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ANALYSIS AND DEVELOPMENT OF REGENERATED DESICCANT
SYSTEMS FOR INDUSTRIAL AND AGRICULTURAL DRYING

Final Report, April—December 1977

By
D. V. Merrifield
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December 1977
Date Published

Work Performed Under Contract No. W-7405-ENG-26

Lockheed Missiles and Space Company, Incorporated
Huntsville Research and Engineering Center
Huntsville, Alabama

MASTER

U. S. DEPARTMENT OF ENERGY

Division of Energy Storage Systems

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FOREWORD

This document presents the results of a research and development study of a regenerated desiccant drying system, performed under Sub-contract No. 7296 to the Union Carbide Corporation at the Oak Ridge National Laboratory (ORNL), Oak Ridge, Tennessee. The work was funded by the U.S. Department of Energy (DOE) under the DOE-STOR Thermal Energy Storage Program. The Program Technical Monitor at ORNL is Dr. Herbert W. Hoffman. The Program Manager at DOE Headquarters is Mr. C. J. Swet. The study was performed by personnel of the Lockheed-Huntsville Research & Engineering Center, Huntsville, Alabama.

The authors wish to acknowledge the valuable contributions to this study made by Dr. S. M. Ko of Lockheed-Huntsville.

ABSTRACT

This report documents the results of a research and development effort to further develop and improve the design of a regenerated desiccant drying and dehumidification system which has more than twice the energy efficiency of conventional warm air dryers. The baseline application is for large-scale commercial grain drying, although several other applications have been identified such as space air conditioning and kiln drying of lumber. The system has excellent potential for providing a significant degree of industrial energy conservation while incurring payback periods of two to eight years to recover the added capital investment over conventional less-efficient drying methods. Although the intended energy source for the system is solar energy, the basic energy saving technique is the liquid desiccant drying process itself such that virtually any thermal energy input can be used. The use of solar energy permits further energy conservation if the additional capital investment is justified.

The major goal of the present study has been to improve the cost effectiveness of the desiccant regeneration system and establish significant design improvements over the regenerated desiccant drying concept developed in previous study phases. This goal has been achieved through the use of a straightforward evaporative technique for regenerating (reconcentrating) the lithium chloride desiccant solution, in place of using the previous packed column technique, and by using a steam condenser for heat recovery. The use of this technique resulted in a system cost reduction of nearly one-third and an efficiency improvement of approximately 10%, as well as a reduction in system size and complexity.

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Section 1

INTRODUCTION

This report documents the results of a research and development effort to further develop and improve the design of a regenerated desiccant drying and dehumidification system which requires less than one-half the energy input of conventional warm air dryers. The baseline application is for large-scale commercial grain drying, although several other applications have been identified such as space air conditioning and kiln drying of lumber. The system has excellent potential for providing a significant degree of industrial energy conservation with payback periods of two to eight years to recover the added capital investment over conventional less-efficient drying methods. Although the intended energy source for the system is solar energy, the basic energy saving technique is the liquid desiccant drying process itself such that virtually any thermal energy input can be used. The use of solar energy permits further energy conservation if the additional capital investment is justified.

The major goal of the present study has been to improve the cost effectiveness of the desiccant regeneration system and establish significant design improvements over the regenerated desiccant drying concept developed in the previous study phases (Refs. 1, 2 and 3). This goal has been achieved through the use of a straightforward evaporative technique for regenerating (reconcentrating) the lithium chloride desiccant solution, in place of using a packed column technique, and by using a steam condenser for heat recovery. These two approaches are compared schematically in Fig. 1-1. As a key part of the study, a laboratory-scale evaporative regeneration subsystem was built and tested. The results of this activity (Ref. 4) clearly demonstrated the technical feasibility and energy-efficiency of this approach to desiccant regeneration and heat recovery.

WITH EVAPORATOR AND AIR-COOLED CONDENSER

WITH PACKED COLUMN

Fig. 1-1 - Comparison of Desiccant Regeneration System Configurations

Section 2

SUMMARY

Major tasks carried out in the study were: (1) analysis and evaluation of improved regeneration techniques; (2) laboratory test and evaluation of an evaporative regeneration subsystem; (3) development of performance prediction models; (4) definition, design analysis, and economic analysis of a commercial-size baseline drying system; and (5) analysis of some specific applications.

The initial effort — to identify the most promising approach to an improved desiccant regeneration system — resulted in the selection of an evaporator/condenser concept in which water evaporated from the desiccant is condensed in a water-cooled or air-cooled condenser from which useful heat is recovered to provide additional drying capacity. This concept was then evaluated through a combined laboratory test program and math modeling analysis. A 5 lb/hr capacity laboratory-scale system was built and successfully tested (see Fig. 5-2). Its performance was essentially as predicted. A specific energy input of approximately 960 Btu/lb of water was obtained*, which is less than 50% of the energy requirement of conventional warm air dryers. The laboratory system also demonstrated the operating flexibility of the concept and its superior heat recovery capability.

The math modeling effort resulted in models with which to predict the system performance under various conditions and to determine principal design and operating parameters. It was determined that with the evaporation technique, an optimized specific energy input value of 900 Btu/lb including thermal losses is achievable, which is 45% of the energy requirement of a conventional system. A baseline, or reference system, was defined and sized to meet the requirements of a large commercial grain dryer. For

* This is 15% better than that obtained from the previous laboratory system which used a packed column for regeneration (see Ref. 1).

this system, with 6000 lb/hr of moisture removal capacity, optimum design parameters were determined and a cost estimate was made and compared with the estimated cost of an equivalent system using the previous packed column regeneration concept. The total cost of the system, including an absorption column, was found to be 31% less than for the previous system. The major reason for the lower cost is that the total required heat transfer surface is considerably less. Using the estimated cost of \$385,000, an economic analysis was made to show the payback and return-on-investment characteristics of the system considering its fuel cost saving capability over a typical conventional system. It was found that for a use-factor greater than about 20% (continuous operation 20% of the year) acceptable economics are obtained. For high use-factors, exceeding 75%, the payback period is less than 3 years with a return-on-investment of over 30%. For use-factors over 40 to 50%, the use of solar energy was found to be economically feasible. In addition to the baseline grain drying application, three other representative applications were analyzed: lumber drying, soybean processing, and space air conditioning, all having good economic potential.

The major conclusion of the study is that the regenerated desiccant drying concept has been significantly improved, in both cost and performance potential, through the use of the evaporative regeneration technique. The overall concept shows excellent potential as a major means of industrial energy conservation with a wide range of applications, and exhibits promising economic characteristics, particularly for high-use applications.

It is recommended that a modular prototype regenerated desiccant system be built and tested to provide a useful demonstration of this technology. A capacity of 100 lb/hr would be an appropriate size. The modular prototype system should incorporate the necessary operational flexibility (particularly with regard to the absorption column) to permit it to be tested over a range of conditions representing three or four application areas (such as for product drying and general air dehumidification). The initial step of this program should be a design requirements and integration analysis to identify, for the applications to be demonstrated, such specific items as performance requirements, any operational problems, sizing and design integration, and development/demonstration plans.

Section 3

BACKGROUND

Lockheed studies of regenerated desiccant drying techniques have been underway since early 1976. The principal objective has been to develop and demonstrate energy-saving drying techniques using desiccant-dehumidified air, with heat recovered from the desiccant regeneration process to provide additional drying capability. Figure 3-1 shows the basic process. The process differs from conventional desiccant dehumidification techniques (Ref. 5) in that useful heat is reclaimed from the regeneration process and can be used for additional drying energy or for other purposes. The result is that energy savings of over 50% can be realized with respect to conventional drying processes such as used for agricultural products and lumber, where the drying is conventionally accomplished using heated ambient air.

Fig. 3-1 - Simplified Schematic of Regenerated Desiccant Process with Heat Recovery

During the previous study phase (Ref. 1), in which a selected baseline system concept was evaluated analytically and experimentally to confirm its energy saving and economic potential, it became apparent that the drying method under study is sufficiently general to be applied to a wide range of drying applications in addition to the initial agricultural applications envisioned. These include drying of textiles, pulp, paper, wood, chemicals, rubber, etc., as well as commercial air conditioning. Section 8 of this report discusses three representative applications.

Other principal outputs from the previous study which provided an overall characterization of the concept are as follows:

- It was demonstrated experimentally and analytically that energy savings of 45 to 50% could be realized.
- The use of solar energy can be treated as an option providing for additional fuel savings if desired.
- The system is both energy efficient and capital intensive so that like other energy saving systems it must pay for its additional cost (over conventional systems) through the cost of the energy saved. This requires that the capital investment be minimized while retaining the energy saving characteristics, and of equal importance, requires that the system receive sufficient annual use to realize appreciable energy savings.

The present study phase, documented in this report, has been addressed primarily to cost effectiveness improvement, and has resulted in a method for substantially reducing the system capital cost and increasing its application potential. As a result the regenerated desiccant drying concept shows excellent potential for use as a practical and cost effective means of energy conservation.

Section 4

ANALYSIS AND EVALUATION OF IMPROVED REGENERATION TECHNIQUES

In the preceding study phase, various system concepts were studied for regenerating (or reconcentrating) the lithium chloride desiccant solution after it has absorbed water during the product drying process. The selected process, which was subjected to a thorough evaluation (including laboratory-scale testing, math modeling, and economic analysis), used a packed column and moist air condenser for regeneration. This system is shown in Fig. 4-1. Although this system showed good promise, it was recognized that there was good potential for reducing its capital cost and simplifying the design to improve its application potential. It appeared that the most likely means for doing this was to either decrease the condenser cost through the use of enhanced heat transfer surfaces or to use an evaporator instead of a packed column.

4.1 ENHANCED SURFACE CONDENSER EVALUATION

An investigation was made to determine whether enhanced surface condenser tubes could be used to decrease the heat transfer area requirement of the condenser shown in Fig. 4-1. The initial step was to survey published literature and other sources on enhanced surface techniques to determine which particular geometry offers the best combination of improved condensation heat transfer and reasonable material cost (Refs. 6 through 9). Very little information was found to be available since the use of enhanced surfaces in promoting steam condensation heat transfer is relatively recent. The most complete and informative publication is by Carnavos (Ref. 6), who classifies enhanced surfaces as those with permanent surfaces as those with permanent surface treatment (hydrophobic coatings or textured or formed surfaces) such that heat transfer coefficients are orders of magnitude greater than conventional surfaces. Typical surfaces for promoting condensation include fluted tubes and spirally or helically grooved tubes. Based on the favorable performance of

Fig. 4-1 - Desiccant Drying Concept with Packed Column and Moist Air Condenser for Desiccant Regeneration

fluted tubes as reported on by Carnavos, and obtained by the Oak Ridge National Laboratory (Ref. 7), this type of enhanced surface was selected for test and evaluation within a desiccant cooled condenser.

It was determined that the most promising geometry would be that of a double fluted tube with relatively high flute density. Several condenser tube manufacturers were contacted but no fluted tubing was available. One tubing manufacturer who manufactures helically grooved tubing stated that they had previously manufactured a corrugated (fluted) tube but were discontinuing it due to high manufacturing cost, reportedly double that of their helically grooved tubing. A decision was made to fabricate a set of fluted copper tubes in-house for use in a test condenser, using a die forming process. This permitted attainment of the desired flute geometry and tube diameter as shown in Fig. 4-2. A unique die was used, which has 12 adjustable pins such that when the tubing is drawn through the die, the pins, which bear on the outside of the tubing compressively form the longitudinal flutes. Twenty-seven fluted tubes each 16 in. long and 0.4 in. o.d. were fabricated and assembled within a test condenser shell. The test condenser was designed so as to have the same condensing side surface area (3.3 ft^2) as the smooth tube condenser used with the experimental packed column.

Fig. 4-2 - Cross Section of 12-Fluted Condenser Tube,
Approximately Twice Full Scale

The performance of this condenser was then compared with the performance of the smooth tube condenser to see if there was an improvement in condensation rate. The method used was to operate each condenser with the packed column regeneration system used in the previous contract phase. For each of the two tests, the same condenser inlet conditions were used and the condensation

rate was determined. In this way, both condensers received virtually the same inlet flow of hot air containing moisture removed from the desiccant flowing through the packed column.

Table 4-1 summarizes the test results. It was found that there was no measurable difference in condensation rates for the two condensers. This result was not totally unexpected since an analysis had indicated that the presence of air (and other noncondensables) can severely retard the water

Table 4-1
RESULTS OF CONDENSER EXPERIMENT

Parameter	Smooth Tube Condenser	Fluted Tube Condenser
Air inlet temperature, average (F)	207	205
Air outlet temperature, average (F)	75	75
Desiccant inlet temperature, average (F)	74	75
Desiccant outlet temperature, average (F)	126	129
Condensation rate, average (lb/hr)	0.95	0.95

vapor condensation process since the water vapor must diffuse through the air to condense on the heat transfer surface. It had been hoped that in spite of this limitation the superior drainage characteristics (Ref. 8) of the fluted configuration would still provide some degree of heat transfer enhancement. Other significant factors involved, which were inherent in the overall nature of the test, were as follows: the flow rate of moist air from the packed column (which was at its maximum value of approximately 5 lb/hr) was such that neither condenser was utilizing its full area. Also the desiccant flow through the condenser shell, which is preheated in the condenser before it flows into the heat exchanger and packed column is process-constrained such that the velocity in the shell is less than 0.01 ft/sec. For this reason, the desiccant side heat transfer coefficient is probably the controlling factor.

It was evident therefore that the test was inconclusive in determining whether there was an improvement in heat transfer within the condenser.

It was concluded, however, that the use of enhanced surfaces will probably not be cost effective for a packed column regeneration system, but should be strongly considered for a desiccant regeneration process that does not depend on air as a moisture transport medium.

4.2 IMPROVED REGENERATION TECHNIQUE SELECTION AND PROCESS DEFINITION

4.2.1 Selection

An analysis was made to determine what improvements in hardware or process techniques (over the previously established baseline system) should be selected for detailed evaluation.

Principal selection criteria were:

1. Potential for cost improvement without efficiency reduction
2. Potential for system simplification (to reduce cost and improve reliability and maintenance)
3. Minimum development risks due to technical problems
4. System flexibility in terms of permitting the system to be applied to a wide range of drying and dehumidification applications, and
5. Ability to operate over a wide range of regeneration temperatures and pressures.

Based on these criteria, it was decided to use an evaporator instead of a packed column. An evaluation plan was prepared to consist of: (1) laboratory-scale testing; (2) performance modeling; (3) full-scale system design and trade-off analysis; (4) cost effectiveness evaluation; (5) solar energy analysis, and (6) application analysis.

4.2.2 Process Definition

Figure 4-2 shows the selected evaporative process configuration. The nominal operating parameters were established as follows:

- Evaporator temperature 250 F
- Evaporator pressure 7.5 psia
- Ratio of evaporator feed rate
to water evaporation rate 6:1
- Desiccant concentration change 37 to 44%
- Condensing steam saturation
temperature 180 F
- Condenser coolant (water) inlet/
outlet temperature 150/160 F
- Temperature of cool desiccant
to heat exchanger 90 F

The major design differences between this regeneration process and the previous one using a packed column and moist air condenser are that: (1) heat transfer rates in the condenser are significantly improved since virtually pure steam is condensed rather than moist air^{*}; (2) the ratio of desiccant flow rate to regeneration rate is much lower, permitting a smaller heat exchanger and associated piping; (3) the equipment is physically smaller and more compact; and (4) only two components – the evaporator and the heat exchanger – are exposed to desiccant solution, as opposed to five in the stripping column process.

*

Permitting the use of enhanced surface condenser tubes.

Fig. 4-2 - Evaporative Regeneration Process (with water-cooled condenser)

Section 5

LABORATORY SYSTEM TEST AND EVALUATION

To assist in evaluation of the evaporative regeneration technique, a laboratory-scale system using lithium chloride desiccant was built and tested. The specific purposes of this effort were to:

- Measure overall performance and energy input characteristics of the desiccant regeneration system
- Demonstrate the process and determine its operating flexibility
- Evaluate problem areas, and
- Obtain design criteria for a full-scale system.

5.1 DESCRIPTION

The system was sized for a regeneration rate of 5 lb/hr, which is seven times the capacity of the previous laboratory-scale regeneration system which used a packed column. A schematic of the system is shown in Fig. 5-1. Principal design features are as follows:

1. The evaporator is electrically heated.
2. The condenser uses enhanced surface tubes (fluted configuration) for increased heat transfer rates.
3. A water circulation loop is used to cool the condenser and heat a stream of drying air.
4. The principal operating mode is: "once-through" or continuous regeneration, wherein the desiccant stream is completely re-concentrated in one pass through the evaporator.

Nominal performance parameters were established as follows:

Evaporation (regeneration) rate	.5 lb/hr
Evaporator temperature	.250 F
Evaporator pressure	.7.5 psia
Condensing temperature	.180 F

Dilute desiccant feed rate	30 lb/hr
Dilute desiccant concentration [*]	38%
Heat input to evaporator.	7500 Btu/hr (max.)
Condenser water inlet/outlet temperature . .	150/160 F
Specific energy input	1380 Btu/lb condensate

Principal physical characteristics are as follows:

- Evaporator height x diameter 20 in. x 3 in.
- Condenser height x diameter 24 in. x 3 in.
- Condenser area 3.3 ft²
- Heat exchanger surface area 1.0 ft²

Photographs of the completed system are shown in Fig. 5-2. The system is shown installed in place of the previously tested packed column regeneration system, and is interfaced with the previously used air heating loop and desiccant supply. The evaporator and condenser are both equipped with flange-mounted sight glasses to permit visual observation of fluid behavior. Instrumentation is provided to permit measurement of temperature, pressures, heater power flow rates and desiccant concentration. To permit nominal system operation at subatmospheric pressure, a condensate receiver and vacuum pump are used.

Operation: In normal operation, dilute desiccant is fed to the evaporator through the heat exchanger which preheats it using the concentrated desiccant outlet stream. The evaporated water, in the form of initially superheated steam, flows through a separator and demistor and into the condenser. The condensing steam then indirectly heats a stream of air by means of the water circulation loop and water/air heat exchanger. The system therefore produces, as outputs, a stream of concentrated desiccant and a stream of heated air.

^{*}Percent by weight lithium chloride in water.

Fig. 5-2a - Laboratory Evaporative Regeneration System

**Fig. 5-2b - Laboratory Evaporative Regeneration System
Before Addition of Insulation**

5.2 INITIAL TESTING

Initial activities consisted of observing the general behavior of the system, experimenting with various operating conditions, and correcting problems. Early in the testing it was found to be desirable to operate primarily in a "one pass" mode in which the desiccant is reconcentrated by one pass through the evaporator, rather than by continuous circulation to and from a single storage tank. This method is more appropriate for system applications with continuous drying and regeneration and permits the attainment of steady state operating conditions. This method was used in most of the testing.

In the initial tests it was found that during vacuum purging of air from the system there was entrainment of desiccant in the steam flow to the condenser. This problem was eliminated by the addition of 3-in. diameter separation chamber and mist removal section between the evaporator and the condenser, such as is the practice with commercial-size evaporators.

Most of the initial test runs were made at subatmospheric pressure and with constant heat input to the evaporator. During these runs the regeneration rate was approximately 4.5 lb/hr and the specific energy input (total energy required to evaporate each pound of water) was within 10% of the predicted value of 1380 Btu/lb of water evaporated. During this time considerable effort was spent in determining the optimum method of removing noncondensibles, (primarily air) from the condenser in order to maintain the desired condensation rate and control the system operating pressure. It was found that the best method was to continuously remove the air through a needle valve to permit fine control of the flow rate. By using this method, which requires only a very small continuous pumping rate (less than 0.005 lb/min) it was possible to maintain the evaporator/condenser pressure within ± 0.25 psia, resulting in good control of system temperatures. The need for air removal is due to slight leakage at some mechanical joints, and to a lesser degree dissolved air and small bubbles in the desiccant feed lines. The air build-up must be controlled since it reduces the evaporation and condensation rates. It was originally believed that the presence of dissolved air in the desiccant

supplied to the evaporator could create a possible problem, however it was found that the solubility of air in the lithium chloride desiccant solution is less than 10 ppm by weight at a temperature of 90 F.

5.3 TEST DATA AND DISCUSSION

Figure 5-3 shows the results of a test conducted with the following conditions:

Evaporator pressure (P_e)	= 6.1 psia
Heat input (Q_e)	= 7167 Btu/hr
Desiccant feed rate (m_1)	= 44 lb/hr
Desiccant feed concentration (C_1)	= 38%
Water inlet temperature to condenser (T_{W1})	= 142 F (average)

During this test evaporator temperature was nearly constant at 236 ± 2 F, versus a predicted temperature of 238 F. The outlet concentration of the desiccant was 43% versus a predicted value of 43.7%. The average water removal rate, m_w , was 4.9 lb/hr. Using the average rate of heat input of 7167 Btu/hr, the specific energy consumption, per pound of condensate, was

$$E = \frac{Q_e}{\dot{m}_w} = \frac{7167}{4.9} = 1462 \text{ Btu/lb}^*$$

This compares with a predicted value of 1370 Btu/lb calculated for these test conditions and is seen to be within 7%. The difference is probably due in large part to thermal losses to ambient. During the test the condenser performance was as follows:

Condensing temperature	= 171 F
Water inlet temperature	= 142 F (avg.)
Water outlet temperature	= 157 F (avg.)
Heat Transfer	= 4900 Btu/hr

*Of this 1462 Btu/lb, approximately 1000 Btu/lb was recovered in the condenser and used to heat a stream of drying air to 130 F.

Fig. 5-3 - Test Data for Operation at 6.1 psia

Based on these data the overall heat transfer coefficient for the condenser during this test was found to be $U = 74 \text{ Btu/hr-ft}^2$. This is over ten times higher than the estimated U value obtained when the same condenser was operated previously (Section 4.1) with humid air instead of steam.

The heat exchanger, whose function is to preheat the cold incoming desiccant using the hot outgoing desiccant from the evaporator, performed satisfactorily. As shown in Fig. 4-2 the cool desiccant was heated to within approximately 14 deg of the predicted temperature of 203 F.

In summary, this test demonstrated very satisfactory performance of the experimental regeneration system and was the first test in which good control of evaporator pressure and temperature was obtained.

Four Hour Test at 7.5 psia: The test which best demonstrated the successful performance of the experimental system was one in which the evaporator operating conditions were very close to those established prior to fabrication of the system. Figure 5-4 shows the principal temperatures and the condensation rate for this test, in which more than four hours of near-steady-state data were obtained.

The operating conditions were:

Evaporator pressure	= 7.5 psia
Heat input	= 7170 Btu/hr
Water inlet to condenser	= 145 F (avg.)
Desiccant feed rate	= 45 lb/hr
Desiccant feed concentration	= 38.5%

The evaporator temperature was 245 ± 3 F versus a predicted temperature of 244 for these test conditions. The measured water removal rate was 5 lb/hr giving a specific energy input of $E = 1434 \text{ Btu/lb}$. This is 6% higher than the predicted value of 1353 Btu/lb. The temperature fluctuations shown

Fig. 5-4 - Four-Hour Test at 7.5 psia

in Fig. 5-4 for the desiccant preheat temperature, the condenser water outlet temperatures, and the desiccant outlet temperature were caused by manual regulation of desiccant flows and not by system instability. The average value of the preheat temperature was within 5 F of the predicted value of 210 F. The desiccant outlet temperature was 5 F of the planned outlet temperature of 100 F. The condenser water outlet temperature was within 2 F of the planned outlet temperature of 160 F. The condensing temperature (not shown on Fig. 5-4) was 180 ± 1 deg F, corresponding to a pressure of 7.5 psia. In this test as in all other tests there was no measurable pressure drop from the evaporator to the bottom of the condenser. The average U value for the condenser was found to be $56.5 \text{ Btu/hr-ft}^2\text{-F}$. In summary, this test provided the most successful demonstration of the performance of the system.

In a previous test of two hours duration which was conducted with approximately the same evaporator conditions, a water removal rate of 5.14 lb/hr was obtained, giving a specific energy value of 1394 Btu, the lowest achieved. In this case a lower desiccant flow rate was used, which should give slightly better efficiency. It was found, however, that at this lower flow rate there was some tendency for crystallization and plugging to occur in the desiccant outlet lines. This was judged to be due to concentration gradients in the laboratory evaporator caused by the lower flow rates. It was learned that commercially available evaporators avoid this problem by using a simple recirculation technique.

Low Temperature Test: At the completion of the tests performed at nominal evaporator temperatures of 240 to 250 F, a test was made to demonstrate the capability of the system to regenerate desiccant at approximately 200 F, which involves operating at a pressure of near 3 psia. For this test the evaporator was operated at a heat input level of 3320 Btu/hr. The average evaporator temperature was 201 F and the average water removal rate was 2.3 lb/hr, giving a specific energy of 1440 Btu/lb which is very close to the corresponding energy values obtained at the higher operating temperatures.

Other values measured during the test are given as follows (average):

Evaporator pressure	= 2.9 psia
Desiccant flow rate	= 16.5 lb/hr
Condenser water outlet temperature	= 100 F
Preheat temperature	= 160 F
Steam condensing temperature	= 140 F

This test demonstrated the capability of the system to operate at low temperature while maintaining virtually the same efficiency as obtained at higher temperature levels.

Condenser Test: A test run was made to determine overall U values for the condenser and examine the effects of internal air (and other non-condensable gases) on condenser performance. This test, run at 14.7 psia, also demonstrated the operation of the system at atmospheric pressure. The method used in running this test was to maintain the condenser vented to ambient at the bottom of the tubes at all times while gradually heating up the system, thus allowing air to be displaced by steam. During this gradual heat-up process, the air blower used to cool the condenser circulating water was on a low speed, permitting the condenser cooling water to be at a higher temperature and reducing the ΔT across the condenser tubes. This was continued until a small amount of steam was observed venting from the condenser at which time the cooling-rate was stabilized with the condenser essentially free of air. Using the measured condensation rate and the fluid inlet/outlet temperature, the condenser heat load and log mean temperature difference, ΔT_m , were calculated and overall U values in $\text{Btu/hr-ft}^2\text{-F}$ were determined. The results are plotted in Fig. 5-5 which shows U versus the average cooling water temperature. This demonstrates how the heat transfer within the condenser increases sharply as air is removed and the effective heat transfer area increases. It is believed that limiting factor in the condenser heat transfer rate was the water side coefficient since the water velocity (in

Fig. 5-5 - Condenser Overall Heat Transfer vs Average Coolant Temperature

the condenser shell) was estimated to be less than 0.1 ft/sec, giving a correspondingly low Reynolds number. Even so, the overall U values obtained were up to 20 times larger than that obtained during operation of the same condenser with a packed column in which case the major resistance was the air film on the tubes. In summary this test demonstrated not only the increasing heat transfer performance of the condenser as air is removed, but in a more general sense demonstrated the ability of the regenerative evaporation concept to provide for heat recovery at a relatively high average temperature level, which in this case was 211 F.

5.4 ASSESSMENT OF LABORATORY TEST RESULTS

Laboratory experience with the 5 lb/hr evaporative regeneration system demonstrated the feasibility, practicality, and performance potential of this particular desiccant regeneration method. The system performed essentially as predicted and no problems were found which would prevent the large-scale use of such a system. The measured specific energy input was typically 1440 Btu/lb of water removed (including heat losses). Considering the 50% additional drying capability achievable with the recovered heat, this equates to 960 Btu/lb of water removed from a product being dried.

Removal of air and other noncondensable gases from the condenser was not a problem and it was found that the pump power involved would be less than 0.5% of the thermal input to a full-size system.

It was shown that operation can be accomplished over a wide range of temperature levels and pressure (subatmospheric and atmospheric) without significant efficiency variations, and that steam condensation is an efficient, flexible means of recovering heat from the desiccant regeneration process.

In comparison with the previously tested 0.75 lb/hr regeneration system which used a packed column and moist air condenser, the evaporative system was found to be more compact, less complicated, and more energy-efficient.

Section 6

PERFORMANCE PREDICTION MODELS

Analytical models were developed to assist in prediction and determination of regeneration system performance and design parameters. Two efforts were undertaken – one to predict the equilibrium performance of a fixed design configuration consisting of an evaporator, heat exchanger, and water-cooled condenser, the other to determine system design parameters for specified performance requirements.

6.1 EQUILIBRIUM PERFORMANCE PREDICTION MODEL

A computer simulation model was developed to handle the case in which specified design geometry, heat transfer correlations, and operating parameters are given to describe the condenser, evaporator, and heat exchanger. The program then determines the equilibrium performance of the system, such as the evaporator and heat exchanger temperatures, the condensation rate and the outlet concentration of the desiccant. The following summarizes the operation of the model and illustrates its use.

For a given desiccant concentration and operating temperature, the rate of water removal from the desiccant in the evaporator is computed from water-desiccant equilibrium data. The steady state evaporator temperature at a given concentration is determined by an iterative procedure which matches the evaporation rate and the condensation rate in the condenser at given input system design conditions. For a system where desiccant is continuously reconcentrated by circulation to and from a single storage tank, the tank desiccant concentration becomes a function of time and therefore the evaporator temperature slowly increases with increasing incoming desiccant concentration to the evaporator. At each point in time, the system is considered to be in equilibrium, even though conditions are slowly changing.

Figures 6-1 and 6-2 show the results of a simulation run to predict the performance of a configuration similar to that of the laboratory system. For initial and final desiccant concentrations of 36 and 43%, the evaporator temperature ranges from 240 to 260 F. The corresponding concentrations of regenerated desiccant from the evaporator range from 45 to 53%. When regeneration is accomplished in a continuous manner such as with the use of two storage tanks as shown in Fig. 6-3, the incoming desiccant concentration to the evaporator is always at the initial concentration and therefore steady state operation of the evaporator can be achieved, which relaxes process control requirements. This is the principal method which was used in the laboratory testing. Typical operating conditions for a steady state system are presented in Fig. 6-3.

The predicted specific energy requirement of the system for the conditions shown in Fig. 6-3 is 1380 Btu/lb condensate or 920 Btu/lb moisture removed from the product being dried.

6.2 DESIGN REQUIREMENTS MODEL

A model was developed to determine system design parameters for the evaporator and heat exchanger. Specifically, the following are determined:

- Evaporator heat load
- Evaporator pressure
- Heat exchanger load
- Desiccant preheat temperature (into evaporator)
- Desiccant outlet concentration
- Specific energy input requirement, and
- Required heat transfer areas.

These are determined as a function of:

- System regeneration rate (condensation rate)
- Desiccant properties

Fig. 6-1 - Performance Simulation of Evaporative Regeneration (Time-Dependent, One Tank Mode)

Fig. 6-2 - Performance Simulation-Evaporative Regeneration
(Time Dependent - One Tank Mode)

Fig. 6-3 - Regeneration System Performance Simulation - Typical Operating Conditions

- Heat transfer coefficients
- Desiccant feed rate and concentration, and
- Desiccant feed temperature and return-temperature.

In Section 7 it will be shown how this model was used to perform parametric studies showing the principal design/performance characteristics of a full-scale regeneration system and the principal design tradeoffs involved.

Section 7

BASELINE SYSTEM DEFINITION AND ANALYSIS

Based on the results of the laboratory testing, system math modeling, and overall system analysis of the evaporative regeneration concept, a baseline or reference system was defined. This system was sized to handle the moisture removal requirements of a commercial grain drying column with a minimum capacity of 1000 bushels per hour. This requires a total drying rate of approximately 6000 lb of water per hour, and a desiccant regeneration rate of 4000 lb/hr. For this reference system, the following were accomplished:

- A parametric and tradeoff analysis was made to seek optimum design parameters
- A cost estimate was made and compared with the estimated cost of a system using the previous packed column regeneration technique
- An economic analysis was made to determine payback and return-on-investment characteristics; and
- A solar energy utilization assessment was made.

7.1 SYSTEM DEFINITION

The 6000 lb/hr reference system is shown schematically in Fig.7-1. Shown also, for completeness, is the absorption column. The desiccant regeneration system consists of three principal components — the evaporator, heat exchanger, and a combined condenser and air heater. Use of the steam condenser for direct air heating instead of using a water-cooled condenser and water/air heat exchanger reduces cost and simplifies the system. This approach is equivalent to that used in the laboratory system configuration but eliminates the need for the water circulation loop components and reduces the required amount of heat transfer surface.

Fig. 7-1 - Regenerated Desiccant Grain Dryer with Evaporative Regeneration

For a general drying application such as depicted in Fig. 7-1, the system shown is believed to be the lowest cost approach because of its inherent simplicity. The system operates in a continuous regeneration mode, requiring no desiccant storage; during operation the dilute desiccant from the absorption column has water evaporated from it at the same rate as moisture from the grain is being removed from the air passing through the absorption column. At the same time, the steam from the evaporator is used to heat ambient air and perform initial drying of the wet grain before it passes through the closed cycle (desiccant) drying section. The drying capacity of the closed cycle portion is 4000 lb/hr of moisture, and the nominal capacity of the pre-drying portion (which is conventional in nature) is 2000 lb/hr. The evaporator is heated by either fossil fuel energy, solar energy or both, using steam, hot water, or hot combustion gas as the heating medium.

It should be noted that this reference configuration is general in nature in that its function is to supply heated and dehumidified air. For this reason, as will later be discussed, it is also applicable to other low-temperature drying applications.

Shown in Fig. 7-2 is a photograph of a typical commercial grain drying installation. The unit shown is a two-stage continuous flow column-type dryer in which the grain flows slowly downward in two 14 in. thick x 20 ft wide columns. The heated drying air flows outward through and perpendicular to each column of grain. The columns are formed by a perforated sheet metal structure.

7.2 PARAMETRIC AND TRADEOFF ANALYSIS

The design requirements model described in Section 6.2 was used to determine, for the reference system, principal evaporator/heat exchanger parameters and their sensitivity to evaporator design temperature and desiccant feed rate. In each case, the evaporation rate was held constant at 4000 lb/hr and the incoming desiccant concentration was held constant at 38%*.

*This concentration is believed to be near optimum from an overall system viewpoint.

Fig. 7-2 - Typical Commercial Grain Drying Installation

Figure 7-3 shows the sensitivity to evaporator temperature, where the desiccant feed rate is held constant at 24,000 lb/hr. Note that although the preheat temperature, T_p , the heat recovery temperature T_s , and the evaporator pressure, P_E , are strong functions of evaporator temperature, the evaporator heat input, and consequently the specific energy input, E_o , are not, and increase only 5% as the evaporator temperature is increased from 200 to 300 F.* Note also from Fig. 7-3 that the heat exchanger load, Q_{HX} , is doubled when the evaporator design temperature is increased from 200 to 300 F.

Figure 7-4 shows the sensitivity to the design value of the desiccant feed flowrate over a range of 20,000 to 30,000 lb/hr (4.8 to 7.2 gpm) with the evaporator temperature held constant at 250 F. The specific energy input is seen to be very insensitive to flow rate although the heat exchanger load increases by 61%. The other parameters shown are only moderately effected by flow rate.

By making use of these design characteristics, an analysis was made to show how the total heat transfer area requirement depends on evaporator temperature, the most important design variable. The result is shown in Fig. 7-5. A principal assumption is that the energy supply temperature to the evaporator is 297 F (50 psig steam). The desiccant feed flow is 24,000 lb/hr. Representative heat transfer coefficients are used for the condenser, heat exchanger and evaporator. Since system cost is roughly proportional to total heat exchange area, this plot shows that the evaporation temperature should be approximately 270 F, with a corresponding pressure of 10 psia (primarily to minimize the condenser area). This result was found to be quite insensitive to the assumptions made, except in the case of energy supply temperature, where an increase in this temperature results is an increase in the optimum evaporator temperature. An interesting characteristic shown in Fig. 7-4 is that the heat exchanger area is relatively insensitive to evaporator temperature, even though Fig. 7-3 showed that Q_{HX} is a strong function of evaporation temperature. This is caused by the offsetting effect of the log mean temperature difference, LMTD, which also determines the required area. Also investigated was the effect of increasing the design value of the cold end ΔT of

*The specific energy, E_o , shown in Figs. 7-3 and 7-4 is based on the total water removal rate of 6000 lb/hr.

Fig. 7-3 - Sensitivity to Evaporator Design Temperature

Fig. 7-4 - Sensitivity to Desiccant Feed Flow

Fig. 7-5 - Regeneration Subsystem Heat Transfer Areas vs Evaporator Temperature

the heat exchanger to decrease its size. It was found that increasing the ΔT from 10 F to 20 F decreases the heat exchanger area by about 40% although the optimum evaporation temperature remains in the range of 260 to 280 F. The specific energy input requirement is increased by only 2%.

7.3 REFERENCE SYSTEM DESIGN PARAMETERS

Based on knowledge of the system characteristics and major tradeoffs, a set of near-optimum design/performance parameters was chosen and used in estimating system cost and economic characteristics. These are shown in Table 7-1. A conceptual design layout of the regenerated desiccant grain drying application is shown in Fig. 7-6. A corresponding isometric view is shown in Fig. 7-7. The grain column which is depicted has a height of approximately 38 ft.

7.4 COST EFFECTIVENESS ANALYSIS

This section presents an analysis of the economic characteristics of a commercial-scale regenerated desiccant drying system with evaporative regeneration. The system includes the absorption drying components with the regeneration subsystem. In addition to the estimated capital cost of this reference system, a major factor included in the analysis is the "use factor" or the percent of time that the system is used continuously at rated capacity. The specific points addressed are; (1) estimated capital cost; (2) comparison with the cost of the previous system which used a packed column for desiccant regeneration; (3) economic payback period versus use factor; and (4) return-on-investment versus use factor. Since the configuration of the reference system is such that it can be used for applications other than grain drying, the results of this analysis show the economic potential of the system for similar applications with use factors ranging from about 10 to 100% and gives a useful perspective on the applicability of the system. A good example is lumber drying, with a use factor of 80 to 90%.

The reference system analyzed has a regeneration capacity of 4000 pounds per hour of water. By using recovered energy the total drying

Table 7-1
OPTIMUM DESIGN/PERFORMANCE PARAMETERS
FOR 6000 LB/HR REFERENCE SYSTEM

Evaporator:

Evaporation rate	4000 lb/hr
Temperature	260 F
Pressure	7 psia
Desiccant inlet temperature (preheat temperature)	215 F
Desiccant feed flow	24,000 lb/hr
Concentration in	38%
Concentration out	45.5
Heat input	5.1 MBtu/hr

Heat Exchanger:

Hot desiccant inlet temperature	260 F
Hot desiccant outlet temperature	110 F
Cool desiccant inlet temperature	90 F
Cool desiccant outlet temperature	215 F
Heat load (duty)	1.9 MBtu/hr

Condenser:

Steam inlet temperature	260 F
Steam condensing temperature	185 F
Air inlet temperature	70 F
Air outlet temperature	120 F
Heat load (duty)	4.0 MBtu/hr
Air flow	74,000 scfm

Top View

Elevation View

Fig. 7-6 - Conceptual Layout of Regenerated Desiccant Grain Drying Installation Showing Major Components

Fig. 7-7 - Conceptual Drawing of Regenerated Desiccant Grain Drying Installation

capacity becomes approximately 6000 pounds per hour. For grain drying, this corresponds to a drying capacity of from 1000 to 2000 bushels per hour depending on grain wetness. For lumber drying, for example, this corresponds to approximately 1500 to 2000 board feet per hour of lumber.

Capital Cost Estimates

For the reference system using lithium chloride desiccant, the total capital cost was estimated to be \$385,000. This total is based largely on cost data obtained from equipment manufacturers and is believed to be somewhat conservative. This includes material and labor costs associated with the evaporator, heat exchanger, condenser/air-heater, absorption column, and auxiliary equipment. For the previous system using a packed column, the total equivalent cost was estimated at \$560,000. Therefore, the use of the evaporator concept provides nearly a one-third cost reduction over the packed column system studied in the previous contractual phase. The principal reason is that heat exchanger area requirements are much smaller. This is shown in Fig. 7-8 which compares the configurations and heat transfer area requirements of the two systems. Given in Table 7-2 is a percentage cost breakdown for the two types of desiccant systems.

Economic Calculations

For the reference system, calculations were made to determine payback periods and equivalent returns on investment. The payback period is defined as the incremental capital investment divided by direct cost savings of the first year, taken to be the value of the energy saved minus the operating and maintenance cost (O&M). The return-on-investment, (ROI) calculation gives the percentage interest rate on the capital investment, where the annual cash flow which amortizes the investment is equal to the gross fuel savings less the total of O&M, depreciation, and income taxes. The basis for the calculations is given as follows:

1. The estimated incremental capital investment for the regenerated desiccant system is \$385,000 (fossil fuel energy source).

WITH EVAPORATOR AND AIR-COOLED CONDENSER

WITH PACKED COLUMN

Fig. 7-8 - Comparison of Desiccant Regeneration System Configurations and Heat Transfer Areas

Table 7-2

**PERCENT COST BREAKDOWN COMPARISON
FOR REGENERATED DESICCANT SYSTEMS**

With Evaporative Regeneration:	Percent
Evaporator unit	28
Heat exchanger	4
Condenser	6
Absorption columns	19
Installation and auxiliaries *	<u>43</u>
	100
With Packed Column Regeneration:	Percent
Packed column (regenerator)	9
Condenser	19
Desiccant heater	3
Heat exchanger	7
Air heater	10
Absorption columns	13
Installation and auxiliaries *	<u>39</u>
	100

* Includes allowance for pumps, piping, controls and foundations.

2. The rate of energy savings is based on a specific energy consumption of 900 Btu/lb water removal from the product being dried, compared to 2000 Btu/lb for a conventional dryer, giving a savings of $(2000 - 900) \times 6000$ or 6.6 MBtu/hr.
3. Fuel cost is taken to be \$5/MBtu, giving a cost savings of \$33/hr or 23,100/month.
4. Depreciation is by the straight line method over 20 years; the federal tax rate is 50%; neither fuel cost escalation nor general inflation is included; no tax or investment incentives are included; annual O&M is assumed to be 2% of the capital investment.

Fig.7-9 shows the results of the payback and ROI calculations, plotted versus the use factor, which is the percent of time the system is used continuously at rated capacity. For example, at a 20% use factor, which could apply to a commercial grain drying operation, the payback period is 7.5 years and the corresponding return on investment is 6.5%. For a grain processing application such as soybean drying, or for a lumber drying application the use factor will exceed 80%, in which case the payback time will be less than two years with a return on investment of over 30%. It should be noted that the ROI results are in current dollars, and do not include fuel cost increases and general inflation . Therefore, the results shown are conservative.

It is evident from Fig.7-9 that the use factor is a major determinant of economic feasibility. Factors which are also significant but of somewhat lesser influence are the tax rate and the fuel cost. The principal conclusions from the data shown is that the regenerated desiccant system has excellent potential for fossil fuel conservation with acceptable economics, but only if the equipment is utilized for at least 15 to 20% of the time. Conversely, if the use factor is very low, there will be no economic justification. Therefore it is important that, in addition to grain drying applications, other applications, such as lumber drying, for example, be also pursued.

Solar Energy Utilization: The payback and ROI data of Fig. 7-9 are for a regenerated desiccant drying system using a fossil fuel energy input and having an energy use rate equal to 45% of that of a conventional heated-air dryer. If the system uses solar collectors to provide a significant portion of the energy requirement (the thermal input to the evaporator) additional fuel conservation is realized but at the expense of an additional capital investment. In this case, the payback periods are extended and the total return-on-investment is reduced. For this reason it is necessary to have a use factor greater than approximately 40% in order to have a reasonable return on investment. For the 6000 pound per year reference system, an analysis was made which showed that for a solar regenerated desiccant drying system in which solar collectors are used to supply 23% of the thermal input requirement of the evaporator and the

Fig. 7-9 - Economic Characteristics of Regenerated Desiccant Drying System with 6000 lb/hr Moisture Removal Capacity

Table 7-3
ECONOMIC CALCULATIONS FOR SOLAR
REGENERATED DESICCANT DRYER

<u>Basis</u>	<ol style="list-style-type: none"> 1. Energy collection rate for concentrating collectors with Atlanta, Ga., weather conditions: 840 Btu/ft² per day, annual average 2. No solar energy storage 3. Collector cost: \$12/ft² 4. Collector area: 37,300 ft² 5. Value of solar energy supplied: \$4750/month 6. Percent solar energy supplied: 23% 	
<u>Results</u>		
Use Factor (%)	Payback Period (yrs)	Return on Investment (%)
40	7.2	7.1
50	5.6	9.7
60	4.6	11.9
70	3.9	14.1
80	3.4	16.3
90	3.0	18.4

use factor is 90%, the total payback period is approximately 3 years and the return on investment is approximately 18%. Table 7-3 shows the basis for the calculations and the results obtained. These data indicate that even with the additional cost of a solar collection system,* attractive economics can be obtained. It is clear however, that for lower use factors such as are associated with commercial grain drying, the additional cost of solar collectors does not appear economically justified.

*In estimating the solar collector cost it was considered reasonable to use a cost corresponding to a collector system cost payback period of approximately 10 years, reflecting the favorable effects of probable future tax and depreciation incentives for the use of solar energy. Using typical Atlanta, Ga., weather conditions, this results in a collector cost of \$12/ft², 20-40% less than typical present day installed costs.

Summary of Economic Analysis: In summary, the economic analysis given in this section shows that the regenerated desiccant drying system exhibits highly promising economic characteristics, particularly for high use applications. For commercial grain drying utilization, the analysis has shown that the system (non-solar) should be used for at least 10 weeks of the year on a continuous basis (ROI = 6%). Since this is longer than many grain dryers are operated, then the application to grain drying may require modified (extended) use procedures. An approach to extending the utilization time is to make use of the system for grain conditioning purposes (final drying and moisture control) which, at large grain facilities, is typically required for two to three months after the initial drying operations. This could decrease the payback period to less than five to six years.

7.5 SOLAR ENERGY UTILIZATION ASSESSMENT

As a part of this study, which has been concerned primarily with development of an improved desiccant regeneration technique which can make use of any type of energy supply, an analysis was made to identify more clearly how solar energy should be utilized. The question is: Since the system has considerable energy conservation potential regardless of energy source, should solar operation be considered and in what manner? It has become clear during this study that the use of solar energy to regenerate the desiccant must be justified solely on its own fuel conservation merits as with other solar applications. Also it is clear that a system use factor of at least 40 to 50% is needed to justify the additional capital cost of solar collectors. As was shown in the previous economic analysis, the fuel saving nature of the basic regenerated desiccant system can, with high use factors, help to pay off the additional solar investment such that attractive economics can still be obtained.

In regard to functional integration of a solar energy source, it has been determined that the lowest cost technique is to design for continuous regeneration and use the solar energy input to augment the use of a conventional energy source. Solar energy storage, if used to increase the percent solar contribution, should be external to the desiccant system rather than using desiccant

as an overnight energy storage medium. The use of concentrating solar collectors, generating steam at 300 to 350 F, will permit an optimum system design at a collector cost competitive with flat plate collectors. The use of phase-change materials appears to be the most promising technique for solar energy storage in this application.

Section 8

REGENERATED DESICCANT DRYING/DEHUMIDIFICATION APPLICATIONS

The regenerated desiccant technique developed and analyzed in this study is applicable to virtually any drying/dehumidification requirement which has the following general characteristics:

- Drying temperature of 80 to 110 F with relative humidity of 30 to 40% (closed cycle drying)
- Requirement for thermal energy in the form of steam or hot water for additional drying capacity or other energy use
- Use-factor of at least 15 to 20% for large capacity application — 40 to 50% for smaller capacity applications.

Since time restraints did not permit analyzing the specific requirements of a broad range of applications, effort was focused on three representative applications in addition to the baseline grain drying application. These are: kiln drying of lumber, soybean processing, and space air conditioning.

Kiln Drying of Lumber: This appears to be a highly promising application because continuous drying is required virtually year round. Also, the steam from the desiccant evaporator can be used within a conventional steam heated kiln. Use of desiccant drying would permit the lumber to be dried at lower temperature with better control of relative humidity and temperature than is possible with current kiln systems, resulting in better quality wood. The major components required for the drying system would be an evaporator, heat exchanger, and absorption column. The condenser normally used to condense steam from the desiccant evaporator will not be an extra cost item since the fin-pipe heat exchanger normally used for steam heating of the kiln can be used for this purpose. Special attention will need to be given to the design and sizing of the absorption column to ensure that it is matched to the drying conditions needed for the lumber.

Figure 8-1 shows a simple block diagram of the kiln drying application. In this approach Kiln 1 is converted to low-temperature desiccant drying with a boiler used to drive the desiccant evaporator. The steam from the evaporator is then used to drive a smaller conventional kiln (with half the capacity of Kiln 1). The boiler would be approximately half the capacity normally required for both kilns, although a higher pressure rating could be required. In the case where Kiln 2 is to have higher capacity, it can be heated by both the steam from the boiler (conventionally) and with steam from the desiccant process^{*}. As indicated from the economic analysis of Section 6, the desiccant equipment should pay for itself within two to three years, depending upon the specific size requirement. For this application, since the use factor will exceed 80%, it may be quite feasible to use solar collectors to provide for additional fuel savings.

Soybean Processing: Processing of soybeans represents a grain drying situation in which the drying equipment is operated over a large fraction of the year, thus representing a favorable economic situation for the use of a regenerated desiccant system. Large plants, such as are operated by the Gold Kist Corporation for example, require that the soybeans be partially dried when received and further dried during later processing. A system such as the reference design analyzed in the previous section of this report could be applied to soybean drying. As in the lumber drying case, the absorption column must be designed to meet the specific drying requirements (of moisture, flow rate, etc.). The use of solar energy should also be considered in this application because of the favorable use factor.

Space Air Conditioning: This application of the regenerated desiccant process differs from the others in that the evaporator, heat exchanger and absorption column would be combined with a standard lithium bromide absorption chiller to obtain an air conditioning process with a high coefficient of performance (COP). A block diagram of this concept is shown in Fig. 8-2. The

^{*}If the desiccant evaporator is operated at sub-atmospheric pressure, the low pressure steam can be injected into the Kiln 2 steam line using an ejector.

Fig. 8-1 - Block Diagram of Lumber Drying Application

Fig. 8-2 - Block Diagram of Regenerated Desiccant Air Conditioning Concept

key feature of this process is that the steam from the evaporator, at about 200 to 220 F, which contains approximately 75% of the energy input to the desiccant evaporator is used to drive the absorption chiller (specifically, to drive the lithium bromide vapor generator). In this manner, the latent load (air dehumidification) is handled by the desiccant equipment^{*} while the sensible load (cooling) handled by the conventional absorption cooling equipment^{**}. If the air conditioning load is such that all of the steam (recovered heat) is utilized, then the COP can be increased by more than 60%, with energy savings of over 40% obtained compared to using a conventional absorption chiller alone. This concept, which represents a hybrid absorption air conditioning machine, appears well suited for use in solar air conditioning applications, where normally a lithium bromide absorption chiller is used. In this case, there would be a substantial decrease in solar collector investment. It is estimated that this would more than offset the increased cost of the air conditioning equipment. On a fuel saving basis (non-solar) it is estimated that the additional cost of the desiccant equipment could be paid back in less than seven years.

^{*}The heat added to the desiccant would be removed, to the maximum degree possible, using a cooling tower.

^{**}Since cooling to the dewpoint is not required in this case, the capacity and COP of the chiller would be somewhat improved.

Section 9

CONCLUSIONS AND RECOMMENDATIONS

The major conclusion of the study is that the regenerated desiccant drying concept is significantly improved, in both cost and performance potential, through the use of the evaporative regeneration technique. The overall concept shows excellent potential as a major means of industrial energy conservation with a wide range of applications, and exhibits promising economic characteristics, particularly for high-use applications. For the baseline commercial grain drying application, acceptable economics will be obtained if the system is used for at least 20% of the year.

It is recommended that a modular prototype regenerated desiccant system be built and tested to provide a useful demonstration of this technology. A capacity of 100 lb/hr would be an appropriate size. The modular prototype system should incorporate the necessary operational flexibility (particularly with regard to the absorption column) to permit it to be tested over a range of conditions representing three or four application areas (such as for product drying and general air dehumidification). The initial step of this program should be a design requirements and integration analysis to identify, for the applications to be demonstrated, such specific items as performance requirements, any operational problems, sizing and design integration, and development/demonstration plans.

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