοκρασίας του διαλύματος μέσα στον αφυγραντή εξασθενεί. Αυτό έχει ως αποτέλεσμα η τάση ατμών του διαλύματος να παραμένει σταθερή, οπότε η διαφορά της τάσης ατμών του διαλύματος και του αέρα αυξάνεται. Έτσι, η απόλυτη υγρασία εξόδου του αέρα μειώνεται και ο ρυθμός αφύγρανσης αυξάνεται.
- Η απόδοση της αφύγρανσης είναι ανάλογη με τη παροχή μάζας του διαλύματος.
 Αυξάνοντας τη παροχή μάζας, ο συντελεστής μεταφοράς μάζας αυξάνεται, επιφέροντας περαιτέρω αύξηση στην απόδοση της αφύγρανσης.
- Ο αφυγραντής με ομορροή και LiCl ως το προσροφητικό διάλυμα έχει την καλύτερη απόδοση, καθώς μειώνει την απόλυτη υγρασία του αέρα κατά 31%, όταν ο αφυγραντής με το LiBr μπορεί να μειώσει την απόλυτη υγρασία του αέρα κατά 12% μόνο, υπό ίδιες αρχικές συνθήκες.
- Η αντιρροή παράγει πολύ πιο ξηρό αέρα από την ομορροή, καθώς η αντιρροή μειώνει την απόλυτη υγρασία του αέρα κατά 44%, ενώ η ομορροή κατά 31%.

ΕΚΤΕΝΗΣ ΠΕΡΙΛΗΨΗ

Το δεύτερο μέρος του κεφαλαίου ξεκινάει με την περιγραφή της πειραματικής διάταξης που έχει εγκατασταθεί στο Εργαστήριο Εφαρμοσμένης Θερμοδυναμικής ΕΜΠ. Σχολιάζεται η λειτουργία της διάταξης, παραθέτονται τα σχέδια της εγκατάστασης και του μετρητικού συστήματος και στη συνέχεια, αναπτύσσεται η υλοποίηση των πειραμάτων. Συνολικά, γίνονται 8 πειράματα αφύγρανσης και 12 πειράματα αναγέννησης. Ακολουθεί η αξιολόγηση των πειραματικών μετρήσεων με εκτεταμένη ανάλυση. Χρησιμοποιούνται δεδομένα από 4 διαφορετικά πειράματα αφύγρανσης και αναγέννησης, αναλύονται τα αποτελέσματα και διαπιστώνεται η αξιοπιστία των μετρήσεων. Δίνεται πίνακας με τα πειραματικά δεδομένα του συνόλου των πειραμάτων της αφύγρανσης και της αναγέννησης.

Για τα πειραματικά δεδομένα της αφύγρανσης, ο ισολογισμός ενέργειας δίνει απόκλιση που κυμαίνεται από -5.36% έως 19.84%, ενώ για την αναγέννηση το ποσοστό αυτό είναι από -3.02% έως 10.57%.

Τα πειραματικά αποτελέσματα δείχνουν ότι ο ρυθμός αφύγρανσης του αέρα για το πειραματικό σύστημα κυμαίνεται από 6.99 έως 11.90 kg νερού/ώρα, ανάλογα με τις εξωτερικές συνθήκες και τις συνθήκες λειτουργίας της μονάδας. Ο ρυθμός ύγρανσης του αέρα (αναγεννητής) κυμαίνεται από 3.55 έως 12.21 kg νερού/ώρα. Η κατανάλωση ενέργειας που σχετίζεται με τον αναγεννητή κυμαίνεται από 6.93 kWh έως 26.22 kWh. Η απαιτούμενη θερμοκρασία θερμού νερού κυμαίνεται από 46.11 °C έως 72.86°C. Ακόμη και με θερμοκρασίες θερμού νερού στους 46.11°C, ο ρυθμός ύγρανσης του αέρα είναι σε ικανοποιητικά επίπεδα, της τάξης των 3.55 kg νερού/ώρα.

Ουσιώδη σημασία για την εργασία έχει η σύγκριση των πειραματικών μετρήσεων με το επικυρωμένο μαθηματικό μοντέλο. Η σύγκριση έγινε μετά από την προσαρμογή των μαθηματικών μοντέλων του αφυγραντή και του αναγεννητή στις αρχικές συνθήκες του κάθε πειράματος και βέβαια, στα γεωμετρικά χαρακτηριστικά της κάθε διάταξης. Τα αποτελέσματα δείχνουν μία λογική απόκλιση μεταξύ της προβλεπόμενης τιμής του μοντέλου και της πειραματικής τιμής. Συνολικά, οι αποκλίσεις κυμαίνονται από -17.39% έως 14.51% για την αφύγρανση και από -20.56% έως 15.63% για την αναγέννηση.

Στην περίπτωση της αφύγρανσης, η πειραματική διάταξη παρήγαγε πιο ξηρό και πιο θερμό αέρα από αυτόν που προβλέπει το μαθηματικό μοντέλο, πιο αραιό και πιο θερμό διάλυμα και πιο θερμό νερό. Στην περίπτωση της αναγέννησης, η πειραματική διάταξη παρήγαγε πιο ξηρό και πιο ψυχρό αέρα από αυτόν που προβλέπει το μαθηματικό μοντέλο, πιο ισχυρό και πιο θερμό διάλυμα και πιο ψυχρό νερό. Η οπτικοποίηση των δεδομένων του αέρα και στις δύο διαδικασίες σε ψυχρομετρικό χάρτη δείχνει ότι το μαθηματικό μοντέλο προβλέπει καλύτερα την απόλυτη υγρασία εξόδου του αέρα, παρά τη θερμοκρασία εξόδου του. Λαμβάνοντας υπόψη τις δυσκολίες που αντιμετωπίστηκαν

κατά τη διάρκεια των πειραμάτων, συμπεραίνεται ότι το μοντέλο προβλέπει τα πειραματικά δεδομένα με ικανοποιητική ακρίβεια.

Αναμφίβολα, η λειτουργία της πειραματικής διάταξης είναι χρονοβόρα και γεμάτη προκλήσεις. Εξωτερικοί, αστάθμητοι παράγοντες προκύπτουν συνεχώς, που οφείλουν να αντιμετωπιστούν με επιμονή, υπομονή και αμεσότητα. Παρόλες τις δυσκολίες, οι εμπειρίες και οι γνώσεις που αποκτώνται από την πειραματική διαδικασία είναι ανεκτίμητες.

Ακολουθούν προτάσεις βελτιστοποίησης της πιλοτικής μονάδας και της πειραματικής διαδικασίας.

ΣΥΜΠΕΡΑΣΜΑΤΑ ΚΑΙ ΜΕΛΛΟΝΤΙΚΗ ΕΡΕΥΝΑ (ΚΕΦΑΛΑΙΟ 7)

Το αντικείμενο της έρευνας αυτής ήταν η μελέτη των φαινομένων μεταφοράς θερμότητας και μάζας μεταξύ του αέρα, του αφυγραντικού υλικού και του νερού που συντελούνται στα κλιματιστικά συστήματα ανοικτού εξατμιστικού κύκλου με υγρά προσροφητικά υλικά. Η εργασία υλοποιήθηκε τόσο σε θεωρητικό όσο και σε πειραματικό επίπεδο και εξέτασε την απόδοση του αφυγραντή και του αναγεννητή ως προς τον ρυθμό αφύγρανσης αέρα και την ενεργειακή κατανάλωση.

Ειδικότεροι στόχοι της εργασίας ήταν οι εξής:

- Η απόκτηση του θεωρητικού υπόβαθρου σχετικά με τα διαθέσιμα προσροφητικά υλικά, τις
 συσκευές αφύγρανσης και αναγέννησης που χρησιμοποιούν υγρά προσροφητικά υλικά και
 τις διατάξεις των κλιματιστικών συστημάτων.
- Η ανάπτυξη μαθηματικών μοντέλων για την εκτίμηση της απόδοσης των αφυγραντικών
 διατάξεων που χρησιμοποιούν υγρά προσροφητικά υλικά.
- Η ανάπτυξη μαθηματικών μοντέλων για την εκτίμηση της απόδοσης των αναγεννητικών
 διατάξεων των υγρών προσροφητικών υλικών.
- Η εξέταση, μέσω εκτενούς παραμετρικής ανάλυσης, των σημαντικότερων παραμέτρων που επηρεάζουν την ενεργειακή απόδοση του αφυγραντή και του αναγεννητή.
- Η πειραματική αξιολόγηση των κλιματιστικών συστημάτων ανοικτού τύπου με υγρά προσροφητικά υλικά.
- Η ανάδειξη μεθόδων βελτιστοποίησης των συστημάτων αυτών, βασιζόμενη στην θεωρητική μελέτη και στην πειραματική διαδικασία.
Η καινοτομία της εργασίας έγκειται στους παρακάτω τομείς έρευνας:

- Επίλυση φαινομένων μεταφοράς θερμότητας και μάζας του αφυγραντή και του αναγεννητή
 χρησιμοποιώντας σύστημα συνήθων διαφορικών εξισώσεων.
- Ανάπτυξη αλγόριθμου επίλυσης των συντελεστών μεταφοράς θερμότητας και μάζας σε
 - Αδιαβατικές συσκευές με πληρωτικό υλικό
 - Μη αδιαβατικές συσκευές με σωληνοειδή εναλλάκτη
 - Μη αδιαβατικές συσκευές με πλακοειδή εναλλάκτη
- Ανάπτυξη μαθηματικών μοντέλων για την προσομοίωση της λειτουργίας των
 - Αφυγραντή και αναγεννητή ως αδιαβατικές συσκευές με πληρωτικό υλικό
 - Αφυγραντή και αναγεννητή ως μη αδιαβατικές συσκευές με σωληνοειδή εναλλάκτη
 - Αφυγραντή και αναγεννητή ως μη αδιαβατικές συσκευές με πλακοειδή εναλλάκτη
- Σύγκριση και επικύρωση των μαθηματικών μοντέλων με πειραματικά δεδομένα από τη βιβλιογραφία.
- Συγκριτική ανάλυση και παραμετρικές προσομοιώσεις του αφυγραντή και του αναγεννητή με
 τις ακόλουθες εξαρτημένες μεταβλητές
 - Είδος προσροφητικού διαλύματος
 - Κατάσταση εισόδου του αέρα
 - Κατάσταση εισόδου του προσροφητικού διαλύματος
 - Κατάσταση εισόδου του νερού
- Ανάπτυξη μαθηματικού μοντέλου για την προσομοίωση των κλιματιστικών συστημάτων
 ανοικτού κύκλου με υγρό προσροφητικό υλικό
- Διεξαγωγή πειραμάτων αφύγρανσης και αναγέννησης με τις ακόλουθες εξαρτημένες
 μεταβλητές:
 - Κατάσταση εισόδου του αέρα
 - Κατάσταση εισόδου του προσροφητικού διαλύματος
 - ο Κατάσταση εισόδου του νερού

Η διατριβή ξεκινάει με μία εκτεταμένη βιβλιογραφική επισκόπηση των μεθόδων επίλυσης των δύο θεμελιωδών λειτουργιών των κλιματιστικών συστημάτων ανοικτού κύκλου με υγρά προσροφητικά υλικά: της αφύγρανσης του αέρα και της αναγέννησης του αφυγραντικού διαλύματος. Περιγράφονται αναλυτικά τα υπολογιστικά μοντέλα που έχουν αναπτυχθεί από τους ερευνητές, οι βασικές υποθέσεις που έγιναν και τα συμπεράσματα που προέκυψαν. Υπάρχουν ποίκιλλες έρευνες σχετικά με την αφύγρανση και την αναγέννηση, με την πρώτη να αναφέρεται το 1969. Η εξέλιξη της έρευνας από τότε είναι μεγάλη: έχουν αναπτυχθεί αρκετά μοντέλα για την προσομοίωση των κλιματιστικών αυτών συστημάτων. Οι περισσότερες εργασίες επικεντρώνονται περισσότερο στην εκτενής περιλήψη

αντιρροή και λιγότερο στην ομορροή ή σταυρωτή ροή, καθώς η αντιρροή έχει αποδειχθεί ότι έχει την καλύτερη απόδοση. Υπάρχουν μερικές πειραματικές εργασίες οι οποίες συγκρίνουν τα θεωρητικά μοντέλα που έχουν αναπτυχθεί και διεξάγουν συνοπτική παραμετρική ανάλυση. Παρόλα αυτά, η επίδραση ορισμένων παραγόντων που επηρεάζουν την απόδοση τέτοιων συστημάτων παραμένει άγνωστη. Περαιτέρω έρευνα και πειραματικά δεδομένα απαιτούνται για τη κατανόηση της λειτουργίας των συστημάτων αυτών, τη βελτίωση της συνολικής τους απόδοσης και συνεπώς,

Η θερμοδυναμική ανάλυση των συστημάτων ανοικτού εξατμιστικού κύκλου με υγρά προσροφητικά υλικά δεν θα μπορούσε να μην ξεκινήσει από τις θερμοδυναμικές ιδιότητες των προσροφητικών μέσων. Καθώς η σύγκριση της απόδοσης των ποικίλλων διαθέσιμων προσροφητικών διαλυμάτων δεν αποτελεί αντικείμενο της παρούσας Διατριβής, μόνο τα συνήθη προσροφητικά διαλύματα επιλέχθηκαν να μελετηθούν θερμοδυναμικά. Έτσι, τα υδατικά διαλύματα του Χλωριούχου Λιθίου (LiBr) και Χλωριούχου Ασβεστίου (CaCl₂) μελετήθηκαν εκτενώς. Επιλέχθηκαν οι θερμοδυναμικές τους ιδιότητες που είναι απαραίτητες για το σχεδιασμό κλιματιστικών συστημάτων, όπως όριο διαλυτότητας, τάση ατμών, επιφανειακή τάση, δυναμική συνεκτικότητα, θερμική αγωγιμότητα, ειδική θερμοχωρητικότητα και διαφορική ενθαλπία διάλυσης. Τα δεδομένα της μελέτης προήλθαν από εκτενή βιβλιογραφική αναφορά.

Η απλούστερη διάταξη του αφυγραντή και του αναγεννητή, η αδιαβατική, μελετήθηκε πρώτα, αναπτύσσοντας το υπολογιστικό μοντέλο προσομοίωσης του αφυγραντή και του αναγεννητή. Οι προβλέψεις του μαθηματικού μοντέλου βρέθηκαν σε συμφωνία με τα πειραματικά δεδομένα της βιβλιογραφίας. Βρέθηκε ότι ο ρυθμός της αφύγρανσης μεταβάλλεται ανάλογα με την απόλυτη υγρασία εισόδου, συγκέντρωση εισόδου διαλύματος, παροχή μάζας αέρα και παροχής μάζας διαλύματος και αντιστρόφως ανάλογα με τη θερμοκρασία εισόδου διαλύματος. Η απόδοση της αφύγρανσης είναι ανάλογη με την παροχή μάζας αέρα. Υψηλή απόδοση στη αφύγρανση επιτυγχάνεται κάτω από μεγάλη απόλυτη υγρασία περιβάλλοντος, μικρή παροχή μάζας αέρα και διάλυμα LiCl ως προσροφητικό μέσο. Υψηλή απόδοση στην αναγέννηση επιτυγχάνεται με μεγάλη παροχή μάζας διαλύματος, μεγάλη παροχή μάζας αέρα, μεγάλη θερμοκρασία εισόδου του διαλύματος, μικρή συγκέντρωση εισόδου του διαλύματος και CaCl₂ ως το προσροφητικό διάλυμα.

Στη συνέχεια, μελετήθηκε ο αφυγραντής και ο αναγεννητής, ως μη αδιαβατικές συσκευές με σωληνοειδή εναλλάκτη. Περιεγράφηκαν εκτενώς τα μαθηματικά μοντέλα, τα οποία στη συνέχεια επικυρώθηκαν με πειραματικά δεδομένα της βιβλιογραφίας. Η παραμετρική ανάλυση έδειξε ότι καθώς αυξάνεται η απόλυτη υγρασία εισόδου του αέρα, η αφυγραντική απόδοση και ο ρυθμός αφύγρανσης του αέρα αυξάνεται. Καθώς αυξάνεται η θερμοκρασία εισόδου του αέρα η ψυκτική

απόδοση αυξάνεται. Για τον αναγεννητή, ο συνδυασμός της χρήσης CaCl₂ ως το προσροφητικό διάλυμα και της αντιρροής (αέρας προς τα πάνω, διάλυμα προς τα κάτω, νερό προς τα κάτω) έδωσε τα καλύτερα αποτελέσματα ως προς την απόδοση και τον ρυθμό ύγρανσης του αέρα, για όλες τις περιπτώσεις. Αντίθετα, η χρήση LiCl στον αναγεννητή είχε ξεκάθαρα μειονεκτήματα σε σχέση με τα υπόλοιπα δύο διαλύματα.

Στη συνέχεια, συζητήθηκε το μοντέλο που αναπτύχθηκε για τον αφυγραντή ως μη αδιαβατική συσκευή με πλακοειδή εναλλάκτη, το οποίο στη συνέχεια επικυρώθηκε με πειραματικά δεδομένα της βιβλιογραφίας. Οι προσομοιώσεις έδειξαν ότι ο ρυθμός αφύγρανσης είναι ανάλογος με την απόλυτη υγρασία εισόδου, την παροχή μάζας αέρα και την παροχή μάζας διαλύματος, ενώ είναι αντιστρόφως ανάλογος με τη θερμοκρασία εισόδου του διαλύματος και τη θερμοκρασία εισόδου του νερού. Ο αφυγραντής με αντιρροή και LiCl ως το προσροφητικό διάλυμα είχε την καλύτερη απόδοση από όλες τις περιπτώσεις που εξετάστηκαν.

Ακολούθησε η περιγραφή της πειραματικής διάταξης του κλιματιστικού συστήματος ανοικτού κύκλου με υγρό προσροφητικό υλικό, που εγκαταστάθηκε στο Εργαστήριο Εφαρμοσμένης Θερμοδυναμικής ΕΜΠ. Σχολιάστηκε η λειτουργία της διάταξης και έγιναν λεπτομερείς αναφορές για τα σχέδια της εγκατάστασης, του μετρητικού συστήματος και της υλοποίησης των πειραμάτων. Η αξιολόγηση των πειραματικών μετρήσεων, μέσω του ισολογισμού ενέργειας, έδειξε απόκλιση - 5.36% έως 19.84% για την αφύγρανση και -3.02% έως 10.57% για την αναγέννηση. Ο ρυθμός αφύγρανσης του αέρα κυμάνθηκε από 6.99 έως 11.90 kg νερού/ώρα, ανάλογα με τις εξωτερικές συνθήκες. Ο ρυθμός ύγρανσης του αέρα (αναγεννητής) κυμάνθηκε από 3.55 έως 12.21 kg νερού/ώρα. Η κατανάλωση ενέργειας που σχετίζεται με τον αναγεννητή ήταν από 6.93 kWh έως 26.22 kWh. Η απαιτούμενη θερμοκρασία θερμού νερού κυμάνθηκε από 46.11 °C έως 72.86°C. Ακόμη και με θερμοκρασίες θερμού νερού στους 46.11°C, ο ρυθμός ύγρανσης του αέρα είναι σε ικανοποιητικά επίπεδα, της τάξης των 3.55 kg νερού/ώρα.

Ουσιώδη σημασία για την εργασία είχε η σύγκριση των πειραματικών μετρήσεων με το επικυρωμένο μαθηματικό μοντέλο. Τα αποτελέσματα έδειξαν μία λογική απόκλιση μεταξύ της προβλεπόμενης τιμής του μοντέλου και της πειραματικής τιμής: από -17.39% έως 14.51% για την αφύγρανση και από -20.56% έως 15.63% για την αναγέννηση. Στην περίπτωση της αφύγρανσης, η πειραματική διάταξη παρήγαγε πιο ξηρό και πιο θερμό αέρα από αυτόν που προβλέπει το μαθηματικό μοντέλο, πιο αραιό και πιο θερμό διάλυμα και πιο θερμό νερό. Στην αναγέννηση, η πειραματική διάταξη παρήγαγε πιο ξηρό και πιο ψυχρό αέρα από αυτόν που προβλέπει το μαθηματικό μοντέλο, πιο ισχυρό και πιο θερμό διάλυμα και πιο ψυχρό νερό. Συμπερασματικά προκύπτει ότι το μαθηματικό μοντέλο προβλέπει καλύτερα την απόλυτη υγρασία εξόδου του αέρα, παρά τη θερμοκρασία εξόδου του.

Αναμφίβολα, η λειτουργία της πειραματικής διάταξης ήταν χρονοβόρα και γεμάτη προκλήσεις. Εξωτερικοί, αστάθμητοι παράγοντες προκύπταν συνεχώς, που όφειλαν να αντιμετωπιστούν με επιμονή, υπομονή και αμεσότητα. Παρόλες τις δυσκολίες, οι εμπειρίες και οι γνώσεις που αποκτήθηκαν από την πειραματική διαδικασία είναι ανεκτίμητες.

Παρόλο που στην εργασία αυτήν παρουσιάστηκαν ικανοποιητικά καινοτόμα στοιχεία, υπάρχουν ακόμη μεγάλα περιθώρια για έρευνα και ανάπτυξη των κλιματιστικών αυτών συστημάτων. Ενδεικτικά παραδείγματα είναι τα εξής:

 Μελέτη οικονομικής απόδοσης κλιματιστικού συστήματος ανοικτού κύκλου με προσροφητικό διάλυμα υποβοηθούμενο από θερμικό ηλιακό σύστημα.

Το θέμα αυτό μπορεί να περιλαμβάνει τη μελέτη της επίδρασης της επιφάνειας των συλλεκτών και της χωρητικότητας της δεξαμενής θερμού νερού στην ετήσια ηλιακή κάλυψη, της ετήσιας απόδοσης των συλλεκτών και του συνολικού κόστους στον κύκλο ζωής του συστήματος. Μπορεί επίσης να περιλαμβάνει υπολογισμούς σχετικά με το αρχικό κόστος, το ετήσιο κόστος λειτουργίας, την περίοδο απόσβεσης επένδυσης και τις εκπομπές CO₂.

 Θεωρητική προσέγγιση στην απόδοση κλιματιστικών συστημάτων ανοικτού κύκλου με προσροφητικό διάλυμα έναντι κλιματιστικών συμπίεσης ατμών.

Το θέμα αυτό σχετίζεται με τη σύγκριση της τεχνολογίας των προσροφητικών διαλυμάτων με τα συμβατικά συστήματα συμπίεσης ατμών. Απαιτεί τον ορισμό του συντελεστή απόδοσης, τόσο του ηλεκτρικού όσο και του θερμικού. Χρήσιμος θα ήταν και ο υπολογισμός του συνολικού ποσού ενέργειας που απαιτείται για την κάλυψη συγκεκριμένου λανθάνοντος και αισθητού φορτίου με τα δύο αυτά συστήματα.

Εκτίμηση της ετήσιας ενεργειακής, οικονομικής και περιβαλλοντικής απόδοσης
 κλιματιστικού συστήματος ανοικτού κύκλου με προσροφητικό διάλυμα.

Το θεωρητικό αυτό θέμα περιέχει τον υπολογισμό της κατανάλωσης πρωτογενούς ενέργειας, ετήσιας εξοικονόμησης λειτουργικού κόστους, περίοδος απόσβεσης επένδυσης. Οι παράμετροι αυτοί θα μπορούσαν επιπλέον να συγκριθούν με ένα συμβατικό κλιματιστικό σύστημα, προκειμένου να εξαχθούν χρήσιμα συμπεράσματα. Πειραματική αξιολόγηση της ενεργειακής απόδοσης κλιματιστικού συστήματος ανοικτού κύκλου με προσροφητικό διάλυμα στην περιοχή της Αθήνας.

Το θέμα αυτό προϋποθέτει την εγκατάσταση ενός τέτοιου συστήματος σε κτήριο της Αθήνας, με σκοπό να ληφθούν δεδομένα από πραγματικές συνθήκες λειτουργίας. Εστιάζει στην παραμετρική ανάλυση και ανάλυση ευαισθησίας, ώστε να εντοπιστούν και να ποσοτικοποιηθούν οι παράγοντες που επηρεάζουν την απόδοση του συστήματος.

 Ανάλυση της απόδοσης της ηλεκτροδιαλυτικής αναγέννησης με φωτοβολταϊκούς συλλέκτες (Photovoltaic electrodialysis (PV–ED) regeneration).

Το θέμα αυτό εισάγει τη νέα μέθοδο αναγέννησης προσροφητικών διαλυμάτων, η οποία απαιτεί ηλεκτρική - και όχι θερμική ενέργεια - και περιλαμβάνει ηλεκτροδιάλυση. Η μέθοδος αυτή βασίζεται στην αρχή ότι τα ιόντα μπορούν να κινηθούν, μέσω μεμβρανών, κάτω από την επίδραση ηλεκτρικού πεδίου. Η εφαρμογή ηλεκτρικού πεδίου κάνει τα ανιόντα και τα κατιόντα να διαπερνούν μεμβράνες ανταλλαγής ανιόντων και κατιόντων, αντίστοιχα. Η ροή αυτή προκαλεί αύξηση της συγκέντρωσης ιόντων σε συγκεκριμένη περιοχή και μείωση της συγκέντρωσης ιόντων σε άλλη περιοχή. Αυτή η διαδικασία μπορεί να διαχωρίσει το αραιό διάλυμα σε πυκνό διάλυμα και καθαρό νερό. Η προσέγγιση του θέματος αυτού μπορεί να είναι είτε θεωρητική είτε πειραματική. Καθώς δεν υπάρχουν μέχρι στιγμής πολλές ερευνητικές εργασίες, το θέμα αυτό θα ήταν ιδιαίτερα ενδιαφέρον.

 Ανάλυση της απόδοσης συστήματος αφύγρανσης αέρα με προσροφητικό διάλυμα και με υπερηχητικό ψεκασμό.

Το θέμα αυτό αποτελεί πρόσφατη ιδέα των ερευνητών που ασχολούνται με προσροφητικά διαλύματα. Η τεχνολογία αυτή αποσκοπεί στη βελτίωση της απόδοσης της αφύγρανσης, μέσω την ενσωμάτωσης ενός συστήματος υπερηχητικού ψεκασμού. Το σύστημα αυτό έχει την ιδιότητα να μετατρέπει το διάλυμα σε εξαιρετικά μικρής διατομής σταγονίδια και έτσι, να αυξάνει δραστικά την επιφάνεια μεταφοράς θερμότητας και μάζας. Η θεωρητική και ιδιαίτερα η πειραματική ανάλυση της απόδοσης του συστήματος αυτού θα ήταν χρήσιμη για την τεχνολογία.

ΠΑΡΑΡΤΗΜΑΤΑ (ΚΕΦΑΛΑΙΟ 8)

Στο κεφάλαιο αυτό παρουσιάζονται επιγραμματικά οι 9 δημοσιεύσεις του συγγραφέα, καθώς και οι 15 αναφορές που έχουν γίνει στις δημοσιεύσεις του συγγραφέα από άλλους ερευνητές έως σήμερα. Προς διευκόλυνση του αναγνώστη, δίνεται λίστα με τις εικόνες που υπάρχουν στην Διατριβή, καθώς και λίστα με τους πίνακες.

INTRODUCTION

1 INTRODUCTION

1.1 BACKGROUND AND MOTIVATION

In recent years, ozone layer depletion has created considerable public concern. The halogenated chlorofluorocarbons CFCs, which have provided refrigeration and air-conditioning for about 60 years, provide the major source of stratospheric chlorine and are a major threat to the ozone layer [1, 2].

These global environmental concerns along with improving air quality standards have all contributed to a change in design thinking. Overheating is now the predominant design consideration for new buildings, mainly because of the computer equipment increase, the inefficient lighting installations and often, architectural fashion. If development strategies compatible with economic and environmental sustainability are sought, then alternative refrigeration methods, such as liquid desiccant cooling systems, are the objective.

The air-conditioning load is the sum of the sensible and latent load [3, 4] and represents the 20-40% of the overall energy consumption in a building [5]. Dehumidification handles the latent load, while sensible cooling handles the sensible load [4, 6]. Traditional vapour compression equipment overcools the air-stream to provide cooling and dehumidification [7, 8, 9, 10, 11, 12]. Air-conditioning operates at a temperature colder than the supply air dew-point temperature [13, 14, 15], since for a typical humidity ratio of 0.007 this occurs at 9^oC [40], so the air needs reheating before entering indoors [16, 17]. Additionally, the ratio between refrigeration capacity and electrical energy required is not higher than unity [18].

Unlike conventional evaporative cooling systems, liquid desiccant cooling systems are an open heat driven cycle that provides indoor comfort even in hot and humid weather, without using refrigerants or relying on electrical energy [5, 18, 19, 20, 21]. Since the air does not have to be cooled below its dew point, utilization of waste heat or solar energy can be realized and that reduction in humidity, corrosion and microbiologic activity is achieved. The entire operation takes place at atmospheric pressure, eliminating the need for capital-intensive, pressure-sealed units [8]. This makes it a very environmentally friendly technology choice if properly designed, sized and managed in use.

The idea behind this research project originated by a question that came up when learning about open cycle desiccant dehumidification and cooling of air; is it possible to dehumidify air by a desiccant solution and chill it by evaporative cooling only and still get an acceptable indoor temperature? At that time, few written sources could give a straight answer. Shortly after, I got the opportunity to study at the Laboratory of Applied Thermal Engineering in the

National Technical University of Athens. Here the doors to the research world were opened for me and I was encouraged to conduct research on this topic. I am truly grateful for that.

1.2 RESEARCH OBJECTIVES AND INNOVATIONS

The general objective of this work is to study the heat and mass transfer processes between air, desiccant solution and water during absorption and desorption processes for applications of liquid desiccant air conditioning. The study is performed both theoretically and experimentally and investigates the performance of these processes in terms of energy efficiency, dehumidification rate and humidification rate.

Specific objectives of this study are:

- To acquire the theoretical background on the various liquid desiccant materials, components, devices and configurations,
- To develop mathematical models for the evaluation of the performance of the most commonly used liquid desiccant dehumidifiers,
- To develop mathematical models for the evaluation of the performance of most commonly used liquid desiccant regenerators,
- To investigate, through extensive parametric analysis, the most important parameters that affect the performance of liquid desiccant dehumidifiers and regenerators,
- To experimentally investigate the performance of liquid desiccant air conditioning systems,
- To find and propose optimization methods and procedures based on the implemented theoretical and experimental study.

The innovation of the implemented work lies on the following fields:

- Solving the heat and mass transfer phenomena during dehumidification and regeneration procedure via ordinary differential equations.
- Development of an algorithm for the calculation of heat and mass transfer coefficients in
 - Adiabatic beds
 - Non-adiabatic tubular heat exchangers
 - Non-adiabatic plate heat exchangers

- Development of mathematical models for the simulation of
 - Dehumidifiers and regenerators as adiabatic beds
 - o Dehumidifiers and regenerators as non-adiabatic tubular heat exchangers
 - Dehumidifiers and regenerators as non-adiabatic plate heat exchangers
- Validation of developed mathematical models with experimental data from the literature.
- Comparative analysis and parametric simulations with the depended variables
 - Type of liquid desiccant solution
 - State of inlet air
 - State of inlet desiccant solution
 - State of inlet water (when applicable)
- Development of a mathematical model for the simulation of a liquid desiccant system
- Carrying out experiments on dehumidification and regeneration, with the depended variables
 - o State of inlet air
 - o State of inlet desiccant solution
 - o State of inlet water

1.3 THESIS STRUCTURE

This thesis consists of eight chapters. The first chapter introduces the reader to the air conditioning technology. A short history of the available methods, the conventional air conditioning systems and the thermal chillers are presented. The objectives and the innovations relating to this work are also mentioned.

The second chapter firstly discusses the operation of the liquid desiccant systems. It then includes an extensive literature review of the mathematical models developed for the coupled heat and mass transfer processes during dehumidification and regeneration process in liquid desiccant systems. The review includes the most common device geometries, flow configurations and desiccant solutions used.

The development of calculation models for the thermophysical properties of aqueous solutions of the chlorides of lithium, bromine and calcium, particularly suited for use as desiccants in sorption based air conditioning systems is discussed in chapter three. Calculation models are described for each of the following properties: Solubility boundary,

vapor pressure, density, surface tension, dynamic viscosity, thermal conductivity, specific thermal capacity and differential enthalpy of dilution.

Chapter four presents the study of the performance of an adiabatic liquid desiccant dehumidifier and regenerator. To this end, a heat and mass transfer model predicting the performance of an adiabatic packed column is developed. A detailed sensitivity analysis is implemented to indicate which input variables mostly affect the output conditions for both the dehumidifier and the regenerator.

The fifth chapter relates to the study of the non-adiabatic dehumidifier and regenerator. The non-adiabatic issue is translated here into a tubular heat exchangers acting as the dehumidifier and the regenerator. The heat and mass transfer theoretical models that predict the performance of the dehumidifier and regenerator under various operating conditions are widely discussed.

Chapter six is dedicated to the liquid desiccant experimental unit that has been installed in the Laboratory of Applied Thermodynamics in the National Technical University of Athens. It involves the theoretical investigation of this system, by introducing the reader to the mathematical model developed for the dehumidifier. It then presents the technical characteristics of the experimental apparatus and the experimental process is explained in details. The results from the dehumidification and regeneration experiments are then presented and a comparison with the theoretical study is implemented.

Chapter seven summarizes all the implemented work and consolidates the general conclusions. Useful information regarding future research work is also proposed.

Finally, Chapter eight includes the author's publications, the citations to the author's work so far and the list of figures and tables.

The diagram below shows the workflow, from the beginning of the idea until the realization of this document.



Figure 1-1 Flowchart of the implemented work

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1 INTRODUCTION

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1 INTRODUCTION

2.1 INTRODUCTION

Liquid desiccant dehumidification systems have been used for many years in specialized applications. The performance of liquid desiccant systems lies heavily on the heat and mass transfer characteristics of the two critical components: the dehumidifier and the regenerator. The purpose of this chapter is to provide a comprehensive review of the heat and mass transfer correlations developed to mathematically model the adiabatic absorption and desorption process. There has been an attempt to describe the most important characteristics of each research, such as the method adopted, the assumptions used, the validation of the data, as well as the most important results. It was found that most work considers the desiccant solution flowing counter currently with the air and the use of structured packing instead of random. Fewer researchers considered the co flow configuration or desiccants other than salts.

Liquid desiccant air conditioning systems have attracted more and more attention in recent years due to their environmentally friendly technology and promising utilization of low-grade thermal energy provided by solar flat-plate collector or waste heat [1]. In liquid desiccant cooling cycles, a sorbent solution is employed to dehumidify the air, circulating between the two critical components; the dehumidifier and the regenerator.

As the strong desiccant solution is sprayed on top of the dehumidifier, it flows down by gravity and comes in contact with the process air. The air can be moving in parallel, counter or cross flow with the solution. The desiccant solution which, by definition, has a strong affinity for water vapor, absorbs moisture from the air. As the water vapor condenses and mixes with the desiccant, heat is released. This heat equals to the latent heat of condensation for water plus the chemical heat of mixing between the desiccant and water. In an adiabatic dehumidifier (Figure 2-1) this heat would raise the temperature of the desiccant and decrease its ability to remove water vapor from the air. An internally cooled dehumidifier (Figure 2-2), by means of cooling water derived from an evaporative cooler, would be a solution to this problem. The cooling water restricts the temperature changes for both the desiccant and air and therefore, increases the dehumidification capacity. The end of the process finds the air cool and dehumidified and the solution diluted.



Figure 2-1 Hydraulic scheme of an adiabatic liquid desiccant cooling system.



Figure 2-2 Hydraulic scheme of an internally cooled liquid desiccant cooling system with plate heat exchangers.

The role of the regenerator is to retrieve the initial concentration of the diluted desiccant solution that exits the dehumidifier. The regeneration is a very important process, since it requires high temperatures and therefore, affects both cooling capacity and energy utilization efficiency of air conditioning systems [1]. The regenerator device can have the same configuration with the dehumidifier; however, the process occurring is just the opposite. The hot and diluted desiccant solution comes into contact with the ambient air. Since the vapor pressure of the desiccant is now higher than that of the air, moisture is evaporated from the solution and transferred to the exhaust air stream. In an adiabatic regenerator (Figure 2-1), the heat is supplied to the desiccant solution before entering the regenerator. This results in the cooling of the solution with the progress of regeneration, so the performance is gradually decreased. Internally heated regeneration (Figure 2-2), by means of hot water derived from a low temperature source, is thus preferred. At the end of the process, the hot humid air is rejected to the ambient and the concentrated solution is driven to the dehumidifier. A liquid-to-liquid heat exchanger is used to precool the warm concentrated solution using the cool dilute solution from the outlet of the dehumidifier. This reduces the heat input to regenerator by 10-15% and improves the system performance [2].

The complex heat and mass transfer phenomena, occurring both in the dehumidifier and regenerator, have been the subject of a great amount of research, especially within the last 25 years [3, 4]. During dehumidification (absorption) and regeneration (desorption) process, heat and mass are transferred through and between the liquid and vapor phases. The driving force for heat transfer between the liquid desiccant solution and the air is their temperature difference, while the driving force for the mass transfer is the difference between the vapor pressure of the desiccant and the partial pressure of water vapor in the air. However, these driving forces change as the process progresses, due to changes in the local temperature and concentration and due to changes in the liquid-vapor interface equilibrium condition [3]. Equilibrium condition is achieved when the air temperature is equal to that of the liquid desiccant solution pressure of the solution [5]. The heat and mass transfer phenomena are thus coupled and so, the accuracy of the mathematical models depends on the method and correlations used to predict these heat and mass transfer processes.

This chapter attempts to provide a wide review on the mathematical models developed for the coupled heat and mass transfer processes during dehumidification and regeneration in adiabatic liquid desiccant systems. This review includes the most common device geometries, flow configurations and desiccant solutions used. However, models which have

not considered the coupled nature of heat and mass transfer processes, which have used additives to enhance the transfer rates or which have considered the effect of nonabsorbable gases are not included in this study.

The following paragraphs have been organized first according to the type of flow configuration (co flow, counter flow or cross flow) and then by date. For each study, the key assumptions, solution method and model validation have been summarized. Although the intention of this chapter is to highlight most of the significant studies, the listing may not be complete, due to space and language limitations.

2.2 LITERATURE REVIEW

2.2.1 Co-flow

Very few authors have studied the heat and mass transfer coefficients of the co-flow configuration. Chen & Sun [6] experimentally evaluated the coefficients in a vertical film dehumidifier employing CaCl₂. To this end, they firstly mathematically determined the temperature and concentration at the desiccant-air interface and then, they expressed the overall heat and mass transfer coefficients, based on the logarithmic mean temperature difference between the inlet and outlet conditions. The comparison of this method with others from the literature showed that the assumption that the bulk desiccant temperature is equal to the interfacial temperature, resulted in an overestimate of mass transfer coefficient and an underestimate of heat transfer coefficient. The authors found that as the Reynolds number of the inlet desiccant increases, the outlet concentration is increased, resulting in a higher mass transfer coefficient and lower heat transfer coefficient. Additionally, increasing the desiccant flow rate slightly increased the absorption mass transfer rate. Increasing the absorber pressure, increased the condensation driving force and the solution potential for absorbing water vapour; however it did not affected the heat and mass transfer coefficients. Correlations for the mass transfer coefficient with respect to the inlet desiccant Reynolds number and physical properties have been also derived.

In the study of Rahamah, Elsayed and Al-Najem [7], the governing equations with the boundary and interfacial conditions are numerically solved. The assumptions used were laminar flow, constant physical properties of desiccant solution and air, air – solution interface in thermodynamic equilibrium, negligible water absorption/desorption rate compared to the desiccant flow, constant film thickness, constant wall temperature, negligible drag between air and liquid and fully developed velocity profile at the entrance. By

using the control volume approach, the equations describing the conservation of momentum, energy and mass for the air and CaCl₂ were transformed into finite difference equations, which were then solved by an iterative algorithm. Correlations, in terms of average Nusselt and Sherwood numbers, were developed for the heat and mass transfer coefficients, both for dehumidification and regeneration processes. The correlation developed for the average Nusselt number was compared with the literature and the maximum deviation was 15%. Their parametric analysis showed that changing the inlet humidity ratio did not affect the air and desiccant outlet temperatures, but increased the dehumidification rate. The variation of the inlet air temperature has negligible effect on the outlet humidity ratio, the solution concentration and the solution temperature. They also found that high solution concentration, low air flow rates and high desiccant flow rates produce better dehumidification.

The study of Liu, Jiang, Xia & Chang [8] extended their knowledge in liquid desiccant systems, as they produced analytical solutions of the air temperature and humidity ratio, the desiccant temperature and equivalent humidity ratio and the moisture efficiency in the parallel flow, counter flow and cross flow dehumidifier/regenerator, using realistic assumptions.

Liu & Jiang [9] focused on the coupled heat and mass transfer interaction in the dehumidifier and regenerator, under co-flow and counter-flow configuration. They agree that the heat transfer process and the mass transfer process are influenced and combined together, so they should not be analyzed independently. After evaluating the uncoupled heat and mass transfer driving force, they concluded that the variance of the desiccant concentration within the dehumidifier/regenerator may be neglected and the variance of the desiccant temperature greatly affects the desiccant absorption ability. The comparison between the two flow patterns showed that the mass transfer performance of the co-flow configuration is lower during the dehumidification process, but higher during regeneration.

Diaz [10] developed a two-dimensional transient model to analyse the effect of temperature and concentration change over time in a parallel flow liquid desiccant dehumidifier. The effect of the wall has been considered for adiabatic and constant temperature boundary conditions. An implicit finite difference scheme was used to solve the set of equations and the model was validated against test data from the literature. The author presented interesting plots of the variation of dimensionless temperature of air, desiccant and wall over time. The variation of the water concentration in the liquid desiccant solution and variation of the air humidity ratio over time were drawn. The instantaneous average Nusselt and Sherwood numbers along the air/liquid desiccant interface were also presented.

More recently, in 2011 Peng and Zhang [11] introduced two new variables, the total temperature difference and the dimensionless heat loss coefficient, and solved the dimensionless heat and mass transfer equations in a solar regenerator, both for co-flow and counter flow. The results of their numerical model gave two possible relations between the dimensionless air temperature and the local number of transfer units; linear and parabolic. An analytical model describing the heat and mass transfer driving potentials between air and solution was then developed and compared against the numerical model. Although the results showed no significant difference between the two models, the authors explained the errors included. The analytical model was also compared against experimental data from the literature, with well agreement. Their parametric analysis of inlet variables proved that counter-flow yields better performance than co-flow regeneration at the same working conditions.

Tanda, Shirai, Matsumura & Kitahara [12] experimentally studied a dehumidifier using polyethylene glycol (PEG) as the desiccant solution, in order to produce a mass transfer correlation that includes the desiccant properties and the nozzle diameter. The performance of the dehumidifier was expressed by the volumetric mass transfer coefficient, which is increased with liquid flow rate, air flow rate and with water concentration, but is decreased with the orifice diameter of the nozzle. Two correlations for the gas phase mass transfer coefficient are suggested here; the first includes physical properties of the liquid and the nozzle diameter and the second includes the desiccant surface tension. The first correlation predicted the experimental results obtained from the literature within 18.1%, while the predictions of the second were within 10.4%. The correlations developed here are applicable to a wide range of operational parameters, but only in dehumidifiers with the specific type of nozzle used.

2.2.2 Counter flow

The researcher's interest in counter flow adiabatic dehumidifiers and regenerators dates back to 1969, when Treybal [13] developed a model for the adiabatic gas absorption and produced the individual heat and mass transfer coefficients for the liquid and gas phases.

Few years later, in 1980, Factor and Grossman [14] studied a counter flow packed column and developed an analytical model to predict the performance of the device under various operating conditions. The assumptions used were low flow conditions, adiabatic operation,

constant gas and liquid properties and interface temperature equal to solution temperature. After deriving the expressions for heat and mass transfer rates across the interface, they assumed an initial exiting liquid temperature and they were able to calculate the local gradients in successive steps of tower height. They found that the regeneration can be enhanced by increased liquid flow rates, but hampered by the solution cooling as it contacts the air. Therefore, they propose that the solution should be maintained at a higher temperature throughout the bed, in order to improve regeneration performance. An experimental apparatus has been also constructed using Monoethylene Glycol and Lithium Bromide, to accurately monitor the changes of the solution and air and to also validate their analytical model. Although MEG experiments yielded inaccurate results and correlations for mass transfer coefficient could not be developed, Factor and Grossman [14] ran into an important conclusion; MEG, having a much lower vapour pressure than the water dissolved in it, can evaporate in small quantities. On the other hand, LiBr experiments showed very good agreement with the theoretical model. They found that the optimal, in terms of mass transfer and pressure drop, liquid to air mass flow rate ratio ranges from 0.5 to 0.8. They suggest solution temperatures above 65°C for regeneration, whereas 25-30°C for dehumidification.

The evaluation of heat and mass transfer coefficients was also the objective of Lof, Lenz and Rao [15] few years later. They firstly expressed the heat and mass transfer rates in differential equations with the following assumptions; heat and mass transfer coefficients are constant over the height, the specific heat of air is constant, the partial and vapor pressure is expressed by Dalton's law, the solution temperature is constant and the interface temperature is equal to the solution temperature. Then, they constructed an experimental apparatus with LiCl as the desiccant solution and they recorded the heat and mass transfer coefficients over a range of several values of air flow rate, air temperature and humidity. Although the expression they used for the heat transfer coefficient was based on experimental measurements with pure water, the degree of agreement between the theoretical model and the experimental results was within 10%. On the contrary, the experimental and theoretical values for the mass transfer coefficient showed considerable variations. They also indicated the desirability of employing high air mass flow rates consistent with pressure drop and flooding velocity constraints and of dimensioning the packed column allowing for uncertainties in the mass transfer coefficient.

Gandhidasan, Kettleborough and Ullah in 1986 [16] demonstrated the performance difference between two types of packing materials, ceramic Raschig rings and carbon Berl

saddles, in a counter flow dehumidifier with CaCl₂ solution. Extending the earlier study of Treybal [13], they calculated the heat and mass transfer coefficients for gas and liquid phase. In fact, their gas phase correlations are valid for any aqueous solution used. They concluded that the carbon rings yield higher heat transfer coefficients than ceramic rings. They also found that the heat and mass transfer coefficients greatly depend on the solution flow rate and inlet concentration, rather than the air flow rate, solution inlet temperature and air inlet temperature. Two years later [17] they set a theoretical limit on the minimum possible humidity ratio at the air outlet, correlating it with the solution inlet temperature and concentration. They also studied the influence of air inlet temperature and humidity ratio, solution inlet temperature and concentration, on dehumidifier's performance quantified by the moisture removal effectiveness. Their results showed that low solution temperature, high solution concentration and low air flow rate is desirable for effective dehumidification.

Research by Stevens, Braun & Klein in 1989 [18] led to the development of a simple, effectiveness model describing the heat and mass transfer processes in the dehumidifier and regenerator of a liquid desiccant unit employing LiBr. The model was based on a previously developed effectiveness model of a cooling tower and the main assumptions were that the saturation enthalpy depends linearly with temperature and that the amount of water lost is neglected from the solution energy balance. The model was validated against the finite difference model of Factor and Grossman [14] with very good agreement. Experimental data were used for further validation, but without as close agreement as the finite difference model. Indeed, the effectiveness model matched the experimental data in terms of air enthalpy, solution enthalpy and solution temperature; however, a disagreement was found in terms of air outlet temperature and humidity ratio, due to the assumption that Lewis number was one. The comparison of the effectiveness model with other experimental data showed 1- 23% difference regarding energy balances.

As an extension of his previous work, Gandhidasan [19] focused on the regeneration process with CaCl₂ solution. He presented an analytical solution to predict the mass of water evaporated from the weak solution, through a simplified vapor pressure correlation and the dimensionless vapor pressure and temperature difference ratios. He concluded that high air flow rate, high solution temperature, low inlet humidity ratio and solution preheating are desirable for increasing the regeneration performance, but the solution flow rate has no significant effect.

The heat and mass transfer at the air and solution interface during dehumidification was the objective of the research by Ertas, Anderson & Kavasogullari in 1991 [20]. Except from the

common CaCl₂ and LiCl, they also tested a new liquid desiccant mixture, CELD, consisting of 50% LiCl and 50% CaCl₂. The authors considered the void space as a crucial parameter for the evaluation of the heat and mass transfer coefficients. The void space is determined from the effective interfacial area and the "liquid holdup", referring to the amount of solution that remains in the packing and cannot be drained. The heat and mass transfer coefficients were calculated theoretically for the air and liquid phase, for each desiccant solution, using packing material of various dimensions. Their parametric analysis showed that the liquid phase mass transfer coefficient increases as the solution flow rate increases, as the desiccant concentration increases and as the as the desiccant inlet temperature decreases. Among the three solutions, LiCl has the higher rate of change of liquid phase mass transfer coefficient. Both liquid phase mass and heat transfer coefficients decrease as the size of the packing increases. Regarding CELD performance, CELD solution greatly improves the mass transfer comparing to CaCl₂, while it demonstrates the same liquid phase heat transfer as the LiCl and CaCl₂.

Dehumidification and regeneration with CaCl₂ was the objective of the next study [21]. The authors developed a finite difference model to calculate the effectiveness of heat and mass transfer under various column heights, air and solution flow rates, inlet air and solution temperatures and solution concentrations. The governing equations were solved numerically using a fourth order Runge-Kutta integrating scheme and the Nachtsiem-Swigert iteration technique. They found that the effectiveness of heat and mass transfer is increased with the increase in solution mass flow rate and decrease in air mass flow rate. Useful charts, predicting the outlet conditions of air and solution for a wide range of operating conditions, were also drawn.

Chung, Ghosh, & Hines [22] tested a counter flow, adiabatic dehumidifier with random packing and LiCl as the desiccant. They saw that the column efficiency was increased with the desiccant flow rate, while it was decreased with the air flow rate. The overall mass transfer coefficient was increased with the air flow rate, while it was slightly affected with the desiccant flow rate. The air flow rate was noticed to greatly affect the height of the transfer unit, when the inlet desiccant concentration was less than 40%. As expected, decreasing the desiccant temperature significantly improved the column performance.

Another attempt to predict the column efficiency was carried out by Chung in 1994 [23]. He firstly developed correlations for different packings and desiccant solutions (40% LiCl or 95% triethylene glycol TEG). Since the driving force of the dehumidification process is the vapour

pressure difference between the inlet air and the liquid desiccant, Chung [23] modified a previous study [17] and replaced the concentration parameter with the fraction of vapour pressure depression to the vapour pressure of pure water. Thus, he gave a more direct description of the relationship between the desiccant solution and the column efficiency. Since his correlation is a function of the liquid desiccant properties, the packing and the column dimensions, it can be used to predict other systems as well. Its validity was experimentally tested for various packing materials and succeeded an average discrepancy of 7%.

Mass transfer correlations were also developed by Potnis and Lenz in 1996 [24] for random and structured packings with varying bed depths, through the construction of an experimental LiBr liquid desiccant system. Since the gas phase mass transfer resistance is negligible compared to the liquid phase mass transfer resistance, a very useful dimensionless mass transfer correlation could be developed. Following the experimental measurement of the diffusivity of water in LiBr solution, the experimental data of mass transfer were converted into dimensionless form. The results indicated that the mass transfer coefficient in random packing is significantly higher than that in the structured packing and undergoes greater changes in the regenerator than in the dehumidifier. Additionally, since the mass transfer rate of the random packing is about 50% higher than the structured packing in the dehumidifier.

Another method to obtain the dimensionless heat and mass transfer correlations for both random and structured packings was the Buckingham Pi method introduced by Chung, Ghosh and Hines in 1996 [25]. Following extensive experimental runs, the authors calculated the overall gas phase mass transfer coefficient, assuming that the water is transferred from air into the LiCl solution through a stagnant film of liquid, and the gas phase heat transfer coefficient, assuming adiabatic operation of the column. The dimensionless correlations were obtained after a dimensional analysis. Experimental data from the literature were used to test the heat and mass transfer correlations both for random and structured packings. The discrepancy between the theoretical and experimental data did not exceed 10%. The authors observed that the column efficiency is increased as the solution flow rate is increased or as the air flow rate is decreased. Additionally, the gas phase mass transfer coefficient is increased with an increase in the air and solution flow rate. Regarding the heat transfer characteristics, the coefficient in random packing was always higher than that in the structured packing. Generally, the heat transfer coefficients were found to be a stronger

function of the air flow rate than the solution flow rate. Also, increasing the desiccant concentration reduces the heat transfer coefficient.

Oberg and Goswami [26] investigated the heat and mass transfer between 95% triethylene glycol and air, using high solution flow rates with the view to assess the dehumidification rate as well as the process effectiveness. Their finite difference model was based on previous studies [13, 14] and assumed adiabatic absorption, concentration and temperature gradients occurring in the flow direction, only water is transferred between the air and the desiccant, the interfacial surface area is the same for heat transfer and mass transfer and equal to the specific surface area of the packing, the heat of mixing is negligible as compared to heat of condensation of the water and the resistance to heat transfer in the liquid phase is negligible. Using empirical correlations for the gas and liquid phase heat and mass transfer coefficients from the literature, they developed a computer program to carry out the finite difference analysis. The convergence criteria were ±0.05°C for the inlet desiccant temperature and ±0.0001kg TEG/kg solution for the inlet desiccant concentration. Although the finite difference model was found to over predict the real performance of the dehumidifier, the discrepancy between them did not exceed 15% in the case of dehumidification effectiveness. The authors report differences with previous studies; they found no dependency between inlet air temperature and dehumidifier performance, inlet air temperature and condensation rate, inlet desiccant temperature and heat and mass transfer correlation and desiccant flow rate and condensation rate.

In 1999, Lazzarin, Gasparella and Longo [27] investigated theoretically and experimentally the dehumidification of air by the liquid desiccants LiBr and CaCl₂ in a counter flow packed tower. Based on previous work [13, 14, 16], the authors developed a computer model, assuming that the system is adiabatic, the thermal resistance in the liquid phase is negligible, the heat and mass transfer occur only in a transverse direction to gas and liquid flows, the interface areas active in heat and mass transfer processes are equal. In the case of humidity reduction, the model overpredicts experimental data with a mean deviation around 20%. The assumed adiabatic conditions are confirmed, since the discrepancy between the enthalpy balance on the solution side and the enthalpy balance on the air side across the tower is less than 20%. Their results also showed that greater dehumidification is obtained at high solution concentration and low solution temperatures. For LiBr this means that its concentration should be greater than 45% and its temperature between 20 and 30°C. A solution to air flow rate ratio between 1 and 2.5 is also preferable.

Radhwan, Elsayed & Gari [28] developed a mathematical model for a solar liquid desiccant system with CaCl₂. Three indices were introduced to evaluate the system performance; desiccant replacement factor, system thermal ratio and solar utilization factor. The authors described the non-dimensional governing equations for the dehumidifier, employing a fourth order Runge-Kutta integration scheme and the Nachtsheim-Swigert iteration to reach a solution. Following a similar process for the regenerator and the evaporative cooler, they developed a computer program that predicts the transient performance of the system. It was found that the system performance is good at humid weather conditions.

The energy consumption of the regenerator has been the subject of the study of Martin and Goswami [29]. They experimentally investigated the heat and mass transfer processes between a liquid desiccant (95% triethylene glycol) and air in an adiabatic counter flow regenerator using high liquid flow rates. Their theoretical model and its associated assumptions were similar to their previous study [26]. Good agreement was found between the experimental findings and model predictions, with the discrepancy being 15%. As in the dehumidifier [26], the regenerator's finite difference model over predicted the real performance, perhaps due to the assumption concerning the heat and mass transfer area and the incomplete wetting of the packing. Design variables found to have the greatest impact on the regenerator performance are the air flow rate and the humidity ratio, the desiccant temperature and concentration and the packed bed height. Namely, the water evaporation rate is increased with the increase in air flow rate and desiccant temperature and with the decrease in desiccant concentration. The humidity effectiveness is increased with the decrease in air flow rate, but it is unaffected with the change in desiccant temperature or concentration. The desiccant flow rate and the air temperature did not have a significant effect on the regenerator performance; however, the desiccant flow rate must be high enough to ensure wetting of the packing. In their next publication [30], the authors extended their research into the system performance, by studying the effectiveness of the dehumidification and regeneration process with LiCl. Their previously developed mathematical model [26] that worked well with TEG, was modified to include an equation for the wetted surface area. This modification was necessary because LiCl has higher surface tension than TEG and so, it provides less sufficient wetting of the packing and therefore, it causes considerably reduction to the mass transfer area. An additional consideration was introduced in the model to account for the non-uniform desiccant distribution at the top of the tower. Regarding air dehumidification, the adapted finite difference model shows very good agreement with the experimental findings. The authors found that the water

condensation rate increases with the desiccant concentration, inlet air humidity ratio and air flow rate, but decreases with the desiccant temperature. The humidity effectiveness was slightly increased with the desiccant flow rate and slightly decreased with air flow rate and air temperature. Regarding regeneration, the water evaporation rate increases with the desiccant temperature, air flow rate and the desiccant flow rate, while it decreases with the inlet desiccant concentration and the inlet air humidity ratio.

The above experimental data were used by Gandhidasan in 2004 [31] to validate a simple model containing dimensionless vapor pressure and temperature difference ratios. The assumptions used were: the water condensation rate is negligible compared to the solution flow rate, the specific heat of the fluids with respect to the temperature is constant, the latent heat of condensation is constant with respect to the desiccant temperature and concentration, the desiccant inlet temperature is different from the inlet temperature of air. The comparison of this model against experimental data [30] had very good agreement, with the maximum discrepancy being 10.5%. He also showed that the water condensation rate decreases with the increase of the cooling water inlet temperature and the decrease of the desiccant-to-water heat exchanger effectiveness.

In 2005, Longo and Gasparella [32] carried out both air dehumidification and solution regeneration experimental tests. The experiments showed that both dehumidification and regeneration rate depend on the mass flow rate ratio with a logarithmic trend. The KCOOH solution showed worse dehumidification but better regeneration performance than LiCl and LiBr solutions. Their theoretical analysis was based on previous work [13], using the following assumptions: adiabatic system, negligible thermal resistance in the liquid phase, heat and mass transfer occurring only in transversal direction to the flows and equal heat and mass transfer interface areas. The outlet desiccant conditions were initially guessed and iterations were carried out until convergence between calculated and real inlet desiccant conditions was obtained. The model appears fully adequate to simulate the real processes, since the mean absolute deviation between the model and the experiments was 8.8% for dehumidification, 14.8% for regeneration tests and 9.4% for pressure drop. Their parametric analysis determined a transition temperature of the solution, which separates the dehumidification from the regeneration process. For example, if inlet solution temperature is up to 40°C for LiBr and LiCl and 37°C for KCOOH, then dehumidification takes place, while higher temperatures result in regeneration. High dehumidification rates are achievable in the temperature range 20-30°C, whereas regeneration process works well around 50°C.

Similarly, they found the transition concentration of the solution; 25% for LiCl, 42% for LiBr and 57% for KCOOH.

Ren, Jiang, Tang and Zhang [33] studied the adiabatic dehumidification and regeneration with CaCl₂ under low flow. Based on previously developed models [14, 16], they developed a controlling equation for the quasi-equilibrium processes, with the results drawn on a psychrometric chart. Numerical simulations of one dimensional heat and mass transfer model under practical conditions were also performed. They showed that effective dehumidification and cooling occur under high desiccant flow rate, whereas effective regeneration requires high desiccant flow rate and high desiccant inlet temperature. In their next publication one year later [34], they rearranged the initial equations to obtain two coupled ordinary differential equations which they solved analytically. The assumptions used were: one dimensional problem, adiabatic process, negligible axial dispersion, no desiccant evaporation, negligible heat absorption/release from water vapor in the interface, the heat of water absorption equal to the heat of water evaporation and the Lewis factor is constant. Additionally, negligible variation of the solution flow rate and concentration during the process was assumed, therefore a linear approximation for the dependence of equilibrium humidity ratio on solution temperature was adopted. This approximation compromised the accuracy of the developed analytical model especially in operating conditions with large solution temperature differences.

Elsarrag, Elmagzoub M. Ali & Jain [35] assessed the performance of a structured packed liquid desiccant system with triethylene glycol, with the view to develop practical design guidelines. They firstly developed a finite difference model and then they validated it against data obtained from their experimental apparatus. They found that the moisture removal rate increased with the increase in the desiccant flow rate, inlet air humidity ratio, increase in packing height and decrease in the equilibrium humidity ratio. The humidity effectiveness increased with the increase in the liquid flow rate, increase in packing height and decrease in the liquid flow rate, increase in packing height and decrease in the desiccant temperature and concentration. It was also found that high desiccant flow rates do not have a significant effect on the system performance, if the desiccant to air flow ratio exceeds the value of 2.

Elsarrag [36] was especially concerned with the pressure drop, droplet carryover, leaks and corrosion occurring in a dehumidifier employing triethylene glycol as the desiccant solution. By assuming that only water is transferred between the air and the desiccant, the heat of mixing is negligible and that the interfacial temperature is equal to the desiccant bulk

temperature, he developed a theoretical model of the coupled heat and mass transfer. Empirical correlations for the heat and mass transfer coefficients in the gas and liquid phases from his previous publication were used. An experimental apparatus was used in order to validate the developed finite difference model, which showed very good agreement. He found that the heat and mass transfer effectiveness increased with the decrease of the airflow rate and the increase of the packing height, while it showed a slight dependency on the liquid flow rate, the equilibrium humidity ratio and the inlet air humidity ratio. Effectiveness correlations found in the literature were also assessed by using his experimental findings. It was concluded that two, out of the three, studies gave good predictions.

Retaining the high desiccant flow rate conditions and the same assumptions, Chen, Li, Jiang & Qu [37] also presented an analytical solution of adiabatic heat and mass transfer in both parallel flow and counter flow. The validation was implemented against experimental data from the literature and the difference was found less than 10% for all parameters considered. This model can deduce an optimum air to desiccant flow rate ratio, since it was found that the ratio is increased with the decrease of the solution temperature or the increase of concentration. Their following study [8] produced analytical solutions also for the counter flow, as described earlier. Experimental data with LiCl by Chung et al [25] and Fumo & Goswami [30] was used. The analytical solutions predicted the data within ±5% for the dehumidifier and ±10% for the regenerator.

In 2007, Yin, Zhang & Chen [38] constructed an experimental facility of a LiCl liquid desiccant system to study the effect of operational parameters on the system performance. They found that the mass transfer coefficients during regeneration increased with the desiccant inlet temperature and decreased with the inlet desiccant concentration, obtaining a maximum value of 7.5 g/m⁻²s under 20% mass concentration and 77.5°C desiccant temperature. The dehumidification rate was found to increase with the inlet humidity ratio, but most importantly, they found that there is a maximum tower efficiency for every dehumidifier, at a specific inlet humidity ratio.

The study of Jain and Bansal [2] presents a parametric analysis of packed bed dehumidifiers for three desiccant materials (TEG, LiCl and CaCl₂) and three different correlations for the dehumidification effectiveness. During their parametric analysis, they revealed that the increase of desiccant to air flow rate up to 4 increases the dehumidification effectiveness of all three solutions. However, they found that the effect of solution inlet temperature, solution concentration, air inlet temperature and packing size on the dehumidification effectiveness has great variations between the three correlations considered. Finally, they indicate the necessity of a standardized testing procedure, as are available for other air conditioning systems.

The next study of Liu, Li & Jiang in 2009 [39] focused on the comparison of air-water direct contact systems with the air-desiccant systems. Studying the properties of liquid desiccant and water, the authors saw that the status of liquid desiccant can reach a larger area in the psychrometric chart compared to water. They found that the outlet air conditions are within a triangle area in the psychrometric chart, which is composed of three boundary lines; the isenthalpic line of inlet air, the iso-concentration line of inlet desiccant and the connecting line of the statuses of inlet air and inlet desiccant. Thus, they divided the chart into four zones; two for dehumidification and two for regeneration. It was found that heating of desiccant, instead of air, and cooling of desiccant, instead of air, is preferred for the regeneration and dehumidification, respectively. Counter-flow pattern is not always the best flow pattern for various inlet air conditions. Following the same zonal method, they also found [40] that in most dehumidification cases the mass transfer driving force in counterflow configuration is more uniform than that in co-flow configuration; the opposite is valid for most of the regeneration cases. The study of the effect of flow pattern on moisture removal rate showed that in most dehumidification cases counter-flow has the best mass transfer performance, whereas parallel-flow is suggested for regeneration. They also proposed the adoption of six zones, instead of four, in order to consider the impact of flow pattern, mass flow rates and heat and mass transfer coefficients.

In the following work [41] a novel energy efficient liquid desiccant system utilizing LiCl solution was simulated, through a modular simulation program. The authors adopted the finite difference model developed by Ren, Jiang & Zhang [34], assuming that the system operates in steady state, the power consumption of pumps and fans is neglected and the wettability factor of the plate is 0.8. According to the results of the simulation, the inlet desiccant temperature in the regenerator should be high and preferably around 80°C, but not higher than 85°C due to the crystallization risk. The increase of air to desiccant flow rate ratio and air inlet temperature leads to lower dehumidification performance, while the increase of air to desiccant flow rate ratio and the decrease of inlet absolute humidity leads to higher regeneration performance.

In 2009, Tretiak & Abdallah [42] developed and tested a desiccant system under multiple counter flow sorption and desorption cycles, using clay and CaCl₂ solution. They determined a regression equation for the mass of water absorbed by the desiccant, in relation to

multiple predictors and they found that the mass of water absorbed mostly depends on the reaction time between the desiccant and the air. An equation for pressure drop through the desiccant was also determined and compared to existing models. The desiccant performed well during the repeated sorption – desorption tests, though some the exterior of some of the desiccant spheres broke off and desiccant dust was accumulated at the bottom of the bed.

Babakhani & Soleymani [43] presented a new analytical solution of heat and mass transfer processes, assuming the equilibrium humidity ratio on the interface as constant. Based on earlier models for the adiabatic gas absorption, heat transfer coefficients, gas phase mass transfer coefficients for random and structured packings and vapor pressure expressions, they defined the differential equations for all characteristic parameters and developed a simulation code. After the analytical solution describing both air and desiccant distribution in an adiabatic simultaneous heat and mass transfer processes was obtained, comparisons were made with reliable experimental data from the literature. Excellent agreement was found, with the maximum difference being less than 5% in the case of air humidity ratio, desiccant temperature and concentration and less than 7% for the air temperature. They concluded that the water condensation rate increases with air flow rate, air humidity ratio and desiccant concentration, but decreases with desiccant temperature. Humidity effectiveness is slightly increased with the air and desiccant flow rate.

Peng & Pan [44] investigated the transient heat and mass transfer at low flow conditions, in order to develop an s analytical tool for liquid desiccant systems design. They developed non-equilibrium heat and mass transfer mathematical models, based on the local volumetric average approach, to describe dehumidification process. Some of the assumptions made were: one-dimensional heat and mass transfer, uniform and steady air velocity, the air-desiccant interface includes diffusion, the desiccant flow follows classic Nusselt falling film theory, there is thermal equilibrium between the desiccant and the packing and the desiccant flow are assumed to experience two consecutive periods: the initial transient and the steady period. They found good agreement between the experimental results and the model for both air relative humidity and air temperature.

The next study of Babakhani & Soleymani in 2010 [45] presented a mathematical model and an analytical solution for the regeneration. When compared with reliable experimental data from the literature the solution showed good accuracy, with the maximum difference being less than 6%. The model proved suitable for high desiccant flow rate conditions and for random and structured packings. Design variables found to have the greatest impact on the
regenerator performance were the desiccant concentration, the desiccant temperature and the air flow rate.

Artificial neural network analysis was firstly proposed by Gandhidasan & Mohandes [46], for a randomly packed dehumidifier with LiCl. The assumptions used were: constant latent heat of condensation, constant specific heat of the fluids, constant heat exchanger effectiveness and desiccant inlet temperatures different from air inlet temperature. The variables found to have the most significant effect on the dehumidification were: flow rates, air and desiccant temperature, desiccant concentration, dimensionless temperature ratio, air humidity ratio and cooling water temperature. The neural network simulated results predicted reasonably well the experimental data regarding the water condensation rate and the desiccant concentration; however, for the outlet desiccant temperature, some adjustments were needed.

Kumar, Dhar & Jain in 2011 [47] experimented on the air dehumidification by CaCl₂, through a new spray tower design. Spray towers have low initial cost and low air side pressure drop; however, low effectiveness and large carryover of the liquid desiccant are their main drawbacks. This carryover results, not only in monetary losses, but in serious health hazards, if the desiccant vapors are inhaled. Therefore, the authors tried to achieve zero carryover for their new spray tower. Three different types of packing have been designed and tested. The first wire mesh packing tested improved the performance of the conventional hollow spray tower by about 30%, due to either reduced liquid droplet velocity or increased wetted surface. The second packing, with larger wetted area than the first, showed almost equal performance with the first. It was concluded that the reduced velocity of liquid droplets was responsible for the performance enhancement. The parametric analysis showed that the three new packings substantially improved the performance of conventional hollow type spray tower at high desiccant mass flow rates and at high humidity ratios. Up to an air velocity 3.4 m/s, the proposed packings have almost the same pressure drop as that of a conventional tower; however at higher air velocities, their pressure drop is substantially increased. Among the proposed packings, the third one is preferred as it combines improved performance with cost efficiency.

2.2.3 Cross flow

Al-Farayedhi, Gandhidasan and Al-Mutairi in 2002 [48] studied the performance of a structured packing column using three different types of desiccants; CaCl₂, LiCl and CELD. They expressed correlations for the heat and mass transfer coefficients of air and desiccant phases with high accuracy. They proved that the air phase correlations are independent of the type of desiccant used and depend only on the air and desiccant flow rate and air inlet temperature. The volumetric heat and mass transfer coefficients for air were compared with the literature, with very good agreement. Their parametric analysis showed that the desiccant phase mass transfer coefficient increases with the desiccant velocity and concentration, while decreases with the desiccant inlet temperature. The heat transfer coefficient was found to decrease with the desiccant inlet temperature, while it was unaffected by the inlet concentration. The LiCl solution presented the highest mass transfer coefficient.

Two years later, Dai, Y.J. & Zhang, H.F. [49] studied the heat and mass transfer processes in a cross flow dehumidifier using CaCl₂ as the desiccant and honeycomb paper as the packing material. A numerical model was developed, based on the assumptions of negligible water vapor diffusion in the process air, laminar falling film, constant film thickness, negligible buoyancy force, air temperature equal to the desiccant temperature at the interface and air humidity in equilibrium with that of the liquid desiccant solution. Non-dimensional parameters were employed for the validation of the model with the experimental data and a good agreement was found. Analysing the heat and mass transfer processes during dehumidification, it was found that the effect of heat transfer is stronger at the inlet of the desiccant, indicated by a large Nusselt number. The authors concluded that the desiccant solution the desiccant the and the desiccant.

The cross flow procedure has been less studied than the counter flow, due to its lower heat and mass transfer effectiveness. However, Liu, Zhang, Qu and Jiang [50] believe that the cross flow devices occupy less space and are easier to install. Therefore, they studied the cross flow dehumidification with structured packing and LiBr as the desiccant solution, both theoretically and experimentally. The mass transfer performance of the dehumidifier was evaluated in terms of the moisture removal rate and the dehumidifier effectiveness. They found that the moisture removal rate increased with the desiccant flow rate, air flow rate, inlet concentration and inlet humidity ratio, but decreased with desiccant inlet temperature. The dehumidifier effectiveness increased with desiccant flow rate and temperature, but decreased with the air flow rate. Similar tendencies of the impacts of the inlet parameters between cross flow and counter flow dehumidifiers are also reported. Correlations were also developed for the dehumidifier performance, which were in good agreement with the experimental findings.

Liu, Qu & Jiang [51] preferred to express the performance as enthalpy and moisture effectiveness, instead of temperature effectiveness. Following their experimental tests using cross flow dehumidifiers with structured packing and LiBr solution, they produced empirical correlations for enthalpy and moisture effectiveness, with an accuracy of 6.3% and 6.0% respectively. They found that the enthalpy effectiveness increases with increasing air inlet temperature and desiccant flow rate, decreases with increasing airflow rate and inlet humidity ratio and changes little with desiccant inlet temperature and decreasing airflow rate. These correlations were also compared against experimental data for counter-flow dehumidifiers available in literature and achieved deviations of $\pm 10.8\%$ and $\pm 5.1\%$ for enthalpy and moisture effectiveness, respectively.

Their next publication [52] focused on the mass transfer performance of the regenerator. A dimensionless mass transfer correlation is proposed, which can predict the regenerator performance with accuracy, since the comparison of the moisture removal rate between the predicted values and the experimental data were within ±10% with an average discrepancy of 3.9%. The experimental investigation of the effects of inlet parameters on the regenerator performance showed that the moisture removal rate increased with air flow rate, desiccant flow rate and desiccant inlet temperature, while decreased with air inlet humidity ratio and desiccant concentration. The regenerator effectiveness increased with desiccant flow rate and concentration, while decreased with air flow rate and desiccant inlet temperature. Performance comparison between cross-flow and counter-flow regenerators indicated that the impacts of inlet parameters have similar tendency.

The study of Liu, Jiang, Xia & Chang [8] produced analytical solutions for the cross flow, as described earlier. Previous experimental data with LiBr [50, 52] were used. The analytical solutions predicted the data within $\pm 6.6\%$ for the dehumidifier and $\pm 5.1\%$ for the regenerator.

Another theoretical model, this time using number of transfer units as the input parameter and the enthalpy and moisture effectiveness as the performance indices was developed by

the same team [53] in 2007. The assumptions used were: the local heat and mass transfer coefficients are uniform throughout the module, the heat and mass transfer areas are equal to the specific surface area of the packing and the contribution of conduction or diffusion is neglected. Apart from the outlet conditions, the model can also predict the temperature and concentration distributions inside the dehumidifier and regenerator. The comparison between the calculated results and the dehumidification experimental data showed an average absolute difference of 7.9% and 8.5% for the enthalpy and moisture effectiveness, respectively. For the regeneration experimental data, the average discrepancy was 5.8% and 6.9% for enthalpy and moisture effectiveness.

In 2008 Yin & Zhang [54] described the dehumidification process with LiCl through the number of transfer units and Lewis number method. The necessary simplifications assumed here were that the desiccant distribution inside the packing is uniform, the resistance of desiccant solution film is ignored and that the parameters of the desiccant and air are uniform in the control volume. The coupled heat and mass transfer coefficients between air and liquid desiccant were obtained based, not only on experimental results, but also on a new method developed by the authors, called separative evaluation method. This method was validated against experiments carried out by the authors; the difference of air humidity, air temperature and solution temperature between the calculated and experimental results was less than 10%, 6%, 12%, respectively. The results showed that the mass transfer coefficient is increased with the increase of air flow rate, air humidity ratio and desiccant concentration and with the decrease of air inlet temperature and desiccant temperature. The heat transfer coefficient is reduced with the decrease of the air flow rate. The Lewis number is increased with the increase of air inlet temperature and the decrease of air humidity ratio. An important conclusion is that, since the Lewis number changes greatly under the variation of the operation parameters, the Lewis number should not be assumed to be constant or unity during a heat and mass transfer process.

Another form of analytical solutions was given by Liu, Jiang, & Qu [55], taking into consideration the analogy between the heat and mass transfer process in the dehumidifier and the heat transfer process in the heat exchanger. They used their former mathematical model and assumed, among others, constant desiccant flow rate and concentration within the dehumidifier, Lewis number as unity and linear change of desiccant enthalpy with desiccant temperature. Since the control equations for the combined heat and mass transfer performance in cross-flow dehumidifier and the heat transfer in cross-flow heat exchanger were completely the same, the solutions for the cross-flow dehumidifier were then derived.

The results given by the analytical solution were compared with numerical solutions and experimental findings, using LiBr as the desiccant. The discrepancy between analytical and experimental results for the enthalpy and moisture efficiency was less than 20%. Their optimization analysis aimed at decreasing the pressure drop, improving the heat and mass transfer performance and minimizing the initial cost. They found that for 3600 m³/h air flow rate, the optimum dehumidifier size is 0.6×0.35×0.83 m³.

More recently, Babakhani & Soleymani [43] based on the assumption of that the solution flow rate and the concentration changes very little during the dehumidification processes and that the equilibrium humidity on the interface is therefore constant, they formed a new analytical solution of heat and mass transfer processes in a dehumidifier. They assumed negligible heat loss through the column, desiccant vaporization and desiccant thermal resistance, heat and mass transfer interface area equal to the specific surface area of the packing, steady state process and constant fluid properties. They used empirical correlations available in the literature for random and structured packings. The analytical solution was compared against experimental data available in the literature, with very good agreement; the predictions of the analytical solution were within 5% for the humidity ratio, desiccant temperature and concentration and within 7% for the air temperature. Design variables such as desiccant concentration, desiccant temperature, air flow rate, and air humidity ratio have the greatest impact on the performance of the dehumidifier.

New mass transfer performance data were presented [56] of a cross flow dehumidifier with structured packing and CaCl₂. The comparison of their experimental data to the correlations of Chung [23] for counter flow and Liu, Zhang, Qu & Jiang [50] for cross flow showed weak agreement. Therefore, they developed a new empirical correlation for the dehumidification effectiveness, which fitted the experimental data to within 10%, in the effectiveness range 0.4-0.8. The effect of varying air and solution inlet conditions and flow rates on the system performance was also quantified in the paper. The moisture removal rate increases with air humidity ratio, air flow rate, desiccant flow rate and concentration, but decreases with desiccant temperature. Dehumidifier effectiveness increases with solution flow rate, but decreases with air flow rate.

Zhang, Hihara, Matsuoka & Dang [57] experimentally studied the mass transfer of a dehumidifier and regenerator with structured packing and LiCl, in the summer and winter mode. Their theoretical analysis produced expressions for the dimensionless overall mass-transfer coefficient of dehumidifiers and regenerators. The authors carried out both dehumidification and regeneration experiments, in typical operating ranges. They found that

the average overall mass-transfer coefficient in the regenerator was considerably lower than that in the dehumidifier. During dehumidification, the moisture mass transfer rate was significantly affected by the inlet air humidity ratio. Also, increasing the air velocity or increasing the desiccant flow rate, increased the overall mass-transfer coefficient both during dehumidification and regeneration. Deviations between predicted values and experimental values were within ±20%.

In 2011 Bansal, Jain & Moon [58] compared the moisture removal rate, the effectiveness and the mass transfer coefficients of an adiabatic structured packed and an internally cooled dehumidifier, both employing CaCl₂ as the desiccant solution. It was found that the optimum desiccant to air flow rate ratio for maximum effectiveness is about 1.0. The internally cooled device presented higher moisture removal rate and higher dehumidification effectiveness. In fact, under the same range of operating parameters, the maximum effectiveness in the internally cooled ranged was 0.55-0.706, whereas in the adiabatic was 0.38-0.55. The mass transfer coefficients are proportional to the dehumidification effectiveness, thus the trends were the same as the effectiveness.

Liu, Yi & Jiang [59] further expanded their knowledge with a mass transfer performance comparison of LiBr and LiCl, through dehumidification and regeneration experiments over a wide operational range. The comparison was carried out on the basis of the same temperature and same vapour pressure. Two methods were employed: the first one used the correlations developed for the moisture removal rate and the second used the correlations developed for volumetric mass transfer coefficient. During LiBr dehumidification experiments, the average absolute differences between the predicted and experimental values are 9.1% for the first method and 7.8% for the second. Using LiCl, these differences become 7.0% and 9.2% respectively. During LiBr regeneration experiments, the average absolute differences between the predicted and experimental values are 3.8% for the first method and 6.9% for the second. Using LiCl, these differences become 7.5% and 6.4% respectively. The authors concluded that during dehumidification, the mass transfer performance of LiCl is better than LiBr, under the same desiccant mass flow rate. During regeneration process, the mass transfer performance of LiBr is better than LiCl, especially under the same desiccant volumetric flow rate, rather than under the same desiccant mass flow rate. The Coefficient of Performance of the liquid desiccant systems using these two salts is similar, while LiCl costs less than LiBr.

Pineda & Diaz [60] focused on the internal heat exchanger that is installed in the diluted solution stream that exits the dehumidifier and the concentrated solution stream that enters

the dehumidifier. They firstly developed a three-dimensional numerical model of a dehumidifier with CaCl₂ operating near freezing conditions, with the main assumptions being laminar flow, adiabatic process, constant fluid properties, steady state conditions and counter-flow configuration in the internal heat exchanger. Then, they expressed the effectiveness of the internal heat exchanger and they analyzed its effect on the outlet air temperature, humidity ratio, desiccant temperature and concentration. The results showed that high internal heat exchanger effectiveness greater than 60% were required to dehumidify the air, otherwise regeneration occurred. The results also showed a significant effect of internal heat exchanger effectiveness in the liquid desiccant temperature at the outlet of the absorber.

Bassuoni [61] experimentally investigated the performance of the structured packing dehumidifier and regenerator, using $CaCl_2$ as the desiccant. It was found that the mass transfer coefficients for both dehumidification and regeneration were increased with the desiccant flow rate. The experiments showed that a 300% increase of the desiccant flow rate resulted in an increase of moisture removal rate by 74% for dehumidification and by 50% for regeneration. It also resulted in an increase of effectiveness by 48% for dehumidification and by 71% for regeneration. Increasing the air flow rate by 600%, increased the mass transfer coefficient by 280%, the moisture removal rate by 20.4%, while decreased the effectiveness by 76.2% for dehumidification. The behavior of the regeneration process is similar, with these values being 280%, 13.6% and 60.7%, respectively. Increasing the solution temperature at the inlet of the regenerator, increased the mass transfer coefficient and the moisture removal rate, but decreased the efficiency in the regenerator. Also, a higher solution temperature at the inlet of the regenerator resulted in an increase of the solution temperature at the inlet of the dehumidifier, which decreased the mass transfer coefficient, reduced the moisture removal rate and increased the efficiency in the dehumidifier. The moisture removal rate was increased for the dehumidifier and decreased for the regenerator, as the solution inlet concentration increased. The life cycle cost analysis with a 1.87kW latent load showed that the payback period of the studied system was 11 months with 31.24% annual running cost savings.

Gao, Liu, Cheng & Zhang [62] experimentally studied a cross-flow dehumidifier with LiCl and structured packing, in terms of enthalpy and moisture efficiency. Based on energy and mass conservation laws, they developed a simple model that calculated the desiccant enthalpy, flow rate, temperature and concentration at the outlet. They observed higher enthalpy and

moisture efficiency at the upper part of the dehumidifier. They found that the moisture efficiency is mainly affected by desiccant flow rate and air flow rate, while enthalpy efficiency is influenced by the desiccant flow, air flow, air temperature and humidity ratio. Increasing the packing thickness improves the dehumidifier efficiency; however the increase of the packing thickness should have a limit, since it increases the pressure loss and fan energy consumption. The increase of the packing width and height did not improve the efficiency, but it reduced the pressure loss.

Bakhtiar, Rokhman & Hwan [63] offered a new method to evaluate the performance of cross flow dehumidification system with LiCl. According to them, the commonly used indices moisture removal rate and dehumidification effectiveness cannot represent the system efficiency. They propose the Coefficient of Performance, the ratio of the sum of the cooling capacity and the net electrical power input, as the index that represents the real system efficiency. The concentration degradation that happens during dehumidification changes the desiccant's vapor pressure, thus reducing the performance over time. This unstable performance necessitates the system efficiency to be calculated by comparing the cooling capacity of the system with the total power consumption. Their dehumidification experiments aimed at calculating the COP, by comparing the energy change of the room with the total energy consumption of all electrical devices in an hour. They found the low air velocity has high dehumidification effectiveness and high COP.

2.3 CONCLUSIONS

The air-conditioning load of a building is the sum of the sensible and latent load and represents the 20-40% of the overall energy consumption in a building. Dehumidification handles the latent load, while cooling handles the sensible load. Traditional vapour compression systems need to overcool the air stream to provide, apart from cooling, dehumidification. This means that air conditioning operates at a temperature colder than the supply air dew-point temperature, so the air needs reheating before entering indoors. Vapour compression systems are therefore not so energy efficient for handling the latent load in buildings.

Open cycle evaporative cooling systems with liquid desiccant mediums, as known as liquid desiccant cooling systems, have been proven to be an energy efficient method for air dehumidification, compared to conventional air conditioning systems. In contrast to vapor compression air conditioning systems, in which the electrical energy drives the cooling cycle and the air is overcooled, desiccant cooling is heat driven and the air does not have to be cooled below its dew point. Desiccant cooling avoids the conventional problem of re-heating to compensate for the over-cooling, as it does not rely on cooling to produce dehumidification. Therefore, liquid desiccant systems have the potential to utilize cleaner energy sources such as waste heat or solar thermal energy. The entire operation takes place at atmospheric pressure, eliminating the need for capital intensive, pressure sealed components. This makes it a very environmentally friendly technology choice if properly designed, sized and managed in use.

Desiccants are hygroscopic materials that absorb or give off moisture to the surrounding air due to a difference between the water vapor pressure at their surface and that of the surrounding air. The moisture content depends on the desiccant and temperature at the same relative humidity. If the desiccant contains more moisture than the surrounding air, it releases moisture, absorbs heat and produces a cooling effect equal to that of evaporation. If it contains less moisture, it absorbs moisture from the air and releases heat equal to the latent heat given off if a corresponding quality of water vapor were condensed.

The liquid desiccant cooling systems consist of two main components; the dehumidifier and the regenerator. Firstly, the strong desiccant solution is sprayed at the top of the dehumidifier. The ambient humid air enters the dehumidifier at the bottom, transfers its moisture to the desiccant and heat is liberated. At the same time, cold water, derived from an evaporative cooler, circulates in coils inside the dehumidifier, in order to maintain the

solution temperature and hence, the solution vapour pressure at low levels. The dehumidified air exits at the top and the warm, now diluted solution leaves the bottom of the dehumidifier and it is pumped for regeneration. More efficient dehumidification of air is achieved at low desiccant inlet temperatures and high desiccant inlet concentration.

The regeneration is the process in which the diluted desiccant solution retrieves its initial concentration. This process has great impact to the energy efficiency of the liquid desiccant cooling systems, since this is where heat is required. The regenerator device has the same configuration with the dehumidifier; however, the process occurring is just the opposite. The diluted desiccant solution comes into contact with the ambient air. Since the vapor pressure of the desiccant is now higher than that of the air, moisture is evaporated from the solution and transferred to the exhaust air stream. At the same time, hot water, derived from a low temperature source, circulates in coils inside the regenerator, in order to maintain the solution temperature and hence, the solution vapour pressure at high levels. At the end of the process, the hot humid air is rejected to the ambient air and the concentrated solution is driven to the dehumidifier.

Liquid desiccant cooling systems are ideally applied where large latent loads are present, where a low dew point is required, where high humidity can damage properties and materials and where high air quality is necessary. Indicative examples are the supermarkets, museums, ice rinks, indoor pools, hospitals, laboratories, archive buildings, food industries and pharmaceutical industries.

This chapter provided a review on the available literature regarding the heat and mass transfer operations in liquid desiccant cooling systems. Plenty of research papers have been written on the absorption (dehumidification) and desorption (regeneration) principles, with the earlier tracing back to 1969. Since then, great strides have been made; insightful theoretical models on simultaneous heat and mass transfer process during dehumidification and regeneration have been established. Most of the studies reviewed concentrated more in counter flow configuration, and less in cross flow or co flow, due to its high heat and mass transfer effectiveness. Researchers have been experimenting with apparatuses, in order to produce the heat and mass transfer correlations, as well as to validate their theoretical models. Practical conceptions, such as comparisons between types of desiccant solutions, system configurations and flow patterns, have been proven very useful to following researchers. The effects however of some inlet parameters on the efficiency remain uncertain up to now; among others, the optimum air to desiccant flow ratio, the wetting ratio of the desiccant solution over the packing, the relationship between the heat transfer

area and the mass transfer area and the distribution of the absorption heat between the desiccant film and the process air. Further analytical and experimental investigations are needed to improve the overall performance, to make systems more predictable and to promote their widespread application.

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THERMOPHYSICAL PROPERTIES OF SALTS

3 THERMOPHYSICAL PROPERTIES OF SALTS

3.1 INTRODUCTION

Desiccants are hygroscopic materials that absorb or give off moisture to the surrounding air due to a difference between the water vapor pressure at their surface and that of the surrounding air [1, 2, 3, 4]. The moisture content depends on the desiccant and temperature at the same relative humidity [5, 6]. If the desiccant contains more moisture than the surrounding air, it releases moisture, absorbs heat and produces a cooling effect equal to that of evaporation. If it contains less moisture, it absorbs moisture from the air and releases heat equal to the latent heat given off if a corresponding quality of water vapor were condensed [5].

Liquid desiccants change chemically when they absorb moisture and despite their less drying capability [7, 8], they have still distinct advantages over solids [9, 10, 11, 12, 13]. They are potentially more efficient than solid desiccant ones: set point COP of 2 compared to 1 [14]. They can store energy [15, 16, 17, 18] with 1000MJ/m³ capacity [12, 19], which is about 4-6 times of the conventional ice storage tank [18] and up to 3.5 times higher compared to solid desiccants [12, 16]. This is achieved with a high air to solution mass ratio corresponding to a small desiccant flow [8, 16, 11, 17] to achieve great differences between salt inlet and outlet concentrations [16, 17]. A maximum storage capacity of 1540MJ/m³ (or 1354 MJ/m³ with 40% LiCl) is theoretically achievable [16]. Other advantages over solids include smaller pressure drop and easy transportation to the source of regeneration heat [7, 14, 20], easier integration within HVAC plants, more efficient removal of bacteria and dust [18, 21, 22] and less cost [13, 21].

However, aging of the liquid desiccant reduces the cooling performance of the system [7]. Additionally, energy storage is performed at the expense of energy and of the coefficient of performance [15]. The energy storage capacity is limited because the moisture removed from the desiccant during regeneration equals to the moisture condensed from the air during dehumidification and because of the crystallization risk [15].

The crystallization line defines the conditions at which salt hydrates or anhydrous salt crystallizes from the solution [23, 34]. In absorption chillers, the crystallization line is usually very close to the working concentrations [25] and is triggered by high ambient temperature, low ambient temperature with full load, air leak or non-absorbable gases produced during corrosion, excessive heat input to the desorber, failed dilution after shutdown, too low chilled water supply temperature with too high ambient and/or exhaust temperature [2, 11, 25]. Therefore, if the solution concentration is increased [11, 25, 26] or the solution

temperature is reduced, crystallization may occur, interrupting machine operation [11, 25]. Therefore, different desiccants have different operating concentration levels (45-60% LiBr, 30-45% LiCl, 35-45% CaCl₂, 90-98% TEG) [2, 11]. Crystallization mostly occurs in the strong solution entering the dehumidifier, at its lowest temperature [25]. Crystallization in the piping network immediately blocks the flow, so the concentrated solution temperature needs to be raised above its saturation point, to dissolve the salt crystals. No need to mention that the recovery of the dehumidifier operation after crystallization is labour and time intensive [25].

The desiccant selection is decisive in the overall performance of the system and depends on the boiling point elevation, energy storage density, regeneration temperature, thermophysical properties, availability and cost [15]. Desiccants used up to now are the salt solutions and glycol solutions. A brief description of their characteristics follows.

3.1.1 Salts

Salts, such as Lithium Chloride (LiCl), Calcium Chloride (CaCl₂), Lithium Bromide (LiBr) and Potassium Formate (KCOOH) are the desiccants most commonly used in cooling and dehumidification applications.

These salts require corrosion resistant materials (plastic, stainless steel), thereby increasing the equipment cost [1, 15, 18]. They have higher surface tension than glycols, so adequate surface wetting is more difficult [27, 28].

LiCl has been commercially available since 1937 [15, 21, 29], as it is the most stable, has large dehydration concentration (30-40%) and can reduce relative humidity levels even by 60%, though it is expensive (8-16 $\frac{1}{2}$ in 1998) [4, 15]. LiCl and LiBr present better dehumidification than the KCOOH [21] and the cheapest CaCl₂ (0.4 $\frac{1}{2}$, 1988) [29, 30, 31].

The poor performance of CaCl₂ is attributed to its high vapor pressure and its dependence on the air inlet conditions [4, 15]. The partial pressure of CaCl₂ increases as its concentration is decreased [31]. An increase in the CaCl₂ mass flow rate decreases the air humidity ratio and increases the primary air temperature [31]. KCOOH performs better than LiCl and LiBr in regeneration tests, it is less corrosive and expensive, fully compatible with the environment and allows significant humidity reductions [21].

Salt mixtures have been also developed to combine the properties of individual materials and improve their characteristics. For example, mixing LiCl with CaCl₂ produces a desiccant solution less costly than LiCl and more stable than CaCl₂ [11, 15, 27]. Saman et al. [31] and Al-Farayedhi et al. [32] found that CELD, a mixture of 50% $CaCl_2$ and 50% LiCl, has higher mass transfer coefficient and performance than $CaCl_2$ alone.

3.1.2 **Glycols**

Triethylene Glycol (TEG), Monoethylene Glycol (MEG) and Polypropylene Glycol (PPG) are the type of glycols used in liquid desiccant cooling systems.

The earliest liquid desiccant system used TEG and solar heated air for regeneration purposes [1, 6, 15]. TEG is preferred to MEG [8, 33], since MEG has yielded inaccurate results and has higher vapour pressure [8]. 95% TEG has similar effectiveness with 40% LiCl [11].

Glycols work well as desiccants and are less corrosive than salts, but their high viscosity increases the pumping power of the system [1, 15, 27]. They have very low surface vapour pressure and their evaporation contaminates the air, whereas salts have essentially zero vapour pressure and cannot evaporate [1, 8, 34]. Though TEG is not particularly toxic, it may cause respiratory irritation [27]. These evaporation losses are unacceptable in an occupied building and increase the initial cost; therefore salts dominate LDCS [8, 21]. However, adding 10% by weight of polystyrene sulfonic acid salts to TEG improves its moisture absorption capacity by 15%, lowers its vapor pressure, thus reducing evaporation [27]. Finally, TEG regeneration requires a higher solution temperature in humid climates compared to dry climates [34].

The purpose of this Thesis is to analyze thermodynamically the absorption and desorption process, so the comparison of all types of desiccant solutions would have been out of scope. To this end, the most commonly used desiccant solutions were selected; namely Lithium Chloride (LiCl), Lithium Bromide (LiBr) and Calcium Chloride (CaCl₂).

This chapter reports the development of calculation models for the thermophysical properties of aqueous solutions of the chlorides of lithium, bromide and calcium, particularly suited for use as desiccants in sorption based air conditioning systems. Calculation models are described here for the following properties: Solubility boundary, vapor pressure, density, surface tension, dynamic viscosity, thermal conductivity, specific thermal capacity and differential enthalpy of dilution.

3.2 LITHIUM CHLORIDE LICI – H₂O

3.2.1 Solubility boundary

The solubility boundary is defined by several lines. For salt concentrations lower than that at the eutectic point, the solubility line defines the conditions at which ice crystals start to form. This is the ice line. For higher concentrations, the solubility boundary defines the conditions at which salt hydrates or anhydrous salt crystalize from the solution. This is the crystallization line. For LiCl–H₂O solutions, the crystallization line, ABCDE in Figure 3-1 defines also the transition points separating the ranges of formation of the various hydrates. Equations for each range and for the crystallization line were taken from Conde [23, 24].

$$eq. 1 \qquad \theta = \sum_{i=0}^{2} A_{i} \cdot \xi^{i}$$

eq. 2
$$\theta = \frac{T}{T_{c,H2O}}$$

where ξ is the mass fraction of the salt in the solution. For the ice line, the equation is:

eq. 3
$$\theta = A_0 + A_1 \cdot \xi + A_2 \cdot \xi^{2.5}$$

The parameters A_i are included in Table 3-1, for each range of the boundary.

Boundary	A ₀	A ₁	A ₂
Ice line	0.422088	-0.90410	-2.936350
LiCl-5H₂O	-0.005340	2.015890	-3.114590
LiCl-3H ₂ O	-0.560360	4.723080	-5.811050
LiCl-2H ₂ O	-0.315220	2.882480	-2.624330
LiCl-H ₂ O	-1.312310	6.177670	-5.034790
LiCl	-1.356800	3.448540	0.0

Table 3-1 Parameters of the equations describing the solubility boundary of LiCl-H₂O solutions



Figure 3-1 Solubility boundary for LiCl-H₂O

The transition points between the boundaries are shown in the following table.

	ξ	Т (°С)
A	0.253	-75.5
В	0.287	-68.2
С	0.369	-19.9
D	0.452	19.1
E	0.558	93.6

Table 3-2 Transition points for LiCl-H₂O solutions

3.2.2 Vapor pressure

The equilibrium pressure of saturated water vapor above aqueous salt solutions is their most common studied property. Yet, there is no consistent formulation for the accurate prediction of this property, with most attempts are limited to short ranges either on concentration or temperature. The formulations reported below are taken from Conde [23, 24] and reproduce quite accurately the known experimental data. In order to calculate the vapor pressure, the relative vapor pressure π (relative to water at the same temperature) is calculated first. The relative vapor pressure π is a function of the mass fraction and temperature, as seen below.

eq. 4
$$\pi \equiv \frac{p_{sol}(\xi, T)}{p_{H2O}(T)} = \pi_{25} \cdot f(\xi, \theta)$$

Where,

$$f(\xi,\theta) = \mathbf{A} + \mathbf{B} \cdot \theta$$
$$\mathbf{A} = 2 - \left[1 + \left(\frac{\xi}{\pi_0}\right)^{\pi_1}\right]^{\pi_2}$$
$$\mathbf{B} = \left[1 + \left(\frac{\xi}{\pi_3}\right)^{\pi_4}\right]^{\pi_5} - 1$$

$$\pi_{25} = 1 - \left[1 + \left(\frac{\xi}{\pi_6}\right)^{\pi_7}\right]^{\pi_8} - \pi_9 \cdot e^{-\frac{(\xi - 0.1)^2}{0.005}}$$

-1

 $\cdot \theta$

	LiCl-H ₂ O
π ₀	0,28
π ₁	4,30
π2	0,60
π3	0,21
π_4	5,10
π ₅	0,49
π ₆	0,362
π ₇	-4,75
π ₈	-0,40
π9	0,03



The parameters of eq. 5 are given in Table 3-3. The graph depicted in

Figure 3-2 shows the relative vapor pressure π along isotherms calculated with the eq. 4 and eq. 5.



Figure 3-2 Relative vapor pressure for LiCl-H₂O

The vapor pressure of the solution $p_{sol}(\xi,T)$ is calculated from eq. 4:

eq. 6
$$p_{sol}(\xi,T) = p_{H2O}(T) \cdot \pi$$

where $p_{H2O}(T)$ is the water vapor pressure.

Two types of charts are commonly used to represent the vapor pressure of solutions, namely Dühring and Othmer charts. The Othmer chart, the diagram of eq. 6, represents the vapor pressure of the solution against the solution temperature, for various mass fractions of the solution.



Figure 3-3 Othmer diagram for LiCl-H₂O

The Dühring chart represents the dew-point temperature of the vapor phase against the bubble point temperature of the solution.

For an aqueous salt solution that is in equilibrium with water vapor, the saturation temperature depends not only on the water vapor pressure but also on the salt concentration. Since the presence of the salt into the solution impedes the evaporation of water, the saturation temperature is always higher than the temperature of the saturated water vapor. The saturation temperature is increased when the salt concentration is increased.

The aforementioned thermodynamic law applies also in the dehumidifier, where the opposite happens; the condensation of water vapor into the solution and the decrease of salt concentration. Since there is thermodynamic equilibrium in the interface of the water

vapor and the solution, the solution temperature depends only on the salt concentration for a given pressure.

The saturation point of the strong solutions is determined by an empirical law, Duhring's law. According to that, the saturation point of the solution is a linear relationship of the saturation point of the solution's solvent under the same pressure. There is no thermodynamic expression describing this linear relationship and the definition of the coefficients is strictly experimental.

In the following table, the dew points of the solution for a mass fraction range of 0-50% and temperature range of 0-120 °C are depicted. This data has been fitted into a linear equation that relates the dew points of the solution with the solution temperature.

		Solution Temperature T, °C											
	0	10	20	30	40	50	60	70	80	90	100	110	120
% LiCl					C	Dew poir	nt temp	erature	T _{dp} , ⁰C				
0	0.0	10.0	20.0	30.	40.0	50.0	60.0	70.0	80.0	90.0	100.0	110.0	120.0
5	-1.4	8.5	18.6	28.6	38.7	48.8	58.8	68.9	79.1	89.2	99.4	109.5	119.7
10	-1.9	8.0	17.9	27.9	37.8	47.7	57.6	67.6	77.5	87.5	97.4	107.4	117.3
15	-3.1	6.7	16.5	26.2	36.0	45.8	55.6	65.3	75.1	84.9	94.6	104.4	114.2
20	-5.2	4.4	14.0	23.6	33.3	42.9	52.5	62.1	71.7	81.3	91.0	100.6	110.2
25	-8.4	1.1	10.5	20.0	29.5	38.9	48.4	57.9	67.3	76.8	86.3	95.8	105.3
30	-12.5	-3.3	6.0	15.3	24.6	33.9	43.3	52.6	62.0	71.3	80.7	90.1	99.5
35	-17.6	-8.5	0.6	9.7	18.9	28.0	37.2	46.5	55.7	65.0	74.3	83.6	93.0
40	-	-14.1	-5.2	3.7	12.7	21.7	30.8	39.9	49.0	58.2	67.4	76.7	86.0
45	-	-19.2	-10.5	-1.7	7.2	16.0	24.9	33.9	42.8	51.9	60.9	70.0	79.2
50	-	-	-13.0	-4.3	4.4	13.1	21.8	30.5	39.2	47.9	56.6	65.4	74.1

Table 3-4 Solution and dew point temperature for $LiCI - H_2O$

The linear equation is the following:

$$eq. 7 \qquad T_{dp} = A + B \cdot T$$

eq. 8
$$A = -0.02162 - 0.45308 \cdot X + 0.04233 \cdot X^{2} - 0.00208 \cdot X^{3}$$
$$+ 2.24276 \cdot 10^{-5} \cdot X^{4}$$

$$+2.34376 \cdot 10^{-5} \cdot X$$

eq.9
$$B = 1.00071 + 0.00406 \cdot X - 6.80357 \cdot 10^{-4} \cdot X^{2} + 2.676 \cdot 10^{-5} \cdot X^{3} - 4.48265 \cdot 10^{-7} \cdot X^{4} + 2.63846 \cdot 10^{-9} \cdot X^{5}$$

Where X is the concentration of the salt into the solution, given by the expression $X\!=\!\xi\!\cdot\!100$





Figure 3-4 Duhring diagram for $LiCI - H_2O$

A new type of chart showing lines of equilibrium of solution vapor pressure with the partial pressure of water vapor in the air at an atmospheric pressure of 101.325 kPa is especially useful. This chart can be drawn from the relationship below:

eq. 10
$$T(\xi, f) = \frac{\left(\frac{f}{\pi_{25}} - A\right)}{B} \cdot T_{cH20} - 273$$



Figure 3-5 Chart of equilibrium between the vapor pressure of LiCl-H2O solutions and the partial pressure of water vapor in the air, at the normal sea level atmospheric pressure

This chart permits the visualization of the required dilution of the solution for a given dehydration of the air, when the efficiency of the dehumidification system is known.

3.2.3 Density

The relative (to saturated liquid water at the same temperature) density is represented by a single cubic function of the mass fraction ratio solute solvent $\xi/(1 - \xi)$.

eq. 11
$$\rho_{sol}(\xi, \mathbf{T}) = \rho_{\mathrm{H2O},l}(\mathbf{T}) \cdot \sum_{i=0}^{3} \rho_i \cdot \left(\frac{\xi}{1-\xi}\right)^i$$

where the ρ_i are given in Table 3-5.

	LiCl-H₂O
ρο	1.0
ρ ₁	0.540966
ρ ₂	-0.303792
ρ₃	0.100791

Table 3-5: Parameters of the density equation for LiCl-H₂O

 $\rho_{H2O,I}(T)$ is the density of saturated liquid water at the temperature T. It is calculated from:

$$eq. 12 \quad \rho_{\text{H2O},l}(\tau) = \rho_c \cdot \left(1 + B_0 \cdot \tau^{\frac{1}{3}} + B_1 \cdot \tau^{\frac{2}{3}} + B_2 \cdot \tau^{\frac{5}{3}} + B_3 \cdot \tau^{\frac{16}{3}} + B_4 \cdot \tau^{\frac{43}{3}} + B_5 \cdot \tau^{\frac{110}{3}} \right)$$

$$eq. \ 13 \quad \tau = 1 - \frac{T}{T_{c,H2O}}$$

The B_i are given in Table 3-6 and the ρ_c is the density of water at the critical point (322 kg/m³). For solutions of LiCl, the scatter is almost negligible. The range of application of the equation is $0 \le \xi \le 0.56$.

i	B _i
0	1.9937718430
1	1.0985211604
2	-0.5094492996
3	-1.7619124270
4	-44.9005480267
5	-723692.2618632

Table 3-6: Parameters for the liquid water density equation



Figure 3-6 Variation of LiCl-H₂O solution density with the solution temperature for various salt concentrations

3.2.4 Surface tension

The surface tension is a function of the reduced temperature and mass fraction of the form:

$$eq. \ 14 \quad \sigma_{sol}(\xi, \theta) = \sigma_{H20}(\theta) \cdot (1 + \sigma_1 \cdot \xi + \sigma_2 \cdot \xi \cdot \theta + \sigma_3 \cdot \xi \cdot \theta^3 + \sigma_4 \cdot \xi^2 + \sigma_5 \cdot \xi^3)$$

where θ is the reduced temperature of water given by eq. 2, and the parameters σ_i are given in Table 3-7.

The surface tension of water $\sigma_{H_{2O}}(\theta)$ is calculated with the equation proposed by the IAPWS (International Association for the Properties of Water and Steam):

eq. 15
$$\sigma_{H20}(\theta) = \sigma_0 [1 - b(1 - \theta)] (1 - \theta)^{\mu}$$

where σ_0 = 235.8 mN/m, b=-0.625 and μ = 1.256.

Figure 3-7 shows the relative surface tension of LiCl-H₂O solution for various solution temperatures.



Fiaure	3-7	Relative	surface	tension	of	LiCl-H ₂ O	solution
i igui c	<i>· ·</i>	nenacive	Juliace	1011011	\sim_{J}	2.0. 1.20	50101011

	σ1	σ2	σ2	σ3	σ4	
LiCl-H₂O	2,757115	-12,011299	14,751818	2,443204	-3,147739	

Table 3-7: Parameters of the equation to calculate the surface tension of LiCl-H₂O.

3.2.5 Dynamic viscosity

The equation giving the dynamic viscosity of LiCl-H₂O solution is the following:

$$eq. 16 \quad \eta_{sol}(\zeta, \theta) = \eta_{H2O}(\theta) \cdot e^{\eta_1 \cdot \zeta^{3.6} + \eta_2 \cdot \zeta + \eta_3 \cdot \frac{\zeta}{\theta} + \eta_4 \cdot \zeta^2}$$

Where:

eq. 17
$$\zeta = \frac{\xi}{(1-\xi)^{\frac{1}{0.6}}}$$

The parameters η_i are seen in Table 3-8.

	LiCl-H₂O
η1	0.090481
η2	1.390262
η₃	0.675875
η₄	-0.583517

Table 3-8: Parameters of viscosity equation for LiCl-H₂O

The dynamic viscosity of water η_{H2O} , considered here at liquid saturation conditions, is calculated from the formulation recommended by the IAPWS for industrial use, for temperatures above 0°C.



Figure 3-8 Dynamic viscosity of LiCl-H₂O
3.2.6 Thermal conductivity

Literature regarding the thermal conductivity is sparse. Riedel [35] defined a characteristic value called "equivalent thermal conductivity depression" and understood it as a value constant for each salt in aqueous solution, at least for diluted solutions:

$$_{eq.\ 18} \quad a_{R} = \frac{\lambda_{H2O}(T) - \lambda_{sol}(T)}{\zeta_{eq}}$$

where ζ_{eq} is the equivalent ionic concentration:

eq. 19
$$\zeta_{eq} = \frac{\xi \times \rho_{sol}(\mathbf{T}, \xi) \times I_s}{M}$$

 I_s is the ionic strength of the species in solution (1 for LiCl) and M is the molecular weight of salt (M_{LiCl} =42.39).

 α_R shows a linear dependence upon concentration for LiCl solutions, decreasing with concentration.

eq. 20 $a_R = a_0 + a_1 \cdot \xi$

The parameters are given in the following table.

	LiCl
αο	10.8958*10 ⁻³
α1	-11.7882*10 ⁻³

Table 3-9: Parameters of the α_R equation for LiCl

 $\lambda_{H2O}(T)$ is the thermal conductivity of pure liquid water. For temperatures $\vartheta \ge 20$ °C, it is calculated using the IAPWS formulation for industrial applications. For subcooled water the linear variation of the thermal conductivity with temperature, given by [23] is followed:

eq. 21
$$\lambda_{\text{H2O}}(\theta) = \lambda_{\text{H2O},20}(0.208495 + 1.747278 \cdot \theta)$$

 $\lambda_{\text{H2O},20}$ is the thermal conductivity at 20°C, calculated with the IAPWS formulation.



Figure 3-9 Thermal conductivity of LiCl-H₂O versus mass fraction



Figure 3-10 Thermal conductivity of LiCl-H₂O versus temperature

3.2.7 Specific thermal capacity

The equation for the specific thermal capacity has the following general form:

eq. 22
$$Cp_{sol}(T,\xi) = Cp_{H2O}(T) \cdot (1 - f_1(\xi) \cdot f_2(T))$$

The specific thermal capacity of liquid water is calculated as follows:

eq. 23
$$Cp_{H2O}(\Theta) = 1.43981.24317\Theta^{0.02} - 0.1207\Theta^{0.04} + 0.12825\Theta^{0.06} + 0.62934\Theta^{1.8} + 58.5225\Theta^{8}$$

where $\Theta = \frac{T}{228} - 1$

The function $f_1(\xi)$ describes the effects of salt concentration upon the specific thermal capacity. For mass fractions up to $\xi \le 0.31$, it is given from the following equation:

eq. 24
$$f_1(\xi) = K \cdot \xi + L \cdot \xi^2 + M \cdot \xi^3$$

For larger mass fractions, it is linear:

eq. 25 $f_1(\xi) = O + P \cdot \xi$

The function $f_2(\xi)$ is given from:

eq. 26
$$f_2(\Theta) = Q \cdot \Theta^{0,02} + R \cdot \Theta^{0,04} + S \cdot \Theta^{0,06}$$

	к	L	м	0	Р	q	R	S
LiCl-H₂O	1.43980	-1.24317	-0.12070	0.12825	0.62934	58.5225	-105.6343	47.7948

Table 3-10: Parameters of the equations for the specific thermal capacity for LiCl-H₂O



Figure 3-11 Specific thermal capacity of LiCl-H₂O versus mass fraction for various temperatures



Figure 3-12 Specific thermal capacity of LiCl-H₂O versus temperature for various mass fractions

3.2.8 Differential enthalpy of dilution

3.2.8.1 Specific enthalpy of the solution

The values of specific enthalpy of the LiCl-H₂O solution are given in the following table [4]. This data have produced a polynomial equation (eq. 27) using the least square method for the temperature range 0-120°C. The parameters of the produced equation are given by eq. 28, eq. 29 and eq. 30. The isothermic lines in the enthalpy – concentration diagram are drawn in Figure 3-13.

		т (°С)											
% LiCl	0	10	20	30	40	50	60	70	80	90	100	110	120
						Ent	halpy (K	(J/Kg)					
0	0.0	41.9	83.8	125.7	167.6	209.5	251.4	293.3	335.2	377.1	419.0	460.9	502.8
5	-27.0	12.5	52.4	92.2	132.1	171.9	211.7	251.4	291.2	330.9	370.7	410.5	450.5
10	-13.4	22.7	58.8	95.1	131.4	167.7	204.6	240.6	277.1	313.6	350.2	387.0	423.9
15	-12.9	20.9	54.8	88.8	122.9	157.0	191.2	225.5	259.7	294.1	328.5	363.1	397.8
20	-14.4	17.8	50.1	82.5	115.0	147.5	180.1	212.6	245.3	277.9	310.7	343.5	376.4
25	-11.6	19.4	50.5	81.5	112.6	143.8	174.9	206.1	237.3	268.5	299.7	331.0	362.4
30	-0.0	29.8	59.6	89.4	119.2	149.0	178.8	208.6	238.4	268.23	298.0	327.8	357.6
35	-	-	80.1	108.5	136.8	165.2	193.5	221.9	250.2	278.6	306.9	335.1	363.3
40	-	-	-	-	166.3	193.0	219.8	246.5	273.3	300.0	326.7	353.4	379.9
45	-	-	-	-	-	230.9	256.0	281.0	306.1	331.2	356.2	381.2	406.0
50	-	-	-	-	-	-	-	319.8	343.3	366.8	390.3	413.7	436.9

Table 3-11: Specific enthalpy of LiCl-H₂O solutions

The graph showing the relationship between the enthalpy and the concentration of the solution uses the following expressions:

eq. 27
$$h = A + B \cdot T + C \cdot T^2$$

eq. 28
$$A = -66.2324 + 11.2711 \cdot X - 0.79853 \cdot X^{2} + (2.1534E - 02) \cdot X^{3} - (1.66352E - 04) \cdot X^{4}$$

eq. 29
$$B = 4.5751 - 0.146914 \cdot X + (6.307226 E - 03) \cdot X^{2} - (1.38054 E - 04) \cdot X^{3} + (1.06690 E - 06) \cdot X^{4}$$

eq. 30

$$C = (-8.09689 E - 04) + (2.18145 E - 04) \cdot X - (1.36194 E - 05) \cdot X^{2} + (3.20998 E - 07) \cdot X^{4} - (2.64266 E - 09) \cdot X^{4}$$



Figure 3-13 Specific enthalpy and concentration diagram of LiCl-H₂O solution

3.2.8.2 Partial enthalpy of water in the salt solution

The partial enthalpy of water in the salt solution is determined by the pressure and the density of steam. This value is calculated by the following Clapeyron equation:

eq. 31
$$h_{ws} = \frac{\partial H}{\partial m_w} = h_v - T\left(v_v - \frac{\partial V_{sol}(p_{sol}, T, m_w, m_{LiCl})}{m_w}\right) \cdot \frac{dp_{sol}}{dT}$$

The enthalpy and the specific volume of steam (h_v, v_v) can be found in the steam characteristics table. *H* is an enthalpy expression measured in kJ.

The partial derivative of the volume of the solution to the mass of water contained is calculated by the following expression:

$$eq. 32 \quad \frac{\partial V_{sol}(p_{sol}, T, m_w, m_{LiCl})}{\partial m_w} = \frac{1}{\rho_{sol}(\xi, T)} \left[1 + \frac{\xi}{\rho_{sol}(\xi, T)} \cdot \frac{\partial \rho_{sol}(\xi, T)}{\partial \xi} \right]$$

Where m_w, m_{LiCl} is the mass of water and salt in the solution and h_{ws} is the partial enthalpy of water in the solution in kJ/kg.



Figure 3-14 Partial enthalpy of water of in the LiCl-H₂O solution

3.2.8.3 Differential enthalpy of dilution

In the dehydration of moist air with aqueous salt solutions, the water vapour is absorbed by the solution in the dehumidifier, to be later desorbed in the regenerator. For the salt solutions considered here, the absorption (dilution) is an exothermic process, while the desorption (regeneration) requires the supply of thermal energy to the solution. This thermal energy required (or liberated) is larger than that corresponding to the vaporization (or condensation) of pure water. This difference constitutes the energy of dilution, and when referred to the unit mass of water is named differential enthalpy of dilution [23]. The differential enthalpy of dilution expresses the difference between the partial enthalpy of the water of the salt solution and the enthalpy of the saturated water at the same temperature:

eq. 33
$$\Delta h_d = h_w(T) - h_{ws}(T, X)$$

Where $X = 100 \xi$

The enthalpy of saturated water (kJ/kg) is calculated by the following expression [4]:

eq. 34 $h_w(T) = b_0 + b_1 T + b_2 T^2 + b_3 T^3$

The coefficients are given in the following table:

i	b _i
0	0.22156863
1	4.19690058
2	-4.8993808*10 ⁻⁴
3	4.00756794*10 ⁻⁶

Table 3-12: Parameters of the equation for the enthalpy of saturated water

We used data from Conde [23] to establish interpolating equations for the differential enthalpy of dilution of aqueous solutions of lithium chloride. The equations are:

$$eq. 35 \quad \Delta h_d = \Delta h_{d,0} \left[1 + \left(\frac{\zeta}{H_1}\right)^{H_2} \right]^{H_3}$$

where ζ is defined from the salt mass fraction as:

$$eq. 36 \quad \zeta = \frac{\xi}{H_4 - \xi}$$

 $\Delta h_{\scriptscriptstyle d,o}$ is related to the temperature as:

eq. 37
$$\Delta h_{d,0} = H_5 + H_6 \cdot \theta$$

The parameters H_i for the above equations are given in the following table.

	LiCl-H₂O
H1	0.845
H ₂	-1.965
H₃	-2.265
H₄	0.6
Hs	169.105
H ₆	457.850

Table 3-13: Parameters of the equation for the differential enthalpy of dilution of LiCl-H₂O



Figure 3-15 Differential enthalpy of dilution for $LiCl-H_2O$

3.2.8.4 Differential enthalpy of absorption

The differential enthalpy of absorption (heat of absorption) expresses the energy that flows into the solution during the condensation of 1 kg steam in equilibrium state and it is given by the expression [36]:

eq. 38
$$\Delta h_{abs} = h_v(p,T) - h_{ws}(T,X)$$

 h_{abs} is always greater than the heat of condensation of saturated water, under the same pressure. h_{abs} is proportional to concentration and inversely proportional to temperature, as seen in the following figure.



Figure 3-16 Differential enthalpy of absorption in relation to the solution temperature for various concentrations

3.3 LITHIUM BROMIDE LiBr – H₂O

3.3.1 **Solubility boundary**

The solubility boundary for LiBr – H₂O is given by the following relationship:

eq. 39 $T = A_0 + A_1 \cdot \xi + A_2 \cdot \xi^2$

For the ice line the above equation becomes:

eq. 40 $T = A_0 + A_1 \cdot \xi + A_2 \cdot \xi^2 + A_3 \cdot \xi^3$

The coefficients A_i are given in the following table:

Boundary	A ₀	A ₁	A ₂	A ₃
Ice line	-0.047	$-3.951*10^{-3}$	-0.026	-4.214*10 ⁻⁴
А	-7.664*10 ³	218.799	-1.541	0
В	-441.544	9.966	-0.039	0
С	-2.599*10 ⁻³	90.733	-0.791	0
D	-416.307	14.511	-0.143	0

Table 3-14 Parameters of the equations describing the solubility boundary of LiBr-H₂O solutions

The transition points between the boundaries are shown in the following table.

	ξ	T (°C)
d	0.395	-66
С	0.495	-48
b	0.575	2.5
а	0.64.6	40

Table 3-15 Transition points for LiBr-H₂O solutions

For CaCl₂–H₂O solutions, the crystallization line is more complex, particularly due to the formation of various tetrahydrates [23].



Figure 3-17 Solubility boundary for LiBr-H₂O

3.3.2 Vapor pressure

The dew point temperature for mass concentration range 5%-76% and temperature range 0-190 °C is given by the following expression [36]:

eq. 41
$$T_{dp}(X,T) = \frac{T - A(X)}{B(X)}$$

The coefficients A(X), B(X) are:

$$A(X) = \sum_{i=0}^{10} a_i \cdot X^i$$

eq. 42
$$B(X) = \sum_{i=0}^{10} b_i \cdot X^i$$

The parameters a_i, b_i are given in the following table:

i	α	b _i
0	0	1
1	1.663456.10	-6.8242821·10 ⁻²
2	-5.5338169·10 ²	5.8736190
3	$1.1228336 \cdot 10^4$	-1.0278186·10 ²
4	-1.1028390·10 ⁵	9.3032374·10 ²
5	$6.2109464 \cdot 10^{5}$	-4.8223940·10 ³
6	-2.1112567·10 ⁶	$1.5189038 \cdot 10^4$
7	4.3851901·10 ⁶	-2.9412863·10 ⁴
8	$-5.4098115 \cdot 10^{6}$	3.4100528·10 ⁴
9	3.6266742·10 ⁶	-2.1671480·10 ⁴
10	-1.0153059·10 ⁶	5.7995604·10 ³

Table 3-16 Parameters for the dew point temperature equation for LiBr-H $_2O$



Figure 3-18 Duhring diagram for LiBr-H₂O

The pressure of the *LiBr*- H_2O solution is given by the following equation:

eq. 43
$$p_{sol}(T, X) = 10^{C + \frac{D}{T_{dp}(T, X) + 273.15} + \frac{E}{(T_{dp}(T, X) + 273.15)^2}} \cdot 10^3$$

with the coefficients being:

$$e_{q.44}$$
 C=7.05, D= -1603.54 , E= -104095.5

The solution pressure depends on the dew point temperature, which is calculated by the eq. 41.

The Othmer diagram for the *LiBr*-*H*₂*O* solution is depicted below.



Figure 3-19 Othmer diagram for LiBr-H₂O

3.3.3 Density

Density depends on the temperature and mass concentration of the salt in the solution, under the expression:

eq. 45
$$\rho_{sol}(X,T) = \sum_{j=1}^{5} X^{j-1} \cdot \left[G_{0,j-1} + T \cdot \left(G_{1,j-1} + T \cdot G_{2,j-1} \right) \right]$$

The coefficients $G_{i,i}$ are given in the following table:

G _{i,j}	1	2	3	4	5
0	9.99100*10 ²	7.74931	5.36509*10 ⁻³	1.34988*10 ⁻³	-3.08671*10 ⁻⁶
1	-2.39865*10 ⁻²	-1.28346*10 ⁻²	2.07232*10 ⁻⁴	-9.08213*10 ⁻⁶	9.94788*10 ⁻⁸
2	-3.90453*10 ⁻³	-5.55855*10 ⁻⁵	1.09879*10 ⁻⁵	-2.39834*10 ⁻⁷	1.53514*10 ⁻⁹

Table 3-17 Parameters for the density equation for LiBr-H₂O



Figure 3-20 Variation of LiBr-H₂O solution density with the solution temperature for various salt concentrations

3.3.4 Dynamic viscosity

The dynamic viscosity of the solution, in mPa*s, is given by the following expression [37]:

eq. 46
$$\eta_{sol}(X,T) = \exp\left(K_1(X) + \frac{K_2(X)}{T} + K_3(X) \cdot \ln(T)\right)$$

Where

$$K_1(X) = -494.122 + 16.3967 \cdot X - 0.14511 \cdot X^2$$

eq. 47
$$K_2(X) = 28606.4 - 934.568 \cdot X + 8.52755 \cdot X^2$$

$$K_3(X) = 70.3848 - 2.35014 \cdot X + 0.0207809 \cdot X^2$$

T is solution temperature in Kelvin.



Figure 3-21 Dynamic viscosity of LiBr-H₂O solution

3.3.5 Thermal conductivity

DiGuilio et al [38] experimentally assessed the thermal conductivity of the LiBr aqueous solution for salt concentration range of 30-65% and temperature range of 20-190°C. The following expression is used for the calculation of the thermal conductivity:

eq. 48
$$\lambda_{sol}(T, X) = L(T) + M(T) \cdot X + N(T) \cdot X^2$$

where

$$L(T) = l_1 + l_2 \cdot T + l_3 \cdot T^2$$

eq. 49
$$M(T) = m_1 + m_2 \cdot T + m_3 \cdot T^2$$
$$N(T) = n_1 + n_2 \cdot T + n_3 \cdot T^2$$

The parameters are given in the following table:

	I	m	n
1	-1407.53	38.9855	-0.265025
2	11.0513	-0.240475	1.51915.10-3
3	-1.46741.10-2	3.48073.10-4	-2.32262.10-6

Table 3-18 Parameters for the thermal conductivity equation for LiBr-H₂O



Figure 3-22 Thermal conductivity of LiBr-H₂O solution versus temperature



Figure 3-23 Thermal conductivity of LiBr-H₂O solution versus mass fraction

3.3.6 Specific thermal capacity

The specific thermal capacity of the LiBr aqueous solution is given by the following expression:

$$eq. 50 \quad Cp_{sol}(T, X) = \sum_{n=0}^{3} p_n \cdot X^n + 2 \cdot T \cdot \sum_{n=0}^{2} q_n \cdot X^n + 3 \cdot T^2 \cdot r_0$$

$$\frac{i \quad p_i \quad q_i \quad r_i}{0 \quad -3.293 \cdot 10^{-1} \quad 7.4285 \cdot 10^{-3} \quad -2.269 \cdot 10^{-6}}{1 \quad 4.076 \cdot 10^{-2} \quad -1.5144 \cdot 10^{-4} \quad 0}{2 \quad -1.36 \cdot 10^{-5} \quad 1.3555 \cdot 10^{-6} \quad 0}{4 \quad -7.1366 \cdot 10^{-6} \quad 0}$$

Table 3-19 Parameters for the specific thermal capacity equation for LiBr-H₂O



Figure 3-24 Thermal conductivity of LiBr-H₂O solution versus temperature

3.3.7 Differential enthalpy of dilution

3.3.7.1 Specific enthalpy of the solution

The expression for the specific enthalpy of the solution is given by McNeely's data [39]. The specific enthalpy is equal to the enthalpy of the salt, plus the enthalpy of the water plus an additional value and follows the relationship:

$$_{eq. 51} \quad h(T,\xi) = \xi \cdot h_1(T) + (1-\xi) \cdot h_w(T) + h_2(T,\xi)$$

$$h_1(T) = \sum_{j=0}^4 a_i \cdot T^j$$

eq. 52

²
$$h_2(T,\xi) = \xi(1-\xi) \sum_{i=0}^4 \sum_{j=0}^3 b_{i,j} \cdot (2\cdot\xi-1)^i \cdot T^j$$

$h_w(T)$ is the specific enthalpy of saturated water, given by eq. 34.

i	$lpha_i$	<i>b</i> _{<i>i</i>,0}	<i>b</i> _{<i>i</i>,1}	b _{i,2}	<i>b</i> _{<i>i</i>,3}
0	$5.08668 \cdot 10^2$	-1.02161·10 ³	3.68773·10	-1.86051·10 ⁻¹	-7.51277·10 ⁻⁶
1	-1.86241.10	-5.33308·10 ²	4.02847·10	-1.91198·10 ⁻¹	0
2	9.85946·10 ⁻²	4.83628·10 ²	3.99142·10	-1.99213·10 ⁻¹	0
3	-2.50979·10 ⁻⁵	$1.15513 \cdot 10^{3}$	3.33572·10	-1.78258·10 ⁻¹	0
4	$4.15801 \cdot 10^{-8}$	6.40622·10 ²	1.31032.10	-7.75101·10 ⁻²	0

Table 3-20 Parameters for the specific enthalpy equation for $LiBr-H_2O$



Figure 3-25 Specific enthalpy of LiBr-H₂O solution versus solution concentration

3.3.7.2 Partial enthalpy of water in the salt solution

The partial enthalpy of water in the LiBr- H_2O solution is calculated similarly to the LiCl- H_2O solution, as such the following diagrams can be drawn:



Figure 3-26 Partial enthalpy of water to the LiBr- H_2O solution versus temperature



Figure 3-27 Partial enthalpy of water to the LiBr-H₂O solution versus solution concentration

3.3.7.3 Differential enthalpy of dilution

The expression of the differential enthalpy of dilution is given in eq. 33 and the corresponding diagram is shown below.



Figure 3-28 Differential enthalpy of dilution of the LiBr-H₂O solution

3.3.7.4 Differential enthalpy of absorption

The differential enthalpy of absorption is given by eq. 38 and the corresponding diagram is shown below.



Figure 3-29 Differential enthalpy of absorption of the LiBr-H₂O solution

3.4 CALCIUM CHLORIDE CaCl₂ - H₂O

3.4.1 Solubility boundary

For CaCl₂–H₂O solutions, the crystallization line is more complex, particularly due to the formation of various tetrahydrates.

$$_{eq. 53} \quad \theta = \sum_{i=0}^{2} A_{i} \cdot \xi^{i}$$

$$_{eq. 54} \quad \theta = \frac{\mathrm{T}}{\mathrm{T}_{c,h2o}}$$

$$_{eq.55} \quad \theta = \sum_{i=0}^{2} A_{i} \cdot \xi^{i} + A_{3} \cdot \xi^{7,5}$$

The parameters of the above equations are shown in the following table.

Boundary	A ₀	A ₁	A ₂	A ₃
Ice line	0.422088	-0.066933	-0.282395	-355.514247
CaCl ₂ -6H ₂ O	-0.378950	3.456900	-3.531310	0.0
$CaCl_2-4H_2O \alpha$	-0.519970	3.400970	-2.851290	0.0
$CaCl_2-4H_2O\beta$	-1.149044	5.509111	-4.642544	0.0
CaCl ₂ -2H ₂ O	-2.385836	8.084829	-5.303476	0.0
CaCl ₂ -H ₂ O	-2.807560	4.678250	0.0	0.0

Table 3-21 Parameters for the solubility boundary equation for CaCl₂–H₂O

The transition points are shown below.

	ξ	T(°C)	
А	0.2985	-54.23	
В	0.4984	28.93	
B'	0.5238	26.41	
с	0.5663	44.98	
C'	0.5809	37.76	
D	0.7486	176.33	

Table 3-22 Transition points for CaCl₂–H₂O



Figure 3-30 Solubility boundary for CaCl₂–H₂O solution

3.4.2 Vapor pressure

Similarly to the LiCl-H₂O solutions, the equations for the vapor pressure are shown below.

eq. 56
$$\pi \equiv \frac{p_{sol}(\xi, T)}{p_{H2O}(T)} = \pi_{25} \cdot f(\xi, \theta)$$

where

$$f(\xi,\theta) = A + B \cdot \theta$$

$$A = 2 - \left[1 + \left(\frac{\xi}{\pi_0}\right)^{\pi_1}\right]^{\pi_2}$$

$$eq. 57 \quad B = \left[1 + \left(\frac{\xi}{\pi_3}\right)^{\pi_4}\right]^{\pi_5} - 1$$

$$\pi_{25} = 1 - \left[1 + \left(\frac{\xi}{\pi_6}\right)^{\pi_7}\right]^{\pi_8} - \pi_9 \cdot e^{-\frac{(\xi - 0.1)^2}{0.005}}$$

The coefficients are shown in the following table [40].

	CaCl ₂ -H ₂ O
π0	0.31
π1	3.698
π2	0.60
π3	0.231
π4	4.584
π5	0.49
π ₆	0.478
π ₇	-5.20
π ₈	-0.40
π9	0.018

Table 3-23 Coefficients of the vapor pressure equation for $CaCl_2-H_2O$

The solution pressure can be calculated from eq. 6.



Figure 3-31 Relative vapour pressure for CaCl₂–H₂O



Figure 3-32 Othmer diagram for CaCl₂–H₂O

3.4.3 Density

	CaCl ₂ -H ₂ O
ρο	1
ρ1	0.836014

Density of CaCl₂-H₂O solutions is given by eq. 11, with the coefficients ρ_i being now

Table 3-24 Coefficients of the density equation for $CaCl_2-H_2O$

-0.436300

0.105642

eq. 11 is used for concentration range $0 \le \le 0.60$.

ρ2

ρ3

Relative density, which is defined as the ratio of the solution density to the density of the saturated water, is seen in the following figure.

eq. 58 relative_density=
$$\frac{\rho_{sol}(\xi,T)}{\rho_{H_{2}O}(T)}$$



Figure 3-33 Variation of CaCl₂–H₂O solution density with the solution temperature for various salt concentrations

3.4.4 Surface tension

Surface tension is well reproduced by a function of the reduced temperature and mass fraction, already shown in eq. 14. The values of the coefficients σ_i are given in the following table.

	σ1	σ2	σ2	σ3	σ ₄
CaCl ₂ -H ₂ O	2,33067	-10,78779	13,56611	1,95017	-1,77990



Table 3-25 Coefficients of the surface tension equation for $CaCl_2-H_2O$

Figure 3-34 Surface tension of CaCl₂–H₂O

3.4.5 **Dynamic viscosity**

Dynamic viscosity of salt solutions, therefore of $CaCl_2-H_2O$ solution too, is given in eq. 16 and eq. 17. However, the coefficients of these equations for $CaCl_2-H_2O$ are seen in the following table.

	CaCl ₂ -H ₂ O
η_1	-0.169310
η₂	0.817350
η₃	0.574230
η4	0.398750

Table 3-26 Coefficients of the dynamic viscosity equation for $CaCl_2-H_2O$



Figure 3-35 Dynamic viscosity of CaCl₂–H₂O
3.4.6 Thermal conductivity

The equation of thermal conductivity for $CaCl_2-H_2O$ is similar to that of LiCl-H₂O (eq. 18, eq. 19). I_s is 2 for CaCl₂ and M is the molecular weight of CaCl₂ (M_{CaCl2}=110.9). The coefficients for the thermal conductivity equation for CaCl₂-H₂O are given below.

	CaCl₂					
α ₀	5.9473*10 ⁻³					
α1	-1.3988*10 ⁻³					

Table 3-27 Coefficients of the thermal conductivity equation for CaCl₂-H₂O



Figure 3-36 Thermal conductivity of CaCl₂–H₂O versus solution mass fraction



Figure 3-37 Thermal conductivity of CaCl₂–H₂O versus solution temperature

3.4.7 Specific thermal capacity

The proposed equations for the calculation of the specific thermal capacity are equal to those of LiCl–H₂O (eq. 22 - eq. 26); the only difference lies in the parameters:

	к	L	м	0	Ρ	q	R	S
CaCl ₂ -H ₂ O	1.63799	-1.69002	1.05124	0	0	58.5225	-105.6343	47.7948

Table 3-28 Parameters of the equations for the specific thermal capacity for $CaCl_2-H_2O$



Figure 3-38 Specific thermal capacity of CaCl₂–H₂O versus mass fraction for various temperatures



Figure 3-39 Specific thermal capacity of CaCl₂-H₂O versus temperature for various mass fractions

3.4.8 Differential enthalpy of dilution

3.4.8.1 Partial enthalpy of water in the salt solution

The equations calculating the partial enthalpy of water in the salt solution are given by eq. 31 and eq. 32, where instead of m_{LiCl} , $m_{CaC_{2}^{l}}$ is now used that expresses the mass of calcium chloride in the solution. The following graphs show the relationship between the partial enthalpy of water versus solution temperature and mass fraction.



Figure 3-40 Partial enthalpy of water into the $CaCl_2-H_2O$ solution versus temperature for various salt concentrations



Figure 3-41 Partial enthalpy of water into the CaCl₂–H₂O solution versus salt concentration for various temperatures

3.4.8.2 Differential enthalpy of dilution

The differential enthalpy of dilution is given by the equations eq. 35, eq. 36 and eq. 37, where now the parameters H_i for the CaCl₂-H₂O solution are shown below.

	CaCl ₂ -H ₂ O
H ₁	0.855
H ₂	-1.965
H ₃	-2.265
H ₄	0.8
H₅	-955.690

Table 3-29 Parameters of the differential enthalpy of dilution equations for the $CaCl_2-H_2O$ solution



Figure 3-42 Differential enthalpy of dilution for the CaCl₂–H₂O solution versus mass fraction for various temperatures

3.4.8.3 Differential enthalpy of absorption

The differential enthalpy of absorption is given by eq. 38 and the corresponding diagrams are shown below.



Figure 3-43 Differential enthalpy of absorption for the CaCl₂–H₂O solution versus solution concentration for various temperatures



Figure 3-44 Differential enthalpy of absorption for the CaCl₂–H₂O solution versus temperature for various solution concentrations

3.5 CONCLUSIONS

Desiccants are hygroscopic materials that absorb or give off moisture to the surrounding air due to a difference between the water vapor pressure at their surface and that of the surrounding air. The moisture content depends on the desiccant and temperature at the same relative humidity. If the desiccant contains more moisture than the surrounding air, it releases moisture, absorbs heat and produces a cooling effect equal to that of evaporation. If it contains less moisture, it absorbs moisture from the air and releases heat equal to the latent heat given off if a corresponding quality of water vapor were condensed.

The selection of the desiccant solution is decisive in the overall performance of the system and depends on the boiling point elevation, energy storage density, regeneration temperature, thermo-physical properties, availability and cost. Liquid desiccants used up to now are the salt and glycol solutions. Prevailing to the desiccant cooling systems market are the salt solutions, such as Lithium Chloride (LiCl), Lithium Bromide (LiBr) and Calcium Chloride (CaCl₂). The earliest studied liquid desiccant system may used Triethylene Glycol (TEG), but now glycols are less frequently used in liquid desiccant applications.

Since the comparison of the performance of the various desiccant solutions is not the objective of this Thesis, only the most commonly used desiccant solutions were selected to be studied thermodynamically.

As such, the aqueous solutions of Lithium Chloride (LiCl), Lithium Bromide (LiBr) and Calcium Chloride (CaCl₂) are observed in this Chapter. A collection of equations for their most important properties, necessary in the design of absorption air conditioning equipment are presented. Calculation models are described here for the following properties: Solubility boundary, vapor pressure, density, surface tension, dynamic viscosity, thermal conductivity, specific thermal capacity and differential enthalpy of dilution. The data considered here originate from detailed literature studies. These equations are considered to be representative and produce acceptable results in engineering calculations for liquid desiccant air conditioning systems.

This chapter is therefore the base for the upcoming mathematical model development and simulation process.

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4.1 INTRODUCTION

Liquid desiccant systems have been proposed as energy saving alternatives to the conventional vapor compression systems for handling the latent load. This chapter presents the study of the performance of a liquid desiccant dehumidifier and regenerator.

A heat and mass transfer model of an adiabatic packed column has been developed to predict the performance of the liquid desiccant dehumidifier under various operating conditions. Good agreement was found between experimental tests from the literature and the developed theoretical model, with the maximum deviation being $\pm 2.9\%$ in air outlet temperature, $\pm 15.9\%$ in air outlet humidity ratio and $\pm 2.8\%$ in solution outlet temperature. Following the model validation, the rate and the efficiency of the dehumidification process were assessed under the effects of variables, such as air temperature and humidity, desiccant temperature and humidity and air and desiccant flow rates. The three liquid desiccant solutions, namely LiCl, LiBr and CaCl₂ were evaluated against each other. The results show that high absorber efficiency and system efficiency could be achieved under humid conditions, low air mass flow rates and LiCl as the desiccant solution.

The performance of the adiabatic regenerator was enabled through the development of a heat and mass transfer numerical model. Good agreement was found between the theoretical model and experimental tests from previous studies, with the deviation range being $\pm 6.2\%$ in air outlet temperature, $\pm 8.1\%$ in air outlet humidity ratio and $\pm 1.4\%$ in solution outlet temperature. Important design variables were defined. The effects of air and desiccant flow rates, air humidity ratio, desiccant temperature and concentration have been reported on the regeneration rate and regeneration effectiveness. A detailed sensitivity analysis has been implemented to indicate which input variables mostly affect the output conditions. The three most commonly used liquid desiccant solutions, namely LiCl, LiBr and CaCl₂ were evaluated against each other. The results show that high regeneration efficiency could be achieved under high desiccant mass flow rates, high air mass flow rate, high desiccant inlet temperature, low desiccant inlet concentration and CaCl₂ as the desiccant solution.

4.2 DEHUMIDIFIER

4.2.1 Overview

The air-conditioning load is the sum of the sensible and latent load and represents the 20-40% of the overall energy consumption in a building [1]. Dehumidification handles the latent load, while sensible cooling handles the sensible load. Traditional vapor compression equipment overcools the air-stream to provide cooling and dehumidification [2]. Airconditioning operates at a temperature colder than the supply air dew-point temperature, so the air needs reheating before entering indoors [3, 4].

Unlike conventional evaporative cooling systems, liquid desiccant systems do not require the air to be cooled below its dew point. Energy savings, relative to conventional vapor compression systems, of up to 40% can be achieved by using a desiccant assisted air conditioning system [5]. Liquid desiccant systems are also attractive due to their flexibility in operation [6] and their ability to operate under a relatively low regeneration temperature, between 60 and 75 °C, making possible the use of solar energy or waste heat. Thus, fossil fuel energy savings can be further increased.

The task of the dehumidification unit of a liquid desiccant system is to remove the moisture of the inlet air, by bringing it into contact with sprinkled liquid desiccant. In a counter flow dehumidifier (absorber), the strong, cool desiccant solution is sprayed at the top of the device. The ambient humid air enters the dehumidifier at the bottom. The end of the process finds the air dehumidified and the desiccant solution diluted, since the moisture is transferred from the air to the desiccant. A low moisture content of the process air can be obtained at the outlet of the absorber, by maintaining a low desiccant temperature or a high desiccant concentration at the inlet of the absorber [7].

The driving force for the mass transfer between the liquid desiccant solution and the air is the difference between the vapor pressure of the desiccant and the partial pressure of water vapor in the air [8]. Namely, moisture transfer takes place from the air to the desiccant solution only when the cool, strong desiccant solution vapor pressure is less than the water vapor pressure in the air.

Salt solutions like LiBr, LiCl or $CaCl_2$ are most commonly used as liquid desiccants, in comparison to glycols, since they have essentially zero vapor pressure [9]. Although glycols are less corrosive than salts, they have very low surface vapor pressure [10] and their

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evaporation contaminates the air, increasing the risks for respiratory irritations [11]. These evaporation losses are unacceptable in an occupied building; therefore salts dominate [12].

The physical properties as well as the dehumidification performance of the aforementioned aqueous solutions, have been investigated by many researchers [9, 13, 14, 15, 16, 17] while the heat and mass transfer performances in the dehumidification/regeneration process have been experimentally tested by others [5, 12, 18, 19, 20, 21].

Three models have been developed for the analysis of packed bed absorption dehumidifiers [8]; the finite difference model [10, 22], the effectiveness NTU model [23] and the model based on fitted algebraic equations [7, 24]. Among the above models, the finite difference model has proved that it provides the most accurate results. Indeed, the finite difference model developed by Elsayed et al [25] could predict the outlet conditions of air and solution in a dehumidifier. The most commonly accepted and widely used model is the onedimensional finite difference model developed by Factor and Grossman [22]. Good agreement also exists between the experimental findings and the finite difference model predictions developed by Martin and Goswami [26]. The adapted finite difference model of Fumo and Goswami [9] shows very good agreement with the experimental findings. Ren et al [27] found a close agreement between the experimental results and the predictions of the finite difference model when using different heat and mass transfer coefficients. Babakhani and Soleymani [28] developed a finite difference analytical solution of heat and mass transfer processes in a packed bed liquid desiccant regenerator, suitable for high desiccant flow rate, both for random and structured packings. When compared with reliable experimental data, the model results had a maximum difference of 6%.

Counter flow configuration is the most commonly used flow pattern for a liquid desiccant dehumidifier /regenerator and several researchers have analyzed it. Lof et al. [18] used experimental data for a counter flow packed column to verify the mass transfer and heat transfer coefficients of his previous analysis. Gandhidasan et al. [13] demonstrated the performance differences between different packing types inside a counter flow packed column using CaCl₂. Lazzarin et al. [14] investigated both theoretically and experimentally the optimum operative conditions in a counter flow packed tower using LiBr and CaCl₂. Raisul Islam et al. [29] validated a simplified linear coupled heat and mass transfer model for LiBr counter-flow absorbers against a numerical turbulent flow model, achieving deviations between 4.2-9.2%.

As shown above, many researchers have developed mathematical models of the coupled heat and mass transfer processes in the dehumidifier or the regenerator. Few researchers have analyzed the dehumidification potential of more than two liquid desiccant materials, by providing the vapor pressure expressions separately for each solution. Vapor pressure is an important property which determines the air humidity ratio in equilibrium with the desiccant at the interface. In addition, only a few studies that have developed the heat and mass transfer coefficients do have analysed the exact geometry of the apparatus and the packing characteristics.

In the present work, a mathematical model has been produced that permits reliable prediction of the absorber efficiency of an adiabatic counter flow packed column, using three different aqueous desiccant solutions (LiCl, LiBr and CaCl₂). Vapor pressure expressions for each desiccant solution were obtained from the work of Cisternas and Lam [30, 31]. This model has also taken into account the heat of absorption, which expresses the attributed energy to the solution during the condensation of the steam and affects the capability of the solution absorbing the water vapor. Chung and Ghosh [20] experimental data for LiCl solution were used to verify this theoretical analysis.

4.2.2 Thermodynamic parameters

The liquid desiccants used in this analysis are aqueous solutions of LiCl, LiBr and CaCl₂. In the standard atmospheric pressure, the boiling temperature of these salt solutions is over 1200 °C, while the boiling temperature of water is only 100 °C. Therefore, the surface pressure of liquid desiccant almost equals to the vapor pressure. The temperatures and vapor pressures of humid air and liquid desiccant should be the same, when liquid desiccant is in equilibrium with the humid air [36, 37]. Thus, vapor pressure, heat of absorption, heat and mass transfer coefficients and pressure drop are the most important parameters influencing the dehumidification potential of every liquid desiccant solution.

4.2.2.1 Vapor Pressure

The water partial vapor pressure of the desiccant solution determines the humidity ratio of the air that is in equilibrium with the desiccant at the interface. Thus, an expression relating the water partial vapor pressure of the desiccant solution with the temperature and the desiccant concentration is necessary.

Although the vapor pressure of single electrolyte aqueous solutions has been reported extensively in the literature, only Cisternas and Lam [30, 31], based on Kumar and

Patwardhan's [38, 39] method, have developed vapor pressure expressions for all three aqueous solutions following the same experimental procedure. Thus, the present study uses this work to provide analytic correlations for the vapor pressure of the examined aqueous desiccant solutions.

Liquid desiccants with lower water partial vapor pressure have limited ability for water evaporation, so they can absorb more moisture than the desiccants with higher water partial vapor pressure, under the same temperature and concentration. Among the examined desiccants, LiCl solution has the lowest water vapor pressure; thus, LiCl has the highest dehumidification capacity, whereas CaCl₂ has the lowest.

After obtaining the vapor pressure correlation equations, the equivalent humidity ratio of liquid desiccant W^{sat} can then be calculated with the equilibrium state of air, as shown in the following equation:

eq. 59
$$W^{sat}(T_s) = 0.622 \frac{p_{w,st}^{sat}(T_s)}{p_{atm} - p_{w,st}^{sat}(T_s)}$$

Based on the above equation, the psychrometric charts for all three liquid desiccant solutions, representing the status of the solution with the equilibrium status of humid air, can been developed. The thermodynamic properties of LiCl and CaCl₂ can be taken from Conde [15], while LiBr properties can be taken from McNeely [40]. The isoconcentration lines of the desiccants are almost coincident with the isorelative humidity lines of the air; however the lines of very low desiccant concentration lie near the saturated line of air and reversely, the lines of very high desiccant concentration lie near the low relative humidity lines of air. From the developed psychrometric charts, it could be derived that the absorber efficiency of the desiccants enhances with lower temperature and higher concentration and that LiCl has better absorption performance than the other examined desiccants.

4.2.2.2 Heat of absorption

Despite the fact that the saturation temperature of pure water vapor is a function only of pressure, the temperature of a solution that is in equilibrium with an overlying water vapor depends not only on pressure, but also on the concentration of salt in the solution. The solution temperature is always greater than the temperature of overlying saturated water vapor, because of the presence of the salt in the solution that prevents the water evaporation. The solution temperature increases with the increase in solution concentration.

During dehumidification (absorption procedure), water vapor molecules are taken up by the volume of the liquid desiccant while water vapor condensation occurs inside the liquid desiccant solution. Thus, the energy added to the liquid desiccant solution, during the condensation of 1 kg of water vapor is expressed by the differential absorption enthalpy (heat of absorption), given by the following expression [32]:

eq. 60
$$\Delta h_{abs} = h_{st}(p,T_s) - h_{w,s}(T_s,X)$$

The heat of absorption is greater than the condensation heat of saturated water vapor at the same pressure. The heat of absorption increases with the increase in concentration while it decreases with the increase in temperature. Figure 4-1 shows the heat of absorption as a function of solution temperature at the same concentration for all three desiccants.



Figure 4-1 Heat of absorption versus solution temperature at three different concentrations for the three desiccant solutions considered.

4.2.2.3 Pressure drop

Many investigators acknowledge the disadvantage of a high pressure drop through the packed bed dehumidifier, since the pressure drop directly relates the power consumed by the fan [43]. Apart from the packing and the bed physical characteristics, the pressure drop depends also on the desiccant and air mass flow rates [43]. The optimization of the flow rates is very important, since it may deliver more cooling with a pressure drop less than one half than that of a conventional system [44]. The pressure drop equation is given by Gandhidasan [43]:

eq. 61
$$\Delta p_{drybed} = 0.125 \left(\frac{c_1}{\text{Re}_a} + \frac{c_2}{\text{Re}_a^{0.5}} + c_3 \right) \frac{\rho_a V_a^2 a}{\varepsilon^{4.65}}$$

Where c_1 , c_2 and c_3 vary with packing type.

4.2.2.4 Heat & Mass Transfer Coefficients

Correlations for the mass transfer coefficients obtained for packed column desiccant air dehumidifiers are available in the literature. Since mass transfer coefficients must be associated with the individual liquid and gas phases, overall coefficients will not serve. Onda [41] developed empirical correlations for the gas and liquid phase mass transfer coefficients, applicable for random packings with organic solvents. When aqueous salt solutions are used, the liquid phase heat transfer correlation includes errors of 20%. Gandhidasan et al [13] calculated the heat and mass transfer coefficients in a CaCl₂ liquid desiccant system with random packing. Lof et al [18] used random packing to compare the experimental values of heat transfer coefficients with theoretical values based on McAdams relationship. He found though an average deviation of ±15% at high liquid flow rates and of ±30% at lower liquid flow rates. Potnis and Lenz [5] developed dimensionless liquid phase mass transfer correlations, based on experimental results with structured and random packings, using LiBr. Their correlation can predict experimental values within a 20% range, under low liquid flow rates.

However, in the present investigation, heat and mass transfer correlations that would be applicable for structure packings and for various types of desiccant solutions, is necessary. Therefore, the study of Chung et al [20] that is based on Treybal [42] and Buckingham Pi method was used. The developed empirical correlations were chosen as they can be used with PVC structured packing and other liquid desiccant materials than LiCl still be able to predict experimental data within ±10% for a wide range of operating conditions.

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The gas phase mass transfer correlation is:

eq. 62
$$\operatorname{K}_{G}a\left(\frac{M_{w}d_{eq}^{2}}{D_{\alpha}\rho_{\alpha}}\right) = 2.25 \cdot 10^{-4} (1-X)^{-0.75} \left(\frac{\dot{m}_{s}}{\dot{m}_{a}}\right)^{0.1} Sc^{0.333} \operatorname{Re}_{a}$$

Where:

eq. 63
$$Sc = \frac{\mu_a}{D\rho_a}, \operatorname{Re}_a = \frac{d\rho_a V_a}{\mu_a}$$

And from Treybal [42]:

$$eq. \ 64 \quad d = \frac{6(1-\varepsilon)}{a}$$

The gas phase heat transfer correlation is:

$$eq. 65 \quad h_G = LeK_GC_{p,ma}$$

4.2.2.5 **Performance indices**

The mass transfer performance of the dehumidifier is evaluated in terms of the dehumidification mass rate and the absorber efficiency. The dehumidification mass rate (kg_w/s) , is calculated by the following expression:

eq. 66
$$\dot{m}_{deh} = \dot{m}_a (W_{in} - W_{out})$$

The absorber efficiency is the ratio of the actual humidity ratio variance of the air passing through the dehumidifier to its variance under ideal conditions, as shown below:

eq. 67
$$n = \frac{W_{in} - W_{out}}{W_{in} - W^{sat}} 100\%$$

4.2.3 Mathematical model development

4.2.3.1 Assumptions

The absorber is considered as a glass column, d=0.1524m internal diameter, packed up to a depth of L=0.41m, with PVC structured packing of a specific surface α =223 m²/m³ [20]. A detailed description of the experimental test rig is available in Chung et al. [35]. The schematic diagram of the counter-flow packed column dehumidifier is shown in the figure below.



weak desiccant



In order to simplify the complexity of the configuration, the assumptions used in the calculations are:

- Adiabatic dehumidification process,
- One-dimensional analysis: heat and mass transfer occur only in the flow direction,
- Uniform flooding of the dehumidifier and heat transfer area equal to the mass transfer area; thus, transfer area equal to the specific surface area of the packing,
- Vapor pressure equilibrium between the vapor and the liquid at the interface,

- Constant Lewis number for every experiment, taken from the study of Chung et al [20],
- The water is transferred from the air into the desiccant solution through a stagnant film of liquid,
- Negligible radiation heat transfer, due to small temperature differences of the process.

4.2.3.2 Mathematical model

The model was developed as an extension of Rogdakis et al. [32] and Papaefthimiou [33, 34] work. Based on the energy and mass conservation laws, the developed model describes coherently the coupled heat and mass transfer processes taking place inside a counter-flow (upward air flow and downward desiccant flow) packed column liquid desiccant dehumidifier.

The packed column height Z is divided into 15 equal segments and the mass and energy balances are solved for each segment, from the top of the column Z = 0, until the bottom of the column, Z=1; thus, gradients of air temperature, humidity ratio, desiccant temperature, desiccant flow rate and concentration are calculated across the height of the column.

In order to begin calculations, the following initial parameters should be assumed:

- Temperature of ambient air
- Humidity ratio of ambient air
- Flow rate of air inlet
- Physical properties of ambient air
- Temperature of desiccant solution inlet
- Concentration of desiccant solution inlet
- Flow rate of inlet desiccant
- Lewis number
- Heat and mass transfer area of dehumidifier



Figure 4-3 Heat and mass transfer during adiabatic dehumidification

Air - desiccant interface

By definition, the humidity ratio is given by the following equation:

eq. 68
$$W = \frac{\dot{m}_w}{\dot{m}_a}$$

or after differentiating with dz:

$$eq. 69 \quad \frac{d\dot{m}_w}{dz} = \dot{m}_a \frac{dW}{dz}$$

However, the moisture that is removed from the air is absorbed by the solution, so $d\dot{m}_s = d\dot{m}_w$. So, the mass flow rate of the desiccant solution is given by the following expression:

eq. 70
$$\frac{d\dot{m}_s}{dz} = \dot{m}_a \frac{dW}{dz}$$

The mass balance equation for the liquid desiccant solution is:

eq. 71
$$d(\dot{m}_{s}X) = 0$$

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Differentiating this equation over height dz and using eq. 70, the concentration of the solution is obtained:

eq. 72
$$\frac{dX}{dz} = -\frac{\dot{m}_a}{\dot{m}_s} X \frac{dW}{dz}$$

The gradient of humidity ratio along column height expresses the rate of water vapour absorbed by the desiccant at the interface within height dz and can be expressed as follows:

eq. 73
$$\frac{dW}{dz} = -\frac{LK_G}{\dot{m}_a} \left(W^{\text{sat}} - W_{in} \right)$$

Where W^{sat} is the humidity ratio of the saturated air that is in equilibrium with the desiccant solution at the local solution temperature and concentration.

Air phase

Figure 4-3 represents, among others, the energy balance of the air phase in an adiabatic counter flow dehumidifier. Accordingly,

eq. 74
$$\dot{m}_a(h_a + dh_a) = \dot{m}_a h_a + dQ_s + dQ_L \Longrightarrow \dot{m}_a dh_a = dQ_s + dQ_L$$

The enthalpy of moist air is given by:

eq. 75
$$h_a = C_{p,da}T_a + W(C_{p,st}^{sat}T_a + \Delta h_{abs})$$

The term dQ_s represents the sensible heat load, which is transferred from the air to the desiccant solution, due to their temperature difference and it can be written as follows:

eq. 76
$$dQ_s = Lh_G(T_a - T_s)dz$$

Whereas the latent heat load dQ_L transferred from the air to the desiccant solution due to the water condensation is:

eq. 77
$$dQ_L = d\dot{m}_s (C_{p,st}^{sat} T_s + \Delta h_{abs})$$

However, the specific heat capacity of the moist air is:

$$eq. 78 \quad C_{p,ma} = C_{p,da} + WC_{p,st}^{sat}$$

Substituting eq. 70, eq. 73, eq. 76, eq. 77 and eq. 78 into eq. 74, the rate of air dry bulb temperature is obtained:

eq. 79
$$\frac{dT_a}{dz} = \frac{T_a - T_s}{C_{p,ma}} \left(\frac{Lh_G}{\dot{m}_a} - C_{p,st} \frac{sat}{dz} \frac{dW}{dz} \right)$$

Liquid desiccant phase

eq. 70 and eq. 73 give the mass of the water that is absorbed by the solution per unit height:

$$eq. 80 \quad \frac{d\dot{m}_s}{dz} = -LK_G \left(W^{sat} - W_{in} \right)$$

Figure 4-3 represents, among others, the energy balance of the liquid desiccant phase in an adiabatic counter flow dehumidifier. Accordingly,

eq. 81
$$\dot{m}_{s}h_{s} + dQ_{s} + dQ_{L} = (\dot{m}_{s} + d\dot{m}_{s})(h_{s} + dh_{s})$$

However, the term $d\dot{m}_s dh_s$ is negligibly small, so

$$eq. 82 \quad \dot{m}_s dh_s + h_s d\dot{m}_s = dQ_s + dQ_L$$

Taking into consideration eq. 70, eq. 74, $h_s = C_{p,s}(T_s - T_{s,in})$, $dh_s = C_{p,s}dT_s$ and differentiating eq. 82 over height dz, the rate of change of desiccant temperature T_s is produced:

$$eq. 83 \quad \frac{dT_s}{dz} = \frac{\dot{m}_a}{\dot{m}_s C_{p,s}} \left[\left(-C_{p,s} (T_s - T_{s,in}) + C_{p,st}^{sat} T_a + \Delta h_{abs} \right) \frac{dW}{dz} + C_{p,ma} \frac{dT_a}{dz} \right]$$

System of Equations

Finally, a system of five ordinary differential equations is constituted:

$$eq. 84 \quad \begin{cases} \frac{dT_a}{dz} = \frac{T_a - T_s}{C_{p,ma}} \left(\frac{Lh_G}{\dot{m}_a} - C_{p,st}^{sat} \frac{dW}{dz} \right) \\ \frac{dT_s}{dz} = \frac{\dot{m}_a}{\dot{m}_s C_{p,s}} \left[\left(-C_{p,s} (T_s - T_{s,in}) + C_{p,st}^{sat} T_a + \Delta h_{abs} \right) \frac{dW}{dz} + C_{p,ma} \frac{dT_a}{dz} \right] \\ \frac{dW}{dz} = -\frac{LK_G}{\dot{m}_a} \left(W^{sat} - W_{in} \right) \\ \frac{d\dot{m}_s}{dz} = \dot{m}_a \frac{dW}{dz} \\ \frac{dX}{dz} = -\frac{\dot{m}_a}{\dot{m}_s} X \frac{dW}{dz} \end{cases}$$

with the following boundary conditions:

eq. 85
$$\begin{cases} (T_a)_{z=0} = T_{a,in} \\ (T_s)_{z=L} = T_{s,in} \\ (W)_{z=0} = W_{in} \\ (\dot{m}_s)_{z=L} = \dot{m}_{s,in} \\ (X)_{z=L} = X_{in} \end{cases}$$

The five-order Runge - Kutta was used to numerically integrate the above five ordinary differential equations by using a trial step at the midpoint of an interval to cancel out lowerorder error terms. The model was developed using the Mathcad software [45] where all the thermodynamic characteristics of the absorber are calculated from the point where the liquid desiccant is sprinkled till the end of the packing.

4.2.4 Validation

Before proceeding to the examination of the effect of various operating parameters on the absorber efficiency of the adiabatic packed column dehumidifier, the validation of the developed computational model should precede. For this purpose, experimental data from Chung and Ghosh [20] were selected. Experimental errors were not attributed, however an average difference of 8% for mass transfer coefficients and 3% for heat transfer coefficients were noted between the measured and the calculated values. During these experiments, the column pressure drop was less than 14 cm of water per meter of packing height [20].

Table 4-1 provides a comparison between the experimental data with the simulated results for a packed column LiCl dehumidifier with PVC structure packing. In the case of outlet dry bulb temperature, the results of the computational program are almost identical to the experimental ones. The maximum deviation is -2.9% (-0.55 °C) and the mean deviation is -0.9% (-0.17 °C). Slightly larger discrepancies, up to 15.9%, (0.0007 kg_w/kg_{dα}) can be seen in the case of outlet humidity ratio; however, the mean deviation is still very small, at 3.0% (0.0001 kg_w/kg_{dα}). The simulated outlet solution temperature differ as much as 2.8% (0.50 °C) from the experimental values, the average difference being about 1.3% (0.24 °C). As it can be observed, the theoretical model shows very good agreement with the experimental results, having an excellent correspondence between most values.

	m _a	ms	T _{a,in}	T _{a,out,exp}	T _{a,out,sim}	Deviation	$\mathbf{W}_{a,in}$	$\mathbf{W}_{a,out,exp}$	$\mathbf{W}_{a,out,sim}$	Deviation	T _{s,in}	T _{s,out,exp}	T _{s,out,sim}	Deviation	v	Lewis
	kg s⁻¹	kg s⁻¹	°C	°C	°C	(%)	kg kg ⁻¹	kg kg ⁻¹	kg kg ⁻¹	(%)	°C	°C	°C	(%)	^	number
1	0.0231	0.3071	21.8	17.8	17.7	-0.50%	0.0112	0.0064	0.0064	0.20%	16.8	16.9	17.2	1.80%	31	1.02
2	0.026	0.3071	21.6	17.6	17.5	-0.20%	0.0111	0.0063	0.0063	0.40%	16.6	16.6	17	2.80%	31	0.97
3	0.0289	0.3071	20.4	17.3	17.2	-0.40%	0.0115	0.0066	0.0064	-2.90%	16.4	16.6	17	2.10%	31	1.00
4	0.0318	0.3071	21	17.8	17.7	-0.30%	0.0117	0.0067	0.0065	-2.40%	17	17.2	17.6	2.40%	31	1.07
5	0.0318	0.1919	24.1	19.1	18.6	-2.40%	0.0149	0.008	0.0074	-8.10%	16.9	18.3	18.3	0.00%	31	0.99
6	0.0318	0.2303	23.4	19.1	18.7	-1.90%	0.0147	0.0077	0.0072	-7.10%	17.5	18.6	18.6	0.40%	31	1.03
7	0.0318	0.2687	23.3	19.1	18.8	-1.60%	0.0147	0.0075	0.007	-6.30%	17.7	18.5	18.6	0.70%	31	0.99
8	0.0318	0.3071	24.1	18.9	18.3	-2.90%	0.0147	0.0073	0.0069	-6.00%	17.1	18	18	0.10%	31	0.99
9	0.0231	0.3071	23.3	20.6	20.5	-0.50%	0.0127	0.0058	0.006	3.00%	19.5	19.8	20	1.30%	37	0.75
10	0.026	0.3071	22.9	19.9	19.9	-0.20%	0.0126	0.0058	0.006	2.80%	18.9	19	19.5	2.60%	37	0.80
11	0.0289	0.3071	20.9	19.8	19.8	0.30%	0.0101	0.0051	0.0056	10.20%	19.4	19.4	19.9	2.50%	37	0.82
12	0.0318	0.3071	20.8	19.5	19.5	-0.10%	0.0105	0.0052	0.0057	9.00%	19.1	19.2	19.6	2.10%	37	0.90
13	0.0318	0.1919	21.1	18.9	18.7	-1.10%	0.0103	0.0049	0.0054	11.20%	17.6	18.4	18.4	0.30%	38	0.73
14	0.0318	0.2303	20.9	18.6	18.4	-1.00%	0.0103	0.0047	0.0053	13.10%	17.5	18.2	18.3	0.50%	38	0.81
15	0.0318	0.2687	20.8	18.5	18.3	-0.90%	0.0103	0.0046	0.0053	14.20%	17.5	18.1	18.2	0.60%	38	0.81
16	0.0318	0.3071	20.7	18.4	18.3	-0.60%	0.0103	0.0045	0.0052	15.90%	17.7	18	18.2	1.40%	38	0.85

Table 4-1 Comparison between the experimental data from literature and the simulated results for an adiabatic dehumidifier

4.2.5 Parametric analysis

4.2.5.1 **Overview**

Following the validation of the model with the experimental results, an extensive theoretical investigation was conducted to examine the effect of various operating parameters on the efficiency of the adiabatic packed column dehumidifier. The parametric study included three commonly used liquid desiccant solutions; LiCl-H₂O, LiBr-H₂O and CaCl₂-H₂O. Table 4-2 provides all the cases considered.

	Inlet values for the parametric analysis								
	m _a	ms	T _{a,in}	W _{a,in}	T _{s,in}	v			
Cases	Kg/s	Kg/s	°C	Kg/kg	°C	~			
			30						
			32						
Verietien of inlate in			34						
variation of inlet air	0.0653	0.0107	36	0.01486	14	30			
temperature			38						
			40						
			42						
				0.01086					
Variation of inlat				0.01286					
bumidity ratio	0.0653	0.0107	30	0.01486	14	30			
numuity ratio				0.01686					
				0.01886					
	0.107								
	0.0856								
Variation of inlet air	0.0653								
mass flow rate	0.0428	0.0107	30	0.01486	14	30			
mass now rate	0.0214								
	0.0107								
	0.00642								
					10				
Variation of inlet					14				
solution temperature	0.0653	0.0107	30	0.01486	20	30			
solution temperature					25				
					30				
						0.3			
						0.32			
Variation of inlet						0.34			
solution	0.0653	0.0107	30	0.01486	14	0.36			
concentration						0.38			
						0.4			
						0.42			
		0.00653							
		0.00816							
Variation of inlet		0.0107							
solution mass flow	0.0653	0.01633	30	0.01486	14	30			
rate		0.03265							
		0.0653							
		0.1306							

Table 4-2 Parametric analysis in adiabatic dehumidification

A much better understanding though of the difference in the dehumidification potential of the three salt solutions would be the study of the distribution of the air humidity ratio along the dehumidifier's height. Fortunately, the developed computational program not only is able to predict the outlet values, but it can also produce precise estimations of these values along the height of the absorber. So, the study of the distribution of the air humidity ratio along the dehumidifier's height was implemented with the starting values: $T_{a,in}=36$ °C, $W_{a,in}=0.01486 \text{ kg}_w/\text{kg}_{d\alpha}$, $\dot{m}_a = 0.0653 \text{ kg/sec}$, $T_{s,in}=14$ °C, $\dot{m}_s = 0.0107 \text{ kg/sec}$ and X= 30%. Similarly, the distribution of the air and desiccant solution temperature along the dehumidifier's height was calculated, by setting the same starting values.

4.2.5.2 Sensitivity analysis

A sensitivity analysis was performed on the inlet and outlet variables of both air and solution, in order to predict the effect of the inlet variables on the performance of the dehumidifier. The independent variables are the inputs; $T_{a,in}$, W_{in} , m_a , $T_{s,in}$, X_{in} , $m_{s,in}$ and the dependent are the outputs; $T_{a,out}$, W_{out} , X_{out} , n, m_w.

The standardized regression coefficients that have been calculated indicate the degree of influence of every input on the output values (Figure 4-4, Table 4-3). The larger coefficient of the independent variable, the greater influence on the dependent variable has, whereas a negative coefficient indicates that the dependent variable is decreased when the corresponding independent variable is increased. According toFigure 4-4 and Table 4-3, T_{a,out} strongly depends on m_a and X_{in}, whereas the absorber efficiency n depends on m_a and W_{in}. On the other hand, the m_w is very little influenced by the change of every independent variable, for all three solutions examined.



Figure 4-4 Coefficients of independent variables for aqueous solutions of LiCl, LiBr and CaCl₂

$Y=a + b^{*}T_{a,in} + c^{*}m_{a} + d^{*}W + e^{*}T_{s,in} + f^{*}m_{s,in} + g^{*}X_{in}$										
LiCl										
Y	а	b	С	d	е	f	g			
T _{a,out}	-13.82772	0.80036	139.38627	-2.72350	0.27722	1.14487	6.94343			
\mathbf{W}_{out}	-0.00523	0.00004	0.07064	0.83500	0.00002	0.00083	-0.00020			
X _{out}	0.00804	-0.00001	-0.01033	-0.29650	0.00001	0.00073	0.98233			
n	0.79761	-0.00377	-7.26283	1.48500	0.00076	-0.08358	-0.21542			
m _w	-0.00012	0.00000	0.00044	0.01077	0.00000	0.00001	0.00019			
LiBr										
Y	а	b	С	d	е	f	g			
T _{a,out}	-13.84502	0.80104	139.38790	-2.71650	0.27676	1.14371	6.98121			
\mathbf{W}_{out}	-0.00443	0.00004	0.05939	0.83600	0.00003	0.00070	0.00046			
X _{out}	0.00650	0.00000	-0.00768	-0.29750	0.00002	0.00057	0.98687			
n	0.78253	-0.00421	-7.24812	2.87500	0.00098	-0.08004	-0.22472			
m _w	-0.00009	0.00000	0.00034	0.01071	0.00000	0.00001	0.00011			
			Ca							
Y	а	b	С	d	е	f	g			
T _{a,out}	-13.86775	0.80155	139.38858	-2.71150	0.27642	1.14306	7.04106			
\mathbf{W}_{out}	-0.00412	0.00003	0.05121	0.83500	0.00003	0.00059	0.00195			
\mathbf{X}_{out}	0.00457	0.00001	-0.00572	-0.29750	0.00004	0.00048	0.99280			
n	0.76920	-0.00465	-7.23548	4.95500	0.00117	-0.07598	-0.27308			
m _w	-0.00006	0.00000	0.00025	0.01077	0.00000	0.00001	0.00000			

 Table 4-3 Linear regression equations for calculating the output thermodynamic properties of air and solution

	L	iCl	Li	iBr	CaCl ₂		
	R Square	Adjusted R Square	R Square	Adjusted R Square	R Square	Adjusted R Square	
T _{a,out}	0.9238	0.9095	0.9236	0.9093	0.9235	0.9092	
\mathbf{W}_{out}	0.8571	0.8303	0.8696	0.8451	0.8823	0.8603	
X _{out}	0.9996	0.9996	0.9998	0.9997	0.9999	0.9998	
n	0.7991	0.7614	0.7944	0.7558	0.7895	0.7500	
m _w	0.9166	0.9009	0.9207	0.9058	0.9212	0.9065	

Table 4-4 R-Square and adjusted R-Square for all three desiccant solutions

The following remarks can be made:

- R-Square and adjusted R-Square, that represent the quality of the linear fit, were found greater than 78.95% and less than 100% (Table 4-4), for each output variable and all three solutions.
- Since the input variables have different measurement scales, the regression coefficient of each input variable found at the end of a regression analysis was transformed to a standardized regression coefficient, in order to compare the relative influence of the variable.
- The influence of the input variables, as defined by the standardized regression coefficients, on the output values has been categorised in descending order: X_{in} (-6.9% to 166.1%), m_a (-87.5% to 73.9%), W_{in} (-1.4% to 78.1%), $T_{a,in}$ (-16.3% to 63.8%), $T_{s,in}$ (-50.0% to 24.9%) and $m_{s,in}$ (-6.4% to 5.6%). The results of the sensitivity analysis show that the output conditions are found to be most sensitive to the air mass flow rate and the inlet solution concentration change, but least sensitive to the solution flow rate.

The following subchapters are dedicated to the thorough examination of the effect of each parameter separately.

4.2.5.3 Air inlet temperature

The air inlet temperature values studied are considerably higher than the desiccant inlet temperature. This means that more sensible heat is transferred to the desiccant stream and so, the desiccant temperature increases, resulting in an increase in interfacial equilibrium humidity. The potential for mass transfer is thus reduced and consequently, both the dehumidification efficiency and the dehumidification mass rate are reduced. However, the higher specific thermal capacity, the less prone to temperature changes a material is. So, since LiBr and CaCl₂ have less specific thermal capacity than LiCl, under the same temperature and concentration, the increase in air inlet temperature has a stronger effect on LiBr and CaCl₂.

This effect is clearly seen in Figure 4-5, where the dehumidification mass rate is decreased by 7 % for LiCl, by 11% for LiBr and by 15% for CaCl₂. A similar trend can be observed with the absorber efficiency.


Figure 4-5 Effect of inlet air dry bulb temperature on the dehumidification mass rate and dehumidification efficiency for an adiabatic dehumidifier

4.2.5.4 Air inlet humidity ratio

Figure 4-6 shows that increasing air inlet humidity ratio by 74%, substantially increases the dehumidification mass rate of LiCl by 161%, of LiBr by 243% and of CaCl₂ by 400%. Similarly, the absorber efficiency is increased by 8% for LiCl, 19% for LiBr and 40% for CaCl₂.This happens because a higher humidity ratio implies a higher air vapor pressure and thus, a higher average water vapor pressure difference between the air and the desiccant. Consequently, higher potential for mass transfer from the gas to the liquid phase is established.



Figure 4-6 Effect of inlet air humidity ratio on the dehumidification mass rate and dehumidification efficiency for an adiabatic dehumidifier

Figure 4-6 also indicates that the dehumidification efficiency of CaCl₂ is subject to greater changes, compared to LiCl, due to the lower water vapor pressure of CaCl₂. This phenomenon implies that the efficiency of a liquid desiccant system installed in an area with low air humidity ratios is highly dependent on the type of desiccant used. In more humid areas however, it is unlikely that the type of desiccant would appreciably affect the system efficiency.

4.2.5.5 Air flow rate

For all desiccant solutions examined, increasing the air flow rate by a factor of 15 increases the dehumidification mass rate by approximately 82%, while decreases the absorber efficiency by almost 90% (Figure 4-7). This is happening because when the air flow rate is increased, the residence time for the air in the dehumidifier is decreased, resulting in a decrease of the mass transfer coefficient, a higher outlet air humidity ratio and thus, a lower absorber efficiency.



Figure 4-7 Effect of inlet air mass flow rate on the dehumidification mass rate and dehumidification efficiency for an adiabatic dehumidifier

Higher outlet humidity ratio means greater difference in the average water vapor pressure and higher dehumidification mass rate. But as shown in Figure 4-7, the slope of the dehumidification mass rate is greater at air flow rates less than 0.05 kg/sec. This happens because under high flow rates, the absorption heat released during the dehumidification process is greater than the cooling effect due to the low solution temperature. So, the solution is heated by the absorption heat and the dehumidification rate is thus decreased.

4.2.5.6 **Desiccant inlet temperature**

As indicated by Figure 4-8, increasing desiccant inlet temperature by 200% decreases the dehumidification mass rate, by 27% in the case of LiCl, 41% in LiBr and by 56% in CaCl₂. This happens because the increase in desiccant temperature increases the water vapour pressure of the desiccant and so, the average water vapor pressure difference between the air and desiccant in the dehumidifier is decreased. This leads to a higher air outlet humidity ratio and consequently, to a lower dehumidification mass rate.

However, increasing desiccant inlet temperature increases the dehumidifier efficiency by 9% for LiCl, 14% for LiBr and 19% for CaCl₂.



Figure 4-8 Effect of desiccant inlet temperature on the dehumidification mass rate and absorber efficiency for an adiabatic dehumidifier

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4.2.5.7 Desiccant inlet concentration

As shown in Figure 4-9, the 40% increase in desiccant inlet concentration caused a significant increase in the dehumidification mass rate, by 22% in the case of LiCl and by 17% in LiBr. The reason may be as follows. Increasing the desiccant inlet concentration decreases the desiccant surface vapor pressure, resulting in a greater driving force for mass transfer from the gas to the liquid phase. This leads to a lower air outlet humidity ratio and, hence, higher dehumidification mass rate. Under real conditions though, this dehumidification mass rate increase is not unlimited, since the desiccant surface vapor pressure does not only depend on the concentration, but also on the temperature of the solution. A very concentrated desiccant would result in an increase of the absorption heat released, which would heat the solution and decrease the average water vapor pressure difference between the air and desiccant.



Figure 4-9 Effect of desiccant inlet concentration on the dehumidification mass rate and absorber efficiency for an adiabatic dehumidifier

However, in the case of $CaCl_2$, the dehumidification mass rate was almost unaffected, recording a decrease by 1%. This is happening because $CaCl_2$ has higher water vapor pressure than LiCl and LiBr, under the same conditions; thus, a water vapor pressure change in $CaCl_2$ will have far less impact than the same change in LiCl or LiBr.

Increasing desiccant inlet concentration decreases both W^{sat} and W_{out} . Since these values offset each other, the absorber efficiency remains almost unchanged.

4.2.5.8 **Desiccant flow rate**

The dehumidification mass rate increased with the desiccant flow rate (Figure 4-10), by 12% for LiCl, 20% for LiBr and 28% for CaCl₂. This is happening because increasing the desiccant flow rate decreases the contact time between the desiccant and the air and so, decreases the variation of the desiccant concentration and temperature through the dehumidifier. As a result, the variation of the surface vapor pressure of the desiccant through the dehumidifier is decreased and, hence, the average water vapor pressure difference between the desiccant and air in the dehumidifier is increased. This leads to a lower air outlet humidity ratio and a higher dehumidification mass rate.



Figure 4-10 Effect of desiccant inlet mass flow rate on the dehumidification mass rate and absorber efficiency for an adiabatic dehumidifier

The absorber efficiency increased with a similar rate, due to the lower air outlet humidity ratio. Also, increasing the desiccant flow rate increases the mass transfer coefficient between the desiccant and the air in the dehumidifier, resulting in a further increase in the dehumidification efficiency.

It has to be noted here that under real low flow conditions, the increase of desiccant flow rate increases the wetting of the packing, thus increasing the mass transfer area between the two phases. At higher flows though, this effect is not observed. Therefore, at lower flow rates, less than 0.04 kg/sec, the absorber efficiency and the dehumidification mass rate increase more rapidly than at higher flow rates.

4.2.5.9 Liquid desiccant solution

The question is which liquid desiccant solution has better performance under the same operating conditions. In this work, some effort was made to compare three salt solutions, namely LiCl, LiBr and CaCl₂, against their dehumidification potential along the absorber's height. Indeed, the humidity ratio distribution (Figure 4-11) confirms that the absorber using LiCl has the best efficiency, as it can reduce humidity ratio by 10%, followed closely the absorber using LiBr (8%). The absorber using CaCl₂ seems to have the worst efficiency, as it reduces humidity ratio by 6%, under the same inlet conditions.



Figure 4-11 Humidity ratio distribution along adiabatic absorber's height

The variation of the moisture content in solution (kg_w/kg_s) along the absorber's height (Figure 4-12) shows the same result. At the desiccant outlet, which is the bottom of the absorber, LiCl has absorbed the largest amount of moisture from the air, whereas CaCl₂ has absorbed the least.



Figure 4-12 Water concentration distribution along adiabatic absorber's height



Figure 4-13 Distribution of air dry bulb temperature and solution temperature along adiabatic absorber's height

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Figure 4-13 shows that the air and desiccant temperature distribution along the absorber's height is almost the same for all three desiccant materials. This result was expected, as the effect of both air and desiccant inlet temperature (Figure 4-5, Figure 4-8) is much weaker than that of the humidity ratio (Figure 4-6) on the dehumidification mass rate of a liquid desiccant solution. However, it must be noted here that the difference between the desiccant outlet and inlet temperature is much higher than that of the air outlet and inlet temperature is much higher than that of the air outlet and inlet temperature is not allow great temperature changes, whereas the lower desiccant flow rate allows more latent heat to be absorbed, resulting in a great difference between the inlet and outlet desiccant temperature.

4.3 REGENERATOR

4.3.1 Overview

Thermally driven liquid desiccant cooling systems have emerged as a potential alternative to conventional vapor compression systems for cooling and air conditioning. The principle of their operation has been known for years. In the dehumidification process, the strong desiccant solution that has been brought into contact with the air absorbs the moisture from the air and gets diluted. After that, the desiccant must be regenerated to a useful level of concentration.

The regenerator is one of the most significant heat and mass transfer components in a liquid desiccant air-conditioning system, whose efficiency directly influences system performance [47, 48, 49]. The largest energy requirement associated with the liquid desiccant system is the heat required for desiccant regeneration, so the effectiveness of the desiccant regeneration process greatly influences the overall system performance. The regeneration of liquid desiccant can be driven by solar energy, waste heat or any other low-grade heat source [16].

Several literature reviews have been dedicated to the investigation of the performance of liquid desiccant dehumidifiers and regenerators [9, 13, 14, 15, 17, 50], while others have experimentally tested the heat and mass transfer performances in the dehumidification/regeneration process [5, 12, 18, 19, 20, 36, 52].

There are three main solutions to predict the heat and mass transfer performance of a packed-type dehumidifier/regenerator; the finite difference model [10, 22], the effectiveness NTU model [23] and the model based on fitted algebraic equations [7, 24].

The finite difference model, which is based on fundamental equations and gives a numerical solution, works well when the processed air and liquid desiccant are in parallel flow [53]. The finite difference model developed by Elsayed et al [25] could predict the heat and mass transfer in packed beds during the air dehumidification mode and the solution regeneration mode. The most commonly accepted and widely used model is the one-dimensional finite difference model developed by Factor and Grossman [22]. Good agreement also exists between the experimental findings and the finite difference model predictions developed by Martin and Goswami [26]. The adapted finite difference model of Fumo and Goswami [9] shows very good agreement with the experimental findings. Ren et al [27] found a close agreement between the experimental results and the predictions of the finite difference

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model when using different heat and mass transfer coefficients. Babakhani and Soleymani [28] developed a finite difference analytical solution of heat and mass transfer processes in a packed bed liquid desiccant regenerator, suitable for high desiccant flow rate, both for random and structured packings. When compared with reliable experimental data, the model results had a maximum difference of 6%.

However, in the counter-flow configuration, which is the most commonly used flow pattern for a liquid desiccant dehumidifier/regenerator, the outlet conditions of the desiccant are unknown. An iterative solution is thus necessary, until the results converge to the known inlet conditions [25]. Thus, the effectiveness NTU model [23] and the model based on fitted algebraic equations [7, 24] are used. Gandhidasan [8] developed a simplified model for the preliminary design of an air dehumidification process. Lof et al [18] studied the reconcentration of a LiCl solution in a solar air collector and used experimental data for a counter flow packed regenerator to verify the mass transfer and heat transfer coefficients. Gandhidasan et al [13] demonstrated the performance differences between different packing types inside a counter flow packed column using CaCl₂. Lazzarin et al [14] investigated both theoretically and experimentally the optimum operative conditions in a counter flow packed tower using LiBr and CaCl₂. Raisul Islam et al. [29] validated a simplified linear coupled heat and mass transfer model for LiBr counter-flow absorbers against a numerical turbulent flow model, achieving deviations between 4.2-9.2%. Chen et al [25] presented an integrated analytical solution of adiabatic heat and mass transfer in packedtype liquid desiccant system, in both parallel-flow and counter-flow configurations. This analytical solution suited only high desiccant flow rate conditions, since the desiccant concentration was assumed to be constant throughout the absorber/regenerator. Based on the work of Factor and Grossman [22], Chengqin et al [54] developed a controlling equation for the quasi-equilibrium conditions of the two fluid streams. Numerical simulations of one dimensional heat and mass transfer model under practical conditions were performed, by dividing the heat and mass transfer area into a number of control volumes. A validated integrated analytical solution of an adiabatic liquid desiccant system is adopted by Xiao et al [55]. The analytical solution, suitable for high desiccant flow rate conditions, showed that the system experiences higher COP when the ambient air temperature and moisture content are higher. Audah et al [56] studied a solar-powered liquid desiccant system in Beirut using parabolic solar concentrators as a heat source for regenerating the liquid desiccant. Although their results agree well with the literature, their fourth order Runge–Kutta model used previously developed mathematical models and assumed constant heat and mass

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transfer coefficients. An analytical solution method was put forward by Peng et al [57] to predict the performance of solar collector / regenerator, based on the correlative assumptions and proved validity by comparing with numerical solution method and experiments. Though the results of numerical and analytical solution agree well, the errors between both still exist; for the outlet concentration, the relative error between the numerical and analytical solutions was 2.7-9.7%.

In the present work, a numerical model has been produced that permits reliable prediction of the regenerator efficiency of an adiabatic counter flow packed column. The advantage of this model is its ability to easily adapt under different operational conditions, different flow configurations (parallel or counter) and different liquid desiccant solutions (salts or glycols). Utilizing the arithmetic method of Runge-Kutta, the model not only produces the outlet values of air and solution, but it is able to predict also the conditions along the height of the device. Fumo and Goswami's [9, 50] experimental data for LiCl solution were used to verify the theoretical model. Another innovation of this research is the comparison method of the regeneration potential between more than two liquid desiccant solutions; namely LiCl-H₂O, LiBr-H₂O and CaCl₂-H₂O. Vapor pressure expressions for each desiccant solution were obtained from the work of Cisternas and Lam [30, 31].

4.3.2 Thermodynamic parameters

4.3.2.1 Vapor Pressure

The difference between the vapor pressure of the desiccant and the partial pressure of water vapor in the air is the actual driving force for the mass transfer between the liquid desiccant solution and the air [8]. During regeneration, moisture transfer takes place from the desiccant solution to the air only when the desiccant's equilibrium water vapor pressure is higher than the water vapor pressure in the air.

An expression relating the water partial vapor pressure of the desiccant solution with the temperature and the desiccant concentration has been already developed by Fumo and Goswami [9, 50], based on the work of Uemura [58, 59] and was used for the validation of the mathematical model.

As the parametric analysis includes the comparison of three different desiccant solutions, it necessitates the use of an expression that would be commonly valid for all solutions and that would have been developed under the same procedure. Unfortunately, the vapor pressure expressions that Uemura [58, 59] developed are valid only for LiCl and LiBr. Only

Cisternas and Lam [30, 31], based on Kumar and Patwardhan's [38, 39] method, have developed the needed expression. Thus, the parametric analysis of the present study uses the analytic correlations of Cisternas and Lam [30, 31] for the vapor pressure.

After obtaining the vapor pressure correlation equations, the equivalent humidity ratio of liquid desiccant $W^{sat}(T_s)$ can then be calculated with the equilibrium state of air, as shown in the following equation:

eq. 86
$$W^{sat}(T_s) = 0.622 \frac{p_{w,s}^{sat}(T_s)}{p_{atm} - p_{w,st}^{sat}(T_s)}$$

4.3.2.2 Heat of evaporation

During regeneration (desorption procedure), water vapor molecules of the liquid desiccant solution evaporate from the liquid-vapor interface and are transferred to the process air. The heat of evaporation expresses the energy required to transform a given quantity of moisture into gas at a given pressure. Thus, the energy required for the evaporation of 1 kg of water from the desiccant solution is expressed by the heat of evaporation, given by the following expression [32]:

eq. 87
$$\Delta h_{evap} = h_v(p,T_s) - h_{w,s}(T_s,X)$$

4.3.2.3 Pressure drop

Many investigators acknowledge the disadvantage of a high pressure drop through the packed bed regenerator, since the pressure drop directly relates the power consumed by the fan [43]. Apart from the packing and the bed physical characteristics, the pressure drop depends also on the desiccant and air mass flow rates [43]. The optimization of the flow rates is very important, since it may deliver more cooling with a pressure drop less than one half than that of a conventional system [44]. The pressure drop equation is given by Gandhidasan [43]:

eq. 88
$$\Delta p_{drybed} = 0.125 \left(\frac{c_1}{\text{Re}_a} + \frac{c_2}{\text{Re}_a^{0.5}} + c_3 \right) \frac{\rho_a V_a^2 a}{\varepsilon^{4.65}}$$

Where c_1 , c_2 and c_3 vary with packing type.

4.3.2.4 Heat & Mass Transfer Coefficients

Correlations for the mass transfer coefficients obtained for packed column desiccant air dehumidifiers/regenerators are available in the literature. Onda et al [41] developed

empirical correlations for the gas and liquid phase mass transfer coefficients, applicable for random packings with organic solvents. When aqueous salt solutions are used, the liquid phase heat transfer correlation includes errors of 20%. Gandhidasan et al [13] calculated the heat and mass transfer coefficients in a CaCl₂ liquid desiccant system with random packing. Lof et al [18] used random packing to compare the experimental values of heat transfer coefficients with theoretical values based on McAdams relationship [51]. He found though an average deviation of ±15% at high liquid flow rates and of ±30% at lower liquid flow rates. Potnis and Lenz [5] developed dimensionless liquid phase mass transfer correlations, based on experimental results with structured and random packings, using LiBr. Their correlation can predict experimental values within a 20% range, under low liquid flow rates.

However, in the present investigation, heat and mass transfer correlations that would be applicable for both structured and random packings, as well as for various types of desiccant solutions, are necessary. Therefore, the empirical correlations of Chung and Ghosh [20], based on Treybal [42] and Buckingham Pi method, were used since they are able to predict experimental data within ±10% for a wide range of operating conditions.

The overall mass transfer correlation for structured packing is:

eq. 89
$$K_G a \left(\frac{M_w d_{eq}^2}{D_a \rho_a} \right) = 2.25 \cdot 10^{-4} (1 - X)^{-0.75} \left(\frac{\dot{m}_s}{\dot{m}_a} \right)^{0.1} Sc^{0.333} Re_a$$

The overall mass transfer correlation for random packing is:

eq. 90
$$K_G a \left(\frac{M_w d_{pack}^2}{D_a \rho_a} \right) = 1.326 \cdot 10^{-4} (1 - X)^{-0.94} \left(\frac{\dot{m}_s}{\dot{m}_a} \right)^{0.27} Sc^{0.333} Re_a^{1.16}$$

Where

eq. 91
$$Sc = \frac{\mu_a}{D\rho_a}, Re_a = \frac{d\rho_a V_a}{\mu_a}$$

And from Treybal [42]:

eq. 92
$$d = \frac{6(1-\varepsilon)}{a}$$

The overall heat transfer correlation is:

eq. 93
$$h_G = LeK_GC_{p,ma}$$

4.3.2.5 Performance indices

The mass transfer performance of the regenerator is evaluated in terms of the regeneration rate and the regenerator efficiency. The regeneration rate (kg_w/s) is calculated by the following expression:

eq. 94
$$\dot{m}_{w} = \dot{m}_{a}(W_{out} - W_{in})$$

Since liquid desiccant regeneration is a process driven by thermal energy, it is significant to have another index to evaluate the energy utilization efficiency for the liquid desiccant regeneration [60]. This index is called regeneration thermal efficiency and is defined as:

eq. 95
$$n = \frac{\dot{m}_a \Delta h_{evap} (W_{out} - W_{in})}{Q_s}$$

where Q_s is the total heat input to the desiccant solution, which is equal to the enthalpy increase of air and desiccant solution.

The expressions for calculating the amount of heat transferred from the desiccant solution to the air in an adiabatic regenerator, due to convection and evaporation, are defined as:

eq. 96
$$\dot{Q}_{conv} = h_G a (T_s - T_a)$$

eq. 97 $\dot{Q}_{evap} = K_G a \Delta h_{evap} (W^{sat} - W_a)$

4.3.3 Mathematical model development

4.3.3.1 Assumptions

In this research, two different geometries of the apparatus have been considered; the first one serves the model validation and the second the parametric analysis.

Since the developed mathematical model has been validated against experimental data taken from Fumo and Goswami [9, 50] the physical dimensions of the model should be the same with the experimental apparatus of Fumo and Goswami [9, 50]. In this case, the regenerator consists of a packed column of d=0.254m diameter (0.24 m internal diameter) and L=0.6m depth. The packing used was polypropylene Rauschert Hiflow rings with specific surface area α =210 m²/m³ and the desiccant solution was LiCl-H₂O. Contrary to the Fumo and Goswami [9, 50], no correction factor has been used in the model to account for the reduction of area for mass transfer in the contact column.

Since this research was conducted as an extension of the authors' previous work [33], the geometry as well as the initial conditions for the parametric study should remain the same, where possible. Therefore, in this case, the regenerator column has d=0.1524m internal diameter and L=0.41m depth. The packing used is PVC structured packing of a specific surface α =223 m²/m³. The schematic diagram of the counter-flow packed column regenerator is shown in Figure 4-14.

In order to simplify the complexity of the configuration, the assumptions used in the calculations are:

- Adiabatic regeneration process,
- One-dimensional analysis: heat and mass transfer occur only in the flow direction,
- Uniform flooding of the regenerator and heat transfer area equal to the mass transfer area; thus, transfer area equal to the specific surface area of the packing,
- Vapor pressure equilibrium between the vapor and the liquid at the interface,
- Laminar desiccant flow,
- Negligible radiation heat transfer, due to small temperature differences of the process.



strong desiccant

Figure 4-14 Regenerator device

4.3.3.2 Mathematical model

The numerical model was developed as an extension of Rogdakis et al [32] and Papaefthimiou et al [33, 34] work. The scope of this approach was to develop a model that minimizes computational time without sacrificing the accuracy, with the aim of developing a validated tool for the design of installed systems. Thus, based on the energy and mass conservation laws, the developed numerical model describes the coupled heat and mass transfer processes taking place inside a counter-flow (upward air flow and downward desiccant flow) packed column liquid desiccant regenerator.

The packed column height Z is divided into equal segments and the mass and energy balances are solved for each segment. These calculations start at the top of the column, Z=0 until the bottom of the column Z = 1. Thus, gradients of air temperature, humidity ratio, desiccant temperature, desiccant flow rate and concentration are calculated across the height of the column. In order to begin calculations, the following initial parameters have been set:

- Temperature of ambient air
- Humidity ratio of ambient air
- Flow rate of air inlet
- Physical properties of ambient air
- Temperature of desiccant solution inlet
- Concentration of desiccant solution inlet
- Flow rate of inlet desiccant
- Lewis number
- Heat and mass transfer area of regenerator.

The developed numerical model was then validated against experimental data taken from Fumo and Goswami [9, 50].



Figure 4-15 Heat and mass transfer during regeneration

Air - desiccant interface

By definition, the humidity ratio is given by the following equation:

eq. 98
$$W = \frac{\dot{m}_w}{\dot{m}_a}$$

or after differentiating with dz:

$$eq. 99 \quad \frac{d\dot{m}_w}{dz} = \dot{m}_a \frac{dW}{dz}$$

During regeneration however, the amount of moisture that is evaporated from the solution is transferred to the air, so:

eq. 100
$$d\dot{m}_s = d\dot{m}_w$$

So, the mass flow rate of the desiccant solution is given by the following expression:

$$eq. \ 101 \quad \frac{d\dot{m}_s}{dz} = \dot{m}_a \frac{dW}{dz}$$

The mass balance equation for the liquid desiccant solution gives:

$$eq. 102 \quad d(\dot{m}_{s}X) = 0$$

Differentiating this equation over height dz and using eq. 101, the concentration of the solution is obtained:

eq. 103
$$\frac{dX}{dz} = -\frac{\dot{m}_a}{\dot{m}_s} X \frac{dW}{dz}$$

The gradient of humidity ratio along column height expresses the rate of water vapour that is evaporated from the desiccant at the interface within height dz and can be expressed as follows:

$$eq. \ 104 \quad \frac{dW}{dz} = \frac{LK_G}{\dot{m}_a} \left(W^{\text{sat}} - W_{in} \right)$$

Where W^{scat} is the humidity ratio of the saturated air that is in equilibrium with the desiccant solution at the local solution temperature and concentration.

Air phase

Figure 4-15 represents the energy and mass balance in an adiabatic counter flow regenerator. Accordingly,

eq. 105
$$\dot{m}_a h_a + dQ_S + dQ_L = \dot{m}_a (h_a + dh_a) \Longrightarrow \dot{m}_a dh_a = dQ_S + dQ_L$$

The enthalpy of moist air is given by:

eq. 106
$$h_a = C_{p,da} T_a + W (C_{p,st}^{sat} T_a + \Delta h_{evap})$$

or

$$eq. 107 \quad dh_a = C_{p,da} dT_a + dW (C_{p,st}^{sat} T_a + \Delta h_{evap}) + W C_{p,st}^{sat} dT_a$$

The term dQ_s in eq. 105 represents the sensible heat load, which is transferred from the desiccant solution to the air, due to their temperature difference and it can be written as follows:

eq. 108
$$dQ_S = Lh_G(T_s - T_a)dz$$

Whereas the latent heat load dQ_L transferred from the desiccant solution to the air due to the water evaporation is:

$$eq. 109 \quad dQ_L = d\dot{m}_s (C_{p,st}^{sat} T_s + \Delta h_{evap})$$

However, the specific heat capacity of the moist air is:

eq. 110
$$C_{p,ma} = C_{p,da} + W C_{p,st}^{sat}$$

Substituting eq. 107, eq. 108 and eq. 109 into eq. 105 and using eq. 101 and eq. 110, the rate of air dry bulb temperature is obtained:

$$eq. 111 \quad \dot{m}_{a} \left[C_{p,da} dT_{a} + dW (C_{p,st}^{sat} T_{a} + \Delta h_{evap}) + W C_{p,st}^{sat} dT_{a} \right] = dQ_{S} + dQ_{L}$$

$$eq. 112 \quad \frac{dT_{a}}{dz} = \frac{T_{s} - T_{a}}{C_{p,ma}} \left(\frac{Lh_{G}}{\dot{m}_{a}} + C_{p,st}^{sat} \frac{dW}{dz} \right)$$

Liquid desiccant phase

Combining eq. 101 with eq. 104 gives the mass of the water evaporated from the solution per unit height:

eq. 113
$$\frac{d\dot{m}_s}{dz} = LK_G \left(W^{sat} - W_{in} \right)$$

According to Figure 4-15,

eq. 114
$$(\dot{m}_s + d\dot{m}_s)(h_s + dh_s) = \dot{m}_s h_s + dQ_s + dQ_L$$

However, the term $d\dot{m}_s dh_s$ is negligibly small, so

eq. 115
$$\dot{m}_s dh_s + h_s d\dot{m}_s = dQ_s + dQ_L$$

Also, the enthalpy of the solution is given by the following expression:

eq. 116
$$h_s = C_{p,s}(T_s - T_{s,in}) \Longrightarrow dh_s = C_{p,s}dT_s$$

Taking into consideration eq. 101, eq. 105, eq. 107 and eq. 116, eq. 115 becomes:

$$\dot{m}_{s}C_{p,s}dT_{s} + C_{p,s}(T_{s} - T_{s,in})\dot{m}_{a}dW = \dot{m}_{a}\left[C_{p,da}dT_{a} + dW(C_{p,st}^{sat}T_{a} + \Delta h_{evap}) + WC_{p,st}^{sat}dT_{a}\right]$$

Differentiating eq. 117 over height dz and using eq. 104 and eq. 112, the rate of change of desiccant temperature T_s is produced:

$$eq. \ 118 \quad \frac{dT_s}{dz} = \frac{\dot{m}_a}{\dot{m}_s C_{p,s}} \left[\left(-C_{p,s}(T_s - T_{s,in}) + C_{p,st}^{sat} T_a + \Delta h_{evap} \right) \frac{dW}{dz} + C_{p,ma} \frac{dT_a}{dz} \right]$$

System of Equations

Finally, a system of five ordinary differential equations is constituted:

$$eq. 119 \begin{cases} \frac{dT_a}{dz} = \frac{T_s - T_a}{C_{p,ma}} \left[\frac{Lh_G}{\dot{m}_a} + C_{p,st}^{sat} \frac{dW}{dz} \right] \\ \frac{dT_s}{dz} = \frac{\dot{m}_a}{\dot{m}_s C_{p,s}} \left[\left(-C_{p,s} (T_s - T_{s,in}) + C_{p,st}^{sat} T_a + \Delta h_{evap} \right) \frac{dW}{dz} + C_{p,ma} \frac{dT_a}{dz} \right] \\ \frac{dW}{dz} = \frac{LK_G}{\dot{m}_a} \left(W^{sat} - W_{in} \right) \\ \frac{d\dot{m}_s}{dz} = \dot{m}_a \frac{dW}{dz} \\ \frac{dX}{dz} = -\frac{\dot{m}_a}{\dot{m}_s} X \frac{dW}{dz} \end{cases}$$

with the following boundary conditions:

$$eq. 120 \begin{cases} (T_a)_{z=0} = T_{a,in} \\ (T_s)_{z=L} = T_{s,in} \\ (W)_{z=0} = W_{in} \\ (\dot{m}_s)_{z=L} = \dot{m}_{s,in} \\ (X)_{z=L} = X_{in} \end{cases}$$

The five-order Runge - Kutta was used to numerically integrate the above five ordinary differential equations by using a trial step at the midpoint of an interval to cancel out lowerorder error terms. The model was developed using the Mathcad software [45] where all the thermodynamic characteristics of the regenerator are calculated from the point where the liquid desiccant is sprinkled till the end of the packing.

4.3.4 Validation

Before proceeding to the examination of the effect of various operating parameters on the regeneration efficiency of the adiabatic packed column regenerator, the validation of the developed computational model should precede. For this purpose, experimental data from Fumo and Goswami [9, 50] were selected. Table 4-5 provides a comparison between the experimental data with the simulated results for a LiCl regenerator column packed with polypropylene Rauschert Hiflow rings.

Figure 4-16 shows the relative deviations of the predicted values from the corresponding experimental values for all 13 regeneration experiments. In the case of outlet dry bulb temperature, the results of the computational program are almost identical to the experimental ones. The maximum deviation is -6.2% (-3.6 °C) and the mean deviation is -1.8% (-1.0 °C). A similar pattern is observed in the case of outlet humidity ratio. The maximum discrepancy is now -6.8% (-0.003 kg_w/kg_{da}) and the mean deviation is still very small, at -3.2% (-0.002 kg_w/kg_{da}). The simulated outlet solution temperature differ as much as 1.4% (0.8 °C) from the experimental values, the average difference being only 0.2% (0.1 °C). The predictions of the model agree very well with the experimental findings in the case of outlet solution concentration; an average discrepancy of only ±0.5% (-0.0017 kg_{Licl}/kg_s) is seen. A similar observation can be made in the case of the water that is evaporated from the liquid desiccant solution; the maximum deviation is 6.6% (0.0001 kg_w/kg_s), whereas the mean deviation is 2.6% (0.00005 kg_w/kg_s). As it can be observed, the theoretical model shows very good agreement with the experimental results, having an excellent correspondence between most values.

	T _{a,in}	T _{a,out,exp}	T _{a,out,sim}	Deviation	W _{a,in}	W _{a,out,exp}	W _{a,out,sim}	Deviation	T _{s,in}	T _{s,out,exp}	T _{s,out,sim}	Deviation	X _{in}	X _{out,exp}	X _{out,sim}	Deviation	m _{evap,exp}	m _{evap,sim}	Deviation
	°C	°C	°C	(%)	kg/kg	kg/kg	kg/kg	(%)	°C	°C	°C	(%)	kg _{salt/} kg _{sol}	kg _{salt/} kg _{sol}	kg _{salt/} kg _{sol}	(%)	kg/sec	kg/sec	(%)
1	30.4	58.9	60.5	2.8%	0.01830	0.05790	0.05650	-2.4%	65.0	58.6	58.6	0.1%	0.340	0.345	0.345	-0.1%	0.00155	0.00160	3.2%
2	30.1	59.3	57.4	-3.2%	0.01800	0.05320	0.05160	-3.0%	65.1	57.8	57.4	-0.7%	0.341	0.348	0.345	-0.7%	0.00181	0.00190	5.0%
3	29.8	57.5	53.9	-6.2%	0.01770	0.04880	0.04590	-5.9%	65.1	56.6	56.9	0.4%	0.345	0.352	0.349	-0.9%	0.00210	0.00210	0.0%
4	35.1	58.5	58.2	-0.5%	0.01800	0.05510	0.05400	-2.0%	65.1	57.4	57.5	0.1%	0.334	0.341	0.339	-0.7%	0.00191	0.00200	4.7%
5	40.0	58.9	59.0	0.2%	0.01780	0.05480	0.05320	-2.9%	65.0	57.6	57.7	0.2%	0.336	0.342	0.341	-0.4%	0.00191	0.00200	4.7%
6	30.2	57.6	57.0	-1.0%	0.01430	0.05130	0.05090	-0.8%	65.2	57.2	57.0	-0.4%	0.340	0.347	0.345	-0.7%	0.00197	0.00210	6.6%
7	29.4	58.5	56.4	-3.6%	0.02100	0.05410	0.05351	-1.1%	65.5	58.3	58.3	0.0%	0.336	0.342	0.340	-0.7%	0.00170	0.00181	6.5%
8	30.3	57.6	57.1	-0.9%	0.01820	0.05070	0.04990	-1.6%	65.4	57.0	56.4	-1.0%	0.344	0.349	0.349	0.0%	0.00171	0.00180	5.3%
9	29.9	59.0	57.8	-2.1%	0.01800	0.05560	0.05230	-5.9%	65.2	57.9	58.7	1.4%	0.343	0.349	0.347	-0.6%	0.00195	0.00190	-2.6%
10	30.0	55.8	54.7	-2.0%	0.01870	0.04470	0.04168	-6.8%	60.3	54.2	54.6	0.7%	0.344	0.348	0.347	-0.2%	0.00136	0.00129	-5.1%
11	29.7	62.6	60.9	-2.7%	0.01840	0.06660	0.06260	-6.0%	70.0	60.0	60.3	0.6%	0.345	0.353	0.351	-0.7%	0.00245	0.00240	-2.0%
12	29.7	57.6	56.0	-2.7%	0.01770	0.05420	0.05374	-0.8%	64.8	56.8	57.1	0.5%	0.328	0.334	0.332	-0.5%	0.00189	0.00200	5.8%
13	30.3	57.9	57.4	-0.9%	0.01820	0.05010	0.04910	-2.0%	65.0	57.5	57.9	0.6%	0.349	0.354	0.353	-0.3%	0.00167	0.00170	1.8%

Table 4-5 Comparison between the experimental data from literature and the simulated results for an adiabatic regenerator



Figure 4-16 Relative deviation between the simulated results and the experimental data from literature for an adiabatic regenerator.



Figure 4-17 Deviations between the experimental data from literature and the simulated results. Top: Relative deviation of air temperature values versus relative deviation of humidity ratio at the outlet of the regenerator. Bottom: Relative deviation of desiccant temperature values versus relative deviation of solution concentration at the outlet of the regenerator.

Figure 4-17 (top graph) plots the relative temperature error versus the relative humidity ratio error for all points. In general, the model has a slight tendency to underpredict the values, since most of the points are positioned in the third quadrant of the coordinate system. It also shows that the relative errors are well below 8%, with the maximum predicted error for humidity ratio being -6.8% (-0.004 kg_w/kg_{da}) and for the outlet temperature -6.2% (-3.56 °C). Taking though into consideration that the uncertainty of the experimental measurements is approximately 0.001 kg_w/kg_{da} and 0.4°C, the relative errors of the theoretical modeling may be even less.

The bottom graph plots the relative temperature error versus the relative concentration error for all points. It shows that the maximum predicted error for desiccant temperature is 1.4% (0.78 °C) and for the outlet concentration is -0.9% (-0.003 kg_{LiCl}/kg_s). Considering the fact that the experimental errors are 0.4°C and 0.004 kg_{LiCl}/kg_s respectively, the predicted values are quite close to the experimental ones. The modeled outlet solution concentration lies in the domain of the uncertainty of the measurements.



Figure 4-18 Psychrometric chart of inlet and outlet air for all 13 regeneration experiments.



Figure 4-19 Psychrometric chart of inlet and outlet LiCl solution for all 13 regeneration experiments.

The inlet and outlet conditions of both air and desiccant solution have been designed in a psychrometric chart, as seen in Figure 4-18 and Figure 4-19. The vapor pressure expressions used to design these psychrometric charts are taken from Fumo and Goswami [9, 50]. The proximity between the outlet values of the theoretical model and the outlet values of the experiments can be clearly seen in both charts. Figure 4-18 also provides valuable information regarding the initial and final states of the air passing through a regenerator; ambient air enters the regenerator and exits at a substantially higher humidity ratio and dry bulb temperature, as a result of heat and mass transfer with the desiccant solution.

Figure 4-19 also provides a simple explanation of the liquid desiccant regeneration procedure; the solution enters the regenerator at lower concentration and higher temperature, and exits at higher concentration and lower temperature, as a result of heat and mass transfer with the ambient air.

4.3.5 Parametric analysis

4.3.5.1 **Overview**

Following the validation of the numerical model, an extensive theoretical investigation was conducted to examine the effect of various operating parameters on the regeneration efficiency of the adiabatic packed column regenerator. The investigation included three commonly used liquid desiccant solutions; LiCl-H₂O, LiBr-H₂O and CaCl₂-H₂O. All the considered cases are described in the following table.

The comparison of the three different liquid desiccant solutions, in terms of their regeneration potential, is implemented also through the evaluation of the loads. The developed computational program can predict the distribution of the loads along the regenerator's height. The study of loads was implemented with the starting values: $T_{a,in}=36$ °C, $W_{a,in}=0.01486$ kg_w/kg_{da}, $\dot{m}_{a}=0.0653$ kg/sec, $T_{s,in}=65$ °C, $\dot{m}_{s}=0.3265$ kg/sec and X= 30%.

	Inlet values for the parametric analysis										
Cases	m _a	ms	T _{a,in}	W _{a,in}	Х						
Cases	kg/s	kg/s	°C	kg/kg	°C	kg _{salt} /kg _{solution}					
			30								
			32								
Variation of			34								
inlet air	0.0653	0.3265	36	0.01486	65	0.3					
temperature			38								
			40								
			42								
				0.01086							
Variation of	0.0050	0.0005		0.01286	. .						
inlet humidity	0.0653	0.3265	30	0.01486	65	0.3					
ratio				0.01686							
	0.02265			0.01886							
	0.03205										
Variation of	0.04858										
inlet air mass	0.0033	0 2265	30	0.01/186	65	0.3					
flow rate	0.00102	0.5205	50	0.01480	05	0.5					
nowrate	0.03733										
	0.1306										
					50						
					55						
Variation of					60						
inlet solution	0.0653	0.3265	30	0.01486	65	0.3					
temperature					70						
					75						
					80						
						0.28					
Variation of	0.0052	0.2265	20	0.01.400	65	0.3					
Inlet solution	0.0653	0.3265	30	0.01486	65	0.32					
concentration						0.34					
		0.653				0.50					
		0.055									
Variation of		0.43535									
inlet solution	0.0653	0.3203	30	0.01486	65	0.3					
mass flow rate	0.0055	0.2012	50	0.01400	05	0.5					
mass now rate		0.21707									
		0.16007									
		0.10325									

Table 4-6 Parametric analysis in adiabatic regeneration

4.3.5.2 Sensitivity analysis

A sensitivity analysis was performed on the inlet and outlet variables of both air and solution, such as temperature, humidity ratio, concentration and mass flow rate. This step was necessary in order to predict the effect of the independent variables (inputs $T_{a,in}$, W_{in} , m_a , $T_{s,in}$, X_{in} , $m_{s,in}$, K_G , Le) on the dependent variables (outputs $T_{a,out}$, W_{out} , RH_{out} , $T_{s,out}$, X_{out} , $m_{s,out}$, n, m_w). A multiple regression analysis has been also conducted, with the view to investigate the relationships between the variables and to ascertain the causal effect of one variable upon another. In other words, the regression was employed to estimate the quantitative effect of the independent variables upon the dependent variable that they influence.

For example, Table 4-7 shows the statistic results, after employing a regression analysis on the dependent variable $T_{a,out}$ for LiCl solution. The regression coefficients are the coefficients of the linear relationship between the $T_{a,out}$ and the independent variables.

		T _{a,out}					
Regression Stati	stics		Regression Coefficients	Standardized Regression Coefficients	Standard Error	t-Stat	P-value
Multiple R	0.9632	Intersept	-11.4816	-	5.9645	-1.9250	0.0619
R-Square	0.9277	T _{a,in}	0.6282	0.5923	0.0473	13.2708	1.2E-15
Adjusted R square	0.9277	m _a	-112.4466	-0.5587	8.9362	-12.5833	6.2E-15
Standard Error 0.8462		W _{in}	24.1550	0.0080	133.7990	0.1805	0.8577
		T _{s,in}	0.2669	0.3689	0.0320	8.3445	5.0E-10
		m _{s,in}	1.9222	0.0427	1.9905	0.9657	0.3405
		X _{in}	6.8225	0.0271	11.1830	0.6101	0.5455
		K _G	0.5814	0.2209	0.1172	4.9619	1.6E-05
		Le	12.2289	0.1554	3.4876	3.5064	0.0012

Table 4-7 Sensitivity analysis of $T_{a,out}$ at the outlet of the regenerator using LiCl, treated as dependent qualitative (dummy) variable

It is difficult to say which of the independent variables is most important in determining the value of the dependent variable $T_{a,out}$, since the value of the regression coefficients depends on the units of measurement of the independent variables. So, the regression coefficients have been standardized and become "metric-free". Thus, the larger standardized regression coefficient of the independent variable, the greater influence on the dependent variable has. Figure 4-20 shows the standardized regression coefficients, for all desiccant solutions and for all dependent variables. Table 4-8 gives the linear regression equations for calculating the output thermodynamic properties of air and solution.



Figure 4-20 Coefficients of independent variables for LiCl, LiBr and CaCl₂

The quality of the linear fit, that is how well the regression line approximates the real data points, is represented by the coefficient of determination, R^2 . An $R^2 = 1$ indicates that the regression line perfectly fits the data; however this is not guaranteed for all cases. So, to increase the precision of the R^2 , the adjusted- R^2 is used instead, since it reflects both the number of independent variables in the model and the sample size. Table 4-9 presents the R^2 and adjusted- R^2 for all desiccant solutions, for all dependent variables. Since they range between 0.83-1.00 and 0.79-1.00, it seems that the true relationship is close to the estimated relationship by the regression line.

The results of the sensitivity analysis show that the output conditions are found to be most sensitive to the solution inlet concentration, inlet mass flow rate and temperature as well as air inlet mass flow rate. The following paragraphs are dedicated to the thorough examination of the effect of each parameter separately.

	Y=a + b*T _{a,in} + c*m _a + d*W _{in} + e*T _{s,in} + f*m _{s,in} + g*X _{in} + h*K _G + i*Le											
LiCl												
Υ	а	b	С	d	e	f	g	h	i			
T _{a,out}	-11.4816	0.5923	-0.5587	0.0080	0.3689	0.0427	0.0271	0.2209	0.1554			
W _{out}	-0.0093	-0.0361	-0.4572	0.0925	0.7862	0.0821	-0.2954	0.1614	0.0021			
RH _{out}	0.4797	-0.0274	-0.6912	0.1420	-0.5248	0.1261	0.1882	0.2735	0.0178			
T _{s,out}	2.9470	0.0455	-0.0326	0.0208	0.9452	0.2516	0.1075	-0.0856	-0.0150			
m _{s,out}	0.0019	7.63E-05	-5.45E-04	2.98E-04	-0.0052	0.9995	0.0018	-9.12E-04	-2.97E-05			
X _{out}	-0.0011	-0.0018	-0.0028	-0.0029	0.0338	-0.0221	0.9984	0.0094	0.0004			
n	0.3581	-0.0118	-0.8492	-0.0037	-0.1544	0.1566	0.0239	0.3074	0.0063			
m _w	-0.0017	-0.01239	0.106607	-0.06169	0.892842	0.09192	-0.3181	0.1607	-0.01027			
LiBr												
Y	а	b	С	d	е	f	g	h	i			
T _{a,out}	-6.8950	0.6381	-0.5468	0.0074	0.2960	0.0675	0.0161	0.1981	0.1521			
Wout	-0.0699	-0.0253	-0.5200	0.0582	0.7566	0.1380	-0.1472	0.2101	0.0276			
RH _{out}	0.6169	-0.0258	-0.7862	0.0870	-0.3961	0.2093	0.0312	0.2479	-0.0028			
T _{s,out}	15.1132	0.0535	-0.0585	0.0217	0.8370	0.4365	0.0837	-0.1807	-0.0374			
m _{s,out}	0.0067	0.0001	-0.0012	0.0004	-0.0085	0.9999	0.0015	-0.0022	-0.0002			
X _{out}	-0.0070	-0.0027	-0.0041	-0.0021	0.0541	-0.0371	0.9964	0.0184	0.0016			
n	0.4792	-0.0127	-0.8142	-0.0027	-0.2705	0.2189	0.0144	0.2561	-0.0050			
m _w	-0.0067	0.0055	0.1161	-0.0352	0.9094	0.1641	-0.1550	0.2250	0.0183			
				Ci	aCl ₂							
Y	а	b	С	d	е	f	g	h	i			
T _{a,out}	-4.5935	0.6689	-0.5303	0.0070	0.2444	0.0818	0.0079	0.1884	0.1529			
Wout	-0.0973	-0.0282	-0.5463	0.0458	0.7375	0.1794	-0.0961	0.1915	0.0170			
RH _{out}	0.6227	-0.0197	-0.7841	0.0661	-0.3777	0.2585	-0.0020	0.2364	0.0000			
T _{s,out}	21.0406	0.0634	-0.0693	0.0221	0.7357	0.5553	0.0590	-0.2212	-0.0362			
m _{s,out}	0.009377	-6.5E-05	-0.00125	0.000373	-0.0106	0.999927	0.001156	-0.00252	-0.00017			
X _{out}	-0.01014	-0.00325	-0.00638	-0.00206	0.066849	-0.04402	0.9949	0.02175	0.002322			
n	0.553688	-0.00905	-0.77134	-0.00277	-0.36598	0.256883	0.003226	0.22578	-0.00423			
m _w	-0.00923	0.004224	0.10961	-0.02733	0.913451	0.220356	-0.09514	0.20705	0.005872			

Table 4-8 Linear regression equations for calculating the output thermodynamic properties of air andsolution for adiabatic regeneration

	L	iCl	Li	iBr	CaCl ₂		
	R Square	Adjusted R Square	R Square	Adjusted R Square	R Square	Adjusted R Square	
T _{a,out}	0.93	0.91	0.92	0.90	0.91	0.89	
W _{out}	0.93	0.92	0.91	0.89	0.90	0.88	
RH_{out}	0.89	0.87	0.87	0.84	0.86	0.84	
T _{s,out}	0.98	0.97	0.94	0.92	0.91	0.89	
m _{s,out}	1.00	1.00	1.00	1.00	1.00	1.00	
X _{out}	1.00	1.00	1.00	1.00	1.00	1.00	
n	0.84	0.81	0.83	0.79	0.83	0.79	
m _w	0.95	0.93	0.94	0.93	0.95	0.94	

Table 4-9 R-Square and adjusted R-Square for all desiccant solutions for adiabatic regeneration

4.3.5.3 Air flow rate

The variation of the regeneration rate as a function of the air flow rate is shown in Figure 4-21. Increasing the air flow rate by 300% increases the regeneration rate by about 27% for all three desiccant solutions, due to the increase in the mass transfer coefficient between the air and desiccant.



Figure 4-21 Variation of the regeneration rate for different air flow rates

The increase in the regeneration rate due to the increase in air flow rate is also testified in Figure 4-22. Under higher air flow rates, there is less water content in the solution outlet, which means that more water has been evaporated, so the solution has been more effectively regenerated.



Figure 4-22 Profile of the moisture content in the solution for different inlet air mass flow rates across the packing's height.

The efficiency of the CaCl₂ regenerator is testified again, since a far larger amount of moisture seems to evaporate from CaCl₂ solution than from LiCl, regardless of the air mass flow rate. In fact, more water is evaporated from CaCl₂ solution with low flow (m_a=0.03265 kg/s) than that of LiCl with high flow (m_a=0.13060 kg/s). Consequently, the choice between CaCl₂ and LiCl may not count on the air flow rate, rather than other parameters. The choice however between CaCl₂ and LiBr would appear more difficult, as CaCl₂ solution with low flow (m_a=0.03265 kg/s) produces the approximately same result with LiBr with high flow (m_a=0.13060 kg/s). Other parameters as well as cost may provide the answer.

4.3.5.4 **Desiccant inlet temperature**

Air outlet humidity ratio, regeneration rate and thermal efficiency increase with increasing desiccant inlet temperature. During regeneration, the partial water vapor pressure in the liquid desiccant solution is always higher than the water vapor pressure of the air, so that water is evaporated from the desiccant to the air. However, the increase in the solution inlet temperature results in a further increase in the partial water vapor pressure in the liquid desiccant solution. So, the difference between the partial water vapor pressure in the liquid desiccant solution and the water vapor pressure of the air is increased, enhancing thus the potential for mass transfer. This results in an increase of the air outlet humidity ratio (Figure 4-23) and consequently, in the increase of regeneration rate and thermal efficiency (Figure 4-24). In the present analysis, the increase in desiccant inlet temperature by 60% increases the regeneration rate, by 523% in the case of LiCl, 367% in LiBr and by 331% in CaCl₂.



Figure 4-23 Profile of the humidity ratio for different desiccant inlet temperatures across the packing's height.



Figure 4-24 Variation of the regeneration rate and the regeneration thermal efficiency for different desiccant inlet temperatures.

All three figures also confirm that the CaCl₂ has the best regenerator capacity among the other solutions. The reasoning is as follows: under the same temperature and concentration, the higher water partial vapour pressure, the greater ability a liquid desiccant has for water evaporation. Therefore, since CaCl₂ has the highest water vapour pressure among the examined desiccants, it presents the highest regeneration capacity, whereas the lowest regeneration capacity should be attributed to LiCl. Indeed, under specific initial conditions, its efficiency is 71% and its regeneration rate is 3.48 gr_w/s. At these conditions, the regenerator using LiBr has an efficiency of 64% and a regeneration rate of 2.63 gr_w/s, whereas LiCl is the least favorable, with an efficiency of 47% and a regeneration rate of 1.39 gr_w/s only.

The higher regeneration rate a solution experiences under higher inlet solution temperature is also obvious in the profile of the moisture content of the solution across the regenerator's height (Figure 4-25). Since the difference between the inlet and the outlet moisture content represents the mass of the water evaporated from the solution, higher inlet solution temperatures result in a steeper slope of the moisture content profile line.



Figure 4-25 Profile of the moisture content in the solution for different desiccant inlet temperatures across the packing's height.

From the results shown in the above figures, it is suggested that higher inlet temperature of liquid desiccant should be preferred to get both higher energy utilization efficiency and more regeneration capacity.
4.3.5.5 Desiccant inlet concentration

Figure 4-26 shows the variation of the solution inlet concentration with respect to the humidity ratio distribution across the regenerator. CaCl₂ solution appears to produce more humid air than LiBr and LiCl, regardless of the solution inlet concentration. In fact, CaCl₂ solution of X=0.36 seems to be better than LiBr of X=0.28. The same goes for LiBr and LiCl. Under these circumstances, the desiccant inlet concentration seems to be far less important than the choice of the desiccant solution itself.



Figure 4-26 Profile of the humidity ratio for different inlet desiccant concentrations across the packing's height.

The regeneration efficiency decreases with the increase of inlet desiccant concentration (Figure 4-27). This may be explained from the fact that vapor pressure of each desiccant is a function of the concentration. Therefore, as the solution concentration is increased, its vapor pressure is decreased. The potential for mass transfer is thus decreased, leaving a negative effect on the regeneration rate and efficiency. The model has predicted that a 29% rise in the solution inlet concentration results in 37% reduction in the regeneration efficiency of LiCl, 15% of LiBr and 10% of CaCl₂.



Figure 4-27 Variation of the regeneration thermal efficiency for different desiccant inlet concentrations.

4.3.5.6 **Desiccant flow rate**

The flow rate of the liquid desiccant is a controllable parameter for regenerator operation. Here the flow rate of the liquid desiccant varied from 0.16325 kg/s to 0.6530 kg/s.

In the adiabatic regenerator, both regeneration thermal efficiency and regeneration rate improve significantly with the increase of desiccant flow rate. Since the flow rate of liquid desiccant determines the available heat for the regeneration, when the desiccant flow rate is kept high, the available heat is plenty and the solution does not have the time to cool down as it flows down through the bed. That is the reason why in Figure 4-28 higher solution flow rates result in more horizontal lines.



Figure 4-28 Profile of the solution temperature for different inlet desiccant mass flow rates across the packing's height.



Figure 4-29 Variation of the regeneration rate for different desiccant flow rates.

The higher desiccant mass flow rate, the higher temperatures will be maintained throughout the regeneration process and the higher water vapour pressure of the desiccant solution will be finally obtained. This gives rise to a higher mass transfer potential for the regeneration processes, resulting in an increase in regeneration rate (Figure 4-29). In the present study, this phenomenon can be observed at desiccant flow rates of over 0.3 kg/sec. It has also been observed that a 75% increase in solution flow rate results in 18% increase in regeneration rate for LiCl, in 26% increase for LiBr and in 31% increase for CaCl₂.

However, when the desiccant flow rate is low, the available heat for the regeneration process is limited and the solution temperature decreases quickly after entering the regenerator, due to the heat and mass transfer with the air. The effect of regeneration is thus compromised. Under very small solution mass flow rates, the solution may be cooled dramatically, so that the water vapour pressure of desiccant solution may become lower than the vapor pressure of moist air. In such a case, the solution would actually be diluted.

4.3.5.7 Liquid desiccant solution

CaCl₂ has already shown clear regeneration performance precedence among the three desiccant solutions examined. The distribution though of the solution and air temperatures, as these are formed across the height of the packing, would give additional information.

Figure 4-30 shows that $CaCl_2$ experiences the biggest difference between inlet and outlet solution temperature (65-54.2°C), comparing to the other two materials. This released heat however, does not seem to be all absorbed by the upcoming air, since the air interacting with $CaCl_2$ is not heated that much (36-43.7°C). Most of this heat should therefore be used for the phase change of the water molecules, contained in the solution, from liquid to vapor.

This is clearly seen in Figure 4-31, where the $CaCl_2$ convection load is the least of all solutions at the exit and at the same time, the $CaCl_2$ evaporation load is the highest.



Figure 4-30 Profile of the solution and air temperatures across the regenerator for the three desiccants examined



Figure 4-31 Loads profile across the mass transfer area of the regenerator a. convection load b. evaporation load

4 ADIABATIC MODEL

Correspondingly, LiCl has a smaller difference between inlet and outlet solution temperature (65-60.2°C), whereas the air interacting with LiCl is significantly heated, (36-45.6°C) compared to the other air streams. Therefore, the heat released from LiCl solution is mostly used for heating the air, rather than evaporating the moisture contained in the solution. That is reason why the convection load of LiCl is the highest of all three materials, but its evaporation load is the least, as seen in Figure 4-31.

It must be also noted here that in general, the difference between the desiccant outlet and inlet temperature is much higher than that of the air outlet and inlet temperature. This is due to the fact that the desiccant releases heat for both the phase change of water molecules (evaporation) and for balancing its temperature with the upcoming air (convection), so its temperature is substantially decreased. However, since the evaporation load causes only phase change and does not result in any temperature difference, the temperature of the air is heated only by the convection load.

In this study, the evaporation load ranges from 6 to 16 kW, whereas the convection load ranges from 0.8 to 0.9 kW, depending on the liquid desiccant material chosen.

As the desiccant solution flows down the regenerator, the amount of water evaporated increases, resulting in the increase of the evaporation load. The large regeneration capability of CaCl₂ is also indicated by the significantly higher evaporation load, in comparison to LiBr and LiCl.

4.4 CONCLUSIONS

This chapter presents the study of the performance of an adiabatic liquid desiccant dehumidifier and regenerator. The "adiabatic" term is used to show that during dehumidification and regeneration there is no interaction with external energy sources (i.e. cooling in dehumidification and heating in regeneration). The dehumidifier and regenerator device therefore employ structured packing material and not any cooling or heating coils. The performance was studied in terms of the heat and mass transfer processes that happen between the air and the desiccant when they counter flow through structured packing in the dehumidifier and the regenerator.

A heat and mass transfer model of an adiabatic packed column has been developed to predict the performance of the liquid desiccant dehumidifier under various operating conditions. Good agreement was found between experimental tests from the literature and the developed theoretical model, with the maximum deviation being $\pm 2.9\%$ in air outlet temperature, $\pm 15.9\%$ in air outlet humidity ratio and $\pm 2.8\%$ in solution outlet temperature.

Following the model validation, the rate and the efficiency of the dehumidification process were assessed under the effects of variables, such as air temperature and humidity, desiccant temperature and humidity and air and desiccant flow rates. Design variables found to have the greatest impact on the dehumidification performance are the air mass flow rate, air humidity ratio, desiccant temperature and desiccant concentration. It was found that the dehumidification mass rate increased with increasing air inlet humidity ratio, desiccant inlet concentration, air and desiccant flow rate; decreased with desiccant inlet temperature; and was almost unaffected with air inlet temperature. The air inlet flow rate has been found to have a critical value for the performance of the dehumidification process at 0.05 kg/sec. The absorber efficiency increased with increasing desiccant flow rate and inlet temperature and humidity ratio and desiccant inlet concentration. High system efficiency could be achieved under humid conditions (>0.013 kgw/kgda), making it ideal for buildings in hot and humid regions, where the risk of condensation is very high due to the low surface temperature in the radiant cooling panels.

The three liquid desiccant solutions, namely LiCl, LiBr and CaCl₂ were then evaluated against each other. The analysis showed that liquid desiccant systems using LiCl seem to have better dehumidification performance than those using LiBr and CaCl₂, under the same operational conditions. Although under high humidity ratios, the three desiccant solutions encounter

similar dehumidification efficiency, the performance of LiCl is much more stable and predictable than that of CaCl₂. The results show that high absorber efficiency and system efficiency could be achieved under humid conditions, low air mass flow rates and LiCl as the desiccant solution.

The performance of the adiabatic regenerator was enabled through the development of a heat and mass transfer numerical model. Good agreement was found between the theoretical model and experimental tests from previous studies, with the deviation range being $\pm 6.2\%$ in air outlet temperature, $\pm 8.1\%$ in air outlet humidity ratio and $\pm 1.4\%$ in solution outlet temperature.

Important design variables for the regenerator were then defined. The effects of air and desiccant flow rates, air humidity ratio, desiccant temperature and concentration have been reported on the regeneration rate and regeneration effectiveness. A detailed sensitivity analysis has been implemented to indicate which input variables mostly affect the output conditions. According to the sensitivity analysis, the output conditions of the regenerator strongly depend on the inlet solution concentration change, but depend less on the inlet humidity ratio. For higher energy utilization efficiency and greater regeneration capacity, higher liquid desiccant inlet temperature is suggested. It was found that higher desiccant mass flow rate results in higher mass transfer potential. This happens because when the desiccant mass flow rate is high, the desiccant temperature does not have enough time to decrease as it flows down through the bed. Since the temperature of the desiccant remains greater than the air, the desiccant surface vapor pressure remains higher than the air vapor pressure. This difference provides the mass transfer force. Also, high system efficiency could be achieved under high air mass flow rates; however the air mass flow rate is a critical parameter only when choosing between CaCl₂ and LiBr. A low inlet desiccant concentration would be also favorable for efficient regeneration, though the concentration seems to be less important than the choice of the desiccant solution itself. It has been observed that the heat released from CaCl₂ solution is mostly used for the evaporation of moisture, whereas the heat released from LiCl is mostly used for heating the air. The high water vapor pressure has thus given to CaCl₂ a considerable advantage over LiBr and LiCl for all the investigated parameters.

The three most commonly used liquid desiccant solutions, namely LiCl, LiBr and CaCl₂ were then evaluated against each other. The results show that high regeneration efficiency could be achieved under high desiccant mass flow rates, high air mass flow rate, high desiccant inlet temperature, low desiccant inlet concentration and CaCl₂ as the desiccant solution.

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4.5 NOMENCLATURE

Cp	specific heat capacity at constant pressure (J/kgK)
d	diameter (m)
D	diffusion coefficient (m ² /s)
h	enthalpy (kJ/kg _{da})
h _G	heat transfer coefficient (W/m ² K)
g	gravitational acceleration (m ² /s)
k	thermal conductivity (kW/mK)
K _G	mass transfer coefficient (kmol/sm ² mole fraction)
L	length of packing (m)
Le	Lewis number
'n	mass flow rate (kg/s)
\dot{m}_{w}	regeneration rate (kg _w /s)
М	molecular weight (kg/kmol)
n	regeneration or absorber efficiency
р	pressure (Pa)
Q	heat (kJ)
Ż	load (kW)
Re	Reynolds number
Sc	Schmidt number
т	temperature (°C)
V	superficial velocity (volumetric flow rate per column cross section) (m/s)
W	air humidity ratio (kg _w /kg _{da})
х	salt concentration in solution (kg _{salt} /kg _s)
Y	moisture content in solution (kg _w /kg _s)
z	column height (m)
Δh_{evap}	latent heat of vaporization for water (kJ/kg)
Greek sym	ibols
α	specific interfacial surface area of packing (m ² /m ³)
3	void fraction (bed porosity) (m ³ /m ³)
λ	liquid hold up in the packed bed (m ³ /m ³)
μ	viscosity (kg/ms)
ρ	density (kg/m³)

Subscripts

α	air
atm	atmospheric
conv	convection
da	dry air
deh	dehumidification
eq	equivalent
exp	experimental
evap	evaporation
in	inlet
L	latent
ma	moist air
out	outlet
pack	packing
S	sensible
S	solution
sim	simulated
st	steam
w	water
Superscripts	

sat saturated

4.6 **REFERENCES**

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5 NON-ADIABATIC TUBULAR MODEL

5.1 INTRODUCTION

Liquid desiccant systems are emerging as promising alternatives to achieve humidity control in a variety of applications with high latent loads and low humidity requirements. Their advantage lies on their ability to handle the latent load without super-cooling and then reheating the air, as happens in a conventional compression-type air conditioning system.

This chapter presents the results from a study of the performance of a non-adiabatic liquid desiccant dehumidifier and regenerator. The non-adiabatic issue is translated into internally cooled dehumidifier and internally heated regenerator via a tubular heat exchanger integrated in each device.

In the non-adiabatic liquid desiccant dehumidifier, the desiccant solution is sprayed from the top and flows down by gravity. At the same time, ambient air is blown from the bottom, counter-flowing with the desiccant solution. The desiccant is in direct contact with the air, allowing for heat and mass transfer. The cold water, flowing inside the tubes of the dehumidifier, removes the heat released during dehumidification.

In the non-adiabatic liquid desiccant regenerator, the desiccant solution is sprayed from the top and flows down by gravity. At the same time, ambient air is blown from the bottom, counter-flowing with the desiccant solution. The desiccant is in direct contact with the air, allowing for heat and mass transfer. The hot water, flowing inside the tubes of the regenerator, provides the necessary heat for regeneration.

The heat and mass transfer theoretical models have been developed, based on the Runge-Kutta fixed step method, to predict the performance of the dehumidifier and regenerator under various operating conditions. Experimental data from previous literature have been used to validate both models. Excellent agreement has been found between experimental tests and the theoretical model of the regenerator, with the deviation not exceeding ±6.1%. Following the validation of the mathematical model, the dominating effects on the desorption process have been discussed in detail. The three most commonly used liquid desiccant solutions (LiCl, LiBr, CaCl₂) and two different flows (DDU: water downward – desiccant downward – air upward, UDU: water upward – desiccant downward – air upward) have been also evaluated against each other. Considering the flow analysis, the type of flow does not affect the regeneration capacity as much as the type of the desiccant solution. It has been concluded that high regeneration rate can be achieved under DDU flow (water downward – desiccant downward – air upward), low solution concentration, high air inlet temperature, high solution inlet temperature, low air inlet humidity ratio and $CaCl_2$ as the desiccant solution.

5.2 DEHUMIDIFIER

5.2.1 Overview

Inside the counter flow dehumidifier, the strong desiccant solution is sprayed at the top of the device. The ambient humid air enters the dehumidifier at the bottom and moisture is transferred from the air to the desiccant. The absorption process is cooled by water flowing inside tubes. The end of the process finds the air dehumidified and cooled, whereas the desiccant solution diluted and stored in the weak tank.

An indirect evaporative cooler is required to supply the necessary cooling water during absorption. To achieve this, the evaporative cooler may use the exhaust air from the building and mains water. Should this configuration is employed, the lower theoretical temperature limit for the cooling water produced by the evaporative cooler is the wet bulb temperature of the exhaust air from the building.

5.2.2 Thermodynamic parameters

The thermodynamic parameters used for the simulation of the non-adiabatic tubular dehumidifier refer to the Chapter 3 of this Thesis.

5.2.2.1 Vapor Pressure

The expression of the partial vapor pressure of the water to the LiCl-H₂O solution is taken from the study of Cisternas and Lam [1, 2].

5.2.2.2 Heat of absorption

The heat of absorption, that is the energy added to the liquid desiccant solution during the condensation of 1 kg of water vapor, is expressed by the following equation:

$$eq. 121 \quad \Delta h_{abs} = h_{st}(p,T_s) - h_{W,s}(T_s,X)$$

5.2.2.3 Pressure drop

eq. 122
$$\Delta p_{drybed} = 0.125 \left(\frac{c_1}{\text{Re}_a} + \frac{c_2}{\text{Re}_a^{0.5}} + c_3 \right) \frac{\rho_a V_a^2 a}{\varepsilon^{4.65}}$$

5.2.2.4 Heat & Mass Transfer Coefficients

The heat and mass transfer coefficients used for the dehumidifier are based on the correlations developed for tubular heat exchangers, as studied by Scalabrin [3, 4], Finlay and Harris [5] and Jain et al [6]. As such, the water-side heat transfer coefficient is expressed as:

$$eq. 123 \quad a_{W} = \frac{Nu_{W}k_{W}}{D_{in}}$$

The Nusselt and Reynolds numbers for the water are given by the following expressions:

$$eq. 124 \quad Nu_{W} = 0.36 \operatorname{Re}_{W}^{0.55} \operatorname{Pr}_{W}^{1/3}$$

$$eq. 125 \quad \text{Re}_{W} = \frac{\dot{m}_{W} D_{in}}{A_{in} \mu_{W}}$$

For water flow inside tubes, the Prandtl number Pr_w is assumed as 7.03. The heat transfer coefficient from the solution to the tube surface is expressed as:

$$eq. \ 126 \quad a_{S} = \frac{Nu_{S}k_{S}}{S}$$

The film thickness S is calculated as in Jain et al [6]. The Nusselt and Reynolds numbers for the solution are given by the following expressions:

eq. 127
$$Nu_s = 0.67 \operatorname{Re}_s^{0.11} \left(\frac{C_p \mu_s S}{k_s H} \right)^{1/3}$$

eq. 128
$$\operatorname{Re}_{S} = \frac{\dot{m}_{S} D_{OUI}}{A_{OUI} \mu_{S}}$$

The heat and mass transfer coefficients between the solution interface and the air stream are expressed as:

eq. 129
$$K_a = \frac{\dot{m}_a}{A_{out}} \frac{0.023 \text{Re}_a^{-0.17}}{Sc_a^{2/3}}$$

$$a_{a} = C_{pm} K_a L e^{2/3}$$

The Reynolds and Schmidt numbers are given by the following expressions:

eq. 131
$$\operatorname{Re}_{a} = \frac{\dot{m}_{a}D_{i}}{A_{out}\mu_{a}}, Sc_{a} = \frac{\mu_{a}}{\rho_{a}\delta}$$

The diffusion coefficient δ of the water vapor to the air is calculated by the expression of Bolz and Tuve [7]. The overall heat transfer coefficient from the solution to the water is given by the following expression:

eq. 132
$$U = \left[\frac{D_{out}}{D_{in}}\frac{1}{a_w} + \ln\left(\frac{D_{out}}{D_{in}}\right)\frac{D_{out}}{2k_{pipes}} + \frac{1}{a_s}\right]^{-1}$$

From all the above relations, the model can evaluate the thermodynamic parameters in every step along the height of the dehumidifier, under any given input conditions. Using the basic mass and energy balance equations, the output conditions are then yielded.

5.2.2.5 **Performance indices**

The mass transfer performance of the dehumidifier is evaluated in terms of the dehumidification mass rate (kg_w/h), which is calculated by the following expression:

$$eq. 133 \quad \dot{m}_{deh} = \dot{m}_a (W_{in} - W_{out})$$

The absorber efficiency is the ratio of the actual humidity ratio variance of the air passing through the dehumidifier to its variance under ideal conditions, as shown below:

eq. 134
$$n = \frac{W_{in} - W_{out}}{W_{in} - W^{\text{sat}}} 100\%$$



Figure 5-1 Tube bank arrangement [8]

5.2.3 Mathematical model development

5.2.3.1 Assumptions

The geometry of the dehumidifier is taken from Scalabrin [3, 4]. Accordingly, it consists of a horizontal bank of staggered tubes of 11 rows with 33 tubes in each row, with tube length 2m, tube inner diameter 0.0016 m, tube outer diameter 0.02 m, pitch 0.03 m and device length 1.015 m. The temperature of the inlet cooling water to the dehumidifier and the inlet sprayed water to the evaporative cooler is 18°C, both with a flow rate of 4.15 kg/sec. The solution inlet temperature to the dehumidifier is assumed to be equal to the ambient air temperature. The concentration and flow rate of the strong solution is 40% w.w. and 0.166 kg/sec, respectively.







5.2.3.2 Mathematical model

Figure 5-3 Heat and mass transfer during dehumidification in a non-adiabatic tubular dehumidifier

The numerical model was developed as an extension of Papaefthimiou et al [9, 10] work. The advantage of this model is its ability to easily adapt under different operational conditions, different flow configurations and different liquid desiccant solutions. Based on the energy and mass conservation laws and the arithmetic method of Runge-Kutta, this numerical model describes the coupled heat and mass transfer processes taking place inside a non-adiabatic tubular dehumidifier.

In order to simplify the complexity of the configuration, the assumptions used in the calculations are:

- One-dimensional analysis: heat and mass transfer occur only in the flow direction,
- Temperature of the liquid desiccant equal to the temperature of the coils,
- Uniform flooding and heat transfer area equal to the mass transfer area,
- Vapor pressure equilibrium between the vapor and the liquid at the interface,
- Negligible heat transfer from liquid to vapor, negligible heat transfer because of radiation, viscous dissipation, pressure gradients, concentration gradients or gravitational effects.

The dehumidifier area has been divided into 15 equal segments and the mass and energy balances are solved for each segment, from the top of the device (z=0) until the bottom (z=A); thus, gradients of air temperature and humidity ratio, water temperature and flow

rate and desiccant temperature, flow rate and concentration, are calculated across the height of the device. In order to begin calculations, the following inlet parameters should be assumed:

- Temperature of air, desiccant and water
- Flow rate of air, desiccant and water
- Humidity ratio of air
- Concentration of desiccant solution
- Physical properties of ambient air, desiccant
- Heat and mass transfer area.

Air - Desiccant interface

The humidity ratio is given by the following equation:

eq. 135
$$\frac{d\dot{m}_W}{dA} = \dot{m}_a \frac{dW}{dA}$$

During dehumidification however, the amount of moisture that is evaporated from the solution is transferred to the air, so the mass flow rate of the desiccant solution is given by the following expression:

eq. 136
$$\frac{d\dot{m}_S}{dA} = \dot{m}_A \frac{dW}{dA}$$

The mass balance equation for the liquid desiccant solution gives:

$$eq. 137 \ d(\dot{m}_S X) = 0$$

Differentiating this equation over area dA, the concentration of the solution is obtained:

eq. 138
$$\frac{dX}{dA} = -\frac{\dot{m}_a}{\dot{m}_s} X \frac{dW}{dA}$$

The gradient of humidity ratio along column height expresses the rate of water vapor absorbed by the desiccant at the interface within area dA and can be expressed as follows:

eq. 139
$$\frac{dW}{dA} = -\frac{K_G}{\dot{m}_a} \left(W^{\text{sat}} - W_{in} \right)$$

Where W^{sat} is the humidity ratio of the saturated air that is in equilibrium with the desiccant solution at the local solution temperature and concentration.

$$_{eq.\ 140}$$
 W^{sat}(T_s)=0.622 $\frac{p_{W,st}^{sat}(T_s)}{p_{atm}-p_{W,st}^{sat}(T_s)}$

Air phase

The energy balance gives:

$$eq. 141 \quad \dot{m}_a dh_a = dQ_S + dQ_L$$

The enthalpy of moist air is given by:

$$eq. 142 \quad h_a = C_{p,da} T_a + W(C_{p,st}^{sat} T_a + \Delta h_{abs})$$

The term dQ_S represents the sensible heat load, which is transferred from the air to the desiccant solution, due to their temperature difference and it can be written as follows:

$$eq. 143 \quad dQ_S = a_a (T_a - T_s) dA$$

Whereas the latent heat load dQ_L transferred from the air to the desiccant solution due to the water evaporation is:

$$eq. 144 \quad dQ_L = d\dot{m}_s (C_{p,st}^{sat} T_s + \Delta h_{abs})$$

However, the specific heat capacity of the moist air is:

$$eq. 145 \quad C_{p,ma} = C_{p,da} + W C_{p,st}^{sat}$$

Using eq. 136, eq. 141, eq. 143, eq. 144 and eq. 145, the rate of change of air dry bulb temperature is obtained:

eq. 146
$$\frac{dT_a}{dA} = \frac{T_a \cdot T_s}{C_{p,ma}} \left(\frac{a_a}{\dot{m}_a} - C_{p,st} \frac{sat}{dA} \frac{dW}{dA} \right)$$

Liquid desiccant phase

Combining eq. 136 with eq. 139, the mass of the water absorbed by the solution per unit area is produced:

$$eq. 147 \quad \frac{d\dot{m}_{S}}{dA} = -K_{G} \left(W^{Sat} - W_{in} \right)$$

The energy balance gives:

$$eq. 148 \quad \dot{m}_{S}dh_{S} + h_{S}d\dot{m}_{S} = dQ_{S,S \rightarrow W} + dQ_{L,S \rightarrow W}$$

Also, the enthalpy of the solution is given by the following expression:

$$eq. 149 \quad h_{S} = -C_{p,S}(T_{S} - T_{S,in}) \Longrightarrow dh_{S} = C_{p,S}dT_{S}$$

And the sensible heat that is transferred from the solution to the water stream is:

$$eq. 150 \quad dQ_{S,s \to W} = U(T_s - T_w) dA$$

Differentiating eq. 149 over dA, the rate of change of desiccant temperature T_s is produced:

$$\frac{dT_s}{dA} = -\frac{1}{\dot{m}_s C_{p,s}} \left[\dot{m}_a \left[C_{p,s} \left(T_s - T_{s,in} \right) + C_{p,st} \frac{sat}{T_s} + \Delta h_{abs} \right] \frac{dW}{dA} \right] + a_a \left(T_s - T_a \right) + U(T_w - T_s)$$

eq. 151

However, the sensible heat that is transferred from the solution to the water stream can be also written as:

$$eq. 152 \quad dQ_{S,s \to w} = \dot{m}_w C_{p,w} dT_w$$

Using eq. 150 and eq. 152 , the rate of change of water temperature T_w is produced:

$$_{eq.\ 153} \ \frac{dT_{W}}{dA} = \frac{U(T_{S} - T_{W})}{\dot{m}_{W}C_{p,W}}$$

Finally, a system of six ordinary differential equations is constituted:

$$eq. 154 \begin{cases} \frac{dT_a}{dA} = \frac{T_a - T_s}{C_{p,ma}} \left(\frac{a_a}{\dot{m}_a} - C_{p,st}^{sat} \frac{dW}{dA} \right) \\ \frac{dT_s}{dA} = -\frac{1}{\dot{m}_s C_{p,s}} \left[\dot{m}_a \left[C_{p,s} (T_s - T_{s,in}) + C_{p,st}^{sat} T_s + \Delta h_{abs} \right] \frac{dW}{dA} \right] \\ \frac{dT_w}{dA} = -\frac{U(T_s - T_w)}{\dot{m}_w C_{p,w}} \\ \frac{dW}{dA} = -\frac{K_G}{\dot{m}_a} \left(W^{sat} - W_{in} \right) \\ \frac{d\dot{m}_s}{dA} = \dot{m}_a \frac{dW}{dA} \\ \frac{dX}{dA} = -\frac{\dot{m}_a}{\dot{m}_s} X \frac{dW}{dA} \end{cases}$$

with the following boundary conditions:

$$eq. 155 \begin{cases} (T_a)_{z=0} = T_{a,in} \\ (T_s)_{z=A} = T_{s,in} \\ (T_w)_{z=A} = T_{w,in} \\ (W)_{z=0} = W_{in} \\ (\dot{m}_s)_{z=A} = \dot{m}_{s,in} \\ (X)_{z=A} = X_{in} \end{cases}$$

The five-order Runge - Kutta was used to numerically integrate the above six ordinary differential equations by using a trial step at the midpoint of an interval to cancel out lowerorder error terms. The model was developed using the Mathcad software [11] where all the thermodynamic characteristics of the dehumidifier are calculated from the point where the liquid desiccant is sprinkled till the bottom of the device.

5.2.4 Parametric analysis

5.2.4.1 Air inlet temperature

The sensible capacity of the dehumidifier was found to depend strongly on air inlet temperature. As the weather becomes hotter, the sensible capacity of the dehumidifier is increased since more sensible heat is transferred to the water stream. For example, at 32° C and 0.012kg_w/kg_{da}, the Q_{sens.deh} is 10.17kW, while at 36° C and 0.012kg_w/kg_{da}, the Q_{sens.deh} is 13.15kW. This fact justifies the necessity of the water tubes. The absence of water would mean higher liquid desiccant temperatures and subsequent rise in interfacial equilibrium humidity. The driving force for mass transfer would thus be decreased and the dehumidifier latent and sensible capacity would be reduced.

5.2.4.2 Air inlet humidity ratio

As the weather becomes more humid, the simulations showed that the latent capacity of the dehumidifier and the dehumidification mass rate is increased. Specifically, at 40°C and 0.011 kg_w/kg_{da}, the Q_{lat.deh} is 9.37kW and the Δm_{deh} is 13.34kg_w/h, while at 40°C and 0.016 kg_w/kg_{da} the Q_{lat.deh} becomes 15.1kW and the Δm_{deh} 21.35kg_w/h. This happens because a higher humidity ratio implies a higher air vapor pressure and thus, a higher average water vapor pressure difference between the air and the desiccant. Consequently, higher potential for mass transfer from the gas to the liquid phase is established. Therefore, the numerator of the eq. 134 is increased, while the denominator remains approximately constant. So, as the weather becomes more humid, the system COP is increased.

The simulation showed also that when the inlet humidity ratio (ambient air) varied from 0.011 to 0.016 kg_w/kg_{da}, the outlet humidity ratio (inlet air to the building) ranged from 0.0065 to 0.0089 kg_w/kg_{da}.

5.3 REGENERATOR

5.3.1 Overview

Recently, liquid desiccant air conditioning systems have attracted more attention due to their environmentally friendly technology and promising utilization of low-grade thermal energy provided by solar flat-plate collector or waste heat [12]. Energy savings, relative to conventional vapor compression systems, of up to 40% can be achieved by using a desiccant assisted air conditioning system [13].

The role of the regenerator device is to restore the concentration of the weak desiccant solution that exits the dehumidifier. At the outlet of the dehumidifier, the desiccant has been diluted with the moisture of the process air stream and has to regain its initial concentration. It therefore flows into the regenerator, where it is sprayed at the top. Gravity forces it to flow downwards, contacting the ambient air that moves upwards, through the tubes. At the same time, heating water is circulated through the tubes and supplies the necessary amount of heat. As the liquid desiccant is heated by the water, water vapor is driven out of the weak solution and carried away by the ambient air stream. At the outlet of the regenerator, the desiccant has gained its initial concentration and it is ready to re-enter the absorber [14, 15].

Peng and Howell [17] developed mathematical models for two liquid desiccant systems powered by solar or geothermal energy and using triethylene glycol as the desiccant solution, as early as 1981. Their parametric analysis showed that the optimum absorber height for the process recirculation mode is 1.2 m, while for the exhaust recirculation mode is 0.9 m. The exhaust recirculation mode has better thermal performance and lower fan power requirements at full capacity, but the process recirculation mode produces cooler and drier air. Three years later, they [17] proposed a liquid desiccant cooling system that uses triethylene glycol, involves low parasitic power requirements and uses thermal energy at a temperature as low as 60°C. Their study analyses and compares the two basic regenerator designs; the open falling film (roof-type solar still or inclined surface) and the closed air circulation (inclined surface or internally heated spray chamber). Following the development of the mathematical models, their parametric analysis has shown that the closed spray chamber with a finned-tube arrangement has a much higher water evaporation rate than the open design. The closed spray chamber with finned-tube coils was found to be the most practical amongst all, allowing for steady operation and compactness in size. It was also found suitable for multipurpose solar applications, as it can be powered by other types of low-grade thermal energy. Scalabrin [3] developed a one-dimensional mathematical model of an internally heated regenerator, in which the air flows counter-currently with the LiCl solution and the heating water enters at the top, flowing inside a horizontal bank of staggered tubes. He managed to predict the behavior of the apparatus both globally and locally and he evaluated the energetic performance of the regenerator via a detailed sensitivity analysis. Since he observed an opposite trend between the energy consumption and the temperature of the heating water, he finally suggested a compromise between the heat quantity and its thermal level. According to his findings, the most important parameters are the air inlet flow rate, the solution inlet concentration and temperature and, particularly for the summer conditions, the solution exchanger effectiveness. His cooperation with Scaltriti [4] lead to the simulation of a liquid desiccant system, employing cooling-tube bank arrangement for both the regenerator and the dehumidifier. LiCl was used as the hygroscopic agent. They developed one-dimensional models of the main components (regenerator, dehumidifier and cooling tower) that a very high accuracy with regard to mass and energy balance. The independent variables controlling the performance of the system and their variation effects are well defined. The performance analysis included three typical working modes; in the case of load variation, the system was found to respond well without changing the solution concentration, the refrigerating power and the COP. Jain et al [18] attempted to evaluate seven liquid desiccant cycles for hot and humid climates. A computer simulation model was developed, based on the constant effectiveness of heat exchangers and evaporative coolers. The effect of various outdoor conditions and the effectiveness of heat exchangers on the cooling capacity have been investigated. It was found that a combination of dehumidifier and wet surface heat exchanger shows better performance, for a wide range of outdoor conditions. Park et al [19] simulated the heat and mass transfer in a liquid desiccant (triethylene glycol) dehumidifier and regenerator. Both devices were plate finned-tube heat exchangers, with air and desiccant in cross flow and water was circulated through the tubes. They presented the governing equations, boundary conditions and interfacial equilibrium conditions and concluded that the problem is more complex than for co-current or counter-current flow, since it is three-dimensional. Their numerical predictions compare quite well with the experimental results. They found that lower air flow rates provide better humidity control and cooling over the investigated conditions.

One of the earliest studies comparing the performance of different desiccant solutions in internally heated/cooled devices was that of Kim et al [20]. Indeed, they tried to evaluate,

both theoretically and experimentally, the performance of LiCl and LiBr in a vertical falling film absorber. They concluded that LiBr outperformed LiCl, in terms of absorption rate; however, solution cost favors LiCl even though LiCl requires a larger absorber surface area. Khan [21] presented heat and mass transfer analysis of an internally cooled liquid desiccant absorber, which was considered as a spray finned tube heat exchanger. A two dimensional steady state model was developed and solved numerically. Based on the NTU-effectiveness methodology, a performance prediction model was developed to estimate the annual energy requirements of the system using an hour-by-hour analysis. Chung and Wu [22] studied and experimentally tested a spray-tower absorber operating with triethylene glycol. The absorber consisted of a U-shape tunnel with eliminators and fin coils, where both air and solution were introduced at the top, in co-current flow. The found that the absorption efficiency increases as the air flow rate decreases, the solution flow rate increases or as the solution temperature decreases. The overall mass transfer coefficient increased with increasing air and solution flow rates, solution concentration and inlet air humidity. Spray towers with and without fin coils using triethylene glycol were compared for their efficiency in the next study of Chung et al [23]. The main parameters varied during the experiments included the air flow rate, the liquid flow rate, the temperature and humidity of inlet air, the average temperature of the fin coils, and the inlet desiccant solution concentration. They found that the efficiency of the absorber with fin coils was 20-30% higher than that of the absorber without fin coils. The overall mass-transfer coefficient was evaluated for both systems, using though the same equation.

Jain et al [24] studied a shell and tube absorber with air and LiBr in parallel flow and with cooling water in counter-flow direction, combined with a counter-flow plate regenerator. For both devices, a numerical solution has been obtained using the finite difference method, by dividing their length into a large number of steps. The heat and mass transfer coefficients were obtained using appropriate correlations from their previous studies. Since their goal was to predict the system cooling capacity and energy requirements, they simulated the whole system adopting the Warner's technique, coupled with an optimization routine. They finally produced an optimized design for a 20 TR unit, with a thermal COP 0.33. At the same year, they [6] experimentally studied a counter flow, internally cooled liquid desiccant system using LiBr as the desiccant solution. The absorber is a falling film tubular type, whereas the regenerator consists of parallel iron sheets. By using appropriate correlations available in literature for the heat and mass transfer coefficients, they developed a numerical model that predicts the experimental data within 30%.

5 NON-ADIABATIC TUBULAR MODEL

A non-adiabatic counter-flow dehumidifier packed with finned tube heat exchangers, using triethylene glycol and cooled with water is investigated in the studies of Chebbah [25, 26]. A simplified theoretical model, based on the logarithmic mean differences, has been developed to investigate the heat and mass transfer during dehumidification and regeneration processes. The heat and mass transfer coefficients of the wetted finned-tube heat exchanger were computed using previously developed correlations. The parametric studies has shown that the cooling of the dehumidification process enhances the mass transfer, makes the operation more efficient and compact and reduces the desiccant strong solution requirements. Obvious consequences are the lower regenerating temperature requirements and the higher performance of the solar powered liquid desiccant air conditioning systems.

Islam et al [27] managed to develop a coupled heat and mass transfer model for a counter flow, tubular absorber using LiBr. The mass transfer coefficients obtained from the coupled model and the log-mean difference model agree within 10%. However, the values given by the two models regarding the heat transfer coefficient significantly differ. In their next study [28] they modified a conventional horizontal tube absorber by introducing film-guiding fins between the tubes to produce a film-inverting arrangement. They succeeded a 100% increase in the absorption rate, compared with the conventional tubular absorber. Their numerical simulation results indicate that the vapor absorption rate can be increased by using a large number of film-inverting segments in the absorber. Following that, they developed a linearized coupled model for the heat and mass transfer in falling-film absorbers [29] and they compared it with a non-linear model and a numerical simulation. Their new model yields analytical expressions that are used to determine heat and mass transfer coefficients from the experimental data for a horizontal tubular absorber and a vertical tube absorber. The mass transfer coefficients obtained using the linearized model and the log-mean difference model agree within about 10%, while the heat transfer coefficients showed large differences. They found that the overall Nusselt and Sherwood number for the tubular absorber increase with increasing film Reynolds number and inlet cooling water temperature.

The study of Kukreja et al [30] provides a new empirical heat transfer correlation for the condensation of pure and blended refrigerants in smooth and micro fin tubes. The correlation predicts the local heat transfer coefficient for pure and blended refrigerants in horizontal micro fin tubes within ±15% range. Kumar et al [31] analyzed the performance of a liquid desiccant system by solving the basic mass and energy balance equations through
the fourth order finite difference Runge-Kutta method. The overall system has been simulated using Warner's technique. Based on the Jain cycle [18], the authors proposed two new liquid desiccant cycles with multiple absorbers. The absorber and regenerator have been considered as shell and tube type heat exchangers, where water (in absorber) or oil (in regenerator) circulates inside the tubes. The results showed that these two new cycles improve the performance by 67-116%, in comparison to conventional Jain cycle. The key design parameters of the proposed cycles have been also identified and are: air mass flow rate, liquid desiccant flow rate and inlet temperature of oil and water. Recently, Jeong et al [32] used a previously validated simulation model to determine the minimum heat source temperature for driving various desiccant dehumidification systems utilizing silica gel. Two of the four considered systems were the adiabatic liquid desiccant tower and the internally cooled liquid desiccant tower. Their results indicate that out of the four systems, the latter can be driven by the lowest heated air temperature, approximately 33°C. The most recent study of Bansal et al [15] compares the moisture removal and effectiveness of an adiabatic dehumidifier with an internally cooled dehumidifier, using CaCl₂. The tubular absorber achieved an effectiveness of 0.55-0.706 and removed 0.005 kg moisture / kg of dry air, while the adiabatic absorber had an effectiveness of 0.38-0.55 and removed 0.0034 kg moisture / kg of dry air. It has been also found that the optimum liquid to gas flow rate ratio for maximum effectiveness is about 1.0. The effectiveness of a cross flow dehumidifier is much less as compared to a counter flow configuration despite internal cooling. The authors suggest that further detailed investigations are required to accurately predict this mass transfer. There is also a need to carry out optimization studies on the design of the cooling coil for achieving maximum effectiveness of the internally cooled dehumidifier.

Less work has been done on the performance comparison of the internally heated regenerator using various liquid desiccant materials. This paper presents a mathematical model that can predict the efficiency of an internally heated liquid desiccant regenerator, using three different aqueous desiccant solutions (LiCl, LiBr and CaCl₂). The model is validated by the experimental findings of Scalabrin [3] and the main factors that influence the internally heated regenerator's performance are discussed in detail.

5.3.2 Thermodynamic parameters

5.3.2.1 Vapor Pressure

The model developed in this study used equations available from the literature, in order to express the water partial vapor pressure of the desiccant solution, the heat of evaporation and the heat and mass transfer coefficients.

Regarding the water partial vapor pressure of the desiccant solution, it determines the humidity ratio of the air that is in equilibrium with the desiccant at the interface. In other words, during regeneration, moisture transfer takes place from the desiccant solution to the air only when the desiccant's equilibrium water vapor pressure is higher than the water vapor pressure in the air. Although the vapor pressure of single electrolyte aqueous solutions has been reported extensively in the literature, only Cisternas and Lam [1, 2] have developed vapor pressure expressions for the three aqueous solutions under investigation following the same experimental procedure. So, the present study uses the vapor pressure expressions from this work to develop the model. The equivalent humidity ratio of liquid desiccant can be calculated through the equilibrium state of air, as shown in the following equation:

$$_{eq.\ 156} \quad W^{sat}\left(T_{s}\right) = 0.62197 \frac{p_{w,st}^{sat}\left(T_{s}\right)}{p_{atm} - p_{w,st}^{sat}\left(T_{s}\right)}$$

5.3.2.2 Heat of evaporation

During regeneration (desorption procedure), water vapor molecules of the liquid desiccant solution evaporate from the liquid-vapor interface and are transferred to the process air. The heat of evaporation expresses the energy required to transform a given quantity of moisture into gas at a given pressure. Thus, the energy required for the evaporation of 1 kg of water from the desiccant solution is expressed by the heat of evaporation, given by the following expression [33]:

$$eq. 157 \quad \Delta h_{evap} = h_v(p, T_s) - h_{w,s}(T_s, X)$$

Since the evaporation heat is not constant, but changes proportionally to the concentration and conversely to the temperature, the above expression was used during the development of the model.

5.3.2.3 Heat & Mass Transfer Coefficients

Correlations for the heat and mass transfer coefficients obtained for internally heated tubular desiccant regenerators are available in the literature. In the present investigation, the heat and mass transfer coefficients at any step are calculated by using the correlations of Jain et al [6]. The water-side heat transfer coefficient is expressed as:

$$_{eq.\ 158} \ a_{W} = \frac{Nu_{W}k_{W}}{D_{in}}$$

The Nusselt and Reynolds numbers for the water are given by the following expressions:

$$_{eq.\ 159}$$
 $Nu_{W} = 0.36 \operatorname{Re}_{W}^{0.55} \operatorname{Pr}_{W}^{1/3}$

$$_{eq.\ 160} \quad \text{Re}_{W} = \frac{\dot{m}_{W} D_{in}}{A_{in} \mu_{W}}$$

For water flow inside tubes, the Prandtl number Pr_w is assumed as 7.03. The heat transfer coefficient from the tube surface to the solution is expressed as:

$$eq. 161 \quad a_{S} = \frac{Nu_{S}k_{S}}{S}$$

The film thickness S is calculated as in [6]. The Nusselt and Reynolds numbers for the solution are given by the following expressions:

eq. 162
$$Nu_s = 0.67 \operatorname{Re}_s \frac{0.11}{k_s H} \left(\frac{C_p \mu_s S}{k_s H} \right)^{1/3}$$

$$_{eq.\ 163} \ \operatorname{Re}_{S} = \frac{\dot{m}_{S} D_{out}}{A_{out} \mu_{S}}$$

The heat and mass transfer coefficients from the solution interface to the air stream are expressed as:

eq. 164
$$K_a = \frac{\dot{m}_a}{A_{out}} \frac{0.023 \text{Re}_a^{-0.17}}{Sc_a^{2/3}}$$

$$a_{a} = C_{pm} K_a L e^{2/3}$$

The Reynolds and Schmidt numbers are given by the following expressions:

$$_{eq.\ 166}$$
 Re $_{a} = \frac{\dot{m}_{a}D_{i}}{A_{out}\mu_{a}}, Sc_{a} = \frac{\mu_{a}}{\rho_{a}\delta}$

The diffusion coefficient δ of the water vapor to the air is calculated by the expression of Bolz and Tuve [7]. The heat transfer coefficient from the water to the solution is given by the following expression:

eq. 167
$$U = \left[\frac{D_{out}}{D_{in}}\frac{1}{a_w} + \ln\left(\frac{D_{out}}{D_{in}}\right)\frac{D_{out}}{2k_{pipes}} + \frac{1}{a_s}\right]^{-1}$$

From all the above relations, the model can evaluate the thermodynamic parameters in every step along the height of the regenerator, under any given input conditions. Using the basic mass and energy balance equations, the output conditions are then yielded.

5.3.2.4 **Performance indices**

The mass transfer performance of the regenerator is evaluated in terms of the regeneration mass rate and the regenerator efficiency. The regeneration mass rate (kg_w/h) , is calculated by the following expression:

$$eq. 168 \quad \dot{m}_{reg} = \dot{m}_a (W_{out} - W_{in})$$

The regenerator efficiency is defined as the ratio of the actual humidity ratio variance of the air passing through the regenerator to its variance under ideal conditions, as shown below:

$$_{eq.\ 169} \quad n = \frac{W_{out} - W_{in}}{W^{sat} - W_{in}} 100\%$$

Where W^{sat} is the humidity ratio of the saturated air that is in equilibrium with the desiccant solution at the local solution temperature and concentration.

5.3.3 Mathematical model development

5.3.3.1 Assumptions

The model was developed as an extension of Rogdakis et al. [33], Papaefthimiou [9, 10] and Christodoulaki [36] work. Based on the energy and mass conservation laws, the developed model describes coherently the coupled heat and mass transfer processes taking place inside an internally heated tubular liquid desiccant regenerator.

The regenerator area A is divided into 15 equal segments and the mass and energy balances are solved for each segment, from the top of the regenerator A = 0, until the bottom of the regenerator, A=1; thus, gradients of air temperature, humidity ratio, desiccant temperature, desiccant flow rate and concentration are calculated across the height of the regenerator. In order to begin calculations, the following initial parameters should be assumed:

- Temperature of inlet air, inlet desiccant solution and inlet heating water,
- Flow rate of inlet air, inlet desiccant solution and inlet heating water,
- Humidity ratio of inlet air,
- Concentration of inlet desiccant solution,
- Physical properties of ambient air, inlet desiccant solution,
- Heat and mass transfer area of regenerator.

The developed model was then validated against experimental data obtained from Scalabrin [3]. The basic geometric characteristics of the regenerator are: 11 rows with 33 tubes each, internal diameter $D_i=0.016$, external diameter $D_{out}=0.02$, total length of tubes 2m, pitch ratio 1.5 [3]. The schematic diagram of the internally heated tubular regenerator is shown in Figure 5-4Figure 4-14.

In order to simplify the complexity of the configuration, the assumptions used in the calculations are:

- One-dimensional analysis: heat and mass transfer occur only in the flow direction,
- Temperature of the liquid desiccant equal to the temperature of the coils,
- Uniform flooding of the regenerator and heat transfer area equal to the mass transfer area; thus, transfer area equal to the specific surface area of the packing,
- Vapor pressure equilibrium between the vapor and the liquid at the interface,

 Negligible heat transfer from liquid to vapour, negligible heat transfer because of radiation, viscous dissipation, pressure gradients, concentration gradients (Dufour effect) or gravitational effects.



Figure 5-4 Non-adiabatic tubular regenerator device

5.3.3.2 Mathematical model



Figure 5-5 Heat and mass transfer during regeneration in a non-adiabatic tubular regenerator

Air - Desiccant interface

By definition, the humidity ratio is given by the following equation:

$$eq. 170 \quad W = \frac{\dot{m}_W}{\dot{m}_a}$$

or after differentiating with dA:

$$eq. 171 \quad \frac{d\dot{m}_{W}}{dA} = \dot{m}_{a} \frac{dW}{dA}$$

During regeneration however, the amount of moisture that is evaporated from the solution is transferred to the air, so:

eq. 172
$$d\dot{m}_S = d\dot{m}_W$$

So, the mass flow rate of the desiccant solution is given by the following expression:

$$eq. 173 \quad \frac{d\dot{m}_S}{dA} = \dot{m}_a \frac{dW}{dA}$$

The mass balance equation for the liquid desiccant solution gives:

$$eq. 174 \quad d(\dot{m}_S X) = 0$$

Differentiating this equation over area dA and using eq. 173, the concentration of the solution is obtained:

eq. 175
$$\frac{dX}{dA} = -\frac{\dot{m}_a}{\dot{m}_s} X \frac{dW}{dA}$$

The gradient of humidity ratio along regenerator height expresses the rate of water vapour that is evaporated from the desiccant at the interface within area dA and can be expressed as follows:

$$_{eq.\ 176} \ \frac{dW}{dA} = \frac{K_a}{\dot{m}_a} \Big(W^{\text{sat}} - W_{in} \Big)$$

Air phase



Figure 5-6 Energy balance of the air phase in a non-adiabatic tubular regenerator

The energy balance gives:

eq. 177
$$\dot{m}_a dh_a = dQ_S + dQ_L$$

The enthalpy of moist air is given by:

$$eq. 178 \quad h_a = C_{p,da} T_a + W(C_{p,st}^{sat} T_a + \Delta h_{evap})$$

The term dQ_s in eq. 177 represents the sensible heat load, which is transferred from the desiccant solution to the air, due to their temperature difference and it can be written as follows:

$$eq. 179 \quad dQ_S = a_a (T_s - T_a) dA$$

Whereas the latent heat load dQ_L transferred from the desiccant solution to the air due to the water evaporation is:

$$eq.\ 180 \quad dQ_L = d\dot{m}_s (C_{p,st}^{sat} T_s + \Delta h_{evap})$$

However, the specific heat capacity of the moist air is:

$$eq. 181 \quad C_{p,ma} = C_{p,da} + WC_{p,st}^{sat}$$

Substituting eq. 179 and eq. 180 into eq. 177 and using eq. 181, the rate of change of air dry bulb temperature is obtained:

$$_{eq.\ 182} \ \frac{dT_a}{dA} = \frac{T_s - T_a}{C_{p,ma}} \left(\frac{a_a}{\dot{m}_a} + C_{p,st} \frac{sat}{dA} \frac{dW}{dA} \right)$$

Liquid desiccant phase

Combining eq. 173 with eq. 176, the mass of the water evaporated from the solution per unit area:

eq. 183
$$\frac{d\dot{m}_s}{dA} = K_a \left(W^{sat} - W_{in} \right)$$



Figure 5-7 Energy balance of the liquid desiccant phase in a non-adiabatic tubular regenerator

The energy balance gives:

$$eq. 184 \quad h_S d\dot{m}_S + dQ_{S,W} = \dot{m}_S dh_S + dQ_L + dQ_S$$

Also, the enthalpy of the solution is given by the following expression:

$$eq. 185 \quad h_s = C_{p,s}(T_s - T_{s,in}) \Longrightarrow dh_s = C_{p,s}dT_s$$

And the sensible heat that is transferred from the water stream to the solution is:

eq. 186
$$dQ_{S,W} = U(T_W - T_S)dA$$

Differentiating eq. 184 over dA and taking into consideration eq. 176, eq. 177, eq. 182, eq. 185 and eq. 186, the rate of change of desiccant temperature T_s is produced:

$$eq. 187 \quad \frac{dT_s}{dA} = -\frac{1}{\dot{m}_s C_{p,s}} \left[\dot{m}_a \frac{dW}{dA} \left[-C_{p,s} (T_s - T_{s,in}) + C_{p,st}^{sat} T_s + \Delta h_{evap} \right] + \right] \\ + a_a (T_a - T_s) + U(T_s - T_w) \right]$$

However, the sensible heat that is transferred from the water stream to the solution can be also written as:

$$eq. 188 \quad dQ_{S,W} = \dot{m}_W C_{p,W} dT_W$$

Using eq. 186 and eq. 188, the rate of change of water temperature $T_{\rm w}$ is produced:

$$_{eq.\ 189} \ \frac{dT_W}{dA} = \frac{U(T_W - T_S)}{\dot{m}_W C_{p,W}}$$

System of Equations

Finally, a system of six ordinary differential equations is constituted:

$$\begin{cases} \frac{dT_a}{dA} = \frac{T_s - T_a}{C_{p,ma}} \left(\frac{a_a}{\dot{m}_a} + C_{p,st}^{sat} \frac{dW}{dA} \right) \\ \frac{dT_s}{dA} = -\frac{1}{\dot{m}_s C_{p,s}} \left[\frac{\dot{m}_a}{dA} \frac{dW}{dA} \left[-C_{p,s}(T_s - T_{s,in}) + C_{p,st}^{sat} T_s + \Delta h_{evap} \right] + \right] \\ \frac{dT_w}{dA} = \frac{U(T_w - T_s)}{\dot{m}_w C_{p,w}} \\ \frac{dW}{dA} = \frac{K_a}{\dot{m}_a} \left(W^{sat} - W_{in} \right) \\ \frac{d\dot{m}_s}{dA} = \dot{m}_a \frac{dW}{dA} \\ \frac{dX}{dA} = -\frac{\dot{m}_a}{\dot{m}_s} X \frac{dW}{dA} \end{cases}$$

with the following boundary conditions:

$$eq. 191 \begin{cases} \left(T_a\right)_{z=0} = T_{a,in} \\ \left(T_s\right)_{z=L} = T_{s,in} \\ \left(T_w\right)_{z=L} = T_{w,in} \\ \left(W\right)_{z=0} = W_{in} \\ \left(\dot{m}_s\right)_{z=L} = \dot{m}_{s,in} \\ \left(X\right)_{z=L} = X_{in} \end{cases} \end{cases}$$

The five-order Runge - Kutta was used to numerically integrate the above six ordinary differential equations. The model was developed using the Mathcad software [11] where all the thermodynamic characteristics of the regenerator are calculated from the point where the liquid desiccant is sprinkled till the end of the coils.

5.3.4 Validation

The developed model has been validated against experimental data obtained from the reference case of Scalabrin [3]. The inlet parameters are: $T_{a,in}=42.3$ °C, $W_{a,in}=0.013$ kg_w/kg_{da}, $\dot{m}_a=2.75$ kg/sec, $T_{s,in}=38$ °C, $\dot{m}_s=2.5$ kg/sec, X=29.55%, $T_{w,in}=53.7$ °C and $\dot{m}_w=5$ kg/sec. Figure 5-8 shows the model predictions, in comparison to the experimental values. As it can be observed, the theoretical model shows very good agreement with the experimental results, having an excellent correspondence between all values.



Figure 5-8 Comparison between the outlet parameters of the experimental data [7] and the mathematical model for the reference case in a non-adiabatic tubular regenerator

The specific heat consumption for water desorption, with respect to the produced difference in concentration ΔX has been plotted in Figure 5-9. Again, the predictions of the model agree very well with the experimental findings, since the maximum discrepancy is only 6.1%.



Figure 5-9 Comparison between the experimental results [7] and the mathematical model, regarding the specific consumption of heat for water desorption in a non-adiabatic tubular regenerator



Figure 5-10 Temperature profiles of air, solution and heating water along the tubular regenerator

5.3.5 Parametric analysis

Following the validation of the model with the experimental results, an extensive theoretical investigation was conducted to examine the effect of various parameters on the regenerator efficiency.

5.3.5.1 Sensitivity analysis

A sensitivity analysis was performed on the inlet and outlet variables of air, water and solution. The independent variables are the inputs; T_{a,in}, W_{in}, m_a, T_{s,in}, X_{in}, m_{s,in}, T_{w,in}, m_{w,in} and the dependent are the outputs; T_{a,out}, W_{out}, RH_{out}, T_{s,out}, X_{out}, m_{s,out}, n, m_w. A regression analysis has been also conducted, with the view to investigate the degree of influence of each independent variable (quantitative or not) on the dependent variables. The linear regression coefficients for the thermodynamic properties of air, water and solution in DDU flow are shown in Table 5-1. Negative coefficient value indicates that the dependent variable is decreased when the corresponding independent variable is increased and vice versa.

From the output results of the multiple linear regression analysis, where all possible combinations of the quantitative and qualitative input variable values were tried, the following remarks can be made:

- R-Square and adjusted R-Square, that represent the quality of the linear fit, were found greater than 84.65% and less than 100%, for each output variable and all three solutions.

- Since the input variables have different measurement scales, the regression coefficient of each input variable found at the end of a regression analysis was transformed to a standardized regression coefficient, in order to compare the relative influence of the variable.

- The standardized regression coefficient of the input variables for all desiccant solutions, in descending influence, are:

 X_{in} (-51% to 100%), W_{in} (-64% to 72%), $T_{w,in}$ (-30% to 100%), $T_{s,in}$ (-10% to 97%), $m_{s,in}$ (-5% to 99%), $T_{a,in}$ (-8% to 48%), m_a (-11% to 11%) and $m_{w,in}$ (-1% to 2%). The results of the sensitivity analysis show that the output conditions are found to be most sensitive to the solution concentration and inlet air humidity ratio change, but least sensitive to the water mass flow rate.

Y=a + b*T _{a,in} + c*m _a + d*W _{in} + e*T _{s,in} + f*m _{s,in} + g*X _{in} + h*T _{w,in} + i*m _w									
LiCl									
Y	а	b	С	d	е	f	g	h	i
T _{a,out}	0.35607	0.29287	-0.39495	116.63714	0.34170	-0.73690	16.66563	0.23492	0.10310
W_{out}	0.01032	0.00007	-0.00070	0.40361	0.00039	-0.00090	-0.08488	0.00030	0.00010
T _{s,out}	-1.07100	0.11253	-1.29921	330.30668	0.10042	-0.07820	47.22634	0.35798	0.28390
m _{s,out}	-0.00534	-0.00019	-0.00616	1.64311	-0.00107	1.00250	0.23282	-0.00084	-0.00040
X _{out}	0.00036	0.00002	0.00074	-0.20161	0.00013	-0.00140	0.98228	0.00010	0.00000
n	0.81007	-0.00354	0.05669	-12.56286	0.03037	-0.09340	-2.03737	-0.00269	-0.01250
m _w	0.00536	0.00019	0.00621	-1.64302	0.00107	-0.00249	-0.23284	0.00084	0.00037
T _{w,out}	-3.62795	0.03059	-0.32539	84.93536	0.12933	-0.19450	13.04167	0.73101	0.75870
LiBr									
Y	а	b	С	d	е	f	g	h	i
T _{a,out}	1.44602	0.29026	-1.92224	266.13307	0.41799	-0.21840	4.70541	0.21134	0.11000
W_{out}	-0.00367	0.00011	-0.00271	0.52814	0.00056	-0.00040	-0.04717	0.00042	0.00020
T _{s,out}	7.27744	0.09395	-2.25821	353.60179	0.09826	0.40760	20.14536	0.28927	0.29750
m _{s,out}	0.05823	-0.00029	-0.00996	1.29807	-0.00153	1.00110	0.12981	-0.00115	-0.00060
X _{out}	-0.00781	0.00004	0.00130	-0.16161	0.00019	-0.00250	1.00428	0.00014	0.00010
n	0.58967	-0.00264	0.06057	-3.86286	0.02401	-0.07210	-0.41957	-0.00281	-0.01540
m _w	-0.05809	0.00029	0.00996	-1.29901	0.00153	-0.00112	-0.12994	0.00115	0.00059
T _{w,out}	-2.33828	0.02571	-0.71317	91.30486	0.09913	0.15570	7.25037	0.72491	0.96940
CaCl ₂									
Y	а	b	С	d	е	f	g	h	i
T _{a,out}	6.81723	0.28846	-1.09321	125.46486	0.25708	0.16550	5.48917	0.15884	0.06970
W_{out}	-0.00383	0.00012	-0.00317	0.51175	0.00058	0.00040	-0.03345	0.00035	0.00020
T _{s,out}	11.41811	0.09647	-2.01738	318.81146	0.03016	0.55290	12.99589	0.22363	0.19760
m _{s,out}	0.06488	-0.00033	-0.01136	1.34411	-0.00161	0.99890	0.09242	-0.00097	-0.00040
X _{out}	-0.00816	0.00004	0.00142	-0.16950	0.00020	-0.00260	1.01150	0.00012	0.00000
n	0.49519	-0.00349	0.04369	-1.36575	0.02916	-0.04550	-0.26275	-0.00425	-0.01260
m _w	-0.06492	0.00034	0.01138	-1.34325	0.00160	0.00106	-0.09232	0.00097	0.00041
T _{w,out}	-2.54961	0.01885	-0.52713	59.58479	0.06611	0.28900	5.33036	0.77603	0.91110

Table 5-1 Linear regression coefficients for calculating the output thermodynamic properties of air,water and solution in the case of DDU flow in a non-adiabatic tubular regenerator

5.3.5.2 Inlet air temperature

As it can be seen in Figure 5-11, the 16% increase in the air inlet temperature causes a slight decrease in the regeneration efficiency (-2.0% to -6.5%) and an increase in the regeneration mass rate (14.9% to 35.3%), for all desiccants and for all flow configurations examined. This happens because higher inlet air temperature implies an extra amount of heat consumed for the regeneration; however, since the extra heat does not bring significant improvement of the regeneration mass rate, the regeneration thermal efficiency is eventually decreased.



Figure 5-11 Effect of water inlet temperature on the regeneration efficiency and the regeneration mass rate of UDU flow in a non-adiabatic tubular regenerator

5.3.5.3 Inlet solution temperature

The results of the mathematical model indicate that higher inlet temperature of the desiccant solution can not only improve the regeneration efficiency but also the regeneration mass rate (Figure 5-12). Indeed, a 12% increase causes a 11.4-21.3% increase in the efficiency and a 13.0-23.7% increase in the regeneration mass rate, depending on the flow configuration and desiccant solution. There is also a slight difference in performance between the two flow types for the CaCl₂ and the LiBr, but this would remain unnoticed in the case of LiCl. The significantly lower performance of LiCl, compared to the other two solutions, is again testified.



Figure 5-12 Effect of solution inlet temperature on the regeneration mass rate of the three desiccant solutions in a non-adiabatic tubular regenerator

5.3.5.4 Inlet humidity ratio

The predominance of CaCl₂ (98%, 201 kg_w/h) towards LiBr (85%, 174 kg_w/h) and LiCl (68%, 89 kg_w/h), in terms of efficiency and regeneration mass rate is also evident for UDU flow (Figure 5-13). This figure also shows that a 50% increase in the air inlet humidity ratio causes a slight decrease in the regeneration efficiency (-8.5% to -0.5%), but a more significant decrease in the regeneration mass rate (-23% to -9%), for all desiccants. This happens because at higher humidity ratios, the partial pressure of water in the air is higher and thus, the difference between the vapor pressure of the desiccant solution and the partial pressure of water in the air is decreased. So, since the driving force for regeneration becomes smaller, both the efficiency and the mass rate are proportionally affected.



Figure 5-13 Effect of inlet humidity ratio on the regeneration efficiency and the regeneration mass rate of UDU flow in a non-adiabatic tubular regenerator

5.3.5.5 Desiccant inlet concentration

The increase of the solution inlet concentration slightly decreases both the regeneration efficiency and the regeneration mass rate for all solutions examined, while this decrease is independent of the flow configuration. This may be explained from the fact that vapor pressure of each desiccant is a function of the concentration. Therefore, as the solution concentration is increased, its vapor pressure is decreased. The potential for mass transfer is thus decreased, leaving a negative effect on the regeneration rate and efficiency. It has been noted that the decrease is more obvious in the case of LiCl (-5% and -17%) and less in CaCl₂ (-0.5% and -3%). The following figure shows the distribution of the air humidity ratio and the solution concentration, under DDU flow and 28% inlet concentration.



Figure 5-14 Distribution of air humidity ratio and solution concentration along the non-adiabatic tubular regenerator, for X_{in} =0.28 and DDU flow

As the solution flows down the regenerator, an amount of water is evaporated and transferred to the air stream. That is the reason why, from top to bottom, the concentration increases and the air humidity ratio increases. The air in contact with $CaCl_2$ solution exits the regenerator at the top with the highest humidity ratio (0.032 kg_w/kg_{da}), which means that the regeneration of $CaCl_2$ is the most efficient of all three solutions (X_{out}=28.65%).

5.3.5.6 Liquid desiccant solution

In this work, some effort was made to compare three salt solutions, namely LiCl, LiBr and CaCl₂, against their regeneration potential under the same initial conditions. Indeed, the humidity ratio distribution in Figure 5-15 confirms that the regenerator using CaCl₂ and DDU flow has the best efficiency, as it can increase the air humidity ratio at the outlet by 166%, followed closely by the UDU flow (161%). The regenerator using LiBr and DDU flow can increase the humidity ratio by 143%, while in UDU flow the result is 138%. The regenerator using LiCl seems to have the worst efficiency, as it increases the humidity ratio by 65% in UDU flow, under the same inlet conditions. From the above results, it can be concluded that the flow configuration is of much less importance than the choice of the desiccant solution itself; LiCl shows a clear deficit, compared to the other two solutions.



Figure 5-15 Humidity ratio distribution across the non-adiabatic tubular regenerator, under DDU and UDU flow, for all three desiccant solutions examined

Figure 5-16 shows the air, water and desiccant temperature distribution along the regenerator's height in DDU flow. At the top of the regenerator, the solution is heated from the hot water; heat transfer therefore occurs from the heated solution to the air, as the air approaches the exit. However, as the solution proceeds downwards, it continues its sensible heat transfer with the air and its latent heat transfer for water evaporation. Considering these heat losses and taking also into account the fact that the hot water temperature decreases as it flows downwards, it is understood that the solution temperature falls as it

approaches the exit. There is a large variation among the desiccant solutions, in terms of air and solution temperature. Starting from a solution temperature of 38° C, CaCl₂ ends up at 33.2° C, while LiCl ends up at 42.6° C. Also, CaCl₂ produces the coldest air (38.2° C), while LiCl the hottest (42.3° C).



Figure 5-16 Temperature profiles of bulk air, solution at the interface and heating water in DDU flow across the non-adiabatic tubular regenerator, for all three desiccant solutions examined

5.4 CONCLUSIONS

This chapter theoretically investigated the heat and mass transfer processes between the air, the desiccant solution and the water circuit in a non-adiabatic dehumidifier and regenerator. The "non-adiabatic" term is used to show that during dehumidification and regeneration there is interaction with external energy sources. These sources are translated into a tubular heat exchanger which is added inside the dehumidifier and the regenerator device, to provide cooling and heating respectively, through a closed water circuit. Cooling water is provided to the dehumidifier through an evaporative cooler and hot water is provided to the regenerator through a low-temperature hot water source.

A heat and mass transfer model of a non-adiabatic tubular dehumidifier has been developed to predict the performance of the liquid desiccant dehumidifier under various operating conditions. The advantage of this model is its ability to easily adapt under different operational conditions, different flow configurations and different liquid desiccant solutions. Excellent agreement was found between the developed computational program and published experimental data.

Following the model development, a parametric analysis was conducted to investigate the parameters which have an important effect on the dehumidifier's performance. The sensible capacity was found to depend strongly on air inlet temperature. As the weather becomes hotter, the sensible capacity of the dehumidifier is increased since more sensible heat is transferred to the water stream. This fact justifies the necessity of the cooling water tubes. The absence of water would mean higher liquid desiccant temperatures and subsequent rise in interfacial equilibrium humidity. The driving force for mass transfer would thus be decreased and the dehumidifier latent and sensible capacity would be reduced. As the weather becomes more humid, the simulations showed that the latent capacity of the dehumidifier and the dehumidification mass rate is increased. This happens because a higher humidity ratio implies a higher air vapor pressure and thus, a higher average water vapor pressure difference between the air and the desiccant. Consequently, higher potential for mass transfer from the gas to the liquid phase is established. So, as the weather becomes more humid, the system coefficient of performance is increased.

The performance of the non-adiabatic regenerator was simulated through the development of a heat and mass transfer numerical model. The theoretical model showed very good agreement with the experimental results, having an excellent correspondence between all values. As such, the deviation between the model predictions and the published experimental values was less than 10%.

The results of the sensitivity analysis showed that the output conditions are most sensitive to the solution concentration, the air humidity ratio, the hot water temperature, the solution temperature and the solution flow rate. It was found that the regeneration mass rate increased with air, solution and water temperature, air flow rate, decreased with humidity ratio and concentration, but was almost unaffected with water flow rate. The regeneration efficiency increased with solution temperature and decreased with water temperature, humidity ratio and solution concentration. It was slightly affected by the air temperature, air flow rate and water flow rate. The results of this study also suggest that:

- The increase in the air inlet temperature causes a slight decrease in the regeneration efficiency and an increase in the regeneration mass rate, for all desiccants and for all flow configurations examined.
- The regeneration potential decreases when the regeneration process takes place in more humid climates. An increase in the air inlet humidity ratio causes a slight decrease in the regeneration efficiency, and a significant decrease in the regeneration mass rate, for all desiccants. This happens because at higher humidity ratios, the partial pressure of water in the air is higher and thus, the difference between the vapor pressure of the desiccant solution and the partial pressure of water in the air is decreased. So, since the driving force for regeneration becomes smaller, both the efficiency and the mass rate are proportionally affected.
- Higher inlet temperature of liquid desiccant should be preferred to get more regeneration capacity, for all desiccants and for all flow configurations examined. The higher inlet temperature of the desiccant solution can not only improve the regeneration efficiency but also the regeneration mass rate.
- The increase of the solution inlet concentration slightly decreases both the regeneration efficiency and the regeneration mass rate for all solutions examined, while this decrease is independent of the flow configuration. This may be explained from the fact that vapor pressure of each desiccant is a function of the concentration. Therefore, as the solution concentration is increased, its vapor pressure is decreased. The potential for mass transfer is thus decreased, leaving a negative effect on the regeneration rate and efficiency.

- The combination of CaCl₂ as the desiccant solution and DDU flow (water downward

 desiccant downward air upward) shows the best regeneration efficiency and
 mass rates among all cases considered, as it can increase the air humidity ratio at
 the outlet by 166%.
- On the contrary, the regenerator using LiCl seems to have the worst efficiency, as it can increase the humidity ratio only by 67% in DDU flow (water downward desiccant downward air upward) and by 65% in UDU flow (water upward desiccant downward air upward).
- The flow configuration is of much less importance than the choice of the desiccant solution itself; LiCl shows a clear deficit, compared to the other two solutions.

5.5 NOMENCLATURE

- a heat transfer coefficient (W/m²K)
- A heat transfer area (m²)
- A_{out} cross section for external flow (m²)
- A_{in} cross section for internal flow (m²)
- C_p specific heat capacity at constant pressure (Jkg⁻¹K⁻¹)
- D diameter
- h enthalpy $(kJ/kg_{d\alpha})$
- k thermal conductivity (W/mK)
- K mass transfer coefficient (kg/m²s)
- K_G mass transfer coefficient (kg/m²s)
- \dot{m}_{reg} regeneration mass rate (kg_w/h)
- \dot{m}_{deh} dehumidification mass rate (kg_w/h)
- \dot{m} mass flow rate (kg/s)
- n regenerator efficiency
- Q heat (J)
- Q capacity or load (kW)
- S film thickness (m)
- T temperature (°C)
- U overall heat transfer coefficient (W/m²K)
- W air humidity ratio ($kg_w/kg_{d\alpha}$)
- X salt concentration in solution (kg_{salt}/kg_s)
- Y moisture content in solution (kg_w/kg_s)
- Δh_{abs} heat of water absorption (J/kg)
- Δh_{evap} latent heat of water evaporation (J/kg)

Greek symbols

- $\delta \qquad \text{ diffusion coefficient of water vapor to air} \\$
- μ dynamic viscosity (kg/ms)
- ρ density (kg/m³)

Subscripts						
α	air					
abs	absorption					
atm	atmospheric					
CW	cooling water					
deh	dehumidifier					
dα	dry air					
eq	equivalent					
evap	evaporation					
exp	experimental					
in	in					
L	latent					
lat	latent					
mα	moist air					
out	out					
reg	regenerator					
S	solution					
S	sensible					
sens	sensible					
sim	simulated					
st	steam					

w water

Superscripts

sat saturated

5.6 **REFERENCES**

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6 LIQUID DESICCANT EXPERIMENTAL UNIT

6.1 INTRODUCTION

This chapter presents the results from a study of the performance of an internally cooled liquid desiccant dehumidifier. A plate heat exchanger is proposed as the internally cooled element of the dehumidifier and water as the cooling fluid. The desiccant solution is sprayed into the internally cooled dehumidifier from the top and flows down by gravity. At the same time, fresh humid air is blown from the bottom or top, counter-flowing or co-flowing with the desiccant solution. The desiccant is in direct contact with the air, allowing for heat and mass transfer. The cooling water, flowing inside the plates of the dehumidifier, carries out the heat of the crossed air and solution. A heat and mass transfer theoretical model has been developed, based on the Runge-Kutta fixed step method, to predict the performance of the device under various operating conditions. Experimental data from previous literature have been used to validate the model. Excellent agreement has been found between experimental tests and the theoretical model, with the deviation not exceeding ±4.1% for outlet air temperature and ±4.0% for outlet humidity ratio. Following the validation of the mathematical model, the dominating effects on the absorption process have been discussed in detail. Namely, effects of flow configuration, air inlet temperature, humidity and flow rate, as well as desiccant inlet temperature, concentration and flow rate have been investigated against the dehumidification rate and the cooling efficiency. The two most commonly used liquid desiccant solutions, namely LiCl and LiBr have been also evaluated against each other. The results suggested that high dehumidification mass rate can be achieved under counter flow between air and solution, low air mass flow rates, low cooling water temperature, low desiccant temperature and LiCl as the desiccant solution.

The second part of this chapter presents the liquid desiccant system that has been installed in the Laboratory of Applied Thermodynamics in the National Technical University of Athens. The 8 dehumidification and 12 regeneration experiments that have been carried out show the performance of the apparatus under different design parameters. Energy and mass balance equations are used to calculate the dehumidification range for the dehumidifier and the energy consumption of the regenerator.

6.2 NON-ADIABATIC PLATE DEHUMIDIFIER

6.2.1 Overview

Recently, liquid desiccant air conditioning systems have attracted more attention due to their environmentally friendly technology and promising utilization of low-grade thermal energy provided by solar flat-plate collector or waste heat [1]. Unlike conventional evaporative cooling systems, liquid desiccant systems do not require the air to be cooled below its dew point. Energy savings, relative to conventional vapor compression systems, of up to 40% can be achieved by using a desiccant assisted air conditioning system [2].

The primary function of the dehumidifier in a liquid desiccant air conditioner is to dehumidify the supply air to the building, by bringing it into contact with a liquid desiccant solution. This can be done in an internally cooled device, which is constructed as a parallel plate heat exchanger that is flooded with desiccant. As the strong desiccant solution is sprayed at the top of the device and onto the plates, it flows down by gravity and comes in contact with the process air that can be moving downwards or upwards, through the passageways. The desiccant which, by definition, has a strong affinity for water vapour absorbs moisture from the air. However, as the water vapour condenses and mixes with the desiccant, heat is released. This heat equals to the latent heat of condensation for water plus the chemical heat of mixing between the desiccant and water [3]. This heat would raise the temperature of the desiccant, decreasing its ability to remove water vapor from the air [4]. For this reason, cooling water flows downwards inside the plates and immediately absorbs this heat. The cooling water restricts the temperature changes for both the desiccant and air and therefore, increases the dehumidification capacity of the absorber.

The driving force for the mass transfer between the liquid desiccant solution and the air is the difference between the vapor pressure of the desiccant and the partial pressure of water vapor in the air [5]. Namely, moisture transfer takes place from the air to the desiccant solution only when the cool, strong desiccant solution vapor pressure is less than the water vapor pressure in the air.

Salt solutions like LiBr and LiCl are most commonly used as liquid desiccants, in comparison to glycols, since they have essentially zero vapor pressure [6]. Although glycols are less corrosive than salts, they have very low surface vapor pressure [7] and their evaporation contaminates the air, increasing the risks for respiratory irritations [8]. These evaporation losses are unacceptable in an occupied building; therefore salts dominate [9].

Due to the configuration complexity of the internally cooled dehumidifier, the heat and mass transfer characteristics have not been fully studied compared with the adiabatic dehumidifier. Grossman [10] performed, as early as 1983, a theoretical analysis of the combined heat and mass transfer process taking place in the absorption of a gas into a laminar liquid film. Two methods of solution (analytical and numerical) were used with very good agreement between them. The heat and mass transfer coefficients for typical wall conditions and the correlations for Nusselt and Sherwood numbers were produced. Under all conditions studied, they observed that the rate of absorption is lower with an adiabatic wall (packed bed) than with a constant temperature wall (internally cooled). Few years later, another study used the standard Galerkin finite element method and developed a model for the coupled heat and mass transfer in laminar film flow over a cooled wall [11]. They also found correlations for the Nusselt and Sherwood number. Assuming the cooling water temperature constant over the plate height, the results showed that the coupled process differs significantly from that for pure heat transfer and pure mass transfer. Lowenstein and Gabruk [3] calculated the heat transfer coefficients from an empirical expression that applies to channels bounded by flat plates with no liquid-gas interactions and assumed the Lewis number as one. They found higher system performance for lower water flow rates and cocurrent flow between the desiccant and the air. Park et al. [12] performed a detailed numerical analysis of the heat and mass transfer between a falling desiccant film (TEG) on a vertical plate and air in cross-flow. The experimental rig consisted of a hybrid vapor compression and liquid desiccant unit. The absorber/regenerator was a plate-fin tube heat exchanger, where water was circulated through the tubes and the air was blown in crossflow through the fins to contact the TEG film. They found that the low air flow rate clearly shows better humidity control and regeneration performance. Lowenstein and Novosel [13] modeled a liquid desiccant application that processes the mixed air stream composed of return air and ventilation air, in an office building in Atlanta. The system incorporates an internally cooled parallel plate absorber and a double-effect regenerator. Using their previously developed simulation model [3], they calculated the seasonal performance of the liquid desiccant air conditioner. They found that seasonal COPs between 1.44 and 2.24 are possible. If however, the liquid desiccant air conditioner processes only ventilation air, it can serve 52% of the cooling load and reduce the peak electrical demand by 47%. Kessling et al [14] developed a steady-state analytical simulation model to investigate the dehumidification potential and energy storage capacity of a counter flow parallel plate dehumidifier, using LiCl and Klimat. They proved that the cooled absorption process exhibits

more efficient dehumidification and significantly higher storage capacities than the adiabatic. In their later work [15], they carried out single channel experiments with a parallel plate, counter flow dehumidifier using LiCl. They experimentally identified the mass transfer coefficients and they developed two numerical models; one differential, based on five differential equations derived from the mass and energy balance and one isothermal, based on the number of transfer units. They found that efficient absorption occurs under air to solution mass ratio 50-100. Khan and Martinez [16] developed a mathematical model to predict the performance of an internally cooled parallel plate LiCl absorber with almost isothermal operation. The model was based on the control volume numerical method and the absorber was divided into a finite number of control volumes in the process air flow direction. They observed negligible performance differences between parallel and counter fluid flow arrangements. At the same year, Khan alone [17] based on the NTU-effectiveness method and solved numerically a two dimensional steady state model. The studied device was a finned-tube liquid desiccant dehumidifier, with air and water in a cross flow with the LiCl solution. He estimated the average heat transfer coefficient by using fully developed turbulent flow correlations reported in previous literature. Jain et al [18] studied a shell and tube absorber with air and LiBr in parallel flow and with cooling water in counter-flow direction, combined with a counter-flow plate regenerator. For both devices, a numerical solution has been obtained using the finite difference method, by dividing their length into a large number of steps. The heat and mass transfer coefficients were obtained using appropriate correlations from their previous studies. Since their goal was to predict the system cooling capacity and energy requirements, they simulated the whole system adopting the Warner's technique, coupled with an optimization routine. They finally produced an optimized design for a 20 TR unit, with a thermal COP 0.33. At the same year, they [19] experimentally studied a counter flow, internally cooled liquid desiccant system using LiBr as the desiccant solution. The absorber is a falling film tubular type, whereas the regenerator consists of parallel iron sheets. By using appropriate correlations available in literature for the heat and mass transfer coefficients, they developed a numerical model that predicts the experimental data within 30%. In 2004, a new regenerator prototype based on a plate heat exchanger design has been developed and tested with a two-dimensional numerical model [20]. In the study of Yoon et al [21], a two-dimensional simultaneous heat and mass transfer numerical model of water cooled vertical plate absorber with LiBr was developed. The model assumed, among others, constant film thickness and upward flow of the cooling water and was solved using the tri-diagonal matrix algorithm. The local heat
transfer coefficient from the liquid to the wall and the local mass transfer coefficient from the air-liquid interface to the wall were also evaluated. It was deducted that the absorption rate increases as the inlet cooling water temperature decreases, the system pressure increases, the inlet solution concentration increases, and the inlet solution temperature decreases. More recently, Lowenstein et al [22] studied and developed a low flow internally cooled liquid desiccant system. The tests confirmed the higher sensible and latent effectiveness of the dehumidifier, the higher COP of the regenerator and elimination of desiccant carryover, compared to adiabatic systems. Ren et al [23] investigated four possible flow arrangements and three types of commonly used liquid desiccants. Their onedimensional analytical model considered the variations of heat and mass transfer coefficients, surface wettability and Lewis factor along the height of the exchanger. Their study however, was confined to the cases were the changes in the solution concentration and the flow rate are relatively small. They found that the flow arrangement with air flowing counter-currently to the fluid and solution streams gives the best performance. At the same year, a two dimensional numerical method was adopted in the work of Chang et al [24] to analyze the performance of a parallel plate cross flow dehumidifier and available experimental results have been used for verification. They found that the moisture removal rate in internally cooled dehumidifier is 20% higher than the adiabatic one under the same operating conditions. The effect of flow pattern, especially the flow direction of air to desiccant on the internally cooled dehumidifier performance is numerically analyzed in their next study [25]. Adopting the heat and mass transfer model by available literature and using experimental results from the literature, the developed model is able to predict the performance of the dehumidifier within 13%. The result shows that counter flow configuration of air to desiccant has better dehumidification performance, while parallel flow performs the poorest. They also proved that the internally cooled dehumidifier has better mass transfer performance compared with the adiabatic plus external heat exchanger, due to less heat transfer process and less desiccant mixing loss in the internally cooled dehumidifier. A more recent research [26] compared the performance of a new plate-fin heat exchanger with an adiabatic dehumidifier / regenerator. They concluded that the cooling efficiency decreases with the increase of the cooling water temperature and that the internally heated regeneration is more efficient than the adiabatic one. The following publication of the same authors [27] presented a uniform mathematical model for a co-flow parallel plate dehumidifier / regenerator, validated by experimental data. They found that

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both internally cooled dehumidifier and regenerator provide better performance comparing with the adiabatic devices.

Most of the previous researches have focused on the performance of a fixed flow pattern of the internally cooled dehumidifier, while less has been done on the performance comparison of various flow patterns and liquid desiccant materials. In the present work, a mathematical model has been produced that permits reliable prediction of the efficiency of an internally cooled liquid desiccant dehumidifier, using two different aqueous desiccant solutions (LiCl, LiBr) and two flow configurations (co-flow and counter-flow). The model is validated by the experimental findings of Mesquita [28, 29] and the main factors that influence the internally cooled dehumidifier's performance are discussed in detail.

6.2.2 Thermodynamic parameters

Since the water partial vapor pressure of the desiccant solution, the heat of absorption released during dehumidification and the heat and mass transfer coefficients are crucial parameters for the dehumidification potential of every liquid desiccant solution, the developed model used relative equations from available literature.

6.2.2.1 Vapor Pressure

Regarding the water partial vapor pressure of the desiccant solution, it determines the humidity ratio of the air that is in equilibrium with the desiccant at the interface. In other words, liquid desiccants with lower water partial vapor pressure have limited ability for water evaporation, so they can absorb more moisture than the desiccants with higher water partial vapor pressure, under the same temperature and concentration. Although the vapor pressure of single electrolyte aqueous solutions has been reported extensively in the literature, only Cisternas and Lam [30, 31] have developed vapor pressure expressions for the two aqueous solutions under investigation following the same experimental procedure. So, the present study uses the vapor pressure expressions from this work to develop the model. The equivalent humidity ratio of liquid desiccant can be calculated through the equilibrium state of air, as shown in the following equation:

$$_{eq. 192} \quad W^{sat}\left(T_{s}\right) = 0.622 \frac{p_{w,st}^{sat}\left(T_{s}\right)}{p_{atm} - p_{w,st}^{sat}\left(T_{s}\right)}$$

6.2.2.2 Heat of absorption

During dehumidification (absorption procedure), water vapor molecules are taken up by the volume of the liquid desiccant while water vapor condensation occurs inside the liquid desiccant solution. The heat of absorption expresses the energy added to the liquid desiccant solution, during the condensation of 1 kg of water vapour and it is given by the following expression [32]:

$$eq. 193 \quad \Delta h_{abs} = h_{st}(p, T_s) - h_{w,s}(T_s, X)$$

Since the absorption heat is not constant, but changes proportionally to the concentration and conversely to the temperature, the above expression was used during the development of the model.

6.2.2.3 Heat & Mass Transfer Coefficients

Correlations for the mass transfer coefficients obtained for internally cooled parallel plate desiccant dehumidifiers are available in the literature. In the present investigation, heat and mass transfer correlations that would be applicable for parallel plates, for various flow configurations and for various liquid desiccant solutions are needed. Therefore, the studies of Jain et al [18, 19], Mesquita [28, 29], Cengel and Boles [33], Incropera et al [34] and Cengel [35] were used to evaluate the heat and mass transfer coefficients.

As such, the water-side heat transfer coefficient is expressed as:

eq. 194
$$a_{W} = \frac{Nu_{W}k_{W}}{D_{h.in}}$$

where the Nusselt number is 7.54 and $D_{h.in}$ is the internal hydraulic diameter of plate

The heat transfer coefficient from the solution to the plate surface is expressed as:

$$a_{S} = \frac{Nu_{S}k_{S}}{D_{h.S}}$$

The Nusselt and Reynolds numbers for the solution are given by the following expressions:

eq. 196
$$Nu_s = 0.680 \cdot \text{Re}_s^{1/2} \cdot \text{Pr}_s^{1/3}$$

eq. 197
$$\operatorname{Re}_{s} = \frac{4 \cdot \dot{m}_{s}}{L \cdot \mu_{s}}$$

The film thickness, necessary for the calculation of D_{hs} is given by the following expression

eq. 198
$$\delta = \left(\frac{3 \cdot v_s^2}{9.81}\right)^{1/3} \cdot \operatorname{Re}_s^{1/3}$$

The heat and mass transfer coefficients between the solution interface and the air stream are expressed as:

eq. 199
$$a_a = \frac{Nu_a k_a}{D_{h.out}}$$

The Nusselt and Reynolds numbers for the air are given by the following expressions:

eq. 200
$$Nu_a = 0.023 \cdot \text{Re}_a^{4/5} \cdot \text{Pr}_a^{1/3}$$

eq. 201
$$\operatorname{Re}_{a} = \frac{V_{a} \cdot D_{h.out}}{\rho_{a} \cdot v_{a}}$$

The overall heat transfer coefficient is given by the following expression:

eq. 202
$$U = \left[\frac{1}{a_w} + \frac{d}{k_p} + \frac{1}{a_s}\right]^{-1}$$

The mass transfer coefficient is expressed as:

$$eq. 203 \quad K_G = \frac{a_a}{C_{p,ma} \cdot Le}$$

6.2.2.4 Performance indices

The mass transfer performance of the dehumidifier is evaluated in terms of the dehumidification mass rate and the absorber efficiency. The dehumidification mass rate (kg_w/h), is calculated by the following expression:

$$eq. 204 \ \dot{m}_{deh} = \dot{m}_{a} (W_{in} - W_{out})$$

The absorber efficiency is the ratio of the actual humidity ratio variance of the air passing through the dehumidifier to its variance under ideal conditions, as shown below:

eq. 205
$$n = \frac{W_{in} - W_{out}}{W_{in} - W^{\text{sat}}} 100\%$$

6.2.3 Mathematical model development

6.2.3.1 Assumptions

The model was developed as an extension of Rogdakis et al. [32], Papaefthimiou [38, 39] and Christodoulaki [40] work. Based on the energy and mass conservation laws, the developed model describes coherently the coupled heat and mass transfer processes taking place inside an internally cooled, parallel plate liquid desiccant dehumidifier.

The dehumidifier height Z is divided into 15 equal segments and the mass and energy balances are solved for each segment, from the top of the dehumidifier Z = 0, until the bottom of the dehumidifier, Z=1; thus, gradients of air temperature, humidity ratio, desiccant temperature, desiccant flow rate and concentration are calculated across the height of the dehumidifier. In order to begin calculations, the following initial parameters should be assumed:

- Temperature of inlet air, inlet desiccant solution and inlet cooling water,
- Flow rate of inlet air, inlet desiccant solution and inlet cooling water,
- Humidity ratio of inlet air,
- Concentration of inlet desiccant solution,
- Physical properties of ambient air, inlet desiccant solution,
- Heat and mass transfer area of dehumidifier.

The developed model was then validated against single channel experimental data obtained from Mesquita [28, 29]. Two different flow configurations were examined co-flow (water downward – desiccant downward – air downward DDD) and counter-flow (water downward – desiccant downward – air upward DDU). The basic geometric characteristics of the absorber test rig are: plate height 0.5m, plate length 1m, plate thickness 4.87mm, wall thickness 0.35mm and distance between two consecutive plates 3.3mm [29]. The schematic diagram of the internally cooled parallel plate dehumidifier is shown in Figure 6-1.



Figure 6-1 Counter flow (DDU) internally cooled parallel plate dehumidifier

In order to simplify the complexity of the configuration, the assumptions used in the calculations are:

- One-dimensional analysis: heat and mass transfer occur only in the flow direction,
- Temperature of the liquid desiccant equal to the temperature of the plates,
- Uniform flooding of the dehumidifier and heat transfer area equal to the mass transfer area; thus, transfer area equal to the specific surface area of the packing,
- Vapor pressure equilibrium between the vapor and the liquid at the interface,
- All heat of absorption is released at the interface,
- Negligible heat transfer from liquid to vapour, negligible heat transfer because of radiation, viscous dissipation, pressure gradients, concentration gradients (Dufour effect) or gravitational effects.

6.2.3.2 Mathematical model

Air - Desiccant interface



Figure 6-2 Air-desiccant interface heat and mass transfer in a counter flow (DDU) internally cooled parallel plate dehumidifier.

By definition, the humidity ratio is given by the following equation:

$$eq.\ 206 \quad W = \frac{\dot{m}_{W}}{\dot{m}_{a}}$$

or after differentiating with dA:

$$eq. \ 207 \ \frac{d\dot{m}_{W}}{dA} = \dot{m}_{a} \ \frac{dW}{dA}$$

During dehumidification however, the moisture that is removed from the air is absorbed by the solution, so:

eq. 208
$$d\dot{m}_S = d\dot{m}_W$$

So, the mass flow rate of the desiccant solution is given by the following expression:

eq. 209
$$\frac{d\dot{m}_s}{dA} = \dot{m}_a \frac{dW}{dA}$$

The mass balance equation for the liquid desiccant solution gives:

$$eq. 210 \ d(\dot{m}_S X) = 0$$

Differentiating this equation over area dA and using eq. 209, the concentration of the solution is obtained:

eq. 211
$$\frac{dX}{dA} = -\frac{\dot{m}_a}{\dot{m}_s} X \frac{dW}{dA}$$

The gradient of humidity ratio along column height expresses the rate of water vapour absorbed by the desiccant at the interface within area dA and can be expressed as follows:

$$eq. 212 \ \frac{dW}{dA} = -\frac{K_G}{\dot{m}_a} \left(W^{\text{sat}} - W_{in} \right)$$

Where W^{sat} is the humidity ratio of the saturated air that is in equilibrium with the desiccant solution at the local solution temperature and concentration.

Air phase



Figure 6-3 Energy balance of the air phase in an internally cooled dehumidifier

The energy balance gives:

$$eq. 213 \qquad \dot{m}_a dh_a = dQ_S + dQ_L$$

The enthalpy of moist air is given by:

eq. 214
$$h_a = C_{p,da}T_a + W(C_{p,st}^{sat}T_a + \Delta h_{abs})$$

The term dQ_s in eq. 213 represents the sensible heat load, which is transferred from the air to the desiccant solution, due to their temperature difference and it can be written as follows:

$$eq. 215 \qquad dQ_S = a_a (T_a - T_s) dA$$

Whereas the latent heat load dQ_L transferred from the air to the desiccant solution due to the water condensation is:

$$eq. 216 \qquad \qquad dQ_L = d\dot{m}_s \left(C_{p,st}^{sat} T_s + \Delta h_{abs}\right)$$

However, the specific heat capacity of the moist air is:

$$eq. 217 C_{p,ma} = C_{p,da} + WC_{p,st}^{sat}$$

Substituting eq. 216 and eq. 217 into eq. 213 and using eq. 209 and eq. 218, the rate of change of air dry bulb temperature is obtained:

eq. 218
$$\frac{dT_a}{dA} = \frac{T_a - T_s}{C_{p,ma}} \left(\frac{a_a}{\dot{m}_a} - C_{p,st} \frac{sat}{dA} \frac{dW}{dA} \right)$$

Liquid desiccant phase



Figure 6-4 Energy balance of the liquid desiccant phase in an internally cooled dehumidifier. Combining eq. 209 with eq. 212 gives the mass of the water absorbed by the solution per unit height:

eq. 219
$$\frac{d\dot{m}_s}{dA} = -K_G \left(W^{sat} - W_{in} \right)$$

The water that flows inside the parallel plates removes the heat generated by the absorption process.

The energy balance gives:

$$eq. 220 \qquad \dot{m}_S dh_S + h_S d\dot{m}_S = dQ_S + dQ_L$$

Also, the enthalpy of the solution is given by the following expression:

$$eq. 221 h_s = -C_{p,s}(T_s - T_{s,in}) \Rightarrow dh_s = C_{p,s}dT_s$$

And the sensible heat that is transferred from the solution to the water stream is:

$$eq. 222 dQ_{S,W} = U(T_S - T_W)dA$$

Differentiating eq. 220 over dA and taking into consideration eq. 209, eq. 212, eq. 213, eq. 218, eq. 221 and eq. 222, the rate of change of desiccant temperature Ts is produced:

$$eq. 223 \qquad \frac{dT_{s}}{dA} = -\frac{1}{\dot{m}_{s}C_{p,s}} \left[\dot{m}_{a} \left[C_{p,s}(T_{s} - T_{s,in}) + C_{p,st} \frac{sat}{T_{s}} + \Delta h_{abs} \right] \frac{dW}{dA} + \right] \\ a_{a}(T_{s} - T_{a}) + U(T_{w} - T_{s})$$

However, the sensible heat that is transferred from the solution to the water stream can be also written as:

$$eq. 224 \qquad dQ_{S,w} = \dot{m}_w C_{p,w} dT_w$$

Using eq. 222 and eq. 224 the rate of change of water temperature Tw is produced:

eq. 225
$$\frac{dT_w}{dA} = \frac{U(T_s - T_w)}{\dot{m}_w C_{p,w}}$$

System of Equations

Finally, a system of six ordinary differential equations is constituted:

$$eq. 226 \begin{cases} \frac{dT_a}{dA} = \frac{T_a - T_s}{C_{p,ma}} \left(\frac{a_a}{\dot{m}_a} - C_{p,st}^{sat} \frac{dW}{dA} \right) \\ \frac{dT_s}{dA} = -\frac{1}{\dot{m}_s C_{p,s}} \left[\frac{\dot{m}_a \left[C_{p,s} (T_s - T_{s,in}) + C_{p,st}^{sat} T_s + \Delta h_{abs} \right] \frac{dW}{dA} + \right] \\ \frac{dT_w}{a_a} \left[\frac{T_s - T_w}{\dot{m}_w C_{p,w}} \right] \\ \frac{dT_w}{dA} = \frac{U(T_s - T_w)}{\dot{m}_w C_{p,w}} \\ \frac{dW}{dA} = -\frac{K_G}{\dot{m}_a} \left(W^{sat} - W_{in} \right) \\ \frac{d\dot{m}_s}{dA} = \dot{m}_a \frac{dW}{dA} \\ \frac{dX}{dA} = -\frac{\dot{m}_a}{\dot{m}_s} X \frac{dW}{dA} \end{cases}$$

with the following boundary conditions:

$$eq. 227 \begin{cases} \left(T_{a}\right)_{z=0} = T_{a,in} \\ \left(T_{s}\right)_{z=L} = T_{s,in} \\ \left(T_{w}\right)_{z=L} = T_{w,in} \end{cases} \\ \begin{pmatrix} W \end{pmatrix}_{z=0} = W_{in} \\ \begin{pmatrix} \dot{m}_{s} \end{pmatrix}_{z=L} = \dot{m}_{s,in} \\ \begin{pmatrix} X \end{pmatrix}_{z=L} = X_{in} \end{cases}$$

The five-order Runge - Kutta was used to numerically integrate the above six ordinary differential equations by using a trial step at the midpoint of an interval to cancel out lowerorder error terms. The model was developed using the Mathcad software [41] where all the thermodynamic characteristics of the dehumidifier are calculated from the point where the liquid desiccant is sprinkled till the end of the packing.

6.2.4 Validation

The developed model has been validated against single channel experimental data obtained from Mesquita [28, 29]. Two different flow configurations were examined; co-current flow (test runs 13, 14 and 15 [29]) and counter flow (test runs 16, 17, 18 [29]). Experimental errors have been taken into consideration.

Figure 6-5 shows the relative deviations of the predicted values from the corresponding experimental values for the six dehumidification experiments for co-current flow and counter flow. In the case of outlet dry bulb temperature, the results of the computational program are almost identical to the experimental ones; the maximum deviation is -4.1% (-0.98 °C). A similar pattern is observed in the case of outlet humidity ratio; the maximum discrepancy is now 4.0% ($0.0003 \text{ kg}_w/\text{kg}_{d\alpha}$). The simulated outlet solution and water temperature differ as much as -11.7% (-2.52 °C) and -6.1% (-1.17 °C) from the experimental values respectively. The predictions of the model agree very well with the experimental findings in the case of outlet solution concentration; an average discrepancy of only ±0.2% ($0.0008 \text{ kg}_{uicl}/\text{kg}_s$) is seen. A similar observation can be made in the case of the water that is evaporated from the liquid desiccant solution; the maximum deviation is -8.2% ($0.000004 \text{ kg}_w/\text{kg}_s$). As it can be observed, the theoretical model shows very good agreement with the experimental results, having an excellent correspondence between most values.



Figure 6-5 Relative deviation between the results of the developed model and the experimental data [28, 29] of the six dehumidification experiments.

Figure 6-6 depicts the experimental data versus the simulation data regarding the air temperature. The experiments constitute of six points with errors on the grey line, whereas the simulation data are depicted as six black dots. It can be observed that the simulation points are very close to the line of experiments, with relative errors well below 5%. Taking into consideration that the uncertainty of the experimental measurements is 0.0003 kg_w/kg_{da} and 0.2°C, the deviation of the theoretical model becomes even less. Figure 6-7 depicts the experimental data versus the simulation data regarding the solution concentration. Again, the experiments constitute of six points with errors on the grey line, whereas the simulation data are depicted as six black dots. The simulation points are almost identical to the experiments, with relative errors being well below 1%. Considering though the experimental errors (0.2°C and 0.001 kg_{LICI}/kg_s), some the predicted values may lie even closer to the real ones.



Figure 6-6 Air temperature experimental runs [28,29] versus simulation results, at the outlet of the regenerator.



Figure 6-7 Solution concentration experimental runs [28,29] versus simulation results, at the outlet of the regenerator

6.2.5 Parametric analysis

6.2.5.1 **Overview**

Following the validation of the model with the experimental results, an extensive theoretical investigation was conducted to examine the effect of various parameters on the dehumidifier efficiency. The parametric study included two flow configurations (DDD and DDU) and the two most common liquid desiccant solutions; LiCl-H₂O and LiBr-H₂O. Table 6-1 provides all the cases considered. The necessary thermodynamic properties were taken from Conde [36] regarding LiCl and from McNeely [37] regarding LiBr.

A much better understanding though of the difference in the dehumidification potential of the three salt solutions would be the study of the distribution of the air humidity ratio along the absorber's height. Fortunately, the developed computational program not only is able to predict the outlet values, but it can also produce precise estimations of these values along the height of the absorber. So, the study of the distribution of the air humidity ratio along the absorber's height was implemented with the starting values: $T_{a,in}=26$ °C, $W_{a,in}=0.012$ kgw/kg_{da}, $\dot{m}_a=0.012$ kg/sec, $T_{s,in}=22$ °C, $\dot{m}_s=0.0003$ kg/sec, X= 40%, $T_{w,in}=20$ °C and $\dot{m}_w=0.32$ kg/sec.

	Inlet values for the parametric analysis							
	m _a , kg/s	m _s , kg/s	m _w , kg/s	T _{a,in} , ^o C	T _{s,in} , °C	T _{w,in} , ^o C	W _{a,in} , kg/kg	Х
Air mass flow rate	0.004 0.008 0.012	0.0003	0.32	26	22	20	0.012	0.4
Solution mass flow rate	0.012	0.0003 0.0006 0.0009	0.32	26	22	20	0.012	0.4
Water mass flow rate	0.012	0.0003	0.3 0.31 0.32	26	22	20	0.012	0.4
Air temperature	0.012	0.0003	0.32	24 26 28	22	20	0.012	0.4
Solution temperature	0.012	0.0003	0.32	26	22 24 26	20	0.012	0.4
Water temperature	0.012	0.0003	0.32	26	22	18 19 20	0.012	0.4
Humidity ratio	0.012	0.0003	0.32	26	22	20	0.011 0.0115 0.012	0.4
Solution concentration	0.012	0.0003	0.32	26	22	20	0.012	0.4 0.405 0.41

Table 6-1 Cases considered for the parametric analysis of LiCl-H₂O and LiBr-H₂O, in both DDD and DDU flow configurations.

6.2.5.2 Air inlet humidity ratio

Figure 6-8 shows that increasing air inlet humidity ratio by 9.1%, increases the dehumidification mass rate of LiCl by 12%, and of LiBr by 32.6%. This happens because a higher humidity ratio implies a higher air vapor pressure and thus, a higher average water vapor pressure difference between the air and the desiccant. Consequently, higher potential for mass transfer from the gas to the liquid phase is established. The absorber efficiency shows a slight change, with that of LiCl being at -0.5% and of LiBr at -6.5%.



Figure 6-8 Effect of inlet humidity ratio on the dehumidification efficiency and the dehumidification mass rate of LiCI-H₂O and LiBr-H₂O, under co-current flow

6.2.5.3 Air flow rate

For both desiccant solutions examined, increasing the air flow rate by 200% increases the dehumidification mass rate by approximately 120%, while decreases the absorber efficiency by almost 30%. This is happening because when the air flow rate is increased, the residence time for the air in the dehumidifier is decreased, resulting in a decrease of the mass transfer coefficient, a higher outlet air humidity ratio and thus, a lower absorber efficiency. At the same time, higher outlet humidification mass rate. So, during the low air flow rate range, more pressure and higher dehumidification mass rate. So, during the low air flow rate range, more air flow rate can achieve better dehumidification performance. Especially for LiCl, the efficiency is reduced by -23% in co-flow and by -18% in counter-flow, while the dehumidification mass rate is increased by 131% in co-flow and by 143% in counter-flow (Figure 6-9). As also shown in Figure 6-9, counter flow configuration tends to show higher efficiency and dehumidification mass rate, under air flow rates greater than 0.0012 kg/s.



Figure 6-9 Effect of inlet air flow rate on the dehumidification efficiency and the dehumidification mass rate of LiCl-H₂O.

6.2.5.4 Inlet water temperature

Figure 6-10 shows that the increase of the cooling water temperature by 11% results in a slight decrease in the dehumidification efficiency (5%) and a greater decrease in the dehumidification mass rate, for both solutions. This happens because as the inlet cooling water increases, the heat transfer rate from the liquid film to cooling water decreases, hence the saturation pressure increases. This causes a decrease in the vapor pressure driving force and thus, in a decrease in the mass transfer rate. LiBr is subject to greater dehumidification mass rate changes (-18%), whereas LiCl seems more stable (-5%).



Figure 6-10 Effect of inlet water temperature on the dehumidification efficiency and the dehumidification mass rate of LiCl-H₂O and LiBr-H₂O, under co-current flow.

6.2.5.5 **Desiccant inlet temperature**

As indicated by Figure 6-11, increasing desiccant inlet temperature by 18.2% decreases the dehumidification mass rate, by 0.1% in the case of LiCl and by 59.2% in LiBr. This happens because the increase in desiccant temperature increases the water vapour pressure of the desiccant and so, the average water vapor pressure difference between the air and desiccant in the dehumidifier is decreased. This leads to a higher air outlet humidity ratio and consequently, to a lower dehumidification mass rate. However, increasing desiccant inlet temperature increases the dehumidifier efficiency by 12.1% for LiCl and by 33.9% for LiBr. Figure 6-11 also indicates that, under the three temperatures considered, LiBr may have higher efficiency than LiCl, but it can remove about 0.1 kg_w/h less than the LiCl. Both the dehumidification efficiency and the mass rate of LiBr is subject to greater changes compared to LiCl, due to the lower water vapor pressure of LiBr.



Figure 6-11 Effect of inlet solution temperature on the dehumidification efficiency and the dehumidification mass rate of LiCl-H₂O and LiBr-H₂O, under co-current flow.

6.2.5.6 **Desiccant flow rate**

The dehumidification mass rate increased with the desiccant flow rate (Figure 6-12), by 8.4% for co-flow and by 4.2% for counter-flow, in the case of LiCl. This is happening because increasing the desiccant flow rate decreases the contact time between the desiccant and the air and so, decreases the variation of the desiccant concentration and temperature through the dehumidifier. As a result, the variation of the surface vapor pressure of the desiccant through the dehumidifier is decreased and, hence, the average water vapor pressure difference between the desiccant and air in the dehumidifier is increased. This leads to a lower air outlet humidity ratio and a higher dehumidification mass rate.

The absorber efficiency increased with a similar rate, due to the lower air outlet humidity ratio. Also, increasing the desiccant flow rate increases the mass transfer coefficient between the desiccant and the air in the dehumidifier, resulting in a further increase in the dehumidification efficiency.



Figure 6-12 Effect of inlet solution flow rate on the dehumidification efficiency and the dehumidification mass rate of $LiCI-H_2O$

6.2.5.7 Liquid desiccant solution

The question is which liquid desiccant solution has better performance under the same operating conditions. In this work, some effort was made to compare two salt solutions, namely LiCl and LiBr, against their dehumidification potential along the absorber's height. Indeed, the humidity ratio distribution (Figure 6-13) confirms that the co-flow absorber using LiCl has the best efficiency, as it can reduce humidity ratio by 31%, while the absorber using LiBr reduces humidity ratio by only 12%, under the same inlet conditions. The numerical study has further shown that counter-flow configuration produces a much drier air than co-flow, since counter flow can reduce humidity ratio by 44% when co-flow reduces it by 31%.



Figure 6-13 Humidity ratio distributions along the dehumidifier, for LiCl-H₂O and LiBr-H₂O, under cocurrent flow

6.3 LIQUID DESICCANT EXPERIMENTAL UNIT

6.3.1 **Description of the experimental unit**

A liquid desiccant system has been installed at the National Technical University of Athens, Laboratory of Applied Thermodynamics, for experimental purposes. The liquid desiccant system has been constructed by the German company L-DCS [42]. The main components of the system are the dehumidifier, the regenerator and the evaporative cooler. The system uses water as the cooling medium and LiCl aqueous solution as the desiccant. It also employs two storage tanks, one for the concentrated solution and one for the diluted.

Inside the counter flow dehumidifier (absorber), the strong desiccant solution (LiCl-H₂O), coming from the strong solution tank, falls from the top of the device. The ambient humid air enters the dehumidifier at the bottom and moisture is transferred from the air to the desiccant. However, as the water vapor condenses and mixes with the desiccant, heat is released. This heat equals to the latent heat of condensation for water plus the chemical heat of mixing between the desiccant and water. This heat would raise the temperature of the desiccant, decreasing its ability to remove water vapor from the air. For this reason, the absorption process is cooled by water flowing inside polypropylene sheets and this heat is immediately absorbed. The cooling water restricts the temperature changes for both the desiccant and air and therefore, increases the dehumidification capacity. The end of the process finds the air dehumidified and cooled, whereas the desiccant solution diluted and stored in the weak solution tank.

The role of the indirect evaporative cooler is to supply the necessary cooling water during absorption. To achieve this, the evaporative cooler uses the exhaust air from the building and mains water. Due to this configuration, the lower theoretical temperature limit for the cooling water produced by the evaporative cooler is the wet bulb temperature of the exhaust air from the building.

After dehumidification, the diluted desiccant has to regain its initial concentration. It therefore flows into the regenerator (desorber), where it falls from the top. Gravity forces it to flow downwards, contacting the ambient air that moves upwards, through the tubes. At the same time, heating water is circulated through the polypropylene sheets and supplies the necessary amount of heat. As the liquid desiccant is heated, water vapor is driven out of the weak solution and carried away by the ambient air stream. At the outlet of the regenerator, the desiccant has gained its initial concentration and it is stored to the strong tank, ready to re-enter the dehumidifier.

The two solution tanks, weak (diluted solution) and strong (concentrated solution), serve to make the dehumidification and regeneration processes independent from each other. The necessary energy for air conditioning can be also stored over any desired period, in the form of concentrated salt solution without any energy losses.

Since the desiccant solution circulates in an open cycle, all desiccant containers could theoretically overflow. To prevent this from happening, L-DCS has installed emergency overflow switches in all containers and tanks of the system. These mechanical overflow switches are operated by a rod, mounted to a float, floating on the surface of each salt solution container. Should the desiccant level exceed the permitted maximum level within a container, the switch is triggered, the power supply for the subsystem gets interrupted and all pump circuits are switched off.



Figure 6-14 Photo of the liquid desiccant L-DCS experimental apparatus installed in the laboratory. Left: Dehumidifier. Right: Regenerator, diluted solution tank, concentrated solution tank



Figure 6-15 Scheme of the liquid desiccant L-DCS experimental unit installed in the laboratory

6.3.1.1 **Dehumidifier**

The following photo shows the main components of the dehumidifier; packing material, water circuit, solution circuit and provisional tanks for the strong and weak solution.



Figure 6-16 Photo of the L-DCS dehumidifier

The packing material of the dehumidifier consists of 60 polypropylene black sheets with horizontal ducts for the water flow and 62 white sheets with vertical ducts for the solution and air flow. The length, depth and height are 2.3*1000*970 mm for the black sheets and 3.3*1000*970mm for the white sheets. Each black sheet has 391 ducts of 2.2*2.2 mm and each white sheet has 312 ducts of 2.4*2.4mm. The geometry of the evaporative cooler is

exactly the same as the dehumidifier, while the regenerator has 25 black sheets and 24 white sheets.



Figure 6-17 Photo of the PPP sheets as packing material inside the dehumidifier



Figure 6-18 Drawing of the PPP sheets inside the dehumidifier

6.3.1.2 Regenerator

The following photo shows the main components of the regenerator; packing material, water circuit, solution circuit and provisional tanks for the strong and weak solution. The regenerator consists of 25 polypropylene black sheets with horizontal ducts for the water flow and 24 white sheets with vertical ducts for the solution and air flow.



Figure 6-19 Photo of the L-DCS regenerator

6.3.1.3 Evaporative cooler

The role of the indirect evaporative cooler is to supply the necessary cooling water during absorption. To achieve this, the evaporative cooler uses room air from the building, the closed water circuit (input – output of the dehumidifier) and the open water circuit (mains). Due to this configuration, the lower theoretical temperature limit for the cooling water produced by the evaporative cooler is the wet bulb temperature of the room air. The open water circuit (mains) is disinfected by a UV light and the entire system is built from corrosion-resistant germ-free material. The packing geometry of the evaporative cooler is exactly the same as the dehumidifier.

6.3.1.4 Heat source

The necessary hot water for the regenerator is supplied from a 24kW electric hot water boiler installed in the laboratory. The following photo shows the configuration of the boiler.



Figure 6-20 Photo of the heat source for the regenerator

6.3.2 Monitoring system

The monitoring system consists of temperature sensors, relative humidity sensors, differential pressure sensors and flow meters. They are installed in the most representative areas, both at the input and at the output of each circuit. The following table summarizes the installed sensors.

Dehumidifier device							
Sensor type	Name	Description					
Kimo TH100-ANS	T_abs_air_in	Temperature of air input					
Kimo TH100-ANS T_abs_air_out		Temperature of air output					
Kimo TH100-ANS	RH_abs_out	Relative Humidity of air output					
Kimo TH100-ANS	RH_abs_in	Relative Humidity of air input					
Kimo SF GT 50	T_abs_ conc	Temperature of concentrated solution tank					
Kimo SF GT 50	T_abs_ dil	Temperature of diluted solution tank					
Elster EQOS - Single let	T_abs_ water_in	Temperature of cooling water input					
Meter	T_abs_ water_out	Temperature of cooling water output					
WELEI	F_abs_water	Mass flow rate of the cooling water					
Kimo SFE 50	T_abs_ water_out	Temperature of cooling water output					
Honeywell PXL 0025 DN	ΔP_abs_air	Overall air pressure drop in the absorber					
ZIEHL-ABEGG, DSG 500	ΔP_abs_pack	Air pressure drop at packing					
Endress+Hauser, Cerabar TPMC131	P_abs_water_bot	Water pressure drop at bottom					
Endress+Hauser, Cerabar	P_abs_water_top	Water pressure drop at top					
Regenerator device							
Sensor type	Name	Description					
Kimo TH100-AND	T reg air in	Temperature of air input					
Kimo TH100-AND	T reg air out 1	Temperature of air output					
Kimo SFE 50	T reg air out 2	Temperature of air output					
Kimo TH100-AND	RH_reg_in	Relative Humidity of air input					
Kimo TH100-AND	RH_reg_out	Relative Humidity of air output					
Kimo SF GT 50	T_reg_ conc	Temperature of concentrated solution tank					
Kimo SF GT 50	T_reg_dil	Temperature of diluted solution tank					
Elatar 5000 Circle Lat	T_reg_ water_in	Temperature of hot water input					
Eister F905 – Single Jet	T_reg_ water_out	Temperature of hot water output					
weter	F_reg_water	Mass flow rate of the hot water					
Kimo SFE 50	T_reg_ water_out	Temperature of hot water output					
Honeywell DCXL 10 DS	ΔP_reg_air	Overall pressure drop in the regenerator					
ZIEHL-ABEGG, DSG 500	ΔP_reg_pack	Air pressure drop at packing					
Endress+Hauser, Cerabar TPMC131	P_reg_water_bot	Water pressure drop at bottom					
Endress+Hauser, Cerabar TPMC131	P_reg_water_top	Water pressure drop at top					
L-DCS Unit							
Sensor type Name		Description					
Honeywell SCX 15AN	P amb	ambient pressure					

Table 6-2 List of sensors installed

The positioning of the sensors inside the dehumidifier is shown in the following scheme.



Figure 6-21 Monitoring system for the dehumidifier

The positioning of the sensors inside the regenerator is shown in the following scheme.



Regenerator

Figure 6-22 Monitoring system for the regenerator

6.3.2.1 Calibration



Figure 6-23 Photos during the calibration procedure

The calibration of the temperature sensors was implemented using the FLUKE 9142 device. All temperature sensors were calibrated against three different temperature values (reference values): 15 °C 30 °C, 45 °C and 70 °C. The sensors showed little deviation from the reference value, as it can be seen in the following figures.



Figure 6-24 Calibration results for the air temperature sensors



Figure 6-25 Calibration results for the desiccant solution temperature sensors



Figure 6-26 Calibration results for the water temperature sensors

6.3.3 Plausibility check

This section provides information on the method followed for each dehumidification and regeneration experiment as well as it describes the plausibility check for one indicative experiment, both for dehumidification and regeneration process.

6.3.3.1 **Dehumidification**

The steps followed for one dehumidification experiment are described below:

- 1. Check the time, then
- 2. Turn on the general switch of the dehumidifier and the evaporative cooler, then
- 3. Turn on the fan switch of the dehumidifier and the evaporative cooler, then
- 4. Turn on the water pump switch of the dehumidifier and the evaporative cooler, then
- 5. Open the strong solution tank, stir the solution and measure the initial solution density, then
- 6. Open the weak solution tank, stir the solution and measure the initial solution density, then
- 7. Turn on the solution pump switch of the dehumidifier (this step should be initiated at least 30 minutes after step 4), then
- 8. Wait until the strong solution wears out (the solution pump switch is turned off automatically), then
- 9. Turn off the water pump switch of the dehumidifier and the evaporative cooler, then
- 10. Turn off the fan switch of the dehumidifier and the evaporative cooler, then
- Open the strong solution tank, stir the leftover solution and measure the final solution density, then
- 12. Open the weak solution tank, stir the solution and measure the final solution density.

The plausibility check is implemented for one representative day, which is 16/07/2013. Due to restrictions in graph formatting, time is shown in units 1000-1800, which correspond to 08:14-14:55. The following graphs show the distribution of temperatures, humidity ratio and loads.



Figure 6-27 Distribution of air temperature during 16/07/2013 dehumidification experiment







Figure 6-29 Distribution of solution temperature during 16/07/2013 dehumidification experiment



Figure 6-30 Distribution of humidity ratio during 16/07/2013 dehumidification experiment According to that day, the state of the air coming in is approximately 28°C and 0.008kg/kg, while the air coming out is 19°C and 0.002kg/kg (Figure 6-27 and Figure 6-30). Therefore, the air is cooled and dehumidified. Namely, 32.73 kg of water were removed during that experiment. The water coming in is 15 °C (Figure 6-28) and this is well justified by the wet bulb temperature of the room air. The only 1 °C temperature change of the solution (Figure 6-29) owns to its contact with the cooling water.

Figure 6-31 depicts the heat transfer process during dehumidification. The solution seems that does not have large effect in the heat transfer process (0.84 kWh), due to its low flow and slight temperature change. In the contrary, a large amount of heat is removed from the air (-37.41 kWh) which is transferred to the cooling water (38.72 kWh). The energy balance shows a deviation of 5.44%.


Figure 6-31 Distribution of all loads during 16/07/2013 dehumidification experiment

6.3.3.2 Regeneration

The steps followed for one dehumidification experiment are described below:

- 13. Check the time, then
- 14. Turn on the general switch of the regenerator and the hot water boiler, then
- 15. Turn on the resistances 1 and 2 of the hot water boiler and choose the desired hot water temperature, then
- 16. Turn on the fan switch of the regenerator, then
- 17. Turn on the water pump switch of the regenerator, then
- Open the strong solution tank, stir the solution and measure the initial solution density, then
- Open the weak solution tank, stir the solution and measure the initial solution density, then
- 20. Turn on the solution pump switch of the regenerator (this step should be initiated at least 30 minutes after step 5), then
- 21. Wait until the diluted solution wears out (the solution pump switch is turned off automatically), then
- 22. Turn off the water pump switch of the regenerator, then
- 23. Turn off the fan switch of the regenerator, then
- 24. Open the weak solution tank, stir the leftover solution and measure the final solution density, then
- 25. Open the strong solution tank, stir the solution and measure the final solution density.



Figure 6-32 Distribution of air temperature during 17/07/2013 regeneration experiment





Figure 6-33 Distribution of water temperature during 17/07/2013 regeneration experiment

Figure 6-34 Distribution of solution temperature during 17/07/2013 regeneration experiment



Figure 6-35 Distribution of humidity ratio during 17/07/2013 regeneration experiment

The plausibility check is implemented for the representative day of 17/07/2013. Due to restrictions in graph formatting, time is shown in units 1000-1600, which correspond to 08:14-13:15.

According to that day, the state of the air coming in is approximately 43°C and 0.01 kg/kg, while the air coming out is 50°C and 0.07 kg/kg (Figure 6-32 and Figure 6-35). Therefore, the air is heated and humidified. Namely, 33.28 kg of water were added to the air during that experiment. The hot water coming in ranges from 65 to 70 °C (Figure 6-33), fluctuating due to the absence of differential thermostat in the boiler. A corresponding fluctuation is observed in the hot water output. The solution coming in is at 29 °C but due to its contact with the hot water and its low flow, it comes out at 42 °C (Figure 6-34).

Figure 6-36 depicts the heat transfer process during regeneration. The solution seems that does not have large effect in the heat transfer process (0.77 kWh), due to its low flow conditions. In the contrary, a large amount of heat is removed from the hot water (-26.22 kWh) which is transferred to the air (25.21 kWh). The energy balance shows a deviation of - 0.9%.



Figure 6-36 Distribution of loads during 17/07/2013 regeneration experiment

6.3.4 Experimental measurements

This section presents the results of all experimental measurements carried out, both for dehumidification and regeneration.

6.3.4.1 **Dehumidification**

This section could include graphs for the distribution of air temperature, water temperature, solution temperature, humidity ratio and loads during each of the 8 experiments; however for space saving purposes, it was decided to analyze and comment on only one experiment and to summarize the average values of all experiments in the Table 6-3.

Since a summer experiment has already been discussed (16/7/2013), a winter experiment, for example the one in 22/2/2014, could be worth of analysis.





Figure 6-37 Distribution of air temperature during 22/02/2014 dehumidification experiment

Figure 6-38 Distribution of water temperature during 22/02/2014 dehumidification experiment



Figure 6-39 Distribution of solution temperature during 22/02/2014 dehumidification experiment



Figure 6-40 Distribution of humidity ratio during 22/02/2014 dehumidification experiment



Figure 6-41 Distribution of all loads during 22/02/2014 dehumidification experiment

According to the experimental results, the state of the air coming in, after the activation of the solution pump, is approximately 16°C and 0.006kg/kg, while the air coming out is 14°C and 0.002kg/kg (Figure 6-37 and Figure 6-40). Therefore, the air is cooled and dehumidified. Namely, 14.88 kg of water were removed during that experiment. The water coming in is 12°C (Figure 6-38Figure 6-28) and this is well justified by the wet bulb temperature of the room air. The 5 °C temperature change of the solution (Figure 6-39) owns to its contact with the cooling water.

Figure 6-41 depicts the heat transfer process during dehumidification. The solution seems that does not have large effect in the heat transfer process (-0.12 kWh), due to its low flow and slight temperature change. In the contrary, a large amount of heat is removed from the air (-12.59 kWh) which is transferred to the cooling water (13.38 kWh). The energy balance shows a deviation of 5.02%.

		AIR								WATER					DESICCANT SOLUTION							Deviation
Date	duration	T_in	T_out	W_in	W_out	Mass flow	Load	Energy	T_in	T_out	Mass flow	Load	Energy	T_in	T_out	C_in	C_out	Mass flow	Load	Energy	water	in Energy Balance
	hr	°C	°C	kg _{H2O} / kg _{dry air}	kg _{H2O} / kg _{dry air}	kg/hr	kW	kWh	°C	°C	kg/hr	kW	kWh	°C	°C	%	%	kg/hr	kW	kWh	kg/hr	%
16/7/2013	3.19	28.16	18.99	0.0084	0.0025	1723	-11.72	-37.41	15.22	25.87	981	12.13	38.72	29.83	28.86	0.41	0.34	49.90	0.26	0.84	10.26	5.44%
24/7/2013	2.63	29.01	20.00	0.0065	0.0032	2913	-14.32	-37.60	14.08	26.14	997	13.96	36.64	27.40	28.25	0.41	0.34	50.00	0.31	0.82	9.76	-0.38%
19/2/2014	1.58	16.08	13.58	0.0063	0.0020	1619	-5.99	-9.48	11.54	19.29	844	7.58	12.00	23.94	19.09	0.41	0.37	65.10	-0.11	-0.17	6.87	19.84%
20/2/2014	1.67	16.26	14.54	0.0065	0.0024	2261	-7.75	-12.92	11.88	20.95	850	8.95	14.91	22.74	18.82	0.41	0.36	65.10	-0.01	-0.02	9.43	13.18%
21/2/2014	1.74	17.35	16.65	0.0068	0.0026	2846	-8.97	-15.63	13.03	21.97	854	8.85	15.42	24.66	19.80	0.41	0.35	65.10	0.00	-0.01	11.90	-1.41%
22/2/2014	1.68	16.62	14.66	0.0063	0.0024	2255	-7.52	-12.59	12.47	20.49	858	7.99	13.38	23.61	18.78	0.38	0.33	64.00	-0.07	-0.12	8.89	5.02%
23/2/2014	2.08	16.14	14.42	0.0061	0.0029	2261	-6.18	-12.88	12.47	18.41	861	5.94	12.38	21.77	17.74	0.35	0.31	62.80	-0.07	-0.15	7.21	-5.36%
27/2/2014	1.00	14.60	14.43	0.0054	0.0023	2283	-5.06	-5.06	12.14	18.23	839	5.92	5.92	22.66	17.90	0.41	0.38	97.30	-0.22	-0.22	6.99	11.19%

Table 6-3 List of dehumidification experiments

6.3.4.2 Regeneration

This section could include graphs for the distribution of air temperature, water temperature, solution temperature, humidity ratio and loads during each of the 12 experiments; however for space saving purposes, it was decided to analyze and comment on only one experiment and to summarize the average values of all experiments in Table 6-4.

Since a summer experiment has already been discussed (17/7/2013), a winter experiment, for example the one in 20/02/2014, could be worth of analysis.



Figure 6-42 Distribution of air temperature during 20/02/2014 regeneration experiment



Figure 6-43 Distribution of water temperature during 20/02/2014 regeneration experiment





Figure 6-44 Distribution of solution temperature during 20/02/2014 regeneration experiment

Figure 6-45 Distribution of humidity ratio during 20/02/2014 regeneration experiment

According to the experimental results, the state of the air coming in is approximately 44°C and 0.008 kg/kg, while the air coming out is 50°C and 0.07 kg/kg (Figure 6-42 and Figure 6-45). Therefore, the air is heated and humidified. Namely, 28.49 kg of water were added to the air during that experiment. The hot water coming in ranges from 72 to 70 °C (Figure 6-43) fluctuating due to the absence of differential thermostat in the boiler. A corresponding fluctuation is observed in the hot water output. The solution coming in is at 22 °C but due to its contact with the hot water and its low flow, it comes out at 41 °C (Figure 6-44).

Figure 6-46 depicts the heat transfer process during regeneration. The solution seems that does not have large effect in the heat transfer process (1.19 kWh), due to its low flow conditions. In the contrary, a large amount of heat is removed from the hot water (-21.54 kWh) which is transferred to the air (21.30 kWh). The energy balance shows a deviation of 4.22%.



Figure 6-46 Distribution of all loads during 20/02/2014 regeneration experiment

			-	-	AIR	_				_	WATE	R				DESIC		OLUTIO	N			Doviation
Date	duration	T_in	T_out	W_in	W_out	Mass flow	Load	Energy	T_in	T_out	Mass flow	Load	Energy	T_in	T_out	C_in	C_out	Mass flow	Load	Energy	ΔM water	in Energy Balance
	hr	°C	°C	kg _{H2O} / kg _{dry air}	kg _{H2O} / kg _{dry air}	kg/hr	kW	kWh	°C	°C	kg/hr	kW	kWh	°C	°C	%	%	kg/hr	kW	kWh	kg/hr	%
17/7/2013	3.36	43.31	50.65	0.0107	0.0692	170	7.51	25.21	67.00	60.12	978	-7.81	-26.22	29.02	40.67	34%	40%	66.00	0.23	0.77	-9.91	-0.90%
19/7/2013	2.92	41.02	46.56	0.0093	0.0538	212	7.10	20.72	63.93	57.19	949	-7.42	-21.64	27.72	38.57	33%	39%	65.66	0.23	0.66	-9.40	-1.22%
25/7/2013	3.18	39.05	44.12	0.0087	0.0392	209	4.88	15.53	58.63	54.02	960	-5.14	-16.35	27.95	36.16	35%	39%	66.42	0.21	0.68	-6.37	-0.89%
18/2/2014	1.71	39.64	47.92	0.0082	0.0512	172	5.72	9.78	66.65	61.02	965	-6.30	-10.77	22.58	37.59	37%	42%	67.45	0.53	0.91	-7.36	-0.80%
19/2/2014	2.17	39.92	46.01	0.0088	0.0472	172	5.03	10.91	62.19	57.03	968	-5.80	-12.56	22.13	37.85	36%	40%	66.99	0.60	1.30	-6.58	-2.90%
20/2/2014	2.33	44.47	49.60	0.0087	0.0658	214	9.13	21.30	72.41	64.27	978	-9.23	-21.54	22.47	41.17	34%	42%	66.18	0.51	1.19	-12.21	4.22%
21/2/2014	1.92	41.37	45.84	0.0090	0.0509	214	6.71	12.87	65.15	58.52	965	-7.42	-14.23	22.63	37.95	32%	37%	65.47	0.49	0.94	-8.94	-3.02%
22/2/2014	2.25	32.63	36.61	0.0081	0.0248	214	2.78	6.26	46.11	43.39	978	-3.08	-6.93	20.85	30.20	32%	34%	65.45	0.40	0.89	-3.55	3.18%
23/2/2014	1.25	44.71	46.29	0.0092	0.0537	217	7.02	8.77	72.76	66.78	999	-6.93	-8.66	22.99	44.14	32%	38%	65.61	0.73	0.92	-9.58	10.57%
27/2/2014	1.58	42.05	47.70	0.0075	0.0539	214	7.49	11.85	66.30	59.69	980	-7.52	-11.90	21.71	39.23	35%	41%	66.70	0.56	0.89	-9.89	6.57%
4/3/2014	2.08	40.69	46.51	0.0073	0.0490	173	5.47	11.40	64.59	59.06	976	-6.26	-13.04	21.16	38.62	36%	40%	67.01	0.67	1.39	-7.19	-1.99%
17/3/2014	2.33	43.22	48.64	0.0071	0.0607	216	8.66	20.22	72.86	65.13	990	-8.87	-20.70	22.17	39.61	34%	41%	66.19	0.49	1.14	-11.55	3.07%

Table 6-4 List of regeneration experiments

6.3.5 **Propositions for optimization**

The following list provides some propositions for optimizing the experimental procedure:

- The dehumidification experiment should begin two days after the regeneration experiment, in order to give some time to the strong solution to cool down. If a dehumidification experiment begins with warm strong solution, then the dehumidification rate and air cooling effect will not be optimal.
- A differential thermostat should have been installed in the hot water boiler in order to produce a steady desired temperature without fluctuations.
- An intermediate heat exchanger solution solution could have been installed between the dehumidifier and the regenerator, in order to accelerate the cooling of the strong solution exiting the regenerator.
- The fan inverter, both at the dehumidifier and the regenerator, caused interference at the energy sensor which produced inaccurate data. This was overcome by the relocation of the energy sensor away from the inverter. Another solution would be to insulate electrically the cables against such signals.

6.4 COMPARISON BETWEEN THEORETICAL & EXPERIMENTAL RESULTS

The accuracy of the developed theoretical model can be best described through the comparison of the experimental data with the corresponding results of the model, obtained both for the dehumidification and the regeneration experiments.

For consolidation purposes, the initial conditions used as inputs to the modeling calculations and the initial conditions of the experiment should be common. To this end, the inputs used in the mathematical model are the same as those applied in the experiments. The initial conditions can be divided into two main categories; the physical characteristics of the device and the initial values of the parameters monitored.

The physical characteristics of the dehumidifier and regenerator device, previously presented in the subchapter, are used as inputs to the mathematical model. The geometry of each device is crucial for the accurate calculation of the heat and mass transfer coefficients (eq. 202 and eq. 203).

The initial values of the monitored parameters are the following:

- Inlet air temperature
- Inlet air humidity ratio
- Inlet air volume flow
- Inlet cooling or heating water temperature
- Inlet cooling or heating water volume flow
- Inlet desiccant temperature
- Inlet desiccant concentration
- Inlet desiccant volume flow

The above values, used as inputs to the mathematical model, can be seen in the Table 6-3 List of dehumidification experiments and Table 6-4 List of regeneration experiments.

Following the adaptation of the mathematical model to the specific initial conditions of each dehumidification and regeneration experiment, the comparison between the experimental and the theoretical results can be realized.

The comparison is applied on the output values of each process. More specifically,

- Outlet air temperature
- Outlet air humidity ratio
- Outlet cooling or heating water temperature
- Outlet desiccant temperature
- Outlet desiccant concentration

The base of the comparisons could not be other than the outlet experimental values, taken from the Table 6-3 List of dehumidification experiments and Table 6-4 List of regeneration experiments. This happens because the outlet experimental values, in comparison to the outlet modeled values, are more close to the real conditions. As such, the comparison is quantified in terms of the deviation percentage, which is expressed by the following equation:

eq. 228 Deviation = $\frac{\text{experimental value} - \text{theoretical value}}{\text{experimental value}}\%$

Obviously, lower deviation values indicate that the results of the mathematical model are closer to the experimental data and that therefore, the developed model can more accurately predict the real conditions.

6.4.1 **Dehumidification**

The mathematical model was adapted to the physical characteristics of the dehumidifier installed at the Laboratory of Applied Thermodynamics, in the National Technical University of Athens. Namely, the geometrical inputs to the model were taken from Figure 6-18 Drawing of the PPP sheets inside the dehumidifier. The inlet values of the monitored parameters had to be the same with the experiments, so they were taken from Table 6-3 List of dehumidification experiments. Thus, the input conditions to the model were the same as the initial conditions of the experiments. There were 8 dehumidification experiments, so 8 simulations were performed and produced the theoretical results.

The results showed a reasonable deviation between the theoretical and the experimental results, with all deviations being in the range of -17.39% to 14.51%.

In the case of the air outlet temperature (Figure 6-47), the deviation ranges from -1.95% to 11.07% for all experiments carried out. Fair difference among the theoretical and experimental results exists in the case of outlet humidity ratio (Figure 6-48), where the deviations start from -3.12% and reach -17.39%. It could be said that the experimental apparatus produced drier but hotter air than expected.



Figure 6-47 Graphical representation of the deviation in outlet air temperature during dehumidification



Figure 6-48 Graphical representation of the deviation in outlet humidity ratio during dehumidification

The theoretical desiccant outlet temperature (Figure 6-49) produces satisfactory results, since its deviation with the experimental data ranges from -1.34% to 14.51%. Reasonable deviation can be also seen in the desiccant solution outlet concentration (Figure 6-50), with the minimum being -1.32% and the maximum 14.50%. The experimental apparatus produced hotter and weaker desiccant than expected.



Figure 6-49 Graphical representation of the deviation in outlet desiccant temperature during dehumidification



Figure 6-50 Graphical representation of the deviation in outlet desiccant concentration during dehumidification

As far as the water loop is concerned (Figure 6-51), the deviation of the outlet temperature shows fair results, with the minimum being -0.43% and the maximum 12.27%. The experimental apparatus produced hotter water than expected.



Figure 6-51 Graphical representation of the deviation in outlet water temperature during dehumidification

Table 6-5 shows the comparison between the theoretical and experimental results of the air, water and desiccant loop during the dehumidification process. The percentage deviation is shown in bold letters.

The results for the air loop are also shown in a psychrometric chart (Figure 6-52), for a more clear understanding. Only 4 experiments, out of the 8 in total, are shown, for clarity purposes. Black rectangular represent the outlet experimental values, whereas the theoretical values are shown in grey dots. The chart indicates that the difference among experimental and theoretical values is greater in the temperature than in the humidity ratio axis. It can be thus concluded that the mathematical model can better predict humidity ratio than temperature values.

		A	IR				WATER		DESICCANT						
Outlet D	ry bulb tempe	erature	Outle	et Humidity ra	atio	Out	let Temperati	ure	Outl	et Temperatı	ıre	Outlet Concentration			
experiment	theoretical model	deviation	experiment	theoretical model	deviation	experiment	theoretical model	deviation	experiment	theoretical model	deviation	experiment	theoretical model	deviation	
°C	°C	%	kg H ₂ O / kg dry air	kg H ₂ O / kg dry air	%	°C	°C	%	°C	°C	%	%	%	%	
18.99	17.32	8.80	0.0025	0.0027	-8.00	25.87	24.32	6.01	28.86	24.67	14.51	34.0%	36.0%	-5.94	
20.00	17.78	11.07	0.0032	0.0033	-3.12	26.14	26.25	-0.43	28.25	26.55	6.01	34.3%	36.3%	-5.92	
13.58	13.19	2.85	0.0020	0.0022	-10.00	19.29	17.01	11.82	19.09	17.09	10.47	37.1%	37.9%	-2.23	
14.54	14.20	2.29	0.0024	0.0028	-16.67	20.95	18.38	12.27	18.82	18.38	2.32	35.8%	37.0%	-3.48	
16.65	16.14	3.02	0.0026	0.0027	-3.85	21.97	20.14	8.32	19.80	20.06	-1.34	34.7%	36.4%	-4.79	
14.66	14.12	3.73	0.0024	0.0027	-12.50	20.49	20.11	1.87	18.78	17.64	6.09	33.4%	34.6%	-3.73	
14.42	13.36	7.36	0.0029	0.0033	-13.79	18.41	17.37	5.67	17.74	16.55	6.71	31.4%	32.1%	-2.38	
14.43	14.71	-1.95	0.0023	0.0027	-17.39	18.23	16.93	7.11	17.90	16.42	8.27	38.3%	39.9%	-4.11	

Table 6-5 Comparison table between theoretical & experimental results in dehumidification



Figure 6-52 Psychrometric chart showing the differences between the theoretical and experimental results in dehumidification

6.4.2 Regeneration

The same process was followed for the assessment of the regeneration model. It was adapted to the physical characteristics of the regenerator that has been installed at the Laboratory of Applied Thermodynamics. Namely, the geometrical inputs to the model were taken from the description in subchapter 6.3.1.2 Regenerator. The inlet values of the monitored parameters were the same with the experiments, taken from Table 6-4 List of regeneration experiments. Thus, the input conditions to the model were the same as the initial conditions of the experiments. There were 12 regeneration experiments, so 12 simulations were performed and produced the theoretical results.

The results showed a reasonable deviation between the theoretical and the experimental results, with all deviations being in the range of -20.56% to 15.63%.

In the case of the air outlet temperature (Figure 6-53), the deviation ranges from -1.65% to - 12.05% for all experiments carried out. Fair difference among the theoretical and experimental results exists in the case of outlet humidity ratio (Figure 6-54), where the deviations start from -2.60% and reach -20.56%. It seems that the experimental apparatus produced colder and drier air than expected.



Figure 6-53 Graphical representation of the deviation in outlet air temperature during regeneration



Figure 6-54 Graphical representation of the deviation in outlet humidity ratio during regeneration

The theoretical desiccant outlet temperature (Figure 6-55) produces satisfactory results, since its deviation with the experimental data ranges from 2.49% to 15.63%. Reasonable deviation can be also seen in the desiccant solution outlet concentration (Figure 6-56), with the minimum being -1.19% and the maximum 12.00%. The experimental apparatus produced hotter and stronger desiccant than expected.



Figure 6-55 Graphical representation of the deviation in outlet desiccant temperature during regeneration



Figure 6-56 Graphical representation of the deviation in outlet desiccant concentration during regeneration

As far as the hot water loop is concerned (Figure 6-57), the deviation of the outlet temperature shows fair results, with the minimum being -1.85% and the maximum -5.98%. The apparatus produced colder water than expected.



Figure 6-57 Graphical representation of the deviation in outlet water temperature during regeneration

Table 6-6 shows the comparison between the theoretical and experimental results of the air, water and desiccant loop during the regeneration process. The percentage deviation is shown in bold letters.

The psychrometric chart in the Figure 6-58 represents the comparison between theory and experiment for the air loop. Only 5 experiments, out of the 12 in total, are shown, for clarity purposes. Black rectangular represent the outlet experimental values, whereas the theoretical values are shown in grey dots. Again, it is visualized that the difference among experimental and theoretical values is greater in the temperature than in the humidity ratio axis. It can be thus concluded that the mathematical model can better predict humidity ratio than temperature values.

		А	IR				WATER		DESICCANT							
Outlet Dr	y bulb temp	erature	Outle	et Humidity ra	itio	Out	let Temperatı	ıre	Outle	et Temperat	ure	Outlet Concentration				
experiment	theoretical model	deviation	experiment	theoretical model	deviation	experiment	theoretical model	deviation	experiment	theoretical model	deviation	experiment	theoretical model	deviation		
°C	°C	%	kg H ₂ O / kg dry air	kg H ₂ O / kg dry air	%	°C	°C	%	°C	°C	%	%	%	%		
50.65	53.71	-6.06	0.0692	0.0710	-2.60	60.12	62.242	-3.53	40.67	39.66	2.49	40.0%	35.7%	10.68		
46.56	48.87	-4.96	0.0538	0.0642	-19.28	57.19	59.767	-4.51	38.57	36.99	4.11	38.5%	36.2%	5.95		
44.12	45.32	-2.72	0.0392	0.0439	-11.93	54.02	56.87	-5.28	36.16	34.97	3.29	38.7%	35.9%	7.29		
47.92	49.89	-4.11	0.0512	0.0608	-18.75	61.02	62.89	-3.06	37.59	36.54	2.80	41.5%	39.1%	5.72		
46.01	49.21	-6.95	0.0472	0.0534	-13.09	57.03	60.432	-5.97	37.85	35.12	7.19	39.9%	37.7%	5.49		
49.60	51.65	-4.15	0.0658	0.0729	-10.79	64.27	65.532	-1.96	41.17	36.13	12.25	41.7%	36.7%	12.00		
45.84	47.99	-4.69	0.0509	0.0562	-10.41	58.52	59.927	-2.41	37.95	32.02	15.63	37.1%	37.5%	-1.19		
36.61	37.21	-1.65	0.0248	0.0299	-20.56	43.39	45.987	-5.98	30.20	26.16	13.38	33.8%	34.5%	-2.07		
46.29	50.00	-8.03	0.0537	0.0573	-6.65	66.78	68.215	-2.15	44.14	42.16	4.50	37.5%	35.2%	6.21		
47.70	53.45	-12.05	0.0539	0.0556	-3.15	59.69	63.079	-5.68	39.23	37.13	5.35	41.1%	37.3%	9.20		
46.51	51.02	-9.70	0.0490	0.0545	-11.22	59.06	61.160	-3.56	38.62	36.52	5.42	40.3%	39.6%	1.71		
48.64	54.33	-11.70	0.0607	0.0633	-4.28	65.13	66.334	-1.85	39.61	35.01	11.61	41.2%	38.5%	6.55		

Table 6-6 Comparison table between theoretical & experimental results in regeneration



Figure 6-58 Psychrometric chart showing the differences between the theoretical and experimental results in regeneration

6.5 CONCLUSIONS

This chapter is divided into two parts; the first provides the theoretical background of the heat and mass transfer processes between the air, the desiccant solution and the water circuit in a non-adiabatic parallel plate dehumidifier and regenerator. The latter provides the findings of the operation of the liquid desiccant experimental apparatus installed at the National Technical University of Athens, Laboratory of Applied Thermodynamics.

The "non-adiabatic" term in this Chapter shows that during dehumidification and regeneration there is interaction with external energy sources, in the form of a parallel plate heat exchanger. Inside the plate heat exchanger, cooling water flows to provide the necessary cooling effect during dehumidification and hot water, coming from a low temperature source, flows to provide heat during regeneration.

Excellent agreement was found between the developed computational programs and published experimental data, with the discrepancy between them not exceeding 12%.

The variables that seem to have the greatest impact on the performance of an internally cooled plate dehumidifier are the air mass flow rate, air humidity ratio, solution temperature and water temperature. More specifically, their dependencies are described below.

- Increasing air flow rate causes an increase in the dehumidification mass rate. When
 the air flow rate is increased, the residence time for the air in the dehumidifier is
 decreased, resulting in a decrease of the mass transfer coefficient, a higher outlet air
 humidity ratio and thus, a lower absorber efficiency. At the same time, higher outlet
 humidity ratio means greater difference in the average water vapor pressure and
 higher dehumidification mass rate. So, during the low air flow rate range, more air
 flow rate can achieve better dehumidification performance.
- Increasing air inlet humidity ratio increases the dehumidification mass rate. A higher inlet humidity ratio implies higher air vapor pressure and thus, a higher average water vapor pressure difference between the air and the desiccant. Consequently, higher potential for mass transfer from the gas to the liquid phase is established.
- The increase of the cooling water temperature results in a decrease in the dehumidification efficiency and the dehumidification mass rate, for all solutions examined. This happens because as the inlet cooling water increases, the heat transfer rate from the liquid film to cooling water is decreased, hence the saturation

pressure is increased. This causes a decrease in the vapor pressure driving force and thus, in a decrease in the mass transfer rate.

- Increasing the desiccant inlet temperature decreases the dehumidification mass rate. This happens because the increase in desiccant temperature increases the water vapour pressure of the desiccant and so, the average water vapor pressure difference between the air and desiccant in the dehumidifier is decreased. This leads to a higher air outlet humidity ratio and consequently, to a lower dehumidification mass rate.
- The dehumidification mass rate is increased with the desiccant flow rate. This is happening because increasing the desiccant flow rate decreases the contact time between the desiccant and the air and so, decreases the variation of the desiccant concentration and temperature through the dehumidifier. As a result, the variation of the surface vapor pressure of the desiccant through the dehumidifier is decreased and, hence, the average water vapor pressure difference between the desiccant and air in the dehumidifier is increased. This leads to a lower air outlet humidity ratio and a higher dehumidification mass rate.
- The absorber efficiency is increased with the desiccant flow rate, due to the lower air outlet humidity ratio. Increasing the desiccant flow rate increases the mass transfer coefficient between the desiccant and the air in the dehumidifier, resulting in a further increase in the dehumidification efficiency.
- The co-flow absorber using LiCl has the best efficiency, as it can reduce humidity ratio by 31%, while the absorber using LiBr reduces humidity ratio by only 12%, under the same inlet conditions.
- The counter-flow configuration produces a much drier air than co-flow, since counter flow can reduce humidity ratio by 44% when co-flow reduces it by 31%.

The second part of this chapter presented the liquid desiccant system that has been installed at the National Technical University of Athens, Laboratory of Applied Thermodynamics. Specific information regarding the characteristics of the experimental apparatus was given. The method followed for each dehumidification and regeneration experiment is explicitly stated. A plausibility check for an indicative dehumidification and regeneration experiment was implemented in order to verify the quality of the experimental data. Since 8 dehumidification and 12 regeneration experiments were carried out, the average values of each experiment were given in a summarizing table. For dehumidification, the deviation in energy balance ranged from -5.36% to 19.84% and for regeneration from -3.02% to 10.57%.

The experimental results showed that the air dehumidification rate (dehumidifier) ranged from 6.99 to 11.90 kg water / hr, depending on the weather and operational conditions. The air humidification rate (regenerator) ranged from 3.55 to 12.21 kg water / hr, depending on the weather and operational conditions. The energy consumption related to the regenerator ranged from 6.93 kWh to 26.22 kWh. The required water inlet temperature was set to a range from 46.11 °C to 72.86°C. Under water inlet temperatures as low as 46.11°C, the regeneration rate was the minimum one, but still at an acceptable value of 3.55 kg water / hr.

The comparison between the obtained experimental data and the results of the developed mathematical model was of particular interest, as it strengthened the validity of the theoretical part of this Thesis. Both mathematical models of the dehumidifier and the regenerator were adapted to the specific physical characteristics and the initial conditions of each experiment. The results showed a reasonable deviation between the theoretical and the experimental results, with all deviations being in the range of -17.39% to 14.51% for the dehumidification process and of -20.56% to 15.63% for the regeneration process. In the dehumidification case, the experimental apparatus produced drier but hotter air than expected, hotter and weaker desiccant than expected and hotter water than expected. In the regeneration case, the experimental apparatus produced colder and drier air than expected, hotter and stronger desiccant than expected and colder water than expected. For both processes, it has been visualized that the mathematical model predicts better the outlet air humidity ratio than the outlet air temperature. Taking into consideration the experimental conditions, it can be educed that the model predicts the experimental data with reasonable accuracy.

Undoubtedly, the operation of the experimental apparatus has been a time consuming and challenging task. External parameters and difficulties appeared that had to be confronted with patience and persistence. However, the information and the practical experience gained regarding the operation of liquid desiccant systems was a rewarding task.

6.6 NOMENCLATURE

- a heat transfer coefficient (W/m²K)
- A wetted area of dehumidifier (m²)
- C_p specific heat capacity at constant pressure (Jkg⁻¹K⁻¹)
- d thickness of plate wall (m)
- $D_{h.in}$ internal hydraulic diameter of plate
- $D_{h.out}$ external hydraulic diameter of plate
- $D_{h.s}$ hydraulic diameter of solution
- h enthalpy $(kJ/kg_{d\alpha})$
- k thermal conductivity (W/mK)
- K_G mass transfer coefficient (kmol/sm²mole fraction)
- L width of plates (m)
- \dot{m}_{deh} dehumidification mass rate (kg_w/h)
- *m* mass flow rate (kg/s)
- n efficiency of dehumidifier or regenerator
- Q heat (J)
- T temperature (°C)
- U overall heat transfer coefficient (W/m²K)
- V cross section velocity (kg/m²s)
- v kinematic viscosity (m²/s)
- W air humidity ratio $(kg_w/kg_{d\alpha})$
- X salt concentration in solution (kg_{salt}/kg_s)
- Y moisture content in solution (kg_w/kg_s)
- z dehumidifier height (m)
- Δh_{abs} latent heat of absorption (J/kg)

Greek symbols

- δ film thickness (m)
- ρ density (kg/m³)
- μ dynamic viscosity (kg/ms)

Subscripts

- α air
- atm atmospheric
- dα dry air
- eq equivalent
- exp experimental
- in inlet
- L latent
- mα moist air
- out outlet
- p plate
- S sensible
- s solution
- sim simulated
- st steam
- w water

Superscripts

sat saturated

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7 CONCLUSIONS AND FURTHER RESEARCH

7.1 CONCLUSIONS

The present work aimed at extending the understanding of liquid desiccant air conditioning systems and at unlocking the heat and mass transfer phenomena in the two most critical components: the dehumidifier and the regenerator. The study was performed both theoretically and experimentally and investigated the performance of these processes in terms of energy efficiency, dehumidification rate and humidification rate.

Specific objectives of this study were:

- To acquire the theoretical background on the various liquid desiccant materials, components, devices and configuration.
- To develop the necessary mathematical models for the evaluation of the performance of liquid desiccant dehumidifiers and regenerators.
- To investigate, through extensive parametric analysis, the most important parameters that affect the performance of liquid desiccant dehumidifiers and regenerators.
- To experimentally investigate the performance of liquid desiccant air conditioning systems.
- To find and propose optimization methods and procedures, based on the implemented theoretical and experimental study.

The innovation of the implemented work lied on the following fields:

- To solve the heat and mass transfer phenomena during dehumidification and regeneration procedure through ordinary differential equations.
- To develop an algorithm for the calculation of heat and mass transfer coefficients in
 - o adiabatic beds with structured packing
 - o non-adiabatic tubular heat exchangers
 - o non-adiabatic parallel plate heat exchangers.
- To develop mathematical models for the simulation of dehumidifiers/regenerators as
 - adiabatic beds
 - o non-adiabatic tubular heat exchangers
 - o non-adiabatic parallel plate heat exchangers.
- To validate the developed mathematical models with published experimental data.
- To perform a comparative analysis of the three aforementioned configurations.

- To perform parametric simulations with predefined depended variables, such as type of liquid desiccant solution, state of inlet air, state of inlet desiccant solution, state of inlet water (when applicable).
- To gain practical experience on dehumidification and regeneration processes through laboratory experiments, with predefined depended variables, such as state of inlet air, state of inlet desiccant solution and state of inlet water.
- To perform a comparative analysis among the experimental results and the developed mathematical models.

The study starts with a comprehensive literature review of the heat and mass transfer operations in liquid desiccant cooling systems. Plenty of research papers have been written on the dehumidification and regeneration principles, with the earlier tracing back to 1969. Since then, great strides have been made; insightful theoretical models on simultaneous heat and mass transfer process have been established. The effects however of some inlet parameters on the efficiency remained uncertain up to now, so further analytical and experimental investigations were needed to assess the overall performance of these systems.

The route to the thermodynamic analysis of the absorption and desorption process and the performance assessment of liquid desiccant systems began with the familiarization with the thermophysical properties of the three most common liquid desiccant solutions; Lithium Chloride (LiCl), Lithium Bromide (LiBr) and Calcium Chloride (CaCl₂). Calculation models were described for the following properties: Solubility boundary, vapor pressure, density, surface tension, dynamic viscosity, thermal conductivity, specific thermal capacity and differential enthalpy of dilution.

The simplest configuration of the dehumidifier and regenerator, which is the adiabatic bed with structured packing, was studied first. Two mathematical models were expressed developed, one for the dehumidifier and one for the regenerator. They both showed an excellent agreement with the available experimental data from the literature. The design variables found to have the greatest impact on the dehumidification performance were the air mass flow rate, air humidity ratio, desiccant temperature and desiccant concentration and on the regeneration performance were the desiccant concentration, desiccant temperature and desiccant mass flow rate. The results showed that high absorber efficiency and system efficiency could be achieved under humid conditions, low air mass flow rates and LiCl as the desiccant solution. On the other hand, high regeneration efficiency could be

achieved under high desiccant mass flow rates, high air mass flow rate, high desiccant inlet temperature, low desiccant inlet concentration and CaCl₂ as the desiccant solution.

The non-adiabatic tubular liquid desiccant dehumidifier and regenerator was then studied. Two more mathematical models were developed and analysed and again, excellent agreement was found between the developed computational program and published experimental data. The sensible capacity of the dehumidifier was found to depend strongly on air inlet temperature. As the weather becomes hotter, the sensible cooling capacity of the dehumidifier and the dehumidifier and the dehumidification mass rate is increased. As the weather becomes more humid, the simulations showed that the latent capacity of the dehumidifier and the dehumidification mass rate is increased. So, higher system coefficient of performance is achieved under hot and humid conditions. The input variables found to have the greatest impact on the regeneration performance are the air humidity ratio, hot water temperature, desiccant concentration, desiccant temperature and the desiccant flow rate; however, humid climate seems to decrease the regeneration potential. The combination of CaCl₂ as the desiccant solution and DDU flow (water downward – desiccant downward – air upward) showed the best regeneration efficiency and mass rates.

The work concerning the liquid desiccant experimental apparatus installed in the Laboratory was then extensively discussed. The mathematical model developed for the non-adiabatic plate liquid desiccant dehumidifier presented excellent agreement with the published experimental data. The simulations showed that the dehumidification mass rate is increased with the air inlet humidity ratio, air flow rate and desiccant flow rate, but it is decreased with the cooling water temperature and desiccant inlet temperature. The dehumidifier using LiCl with counter-flow configuration presented the best efficiency.

Specific information regarding the technical characteristics of the experimental apparatus and the method followed for each dehumidification and regeneration experiment was then explicitly stated. The obtained experimental data are reliable enough, since the deviation in energy balance ranged from -5.36% to 19.84% for dehumidification and from -3.02% to 10.57% for regeneration. The experimental results showed that the air dehumidification rate (dehumidifier) ranged from 6.99 to 11.90 kg water / hr, depending on the weather and operational conditions. The air humidification rate (regenerator) ranged from 3.55 to 12.21 kg water / hr, depending on the regenerator ranged from 6.93 kWh to 26.22 kWh. The required water inlet temperature was set to a range from 46.11 °C to 72.86°C. Under water inlet

temperatures as low as 46.11°C, the regeneration rate was the minimum one, but still at an acceptable value of 3.55 kg water / hr.

The comparison between the obtained experimental data and the results of the developed mathematical model was of particular interest, as it strengthened the validity of the theoretical part of this Thesis. Both mathematical models of the dehumidifier and the regenerator were adapted to the specific physical characteristics and the initial conditions of each experiment. The results showed a reasonable deviation between the theoretical and the experimental results, with all deviations being in the range of -17.39% to 14.51% for the dehumidification process and of -20.56% to 15.63% for the regeneration process. I t can be thus educed that the model predicts the experimental data with reasonable accuracy. In the dehumidification case, the experimental apparatus produced drier but hotter air than expected, hotter and weaker desiccant than expected and hotter water than expected. In the regeneration case, the experimental apparatus produced colder and drier air than expected, hotter and stronger desiccant than expected and colder water than expected. For both processes, it has been visualized that the mathematical model predicts better the outlet air humidity ratio than the outlet air temperature.

7.2 FURTHER RESEARCH AND DEVELOPMENT

Although a fair amount of new knowledge and information is presented in this thesis, several topics can be further investigated, such as:

- Economic performance of a solar assisted liquid desiccant air conditioning system.

This topic may include a study of the influences of the collector area and storage volume of hot water tank on the annual solar fraction, annual collector efficiency and life cycle cost. It may also include calculations of the initial cost, annual operating cost, life cycle cost, payback period and annual CO_2 emissions of the whole liquid desiccant system.

- Theoretical investigation on the performance of liquid desiccant systems versus vapor compression air conditioning systems.

This topic relates to the comparison of liquid desiccant technology versus vapor compression chillers. It requires the definition of the coefficient of performance, both electrical and thermal, and the calculation of the amount of energy consumed for covering a specific amount of sensible and latent load of a building.

- Annual evaluation of energy, environmental and economic performances of a liquid desiccant air conditioning system.

This theoretical topic includes the evaluation of the annual primary energy consumption, annual operation energy savings, payback period and life cycle cost of a liquid desiccant system. These parameters can be then compared to those of a conventional air conditioning system to produce valuable outcomes.

- Experimental investigation of the energy consumption of a liquid desiccant airconditioning system in Athens.

This topic requires the installation of a liquid desiccant system in a building in Athens, in order to elaborate data from real operating conditions. It focuses on a parametric study and sensitivity analysis to quantify the influences of key design and operating parameters on the coefficient of performance of a liquid desiccant system installed in Athens.

- Performance analysis of the Photovoltaic electrodialysis (PV–ED) regeneration.

A new regeneration method that has been recently introduced is the photovoltaic electrodialysis. This method is based on the principle that ions transport through selective membranes under the influence of an electrical field. Under the application of an electrical field, the anions and cations pass through anion exchange membranes and cation exchange membranes, respectively. This flow causes a rise in the ions concentration in the concentrate compartments and fall into the dilute compartment. This process transforms the dilute solution into concentrated (strong) desiccant solution and pure water. Both theoretical and experimental investigation of this new regeneration method would be of great importance, due to the lack of research data in this field.

- Performance analysis of a dehumidification system combined with ultrasonic atomization technology (UADS).

Another method recently introduced for enhancing the liquid desiccant systems efficiency is the integration of an ultrasonic atomization system to the dehumidifier device. The UADS method atomizes desiccant solution into tiny droplets and therefore, it drastically enlarges the liquid-gas contact area. The theoretical and experimental investigation of whether this method actually works would be of great research interest.

8 APPENDICES

8.1 AUTHOR'S PUBLICATIONS

The following graph, exported from Scopus on 18/03/2015, shows the total number of documents published by the author, by year.



Figure 8-1 Documents published by the author, by year

The following list shows the full title of the author's publications, so far.

- R. Christodoulaki, I. Koronaki, E. Rogdakis, Hybrid Liquid Desiccant / Vapour Compression Air-Conditioning Systems: A Critical Review, 9th Biennial ASME Conference on Engineering Systems Design and Analysis ESDA 2008, 7-9 July 2008, Haifa, Israel.
- Koronaki I.P., Christodoulaki R.I., Papaefthimiou V.D., Rogdakis E.D., Performance analysis on the internally cooled dehumidifier using two liquid desiccant solutions, ASME Conference on Engineering Systems Design and Analysis ESDA 2012, 4-7 July 2012, Nantes, France.
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- Koronaki I.P., Christodoulaki R.I., Papaefthimiou V.D., Rogdakis E.D., thermodynamic analysis of a liquid desiccant cooling system under mediterranean climatic conditions, ASME International Mechanical Engineering Congress & Exposition IMECE, November 9-15 2012, Houston, Texas, USA.

- 5. Koronaki I.P., Christodoulaki R.I., Papaefthimiou V.D., Rogdakis E.D., Thermodynamic analysis of a counter flow adiabatic dehumidifier with different liquid desiccant materials, Applied Thermal Engineering, 50, p. 361-373, 2013.
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- Koronaki I.P., Christodoulaki R.I., Papaefthimiou V.D., Rogdakis E.D., Counter flow adiabatic regenerator and comparative study, the 27th ECOS International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems, 15-19 June 2014, Turku, Finland.
- Koronaki I.P., Christodoulaki R.I., Papaefthimiou V.D., Rogdakis E.D., Counter flow adiabatic regenerator and comparative study, International Journal of Thermodynamics, under publication.

8.2 AUTHOR'S CITATIONS

The following graph, exported from Scopus on 18/03/2015, shows the total number of citations to the author, by year.





Note: Scopus is in progress of updating pre-1996 cited references going back to 1970. Pre-1996 citation counts might increase over time.

Figure 8-2 Citations to the author's publications, by year

The following list shows the citations to the author's publications, so far.

- Longo, G.A., Gasparella, A. Three years experimental comparative analysis of a desiccant based air conditioning system for a flower greenhouse: Assessment of different desiccants (2015) Applied Thermal Engineering, 78, pp. 584-590.
- Mota-Babiloni, A., Navarro-Esbrí, J., Barragán-Cervera, Á., Molés, F., Peris, B. Analysis based on EU Regulation No 517/2014 of new HFC/HFO mixtures as alternatives of high GWP refrigerants in refrigeration and HVAC systems (2015) International Journal of Refrigeration, 52, pp. 21-31.
- Sultan, M., El-Sharkawy, I.I., Miyazaki, T., Saha, B.B., Koyama, S. An overview of solid desiccant dehumidification and air conditioning systems (2015) Renewable and Sustainable Energy Reviews, 46, pp. 16-29.
- 4. Luo, Y., Shao, S., Xu, H., Tian, C., Yang, H. Experimental and theoretical research of a fin-tube type internally-cooled liquid desiccant dehumidifier (2014) Applied Energy, 133, pp. 127-134.
- 5. Langroudi, L.O., Pahlavanzadeh, H., Mousavi, S.M. Statistical evaluation of a liquid desiccant dehumidification system using RSM and theoretical study based

on the effectiveness NTU model (2014) Journal of Industrial and Engineering Chemistry, 20 (5), pp. 2975-2983.

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8.5 CURRICULUM VITAE

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Education

National Technical University of Athens NTUA 2009-2015

PhD in Mechanical Engineering

'Thermodynamic analysis of open cycle evaporative cooling systems with liquid desiccants'. Supervisors: Prof. E. Rogdakis, Prof. K. Antonopoulos, Ass. Prof. I. Koronaki.

University College London UCL

2005-2006

MSc Environmental Design & Engineering (Distinction, 72/100)

Modules: Environmental Analysis, Building Solar Design, Efficient Building Services Systems, Energy Management & the Control of Noise, Lit Environment, Health & Comfort, Natural & Mechanical Ventilation, Energy Context. Dissertation: 'Environmental Assessment: The Eden Project'. Supervisor: Dr. Ben Croxford.

University of Athens

1999-2005

1991-1997

BSc Physics (6.51 /10)

Dissertation: 'Internal Boundary Layer'. Supervisor: Prof. K. Helmis.

Italian School of Athens

High School Diploma (16.7/20)

Foreign languages

University of Cambridge, Certificate of Proficiency 2004

Instituto Italiano di Cultura in Atene, Diploma di Lingua Italiana 1997

Professional experience

European Union

Oct 2014

Evaluation of research proposals under the call "Competitive Low-Carbon Energy", of the Horizon 2020.

European Union

Jan 2012

Evaluation of research proposals under the call "Energy Efficient Buildings", of the Framework Program FP7, Specific Programme "Cooperation".

National Technical University of Athens

since 2008

School of Mechanical Engineering, Department of Thermal Energy

- Participation in the NSRF ESPA national project SOLECON regarding the development of an innovative cost-efficient Linear Fresnel solar concentrating collector,
- Participation in the FP7 European project *NRG4CAST* regarding energy efficiency in buildings,
- Participation in the Leonardo Lifelong Learning European project: 'REAL SKILLS EUROPE Refrigerant Emissions And Leakage Skills for Europe',
- Lecturing to parts of modules: Refrigeration II (9th Semester) and Software for Thermodynamics (7th Semester)
- Management and guidance of post-graduate students during their Thesis

Centre for Renewable Energy Sources & Saving 15/04/08-31/12/10

Solar Thermal & Geothermal Energy Department.

Participation in FP6, FP7 & IEE European projects:

- HIGH COMBI 'HIGH solar fraction heating and cooling systems with COMBInation of innovative components and methods,
- SAHC 'Solar Assisted Heating and Cooling in the agrofood sector',
- SOLAIR 'Increasing the market implementation of solar air conditioning systems for small and medium applications in residential and commercial buildings',
- SOLPOOL 'Solar energy use in outdoor swimming pools',
- SolarCombi+ 'Small scale solar cooling systems in combination with traditional solar thermal systems for domestic hot water and space heating',

Participation in the national project Green Island: Agios Eustratios.

Energy Assessments in commercial buildings.

Participation in the International Energy Agency Task 38: Solar Cooling in Buildings.

Proposal writing under the European calls FP7, Intelligent Energy and the national calls ESPA.

ELKEDE Centre of Technology and Design

Research and Development Department.

Participation in Eureka European projects:

- E! SEPAR 3644 'Reclamation of WWTP effluents using a hi-tech parallel solar-powered unit to produce high value quality water',
- E! REWAS 3643 'Arsenic and pesticide removal from natural water by an effective, safe and compact-sized separation system'.
- E! PLASTIC-SEPAR 3887 'Optimisation of plastics recycling technologies: assessment of separation methods'

Proposal writing under the European calls FP7 and EUREKA.

Athens International Airport El. Venizelos 01/03/03-01/05/03

Environmental Department: Student Practice on Differential Optical Absorption Spectroscopy Systems as a tool to measure pollution from airplanes.

Awards -Scholarships

First Award in the 2011 i-bank Technology & Innovation National Competition, organized by the National Bank of Greece. Mutual work with P. Tsekouras. Title of project: Hybrid photovoltaics and Fresnel solar thermal collectors.

3-year scholarship from the National Technical University of Athens, 2011.

Softwares

- Engineering calculations: MathCad
- Data analysis & graphs: Origin Lab.
- Designing: AutoCad
- Solar thermal systems simulation: T-SOL pro, TRNSYS.
- Techno-economic analysis of R.E.S. sytems: RetScreen, PVSYST.
- Building simulation: EDSL TAS Ambiens 9.0.6., EnergyPlusV1.4.0.
- Building Energy Assessment: S.A.P. 2005.
- Computational Fluid Dynamics: EDSL TAS 3D 9.0.7.
- R.E.S. Techno-economic analysis: RetScreen, PVSYST 3.4.
- Artificial & natural lighting: Dialux 4.1, Weather Tool, Solar Tool, Suntect.
- Certification of Microsoft Office knowledge: ECDL

 Publications – R. Christodoulaki, E. Koronaki, E. Rogdakis, 'Hybrid Liquid Desiccant / Speeches Vapour Compression Air-Conditioning Systems: A Critical Review', 9th Biennial ASME Conference on Engineering Systems Design and Analysis ESDA 2008, 7-9/07/08, Haifa, Israel.

R. Christodoulaki, *Solar thermal energy for air condition of buildings*, Pilot Training in the members of the Greek Solar Industry Association EBHE, organized by EBHE, 14 July 2008, Athens.

R. Christodoulaki, *Solar Cooling Systems: Design and installation*, Workshop on Solar thermal systems applications, Organized by CRES and Municipality of Heraklion Crete, 4 November 2008, Heraklion.

R. Christodoulaki, *Heating swimming pools with solar thermal systems*, Workshop on Solar thermal systems applications, Organized by CRES and Municipality of Heraklion Crete, 4 November 2008, Heraklion.

R. Christodoulaki, *Indoor thermal comfort*, Workshop on Renewable Energy Sources for Heating and Cooling in buildings, organized by CRES, 14 November 2008, Hotel Amalia, Athens.

R. Christodoulaki, *Design and installation guidelines of Solar Cooling systems*, Workshop on Renewable Energy Sources for Heating and Cooling in buildings, organized by CRES, 14 November 2008, Hotel Amalia, Athens.

R. Christodoulaki, *Best practice examples on solar cooling*, Workshop on Renewable Energy Sources for Heating and Cooling in buildings, organized by CRES, 14 November 2008, Hotel Amalia, Athens.

R. Christodoulaki, V. Drosou, E. Korma, *Solar Thermal Applications in the Residential Sector*, ENERTECH '08, Session E: Energy Saving & Efficiency, 22/11/08, Athens.

R. Christodoulaki, E. Koronaki, E. Rogdakis, *Thermodynamic Appraisal* of LiCl, LiBr and CaCl2 as common liquid Drying Agents in Open Systems, ENERTECH '08, Session E: Energy Saving & Efficiency, 22/11/08, Athens.

R. Christodoulaki, *Solar thermal systems/a state of the Greek market*, CEE Solar 2008 Shedding light on the emerging solar market, 21st November 2008, Athens Imperial Hotel, Athens.

R. Christodoulaki, Solar thermal applications for swimming pools, Design and installation methods of Solar Cooling systems, Open cycle solar cooling systems, Best installations of Solar cooling in Europe, 2 days intensive Training Seminars to the members of the Greek Solar Industry Association EBHE, organized by EBHE, 24-25 November 2008, Athens.

R. Christodoulaki, *Solar Cooling: Principle of operation and installed systems*, Technica Journal, Issue 250, December 2008, page 63-67.

R. Christodoulaki, *Solar air-conditioning*, International Conference and Business Missions, Trans-Solar project: Transfer of the experience for the development of the solar thermal products, 26 - 27 February 2009, Prague.

R. Christodoulaki, *Solar air-conditioning*, Workshop on Solar air conditioning, organized by CRES, 23 April 2009, Portes Palace Hotel, Halkidiki.

R. Christodoulaki, *Solar thermal applications for swimming pools,* Workshop on Solar air conditioning, organized by CRES, 23 April 2009, Portes Palace Hotel, Halkidiki.

R. Christodoulaki, *Designing solar thermal systems for swimming pools with the software tool «Impact Advisor»,* Workshop on Solar thermal applications in hotels, organized by CRES, Grand Hotel Palace, 11 February 2009, Thessaloniki.

R. Christodoulaki, *Solar thermal heating systems in European Union*, International Conference of Trans-Solar project: Transfer of the experience for the development of the solar thermal products, 16 April 2009, Budapest.

R. Christodoulaki, *Solar thermal applications for swimming pools and hot water*, 8 May 2009, Rhodes.

R. Christodoulaki, E.Villa, A. Thuer, M. Calderoni, E. Wiemken, M.J. Carvalho, Q. Cavalera, O. Ayadi, J.R. Lopez, D. Mugnier, S. Medved, T. Van Steenberghe, M. Proville, *solair – increasing the market implementation of solar air-conditioning systems for small and medium applications in residential and commercial buildings*, 4th European solar thermal energy conference - estec 2009, 25-26 May 2009, Munich, Germany.

R. Christodoulaki, *Solar Cooling Technologies & Applications*, International Conference of Trans-Solar project: Transfer of the experience for the development of the solar thermal products, 18 June 2009, Warsow.

R. Christodoulaki, *Results of SOLAIR project: Solar Cooling market and applications in EU*, CEE Solar 2009 Shedding light on the emerging solar market, 26th June 2009, Park Hotel Athens, Athens.

R. Christodoulaki, *Solar thermal applications in the Agrofood industry*, Exhibition Agrotica 2010, Conference centre Nikolaos Germanos, 6 February 2010, Thessaloniki.

R. Christodoulaki, *Energy efficient technologies in the Agrofood industry*, Exhibition Agrotica 2010, Conference centre Nikolaos Germanos, 6 February 2010, Thessaloniki.

R. Christodoulaki, Solar thermal heating for the Agrofood industry, Workshop organized by CRES, Golden Age Hotel, 22 March 2010,

Athens.

R. Christodoulaki, *Energy efficient technologies in the Agrofood industry*, organized by CRES, Golden Age Hotel, 23 March 2010, Athens.

Tsekouras P., Chasapis D., Christodoulaki R., Karytsas K., Balaras K., Tzivanidis C., Antonopoulos K., *Innovative high solar fraction system for heating and cooling in Athens*, 4th National Conference: Renewable Energy Sources applications for an ambitious and credible National Action Program, 10-12 May 2010, Athens.

R. Christodoulaki, *Environmental effects of refrigerant leakages,* Workshop organized by NTUA, 25 March 2011, Athens.

I. Koronaki, K. Beerman, I. Chaer, D. Cowan, G. Gontarz, K. Kaar, G. Maidment, W. Reulens, R. Christodoulaki, Refrigerant Emissions and Leakage prevention across Europe – Results from the REAL SKILLS EUROPE Project, the 24th ECOS International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems, 4-7 July 2011, Novi Sad, Serbia.

R. Christodoulaki, *Real Skills Europe e-learning program*, Workshop organized by NTUA, 7 September 2011, Athens.

Koronaki I.P., Christodoulaki R.I., Papaefthimiou V.D., Rogdakis E.D., *Performance analysis on the internally cooled dehumidifier using two liquid desiccant solutions*, ASME Conference on Engineering Systems Design and Analysis ESDA 2012, 4-7/07/12, Nantes, France.

Koronaki I.P., Christodoulaki R.I., Papaefthimiou V.D., Rogdakis E.D., *Thermodynamic analysis of an internally cooled dehumidifier using three liquid desiccant solutions*, ASME Summer Heat Transfer Conference HT2012, 9-12/07/12, Puerto Rico, USA.

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Koronaki I.P., Christodoulaki R.I., Papaefthimiou V.D., Rogdakis E.D., Thermodynamic analysis of a counter flow adiabatic dehumidifier with different liquid desiccant materials, Applied Thermal Engineering, 50, p. 361-373, 2013.

Koronaki I.P., Christodoulaki R.I., Papaefthimiou V.D., Rogdakis E.D., Critical review of coupled heat and mass transfer models for a liquid desiccant adiabatic dehumidifier and regenerator, Advances in Building Energy Research, July 2013.

Koronaki I.P., Christodoulaki R.I., Papaefthimiou V.D., Rogdakis E.D., Counter flow adiabatic regenerator and comparative study, the 27th ECOS International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems, 15-19 June 2014, Turku, Finland.

Memberships

American Society of Heating, Refrigerating & Air-conditioning Engineers ASHRAE.

Hellenic Physicists' Society.

Other Interests

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Peer reviewer of ASHRAE International Journal of HVAC & R Research.

Peer reviewer of ASME IMECE 2013 proceedings.

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