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# A Liquid Sorption/Desorption Air Conditioning System Powered by Recovered Heat

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### Introduction

Air conditioning has two ways to be performed: through a compression cycle utilizing secondary energy and through an absorption cycle utilizing primary energy. In both cases the fluids operating in the cycle never come into direct contact with the processed air, so that heat and mass transfer encounters considerable resistances, particularly when air dehumidification is needed. Apart from the particular value of the coefficient of performance of a compression cycle, its primary energy ratio between the refrigeration capacity and the primary energy consumed at the power station to produce the electricity requested ranges roughly around unity, so that the energy balance of such a system is really modest.

While it is not possible to eliminate the above mentioned resistances in a compression air conditioning cycle, this is on the contrary possible in an open absorption air conditioning cycle, and if furthermore the efficiency reached in such a system would be comparable with the overall one of compression, the solution would be of interest for air conditioning.

Moreover, it is not necessary that primary energy has to be produced on purpose from fuel, but, as many industrial processes may be feasible, it can come from heat recovery. If the requested heat supply temperature may be limited enough a large number of industrial cases would be candidates.

In this case the primary energy cost of the air conditioning would be null and the energetic and economic comparison with the compression cycle extremely favourable.

### Proposed dehumidification-humidification process of air conditioning

The process of air conditioning through a dehumidification of air followed by an adiabatic humidification is known and reported by different authors /1/2/3/, which were particularly interested for its powering by solar energy heat.

More recently a modelling of the whole system in a particular configuration, presenting its energy efficiency competitiveness with the conventional closed absorption cycles for air conditioning, is also reported /4/.

The present process is a further improvement of that configuration, looking particularly to the efficiency quality and to the powering with heat at lower temperature. The cooling tower is here preferred to the evaporative cooler, because of the lower cold water temperature produced at same external air conditions.

As the following Fig. 1 shows, the system is composed of two main sections, the air conditioning at left and the solution regeneration at right. The first is composed of three units: the absorber, where the air M to process is dehumidified

in contact with a hygroscopic solution of LiCl over a tube bank cooled by water, the tower, cooling the water from the absorber, and the air washer.

The air F outgoing from the cooling tower is generally colder than the air N from the absorber, particularly where a high efficiency tower is utilized, and the rotary air/air heat exchanger HEx1 lowers the temperature, i.e. the enthalpy, of the dehumidified air to point O. This one is then saturated in the air washer, at the lower possible enthalpy, to the state I of the supply air, which lies on the condition line of the space to be air conditioned. The simulation experience has demonstrated that the tower air G from the heat exchanger HEx1 is cold enough to lower, in the exchanger HEx4, the temperature of the solution entering the absorber, before it will be discharged to the outside.

In the second section the water vapour is desorbed from the diluted solution by a stream of stripping air at direct contact with the diluted solution over a tube bank heated by warm water. This one is external air E preheated in a rotary heat exchanger by the leaving hot and humid air. The lower the specific humidity of the stripping air, the higher the water mass transfer and the higher the air temperature, the lower the fraction of the sensible heat supplied to air over the total heat required at desorber.

The temperature of the heating water at the inlet L of the tube bank of the desorber is modulated through a three way valve controlled by the solution concentration at the outlet: a higher temperature gets a higher solution concentration, so that the temperature at point L can condition easily and at will the solution concentration at the absorber inlet and permits to follow the air conditioning requirements at the absorber time by time. The effect of the solution concentration at the absorber inlet on the final state of the dehumidified air will be shown afterwards.

The heating water L' from the heating system must have a slightly higher temperature than that required at L to allow for a continuous modulation; anyway lower water supply temperature means a wider utilization of the powering heat, if this has a batch source, or otherwise, the applicability of the process to a larger number of cases, if the heat is directly recovered from an industrial process.

Being the outside air E always at higher temperature than the ambient one A, the solution mean temperature is higher in the desorber than in the absorber. In the meantime the first one requires solution temperatures as high as possible while the opposite is true for the absorber. The solution heat exchanger HEx3 economises therefore some sensible heat of the solution flowing from desorber to absorber and vice versa.

#### System analysis and modelling

Each unit operation composing the process has been analysed and modelled to get the simulated behaviour of the whole at different steady state operating conditions. At present the limited available space prevents the description of the models in full extent and therefore only concise references to the literature are made for those published.

The absorber is simulated by a one dimension model /5/ which can predict the air and solution states at any vertical section of the tower as well as the cooling water temperature. The salt of the hygroscopic solution is LiCl, chosen for the favourable thermodynamic properties of the solution and for hygienic reasons, being the air conditioning system open. Other salt solutions are utilisable

for the process, except that the new solution properties have to be introduced in this and the other unit models.

The desorber and the rotary air/air heat exchanger HEx2 are simulated together by a one dimension model /6/. As the former one, it allows to define the operating fluids properties at any vertical section of the tower as well as the heat transfer efficiency of the exchanger for each set of operating conditions. For each solution concentration at outlet S, the temperature of the heating water L is found, being defined the set of geometry, mass flow rate and external air E state.

The model for the cooling tower is one dimensional, not based on the Merkel theory, but on an exact solution of the heat and mass transfer balances section by section. It utilizes surface transfer coefficients, whose correlations are reported from experimental works in literature, and it is able to describe the properties profiles of water and air along the tower.

The heat transfer efficiency of both the air/air heat exchangers HEx1 and HEx2, are calculated according to the procedure reported in /7/. Following the same source /7/ also the solution/solution and solution/air heat exchangers are calculated, but with the difference that here two parametric efficiencies are assumed, being certain that for each one a suitable geometry exists to yield those efficiencies.

The former exchanger was supposed to be a plate one and the second a finned coil. The model of each component operates at first until the reaching of imposed stability conditions in it and, afterwards, iteratively from one unit model to another, searching for properties stability of the fluids flowing between the units.

### System characteristics

The geometries of absorber and desorber are assumed similar. Both are composed of an horizontal bank of bare tubes over which the solution is sprinkled, while air flows upwards. The stainless steel tube has 20 mm outside and 16 mm inside diameter, the tube bank presents a triangular pitch of 1.5 times the tube o.d., is composed of 33 tubes in 11 rows and the tube length is 2.0 m. The exchangers HEx1 and HEx2 have a metallic matrix mesh of steelwire, weighing 103 kg and rotating, the first, at 29.3 RPM and, the second, at 21.5 RPM.

The cooling tower has a fill of corrugated PVC foil with a base square section of  $2.0 \text{ m}^2$  and 1.3 m of height. For the air washer only the particular hypothesis of adiabatic working is assumed, because the required humidification efficiency for the cases examined here is not critical.

### Design conditions

For all the cases analysed the following data were kept constant.

#### - Mass flow rates:

Air at desorber = 3.75 kg/s

Air at cooling tower = 8.0 kg/s

Cooling water at absorber inlet = 5.0 kg/s

Solution at desorber outlet = 2.0 kg/s

#### - Heat transfer efficiencies /7/:

HEx3 solution/solution heat exchanger:  $\eta = 0.80$

HEx4 solution/air heat exchanger:  $\eta = 0.60$

- External air E  
 $t = 34.0 \text{ }^\circ\text{C}$   
 $x = 0.0130 \text{ kg/kg}$   
 $\text{RH} = 39.0\%$

- Ambient air A  
 $t = 26.0 \text{ }^\circ\text{C}$   
 $x = 0.012630 \text{ kg/kg}$   
 $\text{RH} = 60.0\%$

- Ambient air renewal: 20.0%

The external air assumed corresponds to the maximum load design data of several towns in Italy and can be significant for the Mediterranean area.

### Performance analysis

Three series of working modes were studied, aiming at showing the adaptability of the process to typical variations of the operating conditions of an all-air conditioning.

The coefficient of performance of the process is here defined as:

$$\text{COP} = q_o/q_f = [m_{\text{abs}} \cdot (h_A - h_I)] / [m_{\text{wdes}} \cdot C_w \cdot (t_{wi} - t_{wo})] \quad (1)$$

where:

$q_o$  = refrigerating power, kW;

$q_f$  = heating power to desorber, kW;

$m_{\text{abs}}, m_{\text{wdes}}$  = mass flow rates of air at absorber, of heating water to desorber, kg/s;

$t_{wi}, t_{wo}$  = inlet, outlet heating water temperature at desorber,  $^\circ\text{C}$ .

Furthermore:

$\xi_s$  = solution weight concentration at desorber outlet, i.e. absorber inlet, kg<sub>LiCl</sub>/kg<sub>sol</sub>.

Mode A - Constant condition line, constant air mass flow rate at absorber and three different temperature  $t_I$  of the supply air.

Data assumed:

- Air mass rate at absorber = 3.75 kg/s

- Condition line slope:  $\Delta h/\Delta x = 7298.70 \text{ kJ/kg}_{\text{H}_2\text{O}}$

-  $t_I = t_A - \Delta t = 20.0, 18.0 \text{ and } 16.0^\circ\text{C}$

The Fig. 2 refers to the case  $t_I = 20.0^\circ\text{C}$  and shows, on the Mollier diagram, the transformations undergone by the air streams in each of the three unit operations, while letters indicate the corresponding points on the process lay out of Fig. 1. The bold lines MN and PT, EF and UV, BC and QR represent the air and the boundary layer states along the towers, respectively of absorber, cooler and desorber. The figures along both the air and the solution transformations and the dashed lines connecting the corresponding points indicate the air and boundary layer conditions at the lower section ranging from 1 to 51.

The following Table 1 illustrates the main figures for three cases of this working mode, being case A1 that of Fig. 2.

TABLE 1: System performance for the cases of the working mode A

Case	$\Delta h/\Delta x$ kJ/kg <sub>H<sub>2</sub>O</sub>	$m_{\text{abs}}$ kg/s	$\xi_s$ kg <sub>LiCl</sub> /kg <sub>sol</sub>	$t_{wi}$ $^\circ\text{C}$	$t_I$ $^\circ\text{C}$	$\Delta t$		$q_o$		$q_f$ kW	COP	
						$^\circ\text{C}$	0%	kW	0%		-	3%
A1*	7299.70	3.75	0.28204	38.6	20.0	6.0	-	35.49	-	42.63	0.832	-
A2	7298.70	3.75	0.32693	44.8	18.0	3.0	+33.3	47.30	+33.3	62.84	0.753	- 9.6
A3	7298.70	3.75	0.37370	51.6	16.0	10.0	+66.6	54.37	+53.2	84.97	0.699	-16.1

The more interesting results to point out are:

- the strong influence of the solution concentration  $\xi_s$  on the system performance, being this the only independent variable through which to modify, from case to case, the  $t_{wi}$  value; as seen before  $\xi_s$  is controlled through  $t_{wi}$ ;
- the wide (+53.2%) refrigeration power variation in comparison with a modest decrease (-16.1%) of COP;
- the very good values of COP if compared with those of the absorption closed cycles for air conditioning;
- the low temperature levels of the heating water  $t_{wi}$  required, always lower than the values commonly accepted in literature [1], and at which it is impossible to power any conventional absorption cycle;
- the contained increase of  $t_{wi}$  with the  $t_{wi}$  decrease.

For both COP and  $t_{wi}$  it is more convenient to operate at lower  $\Delta t = t_{wi} - t_A$ , i.e. higher  $t_{wi}$ , as possible.

Mode B - Constant condition line, variable air mass rate at absorber and constant air temperature  $t_{wi}$ .

Data assumed:

- Air mass rates at absorber = 3.75, 5.0, 6.25 kg/s
- $t_{wi} = 20.0$  °C,  $\Delta t = 6.0$  °C
- condition line slope, as mode A: 7298.70 kJ/kg<sub>H<sub>2</sub>O</sub>

Table 2 reports the main results, which may be summarized as in the following. Case B1 is the same as A1.

TABLE 2: System performance for the cases of the working mode B

Case	$\Delta h/\Delta x$ kJ/kg <sub>H<sub>2</sub>O</sub>	$m_{abs}$		$\xi_s$ kg <sub>LiCl</sub> /kg <sub>sol</sub>	$t_{wi}$ °C	$\Delta t$ °C	$q_o$		$q_r$ kW	COP	
		kg/s	$\Delta x$				kW	$\Delta t$		-	$\Delta \%$
B1*	7298.70	3.75	-	0.28204	38.8	6.0	35.49	-	42.63	0.832	-
B2	7298.70	5.0	33.3	0.29646	41.9	6.0	47.17	32.9	57.83	0.816	-2.0
B3	7298.70	6.25	66.6	0.31087	45.3	6.0	59.17	66.5	74.54	0.793	-4.7

- the quasi-linear relation between the absorber air mass rate and the refrigeration power, demonstrates that  $m_{abs}$  is the independent variable through which to modify the  $q_o$  value;
- the necessity of a slight adjustment of  $\xi_s$  to get  $\Delta t$  constant;
- the stable value of COP with a strong variation of the system capacity;
- the suitability of the process to work with variable air volume systems.

The comparison between cases A3 and B3 shows the advantage of varying the power  $q_o$  through  $m_{abs}$  instead of  $\Delta t$ : for the same percentual variation of the two variables, case B3 reports a higher COP and even a higher  $q_o$  than case A3.

Mode C - Variable condition line, constant air mass rate at absorber and constant air temperature  $t_{wi}$ .

This mode accounts the quality variation of the heat load.

Being:

$$\Delta m_w = m_{abs} \cdot (x_A - w_{wi}), \text{ kg/s, water mass rate extracted from ambient}$$

$$h_v(t_A) = 1.83 \cdot t_A + 2501, \text{ kJ/kg, enthalpy of water vapour at } t_A$$

$$q_t = q_s + q_l \quad q_t = \text{total heat load}$$

$q_s =$  sensible heat load

$q_l =$  latent heat load

$$\frac{\Delta h}{\Delta x} = \frac{q_t}{\Delta m_w} = \frac{q_s}{\Delta m_w} + h_v (t_A)$$

It is:

$$\frac{q_l}{q_t} = \frac{h_v (t_A)}{\Delta h / \Delta x}$$

$$\frac{q_s}{q_l} = \frac{1}{(q_t/q_s - 1)}$$

Data assumed for the cases of this working mode:

- Air mass rate at absorber = 3.75 kg/s
- $t_I = 20.0$  °C,  $t = 6.0$  °C
- condition line slopes: 4865.8, 7298.7, 9731.6 kJ/kg<sub>H<sub>2</sub>O</sub>

The figures for this mode are reported in Table 3; case C2 is the same as B1 and A1. In Fig. 2 the states of the air introduced I are reported for these three cases. The following comments apply to this mode.

TABLE 3: System performance for the cases of working mode C

Case	$\Delta h / \Delta x$		$m_{\text{abs}}$ kg/s	$q_l / q_t$		$q_s / q_l$		$\zeta_s$ kg <sub>LiCl</sub> /kg <sub>sol</sub>	$t_{w1}$ °C	$\Delta t$ °C	$q_o$		$q_f$ kW	COP	
	kJ/kg <sub>H<sub>2</sub>O</sub>	°K		-	°K	-	Δ%				kW	Δ%		-	Δ%
C1	4865.8	-33.3	3.75	0.524	+50.0	0.909	-51.2	0.33111	45.3	6.0	48.35	+36.2	64.74	0.747	-10.3
C2	7298.7	-	3.75	0.349	-	1.864	-	0.19704	38.6	6.0	35.49	-	42.63	0.832	-
C3	9731.6	+33.3	3.75	0.262	-25.0	2.818	+1.2	0.20591	40.4	6.0	31.28	-11.9	35.67	0.877	+5.3

- the decrease of the latent heat load requests a supply air I with higher enthalpy (Fig. 2) giving therefore a higher COP;
- a strong variation of the latent load fraction affects only slightly the COP;

At constant total load, but with a lower latent load, the humidification of case C1 (Fig. 2) can go on reaching the condition line of case C3 at a point I (C1'). At that point  $t_I = 16.73$  °C, the COP and the total load  $q_o$  are the same as case C1, but  $q_s$  and  $q_l$  become 35.68 kW and 12.67 kW, with a variation respect to the C1 case of Δ% = +55.0 and Δ% = -50.0 respectively.

### Conclusions

The independent variables controlling the system and their variation effects on its performance are well defined.

The open absorption system presented offers so many interesting performance characteristics of adaptability, also with wide variation of the operating conditions, of good energy efficiency and, above all, of low temperatures of the powering heat that it seems to be more suitable for air conditioning applications than the conventional closed absorption cycles. The direct heat and mass transfer of the operating mixture (LiCl - H<sub>2</sub>O) with the processed air at each unit operation reduces to a minimum the gradients, allowing a higher efficiency.

Particularly the low heating temperatures required, mostly in the range 35° + 45 °C for the cases analysed, suggest the utilization of the process in those industrial applications of air conditioning where, as is often the case, consistent amounts of recoverable heat are available: if so, no energetic extra costs and negligible operation costs are needed.

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#### SYSTEME DE CONDITIONNEMENT D'AIR PAR ABSORPTION/DESABSORPTION ALIMENTABLE PAR CHALEUR RECUPERÉE

RESUME: On présente et analyse un procédé de conditionnement estival d'air par déshumidification-humidification. La déshumidification a lieu dans une colonne d'absorption par contact direct avec une solution hygroscopique sur un échangeur horizontal multitubulaire, dont à l'intérieur écoule de l'eau refroidie par tour de refroidissement. Après un échange thermique avec l'air sortante de la tour, l'air déshumidifiée est de suite humidifiée adiabatiquement. La solution diluée est reconcentrée par contact direct avec de l'air extérieure sur un échangeur multitubulaire qui peut être aussi par chaleur un échangeur entre entrée et sortie de l'air economise de l'énergie thermique. Les variables indépendantes qui contrôlent le procédé et les effets de leur variation sur le rendement ont été définis au moyen de l'analyse de trois différentes modes de fonctionnement. Le comportement du système est aussi présenté pour plusieurs cas de fonctionnement.

#### SUMMARY

#### A LIQUID SORPTION/DESORPTION AIR CONDITIONING SYSTEM ACTUATED BY RECOVERED HEAT

An open process of summer air conditioning by dehumidification-humidification is presented and simulated. The air dehumidification is achieved in an absorption column through direct contact with a hygroscopic solution over a cooling tube bank in which water flows from a cooling tower. After a heat exchange with the cooler outlet air the processed air is adiabatically humidified. The diluted solution is reconcentrated in a desorber by direct contact with external air over a heating tube bank, in which water flows from an external heating or heat recovery system. A heat exchanger recovers sensible heat between inlet and outlet air. The independent variables controlling the process and their variation effects on its performance are defined through the study of three different modes of operation. The system performance for a certain number of cases are presented as well.



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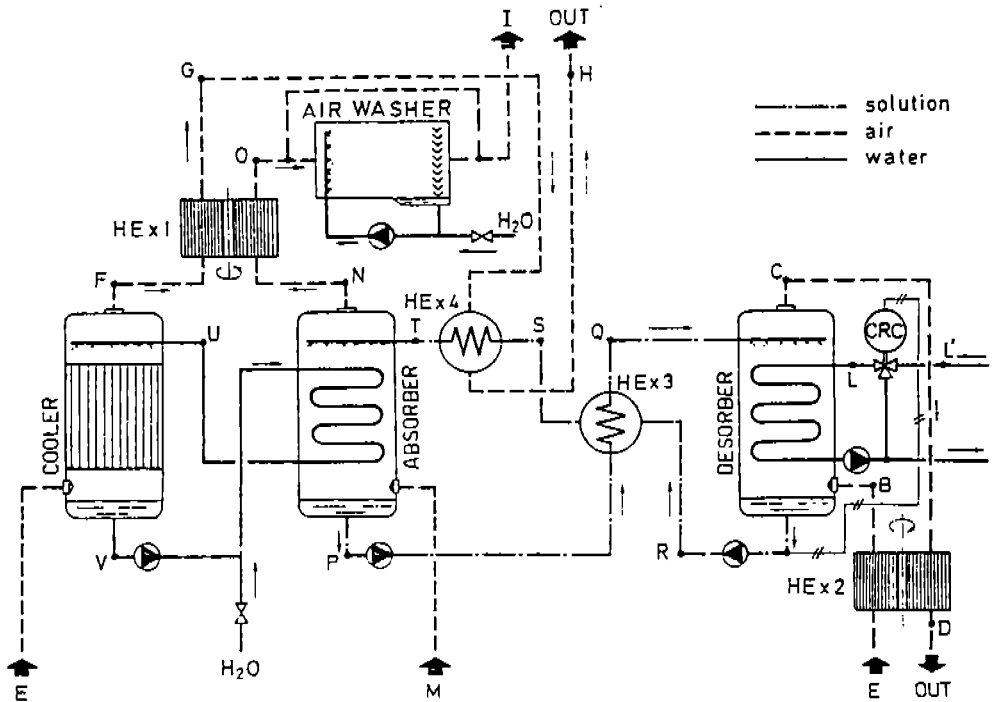
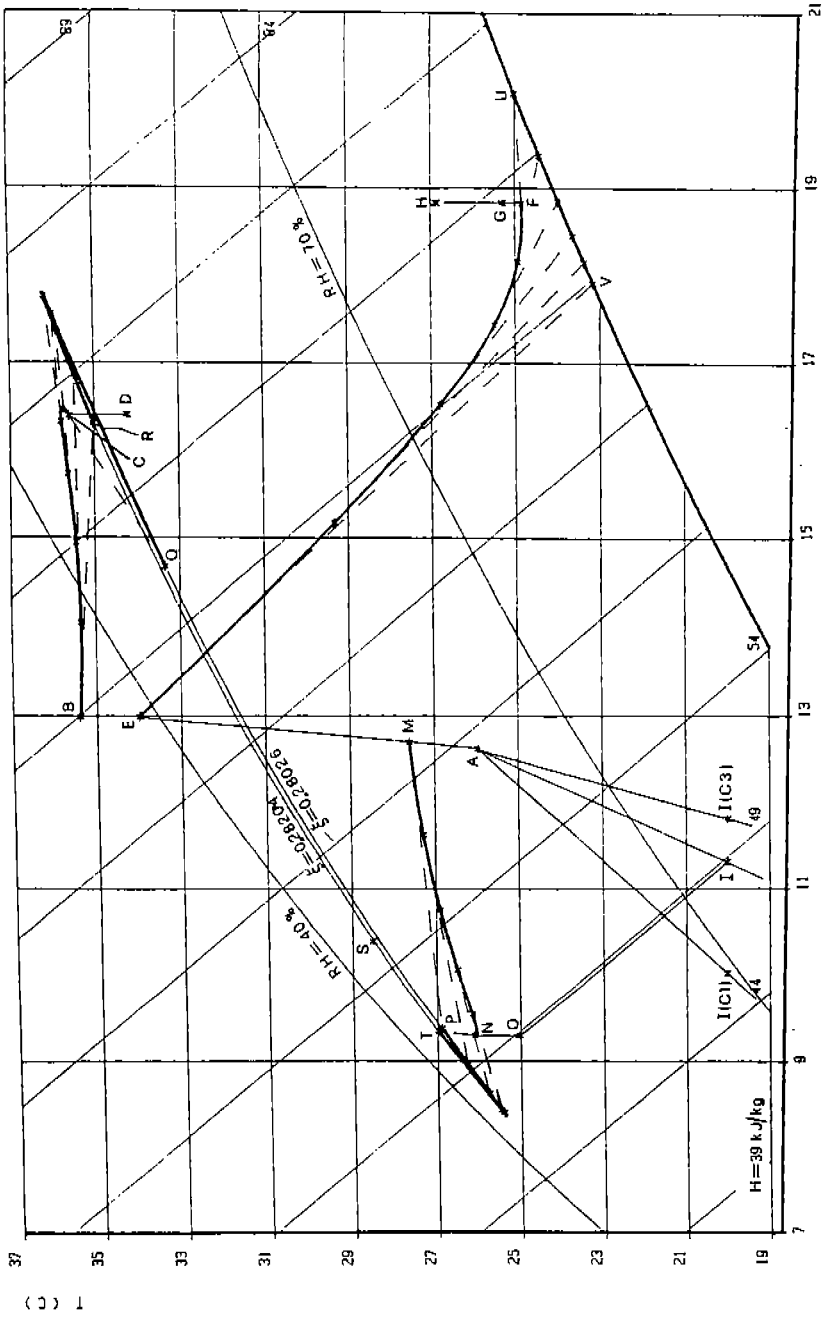


Fig. 1 - Scheme of the air conditioning system proposed



$X$  ( g / kg )

Fig. 2 - Air and solution transformations, at each system unit, on the psychrometric diagram